STEAM AND GAS TURBINES

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PREFACE

The desire for technical information at a fundamental level has been proved by demands of operators and others interested in turbines. This book is in response to the enthusiastic reception given *Power's* special sections, *Steam Turbines*, December, 1945, and *Gas Turbines*, October, 1946. Naturally, the much greater space available in a book allows a more extensive explanation of all phases of turbine operation and selection. However, mathematics and advanced theory have been minimized as much as possible.

A studied effort has been made to explain fundamental relationships in terms of effects and causes that can be appreciated on the basis of everyday experiences. This type of approach has proved helpful also to engineering students studying turbines.

Readers will recognize some of the illustrations as having first appeared in the original sections of *Power*. Much more new material, however, has been added, some of it original with the authors and some contributed by various manufacturers.

The theme of the book centers about the understanding of turbine behavior as applied to stationary power generation. From a simplified consideration of the forces that make a windmill turn, the operating method of the turbine has been derived. With an elementary understanding of the basic working forces and parts involved, the turbines are classified according to mode of application, and then typical constructions are examined.

After the general-assembly methods have been reviewed, attention is given to the arrangement and design of individual turbine parts. This is followed by a discussion of the nature of the lubricating requirements and how they are met in turbines. A brief introduction to steam fundamentals lays the groundwork for an appreciation of the variation in heat performance of turbines. Short-cut methods show how to estimate expected performances of turbines under different controlling conditions.

The chapter on Operation and Maintenance highlights some of the problems in running, starting, and stopping a turbine. Practical requirements concerning turbine operation are discussed in the chapters on Governing and Turbine Auxiliaries.

The final chapter on Gas Turbines recognizes the rapid rise of these units to a practical status over the past 10 to 12 years. Before many years pass, the practical operating and plant designing engineer will have to be well versed in the fundamentals of these prime movers. The review of the performance characteristics and the many forms that gas-turbine plants are now taking should be of great help in this respect.

The authors gratefully acknowledge the cooperation of manufacturers in supplying information and illustrations. We are also indebted to the publishers of *Power* for permission to use their original material. The able direction and coauthorship of the original special sections by L. N. Rowley, Executive Editor of *Power*, have been of value in the preparation of this book.

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CHAPTER 1

TURBINE TYPES

An overwhelming number of the stationary prime movers in use today are steam turbines. Anyone who has had the opportunity to visit even a few power plants will be impressed by the wide variety of forms that steam turbines take. They vary in rating from less than 1 hp to over 200,000 kw; their working speeds range from a few hundred rpm to over 10,000; they are built to run on steam from less than atmospheric pressure to over 2,400 psi; they can run at variable speeds as well as speeds held to a value that won't vary by more than about 0.1 per cent from the specified rpm. Steam turbines are versatile in many respects, and this has made their acceptance well-nigh universal for the production of stationary power by fuels.

How does a turbine work? We know that steam passes through, entering at high pressure and temperature and leaving at lower pressure and temperature, and that as a result the shaft turns. That, of course, is merely an observation of facts but doesn't tell us much of what's happening inside the unit.

Evolution of a Turbine. To understand what's going on inside a turbine, we shall have to review what takes place in some common everyday experiences. Consider the wind for example: we know that when it blows effort is required to walk against it. In high winds we occasionally have found it impossible to walk, because of the immense force exerted on our bodies. In extreme cases, such as hurricanes, the wind has uprooted trees and carried buildings from their foundations.

Some of our smart ancestors, appreciating the usefulness of the force created by the wind, put it to work. The wind has sailed ships and turned windmills for centuries. The amount of human labor the wind has saved is enormous, but, more important, it has made it possible to accomplish things that would be practically impossible with the human and animal labor available.

Just how does the wind make a windmill turn? There are many forms of windmills, but let's examine a very simple one, such as shown in Fig. 1-1. We wouldn't be far wrong in saying that the windmill was the first form of turbine, so it becomes important to our discussion. The sails or arms of the windmill are turned at an angle to the direction from which the wind is coming. The axle on which the sails are mounted faces directly into the wind.

Why are the sails turned at an angle? Well, let's look at two extremes. If the sails were turned edgewise to the wind, the force exerted would be negligible (Fig. 1-1*b*). If the flat side were faced directly into the wind, there would be an enormous force, but it would be in the wrong direction (Fig. 1-1*c*). The sail would bend backward along the length of the shaft and tend to snap in two. Now, if the sail were turned at an angle between

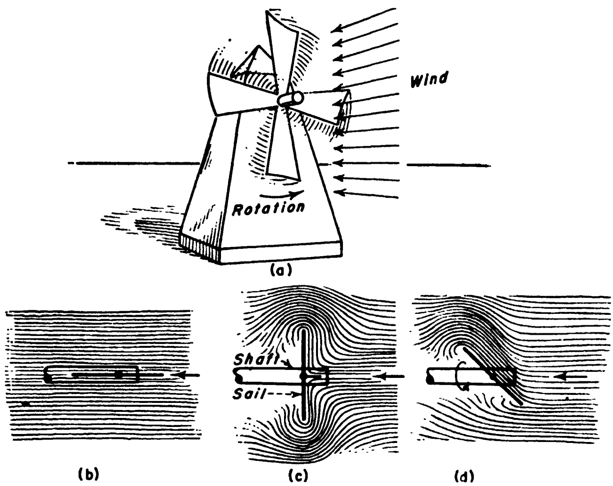


FIG. 1-1. Windmills probably were the first form of turbines. Wind produces the moving force by sliding over the inclined sail surface and momentarily compressing itself.

these two extremes, perhaps somewhere around 45 deg, the wind would slide over the face of the sail as in Fig. 1-1*d*. In doing this, the wind piles up on the windward side of the sail and compresses itself; simultaneously the air behind the sail will be somewhat lower in pressure because of the way in which the wind is sliced. Hence, the sail is dividing two areas of different pressure and will be pushed in the direction of the lower force, in this case downward toward the left in the diagram. Since it is free to rotate only with the shaft, it moves directly downward, as we look at it in Fig. 1-1*d*. It always maintains this same angle to the wind as it rotates; hence, so long as the wind blows constantly, it will have the same force acting on it.

How fast will the shaft rotate? That depends on the wind speed, the angle of the sails to the wind, and the amount of resistance offered against the turning moment by the load that the shaft is carrying.

We see that the force of the wind can do very useful work when the proper equipment is built to utilize it. However, the windmill is at the mercy of the weather; the most efficient one built is useless in a calm. What can we do to get steadier use out of this equipment? Create an artificial wind. Sounds silly! But wait and see. This is a process of evolution. Remember the old teakettle on the fire? The more heat we

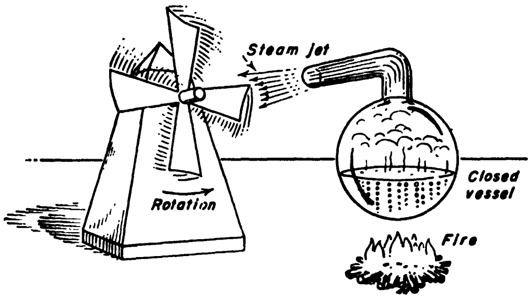


FIG. 1-2. Artificial wind to run a windmill can be produced by boiling water over a fire and directing a steam jet against the sails.

put under it, the faster the steam shot out of the spout, as shown by the louder hissing and the increasing cloud of condensation. In some cases, the lid was even blown off. There's our artificial wind.

As shown in Fig. 1-2, we'll just set up a boiler some distance in front of our windmill, put in enough water, aim the spout at the shaft, light the fire, and as steam pressure builds up in the boiler above the water, the steam leaves in a jet from the spout (nozzle, if you want to be technical) at higher and higher speeds, as we make a hotter and hotter fire. As the jet passes over the sails, the windmill rotates. There's the solution to better utilization of our windmill. Of course, it will take effort and money to get fuel for the fire, so it isn't free by any means; but at least we can get out our production.

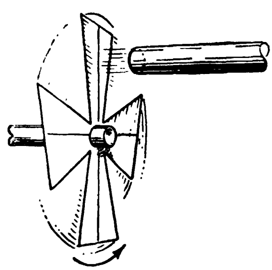


FIG. 1-3. Better utilization of the steam jet can be realized by bringing it up closer to the sails. Between the sails, the jet is blowing free and doing no work.

After a while the cost of the fuel in labor or dollars becomes a problem, and we must look for means of improvement. First let's look at the steam jet. We notice that its speed and force drop off rapidly as it gets farther and farther from the nozzle. Because its energy seems to dissipate as it hits the air, let's move the nozzle up close to the sails as in Fig. 1-3. We do get much better performance, but notice that the jet only acts on the sails, one at a time, for only about half the time in one revolution, because of the gap between the sails. We need more sails, so that none of the jet energy will be wasted. We notice also that the jet acts on only part of the sails, so we proceed to redesign the entire rotor, making the sails smaller and more numerous and mounting them on a disk carried by the shaft, as in Fig. 1-4*b*.

The sails don't look much like windmill sails any more, so we'll call them "blades." Figure 1-4*a* shows how the steam jet acts on the blades.

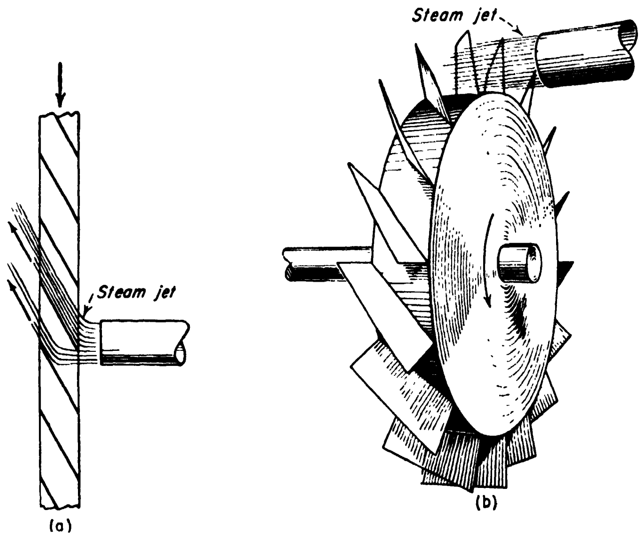


FIG. 1-4. Redesigning a windmill to keep the jet always blowing against a surface takes form as in (*b*) with blades mounted on the rim of a wheel or disk. Developed periphery of wheel and blades appears as in (*a*).

The blades deflect the jet and make its pressure on the upper side of the blades higher. The steam jet will be deflected in the direction shown when the blades are at standstill. Once the blades are in motion (downward on the diagram), the amount of jet deflection is less, and of course the force exerted by the steam jet will be correspondingly less. The speed will settle to a constant value where the amount of energy given up by the jet equals that needed by the load on the shaft.

Something doesn't look quite right with this arrangement. The jet enters on the blade abruptly, sets up turbulence, and still has considerable speed when it leaves the wheel or blades. So let's experiment. We'll set up a blade so that it can be rotated about its vertical axis and so that we can measure the force exerted on it in only one direction, as in Fig. 1-5. Then we'll play a constant-sized steam jet on it and measure the force developed. In (*a*) the blade is parallel with the jet, and we find that the force exerted toward the left is zero. Now we'll turn the blade through 45 deg, and we find the force has increased to a bit less than 0.3 lb. When the blade turns its flat side to the jet, the force has increased to a bit less than 1.0 lb, at which point the maximum force is developed for a flat blade.

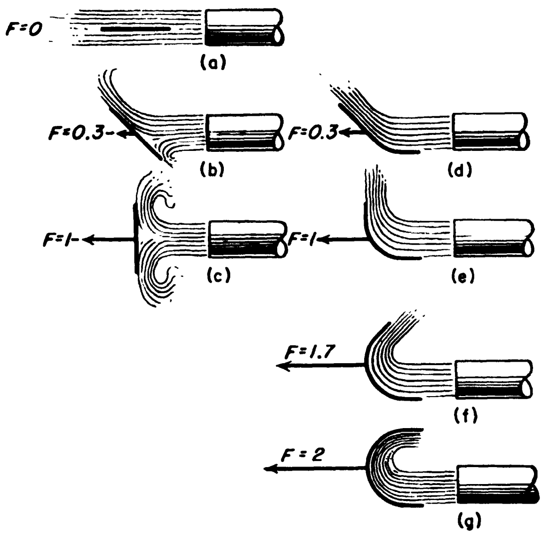


FIG. 1-5. Experimenting with a flat blade in constant-velocity steam jet, no force is developed on the blade as in (*a*); maximum force appears when placed as in (*c*). When curved blades are used, maximum force develops by turning the steam jet back upon itself.

On investigation, we find that considerable turbulence is set up in the steam jet as it hits the flat blade in almost all positions except (*a*). This obviously wastes energy—which we wish to minimize. So let's round off the blade, as in (*d*), so that the steam jet is turned more smoothly. For the same leaving angle as in (*b*), we find slightly more force developed, because of the reduction in jet turbulence. In (*e*) we turn the jet through the same angle, 90 deg, as in (*c*), and again find an improvement with the round-edged blade.

Now we have a new tool and are curious. Let's turn the leaving edge of the blade another 45 deg, as in (*f*). We find that the force has now increased to 1.7 lb. This is remarkable! The more we turn the jet out of its original path, the more force it exerts on the blade that compels it to take a new path. Well! Then let's turn the jet back on itself, as in (*g*). The force jumps up to 2 lb. Evidently, to develop the greatest force, the jet should travel initially in the same direction as the blade and then be turned around a full 180 deg. But wait! All these forces are being developed with the blades at a standstill. To do work, the blades must move while the jet plays on them. The force developed will not be so great, but the main point is that the curved blade will produce more work than the flat blade.

A little reflection will show there are two extremes in relative blade and jet motion: the one at blade standstill, as in Fig. 1-5*g*, developing

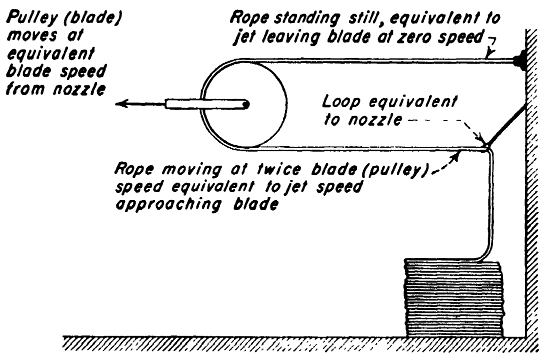


FIG. 1-6. Analogy to demonstrate that a steam jet leaving a semicircular blade has zero speed when the jet approaches the blade with twice the velocity of the blade.

maximum force; the other, when the blade travels at the same speed as the jet. For this condition, the jet can't move through the blade and therefore no force develops. The ideal condition arises when the jet gives up all its energy to the blade and leaves it with zero speed.* This takes place when the blade speed equals one-half the jet speed. This condition can be easily visualized by setting up an analogy, as in Fig. 1-6. A rope is tied to a wall and led through a pulley and then back through a loop. When the pulley is pulled away from the wall, the tied end, as it pays out of the pulley, remains stationary in space; the part entering the pulley moves with just double the speed of the pulley as it passes through the loop. The loop is equivalent to the nozzle, the pulley equivalent to the blade, and the upper part of the rope equivalent to the jet as it leaves the blade at zero speed in relation to the nozzle (loop).

* When the jet moves it has energy, called "kinetic energy"; when its speed reduces to zero, the kinetic energy is zero.

After discovering this very fundamental principle of impulse turbines, we experiment, place curved blades on the edges of disks or wheels, and finally come up with an arrangement such as shown in Fig. 1-7*a*. This is how the edge of the wheel would look if it were unrolled with the blades and nozzle in place. We can't get the jet to enter in exactly the same direction as the blade motion, because the nozzle has to be set off to one

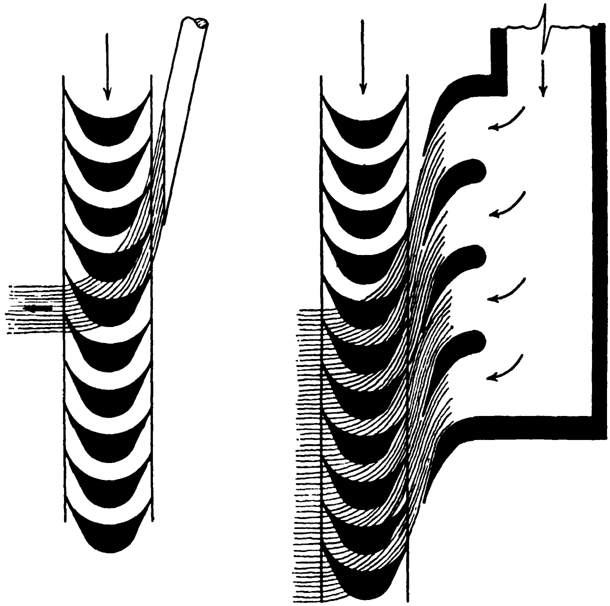


FIG. 1-7. (*a*) Single row of impulse blading with simple nozzle. The steam jet moves faster than the blading, causing the jet to compress itself and exert a force on the blading. The jet follows a curved path as it passes between blading and loses a large part of its speed. (*b*) A given row of blades can be made to do more work by passing greater steam flows through it. The multiple nozzle at the right passes more steam and can be made to cover the entire wheel periphery.

side, but we make the entrance angle as sharp as mechanical limitations will allow. Space has to be left between the exit edges of the blades for the steam jet to leave; hence the full curve in the blade is less than the half circle we should like to make it. As in many machines we have to compromise between theory and practice. Now when we start the steam jet through the blades, put the design load on the shaft, and get the speed up to its most economical point, let's take a high-speed snapshot of the jet passing through the blades.

We'll see something like that in Fig. 1-7a. But what's this? It seems that the jet is passing right through the blades. This is an illusion, of course. We must remember that both blade and jet are moving, but the blade moves slower than the jet and therefore the jet path is deflected because the steam compresses itself against the blades. But here we find the jet leaving the blades at right angles instead of at zero speed. How-

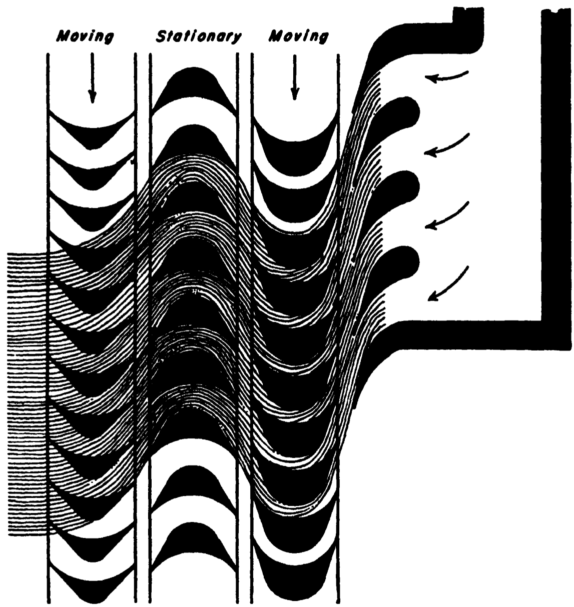


FIG. 1-8. Velocity-compounded impulse stage with two rows of moving blades. Steam jet gives up only part of its energy by partly slowing down in the first row of moving blades. Stationary blades direct steam flow into the second row of moving blades where the steam jet gives up its remaining energy. Steam pressure after the nozzle exit remains constant except for momentary compression against the blading. Velocity compounding allows the use of slower blade speeds.

ever, the jet exit velocity is much less than the speed at which it entered the blades, indicating that it has given up a large part of its energy. The exit speed cannot drop to zero, because the jet must keep moving to get out of the way of the steam following it. So again we have compromised theory for practical considerations.

With only one nozzle working, the blades are idle a large part of the time during one wheel revolution. We can get much more work out of a wheel by having more jets passing through the blades, as in Fig. 1-7b.

Here the jets are ganged up to form a solid wide stream of steam flowing through the blades. We can easily arrange this so that the nozzles are opposite the full periphery of the wheel, and all blades have a continuous force exerted on them during the full revolution.

Under some conditions we may want to use a very high jet speed, but we find this will require a comparably high blade speed and high rpm if we want to absorb a maximum of the jet energy. As the rpm may be higher than we can comfortably use, we'll have to devise means of reducing it to a usable level and still absorb most of the jet energy. One method is called "velocity compounding," in which the steam passes through two rows of moving blades, as in Fig. 1-8. The first row doesn't absorb all the jet energy, as indicated by the amount that the jet is turned back on itself. The intermediate row of stationary blading merely redirects the steam jet at a proper angle into the second row of moving blading, from which it exhausts at low velocity and at right angles to blade motion.

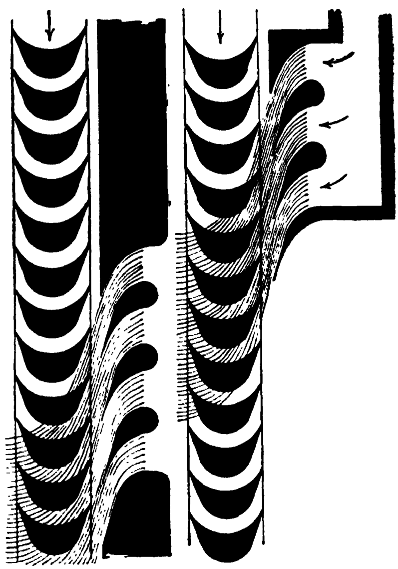


FIG. 1-9. Pressure-compounded impulse stages also allow the use of slower blade speeds for given available steam-pressure drop than a single impulse stage. Here two impulse stages in series drop the steam pressure in two steps through nozzles followed by a moving row of blades.

The second method of reducing blade speed is by pressure compounding. Why do we have a high jet speed? Only because we're using a higher steam pressure in the boiler. To reduce jet speed, we can arrange two nozzles in series, as shown in Fig. 1-9, each nozzle followed by a row of blades. It should be clear that the pressure of the region in which the blades revolve remains constant at both entering and exit edges of the blades. Of course, as the jet passes through the moving blades, there is a momentary compression which creates the working force, but this disappears as the jet leaves the blade. Steam pressure ahead of the first row of nozzles is highest; it drops as it passes through the nozzles into the first row of blades; it drops a second time as it passes through the second row of nozzles; and it reaches its lowest pressure as it enters the chamber where the second row of blades revolves. With the lower jet speeds we'll have lower rpm.

All the foregoing schemes are different arrangements of impulse blading, the force being created by a high-speed jet impinging on a slower moving surface. However, there are other ways of creating a working force on moving surfaces by means of steam jets.

Reaction Blading. Another common means of turning turbine shafts is by the so-called "reaction" force. The term isn't strictly accurate because it makes many tend to misinterpret what takes place. The most familiar everyday application of the reaction force is in some types of garden sprinklers, arranged as in Fig. 1-10. As the water jets leave the nozzles at the end of the arms, the latter rotate backward, opposite to the direction of the jets. Popular misconception says they do this because the jets "kick off" from the nozzle. This hardly explains the real state of affairs.

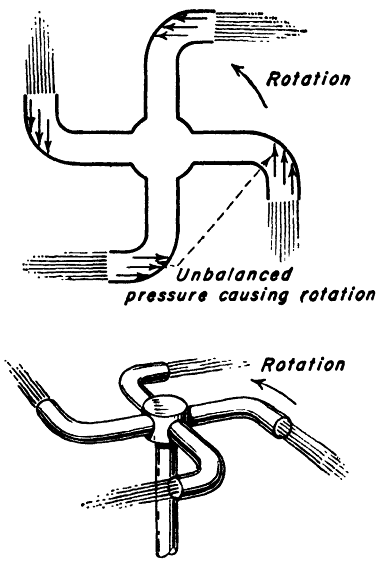


Fig. 1-10. Unbalanced-pressure forces acting in a direction opposite to jet flow in a lawn sprinkler make it rotate. This is the so-called "reaction" force.

Suppose we plug up the four nozzles tightly and turn on the water pressure or air pressure. (They will turn also when compressed air passes through them.) Nothing takes place. Why? Because the fluid under pressure presses on all parts of the sprinkler and the plugs equally. The pressure that acts on the plugs is balanced by the pressure on the curved wall leading into the nozzles. When the plugs are removed, the water can flow out under its pressure. But now the pressure on the curved walls directly behind the nozzles is not balanced by the pressure on the plugs, and since the arms are free to move, these unbalanced forces proceed to make them rotate. In this way work can be done by these unbalanced or "reaction" forces. It's the same sort of force that moves a rocket. The high pressure created in the rocket's combustion chamber by the burning gases presses equally on all parts of the wall of that chamber. But the outlet where the jet issues cannot balance the pressure on the opposite wall, and hence this pressure creates the force that moves the rocket.

Again after experimenting, we come up with a blade and wheel arrangement like that shown in Fig. 1-11. Nozzles are formed on both moving and stationary rows with the result that the steam pressure drops through both. Again, a high-speed snapshot would look like the streamlines in Fig. 1-11 and give the peculiar impression that the steam is flowing "through" the moving blades. As steam pressure drops in a nozzle, the steam jet speeds up. In the stationary nozzles, the jet speeds up enough to smoothly enter the inlets of the moving blades, or nozzles. There may be a moderate amount of "impulse" pressure helping to create the working force, but the main part of the force is created by the unbalanced pressure of the jet leaving the moving blading. The direction of this moving force is indicated by the arrows on each blade. The blades of course can move only downward.

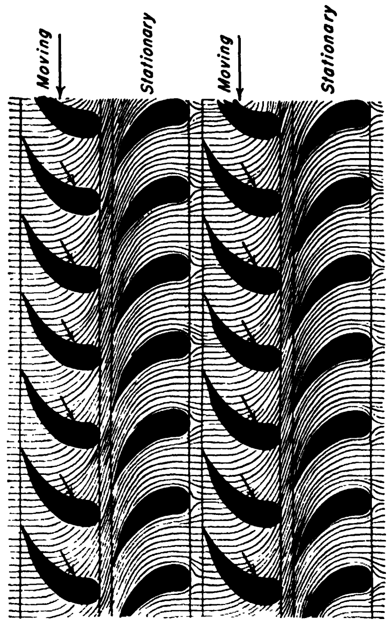


FIG. 1-11. In reaction blading, steam pressure drops in both stationary and moving blading (or stationary and moving nozzles). While flowing through moving blades, steam accelerates because of pressure drop, and unbalanced forces act on the blading in the direction shown by the arrows. Blades can move only directly downward because of mounting on the revolving drum. Note that steam fills all space between blading, a characteristic of reaction turbines.

Again this seems queer. From our viewpoint on solid earth, the steam accelerating through the moving nozzles (blading) is moving from right to left, but the reaction force is acting almost directly downward in the direction of blade motion. It's all in the point of view. In Fig. 1-12*b*, we again reproduce the moving blades and the steam streamlines as they appear from a stationary position. But suppose you were riding on one of the blades. How would things look? This is shown in (*a*). The steam would appear to be coming in parallel, or almost parallel, to the blade (moving nozzle) entrance at a moderate speed and then speeding up as it passes between the blades. The unbalanced pressure must be in the direction opposite to that in which the jet leaves. Now we can see why the reaction force acts in the direction shown in Fig. 1-11.

While we're on the subject, let's make the same investigation for moving impulse blading. In Fig. 1-12*d*, we see how the blading and steam flow appear from a stationary viewpoint, while (*c*) shows how it would appear if we were riding with the blades. The steam would appear to enter from the right parallel with the sides of the blade, flow through at a constant rate, and leave with the same speed at the exit edges of the blades to the left. To keep the record straight, you will often hear impulse blading on the moving wheel referred to as "buckets." Through this book we occasionally use this term.

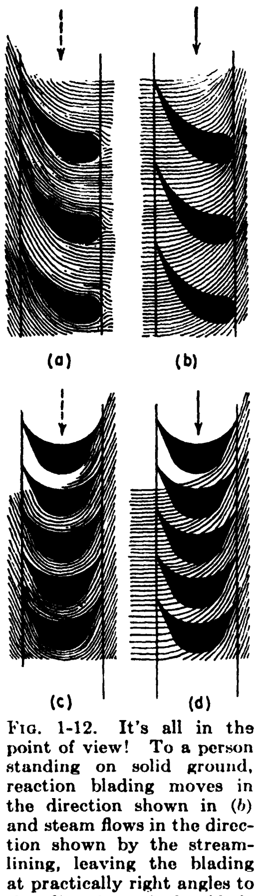
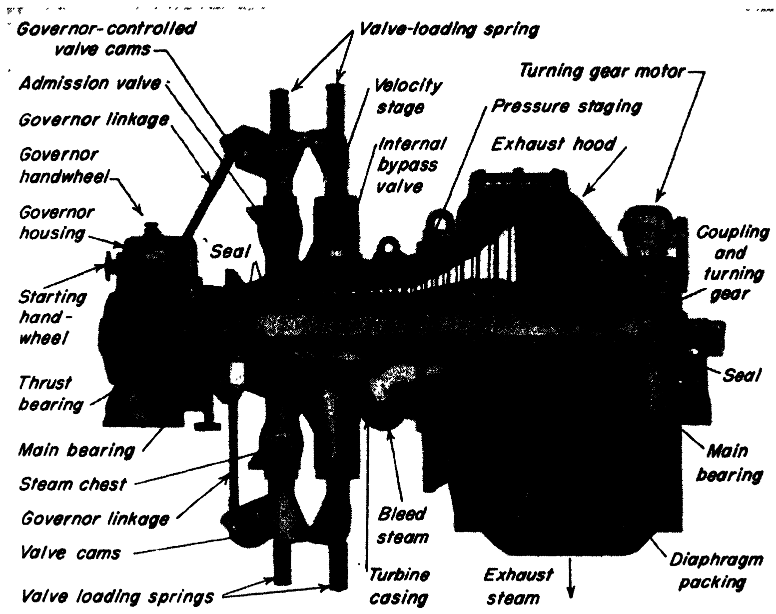


FIG. 1-12. It's all in the point of view! To a person standing on solid ground, reaction blading moves in the direction shown in (*b*) and steam flows in the direction shown by the streamlining, leaving the blading at practically right angles to the direction of the blade motion. However, if he rode on the blading, it would appear to be stationary and the steam would appear to flow as shown in (*a*). The same relations hold for impulse blading; as things would appear to one standing on the ground (*d*) and as they would look to one riding on the blades (*c*).

Turbines. We have discussed the action taking part in the heart of a turbine in a very elementary way. A more technical discussion will be given in Chap. 5, when we talk of steam fundamentals and go into calculations concerning turbine behavior. Also, we have talked of a very few elements in the turbine, the most vital ones, but there are many other parts needed to make the turbine practical. Figures 1-13 and 1-14 show cross sections of impulse and impulse-reaction turbines, respectively. Let's look at these pictures carefully. There must be a *rotor, shaft,* or *spindle* to carry the blades or buckets and a *shell, casing,* or *cylinder* to confine the steam, support the stationary nozzles or blades, and provide a structural frame.

The casing supports the *main bearings* and the *thrust bearings*, which maintain the shaft's axial position. Reaction turbines, characterized by greater end thrust (because of pressure difference across moving blading), use *dummy* or *balance pistons* to counterbalance most of this thrust. Seals and glands at the points where the shaft emerges through the casing minimize steam leakage out of the unit or air leakage into the unit. A lubrication system supplies oil to the bearings and other moving parts.

To control steam admission requires a *throttle* or *stop valve*, a *steam chest*, *steam-admission valves, valve gear,* and a *governor.* An *overspeed governor* and *trip mechanism* usually complete the control equipment. To withdraw steam from an intermediate point in the turbine for feed-water heating or process use, a *bleed* or *extraction connection* must be provided. An *extraction valve* controls extraction steam pressure. Fully expanded steam leaves the turbine through an *exhaust hood.* In Chaps. 2, 3, and 7,

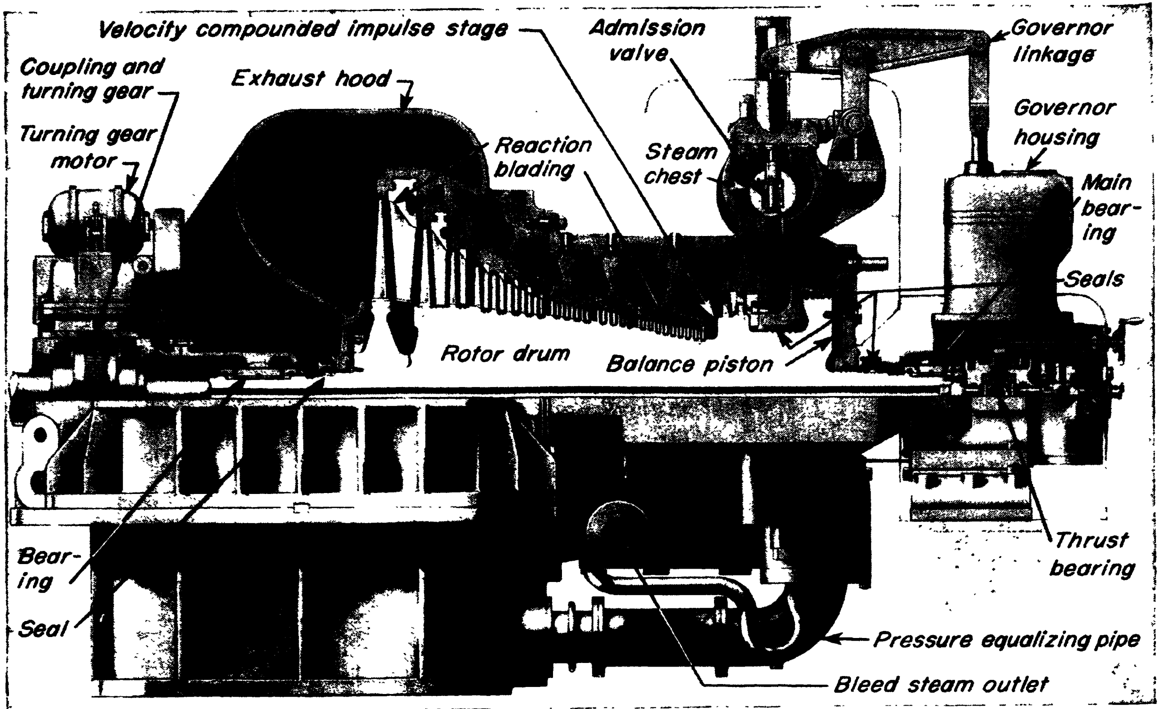


(*Courtesy of General Electric Co.*)

FIG. 1-13. Typical impulse turbine arrangement with first stage built as a velocity-compounded impulse stage. Note the wheels carrying buckets or blades, and the diaphragms carrying nozzles.

we shall talk about these various parts, how they are constructed, and what they are supposed to do.

Turbine Types and Applications. Steam turbines can be fitted into a wide variety of applications in point of size, speed, steam flow, load, and other considerations. Smooth or constant torque characteristics simplify the operation and maintenance of turbines, as compared with reciprocating engines. The nature of the metals of which turbines are made represents a real limitation on the pressure and temperature of the steam that turbines can use; temperature being the critical factor. But as metallurgy produces improved materials, higher and higher temperatures are



(*Courtesy of Westinghouse Electric Corp.*)

FIG. 1-14. Typical reaction turbine arrangement with first stage built as a velocity-compounded impulse stage. Note the drum carrying moving blades.

being used; the highest at this writing is 1050 F, and it is only a matter of time before 1200 F and higher may be put to work.

Ability to take part of the steam from a turbine at an intermediate point (stage) proves a big advantage in favor of the turbine. Since steam, in most instances, is generated by a boiler at one pressure, lower

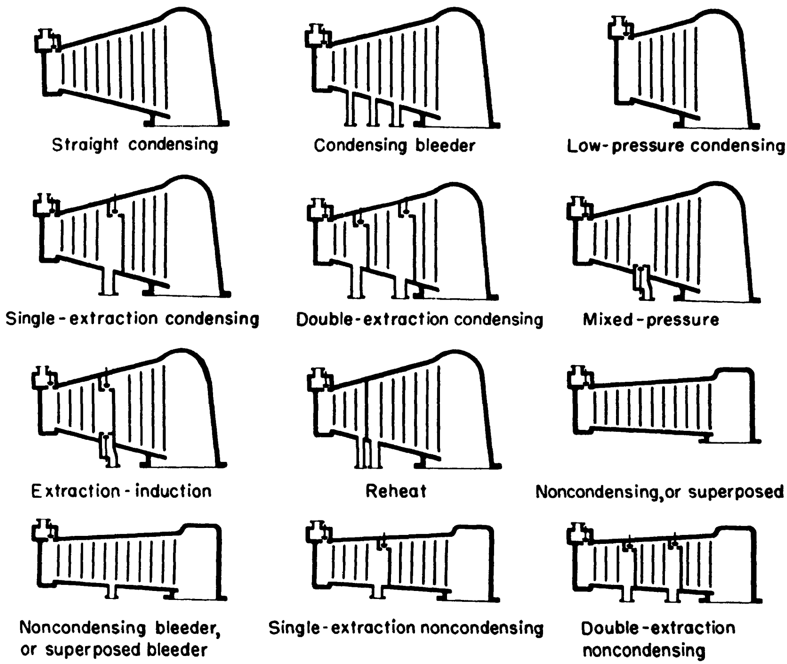


FIG. 1-15. Diagrammatic chart of the principal types of turbine arrangements in regard to steam flow through the unit. Because the diagrams illustrate principles, superposed units are included with the broad class to which they belong, *noncondensing turbines*, although they are commonly thought of as a separate type. *Dual-pressure* units, machines designed to operate at two different pressures or to be converted later to another pressure, might be considered an additional class.

pressure steam can be taken from an appropriate stage of a turbine, thus being made to do work in the pressure-reducing process. This proves more economical than just throttling high-pressure (h-p) steam in order to obtain low-pressure (l-p) steam. Such lower pressure steam, at any level, can be used for process heating work in factories, for space heating in almost any establishment, and for feed-water heating in a straight power-producing turbine plant.

Arrangements of steam flow in and out of turbines are diagrammatically shown in Fig. 1-15. All turbines may be divided into two broad classes:

(1) *condensing* units which operate with back pressures (pressure at exhaust) less than atmospheric pressure (14.7 psia) and (2) *noncondensing* units with back pressures at atmospheric pressure and above.

Each class may be subdivided according to whether full throttle flow continues through the machine to exhaust or whether part of the steam leaves at an intermediate stage on its way through the machine. Units of the latter type are called *bleeder* or *extraction* turbines. These terms are used loosely and more or less interchangeably, but for our purpose we will use *bleeding* to refer to steam withdrawal in minor amounts, without pressure control, as applied in feed-water heating layouts; *extraction* will be used for steam withdrawal in small to large amounts at a controlled constant pressure.

In a bleeder turbine, one or more stages have openings of fixed size through which steam may be taken or bled. Since stage pressures vary with total steam flow through the turbine (rising with increasing flow), the bleed steam pressure will vary. This will be discussed at greater length in Chap. 5, Figs. 5-27 and 5-31.

When steam taken from a turbine must be at a constant pressure, as in process work, some automatic pressure control must be provided. Such turbines are called *automatic-extraction* turbines. In this type of unit, the section or stage following the extraction opening is separated from the section ahead, and steam flow between them is regulated by a valve under automatic control. The action will be described in detail in Chap. 6, Figs. 6-15 to 6-17.

The number of bleed points used, ranging from one to eight, is determined in most cases by an economic analysis of alternate arrangements. Besides fixing the number of points, heat-balance requirements determine the approximate pressure levels.

Extraction turbines are built with one or two extraction openings, the number and pressure levels being determined by process needs.

In some cases, l-p steam is available from process lines in addition to the h-p steam generated in a boiler. Such l-p steam can be put to work in a *mixed-pressure* turbine, by introducing it at an intermediate stage. Whenever it becomes available, the h-p steam entering the throttle is backed off until load requirements are met. When a condition arises where l-p steam is needed sometimes from an extraction turbine and at other times excess is available for introduction to the turbine, an extraction unit is used and called an *extraction-induction* turbine.

A number of other special designs have been developed. Where high throttle pressures have been employed with relatively low temperatures (and in some modern cases with high temperatures) to gain high thermal efficiency, all steam is taken from the turbine at an intermediate stage and resuperheated before being led back into the following stage. Such a unit is called a *reheat* turbine.

Topping or *superposed* turbines have been applied in some instances. Since old moderate-pressure turbines are nearly as efficient as the l-p sections of modern h-p turbines, it sometimes pays to install h-p boilers feeding h-p turbines that exhaust at original station pressures into the existing moderate-pressure turbines. This combination operates at

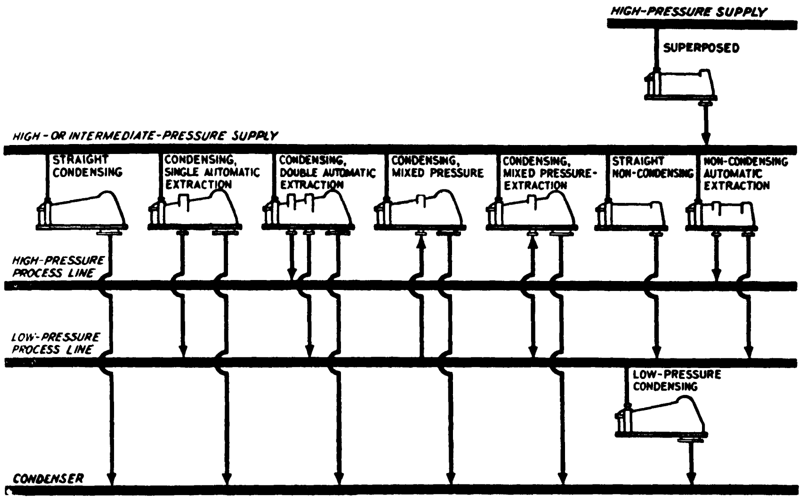


FIG. 1-16. Relative hookups of typical turbine types in regard to steam pressure. Shows how available turbine types fit various heat-balance arrangements. No definite values are assigned to the pressure levels shown as it is intended that they be purely relative and illustrative.

nearly the same efficiency as a complete new h-p plant and salvages investment in existing turbines, condensing systems, and structures. These turbines are similar to standard noncondensing units but operate at higher initial and back pressures.

What might be called *subposed* turbines have been used in some plants where large amounts of exhaust steam became available from engines or process equipment. Working from atmospheric pressure to a low vacuum, an *l-p condensing* turbine does about the same work per pound of steam as a unit expanding steam from 200 psig to atmospheric pressure.

Where plant development plans may later call for higher operating pressure or where two pressures may be available, a *dual-pressure* turbine may be applied. Such a unit may differ from the conventional only in that its design represents a compromise between best performance at each pressure level. Where the difference between two pressures is substantial, the unit may be designed so that the h-p stages may be added later.

Figure 1-16 summarizes the relations between common types of turbine in terms of pressure levels and division of steam flow.

When a turbine is built for higher capacities, it is often found impossi-

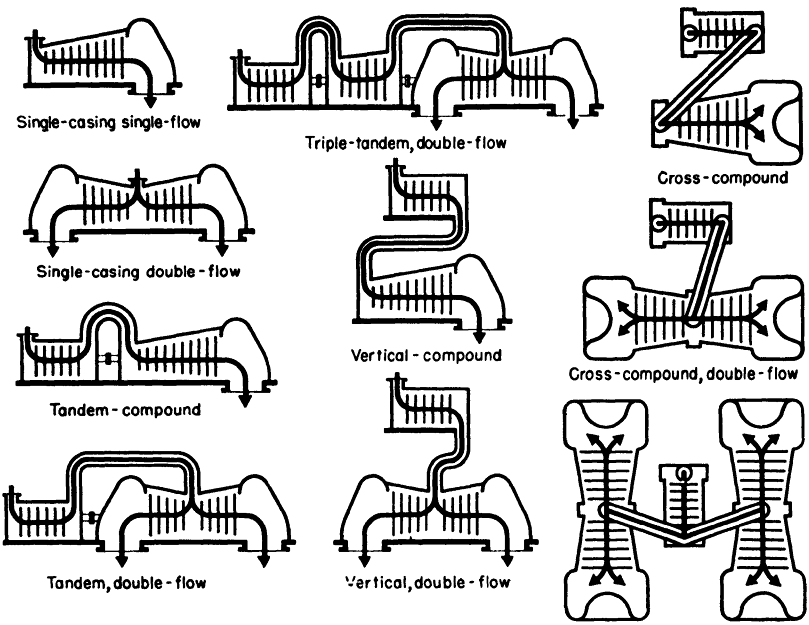
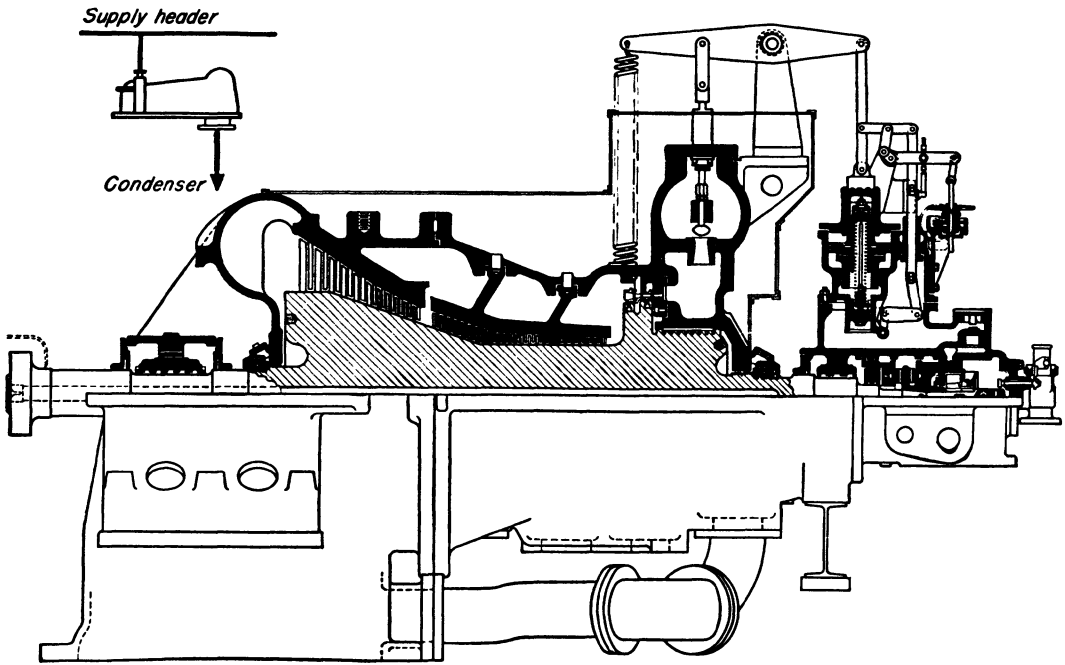


FIG. 1-17. Casing and rotor arrangements for large-capacity turbines. There is a limit to the capacity that can be obtained from the basic single-casing single-flow design. For larger capacities it is necessary to use the double-flow principle and various forms of compounding diagrammed here.

ble to provide it in a single rotor and casing. Figure 1-17 shows various casing and shaft arrangements used in the larger sized units up to 200,000-kw capacity. Actual turbine cross sections for units normally driving electrical generators are shown in Figs. 1-18 to 1-30.

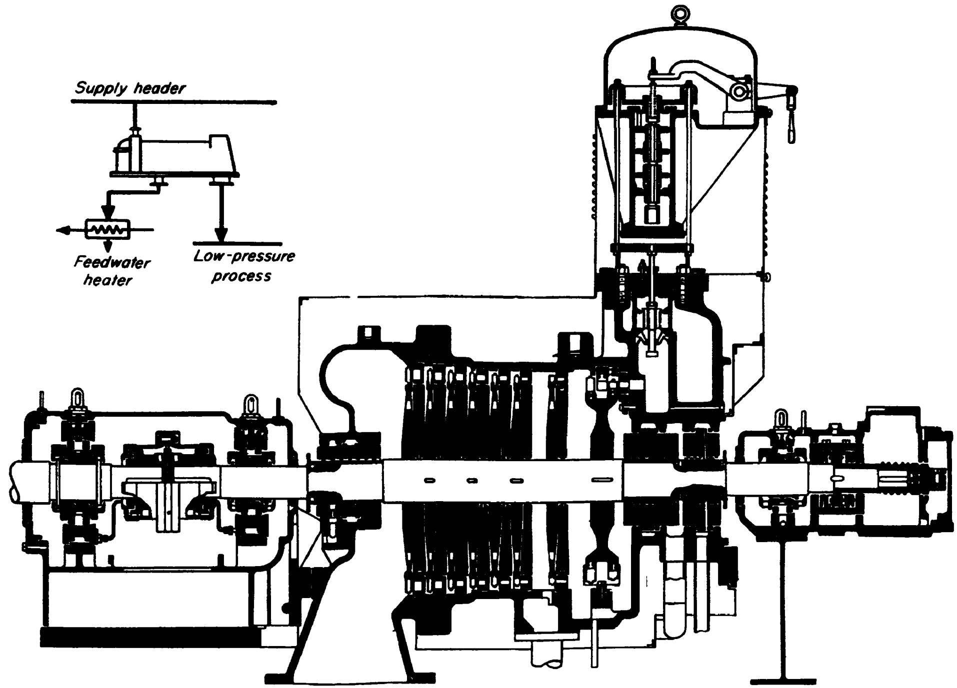
Mechanical-drive Turbines. Turbines thus far described are intended for generator service, usually direct-connected. To meet the varied needs of industry for prime movers to drive fans, pumps, compressors, and other machinery, *mechanical-drive* turbines have been developed. In this service the turbine offers these advantages: (1) easily controlled variable-speed operation, (2) ability to start quickly and without electrical power

(*Continued on p.* 31.)



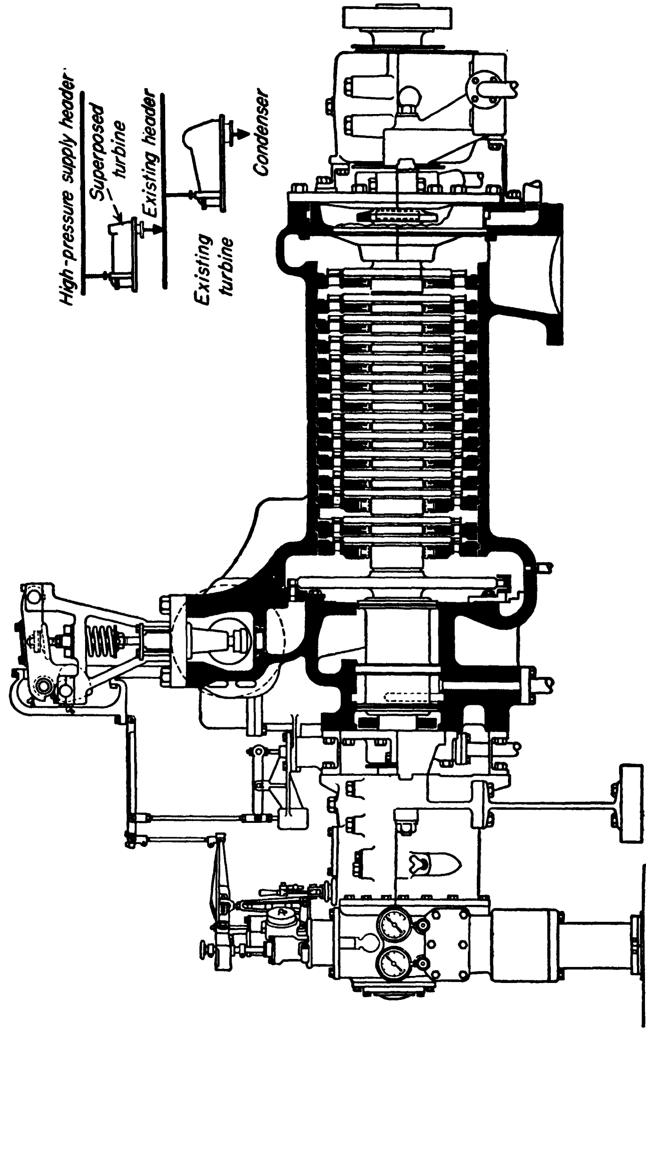
(*Courtesy of Westinghouse Electric Corp.*)

Fig. 1-18. Straight condensing turbine, fits where maximum power from throttle flow is desired, condensing water is available, and there is no need for heating steam. This impulse-reaction design has one impulse pressure stage, velocity compounded, and 36 reaction stages. Two journal bearings carry the solid rotor with its balance piston. Blade ring carries fixed blades of first 25 reaction stages. Integral multivalve steam chest contains single-seated valves actuated by bar-lift mechanism.



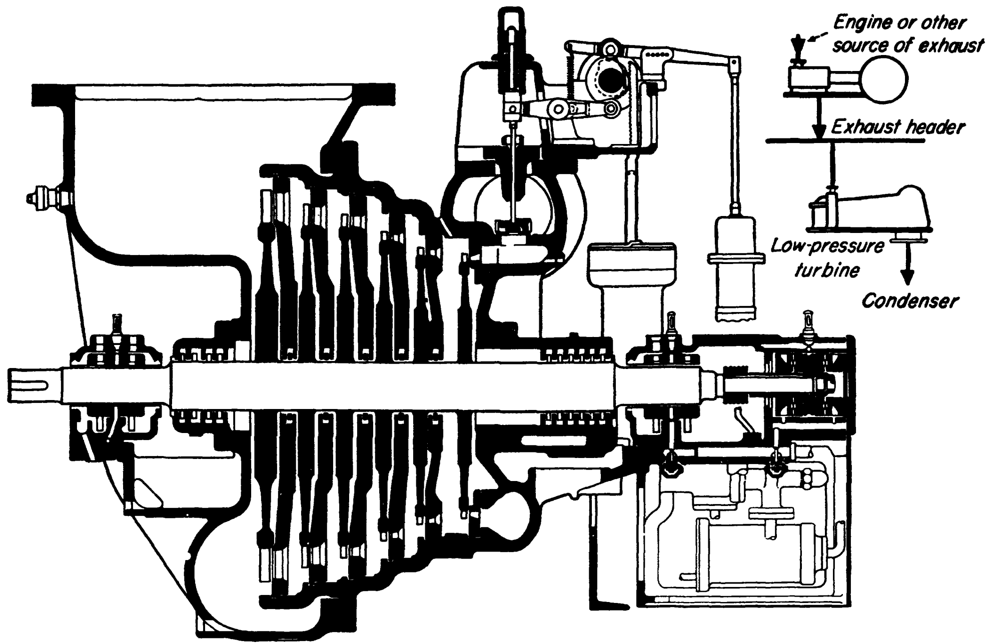
(*Courtesy of Elliott Co.*)

Fig. 1-21. Bleeder noncondensing turbine has velocity-compounded first stage, seven impulse pressure stages, one bleed point. Balanced steam-admission valves are hydraulically actuated from an oil-relay governor. Note shielding of first-row blades. This type supplies relatively large low-pressure steam demand as does the straight noncondensing, and in addition furnishes steam for feed heating.



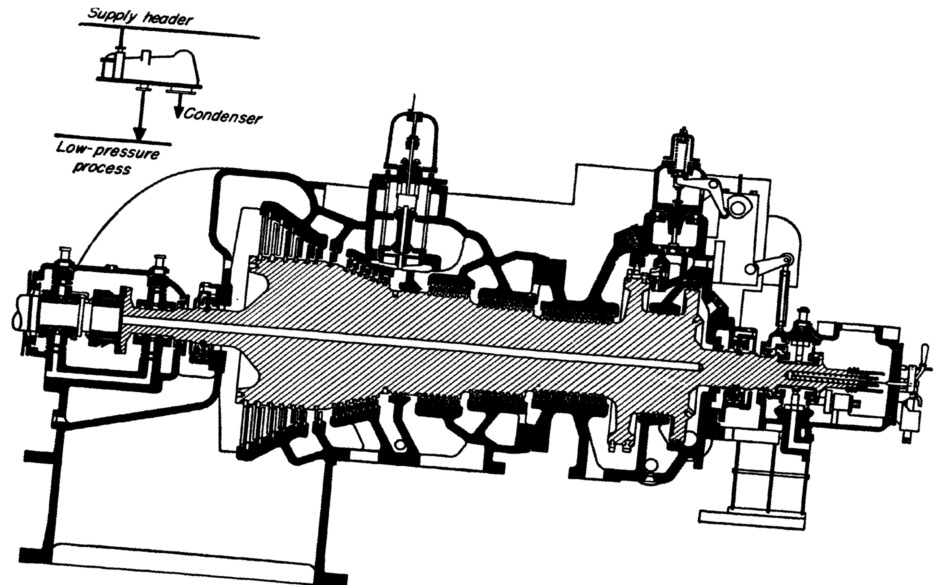
(Courtesy of General Electric Co.)

FIG. 1-22. Superposed or noncondensing turbine operates at high inlet and back pressure. Section shows the heavy, horizontally split casing and 15 impulse stages. Nozzle control to the first stage is used, without velocity compounding. High pressure requires careful sealing. Increase in plant capacity, efficiency, or both results from installation of superposed turbine exhausting to existing turbine-supply main.



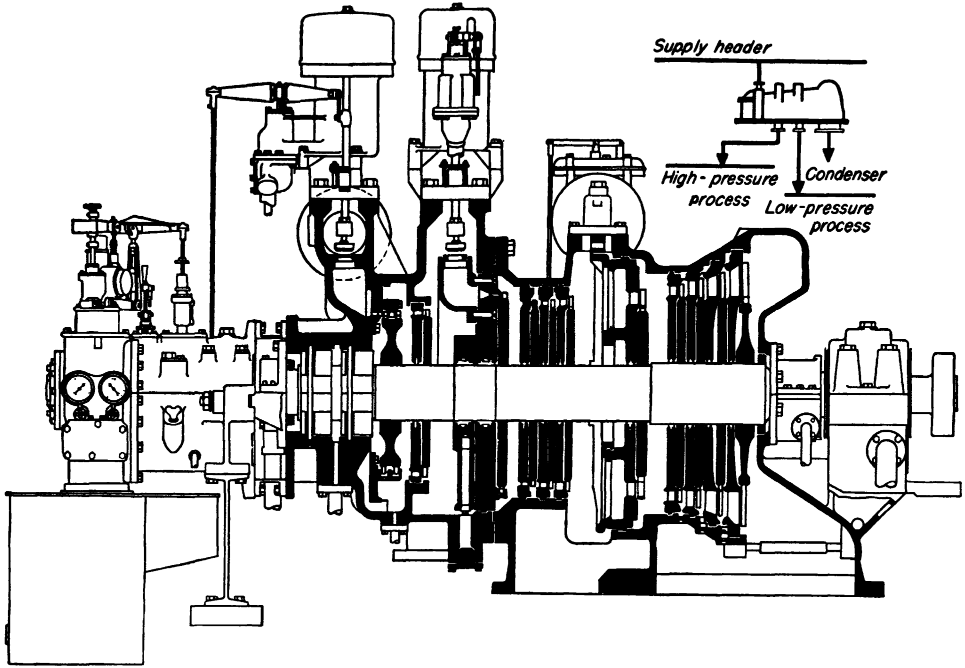
(*Courtesy of Worthington Pump and Machinery Corp.*)

FIG. 1-23. Low-pressure, or "subposed," condensing turbine. Because of the small pressure range over which they operate, they have comparatively few stages. The six-stage impulse machine above may be considered typical except for the upward exhaust, which is dictated by use of a barometric condenser or other special local conditions. The steam-admission valves are cam-lift actuated. Using excess exhaust steam in low-pressure condensing turbines often permits generating additional power without fuel increase.



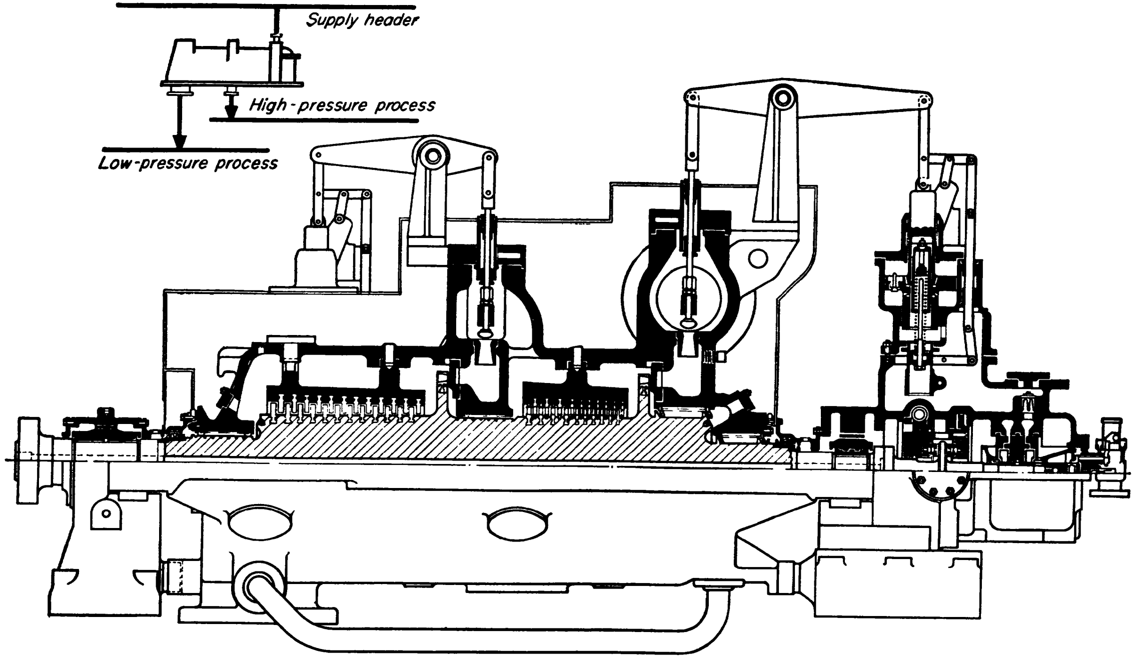
(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 1-24. Single automatic-extraction condensing turbine with both impulse and reaction stages has interstage balanced valve to hold constant extraction pressure. Rotating balance pistons are used. Both ends have labyrinth and water-runner seals. Uncontrolled bleeding for feed heating may be obtained. Turbines of this type suit applications where process-steam demand at one pressure level exists and substantial electrical load must be supplied. Process and electrical loads can vary independently within fixed limits.



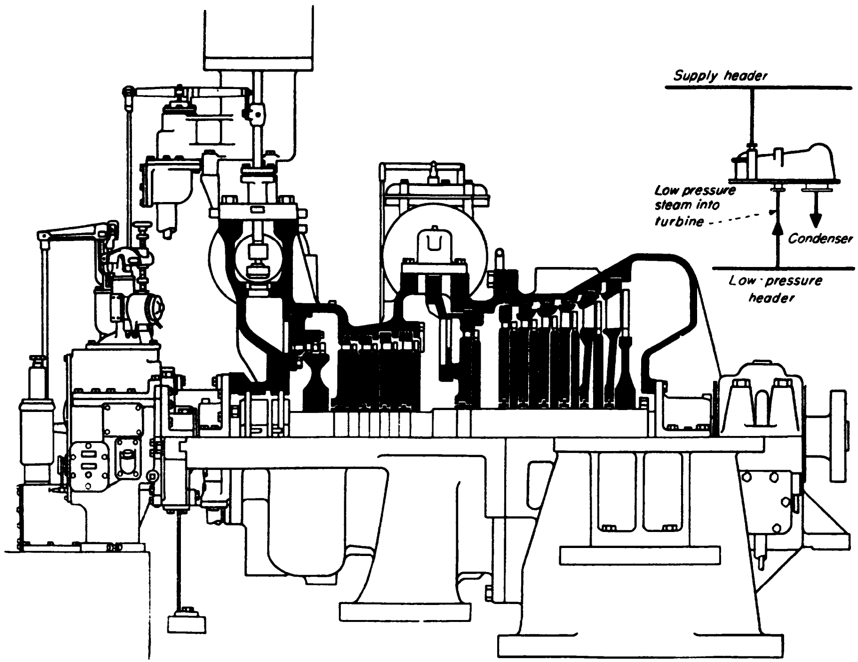
(*Courtesy of General Electric Co.*)

FIG. 1-25. Double-automatic-extraction condensing turbine of impulse design has poppet valve at high-pressure extraction point; grid valve, suited for large steam volumes, at low-pressure point. Extraction and exhaust openings are at bottom of casing, which is split two ways. Three-arm governor mechanism controls extraction pressures and speed. This unit supplies process steam at two pressure levels and operates condenser to give maximum power from net steam flow. Steam and electrical demands can vary independently, within limits.



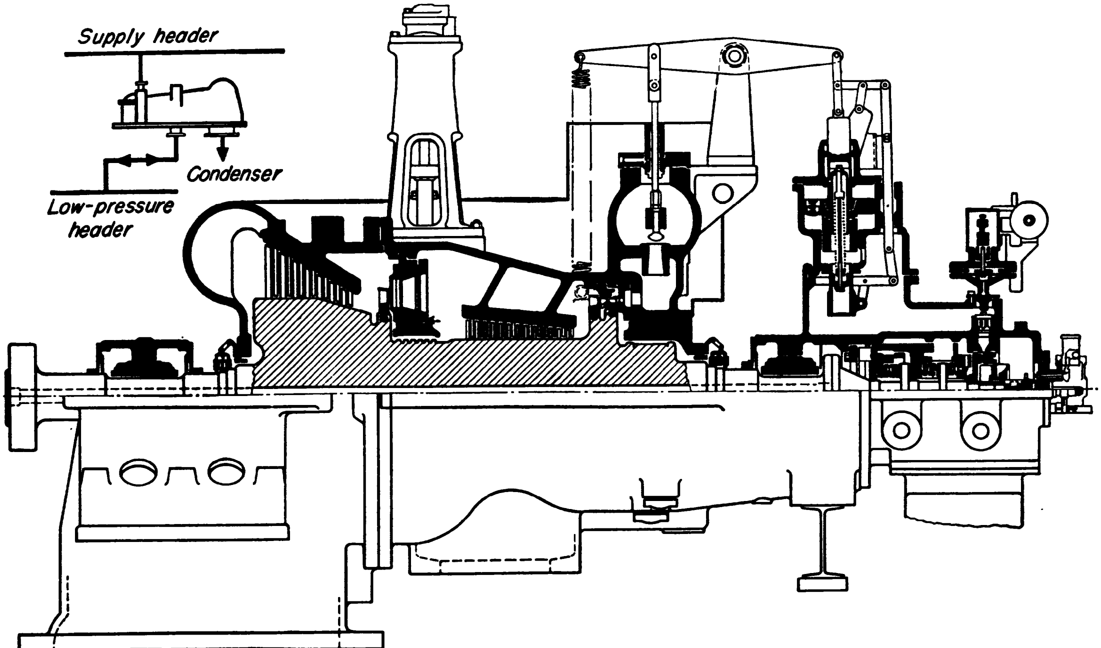
(*Courtesy of Westinghouse Electric Corp.*)

FIG. 1-26. Single automatic-extraction noncondensing turbine has an impulse stage followed by reaction stages in high-pressure section and same arrangement in section after extraction point. Steam-admission and -extraction valves are poppet type. Solid rotor is used in casing having essentially constant diameter. Blade rings supported by the casing hold the stationary blades. Single- or double-extraction noncondensing units supply relatively large amounts of low-pressure steam and smaller amounts of high-pressure steam. Normally extraction and exhaust pressures are controlled and electrical-output variation absorbed by paralleling.



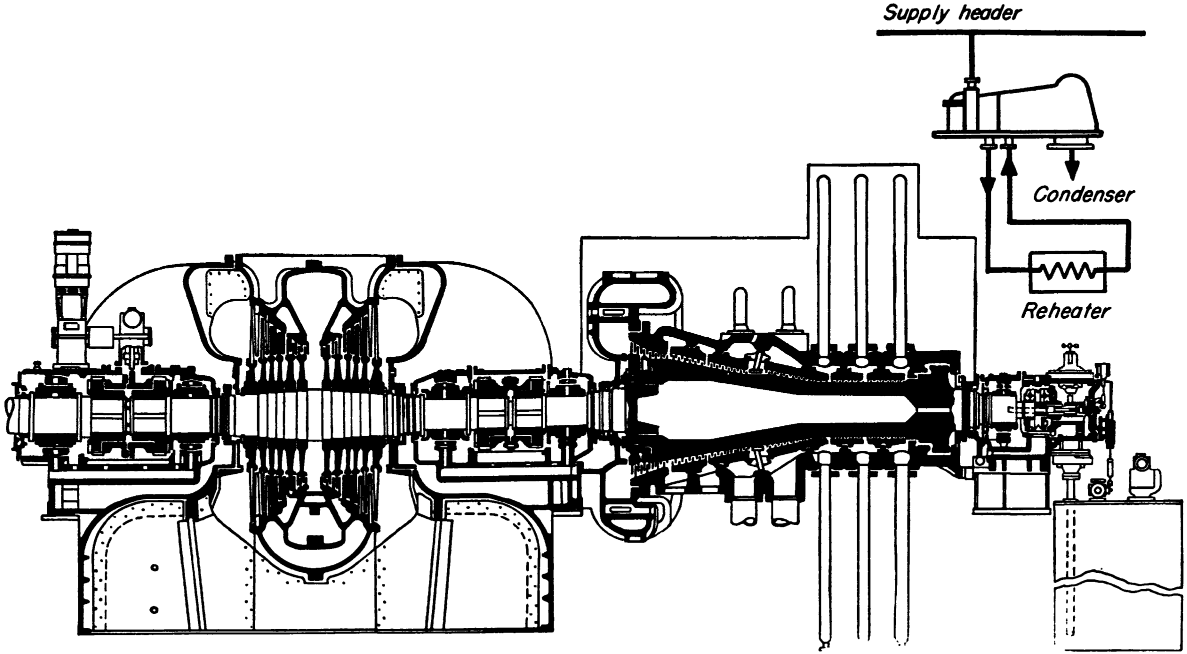
(*Courtesy of General Electric Co.*)

Fig. 1-27. Mixed-pressure condensing turbine. Where excess low-pressure steam is available in varying amounts, it can be utilized in a mixed-pressure turbine, supplementing throttle steam flow and thus effecting substantial operating economies.



(*Courtesy of Westinghouse Electric Corp.*)

FIG. 1-28. Automatic-extraction-induction condensing turbine. Mixed-pressure condensing turbines are, in effect, reversed extraction units. Control valve at point of low-pressure steam admission allows full use of excess steam. Speed governor controls flow of throttle steam to meet electrical load. Extraction-induction condensing turbine has grid valve at mid-casing to control low-pressure steam extraction or induction, poppet valves for throttle flow. Unit has double-split casing, solid rotor, water and labyrinth seals at both ends. High- and low-pressure sections have impulse stage followed by reaction stages. This unit provides for plant conditions where excess low-pressure steam is available at some times while at other times there is a demand for low-pressure steam.

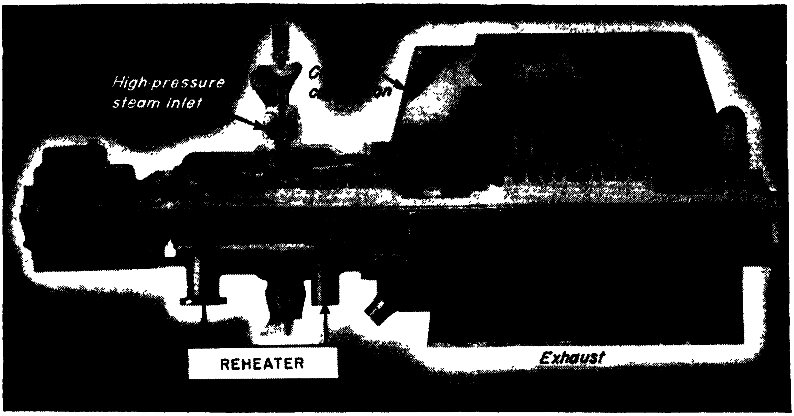


(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 1-29*a*. Reheat bleeder condensing turbine, for large output, is built in tandem-compound arrangement with double-flow low-pressure end. High-pressure section has a hollow-drum rotor; low-pressure blading is carried on individual disks on a solid shaft. All blading is reaction. Outlet and readmission openings for reheating can be seen at mid-casing in high-pressure section. Restoring steam temperature after partial expansion avoids moisture in last stages of units operating with extremely high throttle pressures without requiring excessively high throttle temperatures. Cycle efficiency is bettered.

supply, (3) freedom from spark hazard, and (4) suitability for damp or otherwise adverse conditions.

Even more important, in most cases, is the turbine's flexibility from the heat-balance viewpoint. Throttle steam can be supplied at boiler pressure, or process pressure in a multilevel plant, and exhaust steam can be used for heating and process, or it can be condensed. Turbines can also be arranged for extraction and high back pressure or topping service. Where extraction or exhaust steam can be utilized, the economical advantages of steam drive may prove substantial.



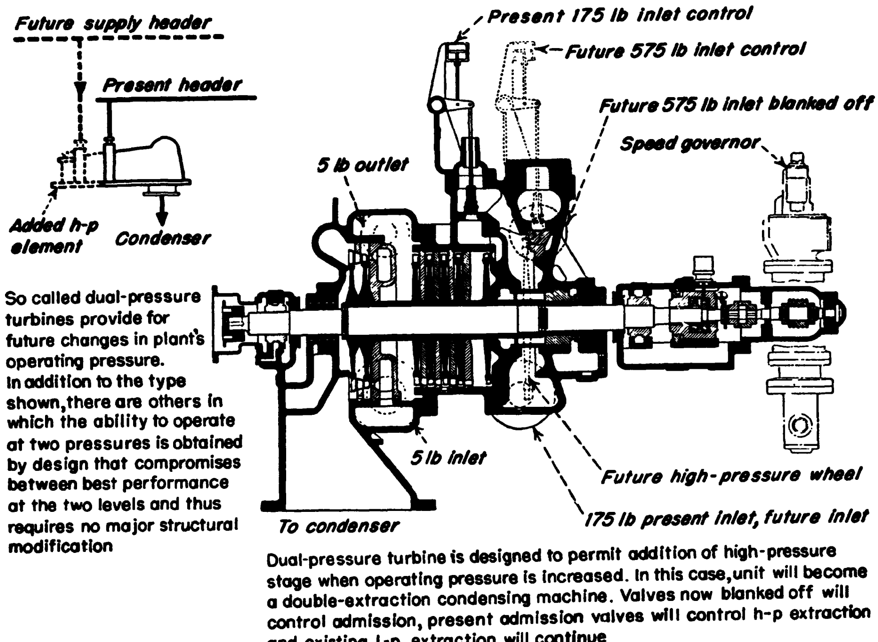
(*Courtesy of General Electric Co.*)

FIG. 1-29b. Reheat bleeder condensing turbine with h-p steam inlet adjacent to reheated steam inlet.

Mechanical-drive service calls for units of relatively small size, occasionally up to about 2,000 hp, but usually below 500 hp. Since most such units operate noncondensing and since exhaust steam is used for heating, maximum efficiency is not essential. In many cases, the best speed of driven equipment is high; where it is not, gearing or belting may be used. Thus, relatively high turbine speeds, up to and above 5,000 rpm, are permissible.

These service conditions dictate designs that feature simplicity and ruggedness to give low first cost, reliable operation, and minimum maintenance. These considerations, in turn, lead to choice of few pressure stages, usually only one. Because impulse wheels operate satisfactorily with partial steam admission and for a given speed take a large pressure drop with good efficiency, virtually all mechanical-drive turbines are of impulse design, frequently velocity-compounded.

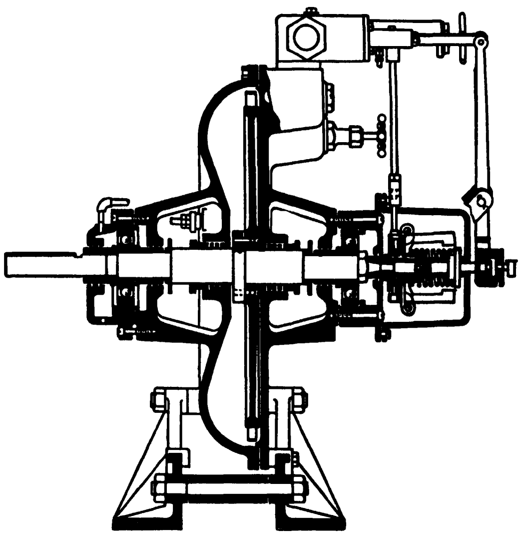
Most mechanical-drive turbines have a single pressure stage, containing either a single wheel with velocity-compounded buckets, or a number of



(*Courtesy of General Electric Co.*)

FIG. 1-30. Dual-pressure turbine for future addition of h-p stages.

wheels. Units of this type are built for capacities up to about 25,000 hp at 1,500 psi and 950 F, but the great majority are designed for capacities of from 5 to 500 hp and maximum steam conditions of 400 to 600 psi and 750 F. Most operate noncondensing at back pressures up to 50 psi, but units can be obtained for high back pressure or topping service. Maximum speed ranges up to 5,000 rpm, although some turbines have been built to operate at much higher speeds. Both horizontal and verti-



(*Courtesy of L. J. Wing Manufacturing Co.*)

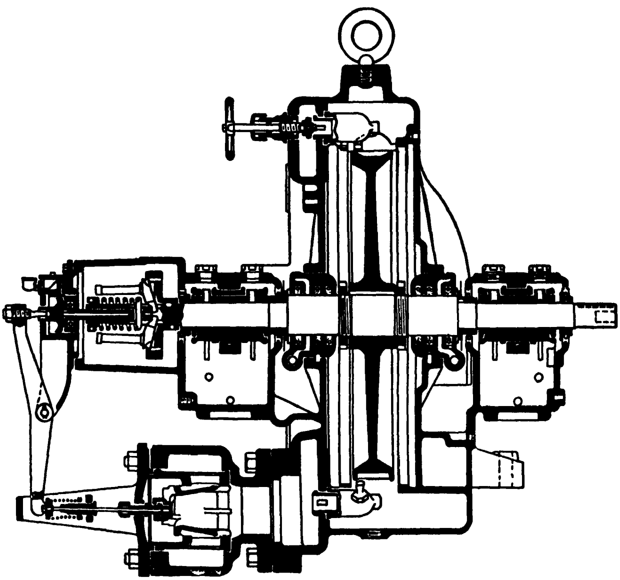
FIG. 1-31. Mechanical-drive turbine with single impulse wheel having reentry-type velocity compounding causing several passes of steam through the rotor blading (see Fig. 2-2*a*).

cal designs are available. Typical single-pressure-stage units are shown in Figs. 1-31 to 1-33.

Velocity compounding may be obtained by use of the reentry principle or by conventional velocity staging. The various reentry designs are discussed in Chap. 2. In Fig. 1-31, steam flows axially, and a chamber cast into the casing redirects the steam back through the buckets. In Fig. 1-32, nozzles direct steam into buckets milled into the rim of a solid wheel, and reversing chambers redirect the steam into following buckets. Because steam flows radially, there is no axial thrust in this design.

Units of Fig. 1-33 employ conventional velocity staging, steam from the nozzles flowing axially through a row of moving buckets and then through fixed buckets which redirect it into another moving row. The most common arrangement features two velocity stages on a single wheel, but a number of designs utilize two or more wheels, each carrying a row of moving buckets, with fixed rows between them supported from the casing.

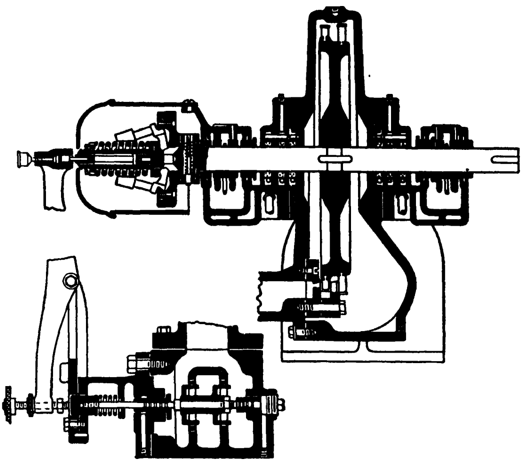
General-purpose Turbines. Small multipressure-stage turbines, designed primarily for mechanical drive but suitable, with appropriate



(Courtesy of Terry Steam Turbine Co.)

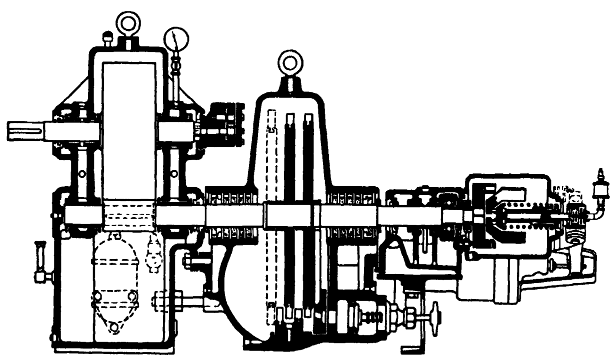
FIG. 1-32. Mechanical-drive turbine with solid rotor carrying milled-out circular-blade passages. Velocity compounding is arranged by reversing the chambers adjacent to each nozzle (see Fig. 2-2*d*).

governors, for generator service, are often considered as a separate class between the mechanical-drive units and those called industrial units, which are designed for generator service and usually built in capacities ranging from 500 to 7,500 kw. Pressure staging permits the attainment of higher efficiency than is possible in a single stage. The designs are essentially simple and more or less standardized. Normally used straight condensing or noncondensing, the multistage design permits extraction. Units may be built for high back pressure or topping service. They are usually of impulse design, for the reasons previously given, and one or two of the pressure stages are often velocity-compounded. Figure 1-34 illustrates typical general-purpose machines.



(*Courtesy of General Electric Co.*)

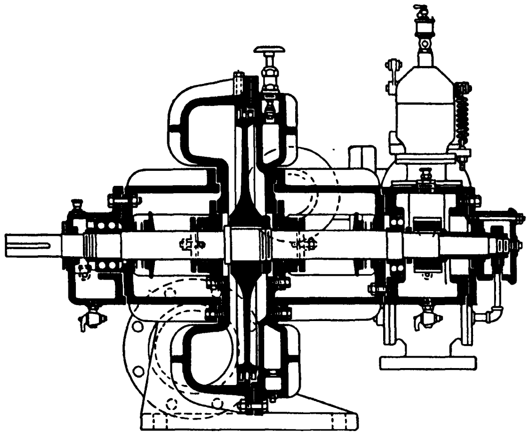
(*a*)



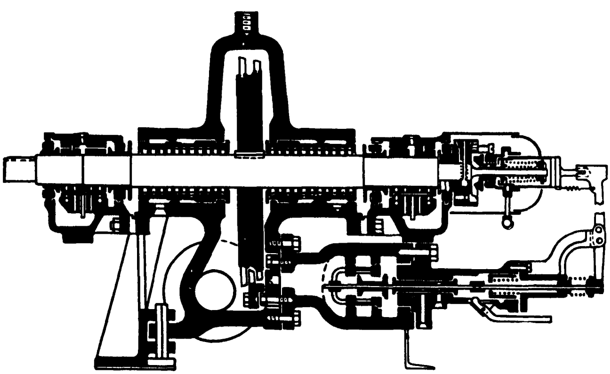
(*Courtesy of Worthington Pump and Machinery Corp.*)

(*b*)

FIG. 1-33. Various types of mechanical-drive turbines with velocity-compounded impulse stages. Most mechanical-drive turbines have a single-pressure stage, which is usually velocity-compounded. A common arrangement features two velocity stages on a single wheel, with the multiwheel construction used less frequently. Single-pressure-stage units are built for capacities up to 2000 hp at 1500 psi and 950 F, but the great majority are designed for capacities of less than 500 hp and maximum steam conditions of 400 to 600 psi and 750 F. Most units operate noncondensing at back pressures up to 50 psi but units can be obtained for higher back pressures. For most units, maximum speed runs about 5,000 rpm; some have integral gears. (*a*) Single-wheel unit with two velocity stages. (*b*) Multiwheel unit with built-in reduction gearing.



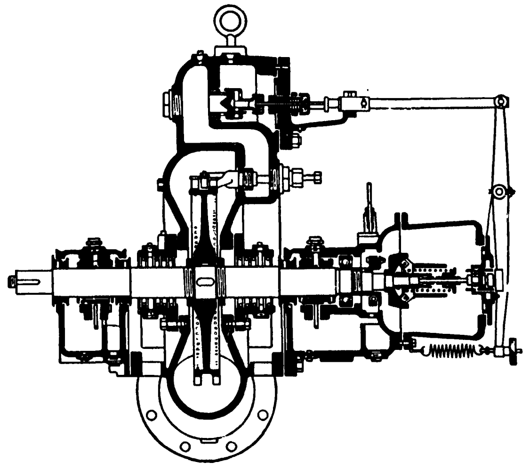
(*Courtesy of Coppus Engineering Corp.*)
(*c*)



(*Courtesy of Westinghouse Electric Corp.*)
(*d*)

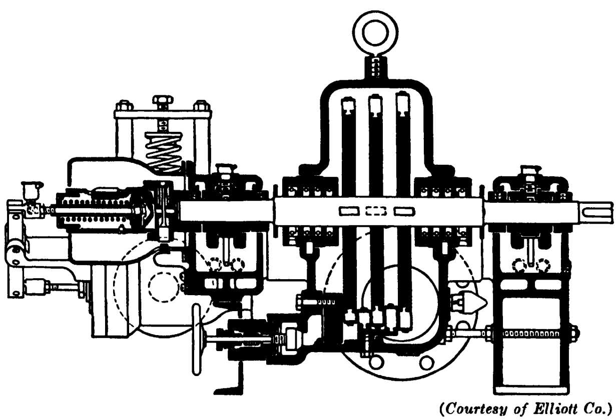
FIG. 1-33. (*Continued.*) (*c*) Another single-wheel velocity-compounded unit. (*d*) Single-wheel two-velocity-stage unit for high back pressure.

Many of the construction elements discussed in Chap. 2 apply to mechanical-drive and general-purpose turbines. Depending on pressures and temperatures, casings are of steel or close-grained cast iron, or a combination of these metals. In the customary design, the casing consists of upper and lower halves bolted together at a horizontal joint, made steamtight without use of gaskets by careful machining of flange surfaces. Making pipe connections to the lower half of the casing permits the upper



(*Courtesy of De Laval Steam Turbine Co.*)

(*e*)



(*Courtesy of Elliott Co.*)

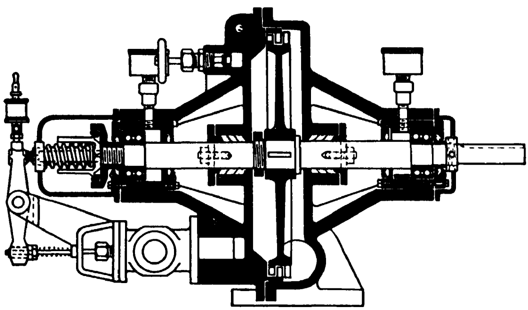
(*f*)

FIG. 1-33. (*Continued.*) (*e*) Velocity-compounded single wheel. (*f*) Turbine with three wheels and three velocity stages.

half to be lifted for internal inspection and maintenance without breaking joints.

Buckets, usually of chrome iron and fitted with shroud rings, mount on a forged-steel disk shrunk or keyed to an alloy-steel shaft. Both dovetail and T-root blade fastenings are used. For high expansion ratios, individual nozzles or drilled and reamed nozzle rings fit into the side of the casing in the nozzle chamber. Cast or assembled curved-vane nozzles serve for low expansion ratios.

Bearings and Seals. Most shafts run in babbitted journal bearings, usually with some provision, such as a babbitt-lined face, to take the normal thrust load. Ball bearings, of both normal and thrust types, are also used. Oil rings handle lubrication on many machines, while pressure systems are used on others. In many units, cored passages surround



(Courtesy of Carling Turbine Blower Co.)

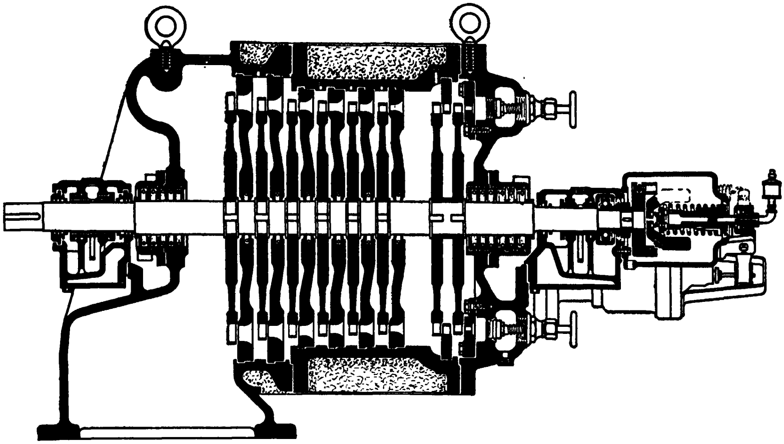
(g)

FIG. 1-33. (*Continued.*) (g) Single-wheel velocity-compounded unit.

the bottom and sides of the bearing brackets, so that cooling water can be circulated if required. Where the shaft passes through the casing, carbon packing rings are usually installed, and a space for removing leakoff steam, or supplying sealing steam, is provided. Both labyrinth and carbon diaphragm seals are used in multistage turbines.

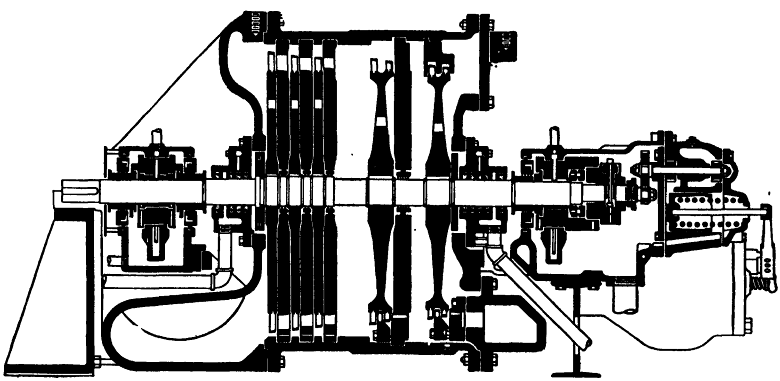
In most units a single governor valve controls steam flow, the valve being directly actuated by the governor through a short linkage. Steam passes through a strainer before the governor valve. Hand-operated stop valves for each nozzle or nozzle group permit changing the number of nozzles to meet loads above normal rating or to give higher efficiency at partial loads. To meet the desired speed characteristics of driven equipment, a number of different governor arrangements are used, as described in Chap. 7.

The method of connection to driven equipment depends on speed characteristics. High-speed machinery, such as fans, pumps, and other centrifugal or rotary machinery, is often direct-connected. Some builders offer complete units comprising both turbine and driven equipment. For many other applications, speed reduction is essential. This



(Courtesy of Worthington Pump and Machinery Corp.)

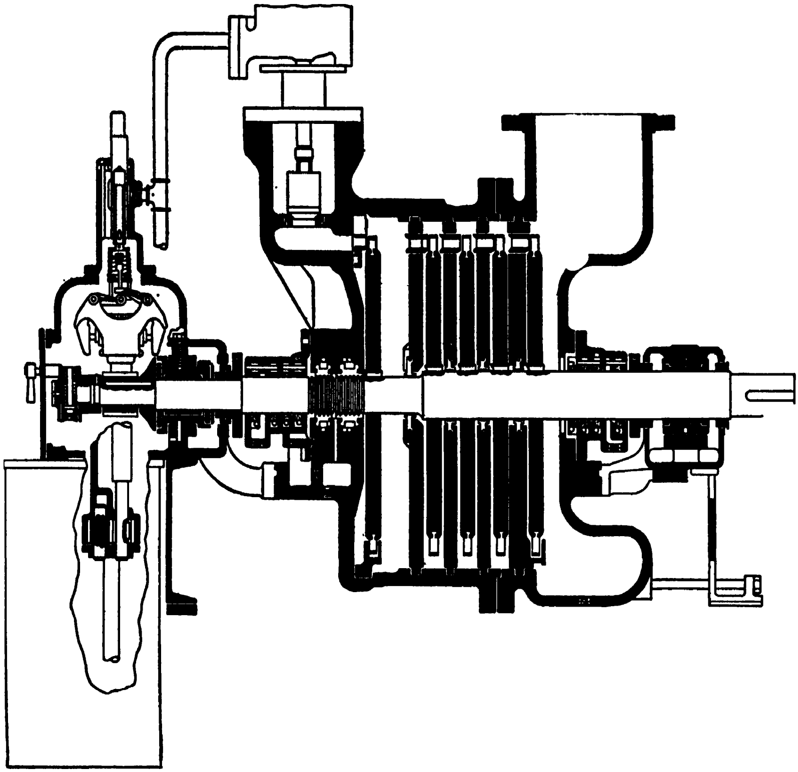
(a)



(Courtesy of Westinghouse Electric Corp.)

(b)

FIG. 1-34. Mechanical-drive or general-purpose turbines for larger capacities with pressure-compounded impulse stages. Although there is no clear-cut distinction, small multistage turbines (up to about 2,000 hp), designed primarily for mechanical drive but suitable for generator service, are often classed as general-purpose units. They are characterized by a relatively small number of pressure stages and more or less standardized design of an essentially simple and rugged nature. Normally straight condensing or noncondensing, they may be equipped for extraction and for high-back-pressure or topping service. They are usually of the impulse type, with one or two velocity-compounded stages. (a) Unit with seven stages, first velocity-compounded. (b) Of five pressure stages, the first two are velocity-compounded.



(*Courtesy of General Electric Co.*)

(*c*)

FIG. 1-34. (*Continued*). (*c*) Turbine has five pressure stages.

is most commonly obtained by gearing, but V- and flat-belt drives are also used. A number of turbines are built with reduction gearing as an integral part of the unit, offering compactness and insuring correct and permanent alignment. Gears are usually cut helically from forged steel with low tooth pressures and pitch-line velocities for quietness and long life.

CHAPTER 2

CONSTRUCTIONAL DETAILS

Nozzles. There are two basic types of nozzle: (1) the converging-diverging design for high steam-expansion ratios, and (2) the converging design for low expansion ratios. These may be built in a number of ways, depending on whether they are to be used in the first or later stages. The nozzle material is selected for use under specified operating conditions and is often a chrome-iron alloy, or at times of bronze. Nozzles for single-stage impulse turbines are usually of the converging-diverging type. Figure 2-1 illustrates such a nozzle. A number of these are generally placed at intervals around the turbine wheel. The nozzle shown is integral with a set of stationary reversing blades. These, as indicated, are required when the single-stage units are velocity-compounded. In the conventional arrangement, steam from the nozzle passes through a row of moving buckets or blades, then through a fixed row, and then through another moving row. Reversing blades are required only in the vicinity of the nozzles and do not extend all the way around the blade-wheel periphery.

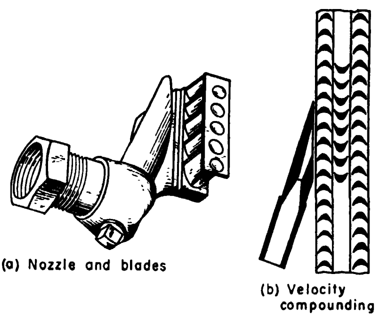


FIG. 2-1. High-expansion-ratio nozzle and attached reversing blade.

Velocity compounding may be obtained with a single row of moving blades, by use of the reentry principle. The nozzles for these designs are commonly part of a reversing chamber assembly, which redirects the steam back through the blades. In Fig. 2-2*a*, such a chamber is shown with the steam flow in an axial direction. In Fig. 2-2*c* and *d* the turbine has a solid wheel, and buckets are milled out of the rim. Steam flows from the nozzle radially and is redirected by reversing chambers, as shown in phantom. Figure 2-2*d* shows another design with a reversing chamber for a similar type of turbine wheel.

In smaller units with a single governor valve, the nozzles placed around the wheel are commonly designed with individual hand- or automatic-operated nozzle valves, as in Fig. 2-3. This permits changing the number

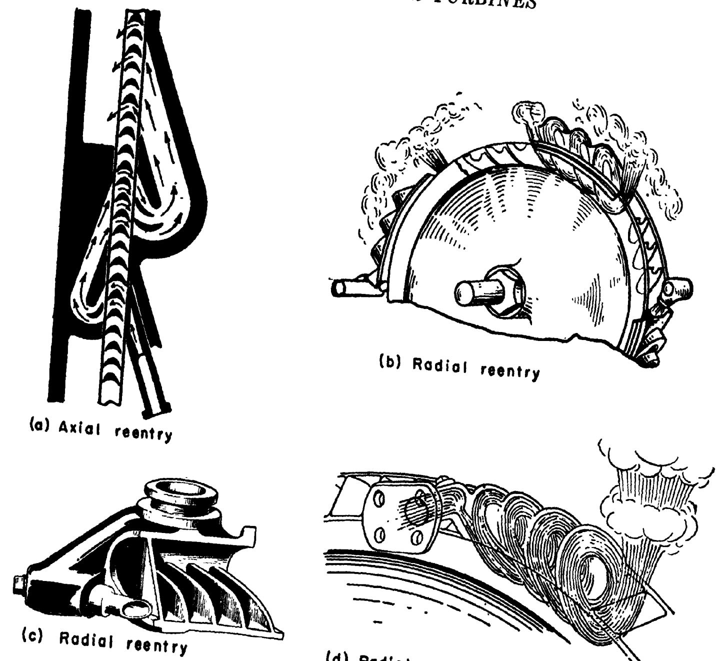


FIG. 2-2. Simple nozzles used in reentry designs.

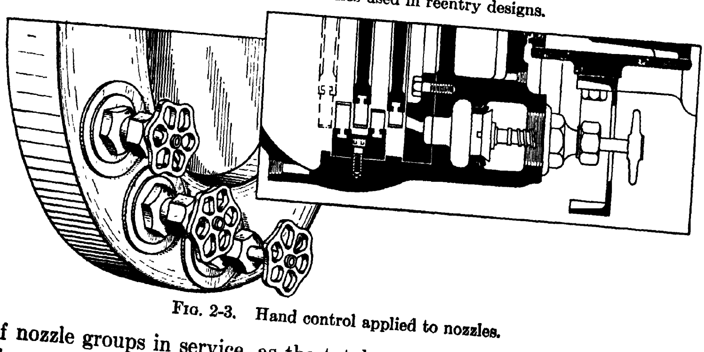


FIG. 2-3. Hand control applied to nozzles.

of nozzle groups in service, as the total steam flow changes. Changing the number of nozzles in service enables a turbine to deliver its rated capacity with reduced steam pressure, permits liberal overload capacity with normal steam pressure, and achieves good economy at low loads by cutting some nozzles out of service. The closing of hand-controlled

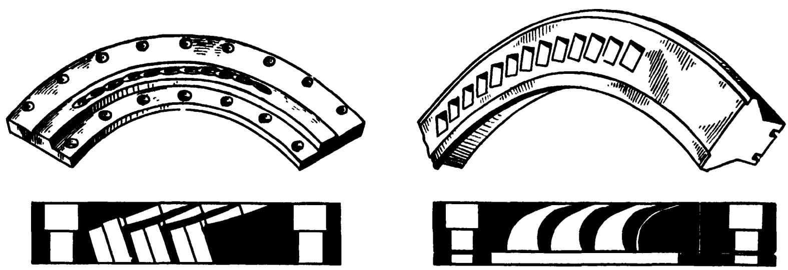


FIG. 2-4. Nozzle designs for large turbines.

nozzles for part-load and normal steam pressure reduces the effect of throttling through the main governor-controlled valve and so provides practically full-load economy at partial loads.

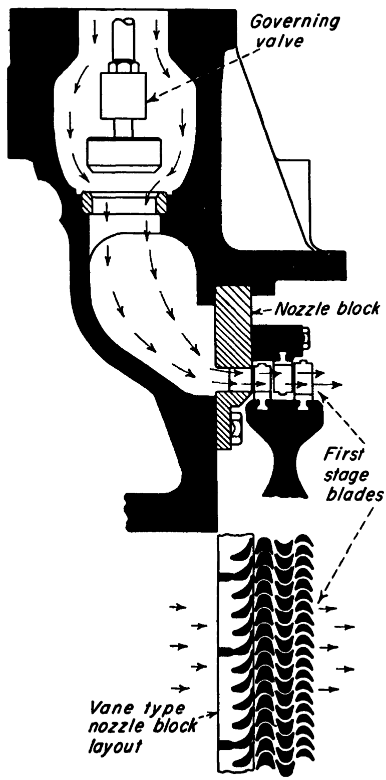
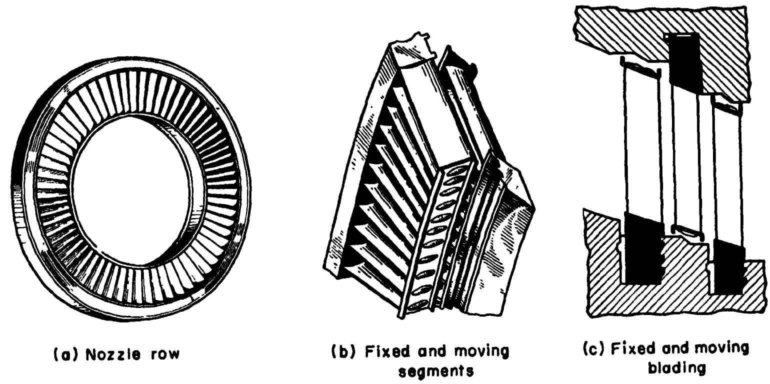


FIG. 2-5. Nozzle block in relation to governing valve and first-stage blades.

In some single-stage units and the first stage of multistage turbines, nozzles are grouped to form segments of a circle. These are assembled in the casing with blanks between. The nozzles occupy only enough of the circumference to provide the area needed for steam flow. For converging-diverging designs, as in Fig. 2-4*a*, the blocks or rings are usually drilled and reamed in a solid block of steel to the correct size. Nozzle blocks or rings may be bolted or welded in the casing. Converging nozzles are generally formed as a vane type of nozzle block, as in Fig. 2-4*b*. Nozzles of the vane type are constructed of accurately machined segments and are commonly secured to support rings by welding. This type of nozzle may be cast when used in the diaphragms of the lower pressure stages of a turbine. The arrangement of nozzle blocks



(a) (b) (c)

FIG. 2-6. Typical reaction nozzles.

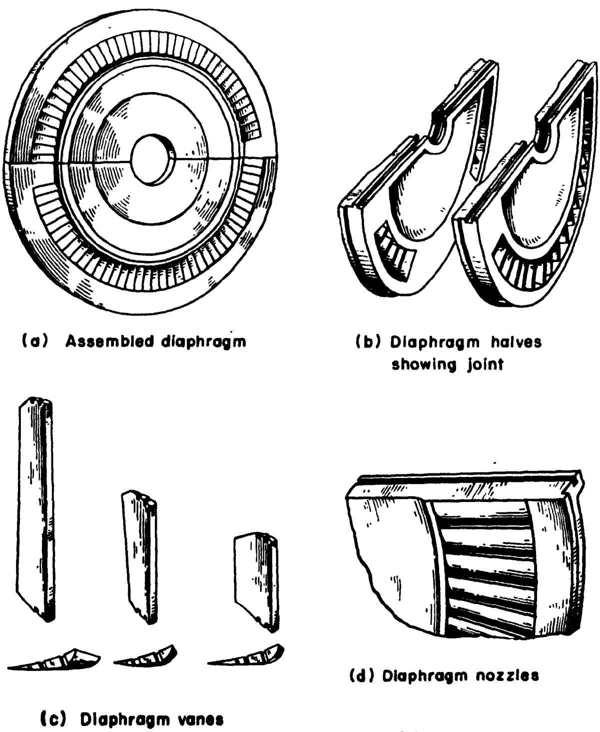


FIG. 2-7. Diaphragm structural features.

in an impulse turbine is shown in Fig. 2-5. Here, inlet steam enters through the governor valves in the steam chest and flows to the nozzle blocks. The nozzles, as shown below, direct the high-velocity steam to the first-stage blades.

In reaction turbines the alternating fixed and moving blades are in effect all nozzles. Reaction blading requires that steam be admitted to 100 per cent of the periphery. The fixed blades are virtually a ring of vane-formed nozzles, as shown in Fig. 2-6. Blade roots may fit directly into the casing or into blade rings supported from the casing. The blades or vane nozzles may be individual or grouped in sections with a common root and shroud ring.

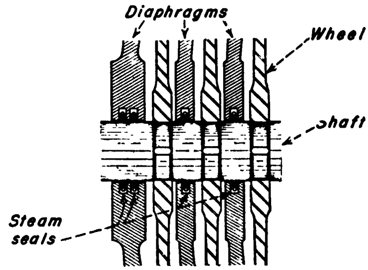


FIG. 2-8. Shaft seals.

Diaphragms and Seals. The stages of an impulse turbine are formed by stationary diaphragms and rotating turbine wheels. Nozzles for stages following the first are built into the diaphragms. As shown in Fig. 2-7, the diaphragms are split horizontally at the center line. The upper and lower halves of the diaphragms are commonly fitted into grooves machined into the turbine casing. The upper half is usually arranged so that it will lift with the upper half of the turbine casing. Correct alignment is assured, and interstage leakage prevented, by the tongue-and-groove joint shown in Fig. 2-7*b*. Diaphragms are made of close-grained cast iron or cast steel, depending on the temperature and pressure of the steam and the pressure drop across the diaphragm. The nozzles are formed of vanes commonly of chrome-iron alloy in the higher pressure stages. These nozzle vanes, as in Fig. 2-7*c*, are machined to exact shapes and are commonly welded into inner and outer bands; then the assemblies are welded into an outer ring and inner web. In the lower pressure stages, the nozzles are formed by casting nozzles or vanes into the diaphragm, which is then annealed and machined.

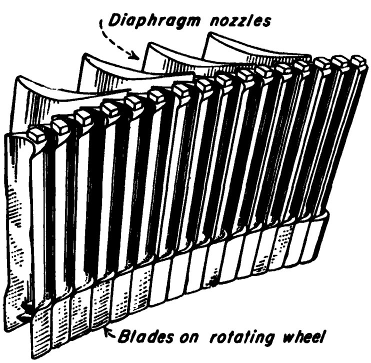


FIG. 2-9. Relation of diaphragm nozzles and turbine blades.

The rotating shaft carrying the turbine wheels, as in Fig. 2-8, passes through the stationary diaphragm. Shaft packing or seals are required to prevent steam leakage from one stage to the next lower one. The type of packing depends on steam conditions and may consist of carbon rings, metallic labyrinth, or a combination of both. The relation of the diaphragm nozzles and turbine wheel is clearly indicated in Fig. 2-9. Nozzle passageways direct the steam into the blades of the succeeding impulse wheel.

Blades or Buckets. The moving elements which are directly subjected to steam action are known commercially as "blades" or "buckets."

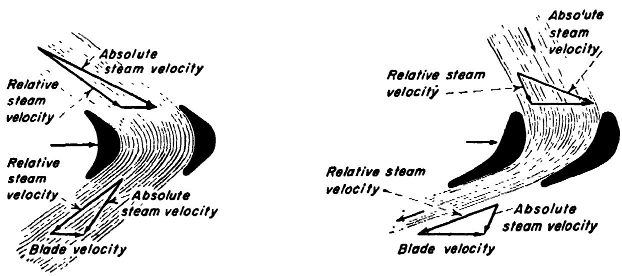


FIG. 2-10. Steam action in impulse and reaction blading.

Figure 2-10 illustrates the basic action of steam on the average impulse and reaction type of blade. The moving impulse blades form passageways through which high-velocity steam passes and gives up the major part of its kinetic energy to the blades, as will be seen by the decrease in the absolute velocity of the steam leaving the blades. This action is generally studied in terms of the velocity diagrams shown. Reaction blades, because of their shape, allow the steam to expand while passing through them. Both the kinetic energy that the steam had at entrance and the work done by the steam while expanding are in great part delivered to the blades and cause them to move. All blades must be carefully made to have accurate cross section and entrance and leaving angles. They must also be smooth to reduce friction losses as the steam passes over them. The materials used in their construction must be selected for their strength and resistance to corrosion and erosion. Some of those commonly used are chrome-alloy steel, nickel steel, cupro-nickel, stainless steel, and bronze.

In cross section, impulse blades are more or less crescent-shaped and may not always be symmetrical, as in Fig. 2-10. Modern reaction blades have a blunt-nosed teardrop shape. Blades or buckets may be forged and milled, milled from solid stock, or drawn. Precision casting is contemplated for small turbine blades. Blades are made in various lengths to accommodate the steam volumes, as expansion continues through the turbine. Short blades are generally uniform in width, while longer ones show some taper. Long blades in the final l-p stages may be "warped," or of nonuniform section, as in Fig. 2-11. This permits smooth entry of the steam to the blades. For long blades the speed or velocity of the tip is much greater than that of the root. Hence, assuming a constant steam velocity over the length of the blade, as seen in Fig. 2-10, the blade angle must change from root to tip. Such changing contours are shown in the longest blade of Fig. 2-11. The leading edges of last stage blades are often constructed with stellite erosion shields as shown in Fig. 2-11. The stellite, shaped to fit the blade is silver-soldered to it. The shield resists the erosion effect of water droplets in the wet steam flowing through the last stages at the end of the steam expansion in the turbine. The shields are renewable in the field. Figure 2-11 also indicates the use of lashing wire in the large blade for stiffening. In this case the wire cross section is streamlined to reduce the turbulence in the steam-flow to a minimum.

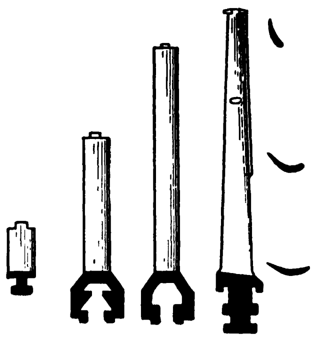
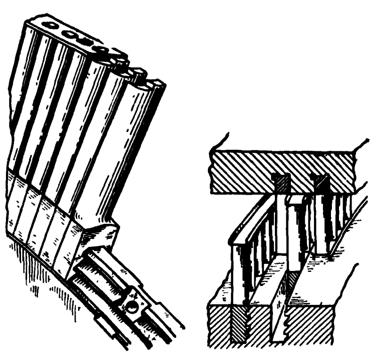


FIG. 2-11. Blades of various sizes and contours.

Moving impulse blades are mounted on individual disks. Reaction blades usually are mounted directly on the rotor. A typical arrangement for such blades is shown in Fig. 2-12. A wide variety of blade-fastening methods has been developed to meet varying conditions. Figure 2-13 illustrates inside and outside dovetails and T-slot arrangements, with and without calking strips. Where centrifugal stresses and temperatures are low, dovetailed root fastenings provide ample strength. Blades subjected to high stresses are commonly secured by T-root fastenings of either a single or double T-root. A buttress or side-entry type of root, shown in Fig. 2-13, is often used for the longest l-p blading and permits



(a) Impulse (b) Reaction

FIG. 2-12. Typical methods of blade assemblies.

longer blades and a greater last row annulus for steam flow. This permits the design of large single-cylinder condensing machines. Blades are commonly spaced by the design and proper thickness of the root sections; in some cases, spacers are used. Blades are frequently secured

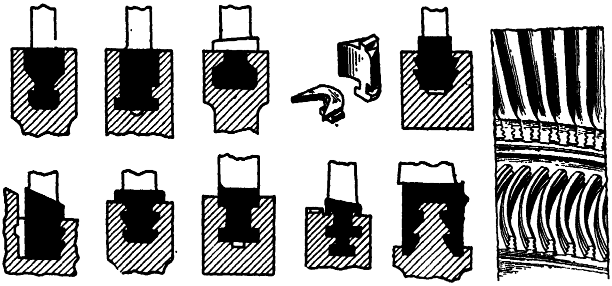


FIG. 2-13. Blade-fastening arrangements of modern turbines.

by being held against a shoulder in the groove by half-round sections, calked in place at the bottom or sides, as shown in Fig. 2-13. The blades or buckets are usually assembled on the wheels or rotors by special blading machines in the turbine builders' shops.

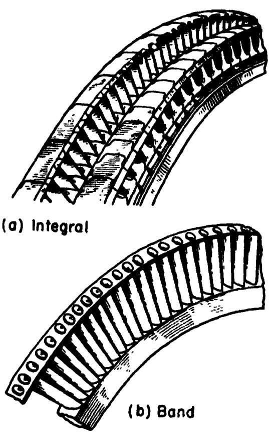


FIG. 2-14. Shrouding.

Shrouding. The tips of turbine blades are often connected by some form of shrouding. This shrouding, as in Fig. 2-14, serves to prevent steam from spilling over the end of the blades as it is discharging from the nozzles and flows over the blades. Shrouding also serves to stiffen the blades or buckets and so prevents vibration of the blades, particularly where steam does not flow over the entire annulus of the blades on a wheel, as in the h-p stages of an impulse turbine. Shrouding is made of chrome-iron alloy, stainless steel, and other materials. In many designs the shrouding is a band slotted to take a projection of the blade tip. This band is then riveted or welded in place. Shorter blades often have integral shrouding formed by the tip of each. An integral, welded construction consisting of a short strip over three or four blades is also used. In some reaction designs, as in Fig. 2-16, the edges of the shroud ring serve also as sealing strips between turbine stages.

Blade Shielding and Sealing. Shielding of blades is indicated in Fig. 2-15. It is used in the first stage where the nozzles cover only a fraction of the circumference of the turbine wheel. As indicated, they are strips and sections between which the blades pass when not in the path of the steam jet. Their effect is to reduce turbulence in these areas and so reduce turbine windage losses. The term "shielding" may also refer to the use of stellite strips on the leading edges of last-stage blades, as discussed in connection with Fig. 2-11.

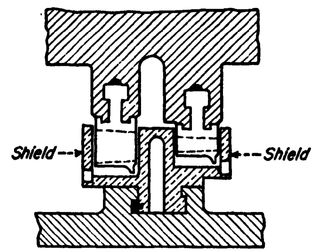


FIG. 2-15. Blade shielding.

Sealing strips may be used on impulse turbines, as in Fig. 2-16*a*, to help direct steam flow in h-p stages and prevent leakage across the stage. Blade sealing is used principally in reaction stages to retard leakage across the blade tips, as there is a pressure difference across this type of blade. Various arrangements are shown in Fig. 2-16*b*

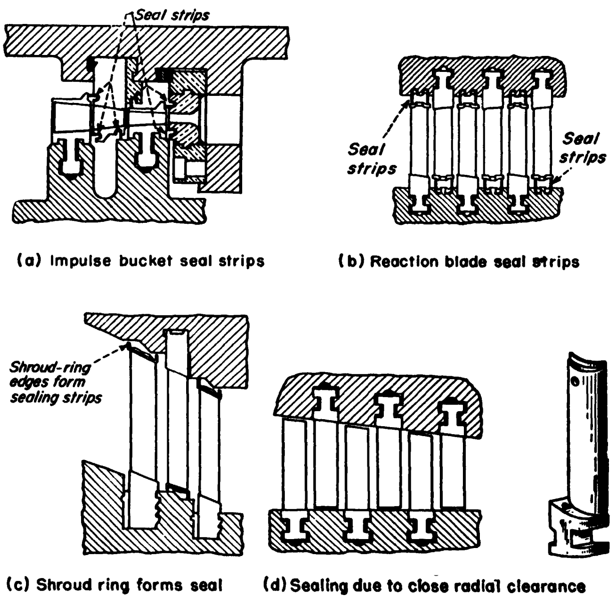


FIG. 2-16. Sealing arrangements on turbine blades.

and *c*. Close radial clearance, as in Fig. 2-16*d*, is also used to retard leakage. The thin edges and sealing strips permit close running clearances with little hazard from accidental contact. In effect, the seals allow close running clearances with relatively large clearances between heavy parts. Seal strips are often of chrome-iron alloy and may be approximately 0.01 in. thick at the sealing edge and 0.03 in. at the base. Where this design is feasible, the strips are rolled to shape and held in grooves by soft steel calking strips.

Seals and Glands. Steam leakage tends to occur in a turbine between rotating and stationary parts. Under certain conditions, air infiltration may also take place. Both effects are detrimental to turbine performance. Figure 2-17 shows the points in a turbine where leakage occurs in a unit of the impulse type. At the h-p end, steam leakage, usually, is outward. During starting conditions, however, and for very low inlet pressure in condensing turbines, the first-stage pressure may be below atmospheric. In this case, there is possible air leakage inward, to mix with the steam. At the l-p end of condensing turbines, air normally leaks into the unit. In noncondensing back-pressure turbines, steam usually leaks outward. Since there is a lower pressure in each successive stage of an impulse turbine, steam leaks from one stage to the following, where the shaft passes through the diaphragm. Similar leakage conditions exist for reaction turbines where the shaft or drum passes through the casing. In place of diaphragm leakage, however, there is a tendency toward leakage across the ends of each set of fixed or moving blades, because the blades are in effect nozzles and a pressure difference exists across them. Any steam leakage out of the turbine, as at the shaft and casing location, represents a complete loss and increases the steam rate of the unit. Steam leakage, such as at diaphragm openings and across reaction blade tips, is not a complete loss since it can be used in the next lower pressure stage nozzles. However, even in such a leakage, the steam pressure drop reduces the effectiveness of this steam and tends to increase steam rates, greater amounts of a steam being required for the same power output. These effects are known technically as "throttling losses." Air infiltration manifests itself in the condenser of the power plant, where an extra load is thrown on the air-removal equipment and the air raises the back pressure of the turbine. This again means an increase in steam flow to produce a given amount of power. Steam leakage and air infiltration must be held to a minimum; therefore, various forms of seals and glands have been designed and are installed in appropriate sections of the turbine.

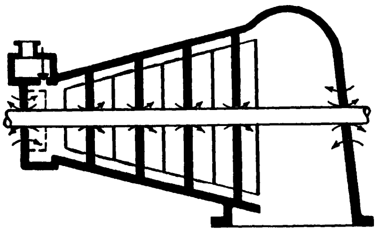


FIG. 2-17. Leakage points in an impulse turbine.

There are four principal types of seals or glands used in turbine practice: (1) stuffing boxes, (2) carbon-ring packing, (3) labyrinth seals, and (4) water glands.

The typical stuffing box is fitted with soft metallic packing rings or a simple bronze bushing. This arrangement is used in some small machines for shaft sealing at low pressure and, particularly, when exhaust is at atmospheric pressure. Most small turbines and larger units employ carbon rings, labyrinths, water glands, or a combination of them. Typical seals are shown in Figs. 2-18 to 2-20.

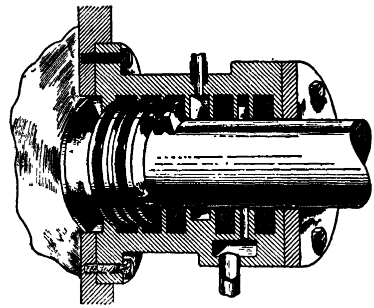


Fig. 2-18. Carbon packing.

Figure 2-18 shows a carbon packing or seal. In this construction, a series of carbon rings, set with close shaft clearance, produce an effective seal. The ring consists of several circular segments, generally rectangular, in cross section. These are held together and hug the shaft under the action of a garter spring or some other design of flat spring. A stop piece in the housing prevents rotation of the ring. Carbon rings are used alone or in combination with other types of seals. The packing cases at the ends of the turbine are generally split horizontally and can be opened readily to allow inspection or cleaning of the carbon rings.

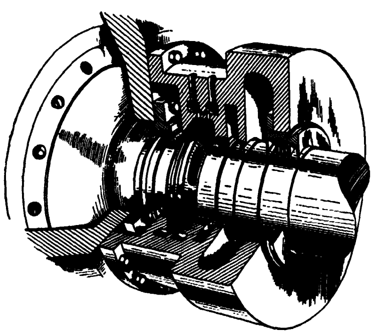


Fig. 2-19. Labyrinth seals (water gland also shown).

Labyrinth seals, as in Fig. 2-19, consist essentially of a number of thin circular strips or serrations fastened to the casing, or a member supported from the casing, and positioned so that the clearance between the shaft and the edges of the strips is small. These seals are generally thin chrome-alloy strips and are calked in place. An accidental rub usually causes little damage, as the thin strips readily dissipate the friction heat. The strips do not tear when contact occurs. The usual result is that the clearance is slightly increased. Worn strips can be replaced easily, during a regular shutdown. Labyrinth strips can be placed to act in either a radial or an axial direction. They are widely used in all types of leakage conditions in turbines, as the resistance offered by this series of strips to steam flow is enough to hold leakage to a minimum. Labyrinths may be used alone and, very frequently, are used in combination with water glands. Figure 2-19 shows labyrinth seals in such a combination.

A water gland, as in Fig. 2-20, is basically a centrifugal pump runner rotating with the turbine shaft and confined in a housing attached to the casing. Condensate should be used as the sealing medium, so that discharged gland water can be returned to the system. The use of condensate also avoids scale deposits in the glands. The runner holds the water in a ring at its periphery and forms a positive seal. In most cases water glands are used in combination with some other form of seal, generally of the labyrinth type, as indicated in Fig. 2-20. This is done because the water gland is effective only at high speeds (*viz.*, 2,000 rpm for a 3,600-rpm unit). The additional seal also reduces the casing-side steam pressure on the water gland, when used at the h-p end.

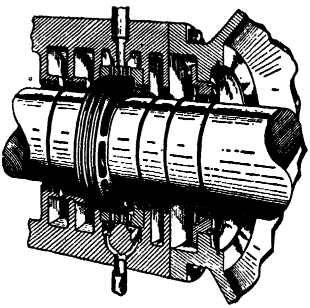


FIG. 2-20. Water gland (labyrinth seals also shown).

The operation of carbon rings or labyrinth seals is improved by the addition of steam sealing and gland leakoffs. Figures 2-21 and 2-22 illustrate the principles of these arrangements diagrammatically. Steam

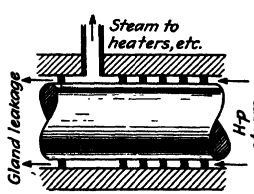


FIG. 2-21. Gland steam leakoff.

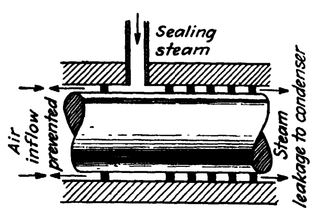
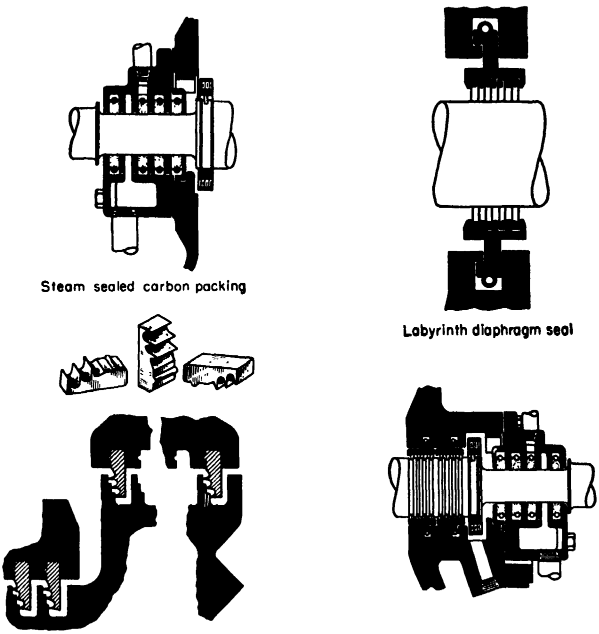


FIG. 2-22. Steam-sealed gland.

that leaks past the gland from a region of high pressure is known as "leakoff" steam and is commonly led to a feed-water heater. Its heat may be recovered elsewhere in the station cycle, or it may be led back to a turbine stage at lower pressure. During starting, steam sealing positively checks air flow into the condenser at the exhaust-end gland and into the turbine at the h-p end. Steam enters an exhaust-end gland, as in Fig. 2-22. Some steam leaks to the atmosphere and prevents air inflow; the remainder leaks into the turbine and then to the condenser.

Applications of these various types of glands or seals vary considerably in actual practice. Labyrinth packings, carbon rings, or combinations of them (Fig. 2-23) are commonly used as diaphragm seals, and usually they are used with steam sealing at shaft ends. In reaction turbines with balance pistons, the pistons are sealed with the labyrinth type of seal.



Steam sealed carbon packing

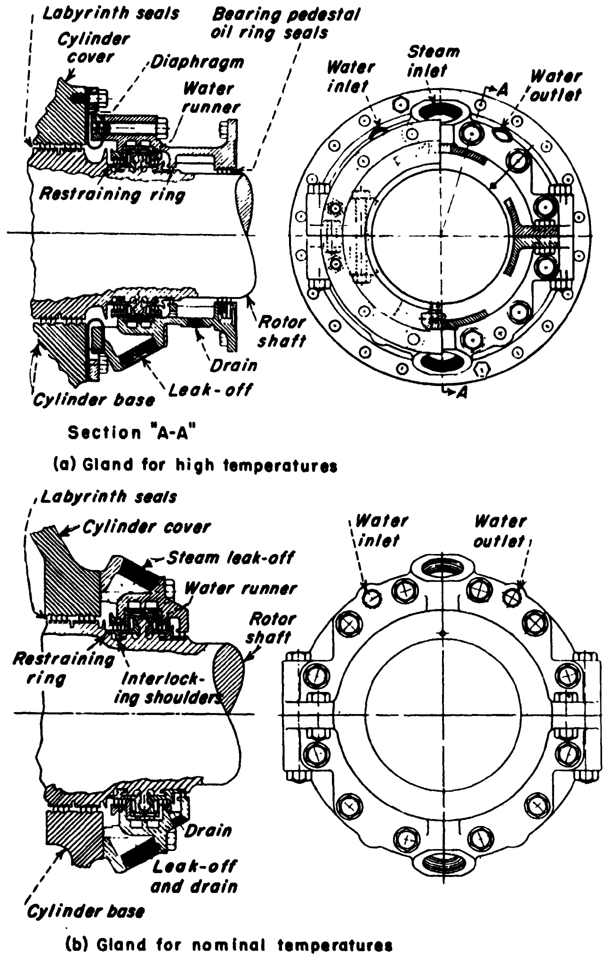
Labyrinth diaphragm seal

Dummy packing on reaction-turbine h-p and l-p balance piston

Labyrinth with leakoff plus steam-sealed carbon-ring gland

FIG. 2-23. Typical sealing arrangements.

Figure 2-24 shows a typical application of water glands to the h-p and l-p or corresponding temperature sections of a turbine. For high-temperature operation, the gland case is supported on the bearing housing and is connected to the hot casing wall by a flexible diaphragm. This reduces any distortion that may be caused by the temperature difference between the casing wall and the gland housing. It should be noted that labyrinth seals are used between the water gland and the inside of the turbine to reduce the pressure difference across the water gland. Radial seals are used on the water-gland section to reduce water leakage to a minimum. Drains remove whatever water leakage does occur. The

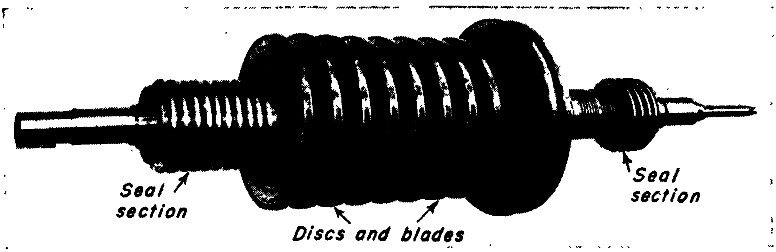


(*Courtesy of Westinghouse Electric Corp.*)

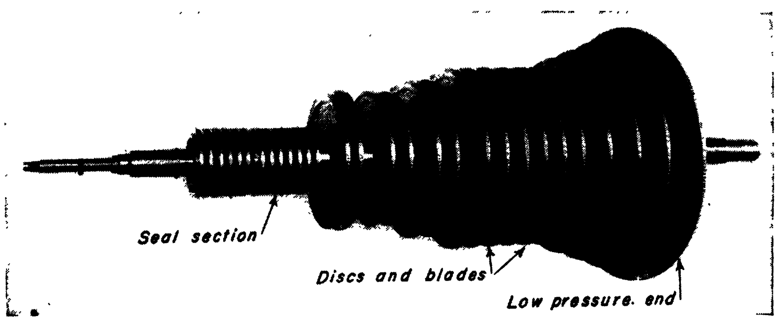
FIG. 2-24. Typical water gland applications.

gland has two openings to which the sealing-water inlet and discharge lines are connected.

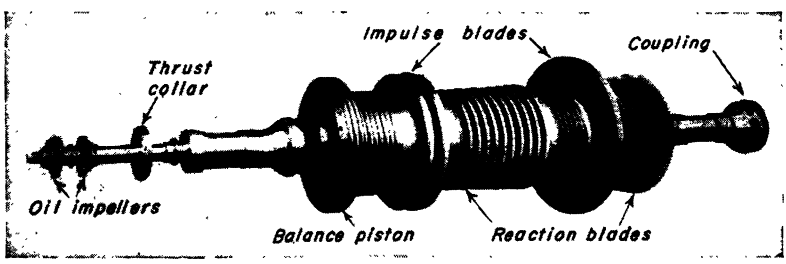
Rotors. The general appearance of turbine rotors when removed from the casing is shown in Figs. 2-25 to 2-28. Both condensing and noncondensing units of straight and extraction types of turbines are indicated. Impulse rotors for 1,800-rpm units generally consist of a shaft



(Courtesy of Worthington Pump and Machinery Corp.)
FIG. 2-25. Noncondensing impulse turbine rotor.



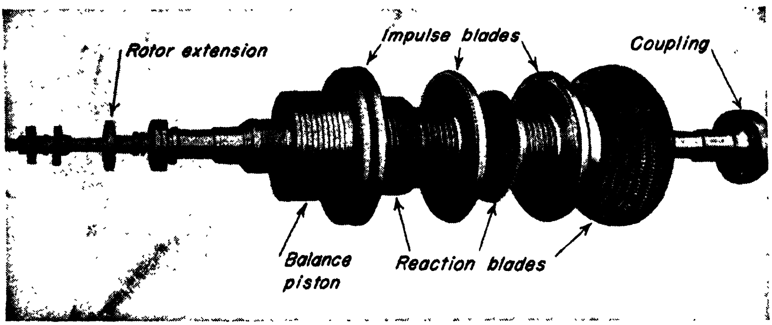
(Courtesy of Worthington Pump and Machinery Corp.)
FIG. 2-26. Condensing impulse turbine rotor.



(Courtesy of Westinghouse Electric Corp.)
FIG. 2-27. Noncondensing impulse-reaction single-extraction turbine rotor.

and wheels. The shaft is made from selected billets which are forged, heat-treated, and machined to proper sizes. After machining, the shaft is accurately ground to size. It is finished to close tolerances to fit accurately the turbine wheels. The steel disks are assembled on the shaft with heavy shrink fits and are secured by keys in the low-temperature section. In the high-temperature section, the wheels tend to leave the shaft under the influence of centrifugal force and sudden temperature increases.

To keep the wheel centered and still drive the shaft, a pin bushing is keyed to the shaft. Radial pins on the bushing drive the wheel through the hub. Rotors for 3,600-rpm units are generally fabricated from a solid forging. This overcomes some difficulties of separate wheels and makes a short rigid rotor that will operate below the critical speed. All solid and wheel types of rotors must be free of metallurgical defects and are checked by the magnaflux method after machining.



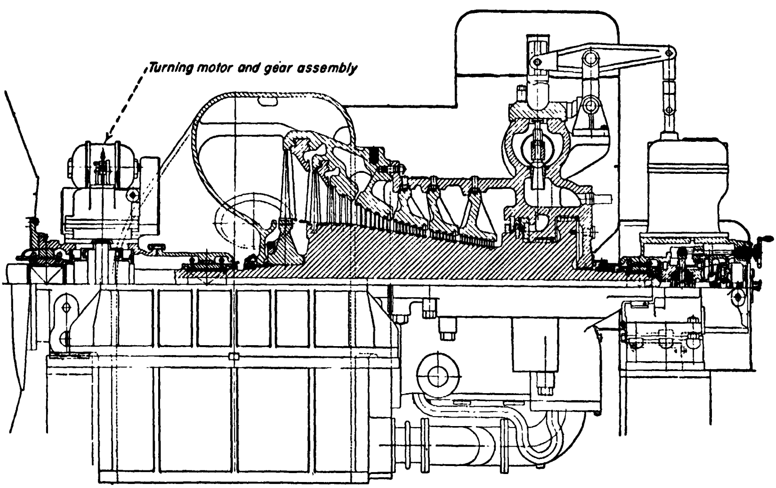
(*Courtesy of Westinghouse Electric Corp.*)

FIG. 2-28. Condensing impulse-reaction double-extraction turbine rotor.

Rotors for reaction or impulse-reaction turbines are commonly one-piece solid forgings of steel, or hollow drum sections, depending upon the turbine capacity. Impulse stages may be placed upon disks that are an integral part of the rotor, or on separate disks on the rotor shaft. Low-pressure reaction blades may also be simultaneously mounted on separate disks fastened to the l-p rotor body. In the case of the solid rotor, a hole for internal inspection is bored through its axis. A shaft extension, as shown, is bolted to the front end of the solid h-p turbine rotor. This extension carries the thrust-bearing collars, the oil impellers, and the overspeed trip. Balance pistons are part of the h-p turbine rotor with reaction blading. These balance pistons are rotating sections which counterbalance the longitudinal thrust of the reaction rotor. Higher steam pressures on the moving blade entering edges develop the rotor thrust. The balancing forces are brought about by allowing steam of suitable pressures to act on the balance-piston faces, and so to create a force in an axial direction. Steam from turbine stages or from the condenser may be piped to the balance-piston faces to obtain the required pressures.

Rotors are commonly made of carbon and nickel steel, for temperatures up to about 750 F, the latter being used for designs where high stresses are encountered. For temperatures above 750 F and for special cases of high stress at lower temperatures, a nickel-chromemoly steel is used. After the rotors are completed and the blades are in place, they are dynamically balanced and then tested in the turbine at full speed and usually at 20 per cent overspeed. Couplings are part of the rotor and may be either pressed and keyed to the rotor or machined as an integral part of it.

Turning Gears. The turning gear, as in Fig. 2-29, consists generally of a small-capacity motor (5 to 20 hp), a speed reducer, and a driving gear



(Courtesy of General Electric Co.)

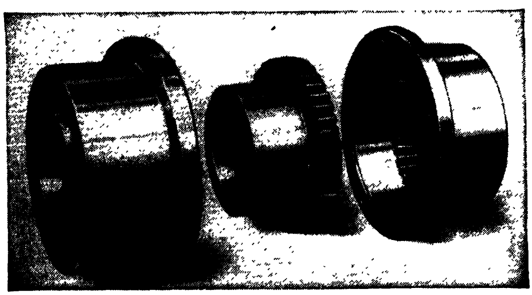
FIG. 2-29. Turning gear.

assembly. It is commonly mounted on the bearing or coupling cover at the l-p end of the turbine and is used for turning the rotating parts at slow speeds (2 to 20 rpm). It is used, during the cooling-off period, after the turbine has been shut down, to prevent distortion in the turbine elements and to avoid localized cooling of the turbine shafts. It is used also in the warming-up period, before starting the turbine, so that the rotating parts will be uniformly heated. When used on a turbine with reduction gears, it permits slow rotation, so that the gears may be inspected. The turning gear is commonly equipped with a shutoff, in case turbine lubrication should fail; a limit switch that will shut off the motor when the turning gear is disengaged; a safeguard against engaging the motor when the main unit is operating; and a disengagement device in case the unit speeds up. Turning gears are supplied on large-capacity units and are generally optional on small units.

Following is a list of the speeds at which the various turbine parts start to function during turbine starts. The values are for two turbine speeds, and the speeds shown are approximate.

	1,800-rpm unit, rpm	3,600-rpm unit, rpm
Turning gear	1½–3	2–5
Slow roll while heating with steam	200	200
Water glands	1,000	2,000
Main oil pumps	1,500	2,700
Governor	1,700	3,400
Overspeed	2,000	4,000

Couplings. In most turbine applications, the connection between the turbine and the driven equipment is made by a coupling. Couplings are used also to connect the h-p and l-p spindles in some tandem types of turbines. Couplings may be of simple, rigid construction, bolted or otherwise fastened together. Most often, however, some form of flexible

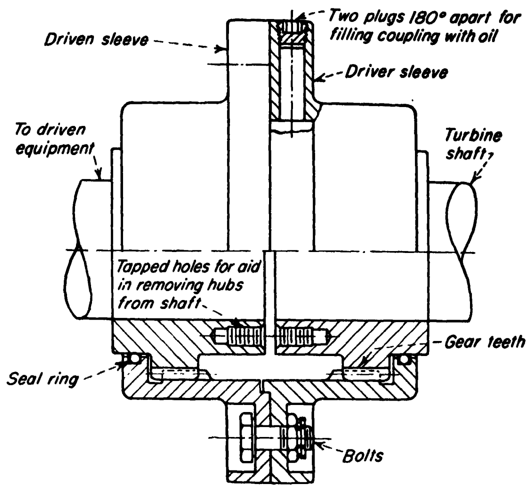


(Courtesy of De Laval Steam Turbine Co.)

FIG. 2-30. Elements of simple gear-type flexible coupling.

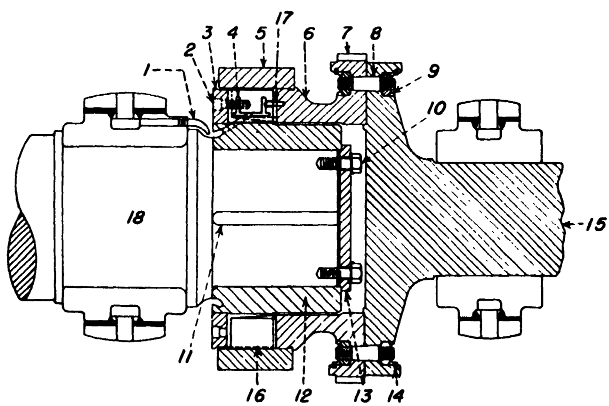
coupling is employed. The elements of a simple, gear-type, flexible coupling are shown in Fig. 2-30. The smaller sections have gear teeth around the outside and are keyed to the shafts. These teeth, in turn, mate with gear teeth cut on the inside of the sleeve, both halves of which are then bolted together. As shown in Fig. 2-31, the driven shaft transmits its power through the gear teeth to the sleeve, and this in turn is bolted to and rotates the sleeve of the driven half.

The coupling is built of carbon steel. A small amount of misalignment



(*Courtesy of De Laval Steam Turbine Co.*)

FIG. 2-31. Construction of simple gear-type flexible coupling.



1. Oil supply pipe
2. Cap screw
3. Coupling ring
4. Oil grooves
5. Shrink ring
6. Coupling sleeve
7. Coupling gear
8. Taper bolts
9. Special nut
10. Tap bolt
11. Coupling key
12. Coupling end
13. Coupling plate
14. Nut lock half ring
15. Turbine spindle
16. Wearing surface
17. Dowel pin
18. Generator rotor

(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 2-32. Single-sleeve flexible coupling.

can be tolerated, but the degree of misalignment results in a proportionate amount of wear. Even with this type of coupling, alignment, in setting up the unit, must be as perfect as possible. Provision is made to keep the coupling full of oil, retained by a seal ring.

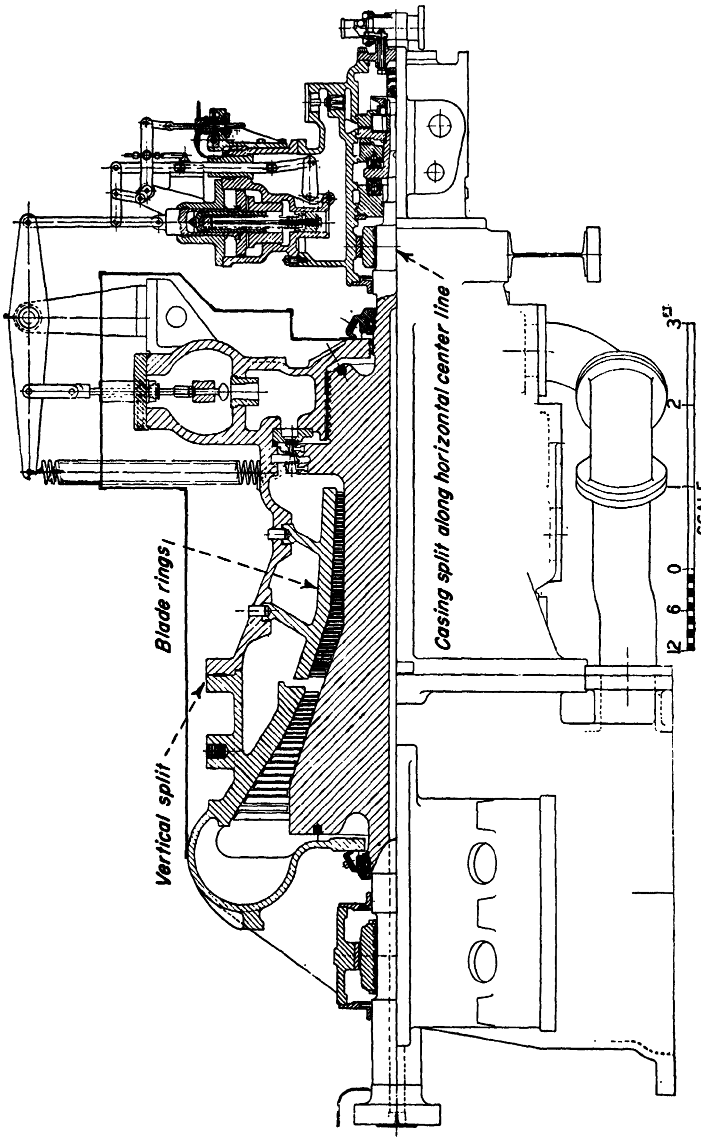
A coupling found on a larger turbine is shown in Fig. 2-32. It consists of a coupling end keyed to the driven shaft. A sleeve is bolted to the turbine shaft. The coupling ends are provided with radial claws and the coupling sleeve with axial claws. The radial claws have inserts made of hard bronze. When connected, the axial claws are located between the radial claws of the coupling ends. A lubricating system allows lube oil to pass between the claw faces of the coupling. To prevent excessive stresses from the centrifugal forces in the axial claws, steel rings are shrunk over the ends of the claws. As shown in Fig. 2-32, the flange of the coupling sleeve has spur gear teeth cut in it. This gear engages with the turning gear for rotation of the unit at low speed during shutdown.

Cylinders, Casings, or Shells. The development of the turbine has depended among other things upon developments in metallurgy. Over the years the major parts of a turbine have been made of different materials. In the beginning of the turbine industry, cast iron was used, allowing about 175 psig and 500 F as an upper temperature. Higher temperature causes cast iron to grow and causes parts to seize. Today, cast iron for turbines of any size is limited to 450 F or less. Steel castings and forgings came into use about 1910. Pressures of 1,250 psig, or higher, and temperatures to about 825 F were made possible with carbon-steel castings. Casings for turbines made from carbon steel are either cast or fabricated from welded steel plates. Carbon-molybdenum alloys were available about 1932. Later, chrome-molybdenum steels for temperature through 950 F were developed. Castings of 2¼ per cent chrome and 1 per cent molybdenum are used for temperatures of 1000 F. Alloys with 18 per cent chrome and 8 per cent nickel are used for gas turbines operated at temperatures of about 1050 F. Table 2-1 summarizes this information. ASTM numbers are indicated where possible; in some of the more recent materials, they are not yet available. Cylinders for operation at temperatures above 825 F are hard to weld and are made as castings. Repair welding is used, however, to correct blowholes or slag inclusions. To accomplish this the casting must be preheated and thoroughly annealed. Cast iron is used for the l-p end of high-temperature turbines operating at 450 F or less. Through modern turbines of high pressure and temperature (*viz.*, 1,200 psig and 900 F), the steam passes in a fraction of a second. One end of the shell will glow red, and the other, a few feet away, may be at a tepid temperature.

TABLE 2-1. STEAM-TURBINE MATERIALS TO BE USED FOR VARIOUS TEMPERATURE RANGES*

Exposed temperature, deg F	Kind of steel	Castings							Forging	Straight pipe, flanged pipe, and pipe bends
		Throttle valve	Steam chest, separate	Steam chest, integral	Nozzle chamber	Inner casing	Outer casing	Blade rings, separate		
451–800	Carbon	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-105 Grade 1	ASTM A-106 Grade A
801–825	Carbon		ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-216-WCb	ASTM A-126-WCb	ASTM A-216-WCb	ASTM A-216-WCb		
	½ % chrome–½ % moly	ASTM A-217WC-3								
	1 % chrome–½ % moly								ASTM A-182-F7†	ASTM A-213-T12
826–900	½ % chrome–½ % moly	ASTM A-217-WC3	ASTM A-217-WC3	ASTM A-217-WC3	ASTM A-217-WC3	ASTM A-217-WC3	ASTM A-217-WC3	ASTM A-217-WC3		
	1 % chrome–½ % moly								ASTM A-182-F7†	ASTM A-213-T12
901–950	1 % chrome–½ % moly	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-217-WC6	ASTM A-182-F7†	ASTM A-213-T12
951–1000	2¼ % chrome–1 % moly				‡	‡		‡	‡	ASTM A-213-T22
1001–1050	18 % chrome–8 % nickel				‡				ASTM A-182-F8c	ASTM A-158 Grade P8d

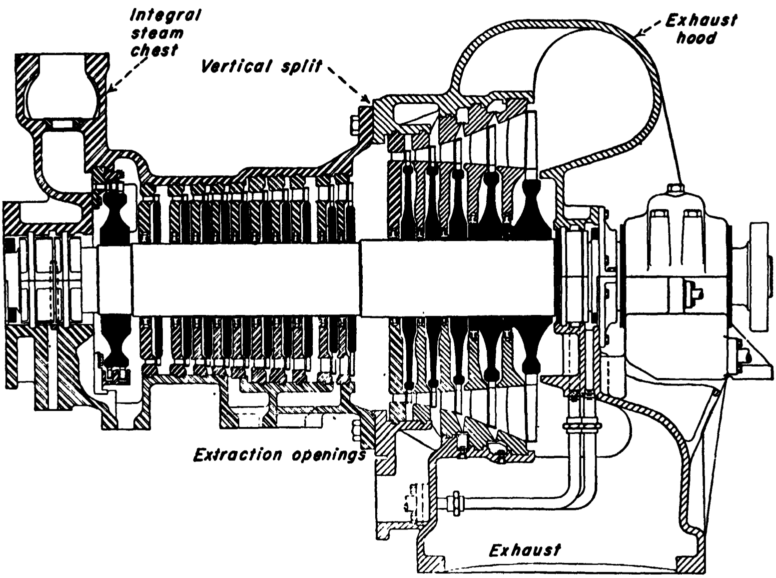
* *Courtesy of Westinghouse Electric Corp.*
† Except ½ per cent moly.
‡ No ASTM number yet.



(Courtesy of Westinghouse Electric Corp.)

FIG. 2-33. Casing construction of an impulse-reaction turbine.

Figures 2-33 and 2-34 show the general features of a single-shell turbine of the condensing type with a fairly standard construction, consisting of a h-p and a l-p section and having the steam chest cast integrally with the h-p section. The casings are split at the horizontal center line to allow lifting the top half for repair, fabrication, and inspection. The halves are fairly symmetrical and relatively uniform in thickness. As shown, the

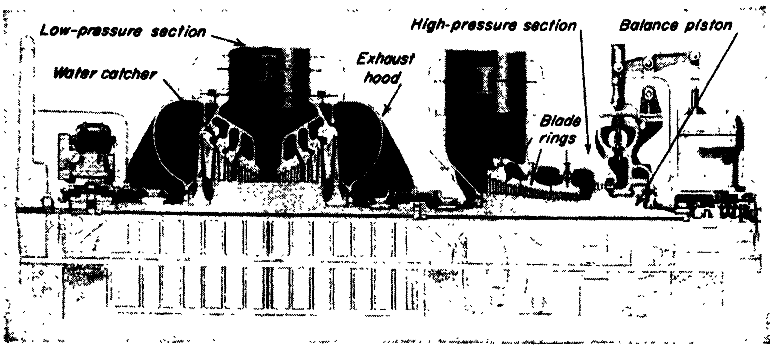


(*Courtesy of General Electric Co.*)

FIG. 2-34. Casing construction of an impulse condensing turbine.

shells also are split vertically; and depending upon steam conditions, the h-p castings are of steel while the exhaust portions are made of cast iron. Turbine halves are bolted together at a gasketless metal-to-metal joint. The joint flanges at these sections are hand-scraped to give full metal-to-metal contact. Steam tightness at the joints is obtained by a coordinated flange and bolt design.

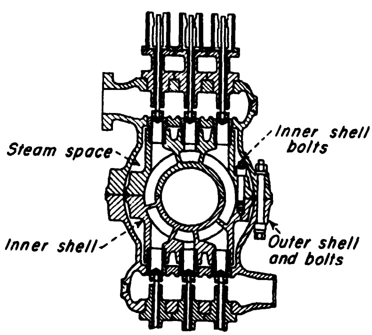
The casing, or shell construction, of an h-p and compound type of turbine is indicated in Fig. 2-35. As in single-cylinder units, the casings are split and bolted together, and metals suitable to steam conditions are used. A double-shell construction is commonly used on h-p and high-temperature casings, as shown in Fig. 2-36. It consists of an inner shell with its own bolting, as in Fig. 2-37, and an outer shell surrounding it. The in-between space is connected to a lower stage of the turbine at a pressure about halfway between inlet-steam pressure and the atmosphere. Neither shell is, therefore, as thick as an equivalent single-shell machine and the inner shell is steam-heated on both sides. The practical



(*Courtesy of Westinghouse Electric Corp.*)
FIG. 2-35. Tandem compound turbine cylinder.

result is a decrease in distortion when heated, and the unit may be rapidly brought up to operating conditions.

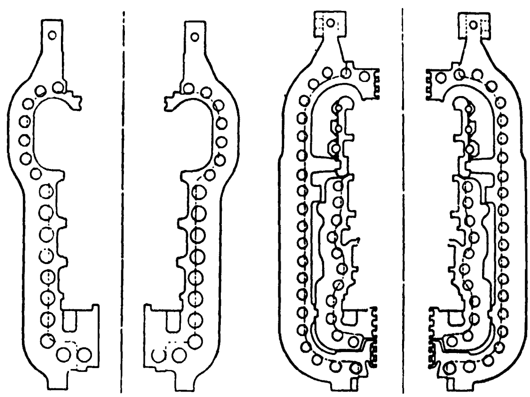
The flanges holding the halves of the casing together, and their bolting arrangements of studs, nuts, and bolts, are important parts of turbine design. No gaskets are used and the bolts are put in as close to the inner diameter as feasible—about 1 in. as a minimum. Turbine studs and bolts are commonly threaded by a milling operation to obtain smooth threads. Other refinements to aid in easy removal are to copper-plate the nut threads and to use liberal clearances between the nut and the stud. Studs over 2 in. in diameter are made hollow so as to allow heating for ease in setting up and loosening. Threads on nuts or in castings for studs are cut with a very slight taper which tends to give a more uniform loading when tight than results with parallel threads. When bolts are put in place, the better practice is to expand the bolts by heating and then to tighten the nuts by light wrenching. A stressed bolt will tend gradually to relax its hold on the flange, because of the creep effect in



(*Courtesy of General Electric Co.*)
FIG. 2-36. Vertical turbine cross section illustrating double shell.

stressed metals at high temperatures. Metals with high relaxation stress characteristics are used.

In manufacturing, the rough castings are laid out and roughly



(*Courtesy of General Electric Co.*)

FIG. 2-37. Comparable bolting flange of single- and double-shell turbines.

machined. After this, the h-p shell is X-rayed, and magnetic powder is used for checking surface flaws. Defects are chipped out, welded, and annealed. The castings are laid out for final machining. Flanges are hand-scraped and bolted. A hydrostatic test at $1\frac{1}{2}$ times normal steam pressure is used. After this, the shell is ready for the various assemblies necessary to complete the turbine unit.

Bearings. The main bearings for small turbines are usually of the horizontally split, babbitt-lined type with bronze or cast-iron shells. Figure 2-38 shows a typical design with a slot and oil ring for bearing lubrication. The oil chamber at the base is designed with passages for cooling water. Bearings of this type may be fitted with more than one oil ring. The shaft, as shown, passes first through the turbine casing and some sealing gland before the bearing is reached. Sealing rings of some form are used also on the bearing, to retain the oil. Small turbines

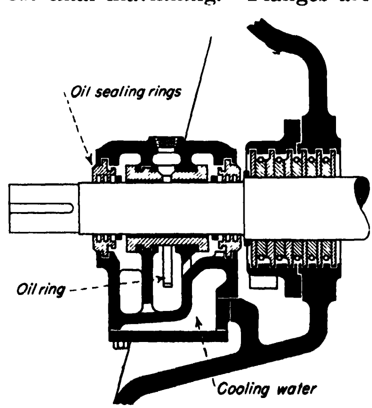


FIG. 2-38. Journal bearing for a small turbine.

are fitted often with single- or double-row ball bearings, as in Fig. 2-39. These are designed either as straight journal bearings, or with features to take up some thrust. Oil-retaining seal rings are used also with this type of bearing.

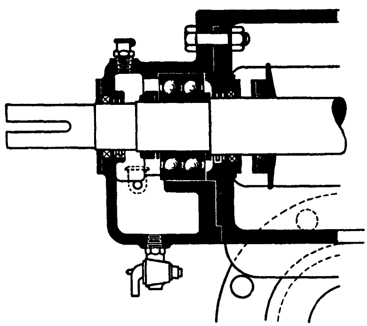


FIG. 2-39. Ball bearing for a small turbine.

Figure 2-40 shows a typical main journal bearing for a large turbine. It has a cast shell of steel with a babbitt lining. Four steel blocks or keys are machined to fit a spherically bored seat in the pedestal, permitting self-alignment. Liners between the key and the bearing shell allow the bearing to be moved and center the rotor in the cylinder. A dowel in the shell fits a notch in the pedestal and prevents rotation of the bearing. Oil under pressure comes through one of the blocks and is distributed by longitudinal grooves along about 85 per cent of the bearing length. To assist oil distribution and formation of a wedge film of oil,

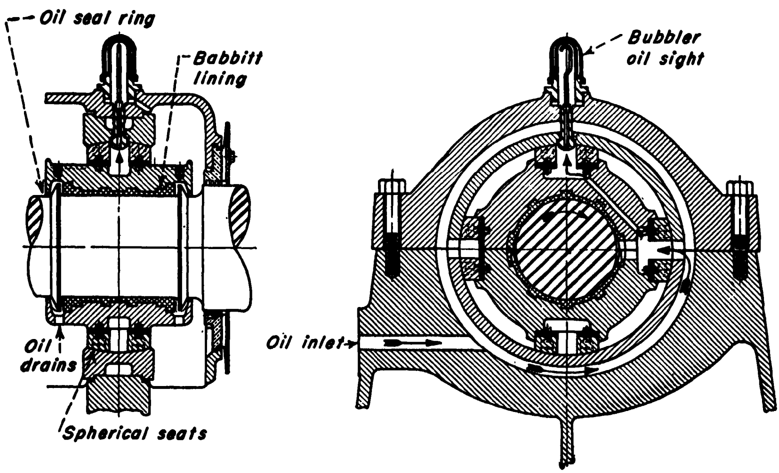


FIG. 2-40. Typical force-feed journal bearing.

the babbitt in the upper half and just below the horizontal joint is relieved to give large radial clearance between the shaft and the babbitt. Oil drains into the pedestal through holes in the bottom of the end annular grooves. The quantity of oil circulated is determined by the cooling needs rather than by the lubrication requirements. A bubbler oil sight gives visual indication of the existence of lube-oil pressure and may have a thermometer incorporated to indicate oil temperatures. Radial oil-sealing strips at the end prevent oil from being thrown along the shaft. The bearings split horizontally are accurately bored with proper clearance.

Thrust bearings of several types take care of the forces acting along the shaft axis. In the tapered-land bearing (Fig. 2-41) a hardened and ground rotating collar bears on babbitted lands. There is a wedge of oil between

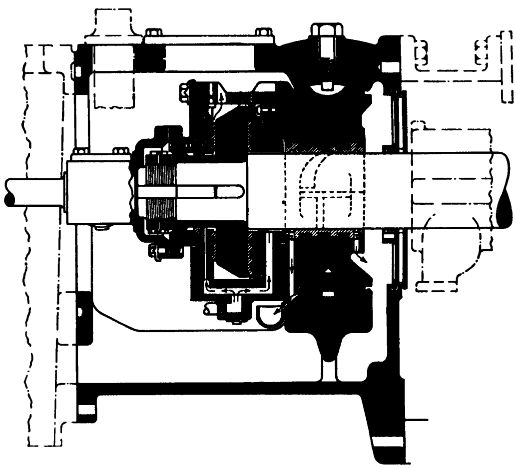


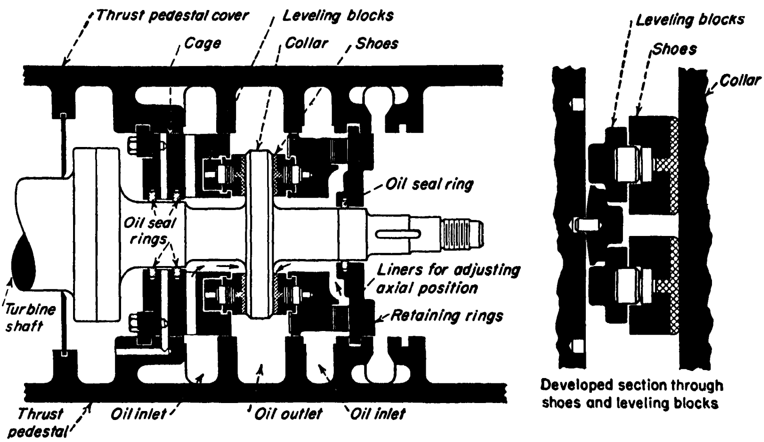
FIG. 2-41. Tapered-land thrust journal bearing.

the rotating collar and the thrust plates. Cooled oil under pressure enters radial grooves in the thrust plate, and the rotating collar carries it over the surface. The thrust collar commonly mounts at the h-p end of the turbine and serves for axial adjustment of the rotor. Squealer rings, which rub before a dangerous degree of wear occurs, give warning of failure. The bearing splits horizontally for ease of assembly.

The Kingsbury type of thrust bearing (Fig. 2-42) has its total surface divided into a number of segments. These are free to rock and take a position set by the formation of an oil wedge between them and the rotating collar. The thrust load distributes evenly among the segments. Axial adjustment of the bearing and, therefore, of the rotor can be made. Oil under pressure circulates to all moving parts.

Expansion. A turbine, in order to operate, must maintain its various clearances. A turbine about 20 ft long, with a 900 F inlet temperature and 100 F at the outlet, will expand about $\frac{5}{8}$ in. over its length at room temperature. This axial expansion must be considered in turbine design for starting, stopping, and running operations, and must be guided and controlled so as not to cause distortions, by restrictions of any kind.

Axial *differential* expansion of the machine is usually allowed for by having a point of fixation between *rotor* and *casing*, near the hot end and by providing proper axial clearance between the stationary and rotating parts. The common fixed point between the casing and the rotor is the thrust bearing. Axial and transverse expansion of the *casing* is usually allowed for by keeping one end of the casing, usually the cold end, keyed

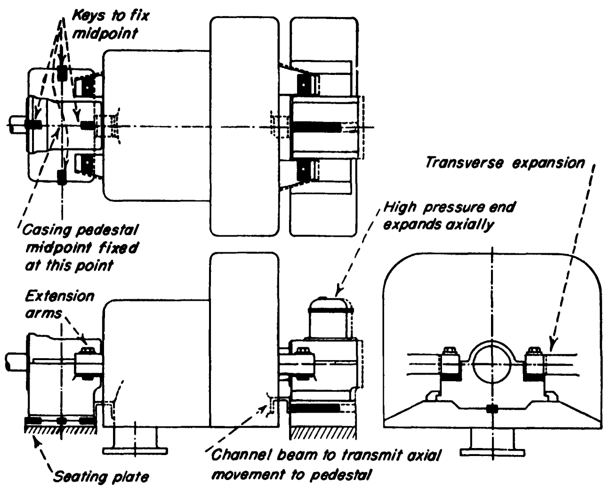


(*Courtesy of Westinghouse Electric Corp.*)

FIG. 2-42. Kingsbury type of thrust bearing.

to the foundation, as shown in Fig. 2-43. Thermal expansion takes place in the horizontal plane at the top of the bearing-pedestal seating plate. At the h-p end of the turbine, the thrust-bearing pedestal is mounted so that it is free to slide axially on its base guided by a key. Hence, when steam enters the cold machine, the casing, which is anchored at the cold end, tends to expand toward the opposite end and pushes the rotor with it. This tends to decrease the axial clearance between stationary and rotating parts, but it is compensated for since the rotor expands at the same time. The axial expansion and contraction of both casing and rotor follow each other closely. Four extension arms, as shown in Fig. 2-43, carry the casing on the pedestal at its horizontal center line. These arms are free to slide on, but are prevented from lifting off, their supports. The axial movement of the casing is transmitted to the pedestals through channel beams. These beams are rigid in the transverse direction and maintain the transverse alignment of the cylinder. In the vertical direction, these beams are flexible and permit movement of the cylinder relative to the bearing pedestals. The flexible channel beams maintain transverse alignment but permit radial expansion. In general this mounting permits a turbine to expand or contract freely, but with the movement guided to keep the turbine in alignment and to maintain clearances.

Another form of expansion in turbines is differential radial expansion between casing and rotor which, in starting, or stopping, might cause rubbing, if unusually large. Turbines are designed to overcome this by the use of separate blade rings, as in Fig. 2-35. The rings are supported



(*Courtesy of Westinghouse Electric Corp.*)

FIG. 2-43. Turbine support and provision for expansion.

by lugs and are held in alignment by pins. The blade rings are free to expand uniformly in all directions. Steam is allowed to flow around the blade rings which expand at nearly the same rate as the rotor. They also expand without distorting forces being produced on the casing. Leakage past the blade rings is prevented by interstage sealing rings.

Lagging and Insulation. Large-capacity turbines are built with a thick layer of heat-retarding insulation, commonly magnesia, over the high-temperature sections. The magnesia acts as an insulation to prevent heat loss and is securely fastened in place. The fastening is commonly made to lugs welded to the shell. Steel plates, rolled and shaped to the form required, are applied over the insulation. These protect it and give a smooth, finished contour to the turbine, varying in appearance from the contour of the casing itself.

Stop and Throttle Valves. Steam flow from the steam main to the turbine proper is generally controlled by stop valves for open-and-shut service, and a throttle valve to regulate steam flow during starting and stopping. These functions are at times combined in one valve. These valves may be manually operated; for large units they are usually hydraulically actuated. Figure 2-44 shows a stop valve hydraulically operated and designed for quick opening and closing. It is tied in with oil pressure, so that a failure of lubricating oil pressure will cause the unit to shut down. If this valve is used alone, starting is controlled by the turbine governing valves. This type of valve is also used in reheat steam lines, to and from the turbine, to close off steam flow when overspeed occurs.

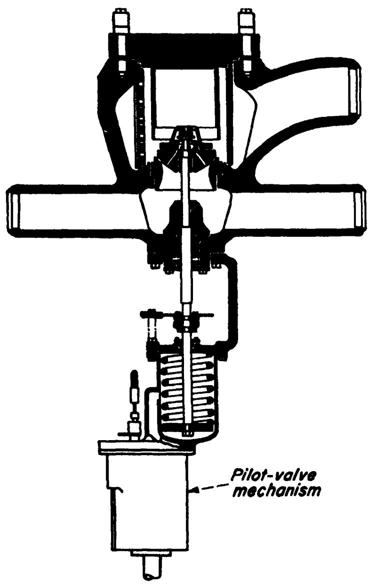
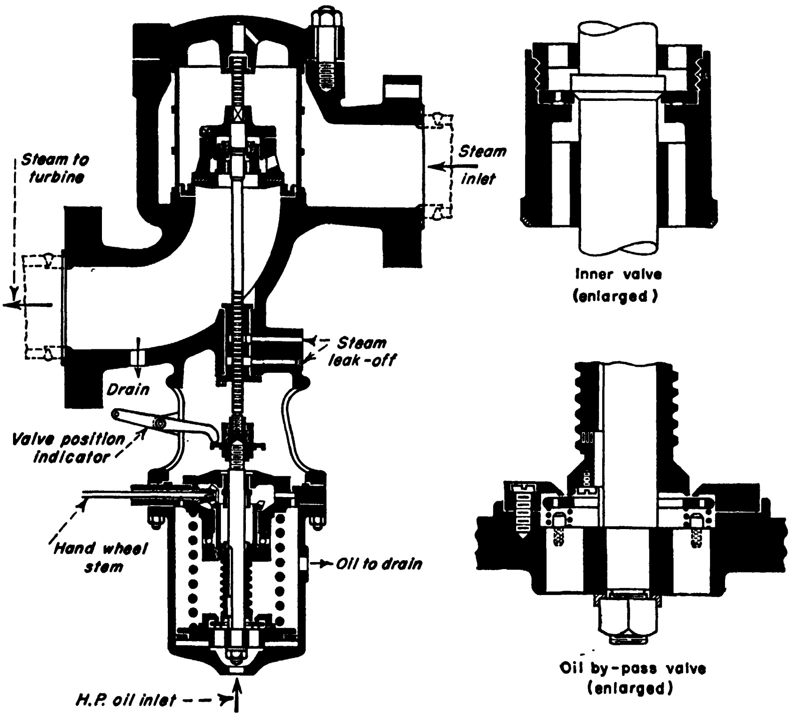


FIG. 2-44. Typical hydraulically operated stop valve.

Figure 2-45 shows a throttle valve with automatic stop features. Throttle valves may be used, sometimes, as both stop and throttle valves, but most often, in addition to stop valves. In Fig. 2-45, there are two parts: the valve in the steam line above, and the operating mechanism below. The steam valve is made of two single-seated valves, one within the other. Both steam pressure and the spring on the operating piston tend to close the steam valve. The valve is operated by turning the handwheel which controls an oil bypass valve, shown enlarged. With the by-pass closed, oil pressure under the piston raises the valve stem as fast as the by-pass valve rises. Oil is supplied to the inlet through an orifice. Hence, if the oil by-pass is raised too quickly, the main piston cannot follow. The oil beneath the piston is released to the space above, and the valve drops to the closed position. The handwheel must then be reset and the starting procedure repeated. The inner steam valve is moved against full steam pressure at the start and allows enough steam to pass to bring the turbine up to speed. At full speed, the governing valve will have closed considerably, and the pressure difference across the throttle steam valves is reduced to a small amount. Further steam flow results as the inner valve engages the outer valve, after which both valves lift. The oil piston serves also as automatic stop valve in case of oil-pressure failure. The oil cylinder is placed below to reduce fire hazard if oil leakage occurs. A strainer surrounds the steam valves. No packing is used, but leakoffs prevent escaping steam. The valve is hardened chrome steel with a spherical contact on a stellited seating ring. Depending on size, the valve may be bolted to the steam chest or mounted separately. In the latter case, it is commonly placed below the floor in front of the turbine.



(*Courtesy of Westinghouse Electric Corp.*)
FIG. 2-45. Typical automatic-stop throttle valve.

Steam Chest and Admission Valves. Steam passes from the throttle valve to the steam chest in turbines. The steam chest may be an integral part of the turbine casing or may be bolted to it. In some large turbines, the entire steam chest assembly is separated from the casing to permit symmetrical cylinder design and reduce distortion. For the smaller mechanical-drive turbines, the usual construction of the steam chest and valve is shown in Fig. 2-46. This shows a design of valve, called a "balanced" valve, positioned by the governor. Steam flows through a strainer to the space between the two valve seats and so leaves practically no net force, due to steam pressure, tending to open or close the valve. Hence, the force necessary to move the valve is very much reduced, and the governor can be more sensitive. In small units with such a single governor valve, hand-operated nozzle valves are also used, as shown in

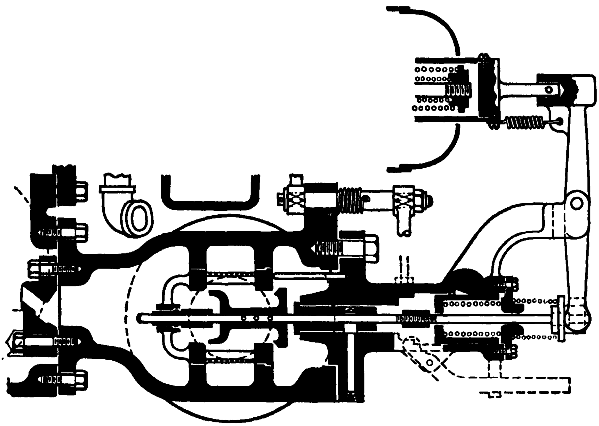


FIG. 2-46. Simple single inlet valve.

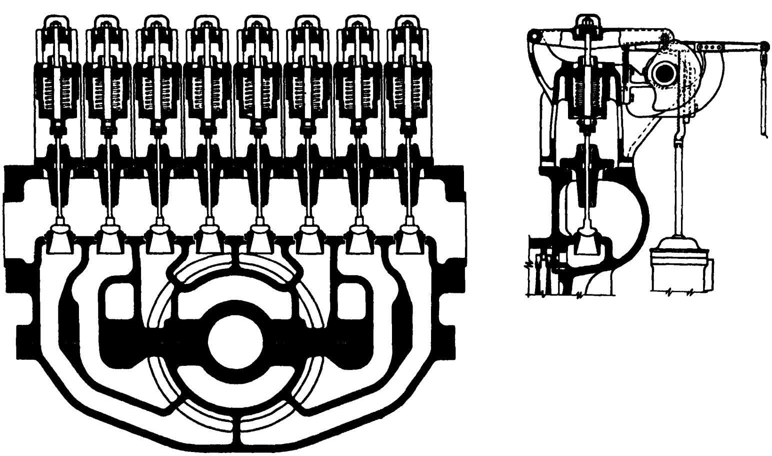


FIG. 2-47. Cam lift inlet valves.

Fig. 2-3. These permit changing the number of working nozzles for better efficiency at part-load and overload operation.

Commercial designs of large turbines, either completely impulse or with first-stage impulse, are shown in Figs. 2-47 and 2-48. Here each valve and seat lead to separate nozzle blocks. Two typical designs of valves and seats and two arrangements commonly used to operate the valves are shown. In both cases the valves open in a certain sequence. In Fig. 2-47 the governor mechanism actuates a camshaft which carries a cam for each steam-admission valve. Cam position and shape determine the sequence and rate of valve opening. The bar lift mechanism of

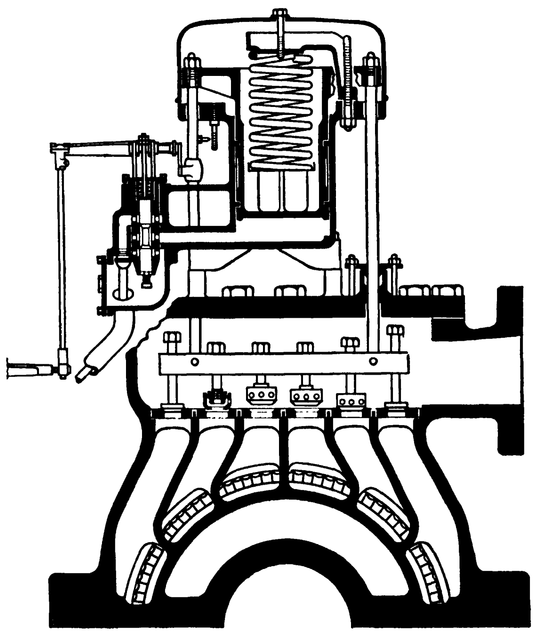
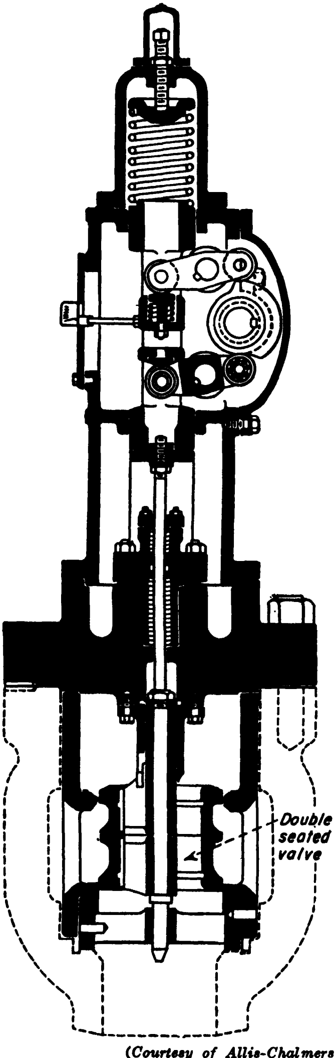


FIG. 2-48. Bar lift inlet valves.

Fig. 2-48 opens the valves in desired sequence with the same rate of opening for all valves. Valve stems pass through holes in a bar which is raised and lowered by the governor mechanism. Collars on each valve stem are set at different heights. Hence, as the bar rises, valves lift in a sequence determined by the collar settings. This opening of the nozzle groups, in series, gives more efficient operation at part loads.

The valves and seats for high temperatures and pressures (above about 600 psi and 750 F) are generally stellite-trimmed. The valve seats are fastened by a heavy press fit in the steam chest. For high temperatures the seat may be threaded in place. The diffuser type of valve seat permits high steam velocities through the throat with small pressure drop, and so helps to reduce the size of valves and valve gear.

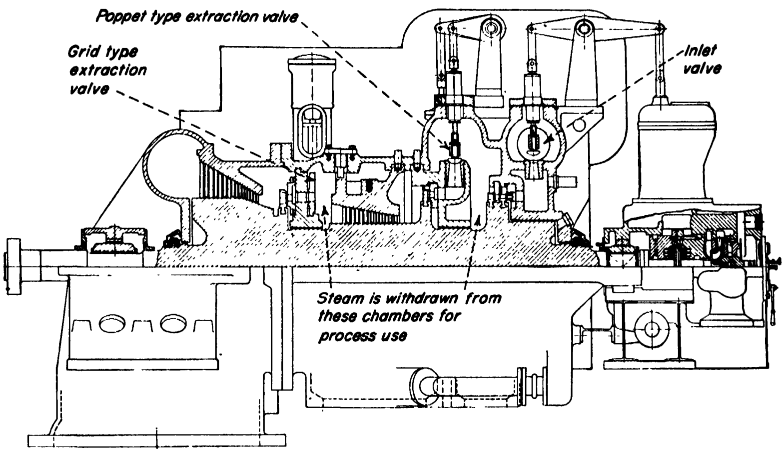
Straight reaction turbines require that steam be admitted to 100 per cent of the drum periphery. Hence no multivalve arrangement for steam admission to nozzle groups is used and a single-inlet valve, as in Fig. 2-49, is typical of such a unit. This type of valve can be used, also, if steam is admitted to successive lower pressure stages in a turbine. As shown, the valves are of the double-seated type and are practically balanced. This reduces the power required for their operation. Here also, the seating surfaces of the valves and seats are stellited. As shown, a gland is used for the valve stems with leakoff between the inner labyrinth and the outer packing gland. The mechanism for opening and closing each valve has a cam mounted on a shaft which is rotated by the governor. There are two levers pivoted in the center for each valve. The ends of the levers are fitted with rollers, one against the cam and the other against a guide in the valve stem. Two levers make both opening and closing occur by positive action of the cam. The coil spring on top assists in closing the valve.



(*Courtesy of Allis-Chalmers Manufacturing Co.*)

Fig. 2-49. Reaction-turbine inlet valve.

Extraction-pressure Control Valves. In turbines designed for the withdrawal of steam for process use, it is generally necessary to maintain the pressure of the extraction steam constant. If the extracted steam is dry or wet, constant pressure means that the temperature is essentially constant, which is highly desirable in many commercial uses. A turbine supplying such steam consists essentially of h-p and l-p, or h-p, l-p, and intermediate-pressure sections. Between these sections are chambers from which the steam is withdrawn. An extraction valve regulates the steam flow from



(*Courtesy of Westinghouse Electric Corp.*)

FIG. 2-50. Condensing double-extraction turbine showing the location of valves.

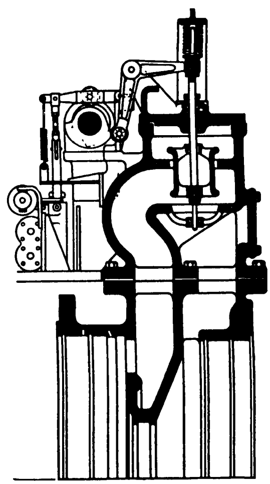


FIG. 2-51. Balanced-type extraction valve.

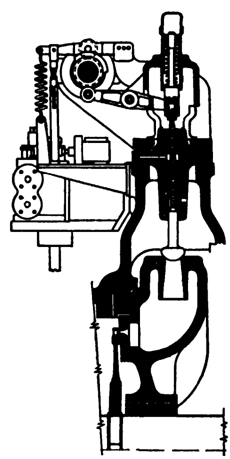


FIG. 2-52. Poppet-type extraction valve.

the chamber into the following stages, so that the steam pressure in the chamber and hence, in the extraction lines, remains constant. Figure 2-50 shows a typical two-extraction-stage turbine. The chambers and extraction valves are indicated.

Extraction valves are of two general types: single-ported and multiported. Figure 2-51 shows a balanced type of valve, of which one or more may be used. Figure 2-52 shows a single-seated poppet type of valve, commonly used in a multivalve steam chest, with the valves operating in sequence as the extraction steam flow varies. At low pressures, where steam volume is large, it is customary to use a multiported valve of the grid type, as shown in Fig. 2-53. This consists of a diaphragm containing a number of openings and a rotating shutter with matching openings. Movement of the shutter increases or decreases the

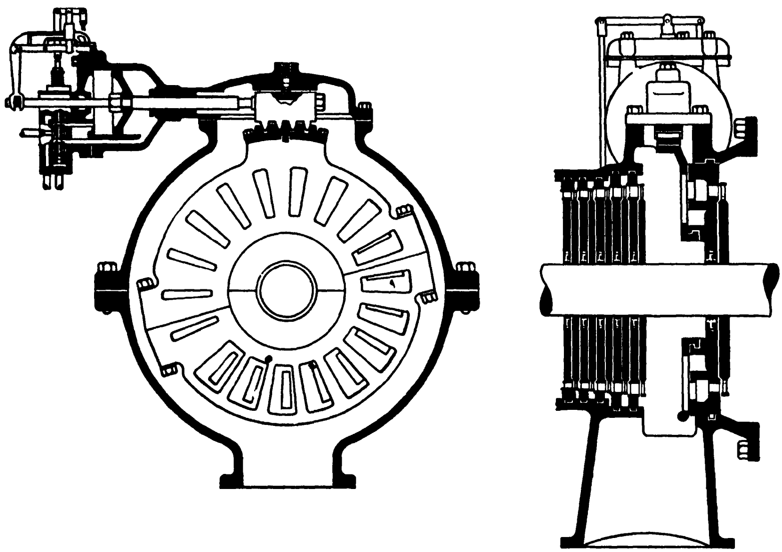


FIG. 2-53. Grid-type extraction valve.

port area as required. The location of this valve in a turbine is indicated in Fig. 2-50 and in the partial section to the right of Fig. 2-53.

Miscellaneous Turbine Valves. A number of valves, other than those already considered, may be used in turbines. In some turbines, by-pass or overload valves, as in Fig. 1-13, are provided to admit steam to the second stage, after the first-stage nozzles have reached maximum capacity. Intercepting valves, similar to inlet valves, may be used in the reheat steam pipes of a reheat turbine. They close when the steam turbine exceeds a predetermined speed.

Condensing turbines are often equipped with atmospheric relief valves. They are mounted in the exhaust line and held closed by atmospheric pressure. If the condenser vacuum fails and if exhaust pressure rises, this valve opens and vents steam to the atmosphere. Most units of large size are now equipped with explosion doors, or vents, to reduce excessive exhaust pressure and with a vacuum trip as the first relief device.

CHAPTER 3

TURBINE LUBRICANTS AND LUBRICATION

LUBRICATION PRINCIPLES

The present-day steam turbine is a carefully designed piece of mechanical equipment constructed of well-selected materials. Its satisfactory performance and useful life in service depend, among other things, on the maintenance of proper lubrication, one of the best insurances against turbine outage. The lubricating oil in turbines does three distinct jobs:

1. It reduces friction losses between all rubbing surfaces as in bearings, reduction gears, and couplings, and so reduces wear and improves efficiency.
2. It aids in removing the heat developed in these parts.
3. It acts as a pressure fluid to operate the governing and stop valves of larger turbines.

The lubrication engineers of turbine manufacturers and oil companies specify oils and diagnose lubrication difficulties. To cooperate intelligently with them and to operate a turbine unit properly, the power engineer and operator must understand certain lubrication principles. They must also be familiar with lubrication systems and oil characteristics. This material will be covered in this chapter.

Fundamentals of Bearing Lubrication. When the surface of one body slides across that of another, there is a resistance to the motion, called "friction." If the surfaces are rough, this resistance is large and never disappears completely, even if they are machined or ground. These surfaces, when looked at through a microscope, still show some roughness. The *solid friction* between such surfaces can be reduced considerably if they are separated by a film of fluid, such as a lubricating oil. The resistance to be overcome in moving the surfaces separated by such an oil film is called *fluid friction*. The oil forming the film tends to "wet" the metal and stick to it, but the oil itself is easily deformed and offers little resistance. Hence, in the case of bearings and other moving parts, an oil film is always used to separate them. This, in turn, causes less wear to the moving parts; and less friction work must be done, resulting in a cooler and more efficient bearing. Figure 3-1 shows the action of a normal turbine journal bearing in forming an oil film. The clearance is much exaggerated. In practice it is about 0.001 to 0.002 in. per in. of journal diameter. When at rest, metal-to-metal contact occurs between journal and bearing; but when starting, with oil being supplied, the journal tends to creep up the bearing wall opposite to the direction of rotation. As the speed picks up, the journal falls back and finally, at full speed, it drags the oil film completely across the rubbing surfaces.

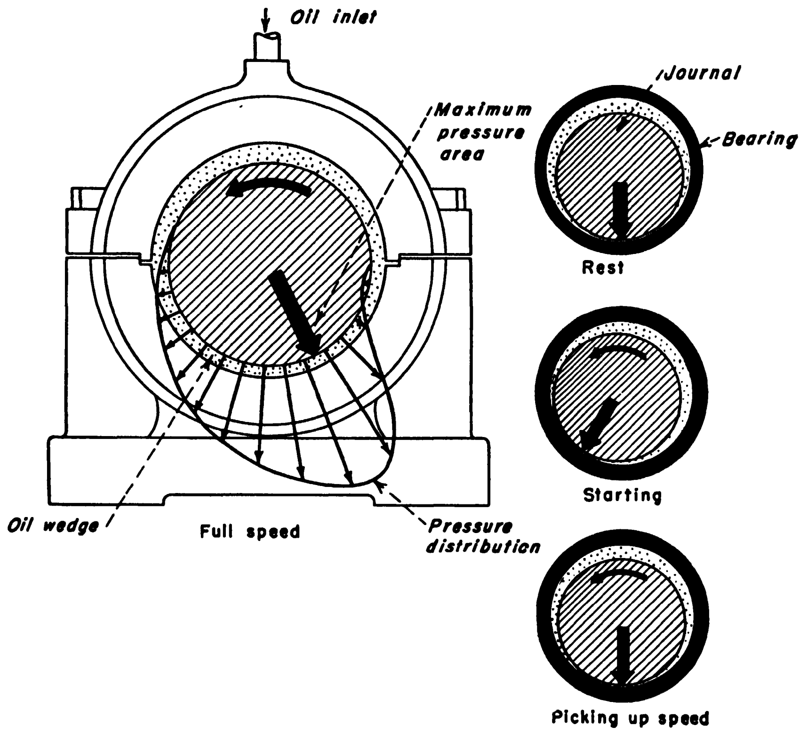


FIG. 3-1. Normal bearing-oil film and pressure distribution.

The oil film becomes a wedge with pressures sufficient to float the journal. These pressures are distributed approximately as shown in Fig. 3-1. Turbine journal speeds are high (often 3,600 rpm and up), and the unit bearing pressures are low. Bearing pressures in pounds per square inch are obtained by dividing the total load per bearing by the projected bearing area. Turbine shaft diameters are large, to enable delivery of the power. Keeping the shaft deflection small is necessary for small clearances. A unit bearing pressure of about 200 psi is the upper limit at present. Turbine bearings are about 0.8 to 1.3 as long as the diameter. The high speed and low loading pressure combine to make the formation of the oil wedge easy. The problem of lubricating the bearings is not difficult, so far as the bearings themselves are concerned.

Any end thrust in a turbine rotor, when not offset by a balance piston, is supported by a thrust bearing, the principles of which are shown in Fig. 3-2. Here, an effective oil wedge is formed, because of either tilting pads or beveled surfaces on the stationary part. This oil wedge can support heavy loads; and these bearings are used where large axial thrust loads are met. Other types of thrust bearings for light loads are the simple collar and the ball thrust bearing; but the most common form used in turbines is that shown in Fig. 3-2.

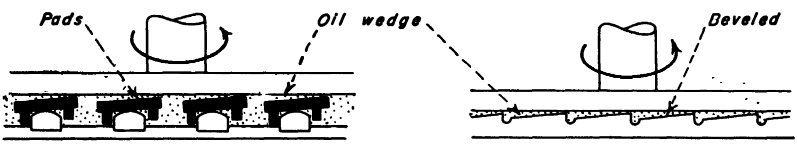
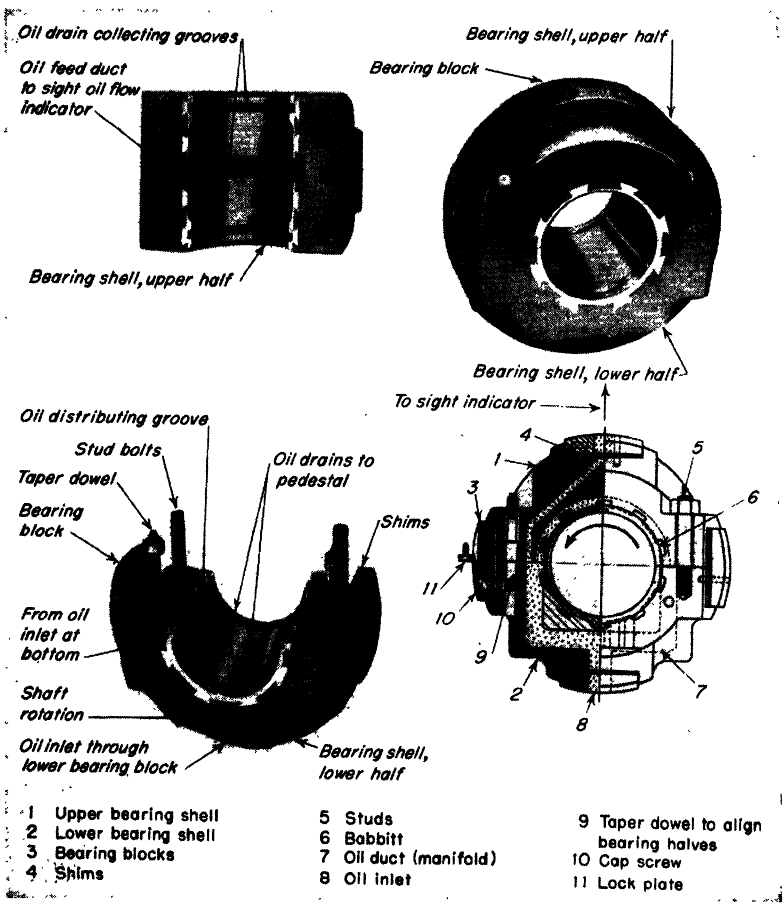


FIG. 3-2. Typical turbine thrust-bearing oil films.

Bearing Grooves. In the design of normal bearings, grooves are cut in the bearing to distribute oil along the length of the journal. These grooves extend to within approximately ½ in. of the bearing ends and are made with chamfered or rounded edges, in the direction of rotation, so as not to scrape off the oil. Most important, these grooves are never cut in the h-p area of the oil wedge but are placed parallel to the shaft in the l-p area. Oil is supplied to the bearing at a point not under pressure and is commonly distributed by a groove. Figure 3-3 shows the upper and lower halves of a typical large turbine bearing. The assembled bearing and a section through it are also shown. The oil entering at the bottom flows upward to an oil-distributing groove in the l-p area and leaves the bearing through the oil drains to the pedestal. In small turbines, the ring-oiled bearing, discussed later, is used in preference to the force-feed type shown in Fig. 3-3.

Additional Uses of Lubrication Oil. A turbine bearing becomes hot because of friction. In addition, heat from the high-temperature parts of a rotor flows through the shaft and increases the bearing temperatures. To run large high-speed bearings and not have them overheat, cool oil must be used, and considerably more than the minimum required for lubrication must be supplied. The amount of oil can be varied so as to limit the temperature rise of the bearing. Large turbines with an oil supply temperature of about 110 to 120 F circulate sufficient oil so that the oil temperature rises about 20 to 30 deg. This heat is subsequently removed by the use of oil coolers. The bearing is cooled, also, by heat radiated from the pedestal; but this amount, in large bearings, is negligible compared to the total heat generated.



(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 3-3. Typical large turbine bearing.

When the power unit consists of a turbine driving through speed-reduction gear sets, the gear and turbine lube-oil systems are commonly integral. In this case, the oil does additional duty in gear lubrication.

Governor mechanisms for speed, back pressure, and extraction valves, on turbines of large capacity, are hydraulically operated; *i.e.*, oil under pressure acts on a piston which, in turn, by means of the piston rod, opens or closes steam-inlet valves. This method is used because the steam valves are bulky and governors cannot be sensitive and at the same time develop large enough forces to operate the valves directly. Instead, the governor controls a pilot valve, which regulates flow of oil to the operating piston. The turbine lubrication system supplies this oil to operate the governor valve mechanism. Hence, a supply line to the governor and a return line are required as parts of a lubrication system.

Lube oil is used in certain turbines for emergency overspeed trips to close a main-steam-line valve. Oil also serves as a hydraulic fluid to operate the main-stop or throttle valve in the steam line leading to a large turbine. Hydrogen-cooled generators use lube oil for sealing purposes.

Types of Turbine Lubricants. Lubricating oils for turbines are mineral oils not compounded with animal or vegetable oils. They can be manufactured from crude petroleum from various fields. The Pennsylvania fields produce a *paraffin-base* crude oil; the western fields, an *asphalt-base* crude; and the mid-continent fields, a *mixed-base* crude. Some turbine oils are produced by conventional distillation processes and treated with acids to take out unwanted compounds. Then they are filtered through fuller's earth, which makes them better able to separate from water. The quality of these oils depends upon the quality of the crude and the experience and refining methods available to the manufacturer. Processes using other solvents than acids to remove undesirable compounds from the refined oil are now common. The solvent used depends on technical considerations of the crude-oil characteristics. At times more than one solvent is used. Some of these processes produce an oil that tends to oxidize easily. Turbine oils commonly have *inhibitors* or *additives* mixed with them. These additives, aside from reducing the tendency to oxidize, may retard rusting in the lube-oil system, improve the oil viscosity index, improve the oiliness of lube oil, and bring about other beneficial results.

Mixing of turbine oils is commonly done in turbine operation. It is not recommended, however, with oils that have additives, because they may react on each other and offset their good effects. Recommendations for the proper lube oil will be given later. Oil companies generally market turbine lube oils under distinct brand names. Usually they are available in a variety of grades of different viscosities. Greases are used, to some extent, in lubricating certain small turbines of the mechanical-drive type.

PROPERTIES OF LUBRICATING OIL

Lubricating oils are necessary for so wide a variety of uses that a large number of tests and specifications have been set up and standardized to give their various properties. Some tests, or analyses, deal mainly with new oils, others with used oils, and many, with both new and used oils. In relation to turbine oils, the following terms are commonly used.

1. Viscosity
2. Viscosity index
3. Neutralization or electrometric acid number
4. Steam emulsion number
5. Demulsibility
6. Precipitation number
7. SAE number
8. Flash and fire points
9. Carbon residue
10. Pour point
11. Gravity API
12. Saponification number
13. Oxidation tests
14. Corrosion tests
15. Color
16. Oiliness

1. *Viscosity:* An indication of the fluidity, or body, of an oil and a measurement of the resistance offered by a fluid to flow, or its internal friction. The viscosity of any oil changes as the oil becomes heated or cooled. Viscosity is useful, as it affects the heat produced, the rate of oil flow, and the fluid friction in a bearing. Low-viscosity oils lower friction loss and distribute well over bearings but must be viscous enough to keep an oil film in a bearing. The common unit used in the United States for measuring this property is Saybolt Universal Seconds (SUS or SSU) which gives the number of seconds required for 60 ml of an oil to flow through the orifice of a standard Saybolt Universal Viscosimeter, at a given temperature. Standard temperatures are 70, 100, 130, or 210 F.

2. *Viscosity index* (V.I.): A number which indicates the viscosity changes of an oil as its temperature changes. It is obtained by comparing the viscosity and temperature changes of a given oil to those for two standard oils. Oils thin out (low viscosity) at high temperature and thicken (high viscosity) at low temperature. A high V.I. oil does not become excessively thin when heated or thicken when cooled and is desirable for turbines.

3. *Neutralization number:* A number indicating the free acidity in an oil. It is the weight in milligrams of potassium hydroxide (a strong alkali) required to make a 1-gram sample of oil neutral. This number helps to show how much a new oil has been refined or the extent to which a used oil may tend to corrode or form sludges and emulsions. Acidity is now, sometimes, given as the electrometric acid number which is

equivalent to the neutralization number but depends on different testing techniques.

4. *Steam emulsion number:* The result of a test for the ability of an oil and water to separate. Oil and steam that condenses are mixed. The number of seconds required to separate is the steam emulsion number.

5. *Demulsibility:* Another test for the ability of an oil to separate from water. Oil and distilled water are mixed, and the rate in milliliters per hour at which the oil settles out is the demulsibility.

6. *Precipitation number:* The number of milliliters of sludge formed when 10 ml of an oil are mixed with 90 ml of naphtha and centrifuged. It is used to measure the amount of sludge in used oils.

7. *SAE number:* A classification of lubricating oils based on their viscosity. SAE No. 30 is an oil with a viscosity of 185 to 225 SSU at 130 F.

8. *Flash and fire points:* Temperatures of oil at which its vapor will flash or burn continuously in a standard test.

9. *Carbon residue:* The result of a test to determine the carbon residue remaining after controlled distillation of an oil sample.

10. *Pour point:* Lowest temperature at which an oil will flow under prescribed test conditions.

11. *Gravity API:* An indication of the density of an oil. It is the number read on an API calibrated hydrometer float placed in oil. Conversion of this number to specific gravity is made by the following calculation:

$$\text{Sp gr at 60 F} = \frac{141.5}{131.5 + \text{deg API at 60 F}}$$

The API number varies with temperature. 60 F is used in the above. The heavier the oil, the lower is the API reading.

12. *Saponification number:* The result of a test in which oil is saponified with potassium hydroxide. It measures all the material in an oil reacting with the alkali and not just the free acid as in the neutralization number test. Specifically it is the number of grams of potassium hydroxide used for 1 gram of oil.

13. *Oxidation tests* (Indians, Funk, Staeger, Sligh, Brown Boveri, Farmer, Gulf, Rogers, ASTM): These are all variations of tests to determine the amount of oxidation of an oil, when mixed with air and some catalyst at high temperatures. These tests are used to judge the possible performance of an oil in service.

14. *Corrosion tests* (ASTM, sea-water rusting test, Navy, Shell film tenacity test, MacCoull): These are tests of the effect of an oil in corroding a steel specimen under standard conditions of moisture and temperature. They indicate the ability of a turbine oil to prevent rusting.

15. *Color* (Saybolt, ASTM, Robinson): These are methods to give a number rating of the color of an oil by matching it with standard colors.

16. *Oiliness:* An oil property that has not been measured. It refers to the difference in fluid friction in oils not accounted for by viscosity difference alone.

Other terms than the above are used for lube oils in general. However, the above are common for turbine oils. Of these, viscosity, viscosity index, neutralization number, flash and fire points, gravity API, and the SAE numbers are most used in operation. Unless otherwise indicated, most values shown are based on tests made in accordance with the latest approved standards of the American Society for Testing Materials (ASTM).

PERFORMANCE CHARACTERISTICS OF TURBINE OILS

Lubrication-oil specifications for turbines are commonly furnished by builders. The specifications are fairly broad, and selecting an oil to meet them is a relatively simple matter. Lube oils, however, tend to deteriorate by changing their chemical and physical properties when in use in a turbine. The principal problem is to select an oil that will deteriorate to only a small extent when in service.

Causes of Oil Impairment. Turbine lube oils lose their lubricating value because of many factors some of which are easily recognized and others are somewhat harder to determine. The several causes of trouble with lube oils are the following:

1. *Moisture:* Due to leakage from glands; leaks in oil-cooler tubes; condensation forming on inner sides of sumps, gearing and bearing castings, and housings, because of different temperatures inside and out.

2. *Air:* Due to leakage into suction line to oil pump; inadequate venting; low oil level so that the suction line draws air; splashing of oil flowing into reservoir or through cleaning system; air contact with oil in bearing housing.

3. *High temperatures:* Result of insufficient cooling; too little oil; improper oil being used; mechanical defects of bearing surfaces; incorrect bearing clearances; improper alignment; oil pipes and reservoir too close to casing.

4. *Catalytic effect of metals:* Metals in general exhibit a catalytic effect on the oxidation of oils; *i.e.*, metals in contact with oils help to speed up the oxidation of the oil. Copper (often found in screens) is the most active of all the common metals and should be avoided.

5. *Mixing of oils:* This results either from deliberate mixing or adding new oil when some old oil remains in the system. Mixing usually shortens the life of the new oil but does have the effect of helping the

new oil to adhere to, or wet, the metal surfaces and thus acts as an effective rust inhibitor. As little as 1 per cent of used oil is effective in this respect, although up to 10 per cent is at times used.

6. *Foreign matter as dust, dirt, fly ash, core sand, metal particles, or rust:* Such matter gains entrance to the oil through oil-well covers or by scale or sand in oil passageways. Rusting is due to decomposition products of oil or from water reaching iron surfaces.

7. *Insufficient oil supply:* Due to excessive air or water, dirty strainers, clogged pipes, or too little oil in the system.

8. *Low oil pressure:* Resulting from too little oil in system; faulty relief valves.

Common Oil Troubles. When turbine oils are subjected to any of the above factors, they tend to lose their value as lubricants. The following conditions often develop:

1. *Foaming or frothing:* Caused by air mixing with the oil, or by adding new oil too quickly to a system. Foaming may cause a drop of oil flow to bearings, faulty governor action, overflow from a system. It is generally dangerous and messy. Elimination of the causes of air leaks, a slight change in oil pressure, or running oil a little hotter is helpful in foam prevention. Fall of oil in high vertical pipes also has been a cause of foaming. Oil should be allowed to rise quickly to operating temperatures as a preventive.

2. *Emulsification:* Caused by the mixing of oil and water, usually with air and solids being also present. The effect lowers the lubricating value of the oil and may cause deposits to settle out in bearings. This condition is offset by the choice of an oil with high demulsibility and by removal of water, acids, and colloidal solids due to oil oxidation. Heat helps to break an emulsion. This practice, not common in land turbines, is used in many marine units by heating the oil in a reserve tank. The tendency of a used oil to emulsify is not easily found from laboratory tests.

3. *Oxidation:* This results from air acting chemically on oil. The action is hastened by high temperatures and the presence of water. The oxidation rate is doubled for approximately each increase in oil temperature of 20 F. Oxidation produces such substances as alcohols, ketones, organic acids, and metallic soaps. When these break down, they form sludges some of which dissolve in the oil at operating temperatures, while others do not. The insoluble material tends to clog oil lines. The soluble material tends to settle out at low temperature in such parts as oil coolers, oil reservoirs, and the governing system. Much of this material from oxidation is soluble in water. Low oil temperatures and elimination of air are helpful in retarding oxidation. Some oxidation always occurs, however, and the oil can be improved if the solid materials

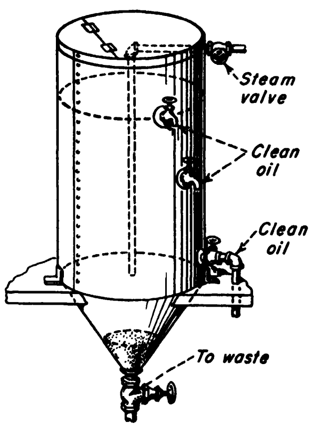
and those soluble in water are removed. All modern turbine oils have inhibitors added which tend to retard the oxidation; but it cannot be completely eliminated. The neutralization number is the index of the extent to which oxidation has occurred.

4. *Corrosion:* Caused by the presence of water which corrodes gears, governor control mechanisms, journals, pumps, housings, reservoirs, etc. Acids formed as a result of oxidation are other possible causes of corrosion. When rust forms as a result of corrosion, it often gets into the oil and has an abrasive effect; some forms of it also help the oil to oxidize by acting on the oxidation inhibitors.

5. *Sludges and deposits:* The results of oil oxidation and also of foreign matter getting into a system. They act to clog flow lines; cover cooler surfaces; become abrasive on bearing surfaces; and clog governors and other valve mechanisms. Elimination depends upon retarding oxidation; stopping entrance of foreign matter; and removing the deposits that form despite proper oil maintenance.

RECONDITIONING AND MAINTENANCE OF LUBRICATING OIL

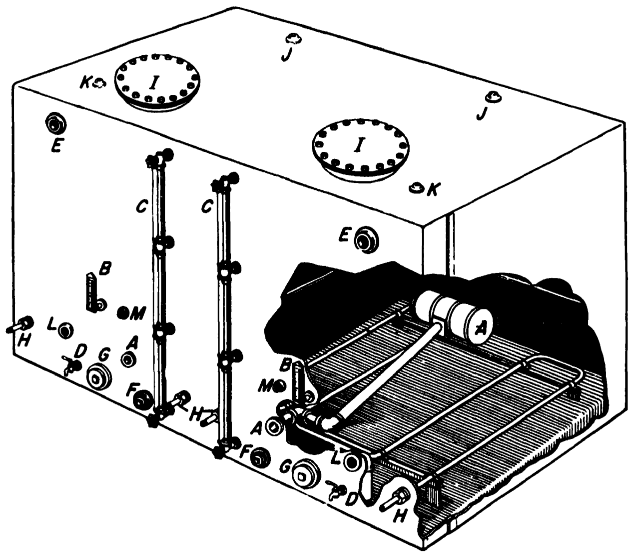
After being in service, lubricating oils of turbines become contaminated with water, soluble impurities, and deposits known as "sludge." Sludges form after an oil has been in use for considerable time and are due to changes resulting from oxidation, foreign matter, and emulsions. Sludge may be a comparatively dry deposit mainly of oxidized hydrocarbons, or it may be a liverlike, slimy substance consisting of oxidized hydrocarbons, foreign matter, some emulsions, and some good oil. Oil deterioration products are removed in a variety of ways depending upon whether they are insoluble sludges or soluble products. Several methods are commercially used for oil purification of turbine oils. These methods require one or more of the following types of equipment:



(*Courtesy of American Machinist.*)
FIG. 3-4. Oil settling tank.

1. Settling or precipitation tanks
2. Centrifuges
3. Filters
4. Washers

1. *Settling or precipitation tank:* The simplest means of clarification is to allow the used oil to stand in a tank quietly for a period from one night to one or more weeks. Settling is mainly intended to remove water and heavier sludges. Gravity alone causes the separation. Tanks are usually equipped with steam heating coils as the warmer, less dense



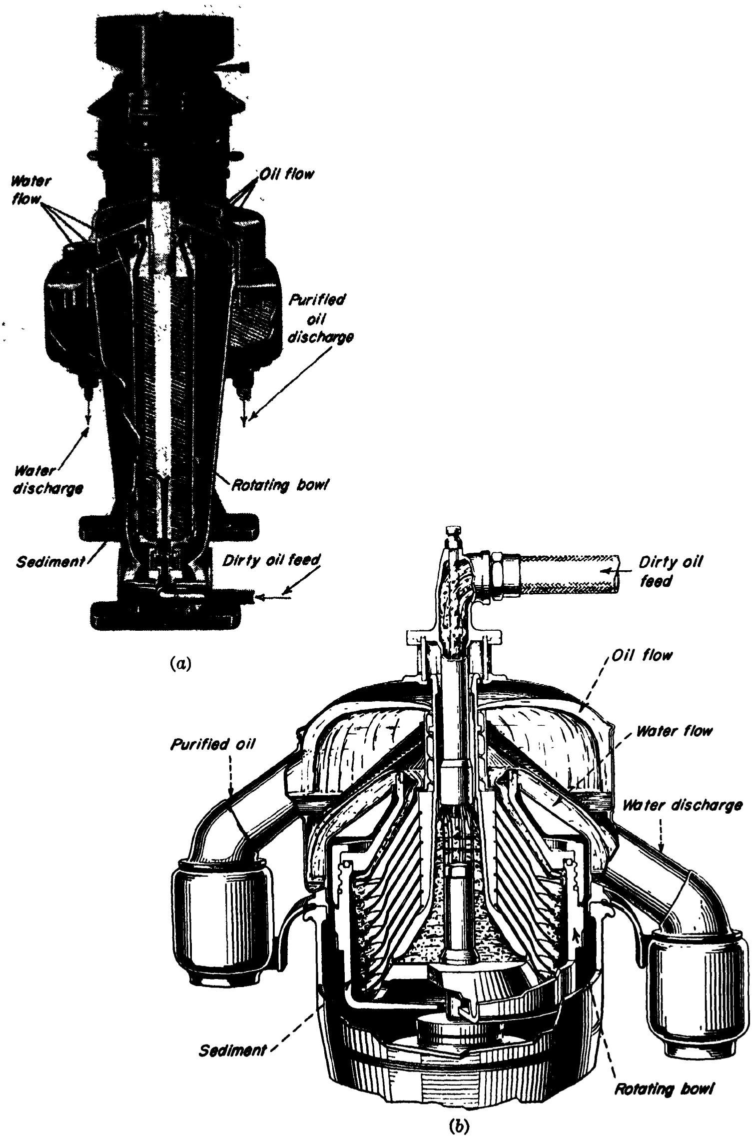
A - Floating suction
B - Thermometer
C - Oil gage
D - Test cock
E - Inlet flange
F - Drain flange
G - Cleanout flange
H - Steam coil
I - Manhole
J - Vent
K - Oil level indicator flange
L - Suction connection
M - Temperature regulator connections

(*Courtesy of Bowser, Inc.*)

FIG. 3-5. Oil precipitation tank.

oil clarifies more easily. High temperatures should be avoided. A temperature of 120 to 180 F is used. Heat should be applied only at the start, and then the oil should be allowed to cool and settle. During heating, convection currents in the oil prevent separation. Settling tanks are also convenient to use as dump tanks in case of fire, or for repairs. Figure 3-4 shows a simple form of settling tank with suitable valves for oil and waste removal.

In Fig. 3-5 an effective form of flat tank is shown. Here, the tank is divided into two sections, each fitted with a floating suction A, thermometer B, oil gauge C, test cock D, inlet flange E, drain flange F, clean-out flange G, steam coil H, large manhole and cover I, vent J, oil-level



[Courtesy of (a) Sharples Co. (b) De Laval Separator Co.]

FIG. 3-6. Typical centrifuge bowl sections.

wall indicator flange K, deep suction connections L, and temperature regulator connections M.

Settling tanks necessitate two batches of oil for the turbine, which must usually be shut down when oil is removed. When settling tanks are used, the batch of oil in service builds up impurities to some extent before a cleaning cycle occurs.

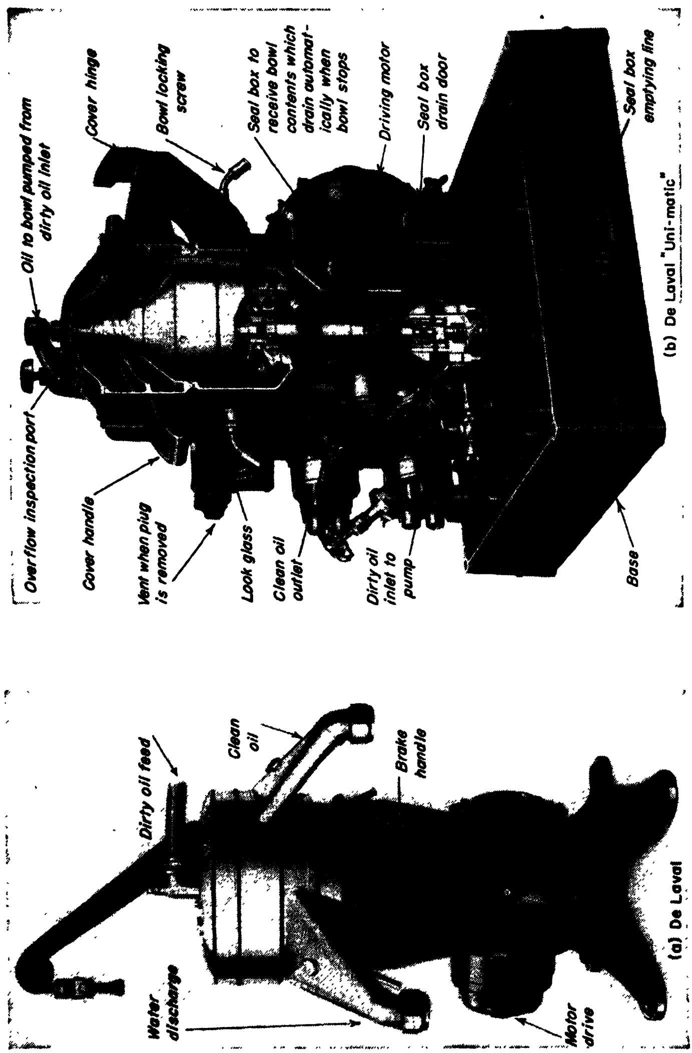
2. *Centrifuges:* Centrifuging, as a means of cleaning oil, is used to remove water, solid matter, and some of the water-soluble oxidation products. Dirty oil is fed to a rapidly rotating bowl, as in the designs in Fig. 3-6. Centrifugal force tends to throw solid matter and water, which are both heavier than oil to, the walls of the bowl. The solid matter must be removed at intervals by stopping the centrifuge and cleaning the bowl. The dirty oil flows in steadily, since the water that separates from the oil and the resulting clean oil can be made to flow continually out of the bowl, over rims, as shown in the figures.

Hot water, usually condensate, may be added to the dirty oil before it enters the bowl. That is called "wet" centrifuging; "dry" centrifuging means handling oil as it comes from the reservoir. Wet centrifuging, when feasible, produces a cleaner oil, because the added water helps remove acids and impurities, such as fly ash and dust, by causing this material to form into large particles which centrifuge more easily. Since emulsions may be formed, there is some danger that the oil, if not watched, may come out of the water discharge pipe. The system may, in this way, be drained of oil.

Figure 3-7 shows the outside appearance of a typical centrifuge and also sections through two commercial types used for turbine oil cleaning. Dirty oil feed lines and the discharge lines are indicated, together with additional features for practical operation. Using increased oil temperatures in operation gives better cleaning results. Temperatures of about 130 F are common, although occasional cleaning at temperatures up to 180 F is recommended. It is good practice to operate a centrifuge not at its rated capacity but rather at a lower rate. Good results are obtained when operating at about one-fourth to one-third of the rated gpm flow. When centrifuges are installed, it is advisable to have considerable excess capacity for part-load operation, as mentioned above, and also to provide for unusual quantities of dirt in the oil.

3. *Filters:* A wide variety of filters is used for lube-oil cleaning. They may be roughly grouped into the following forms:

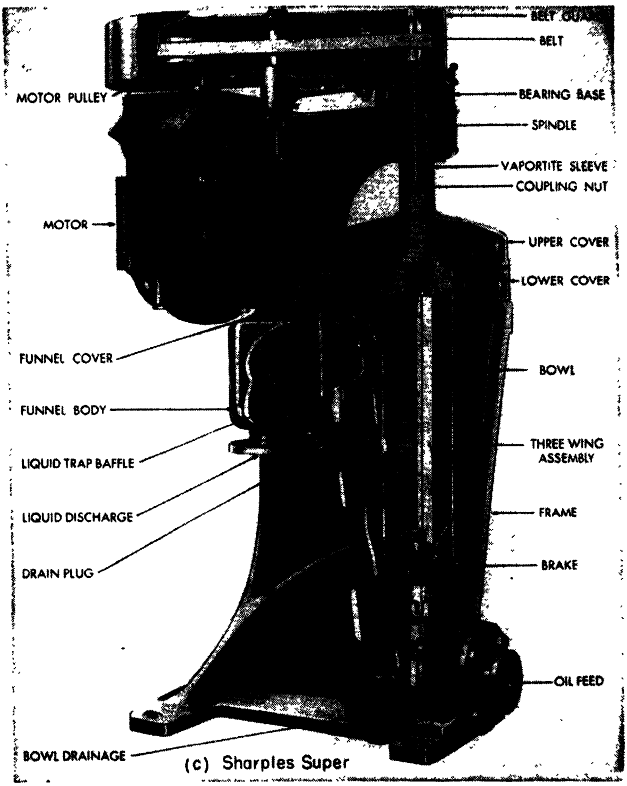
a. Strainers
b. Absorbent filters
c. Adsorbent filters



(Courtesy of De Laval Separator Co.)

FIG. 3-7. Typical oil centrifuges.

Strainers are made of screens, metal disks, steel wool, copper ribbons, cloth, or blotting paper. They filter out the coarse solid impurities. Water, except in blotter filters, is not removed. Soluble oxidized material and material such as dust, and fine soluble oxidized material is not

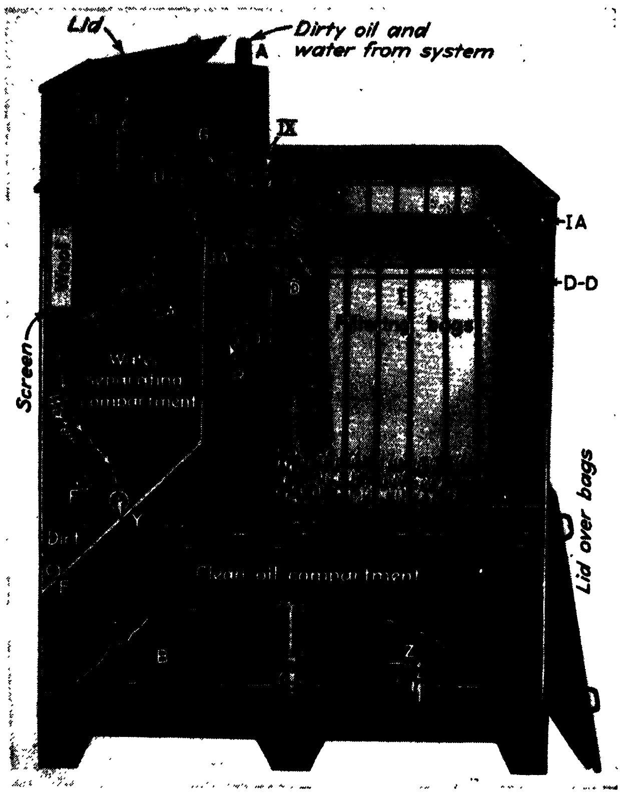


(*Courtesy of Sharples Co.*)

FIG. 3-7. (*Continued*).

retained by them. Strainers are often part of a filter using a combination of methods for cleaning. Figure 3-8 shows a gravity-fed filter of this type, with cloth bags. Oil and water enter at *A* and pass over a steam or electric heater at *G*. Distributing holes *M* lead into a wool screen and from here to a water separating chamber. Here, the oil flows obliquely to the skimming weir *N* and then through valves *O* to the filtering bags. Clean oil leaves at *B*. Separated water leaves at *Y* to be discarded. Sludge and sediment leave at *F*. Excess dirty oil overflows at *C* through

CA to general overflow R to the sump tank. Filter bags are never submerged; if they are, they overflow into at rough J and ring an alarm bell on the outside. The bags are then manually raised. This closes valve O. The bags are then cleaned. They may also be hung on IA, as



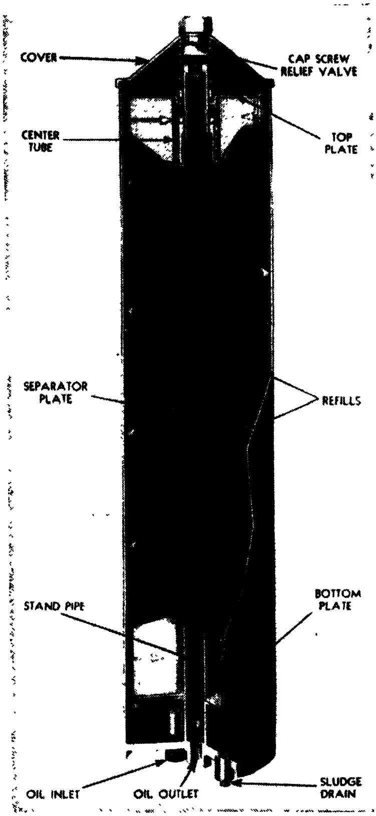
(Courtesy of Wm. W. Nugent & Co., Inc.)
FIG. 3-8. Gravity-type oil filter.

shown, to drain. Filters of this type are built in various sizes to clean from forty to several thousand gallons per hour.

Absorbent filters are made of cotton waste and other types of absorbent cellulose material. A filter of this type, in single cartridge form, is shown in Fig. 3-9. The filter and oil flows are clearly indicated. These filters remove coarse and fine solid material; certain forms also take out water and mineral acids. Oil is pumped through these filters. They are always built with a relief valve, as shown, to by-pass the oil if the filter becomes too clogged. Figure 3-10 shows a multicartridge form of this type of filter. Filters of this type require refill cartridges when the difference in pressure between inlet and outlet becomes large. Manufacturers' recommendations, modified by the operator's experience will determine these pressure conditions.

Adsorbent filters use a filter medium of fuller's earth, diatomaceous earth, charcoal, or other similar material. These filters remove coarse and fine material, also water and some soluble oxidized materials. In use, this material may be mixed with the oil, and the mixture forced through a filter press. These materials are also used by having cartridges constructed of them and used as refills in filters, as shown in Figs. 3-9 and 3-10. The adsorbent type of filter is detrimental in that it removes many of the chemical additives used as oxidation or corrosion inhibitors; it should, therefore, be used with caution in filtering turbine oils.

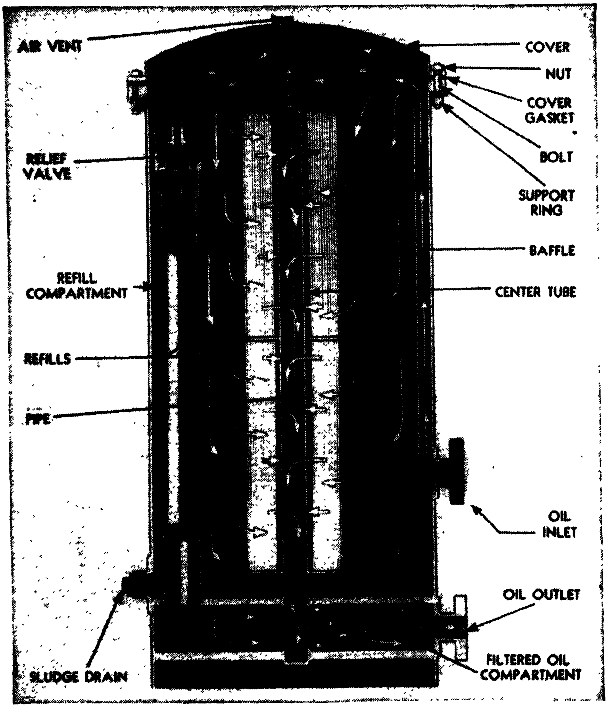
4. *Washers:* Used lube oil can be cleaned of many of its oxidized impurities if it is passed through a water bath since much of this material is more soluble in water than in oil. A separate wash tower, or a washing section, is part of some maintenance or conditioner systems. Figure 3-11 illustrates a typical oil-washing section, as part of such a system.



(*Courtesy of Briggs Clarifier Co.*)

FIG. 3-9. Single-cartridge pressure-type oil filter.

In this system, about 25 per cent of the oil to the conditioner is put through the washer for acid correction. Operation is as follows: oil for washing enters *A* and passes diffusing plate *B*, upward through heated water. Dehydration elements *G*, consisting of a perforated core *C*, mass *D*, cloth bag *E*, and screen *F*, remove water from the oil. After passing over weir *H*, it joins with the remaining oil to be cleaned, which enters at *I*, and flows downward in the precipitation compartment, under baffle *J*, and up through screens *K*, over weir *L*, and under baffle *N*, to the filtration compartment. The oil then passes through cloth-covered filtering elements *O* and, via outlet nozzles, through trough *P* and pipe *Q*



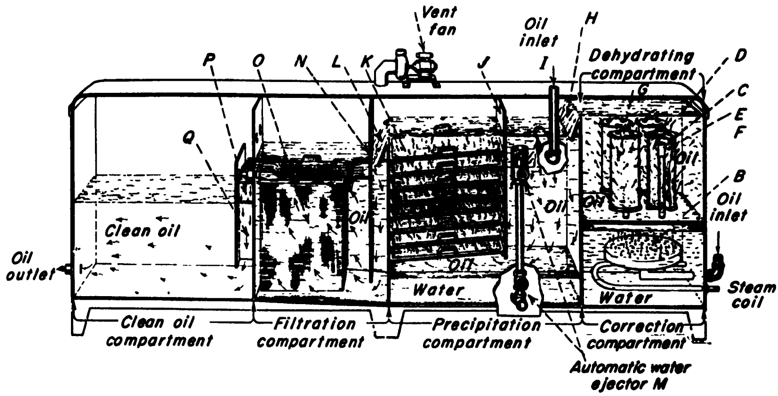
(*Courtesy of Briggs Clarifier Co.*)

FIG. 3-10. Multicartridge pressure-type oil filter.

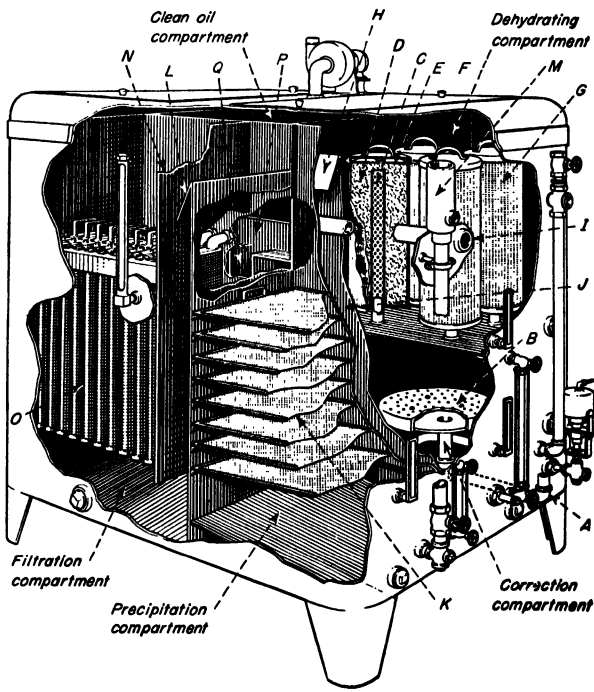
into the clean-oil compartment from which it is pumped to the oil reservoir.

Turbine-oil Maintenance Systems. Several methods for oil maintenance are in use, depending upon the type of turbine and service conditions. They may be grouped as follows:

1. Make-up
2. Sweetening
3. Batch treatment
4. Continuous treatment
5. Continuous by-pass treatment
6. Combination by-pass and batch treatment



(a)



(b)

(*Courtesy of Bowser, Inc.*)

FIG. 3-11. Oil washer and conditioning system. (*a*) Schematic section. (*b*) Actual construction.

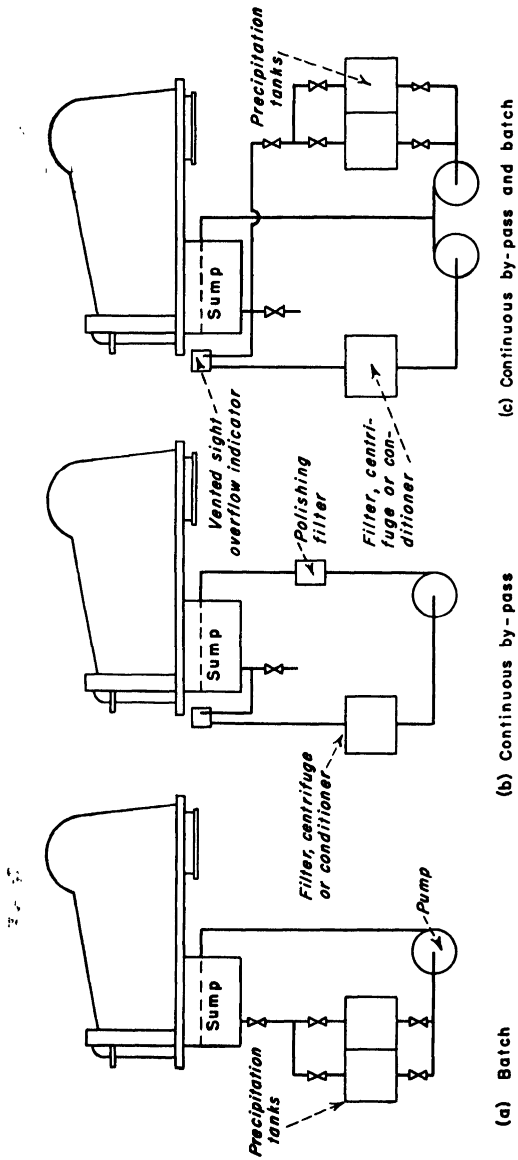


FIG. 3-12. Conventional lube-oil treatment systems.

1. *Make-up:* Small ring-oiled turbines are most commonly lubricated by adding oil to the system periodically to make up for lube-oil losses. In these turbines the oil is frequently changed completely and replaced with a charge of fresh oil.

2. *Sweetening:* In this method about 10 per cent of the oil in the system is withdrawn at intervals from the bottom of the oil reservoir and sump. New make-up oil is added, and the drained oil is either discarded or cleaned and used in other equipment. The method is not favored for

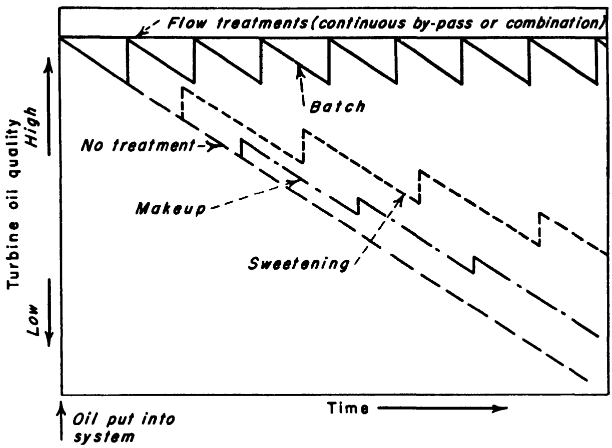


FIG. 3-13. Comparison of various lube-oil treatments.

turbines. It allows the oil to deteriorate between sweetenings. The make-up oil is costly, and adding it to the old oil may at times cause solids to come out of the old oil. Even with sweetening, the oil charge tends in time to lose its lubricating value.

3. *Batch treatment:* The use of settling tanks, as in Fig. 3-12, to treat the entire batch of oil in a turbine system is one of the simplest methods for oil maintenance. Two tanks, each large enough to hold all the oil in the turbine, are used. This system removes both solid matter and water from the oil charge.

4, 5, and 6. *Flow treatments:* Better results are obtained in oil maintenance if the oil charge is treated continuously. Instead of passing the entire oil flow (*continuous treatment*) handled by the main turbine lube-oil pump, through a cleaner, as centrifuge, filter, or conditioner, only part of the oil (*continuous by-pass*) is treated. The latter method is effective and less costly, and is the favorite of the two methods. Figure 3-12 shows this system and an improvement on it by using it in combination with a batch method. This is a better method since some oil contaminants are soluble at operating temperatures, but precipitate out when allowed to stand in tanks at room temperature. Depending on the filter, centrifuge, or conditioner used, the flow treatments remove solid impurities, water, and many of the soluble products of oil oxidation, and so give a continuous life to the oil change.

There are specially designed filters for the removal of the extremely fine solids that pass through many of the conventional filters. These special filters have elements made of a sheet of cellulose impregnated with neutral synthetic phenol resin. They are referred to as "polishing filters" and are employed, as shown in Fig. 3-12, after the usual filters, centrifuges, or conditioners.

The effects of these various treatment methods on oil life are compared graphically in Fig. 3-13. Here, it is seen that flow and batch treatment keep up oil quality, but that make-up and sweetening delay somewhat the oil deterioration, depending on the intervals at which clean oil is added.

SYSTEMS OF TURBINE LUBRICATION

The lubricating systems of steam turbines vary widely, depending on the size and type of the unit and, to a great extent, on the age of the turbine. The general mechanical designs of turbines can be classified as follows:

Direct-drive
Geared

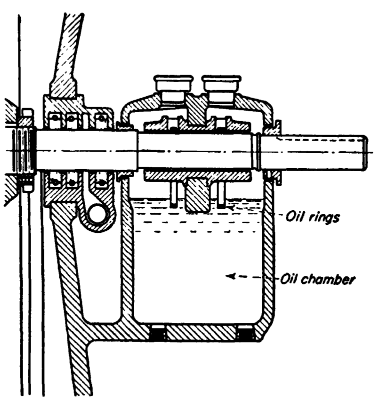


FIG. 3-14. Simple ring-oiled bearing.

Direct-drive. This term means that the main turbine shaft is coupled directly to the shaft of the equipment being driven. No intermediate gears are used, and the lubricating oil system provides mainly for the bearings supporting the shaft. Oil supply for the governor system is commonly an essential part. Incidental lubrication, as of the couplings, and certain functional uses, *e.g.*, the medium for opening the main stop valve, may be part of a complete system.

The lubrication systems for these turbines fall into either one or a combination of the following methods:

1. Ring oiling
2. Circulating (pressure or gravity)

Ring oiling: In small turbines, the oil quantity required for lubrication is small, and the supply system is relatively simple. Figure 3-14 illustrates a ring-oiled bearing. The ring rides on the journal, and its lower arc dips into the oil reservoir. As the shaft rotates, the ring revolves correspondingly. Oil from the reservoir clings to the ring and is carried to the top of the journal to which it transfers by direct contact. At high speeds some form of scraper is common to remove oil from the rings and direct it to the bearing. Grooves in the bearing permit the oil to travel in axial directions and so provide proper lubrication at the bearing areas along the length of the journal. Excess oil drains back to the reservoir beneath the bearing and aids in cooling the journal and bearing. In small turbines, the surface areas of the reservoir are large enough to radiate this accumulated heat and maintain the oil at a safe temperature. For some turbines it is necessary to cool the oil by passing circulating water through a chamber adjacent to the reservoir or through a coil immersed in the oil, as in Fig. 3-15. In this way high operating temperatures, which would seriously shorten oil life, are prevented.

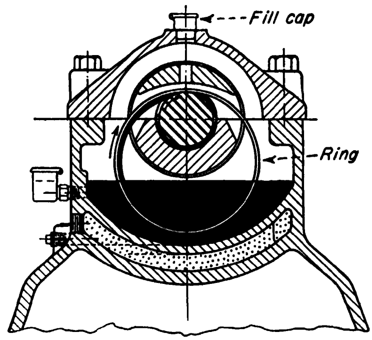
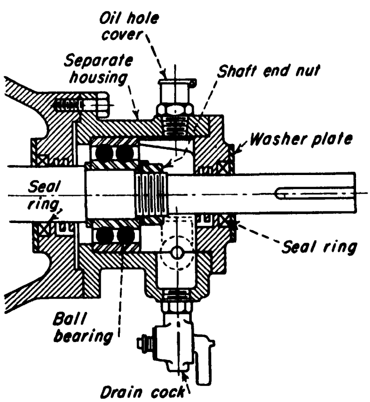


FIG. 3-15. Typical water-cooled ring-oil bearing.

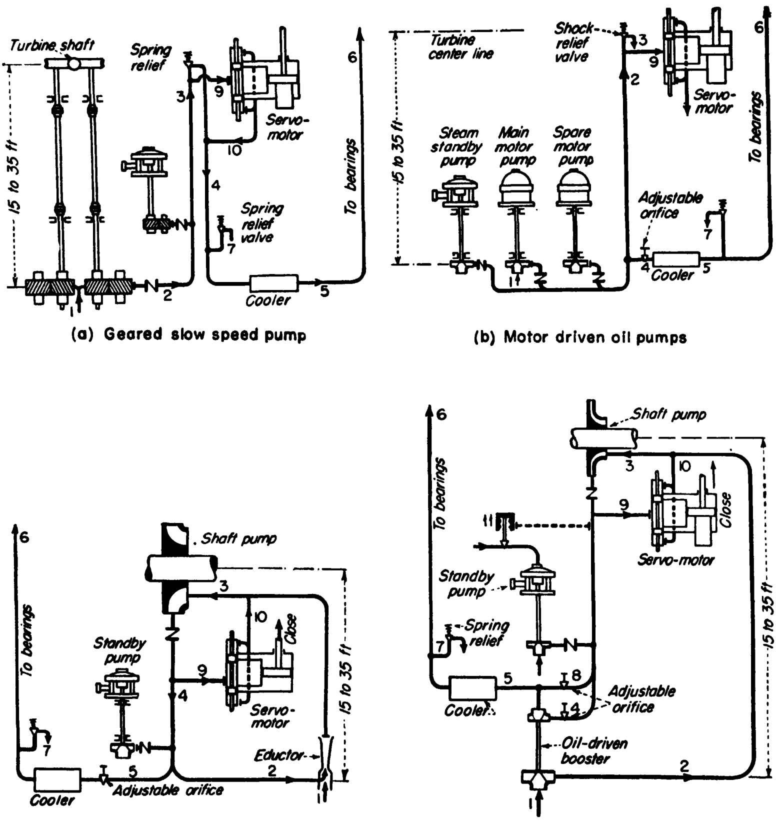
Small turbines often have ball bearings of the double-row deep-groove, or double-row self-aligning type. They are commonly mounted in separate bearing housings, as in Fig. 3-16. Lubrication is provided by an oil reservoir filled through an oilhole as indicated in the figure.



(Courtesy of Coppus Engineering Corp.)

FIG. 3-16. Typical double-row ball bearing.

Principles of Pressure Oil Systems. Oil reservoirs of modern turbines are generally located from 10 to 35 ft below the turbine, to provide proper drainage from bearings. Turbine manufacturers have grouped closely, in this locality, the accessories of the system, placing oil pumps and coolers within the tank. Centralization and removal of oil equipment from the turbine level have made possible the trim appearance of present-day large units. Locating the tank below all main and extraction steam lines also aids in reducing fire hazard from a broken oil line pouring on



(a) Geared slow speed pump

(b) Motor driven oil pumps

(c) Shaft pump with eductor

(d) Shaft pump with oil driven booster

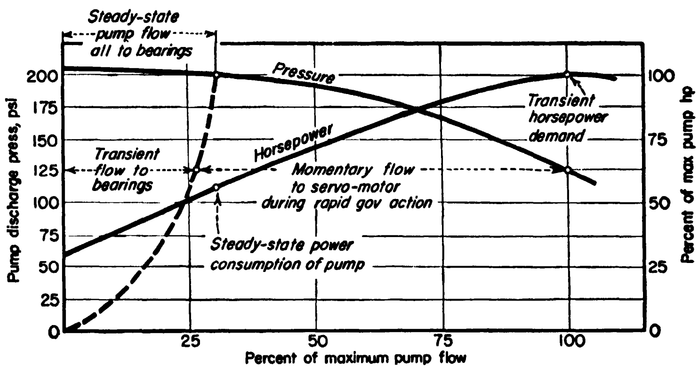
(*Courtesy of Power.*)

FIG. 3-17. Pressure oil system elements.

high-temperature piping. All oil lines are shielded by double piping to prevent oil leaks, as a pipe at about 750 F would cause oil, spilling on it, to burst into flames.

Various systems for oil circulation have been used by turbine manufacturers. The basic elements and principles of these systems are shown in the Fig. 3-17.

Figure 3-17*a* shows a system using a positive-displacement pump, driven at slow speed by a gear reducer from the turbine shaft. The long shaft rotates in an oiltight vertical duct which serves, also, as an oil drain. The main pump delivers oil into line 2-3 at about 200 psig, regulated by relief valve 3. All the pumped oil passes valve 3, but a large part is by-passed through valve 7 to the tank. The main oil pump has a capacity three times the needs of the bearings. When the servomotor acts during rapid governor action, the excess oil is required for its functioning. The system meets the needs of the servomotor and bearings, but the pumps must continually handle a large volume of oil which is only periodically needed.



(*Courtesy of Power.*)

FIG. 3-18. Centrifugal oil impeller characteristics for steady and transient turbine load conditions.

Figure 3-17*b* shows a more modern system using a motor-driven centrifugal main pump. Oil from the reservoir is pumped through line 1-4-5-6, just for bearing requirements, with a little overflow through valve 7. Pressure is maintained at 40 psig in line 5-6 by valve 4. Oil in line 2-9 is at 200 psig, and when the governor is not operating, is essentially at rest. Upon governor operation, flow occurs through line 2-9, and the increased quantity is delivered by the pump. Being of the centrifugal type, it will operate on its characteristic curve, as in Fig. 3-18, and deliver greater volume at somewhat reduced pressure but still at usable values of pressure and volume. Valve 3 is a shock valve to prevent hammer if the servomotor stops suddenly. This system has the advantage of a spare pump for maintenance requirements. Only two pressure lines need be run from the pump in the reservoir to the front bearing housing, although the servomotor line must be large in diameter.

Figure 3-17*c* shows a modern pressure system with a single-stage centrifugal impeller mounted directly on the turbine shaft. An eductor in the reservoir is used to prime the pump suction. Oil is discharged from the pump down line 4. Some of it (about one-half) is used, after metering through an adjustable orifice, for the bearings. The remainder is used for actuating the eductor. Again, during rapid governor action, the increased needs through line 9-10-3 (from about zero to several hundred gallons per minute) are accommodated by the inherent pump characteristic, as in Fig. 3-18. In starting, the steam-driven stand-by pump supplies oil until the turbine, and hence the shaft pump, are up to speed, after which the stand-by pump stops as its regulator closes the steam supply.

Figure 3-17*d*, a newer arrangement, uses a centrifugal type of pump of smaller capacity mounted on the turbine shaft. The suction of this turbine-shaft pump is supplied with oil from a booster pump, as shown through line 2-10-3, at about 10 to 20 psig. The booster pump is driven by a 1,500- to 1,800-rpm oil-turbine wheel. This wheel develops its power by being driven by part of the oil discharged from the main turbine-shaft impeller at 200 psig. The oil through the oil turbine drops from 200 to 30 or 40 psig and then passes through the oil cooler to the bearings. To develop the power needed for the booster pump, only part of the oil discharged from the main pump is required. This part is regulated by orifice 4 while the remainder by-passes through orifice 8. These orifices are adjusted to supply bearing needs and a slight, regulating flow through relief valve 7 and provide a flexible means for adjustment of suction pressure to the main oil-pump impeller to care for variations in elevations and suction-line drops. Under some conditions, flow in circuit 9-10-3 may go from zero to 500 gpm. Again the pump characteristics accommodate this change. Also, the flow in lines 1-2-3-4 and 8-5-6 is sensibly constant, even during peak flows through circuit 9-10-3. During starting, the stand-by pump, either motor- or steam-driven, supplies oil until the main impeller comes up to speed. This general arrangement of a pressure system is economical in terms of oil continuously pumped. It permits smaller oil-pipe sizes from tank to main shaft, lower circulation rates to allow greater settling time, adjustable suction pressure, mechanical simplicity, and latitude in the location of an oil reservoir. The various arrangements of pressure systems shown in these foregoing figures are used by American turbine manufacturers, and some of the principles illustrated are pictured in Figs. 3-19 to 3-22.

Pressure Circulating Systems. In larger units, the need for greater oil flow, larger heat dissipation, and a convenient governor-relaying fluid dictates the use of the oil-circulating system. Figure 3-19 shows the principal elements of such a system for one type of turbine. The main oil pump, a positive-displacement gear unit, is mounted on and driven by the turbine shaft. It draws oil from the reservoir through a strainer and check valve, and discharges it at a pressure of 40 to 50 psig. Part of the oil at this pressure flows to the governor servomotor, or power cylinder, to serve as the relay fluid. The remainder passes through a relief valve, which reduces the pressure to about 5 to 15 psig, depending upon the particular turbine and bearing design.

At the reduced pressure, the oil flows through a twin strainer arranged

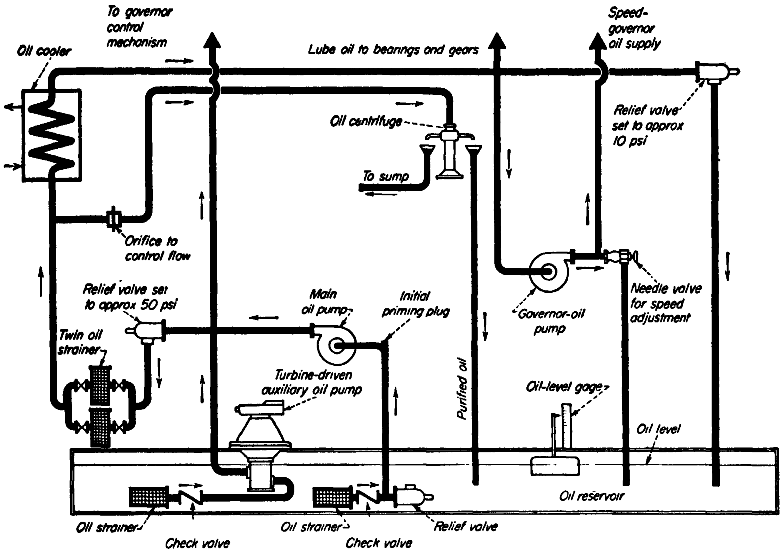
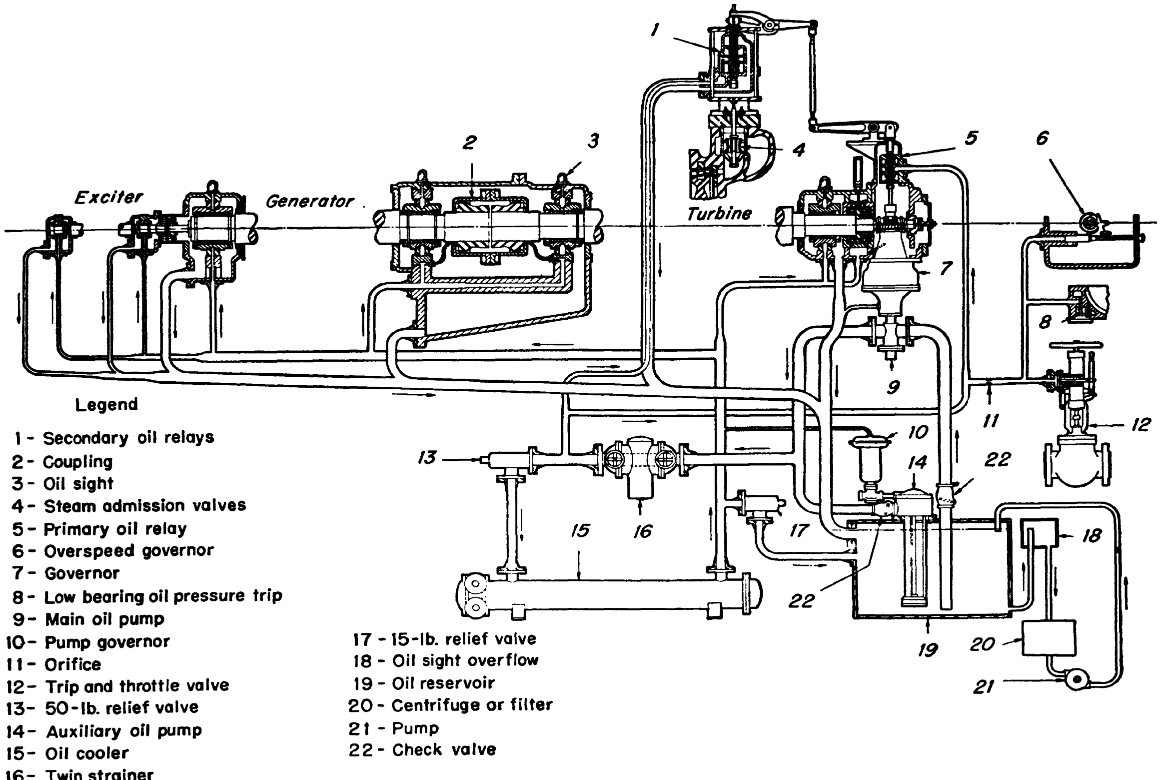


FIG. 3-19. Elements of steam-turbine circulation oil system.

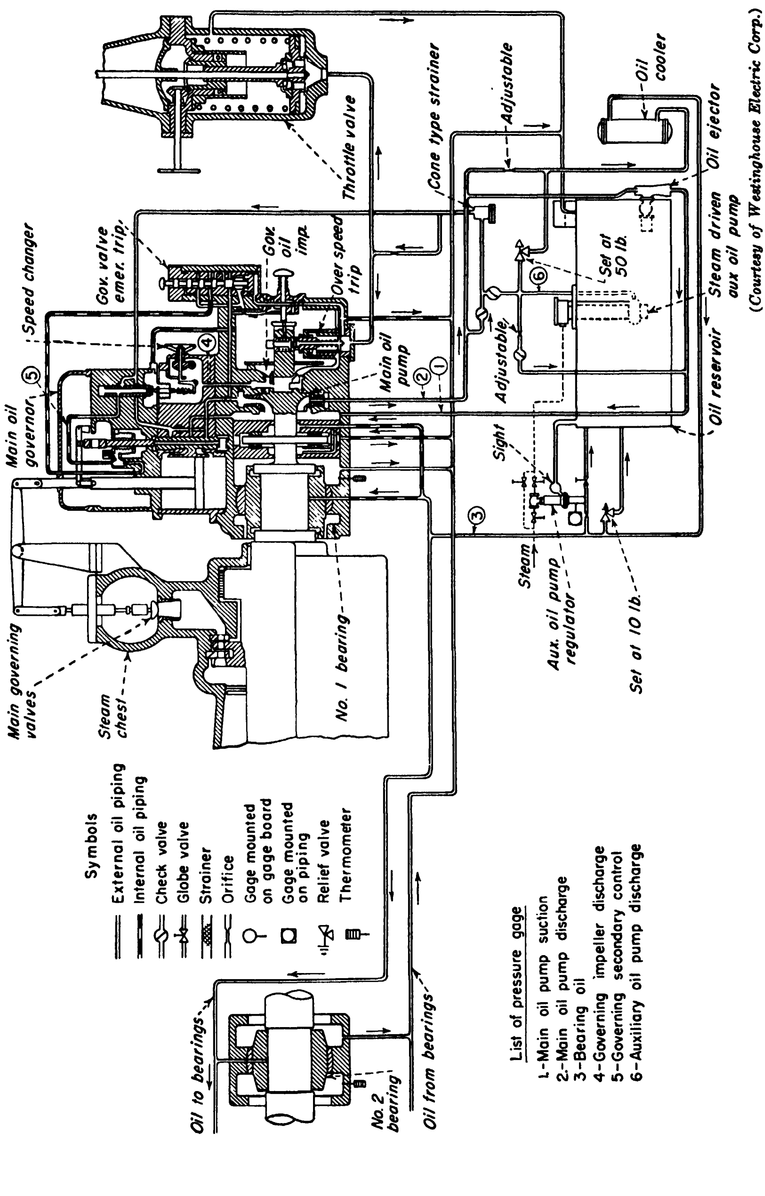
so that the screens can be cleaned without shutting down the turbine. In this continuous by-pass cleaning arrangement, part of the oil is drawn off to the purifier, before entering the oil cooler. The oil that has passed through the cooler is then led to the bearings and reduction gear, if used, and also to the pump forming the speed-responsive element of the governor. From the bearings, reducing gear, governor system, and purifier, the oil drains back, by gravity, to the reservoir, where any heavy impurities that may have been picked up will tend to settle out.

During starting and stopping operations of the turbine, when the rotor and main oil pump are not up to rated speed, a separately driven pump is needed to supply the oil. This auxiliary pump may be driven by either a small auxiliary turbine or an electric motor. During starting procedure, the pump automatically cuts out of operation when the main oil pump develops rated pressure and speed; during shutdown the auxiliary oil



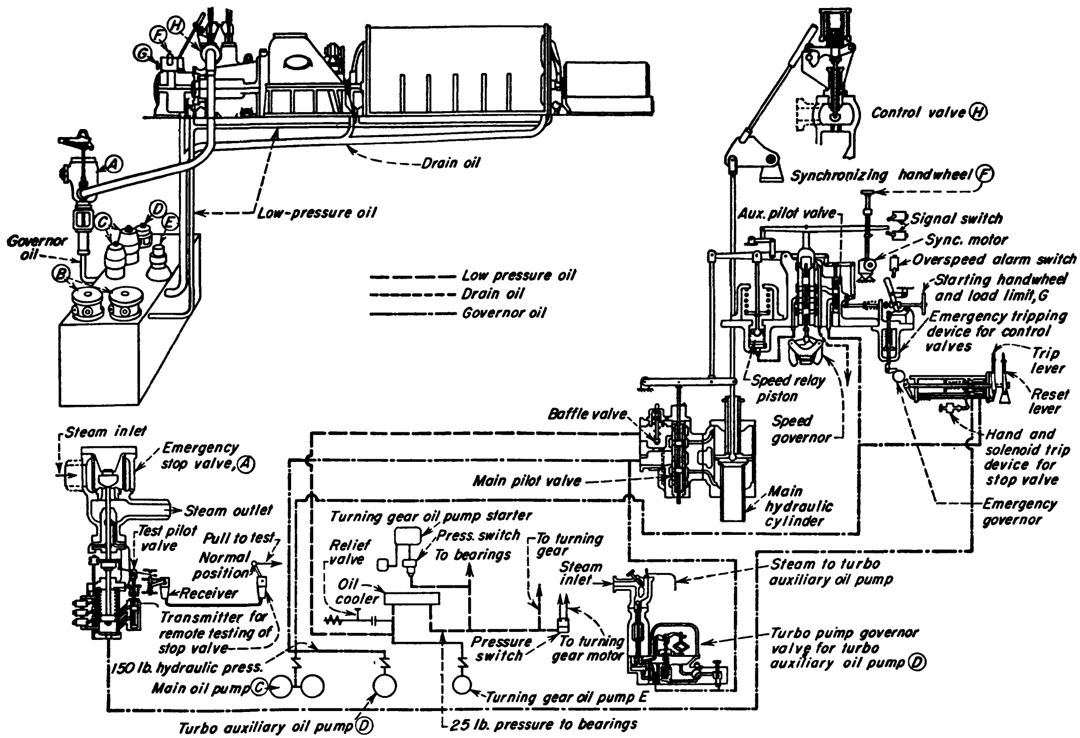
(Courtesy of Elliott Co.)

FIG. 3-20. Typical oiling system for a turbine generator unit.



(Courtesy of Westinghouse Electric Corp.)

FIG. 3-21. Details of a typical steam-turbine oiling system.



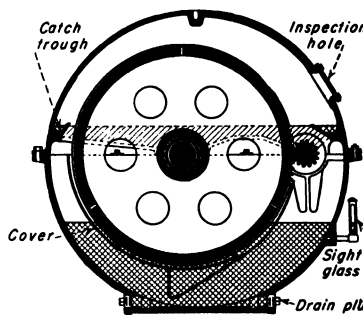
(Courtesy of General Electric Co.)

FIG. 3-22. A typical steam turbine showing details of the l-p and h-p sides of the lubrication system.

pump automatically cuts in when the main oil-pump speed and pressure drop below rated values, thus ensuring a steady oil supply while the turbine rotor is still turning.

Figure 3-20 shows diagrammatically another circulation system. High-pressure oil, at about 50 psig, is supplied for the operation of the hydraulic relays in the governor and valve control mechanism. Lower pressure oil, about 15 psig, is supplied through a pressure relief valve to the bearings. At about half normal speed, after starting, the main oil pump takes over from the auxiliary pump. Drain lines from the relays and bearings return the used oil to the reservoir. A continuous by-pass system of oil cleaning is indicated in the diagram. For added protection a low-oil-pressure trip is used. The overspeed governor, oil-operated trip, and throttle valve act in conjunction with each other. High-pressure oil passes through an orifice to the operating cylinder of the trip valve, which holds the trip latch engaged. In case of overspeed or low oil pressure, the oil in this line is discharged, causing the throttle valve to trip shut and stop the turbine.

Oil lines are commonly welded and may be formed with a core in the bearing and governor casing. Many installations obtain protection against oil fires by enclosing h-p oil piping inside l-p lines to drain back to the reservoir in case of a leak, as in Fig. 3-20.



(Courtesy of Socony-Vacuum Oil Co.)
FIG. 3-23. Splash-oil gear set.

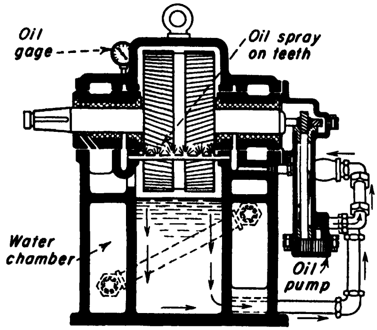
The typical lubricating systems of large turbines are further shown in Figs. 3-21 and 3-22. These indicate how normal lubrication and governing systems of turbines are interconnected.

Geared Turbines. Reduction gears are used to deliver the power from turbines to equipment that operates at less than turbine speed. These gears are usually of the herringbone type, enclosed in an oiltight casing, with babbitted bearings for the shafts. Flexible couplings usually connect the pinion and gear shafts to the turbine and driven machine. Gears are commonly lubricated by the use of

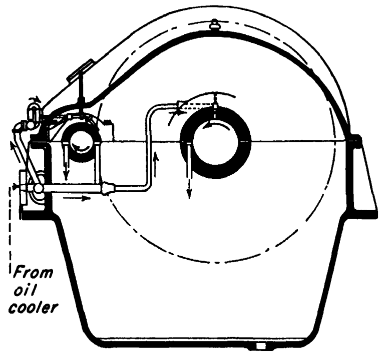
1. Self-contained splash system
2. Self-contained circulation system
3. Turbine main circulation system

In the splash system, as in Fig. 3-23, the lower half of the gear is immersed in oil, and its rotation carries the oil upward and throws it against the upper casing, where it is cooled by radiation. Troughs on the side catch the oil for gravity feed to the bearings. Enough oil adheres to the gear teeth for their lubrication. Cooling coils may be used in the reservoir or in the top of the casing.

The self-contained lubrication system, as in Fig. 3-24, has an oil reservoir in the lower housing with provision for cooling the oil by water



(Courtesy of Socony-Vacuum Oil Co.)
FIG. 3-24. Self-contained gear set.

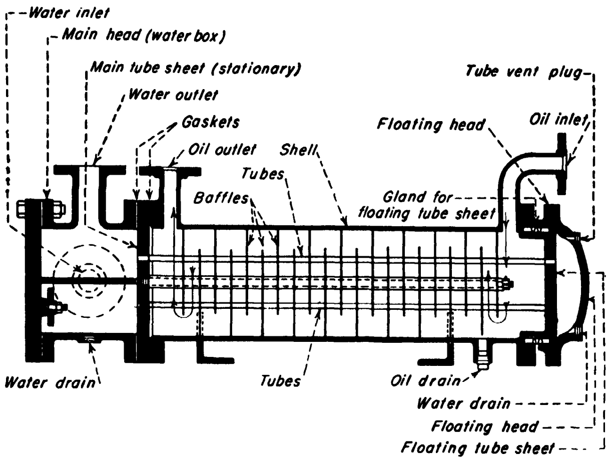


(Courtesy of Socony-Vacuum Oil Co.)
FIG. 3-25. Reduction gear set lubricated from main circulation oiling system.

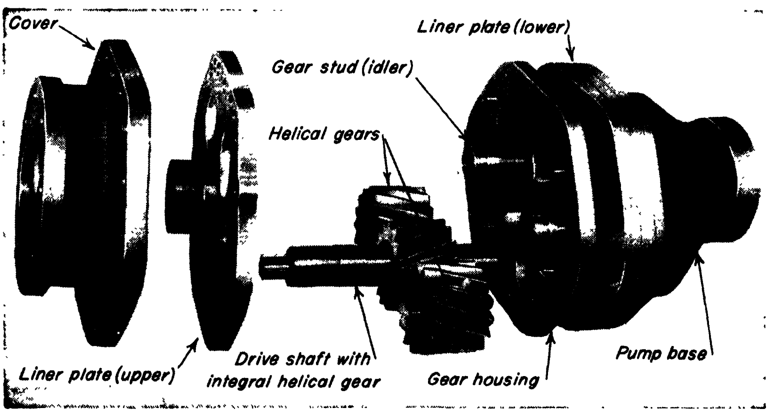
chambers or coils. A pump delivers oil to the bearings and, through spray nozzles, to the gears.

Figure 3-25 shows a gear lubricated by a line from the circulation system of the main turbine. This line leads to the bearings and to a spray for the gear. Used oil drains back to the reservoir.

Oil Coolers. Some form of oil cooling may be found in any type of lubrication system. While direct radiation, water jackets, and simple coils are common in small turbines, separate oil coolers, as in Fig. 3-26, are usual for circulation systems. They serve to remove the heat developed in bearings and so maintain low temperatures and retard oxidation of the oil. For best performance they are built as counterflow coolers. They are installed either horizontally or vertically and should not be below the reservoirs as they tend to collect excessive deposits and require frequent cleaning. The water pressure should be less than the oil pressure so that, in case of a leak, no water will enter the oil. The tubes are commonly of brass, but no copper should be used. Excessive cooling of the oil is detrimental and causes the precipitation of the soluble products of oxidation on the oil cooler tubes.

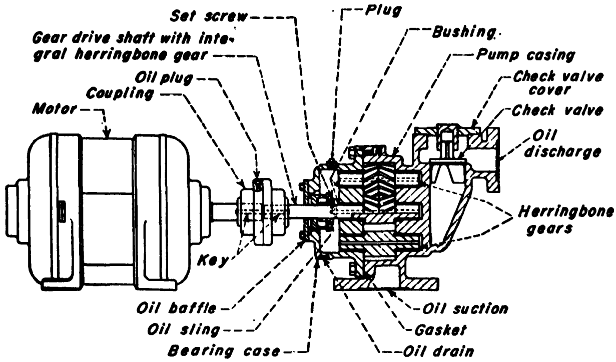


(*Courtesy of Allis-Chalmers Manufacturing Co.*)
FIG. 3-26. Oil cooler.



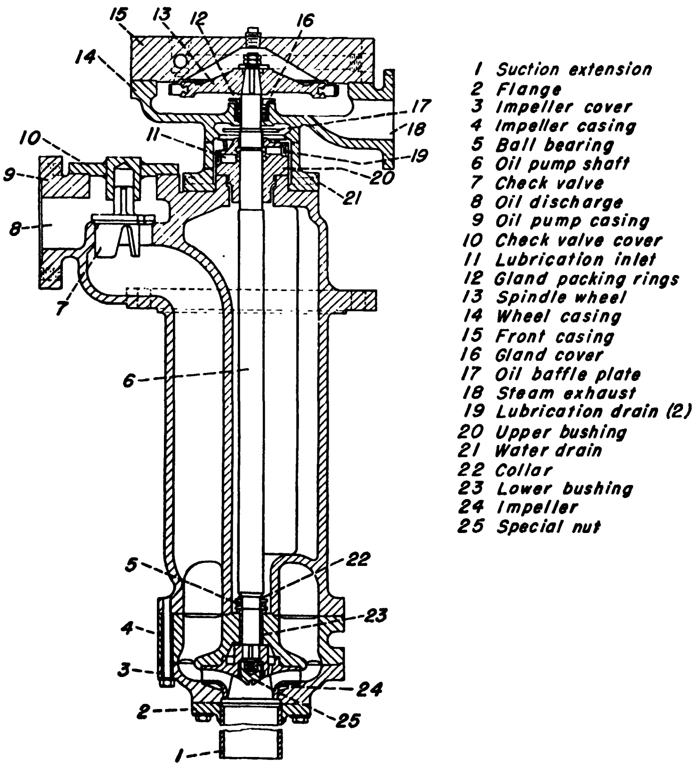
(*Courtesy of Allis-Chalmers Manufacturing Co.*)
FIG. 3-27. Gear-type oil pump.

Oil Pumps. Circulation systems are designed with main oil pumps for the bearing oil supply and for the governor relay supply. The pumps are driven from the main shaft and are commonly of the positive-displacement gear type with helical teeth for quiet operation, as in Fig. 3-27, although centrifugal types are also used. Auxiliary oil pumps for lubrication, when the turbine is not up to speed, may be of the motor-driven types, as in Fig. 3-28. Another common form is the vertical, centrifugal,



(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 3-28. Motor-driven auxiliary oil pump.



(*Courtesy of Allis-Chalmers Manufacturing Co.*)

FIG. 3-29. Turbine-driven auxiliary oil pump.

submerged type, driven by a small impulse steam turbine mounted on the same shaft, as in Fig. 3-29. The auxiliary pumps are furnished with an oil-pressure regulator for starting the pumps if the pressure from the main pumps, for any reason, falls too low. The auxiliary pumps are particularily useful during starting and stopping of the turbine.

Oil Reservoirs. The oil sumps, or reservoirs, of recent turbines are generally made of welded or riveted steel plates, and a tall narrow type is preferred, as there is less exposed metal above the oil level to rust. Older designs have cast-iron sumps with large roofs and large oil surface areas, both of which are detrimental to the oil. Sumps should be fitted with a lid to keep out dirt. The bottoms of older turbine sumps are most often flat, although the newer units have sloping bottoms to aid in removing solids and water. It is essential to be able to remove this dirt and water, and a sludge receiver, as in Fig. 3-30, is useful. This can be drained periodically for cleaning. Sediment and water also, can be pumped out of the sumps, if they are not equipped with a means for drawing off this matter. The sumps should be installed away from the turbine casing to prevent overheating and should be vented of oil vapors by an exhaust fan or some type of steam- or air-operated vapor ejector. Oil fill pipes to the reservoirs should be fitted with a bronze or brass strainer.

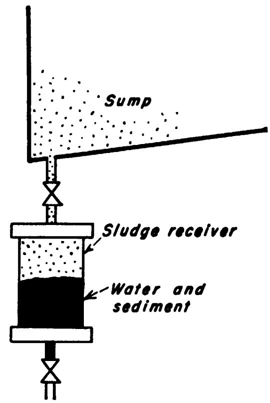


FIG. 3-30. Oil sump sludge receiver.

The interior of sumps, as well as of bearing housings, gear cases, and governor housings, is often coated with some rust preventive coating to overcome the effects of corrosion on these surfaces.

OPERATING PROCEDURES

Oil Selection. High-quality oils, prepared especially for turbine use, should be selected at all times. An oil having high resistance to oxidation and also giving good protection against rusting is desirable, as well as an oil that separates readily from water. Standard demulsibility tests on new oil are not so reliable as a check on how an oil will perform in service. This is because the oxidation characteristics of an oil, when in service, greatly offset its ability to separate from water. Aside from these properties, the choice of a turbine oil is made on the basis of its viscosity characteristics. Different turbine types require oils of certain viscosities. Manufacturers' recommendations for the different types are best followed in all cases. Table 3-1 shows the recommendations of a large turbine manufacturer.

Oil companies usually have a special line of turbine oils which can generally be matched to the manufacturer's recommendations. Table 3-2 shows the recommendations of a large oil company for steam-turbine oils.

TABLE 3-1. OIL RECOMMENDATIONS FOR VARIOUS TYPES OF SERVICE*

Properties	Forced circulation, land and marine			Ring-oiled bearings		ASTM standard method of test
	Direct-connected units	Small geared units, single reduction gears up to 500 kw†	Large geared units, including marine propulsion units‡	Ring-oiled for starting with separate oil cooler and forced circulation	Maximum bearing temperature less than 180 F§	
Saybolt viscosity sec:						
100 F	140–200	250–350	400–500	140–200	400–500	D-88
130 F	76–105	120–165	175–220	76–105	175–220	D-88
210 F	40–46	48–55	54–64	40–46	54–64	D-88
Flash point, deg F	330	350	360	330	360	D-92
Neutralization No.	0.10 max.	0.10 max.	0.10 max.	0.10 max.	0.10 max.	D-663
Steam emulsion No.	120 max.	150 max.	180 max.	120 max.	180 max.	D-157
Corrosion resistance test	Shall pass	Shall pass	Shall pass	Shall pass	Shall pass	D-665
Normal temperature of oil to bearings, deg F	100–120	100–120	100–120	100–120		
Minimum oil temperature when starting, deg F	50	70	80	50		

* *Courtesy of Allis-Chalmers Mfg. Co.*

† For marine service, where it is sometimes desirable to have only one grade of oil for both main propulsion and auxiliary units, an oil having a viscosity of 375 to 425 SSU at 100 F can be used with the understanding that power losses in bearings will be increased when higher viscosity oils are used.

‡ Oil conforming to Navy symbol 2190-T is satisfactory for this service.

§ In some installations, normal bearing temperatures may exceed 180 F due to high ambient temperature, restricted ventilation, etc. For bearing temperatures higher than 180 F, consult the manufacturer or turbine oil supplier for recommendation.

Based upon the recommendations of turbine oil suppliers and builders, the range of viscosities for turbine oils is approximately as in Table 3-3. Specific instructions from a builder are always to be preferred to these recommendations, although they do follow closely what manufacturers suggest.

Care and Supervision of a Turbine Lubrication System. A careful check on the operating conditions of lubrication systems and proper treatment of an oil are the operational means of prolonging oil life and maintaining adequate lubrication. The turbine manufacturer will

TABLE 3-2. TYPICAL TURBINE OIL SPECIFICATIONS*

	Grade No.			
	27	34	38	41
Gravity, API	31.9	31.3	30.8	29.2
Color (Saybolt or ASTM)	16(S)	16(S)	10(S)	1(ASTM)
Pour point, deg F	5	5	5	5
Flash, deg F	385	440	465	470
Fire, deg F	445	500	530	545
SUS viscosity at 100 F	156	314	414	518
SUS viscosity at 210 F	43.5	53	58.5	64
Viscosity index	97	96	96	95
Electrometric acid No	0.07	0.07	0.07	0.07
Navy emulsion test (min. for separation):				
Distilled water	12	9	8	14
Salt water	13	11	9	7
Steam emulsion No	90	160	250	250
ASTM rust test	Passes	Passes	Passes	Passes
Seawater rust test	Passes	Passes	Passes	Passes
Proposed ASTM turbine oil stability test	Over 1,000 hr			

* *Courtesy of Shell Oil Co.*

The usual recommendations for the grades listed above are as follows:

Grade No. 27: This grade is recommended for direct-connected sets equipped with oil-circulating systems.

Grade No. 34: This grade is generally recommended for sets equipped with oil-circulating systems and single reduction gears. It is also widely used in direct-connected marine turboelectric propulsion sets.

Grade No. 38: This grade is recommended for larger sets equipped with oil-circulating systems and double reduction gears. It is almost universally used in geared marine propulsion turbines and conforms to Navy symbol 2190-T turbine oil which is used in all marine turbines.

Grade No. 41: Turbine manufacturers' recommendations for ring-oiled turbines vary widely; however, Grade No. 41 is used in many such units operating at normal bearing temperatures.

TABLE 3-3. RECOMMENDED TURBINE-OIL VISCOSITIES

Operating Conditions	*Saybolt Viscosity at 100 F*
Direct-connected turbines:	
Circulating system	140–250
Ring-oiled bearings,	
Temperatures below 180 F	150–500
Temperatures above 180 F	600–2000
Geared turbines:	
Circulating system supplying both gear and turbine	250–500
Separate lube system for gears	300–600

furnish instructions for the care of a turbine. In general, the following basic procedures are included in all such recommendations:

1. Practice a regular method of purifying oil in service by whatever oil-purification or treatment system is used. Maintain treatment equipment in good order, for example, by cleaning or changing filters, and cleaning centrifuges and precipitation tanks.
2. Clean oil strainers regularly, usually daily.
3. Drain the bottom of the oil reservoir every day to remove water and other contaminants.
4. Check on water leakage by inspecting the turbine steam and water glands, and oil cooler.
5. Check on oil leakage by inspecting the bearing seals and oil piping.
6. Follow the manufacturer's recommendations for periodic cleaning and overhaul, as the frequency varies widely in different plants with service conditions. Yearly cleaning is generally good practice for average turbines. Drain the system as soon as possible after shutdown, as the warm oil carries away the bulk of the dirt. When possible, disconnect and clean all piping, using clean dry linen rags. Flushing with water under pressure, dragging rags through, or using cleaning fluids such as kerosene or carbon tetrachloride, is common practice. Cleaning fluids should be used only on parts that are accessible. Bearings, pedestals, reservoirs, and all oil grooves should be cleaned, also the hydraulic governor system. The oil cooler should be taken apart, and the tubes and shell cleaned with hot water or an air hose. Good practice, when these means do not remove the deposits, is to boil the tube bundle in a cleaning compound. Commonly used is a 50-50 soda ash and trisodium phosphate compound, 5 parts of water to 1 of compound. The entire system should be thoroughly cleaned of any compound as the oil charge may be affected by it.

Check on oil pumps and possible deposits on inlet and outlet. When starting, charge the system with oil so that it can be circulated by the auxiliary pump for about 6 hr with the turbine not running. Drain and clean the oil which can then be used for make-up. Remove all dirt from reservoir bottom and strainers. Recharge with oil, and the turbine is ready for service.

Various solvents are available for turbine cleaning. They reduce the amount of manual cleaning and dismantling usually done. The oil is drained from a system, and solvent equal to about a half-full charge is added and circulated. Small-mesh screens are used, instead of the standard mesh, and the sediment loosened by the solvent is screened out. The system is subsequently flushed with regular oil, drained, and refilled. A certain amount of hand cleaning of the tank, sump, cooler, and gear housing is necessary with these cleaners.

7. Check the oil level daily, and keep a record of the type and quantity of make-up oil added. Reservoirs of ring-oiled bearings should be drained monthly, and the oil replaced by new or clean oil. In circulation systems, users of inhibited oils should secure a brand of oil for make-up, similar to that in the system.

8. In starting, follow manufacturer's recommendations, but check on whether the main oil pump is functioning and whether the correct oil pressure is developed. When the temperature of oil leaving the bearings reaches 110 to 120 F, start the water circulating to the coolers and allow the oil to rise quickly to an operating temperature of about 130 to 140 F. By so doing any trend toward foaming is decreased.

9. Oil sampling: (1) Take a weekly sample of about 4 oz of the oil leaving the bearings. A visual check of these samples for color change and impurities may reveal trouble. (2) Take a sample of about 1 qt of oil periodically (about every 2,000 hr of operation) for analysis by the oil company or other laboratory.

10. Keep a daily log of bearing-oil pressure and oil pressure to the governor mechanism.

11. Keep a daily log of water temperatures (1) entering the cooler and (2) leaving the cooler. A decrease from normal temperature difference of these indicates sludge or water scale in the cooler.

12. Keep a daily log of oil temperatures (1) entering the bearings, (2) leaving the bearings, (3) entering the cooler, (4) leaving the cooler, (5) entering the gear case, (6) leaving the gear case.

13. Keep a log of all maintenance items such as (1) overhauls, (2) oil changes, (3) oil additions, (4) filter cleanings, (5) centrifuge cleanings.

STARTING A NEW TURBINE

Before a turbine is put into continuous service, it is necessary to make certain that the lubrication system is absolutely clean. Manufacturers' instructions are to be followed for starting. The following are recommended as a guide to correct procedure:

1. Inspect and clean, with compressed air, all oil and water piping. Check loose oil and water joints, and use water under full pressure.

2. Add a small quantity of oil to the reservoir, and circulate with the auxiliary oil pump. For new turbines, the water content of the oil used should be very small, to prevent corrosion of the new metal surfaces. If the oil used is not rust-inhibited, a 10 per cent addition of a well-used oil of the same type should be added to the new oil charge, with the oil supplier's approval. The oil lines can be pounded during this time to loosen any pipe scale or rust. After about an hour's operation of this flushing, the system should be drained and cleaned. The drained oil can be purified and used for make-up.

3. Fill the system, and circulate the oil for about 8 hr with the turbine idle, and check on oil pressures, level, and gauges. It is good practice to add oil through a strainer, and be sure that dirt, which commonly adheres to the outside of oil barrels, does not get in. Allow the oil to stand for about 10 hr and drain the dirt from the reservoir, after which make-up is added. Lubricate all turbine parts as may be required on the governor and steam-admission valve.

4. Bring the turbine up to speed, and check oil pumps, gauges, oil level, bearings, and all related parts. Start the cooling water when bearing oil reaches about 110 to 120 F, bring quickly to an operating temperature of 130 to 140 F, and continue checking water and oil temperatures. Clean the main reservoir strainers repeatedly during this period.

5. After a service period of some 300 hr, unless the turbine has a continuous by-pass filtration system, the system should be drained. The drained oil can be used for make-up after it has been cleaned. Before recharging, clean the reservoir cooler, and bearing pedestals.

CHAPTER 4

OPERATION AND MAINTENANCE

We have talked about all the principal parts of the turbine, their details, and assembly. By understanding the construction, we gain some idea of the limitations involved in operating a turbine. Fundamentally, it is built to run under certain operating conditions: steam flows through nozzles and blades at certain velocities and angles; the clearance between stationary and moving parts is large enough to avoid mechanical interference but small enough to minimize leakage; the parts are strong enough to withstand forces developed by steam pressures and centrifugal motions; rotating parts can turn freely without binding, working only against the resistance put up by the load; no foreign matter is allowed to enter any part of the mechanism.

While the turbine is designed to do a job with certain conditions prevailing, the problem immediately arises of just how to bring it from a cold standstill condition to a hot running condition. Proper methods of doing this were developed over a long period of experience. Designers found that they were burdened with the added considerations of starting and stopping the machine *safely*, as well as working out "bugs" that appeared unexpectedly at different development phases, as units were built for larger capacities and higher steam conditions. So, when looking over operating instruction sheets for a unit, follow them to the letter. You may be sure, not to do so will result in disaster for the machine. Many of the rules were formulated from sad experiences in the past. Most power-producing machines cost thousands of dollars, and, unfortunately, they can be reduced to scrap metal, or at best, seriously damaged by a momentary lapse in following the rules.

Before running through a sample set of starting and stopping rules, let's talk about some of the conditions that must be guarded against in turbine operation. The designer does all he can to minimize or eliminate the following effects, but in the end he must depend on the intelligence of the operator to appreciate what causes these effects and what to do when they appear. The effects are

1. Unequal expansion
2. Vibrations
3. Congestions

4. Lack of lubrication
5. Overspeeding

Unequal Expansion. Close clearances between stationary and moving parts characterize turbine construction. These, coupled with high speeds, make the hazards of misalignment an ever-present threat to machine safety. We all know that metals change their dimensions with change in temperature; *i.e.*, they expand as temperatures increase and contract as temperatures decrease; at least, the ones used in turbine construction do. Consider then that, when a turbine is idle for any length of time, all its parts will be at room temperature (if the steam valves are tight); when it is brought into operation, the front end of the turbine (Fig. 4-1) will be heated far above room temperature. In contrast, the rear end warms slightly. The heating of the turbine is, of course, done by the steam. For some units, temperatures range from red hot, at the front end, to that of a warm rain at the exhaust.

Clearances must be maintained while the turbine parts enlarge their dimensions with temperature increase. Dimensional changes are further complicated by the steam temperature dropping as the steam flows from inlet to exhaust. Temperatures along the turbine length will change with load variation. After the first-stage nozzles, temperatures rise as the turbine load increases.

Take a look back through the turbine cross sections shown in the preceding chapters and at Fig. 4-1, and it will be seen that the rotor of any turbine is completely surrounded by the steam, except the shaft ends passing through the seals and bearings. On the other hand, the casing interior is subjected to steam temperature, while its outside surface is exposed to room temperature. However, this condition is largely corrected, at the high-temperature end, by the lagging usually wrapped around the casing to reduce heat radiation. Normally, this will keep casing temperatures nearly alike on inside and outside surfaces. On the average, casing temperatures are likely to be somewhat lower than rotor temperatures during operation because of heat flow to atmosphere.

While blades and nozzles are kept in close proximity, the critical clearances ordinarily occur between moving and stationary strips in the seals, and sealing strips at blade tips and on dummies. These strips are made with thin edges, in order to wear quickly if physical contact should be made between stationary and moving parts. Rubbing of such parts generates heat at the point of contact by friction, which aggravates the expansion of the rubbing parts until they are worn down.

Such accidental rubbing obviously must be caused by warping or unequal expansion of rotor and casing elements. Designers must control the shape and thicknesses of all parts, so that they will tend to expand or contract equally in the same direction with changing steam temperature. To ensure such equal motion, manufacturers universally recommend that temperature changes be made gradually, to ensure uniform heating of all metal. Sudden temperature changes are referred to as "thermal shocks"

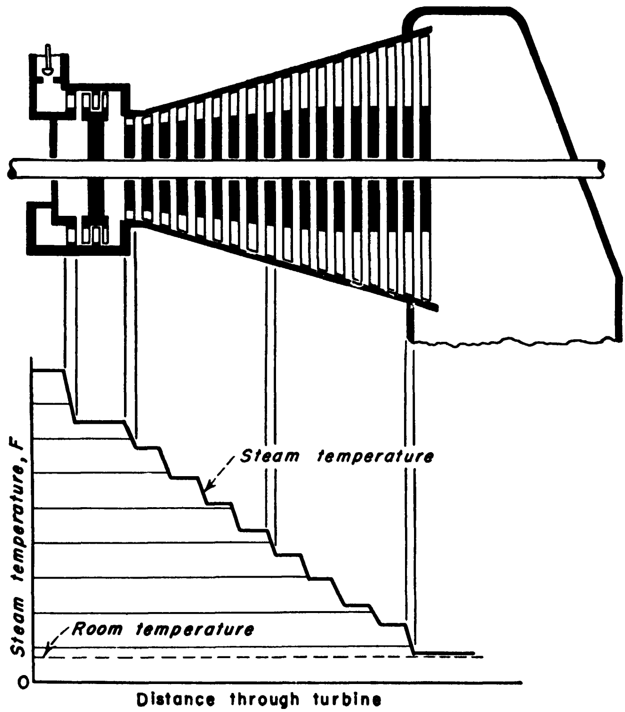


FIG. 4-1. Turbine casings and rotors are heated by the temperature of the steam passing through the unit. Hence each section of these elements will have a different temperature, because steam cools as it gives up energy in the succeeding stages.

and will cause sudden unequal expansions, usually of a momentary duration.

Turbines are usually anchored to the foundation at their exhaust end and allowed to expand toward the inlet end, as they heat up. This is the only practicable arrangement, since the driven apparatus requires the coupling to stay in a constant location. The front end of the turbine, normally supported on a flat plate, slides on the plate as the unit expands and contracts with heating and cooling during starting and stopping operations. Steam pipe connections to the turbine are always made through long easy bends that have sufficient flexibility not to place any undue strain on the connecting flange as the turbine dimensions change with heating.

Great difficulty generally arises in trying to keep h-p steam valves at the head end of the turbine absolutely tight. In most cases a small amount of steam leaks into the turbine casing when the unit is shut down. Such steam tends to remain in the upper part of the casing, resulting in heating that portion of the rotor and casing. Needless to say, this causes unequal expansion of upper and lower halves of the machine elements. Besides creating the danger of rubs, such a condition causes an unbalance of the rotor, which can cause dangerous vibration of the spindle as it is brought up to speed.

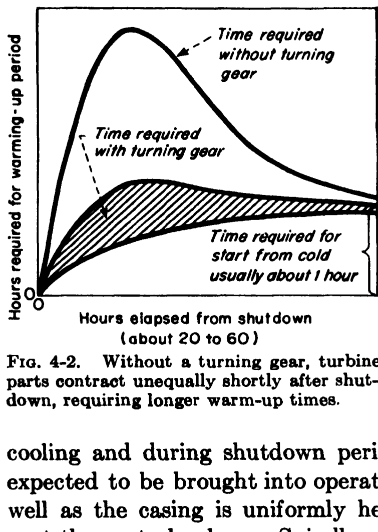


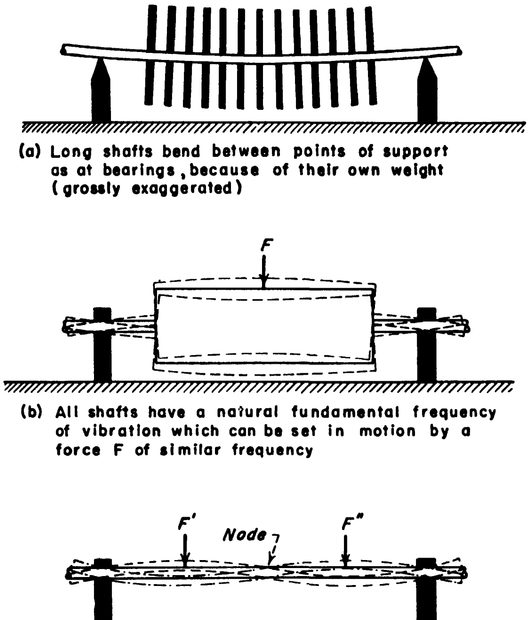
Fig. 4-2. Without a turning gear, turbine parts contract unequally shortly after shutdown, requiring longer warm-up times.

To prevent this condition, large turbines are now fitted with turning gears. These consist essentially of a large gear built into the turbine shaft near the coupling end driven by an electric motor through a suitable reducing gear arrangement. The motor usually mounts on the turbine exhaust hood and drives the turbine spindle at speeds of 1½ to 3 rpm, while warming and cooling and during shutdown periods of several days when the unit is expected to be brought into operation. By such rotation, the spindle as well as the casing is uniformly heated by any leakage steam that gets past the control valves. Spindle motion distributes the steam uniformly throughout all parts of the casing, prevents the spindle from becoming bowed, and consequently allows more rapid starting.

When the spindle is revolved at this low speed, only l-p oil need be supplied to the bearings, at about half the normal operating flow rate. This supplies sufficient lubricant to prevent any damage to bearings and journals.

Besides eliminating mechanical difficulties of unbalance, the turning gear proves economical by saving on the amount of starting steam needed because of the shorter starting periods. Figure 4-2 shows approximate relations between shutdown hours and warm-up hours for turbines with and without turning gears.

Vibrations. Theoretically, the turbine rotor, being a perfectly homogeneous substance, should be able to rotate at any speed, up to the limit of its strength to resist centrifugal forces, without vibrating. However, metals are not homogeneous, *i.e.*, uniform throughout their mass. Also it is necessary to cut slots, boltholes, and the like for constructional needs. In addition, all materials exhibit a certain amount of flexibility, and when supported at two points, as at the ends of a turbine spindle, will bend of their own weight (Fig. 4-3*a*).



(a) Long shafts bend between points of support as at bearings, because of their own weight (grossly exaggerated)

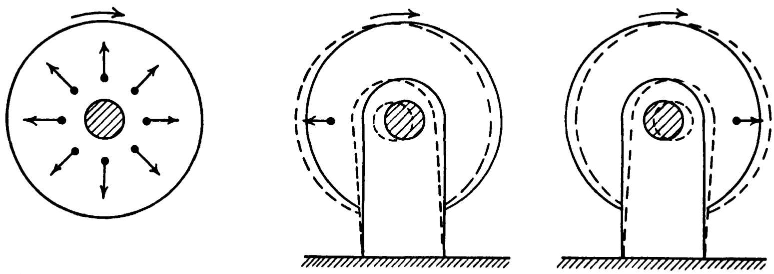
(b) All shafts have a natural fundamental frequency of vibration which can be set in motion by a force F of similar frequency

(c) All shafts also have natural frequencies double (above), triple, quadruple, etc. of the fundamental frequency (see (b)) which can be initiated by properly applied force F' or F"

FIG. 4-3. Turbine rotors cannot retain their designed shape because of the force of gravity and other vibrating forces that may act on them.

All flexible structures will vibrate at a certain frequency, depending on their own mass, its distribution between points of support, and its temperature. All that's needed to start these vibrations is the application of a force that varies in strength at the same frequency. If the turbine drum in Fig. 4-3*b* has a force *F*, varying with a frequency the same as its own fundamental frequency, pushing on it, it will vibrate as shown by the dotted lines. The extent of the vibration is grossly exaggerated in the figure. A spindle with disks, as in Fig. 4-3*a*, will also vibrate under a similar influence.

There is a whole series of natural frequencies for all flexible structures and turbine spindles or rotors. They are even multiples of the fundamental or lowest frequency. Figure 4-3*c* shows how a shaft vibrates with double its fundamental frequency under application of *either* a force F' or a force F'', varying at double the fundamental frequency. A point with no vibration appears midway between the points of support and is called a "node." When the left half bends downward, the right half bends upward, and vice versa. At triple the fundamental frequency, two nodes appear between the end points of support, and three vibrating loops.



(a) Perfectly balanced rotor has exactly balanced centripetal forces developed by rotation

(b) Rotor with residual unbalanced centripetal force causes bearings to vibrate as well as rotor itself. At critical speeds vibrations become violent, can wreck rotor

FIG. 4-4. Slight unbalance in a turbine rotor can often be tolerated if the machine does not operate near one of its critical speeds.

All these vibrations can be induced, while the rotor is at rest, under the action of the proper vibrating force. However, this vibration problem does not arise with the turbine at rest, since no forces then act on the rotor. The serious aspect of vibration arises when the rotor is spinning. When any object is spinning, centrifugal forces act on all moving portions because of the motion. If a rotor is *perfectly* balanced, the centrifugal forces, acting, will balance each other, and rotation proceeds smoothly. However, such *perfect* balance is practically impossible to achieve. Consequently, the centrifugal (sometimes called "centripetal") forces do not balance each other. They leave a small net force acting somewhere on the rotor at right angles to the direction of rotation, *i.e.*, in an outward radial direction, as shown in Fig. 4-4*b*. This unbalanced force pulls the rotor shaft and structure out of alignment and prevents it from turning about its true center. This force can be counterbalanced only by the bearings, which will be vibrated by the continuously changing direction of the force. This vibration, in turn, will be communicated to the rotor. The vibration can always be minimized to tolerable limits by placing balancing weights on the rotor periphery. However, no matter how small this unbalanced force, it becomes dangerous when the speed of rotation is such that the unbalanced force passes a given point in the bearing with a frequency equal to the natural fundamental frequency of the spinning rotor. This speed is known as the *critical speed.* If the rotor is kept turning at the critical speed, it will vibrate violently and eventually fail by breaking at the weakest point. The second critical speed occurs at twice the rpm of the fundamental, the third critical speed at three times the fundamental, etc.

Obviously, operation at any of the critical speeds must be avoided. Modern turbine rotors are designed to operate normally between their fundamental and second critical speeds. Then, in starting, it becomes important to pass through the critical speed rapidly to avoid building up vibrations. At first thought, it might appear wiser to operate turbines entirely below the critical speed, but this would require such heavy rotor construction, to achieve the necessary stiffness, as to make the machine unduly expensive and bulky.

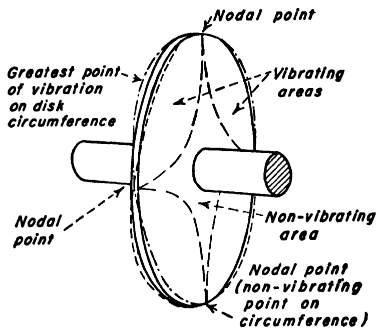


FIG. 4-5. Turbine disks also have critical speeds and vibrate with different frequencies at their edges. Disks must be designed to run at speeds well away from their own critical speeds.

Another form of vibration that may appear in turbines is that of the disk edges. This can be caused by the force of a steam jet leaving a nozzle, passing through the blades on the edge of the disk as it turns past one location in the casing. Again these vibrations depend on the dimensions and weights of the disks and are avoided by proper design. Figure 4-5 shows how these vibrations act for the lowest natural frequency of a disk. No vibration takes place at the nodal points on the periphery of the disk, or on the dotted lines connecting these points on the surface of the disk, or in the area defined by the shaft and these dotted lines. The dash and dot-dash lines on the periphery show alternate positions of the periphery's center line, as it vibrates from one extreme to the other. The edge of the disk at any one instant will have a wavy appearance for the grossly exaggerated condition shown. Natural frequencies exist at higher levels such that 6, 8, 10 nodal points will be established. Sustained running at any speed that will excite these natural frequencies will cause violent vibrations of the disk with disastrous results.

Unshrouded and unlashed blades will have natural frequencies under which they can assume detrimental vibrations, as shown in Fig. 4-6. These vibrations can be entirely independent of the disk characteristics. They are usually caused by a steam jet passing through and stressing the blade for only part of its travel through one revolution of the disk. Successive passes through the steam jet (under proper timing) can sustain these vibrations on a violent scale until the blades fail. On some h-p units the first-stage blading has lived only 5 hr under such conditions. The condition is corrected by "tuning" the blades, *i.e.*, by changing their shape and weight so that their natural frequencies will not be within the operating range of the turbine.

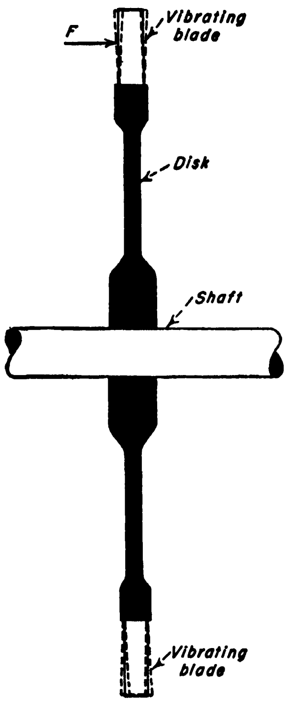


FIG. 4-6. Blades can vibrate under the impact of steam passing through them. They should be built to avoid operation near their own natural frequency.

Other forms of vibration are possible in flexible structures, such as turbine rotors, but the foregoing are those that have caused trouble in the past and are guarded against in modern designs. Instant torsional vibrations are a possibility, but no trouble ever has been reported from this phenomenon. In torsional vibration the shaft twists in one direction, untwists, and then twists in the other direction while it rotates. Since the forces acting on a turbine rotor, as developed by the steam jets, are all continuous and essentially in the same direction, this trouble has not arisen. There have been other troubles such as an oscillating rotor that vibrated axially, but these are of unusual character.

Congestions. There are three ways in which steam passages in a turbine may be blocked: (1) by carryover of solid matter in the steam from the boiler, (2) by accumulation of condensate, (3) by foreign objects inadvertently left in the turbine or steam mains. In the majority of power plants it has been necessary to treat the feed water to the boiler chemically in order to prevent formation of scale on the boiler interior and in order to prevent corrosion. As steam bubbles break through the water surface in the boiler drum, they carry with them minute droplets of water which contain these dissolved chemicals. These droplets evaporate in the superheater, and the chemicals deposit on the valves, nozzles, and blades of the turbine, reducing the area available for steam flow. They also change the shape of the nozzle and blade passages. These two effects reduce both the capacity and the efficiency of the unit.

In present-day plants elaborate methods of washing and separation are used to eliminate the water droplets in steam leaving the boiler. These methods reduce the water content of the steam by 99.9 per cent, but the remaining 0.1 per cent will carry over enough chemicals to form undesirable deposits in the turbine, over a period of time. Measures must be taken to clean the turbine periodically. We shall talk about these later.

Ordinarily, condensate accumulation can occur only when the turbine is shut down. It will build up in the steam main to the turbine, in the steam chests, and in the bottom of turbine casings. When a unit is started up, all the drain lines must be left open a sufficient length of time until all parts are uniformly heated and all water has been removed. A slug of water picked up by a high-velocity steam jet can seriously damage mains, valves, nozzles, and blading.

The careless workman who leaves nuts, bolts, gloves, pencils, and other objects in steam mains and turbines that have been opened for inspection or repair cannot be condemned too strongly. However, there are enough incidents of this sort on record to make it mandatory to inspect thoroughly all equipment before finally assembling it. Steam strainers installed before the turbine stop valve help to guard the turbine against anything left in boiler or main piping, but no such safeguard can be provided beyond this point.

Lack of Lubrication. One of the outstanding advantages of turbines is their relatively small consumption of lubricating oil. However, as small as this demand is, without oil the turbine quickly becomes inoperative. By lack of lubrication we mean not only the supply of sufficient oil but also oil of the proper characteristics. An improper selection of oil can be as fatal as no oil at all. Operators should seek competent advice from lubricating engineers and turbine manufacturers in this respect and follow their advice religiously.

The importance of the lubricating system is high-lighted by the recommended procedure for starting one on a new unit. These recommendations, briefed below, were prepared by a committee of experts of the ASTM.

When the new turbine is in place, see that all rust-preventive coatings in piping and tanks, as well as dirt, are *completely* removed. Check for any peculiarities such as air pockets or poor drainage arrangements which may cause trouble. Wire brushing, blasting, and sweeping are often necessary to remove welding shot, chips, sand, etc. Always use lintless cloths for wiping. Lint accumulations have plugged oil lines and filters.

Next, make provisions to prevent flushing oil from entering the bearings, either by blanking off at flanges and providing by-passes around the bearings, or in small units, by removing the bearing shells or rotating them and leaving a passage for oil flow.

Depending on the system size and layout and the capacity of the circulating pump, it may be best to sectionalize piping and flush each section separately. One function of flushing procedure is to remove any rust preventive present. If this coating does not dissolve in turbine oil, it should be removed completely by suitable solvent before assembling the piping. When this coating is oil-soluble, use a flushing oil of the same or lower viscosity than the recommended lubricating oil. Use an oil with the same type of rust inhibitor as the final oil charge. Use a sufficient amount to ensure continuous circulation with the auxiliary oil pump.

The circulating flushing oil should be heated to 125 to 180 F. Heating above the normal operating temperature expands the piping to a greater degree than when in service. This loosens scale and other adhering matter. Oil may be heated by (1) steam at 3 to 5 psig fed to the water side of the main lubricating-oil cooler, (2) a temporary heating coil in the lube-oil sump tank, (3) a temporary heater in the piping system.

Be sure to use a filter or centrifuge continuously during the flushing operation. Continue flushing hot oil through the system from 4 hr for small units up to 96 hr or longer for larger turbines. Then drain all flushing oil from the system completely. Remove heavy solids that have flushed into the oil reservoir. Inspect and clean the bearing pedestals and governor hydraulic mechanism.

On small systems and those where no rust-preventive materials have been used, the flushing charge may be the same type of oil as that used later for lubrication. To remove all flushing oil and contaminants, it may be desirable to follow flushing with a charge of displacement oil. Follow the recommendations of the oil supplier and turbine maker. Displacement oil should be of the same type as the recommended lubricant, heated to 130 to 150 F, and circulated for about 2 hr. Since viscosity of the oil is often higher than the flushing oil, some additional contaminants may be suspended in the oil and flushed out.

Inspection of the centrifuge or filter will tell if the system is clean. If so, connect in the bearings and continue the circulation for an additional hour or two. During this period, operate the governor mechanism to flush it completely.

Next, remove and inspect all accessible bearings. If they are in satisfactory condition, the system may be considered clean. During drainage of the displacement oil, give attention to all the low points in the system. Wipe out the oil reservoir with lintless cloths and remove any loose solids.

Care at this point will save time and trouble when the turbine is in service.

Overspeeding. An ever-present danger to a turbine in operation is the sudden loss of load. Under such circumstances, if steam flow stays constant, the turbine will quickly overspeed. Speed would continue increasing until the turbine losses equaled the energy put in by the steam. Long before this point would be reached, however, the centrifugal forces would exceed the stresses that the rotor could withstand, with the result that the rotor would be torn apart, literally explode, with great danger to everything in the vicinity.

Since the centrifugal force varies as the square of rotational speed, this force doubles with only a 41 per cent increase in rpm; at double the speed, the force quadruples. Shaft speed accelerates very quickly, so the turbine's overspeed trip on the throttle valve must act within a very short time. For this reason, the overspeed trip should be tested regularly and at every opportunity to ensure its being in proper operating condition.

Turbine Instruments. To supervise properly the operation of a turbine, instruments must be provided to indicate the condition of the steam, oil, and turbine at the various locations. For convenient and fast control the instruments are all mounted on a common panel in so far as conditions permit. Depending on individual preferences, instruments may be provided in the various forms as indicators, recorders, or integrators, or any combination of the three.

Pressure gauges are needed to indicate the state of the steam at various points in its path through the turbine and past the valves. Individual gauges are usually provided for pressure at the stop valve, pressure in the steam chest, in the first stage, at any following stages that receive steam directly from control or by-pass valves, at stages supplying bleed and extraction points, at turbine exhaust (vacuum gauge), at air-ejector connection in the condenser. Of equal importance is information on oil pressure to the bearings and governor system. Pressure gauges also measure conditions of steam and water to gland seals. If a vacuum gauge is used at exhaust, a barometer should be provided for calculation of absolute pressure.

Thermometers are needed to determine the temperature of the steam in the various points of the turbine circuit. They are usually provided at the throttle, the bleed or extraction points, the exhaust, and the hot well. Oil temperatures are required at the bearing outlet and the oil-cooler inlet and outlet.

If the turbine drives a generator, a wattmeter with accompanying volt and ammeters are usually provided at the turbine control board. These meters, together with a flowmeter for either or both the throttle steam flow and the condensate or feed-water flow permit the checking of steam-rate and heat-rate performance of the unit.

A tachometer is required to indicate the speed of the turbine rotor, especially during warm-up periods.

On the larger turbines there is an increasing application of shaft eccentricity, bearing vibration, cylinder axial expansion, and noise

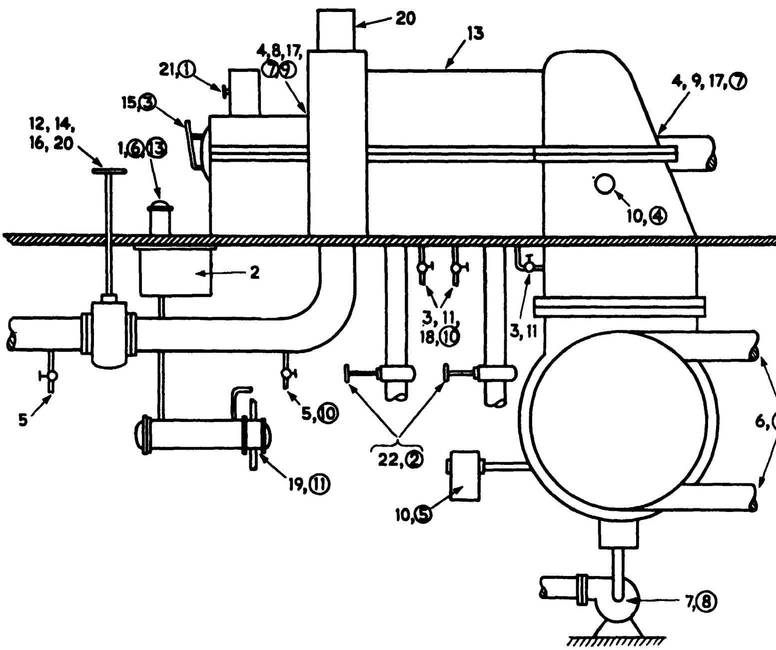


FIG. 4-7. A hypothetical turbine showing the points needing attention during starting and shutdown procedures. Numbers in circles refer to shutting down, others to starting.

meters. The latter are used to detect unusual noises caused by the rubbing of blades or glands. These meters are of especial value during warm-up periods. A watchful study of the records from these meters often detects conditions that may lead to trouble. To aid in efficient loading, there is often applied a system of signals that indicate governing-valve positions.

Turbine Operation. All manufacturers supply detailed instruction sheets for the operation of the turbines they build and install. It is important that operators follow these directions minutely. Although turbines are inherently rugged in their construction, they are delicate in the very small clearances that must be maintained between stationary and moving parts despite the large variation of temperatures they experience during starting operations. Figure 4-7 and the tables below illustrate in a general way the operating procedures in starting and shutting down a condensing turbine. The exact order of the steps shown varies with the manufacturer and the particular turbine.

Turbine Starting Procedure

1. Start auxiliary oil pump and check oil pressure.
2. Check level in oil reservoir.
3. Open all cylinder drain valves.
4. Open gland leakoff valves.
5. Drain condensate from the main steam header and the steam leads.
6. Establish circulating-water flow through condenser.
7. Start condensate pump.
8. Establish seal on h-p gland for starting condition.
9. Establish seal on l-p gland for starting condition.
10. Start condenser air ejector and close vacuum breaker.
11. Close cylinder drains to stages under vacuum.
12. With partial vacuum established, quickly admit enough steam to start rotor and then shut off.
13. Listen for rubs on casing and at seal locations.
14. If no rubs are evident, admit enough steam to establish rotor speed of about 200 rpm. Maintain about ½ hr to warm up rotor and casing evenly.
15. Trip emergency hand control to check operation.
16. Reestablish steam flow and slowly increase speed toward rated rpm during next 15 min. If rotor vibrates severely, decrease speed and continue warming up until no objectionable vibration appears on speed increase.
17. Adjust h-p and l-p seals for operating condition.
18. When cylinder condensation ceases, close drain valves.
19. Turn on cooling water to oil cooler to maintain outlet oil temperature at about 110 F.
20. As turbine reaches rated speed, make sure that the governor takes control.
21. Place unit on line quickly and apply about 20 per cent load.
22. Open bleed-line valves and place heaters in operation.

Turbine Shutdown Procedure (encircled numbers)

1. Reduce turbine load gradually to zero and quickly take the unit off the line.
2. Close bleed-line valves and take heaters out of service.
3. Shut off steam by manual tripping of overspeed trip.
4. Open vacuum breaker.
5. Shut off air ejector.
6. Check that auxiliary oil pump starts at proper speed.
7. Shut off gland seal water.
8. Shut down condensate pump.
9. Shut off gland sealing steam.
10. Open all atmospheric drains.
11. Shut off water to oil coolers.
12. Shut down condenser circulating-water pumps.
13. Keep auxiliary oil pump in operation until unit is cool.

Noncondensing turbines follow the same general procedure as condensing turbines, for starting and stopping, with the variation that follows from exhausting into a steam main as compared to exhausting into a condenser. For example, during the warm-up period, the steam pressure at exhaust is less than the rated pressure for the usual case, and the warming steam must be vented to atmosphere. Before the turbine is placed on the line, the atmospheric relief valve is throttled to raise exhaust pressure to rated value.

In placing automatic-extraction turbines on the line, they are brought up to speed and loaded first as straight condensing or noncondensing machines, as the case may be, and then the extraction valves are put in operation. Order of procedure is reversed for shutting down.

Good Operating Practice. Careful studies of shutdown and starting procedures show that definite reasons exist for following the recommended methods. We suggest that if an operator does not understand the reason for any step, he should inquire of the manufacturer. It pays to become thoroughly familiar with your machine. Points that should be continuously in the operator's mind are given in the following list of precautions.

1. Because of close clearances, avoid operations that cause unequal expansions.
2. Avoid passing steam through a turbine with the rotor at rest.
3. Avoid drawing air through the glands with the rotor at rest.
4. Avoid unnecessary heating of the l-p end of a turbine by running it for prolonged periods below 20 per cent of rated capacity.
5. Always check the stop-valve tripping mechanism during startups and shutdowns.
6. Check the governor and automatic control operation at every opportunity.
7. During normal operation, occasionally check operation of the auxiliary oil pump.
8. Be sure the cylinder and steam mains are drained free of condensate before admitting steam to the turbine.
9. Periodically check oil in the system for contamination.
10. Immediately repair leaks in any part of the oil system.
11. Log half-hourly readings of steam and oil pressures and temperatures.

Operating Troubles. If a turbine falls off in efficiency, the cause may be (1) failure of interstage packing, sealing strips, or gland seals, (2) failure of nozzles or blades, (3) accumulation of chemical deposits on blades and nozzles. Although these troubles can be found by calculating the steam rate from watt-hour and flowmeter records, a quicker indication will be given by an upsetting of stage-pressure distribution within the turbine at any given load.

Turbine manufacturers will supply stage-pressure characteristics for their multistage turbines, showing the steam pressures in various convenient stages for the entire range of turbine loads. These may look as in Fig. 4-8. Together with such curves, the manufacturer will supply correction curves so that, if throttle pressures and temperatures and exhaust pressure do not correspond to those for which the curves are drawn, the actual stage pressures can be corrected back to such conditions before comparing them with the pressures from the curves.*

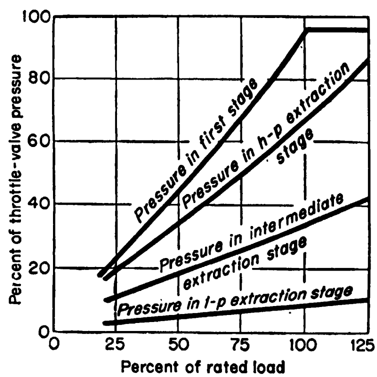


FIG. 4-8. Typical stage-pressure variation with turbine load for some types of units. Fouling and leakage change pressure relations.

Turbine Fouling. In many plants generating steam at high pressures or having difficult make-up water conditions, the most frequent cause of loss of turbine efficiency is fouling of the nozzles, blades, and valves by chemical deposits carried over from the boiler-drum water. Efficiency loss occurs because of increased friction to steam flow, change of nozzle and blade shape causing shock losses, reduction in cross-sectional flow areas, and upsetting of the stage energy distribution. In addition, turbine maximum capacity falls off.

Adjustment of feed-water treatment method should be the first thing investigated, wherever this condition arises. However, in most cases this only helps to minimize the deposits but not to eliminate them altogether. Then, to restore efficiency, the turbine must be internally cleaned. The method depends on the chemical nature of the deposits: whether they are soft and friable, or hard and firmly attached to nozzle and blade surfaces. Experimenting largely determines the cleaning method. Several methods have been developed and have been extensively reported by G. B. Warren and T. W. Howard in The Removal of Deposits from Steam Turbine Passages, *Transactions ASME*, April, 1947.

* For more details on this procedure, refer to Indexes That Point to Turbine Overhaul, by E. E. Harris and G. B. Warren, *Electrical World*, Oct. 31, 1942, also available as General Electric Co. *Bulletin* GEA-3980; also consult Indexes for Noncondensing Turbine Overhaul, by E. E. Harris, *Electrical World*, Oct. 30, 1943, also available as General Electric Co. *Bulletin* GET-1045.

1. *Removing water-soluble deposits by shutdown.* In some cases effective washing can be done by shutting down a unit for 2 or 3 days to cool off. When restarting, condensation of the incoming steam will pick up some of the solids to a satisfactory degree.

2. *Removing water-soluble deposits by light-load operation.* Where boiler-outlet steam temperature drops appreciably at light loads, such operation may do an effective cleaning job if practiced several hours a week. The turbine should operate on one control valve with the valve in wide-open position.

3. *Saturated-steam wash by water injection.* A special piping arrangement between boiler superheater outlet and turbine stop valve *sprays* feed water into the steam. All the steam entering the turbine passes through this special piping, the stop and control valves being kept wide open. The turbine is run at 20 to 25 per cent normal speed and the feed water is carefully controlled to reduce the steam temperature at a *uniform* rate not greater than 100 F per hr.* The feed-water injection rate is continued until the steam entering the turbine contains several per cent of moisture. This steam flow then continues until the condensate being thrown overboard shows complete absence of the chemicals being washed out of the turbine. Feed-water injection is then slowly reduced, in order to raise the steam temperature at a uniform rate of no more than 100 F per hr, until rated temperature is reestablished, after which the unit can be placed under load.

4. *Hot-water wash on turning gear.* This method, not recommended by manufacturers, consists of filling the lower half of the turbine casing with hot water and slowly rotating the spindle. This method does not wash the upper half of the turbine casing, and additional supports are needed for the condenser with the unusual load of water.

5. *Saturated-steam wash under load by lowering boiler temperature.* Modern h-p boilers have superheat control devices which in some cases can be used to desuperheat the steam and allow washing the turbine under load. Under this condition the steam temperature can be changed at a more rapid rate, but the boiler will be found to be the limiting factor in such an operation. Maximum change usually ranges about 50 F per hr. If superheat control does not permit bringing steam down to saturation, then feed water must be *sprayed* into the steam. This becomes hazardous from the standpoint of possible loss of water supply during washing, allowing a sudden rise in temperature. While this method does a rapid washing job, it produces the possibility of washing the deposits

* This rate is recommended for units of 10,000 kw and larger. For small units, 1,000 kw, the temperature change rate can be as high as 200 F per hr. This rate should be proportionally reduced for sizes up to 10,000 kw.

from an earlier stage to accumulate in a lower stage, with consequent plugging and dangerous pressure rise ahead of that stage. This occurrence under load and at rated speed can cause disaster.

6. *Washing out water-insoluble deposits with caustic soda.* This method in manipulation closely follows that outlined in (3) above, but after the desired moisture content in steam is established, this is slowly changed over to be partly replaced by a 10 per cent solution of sodium hydroxide for the removal of silica deposits. After silica no longer shows up in the condensate going overboard, the unit must be thoroughly washed with saturated steam until all traces of caustic disappear.

7. *Acid cleaning of water-insoluble deposits.* Hydrofluoric acid has been suggested for removing silica, but the process requires disassembling the turbine. Such treatment must be followed by a suitable neutralizing wash and finally by a thorough water wash. This method is not recommended because of the danger of acid working its way between the close-fitting parts of the turbine where it will be extremely difficult to remove.

8. *Water-insoluble deposit removal by raised steam temperature.* Another suggestion has been to raise steam temperatures in the region where silica deposits form. Theoretically this would redissolve the silica and carry it out of the turbine. No practical means of accomplishing such heating has been advanced.

9. *Mechanical cleaning of all deposits.* This is usually done as a matter of course during periodic overhauls. The most satisfactory method has proved to be the use of fly ash in a h-p air blast directed against the fouled surfaces. This leaves surfaces with a satin-smooth finish and removes very little of the metal.

More details of these nine methods can be found by referring to the article mentioned above. A note regarding temperature reduction for washing should be emphasized. Much greater rates of temperature change can be tolerated during normal variable-load operation than during washing. The much higher rate of heat transfer and conduction from wet steam to cold surfaces, or from hot surfaces to wet steam, can result in severe temperature strains and distortions of the turbine parts.

Because of the potential dangers of washing turbines, manufacturers generally favor holding these operations to a minimum. Just where to hold the line is problematical, because allowing deposits to build up too heavily can be just as detrimental to the welfare of the unit. One way of treating the problem is on an economic basis.

Frequency of turbine washing is a problem in economics. Washing is an expense in material and labor, but spending this money reduces the increased fuel consumption caused by fouling. There are a certain number of operating hours between washings at which the average total cost per hour of operation for washing and fuel loss is a minimum. At one extreme, washing at short intervals, the fuel cost saved is very small compared to the high cost of frequent washing. At the other extreme,

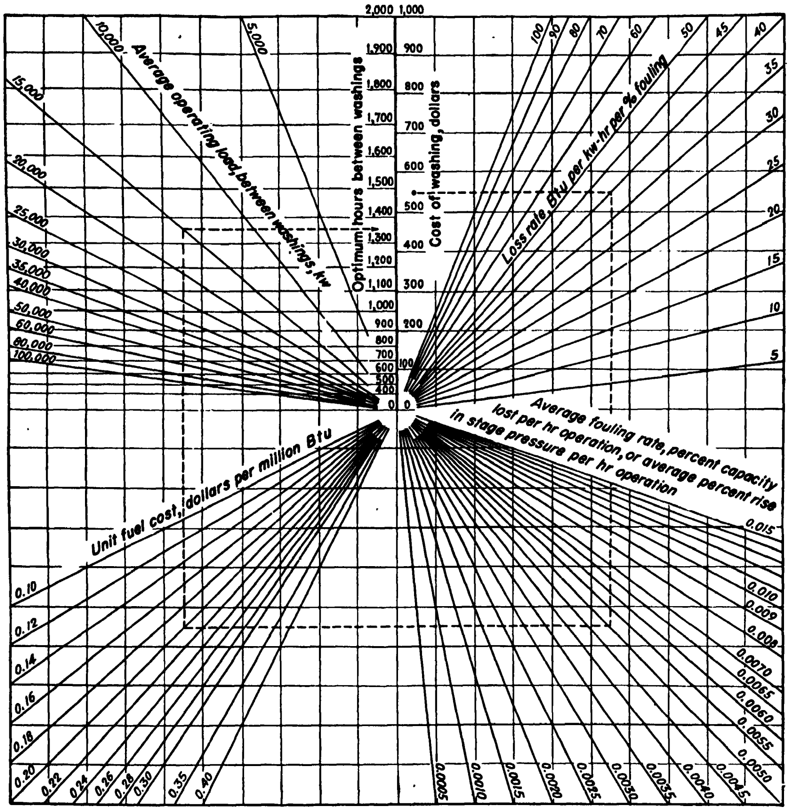


FIG. 4-9. Chart for determining the optimum number of operating hours between washings when cleaning cost, fouling rate, and fouling loss have been established. For example, the washing operation costs $550 for labor, supplies, and outage; the fouling loss rate by test is 40 Btu per kwhr per cent of fouling; the fouling rate occurs at 0.005 per cent per hr of operation; fuel costs 20 cents per million Btu; the average operating load is 15,000 kw. From trace, find 1,360 hr of operation between washings. This applies only to constant fouling-rate conditions.

washing at very long intervals, deferring cleaning expense causes large losses in added fuel cost.

Degree of fouling can be measured either by the percentage loss in capacity with wide-open governor valves or by the average percentage rise in stage pressures at a given load. From tests or carefully compiled operating data, the relation between degree of fouling (measured by either method) and increase in heat rate can be established. Rate of fouling is a matter of record and often is constant as long as make-up water and feed-water treatment remain constant. Figure 4-9 is a chart for the convenient calculation of the number of operating hours to allow between washings of a turbine and reduce the total cost of washing and fuel loss to a minimum. The number of operating hours between washings is not critical; usually a decrease or increase of 20 per cent in this period will not materially affect the economy.

In one large metropolitan central station, it had been the practice arbitrarily to wash turbines when they had lost 10 per cent of their maximum possible generating capacity (with wide-open governor-controlled valves) because of silica fouling. This economic analysis showed that for their conditions of heat-rate loss, washing cost, fouling rate, fuel cost and loading, the turbines should be washed whenever they lose only 3 per cent of maximum capacity. By allowing the turbines to foul the additional 7 per cent, an increase in operating cost of $500 monthly per unit was incurred.

Turbine Overhaul. Turbines being such finely made machines, it should be obvious that their dismantling and repair should be done under the supervision of an experienced erection engineer, who can be engaged from the service staff of the turbine builder.

Frequency of removing turbines from service for overhaul and inspection depends on variations in peak-load demands, the personnel available, and the number of units in the plant. In prewar times many plants made annual inspections of turbines, but pressure of war loads often made inspection outages inadvisable. Experience has indicated that inspections every 2 years are often adequate; in some instances, inspections are made only every third year.

To minimize outage time the inspection program should be minutely outlined, and records of previous inspections studied, before the turbine is taken out of service. In this way parts that have shown signs of deterioration will be earmarked for attention and possible replacement.

Clearances. As soon as the turbine has been taken off the line and the rotor has stopped spinning, and while it is still hot, check the axial clearance of the rotor by pulling and pushing it as far as it will go in each direction.

When the unit has cooled sufficiently, the lagging and upper half of the casing is removed as well as the bearing covers. Axial and radial clearances of blading, axial clearance between balance piston and dummies, and radial clearances between rotor shaft and gland and oil-baffle dummies should be taken before removing the rotor from the casing.

Measurements should be taken on both sides of turbine rotor and also the vertical clearances where possible. Rotate the spindle slowly in the casing and note if it is running true.

Blades. Remove the spindle from the casing and inspect all blades for damaged edges and cracks. Straighten the edges and regauge the blades where needed. Cracks on blade surfaces can be detected by visual inspection. These and subsurface cracks are most easily detected by the Magnaflux method if the blades are made of magnetic materials.

Replace any badly cracked blades. Examine the last few rows of blading for erosion effects.

Remove the diaphragms and stationary blade rings from casing, and inspect also for damage and cracks. Regauge and replace as needed.

Seals and Dummies. Inspect the dummy and gland sealing rings for damage and replace as needed. Gland runners should also be checked.

Cylinder. Check the cylinder bore for evidence of warping. Correct the condition where possible. Joining surfaces between upper and lower half of casing should be closely inspected. All high spots should be filed down to aid perfect contact between surfaces.

Governor. Before dismantling the governor, check the clearance or backlash of the driving gears. Examine the various governor parts, such as pins, bushings, spindle bearings, and all links and levers that connect the governor and valves. Replace wearing parts as needed.

Throttle Valve. Before dismantling the throttle valve, determine the time taken for the valve to become fully seated from the instant of tripping the hand lever. Take the valve down and check conditions of the valve seat, guides, and bushings. Take measurements to determine the amount of wear. Check the valve stem for trueness. If the valve has shown signs of leaking, grind-in valve seats with the valve in place.

Control valve. Where needed, grind-in valve seats. Check clearances in the guides and bushings. Too large clearance causes valve vibration when the valve begins to open. On reassembling, adjust the glands carefully to avoid excessive friction.

Bearings. Measure the clearance between the journal and bearing bore. Inspect the condition of babbitt surface in the bearing and correct as needed. If clearances are too large, machine off the joint between the bearing halves; if too small, install shims in the joint.

Thrust Bearing. Measure the axial clearance between thrust faces before dismantling. Next examine the faces carefully for flaws. Also inspect the thrust-bearing adjusting gear for wear.

Couplings. Examine all wearing parts for defects. In flexible couplings check all oil channels and holes for clogging.

Auxiliaries. Such auxiliaries as the main oil pump, auxiliary oil pump, and oil-pressure regulator should be disassembled and examined for wear. The oil cooler should be taken down and inspected for tightness and cleaned.

During the inspection period all flowmeters, pressure gauges, and thermometers should be dismounted, cleaned, and calibrated. Depending on the time and man power available, the generator, or whatever machine the turbine drives, can be inspected and overhauled at the same time. The condenser and its auxiliaries should also be inspected and cleaned.

Unusual Operation. In some central-station systems there is occasional need for supplying reactive kva. This can be done economically by a synchronous condenser. Generators driven by turbines can act in this capacity also. In ordinary operation synchronous condensers draw their power from the lines to which they supply the reactive power.

To have turbine generators act in this way, some systems bring the generators up to speed with the turbine driving in the ordinary manner and synchronize the unit on the bus. The turbine steam input is then backed down until the generator motorizes, and only enough steam is passed through the turbine to prevent overheating of the blades and nozzles. Such steam is "cooling steam" and does not contribute to supplying any of the energy losses in the turbine. It ranges from about 5,000 to 15,000 lb per hr, depending on unit size.

Other systems obviate the need for cooling steam by keeping the turbine sealing glands in top shape and pumping down the condenser with the air pumps to keep a very high vacuum. The steam-inlet valves must also be kept absolutely tight for best operation. In this way the amount of vapor left in the turbine stages is so thin that the windage losses are very small and practically no heating occurs.

Another method of making turbine generators available for synchronous condenser service is to uncouple them from their driving turbine. Such a generator is then tied in electrically to another turbine generator unit at rest. The turbine of the latter unit then brings both generators up to speed simultaneously and synchronizes both units to the bus. The starting unit can then be disconnected electrically and shut down, with the first generator taking its driving power from the bus.

CHAPTER 5

STEAM FUNDAMENTALS

Energy and Work. Some ideas are so fundamental that they are difficult to explain in simple terms. Energy is often defined as the ability to perform work or to produce some type of effect. Then, what is work? In our everyday world, work means to perform some specific job. However, there is a very specialized meaning for work in the subject we are discussing. Work, here, is defined as the application of a force moving through a distance.

Here again, force is a very fundamental idea. About the quickest way to visualize it is to think of pushing. For instance, if we push some movable object, such as a handcart or a baby carriage, we are exerting or applying a force that moves through a distance. In the sense of our specialized meaning, we are then doing work, or rather, the force we are exerting is doing work.

Work done is measured by multiplying the size of the force in pounds by the distance through which the force acted or was applied in feet, *i.e.*,

$$\text{Work, ft-lb} = \text{force, lb} \times \text{distance, ft} \qquad (1)$$

Notice that if our force, no matter how big, does not move, it cannot do work according to our meaning. For instance, we can push with all our might against a brick wall, exerting considerable force, but we won't do a lick of work, no matter how tired we get.

According to our definition, then, anything that can exert this moving force to do work has energy. Now, anything that does such work is giving up some of its energy to do this. Furthermore, anything that has work done on it absorbs or gains energy. Ideally, this energy gained equals the energy lost by the body doing the work. This gives us the next important idea, namely, that work is the transfer of energy from one body or substance to another body or substance.

Work, or the *transfer* of energy, is always noticeable by the movement associated with it, but energy itself does not necessarily meet the eye. Energy exists in many forms, and may be changed from one form to another, but it cannot be destroyed. This fact is often called the first law of thermodynamics.

Potential Energy. The simplest form of energy is known as "potential energy." It depends on the attractive or gravitational forces that exist between *all* bodies. Gravitational forces between the earth and everything on it causes these objects to fall to the earth when they are not supported. This is common everyday experience. Any object gains potential energy by having work done on it that raises it above the earth's surface. Allowing such an object to fall to the earth will let it do work on any resisting force it meets and in this way give up its potential energy.

Figure 5-1 shows the general relations between gravitational forces, work, and potential energy for bodies about the same size. The assump-

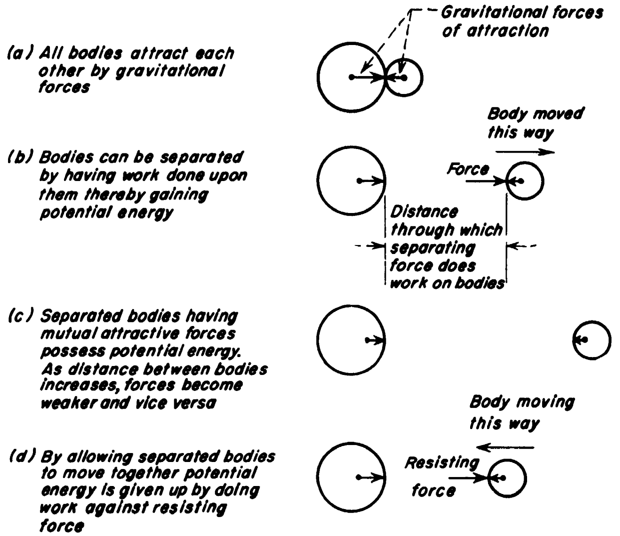


FIG. 5-1. Gravitational force can best be understood by considering two isolated bodies out in space, away from the influence of the earth or other planets.

tion is that the bodies are way out in space, far removed from the earth. Gravitational forces become weaker and weaker with greater separation of the bodies; less work is done by moving bodies 1 ft farther apart when they are 10 ft away from each other, than 1 ft additional when they are only 2 ft apart.

The effect of varying gravitational forces with distance is not noticeable ordinarily on the surface of the earth, because these forces also depend on the relative sizes of the bodies. The enormous size of the earth compared with anything on its surface makes the distance effect relatively insignificant. Only very sensitive instruments can detect the difference in the force needed to lift a 1-lb mass at sea level on the earth's surface and the force required to lift the same mass of material on top of a 10,000-ft mountain. The difference is less than 0.1 of 1 per cent.

Example 1: Find the potential energy of a body weighing 150 lb and suspended 50 ft above the earth's surface.

We measure the potential energy in terms of the work that the body could do if it were allowed to fall to the earth. The force this body would exert is its weight in pounds. We can use Eq. (1) to find the potential energy, or

$$\begin{aligned}\text{Potential energy} &= \text{work} = \text{force} \times \text{distance}\\ &= 150 \times 50 = 7{,}500 \text{ ft-lb}\end{aligned}$$

Kinetic Energy. If we take a handcart with perfect wheels and well-oiled bearings and give it a single push along level smooth ground, we

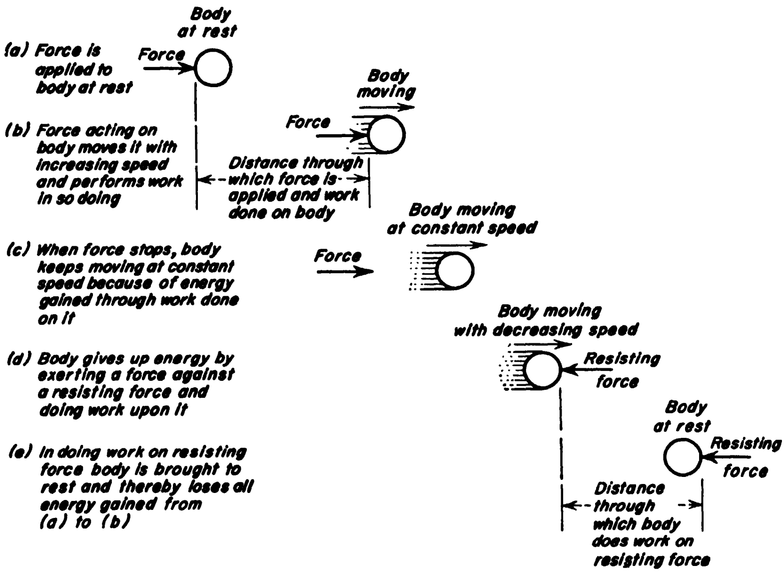


FIG. 5-2. Force is the medium that does work by acting on moving bodies. Work is one method of transferring energy.

know it will travel some distance before coming to a stop. What has happened? Work has been done on the cart by the force exerted by the hands pushing it; *i.e.*, there has been a transfer of energy from the hands to the *moving* cart. The cart absorbed this energy and exhibits it by its motion relative to the ground. This type of energy, energy due to motion, is called *kinetic energy*.

The cart gives up its energy by coming to rest. As long as it keeps moving at the speed it acquired just after the force stopped pushing, the kinetic energy stays with the cart. But we know that the cart will come to rest. This means that the cart gives up this kinetic energy. This is ordinarily done in two ways: (1) part of the energy goes to pushing aside the air (wind resistance) through which it moves, (2) the remainder changes to heat in the shaft journals and wheel bearings because of the friction between them. We shall talk about the latter change in a moment.

Kinetic energy depends on the mass of the body possessing it as well as on the speed of the body. It is measured by

$$\text{Kinetic energy, ft-lb} = \frac{\text{mass, lb} \times (\text{speed, ft/sec})^2}{64.34} \tag{2}$$

Example 2: If the handcart just mentioned moves with a speed of 8.79 ft per sec just after it has been pushed and the cart weighs 100 lb, find its kinetic energy relative to the earth.

$$\text{Kinetic energy} = \frac{100 \times 8.79 \times 8.79}{64.34} = 120 \text{ ft-lb}$$

Example 3: If the force pushing the cart was 20 lb constantly while it was applied, through how many feet must it have acted to give the cart 120 ft-lb of kinetic energy?

$$\text{Kinetic energy} = \text{work done} = \text{force} \times \text{distance}$$

then,

$$\text{Distance} = \frac{\text{work done}}{\text{force}} = \frac{120}{20} = 6 \text{ ft}$$

One very important idea to remember is that the cart would keep on moving forever if it met no obstacles such as wind and friction to dissipate its kinetic energy (Fig. 5-2).

When any body has potential energy because it is located some distance above ground level, that potential energy can be converted to kinetic energy very easily. Removing the support of such a body allows the gravitational forces to do work upon it and causes it to fall. This simply means that the potential energy is being converted to kinetic energy (energy of motion). All falling bodies eventually hit the earth, which brings them to a stop and very evidently removes their kinetic energy. Where did this energy go? It will be found that the body and the earth, both, have been heated at the point of impact. Thus, the kinetic energy of the falling body has been converted to heat.

Here, then, we have an example of the conversion of potential energy to kinetic energy through the medium of work done by gravitational forces and the final conversion of kinetic energy into heat energy.

Heat. Heat, a form of energy, is a state or condition of matter. All matter (solids, liquids, vapors, and gases) is made up of molecules, the smallest particles of any type of body that can exist individually. Molecules are made up of atoms, consisting of neutrons, protons, and electrons, which seem to be the fundamental "building blocks" of nature in *all* substances. To understand heat it is not necessary to investigate the activity that takes place between neutrons, protons, and electrons, and they will not be considered further.

Molecules may be imagined as little balls or spheres. In a solid they are packed relatively close together and, at usual temperatures, vibrate continuously in all directions (Fig. 5-3*a*). Despite their motions they remain in the same relative positions because they are held together by mutual forces of attraction, *i.e.*, by gravitational forces. The motion of the molecules comes from the kinetic energy they possess. *This individual molecular energy is the condition commonly called heat.* As heat is added to a solid from a fire or other source, the kinetic energy possessed by the individual molecules increases and they vibrate with increasing speed and through greater distances. This increase in molecular speed is commonly known as a rise in temperature; *i.e.*, temperature measures the average molecular speed of a substance.

All molecules of a given substance do not have the same speed of motion at a given value of temperature. Some speeds are lower and other speeds are higher than the average.

With additions of heat a solid will eventually melt to a liquid. Heat is absorbed as an increasing vibratory motion of the molecules and causes them progressively to break away from their fixed localities and wander at various speeds among each other, but they still remain close to each other. The condition changes from that shown in Fig. 5-3*a* to that in Fig. 5-3*b* during melting. Constant temperature during melting (or freezing) indicates that the average molecular speed remains constant. Energy absorbed during melting goes toward moving the molecules farther apart against their forces of mutual attraction. The work done against these forces increases their potential energy relative to each other.

When all the molecules can wander at will, the substance is in liquid form (Fig. 5-3*b*). However they are still relatively close together and tend to stay as a fluid mass in the bottom of the container. As the liquid is heated, its temperature rises because of increased molecular velocity. When boiling begins, the faster molecules gain sufficient kinetic energy to break away from the mutual attractive forces of the other liquid molecules into the space above the liquid to start the formation of vapor (Fig. 5-3*c*). During the boiling process the temperature remains constant, because here again the energy goes to moving the molecules farther apart, against their mutually attractive forces. This time the separation is much greater than during melting, and much more energy is required for boiling than for melting.

If the liquid is in a closed container, the vapor molecules wander in straight lines through the space at high velocity, colliding with each other and the walls of the container. From each collision the molecules rebound and speed off in different directions. Vapor molecule impacts on the container walls produce the effect known as "pressure." After all the liquid is boiled to a vapor, additional heating causes the temperature to rise, *i.e.*, increases the molecular velocity. Higher velocities produce more forceful and more impacts on the walls, thereby increasing the

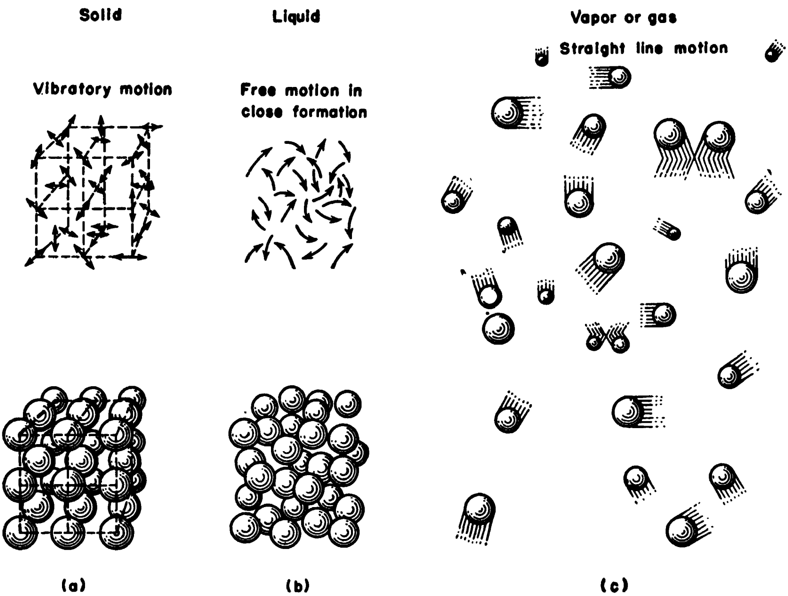


FIG. 5-3. Molecules of any substance behave differently when it is a solid, liquid, or vapor.

pressure. In this condition a vapor is said to be "superheated." At high superheats the vapor may be termed a gas; there is no sharp distinction between the two. Usually a substance is called a gas if at ordinary ranges of temperature it doesn't condense to a liquid.

Vapor pressure allowed to act on a moving piston or turbine blade will exert a force and do work. A little reflection will show that molecules hitting a wall or piston moving in the same direction will rebound at a lower velocity. The faster the piston moves relative to the molecules, the less will be the molecule speeds after rebounding. The difference in speeds before and after colliding represents a change in kinetic energy given up in doing work on the moving piston.

Vapor molecules travel at terrific speeds. For steam at 212 F and atmospheric pressure, they move at a velocity of about 1,600 mph. They collide with each other many thousands of times per second. One pound of steam contains a staggering number of molecules, about 15 million billion billions (15×10^{24}).

Steam and Water. Definite relationships exist between the temperature, pressure, and energy added for all substances. In the case of water this relation would appear as shown in Fig. 5-4 if the pressure is kept constant. This could be done as indicated in the small figure by confining the water under a piston loaded with a weight of constant value while heat (Q) is added through the wall of the cylinder. As heat is

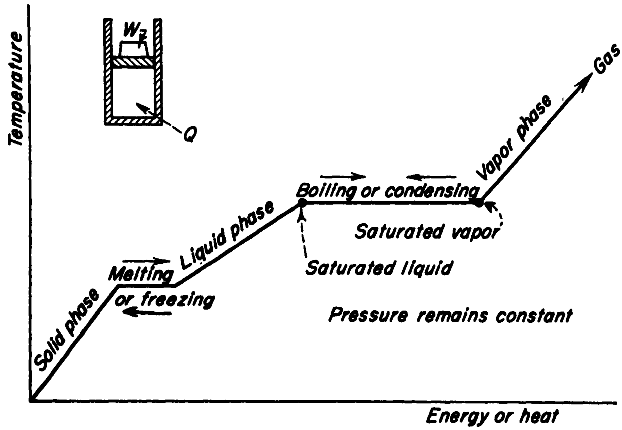


FIG. 5-4. Typical variation of substance temperature with heat energy absorbed or given up, indicating the phases and constant temperature during phase changes.

added to ice at a low temperature, the temperature rises until melting starts. During melting the temperature remains constant for further addition of heat until all the ice becomes liquid water. Melting energy is absorbed by the molecules in overcoming the forces holding them in fixed localities and allowing them to wander closely about each other.

When heat is added to a liquid, the temperature again rises until boiling begins. Here again the temperature remains constant for continued addition of heat. The energy is absorbed during vaporization (boiling) by completely breaking the molecules away from their mutually attractive forces and allowing them to soar through space. When water is just about to start to boil, it is called *saturated water;* when *all the water* has been boiled to vapor, the steam is known as *saturated steam.* Adding heat to saturated steam continuously increases its temperature; in this condition steam is called *superheated steam.*

A vapor can be changed back to a solid by extracting heat, *i.e.,* by cooling. Extracting heat from superheated steam at constant pressure drops its temperature until it becomes saturated steam. It then condenses at constant temperature until it becomes saturated water. Cooling the waetr at constant pressure reduces its temperature, and it is then known as *subcooled liquid.* Finally the subcooled water freezes at constant temperature until it becomes ice. At subsequent heat removal the temperature drops and it is finally a subcooled ice.

Measuring Energy. Heat and work, being different forms of energy, are interchangeable. This means they could be measured by the same units. We have already learned that work is measured in terms of foot-pounds; it follows that heat energy could also be measured in the same unit. However, this is not a convenient unit to use with heat; instead, another unit is used, the British thermal unit, commonly termed the "Btu." Nominally, the Btu is the amount of heat energy that raises the temperature of one pound of water by one degree Fahrenheit. It follows then that there must be a definite relation between a foot-pound and a Btu, namely,

$$1 \text{ Btu} = 778.3 \text{ ft-lb} \tag{3}$$

This relationship is known as the "mechanical equivalent of heat." It has been determined by many different experiments.

Energy and work are essentially total quantities not related to time; *i.e.*, a given quantity of energy could be transferred, or work done, over a period of 1 min or 1 hr or 1 day, year, etc., for one given process. Now the *rate of doing work*, or absorbing and giving up energy, is known as *power* which may be expressed in terms of horsepower or kilowatts. They are related as follows:

$$\begin{aligned} 1 \text{ hp} &= 33{,}000 \text{ ft-lb per min} && (4)\\ &= 550 \text{ ft-lb per sec}\\ &= 2{,}544 \text{ Btu per hr}\\ &= 0.746 \text{ kw}\\ 1 \text{ kw} &= 3{,}413 \text{ Btu per hour} && (5)\\ &= 1.34 \text{ hp} \end{aligned}$$

Forces are most usually measured as pressures, the common unit being the pounds of force acting per square inch of area, abbreviated as psi. Pressures developed by liquids, vapors, and gases are often referred to pressures of the atmospheric air, the latter being taken as the datum point, or relatively zero. When so measured, the pressure is termed pounds per square inch gauge, or psig.

A perfect vacuum means that absolutely no molecules are present; hence, the pressure on the interior sides of a container having such a condition is absolutely zero. When such a container is opened to the atmosphere, the air will enter and exert a pressure of 14.7 psi on the container sides. Hence, the atmosphere is said to have an absolute pressure of 14.7 psi, or pounds per square inch absolute, abbreviated psia. All ordinary gauges and manometers read psig. For most calculating purposes we need to know the pressure in psia; then to convert a gauge pressure to an absolute pressure, we merely add 14.7 psi, *i.e.*,

$$\text{Pressure, psia} = \text{pressure, psig} + 14.7 \qquad (6)$$

To design steam engines and turbines exact knowledge must be available of steam and water behavior in terms of pressure, temperature,

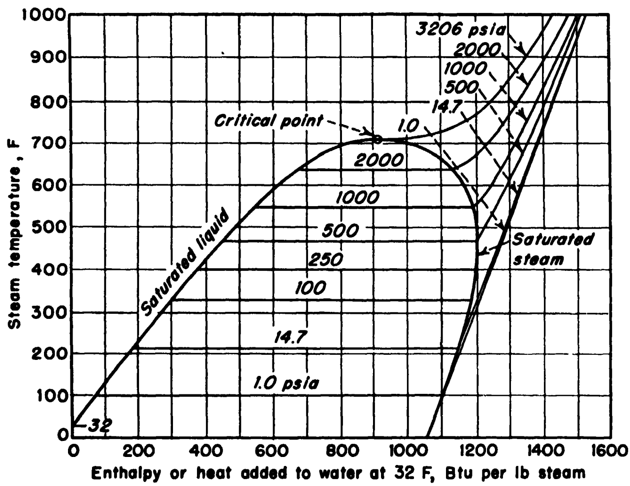


FIG. 5-5. Relation of enthalpy and temperature for water and steam over the usual range used in power plants.

volume, and energy. Heat energy in steam and water is termed "enthalpy," and it is arbitrarily taken as zero for water at 32 F.

The relation between temperature, pressure, and enthalpy for steam and water is given in Fig. 5-5. This figure is similar to Fig. 5-4, except that it does not indicate the ice and melting phases of water. Let's examine these relations; for example, at a constant pressure of 1.0 psia, as water at 32 F is heated, the temperature rises to 101 F with the addition of 69.7 Btu. It starts to boil at this temperature with continued heating. It takes 1,035 Btu per lb to vaporize water at 101 F to steam at 101 F. As more heat is added, the temperature rises rapidly to form superheated steam.

At constant 250 psia pressure, 376 Btu added to water at 32 F forms saturated water at 401 F; 825 Btu more boils the water to saturated steam at 401 F constant temperature; further heating superheats the steam.

As the pressure increases, the heat needed to form saturated liquid increases, but the energy or heating needed for vaporization or boiling decreases. In fact vaporization energy reduces to zero at the *critical pressure* and *temperature*, 3,206.2 psia and 705.4 F. For pressures and temperatures higher than the critical, there is no visible difference between liquid and vapor. These pressures are not used in practice.

Saturated steam has the highest enthalpy at 444 psia pressure and 455 F temperature; whereas saturated water has the highest enthalpy at the critical point. Note that at the critical point the enthalpy of saturated water is the same as that of saturated steam.

Steam Tables. To aid rapid calculation of the performance of steam machinery the properties of steam are arranged in tables called "steam tables." The ones most commonly used in the United States are "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G. Keyes, published by John Wiley & Sons, Inc., New York, 1937. Very abridged tables abstracted from these are given in Tables 5-1 to 5-3. The first table has the properties arranged against temperature in degrees Fahrenheit for saturated water and saturated steam. In the first column are the values of steam and water temperature (in degrees F) against which the other properties are listed; in the second column are the corresponding pressures in psia of steam and water. In the third column under symbol v_f is the specific volume of saturated water in cubic feet per pound. The fourth column (with symbol v_{fg}) has the increase in specific volume as 1 lb of saturated water is heated at constant temperature and pressure to saturated steam. The fifth column (with symbol v_g) is the specific volume of saturated steam. Note that $v_f + v_{fg} = v_g$. The ninth column has the enthalpy, h_f, of saturated water above 32 F in Btu per pound. The tenth column, h_{fg}, is the additional enthalpy required to heat saturated liquid to saturated steam also known as heat of vaporization. The eleventh column, h_g, has the enthalpy of saturated steam. Note that $h_f + h_{fg} = h_g$. The last three columns have the entropy for saturated liquid, vaporization, and saturated steam, the use of which will be demonstrated later.

Table 5-2 has the same properties listed in Table 5-1 except that the pressure and temperature columns are interchanged, thus making it easier to select values corresponding to given pressures. Additional columns for internal energy of saturated water and saturated steam are included in Tables 5-1 and 5-2. These properties are used for certain steam processes not considered in this book.

Table 5-3 lists the properties of superheated steam against pressure

TABLE 5-1. PROPERTIES OF SATURATED STEAM VS. TEMPERATURE*

Temp, °F	Press, psia	Volume, ft³/lb			Internal energy, Btu/lb			Enthalpy, Btu/lb			Entropy, Btu/(lb × °R)		
		v_f	v_{fg}	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
Solid—Vapor													
−40	0.0019	0.01737	133,900	133,900	−177.00	1174.9	1021.6	−177.00	1221.2	1044.2	−0.3654	2.9087	2.5433
−20	0.0062	0.01739	42,200	42,200	−168.16	1172.7	1017.7	−168.16	1221.2	1053.0	−0.3448	2.7764	2.4316
0	0.0185	0.01742	14,770	14,770	−158.93	1170.0	1011.1	−158.93	1220.7	1061.8	−0.3241	2.6546	2.3305
20	0.0505	0.01745	5,658	5,658	−149.31	1167.0	1004.5	−149.31	1219.9	1070.6	−0.3038	2.5425	2.2387
32	0.0885	0.01747	3,306	3,306	−143.35	1164.9	997.9	−143.35	1219.1	1075.8	−0.2916	2.4793	2.1877
Liquid—Vapor													
32	0.08854	0.01602	3,306	3,306	0.00	1021.7	1021.7	0.00	1075.8	1075.8	0.00	2.1877	2.1877
35	0.09995	0.01602	2,947	2,947	3.02	1019.5	1022.6	3.02	1074.1	1077.1	0.0061	2.1709	2.1770
40	0.12170	0.01602	2,444	2,444	7.95	1016.4	1024.3	8.05	1071.3	1079.3	0.0162	2.1435	2.1597
45	0.14752	0.01602	2,036	2,036	13.05	1012.8	1025.9	13.06	1068.4	1081.5	0.0262	2.1167	2.1429
50	0.17811	0.01603	1,703	1,703	18.07	1009.5	1027.6	18.07	1065.6	1083.7	0.0361	2.0903	2.1264
60	0.2563	0.01604	1,206.6	1,206.7	28.06	1002.9	1031.0	28.06	1059.9	1088.0	0.0555	2.0393	2.0948
70	0.3631	0.01606	867.8	867.9	38.04	995.9	1034.0	38.04	1054.3	1092.3	0.0745	1.9902	2.0647
80	0.5069	0.01608	633.1	633.1	48.02	989.2	1037.2	48.02	1048.6	1096.6	0.0932	1.9428	2.0360
90	0.6982	0.01610	468.0	468.0	57.99	982.5	1040.5	57.99	1042.9	1100.9	0.1115	1.8972	2.0087
100	0.9492	0.01613	350.3	350.4	67.97	975.7	1043.7	67.97	1037.2	1105.2	0.1295	1.8531	1.9826
120	1.6924	0.01620	203.25	203.27	87.91	962.1	1050.0	87.92	1025.8	1113.7	0.1645	1.7694	1.9339
140	2.8886	0.01629	122.99	123.01	107.88	948.4	1056.3	107.89	1014.1	1122.0	0.1984	1.6910	1.8894
160	4.741	0.01639	77.27	77.29	127.88	934.5	1062.4	127.89	1002.3	1130.2	0.2311	1.6174	1.8485
180	7.510	0.01651	50.21	50.23	147.90	920.4	1068.3	147.92	990.2	1138.1	0.2630	1.5480	1.8109
200	11.526	0.01663	33.62	33.64	167.95	906.3	1074.2	167.99	977.9	1145.9	0.2938	1.4824	1.7762
212	14.696	0.01672	26.78	26.80	180.02	897.5	1077.5	180.07	970.3	1150.4	0.3120	1.4446	1.7566
220	17.186	0.01677	23.13	23.15	188.08	891.7	1079.8	188.13	965.2	1153.4	0.3239	1.4201	1.7440
240	24.969	0.01692	16.306	16.323	208.26	876.8	1085.1	208.34	952.2	1160.5	0.3531	1.3609	1.7140
260	35.429	0.01709	11.746	11.763	228.53	861.7	1090.2	228.64	938.7	1167.3	0.3817	1.3043	1.6860
280	49.203	0.01726	8.628	8.645	248.90	846.2	1095.1	249.06	924.7	1173.8	0.4096	1.2501	1.6597
300	67.013	0.01745	6.449	6.466	269.37	830.1	1099.5	269.59	910.1	1179.7	0.4369	1.1980	1.6350
320	89.66	0.01765	4.896	4.914	289.99	813.7	1103.7	290.28	894.9	1185.2	0.4637	1.1478	1.6115
340	118.01	0.01787	3.770	3.788	310.74	796.7	1107.4	311.13	879.0	1190.1	0.4900	1.0992	1.5891
360	153.04	0.01811	2.939	2.957	331.67	779.0	1110.7	332.18	862.2	1194.4	0.5158	1.0519	1.5677
380	195.77	0.01836	2.317	2.335	352.78	760.7	1113.5	353.45	844.6	1198.1	0.5413	1.0059	1.5471
400	247.31	0.01864	1.8447	1.8633	374.12	741.6	1115.7	374.97	826.0	1201.0	0.5664	0.9608	1.5272
450	422.6	0.0194	1.0799	1.0993	428.6	690.0	1118.6	430.1	774.5	1204.6	0.6280	0.8513	1.4793
500	680.8	0.0204	0.6545	0.6749	485.2	631.5	1116.7	487.8	713.9	1201.7	0.6887	0.7438	1.4325
600	1,542.9	0.0236	0.2432	0.2668	610.3	479.0	1089.3	617.0	548.5	1165.5	0.8131	0.5176	1.3307
700	3,093.7	0.0369	0.0392	0.0761	802.2	149.6	951.8	823.3	172.1	995.4	0.9905	0.1484	1.1389
705.4	3,206.2	0.0503	0	0.0503	872.9	0	872.9	902.7	0	902.7	1.0580	0	1.0580

* Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York, 1936.

TABLE 5-2. PROPERTIES OF SATURATED STEAM VS. PRESSURE*

Press, psia	Temp, °F	Volume, ft³/lb			Internal energy, Btu/lb			Enthalpy, Btu/lb			Entropy, Btu/(lb × °R)		
		v_f	v_{fg}	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
In. Hg. 1	79.03	0.01608	652.28	652.3	47.05	990.0	1037.0	47.05	1049.2	1096.3	0.0914	1.9473	2.0387
In. Hg. 2	101.14	0.01614	339.18	339.2	69.10	974.9	1044.0	69.10	1036.6	1105.7	0.1316	1.8481	1.9797
In. Hg. 3	115.06	0.01618	231.58	231.6	82.99	965.5	1048.5	82.99	1028.6	1111.6	0.1560	1.7896	1.9456
3	141.48	0.01630	118.69	118.71	109.36	947.3	1056.7	109.37	1013.2	1122.6	0.2008	1.6855	1.8863
5	162.24	0.01640	73.50	73.52	130.12	933.0	1063.1	130.13	1001.0	1131.1	0.2347	1.6094	1.8441
10	193.21	0.01659	38.40	38.42	161.14	911.1	1072.2	161.17	982.1	1143.3	0.2835	1.5041	1.7876
15	213.03	0.01672	26.27	26.29	181.06	896.7	1077.8	181.11	969.7	1150.8	0.3135	1.4415	1.7549
20	227.96	0.01683	20.07	20.089	196.10	885.8	1081.9	196.16	960.1	1156.3	0.3356	1.3962	1.7319
25	240.07	0.01692	16.286	16.303	208.34	876.8	1085.1	208.42	952.1	1160.6	0.3533	1.3606	1.7139
30	250.33	0.01701	13.729	13.746	218.73	869.1	1087.8	218.82	945.3	1164.1	0.3680	1.3313	1.6993
40	267.25	0.01715	10.482	10.498	235.90	856.1	1092.0	236.03	933.7	1169.7	0.3919	1.2844	1.6763
50	281.01	0.01727	8.498	8.515	249.93	845.4	1095.3	250.09	924.0	1174.1	0.4110	1.2474	1.6585
60	292.71	0.01738	7.158	7.175	261.90	836.0	1097.9	262.09	915.5	1177.6	0.4270	1.2168	1.6438
70	302.92	0.01748	6.189	6.206	272.38	827.8	1100.2	272.61	907.9	1180.6	0.4409	1.1906	1.6315
80	312.03	0.01757	5.465	5.472	281.76	820.3	1102.1	282.02	901.1	1183.1	0.4531	1.1676	1.6207
100	327.81	0.01774	4.415	4.432	298.08	807.1	1105.2	298.40	888.8	1187.2	0.4740	1.1286	1.6026
120	341.25	0.01789	3.712	3.728	312.05	795.6	1107.6	312.44	877.9	1190.4	0.4916	1.0962	1.5878
140	353.02	0.01802	3.202	3.220	324.35	785.2	1109.6	324.82	868.2	1193.0	0.5069	1.0682	1.5751
160	363.53	0.01815	2.816	2.834	335.39	775.8	1111.2	335.93	859.2	1195.1	0.5204	1.0436	1.5640
180	373.06	0.01827	2.514	2.532	345.42	767.1	1112.5	346.03	850.8	1196.9	0.5325	1.0217	1.5542
200	381.79	0.01839	2.270	2.288	354.68	759.0	1113.7	355.36	843.0	1198.4	0.5435	1.0018	1.5453
220	389.86	0.01850	2.069	2.087	363.27	751.3	1114.6	364.02	835.6	1199.6	0.5537	0.9835	1.5372
240	397.37	0.01860	1.900	1.9183	371.29	744.1	1115.4	372.12	828.5	1200.6	0.5631	0.9667	1.5298
260	404.42	0.01870	1.756	1.7748	378.86	737.3	1116.1	379.76	821.8	1201.5	0.5719	0.9510	1.5229
280	411.05	0.01880	1.6323	1.6511	386.01	730.7	1116.7	386.90	815.3	1202.3	0.5801	0.9363	1.5164
300	417.33	0.01890	1.5244	1.5433	392.79	724.3	1117.1	393.84	809.0	1202.8	0.5879	0.9225	1.5104
320	423.29	0.01899	1.4296	1.4485	399.26	718.3	1117.6	400.39	803.0	1203.4	0.5952	0.9094	1.5046
340	428.97	0.01908	1.3435	1.3645	405.46	712.4	1117.9	406.66	797.1	1203.7	0.6022	0.8970	1.4992
360	434.40	0.01917	1.2704	1.2895	411.39	706.8	1118.2	412.67	791.4	1204.1	0.6090	0.8851	1.4941
380	439.60	0.01925	1.2030	1.2222	417.10	701.3	1118.4	418.45	785.8	1204.3	0.6153	0.8738	1.4891
400	444.59	0.0193	1.1420	1.1613	422.6	695.9	1118.5	424.0	780.5	1204.5	0.6214	0.8630	1.4844
500	467.01	0.0197	0.9081	0.9278	447.6	671.0	1118.6	449.4	755.0	1204.4	0.6487	0.8147	1.4634
600	486.21	0.0201	0.7497	0.7698	469.4	648.3	1117.7	471.6	731.6	1203.2	0.6720	0.7734	1.4454
700	503.10	0.0205	0.6349	0.6554	488.8	627.5	1116.3	491.5	709.7	1201.2	0.6925	0.7371	1.4296
800	518.23	0.0209	0.5478	0.5687	506.6	607.8	1114.4	509.7	688.9	1198.6	0.7108	0.7045	1.4153
1,000	544.61	0.0216	0.4240	0.4456	538.4	571.0	1109.4	542.4	649.4	1191.8	0.7430	0.6467	1.3897
1,500	596.23	0.0235	0.2530	0.2765	605.1	486.1	1091.2	611.6	556.3	1167.9	0.8082	0.5269	1.3351
2,000	635.82	0.0257	0.1621	0.1878	662.2	403.4	1065.6	671.7	463.4	1135.1	0.8619	0.4230	1.2849
2,500	668.13	0.0287	0.1021	0.1307	717.3	313.3	1030.6	730.6	360.5	1091.1	0.9126	0.3197	1.2322
3,000	695.36	0.0346	0.0512	0.0858	783.4	189.3	972.9	802.5	217.8	1020.3	0.9731	0.1885	1.1615

* Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G. Keyes. John Wiley & Sons, Inc., New York, 1936.

TABLE 5-3. PROPERTIES OF SUPERHEATED STEAM*

p, psia; t, °F; v, ft³/lb; u and h, Btu/lb; s, Btu/(lb × °R)

Press, psia (sat temp)	Prop	Temperature, °F											
		200	300	400	500	600	700	800	900	1000	1200	1400	1600
1 (101.74)	v	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	988.7	1107.8	1227.0
	u	1077.7	1112.1	1146.9	1182.5	1218.9	1255.9	1293.0	1332.8	1372.6	1455.7	1540.7	1630.4
	h	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1637.7	1745.7	1857.5
	s	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4952	2.5566	2.6137
5 (162.24)	v	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	197.71	221.6	245.4
	u	1076.5	1111.5	1146.6	1182.4	1218.6	1255.8	1293.8	1332.7	1372.6	1455.7	1540.7	1630.4
	h	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1637.7	1745.7	1857.4
	s	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.3178	2.3792	2.4363
14.696 (212.00)	v	....	30.53	34.68	38.78	42.86	49.64	51.00	55.07	59.13	67.25	75.37	83.48
	u	....	1109.7	1145.7	1181.7	1218.2	1255.5	1293.6	1332.5	1372.3	1455.5	1540.6	1630.3
	h	....	1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1637.5	1745.5	1857.3
	s	....	1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	2.1989	2.2603	2.3174
30 (250.33)	v	....	14.816	16.897	18.933	20.95	22.96	24.96	26.95	28.95	32.93	36.91	40.89
	u	....	1107.1	1144.1	1180.6	1217.5	1254.9	1293.1	1332.2	1372.0	1454.7	1540.4	1630.0
	h	....	1189.3	1237.9	1285.7	1333.8	1382.4	1431.7	1481.8	1532.7	1637.2	1745.3	1857.1
	s	....	1.7336	1.7937	1.8464	1.8940	1.9379	1.9786	2.0169	2.0530	2.1201	2.1815	2.2386
50 (281.01)	v	....	8.773	10.065	11.309	12.532	13.744	14.950	16.152	17.352	19.747	22.14	24.53
	u	....	1103.1	1142.0	1179.2	1216.5	1254.2	1292.6	1331.6	1371.5	1454.1	1540.1	1629.8
	h	....	1184.3	1235.1	1283.9	1332.5	1381.4	1430.9	1481.1	1532.1	1636.8	1745.0	1856.8
	s	....	1.6721	1.7349	1.7887	1.8368	1.8809	1.9219	1.9602	1.9964	2.0636	2.1251	2.1822
100 (327.81)	v	....	....	4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.860	11.060	12.258
	u	....	....	1136.2	1175.7	1214.0	1252.3	1291.1	1330.5	1370.7	1453.2	1539.5	1629.3
	h	....	....	1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1635.7	1744.2	1856.2
	s	....	....	1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9867	2.0484	2.1056
200 (381.59)	v	....	....	2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.917	5.521	6.123
	u	....	....	1123.0	1168.0	1208.8	1248.5	1288.1	1328.1	1368.5	1451.7	1538.2	1628.3
	h	....	....	1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1633.7	1742.6	1855.0
	s	....	....	1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.9044	1.9713	2.0287

* Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., 1936. u values computed from data in reference.

TABLE 5-3. PROPERTIES OF SUPERHEATED STEAM* (*Continued*)

Press, psia (sat temp)	Prop	Temperature, °F											
		200	300	400	500	600	700	800	900	1000	1200	1400	1600
300 (417.33)	*v*	...	...	...	1.7675	2.005	2.227	2.442	2.652	2.859	3.269	3.674	4.078
	u	...	...	...	1159.4	1203.4	1244.6	1285.0	1325.5	1366.5	1450.2	1537.0	1627.3
	h	...	...	...	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1631.7	1741.0	1853.7
	s	...	...	...	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8638	1.9260	1.9835
400 (444.59)	*v*	...	...	...	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.445	2.751	3.055
	u	...	...	...	1150.0	1197.5	1240.5	1281.9	1323.0	1364.4	1448.6	1535.8	1626.3
	h	...	...	...	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1629.6	1739.5	1852.5
	s	...	...	...	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.8311	1.8936	1.9485
500 (467.01)	*v*	...	...	...	0.9927	1.1591	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
	u	...	...	...	1139.4	1191.3	1236.3	1278.8	1320.6	1362.3	1447.1	1534.6	1625.3
	h	...	...	...	1231.3	1298.6	1357.0	1412.1	1466.0	1519.6	1627.6	1737.7	1851.3
	s	...	...	...	1.4919	1.5588	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
1000 (544.61)	*v*	...	...	...	...	0.5140	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
	u	...	...	...	...	1153.7	1212.7	1261.9	1307.5	1351.6	1439.3	1528.4	1620.5
	h	...	...	...	...	1248.8	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
	s	...	...	...	...	1.4450	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
2000 (635.82)	*v*	...	...	...	...	...	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
	u	...	...	...	...	...	1147.9	1221.7	1278.5	1328.8	1423.3	1516.0	1610.0
	h	...	...	...	...	...	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
	s	...	...	...	...	...	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
3000 (695.36)	*v*	...	...	...	...	...	0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
	u	...	...	...	...	...	1006.1	1169.5	1245.1	1304.3	1406.7	1503.4	1599.7
	h	...	...	...	...	...	1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
	s	...	...	...	...	...	1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
4000	*v*	...	...	...	...	...	0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
	u	...	...	...	...	...	742.6	1096.9	1206.1	1277.8	1389.8	1490.6	1589.3
	h	...	...	...	...	...	763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s	...	...	...	...	...	0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
5000	*v*	...	...	...	...	...	0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
	u	...	...	...	...	...	721.6	992.2	1160.6	1248.9	1372.5	1477.7	1579.0
	h	...	...	...	...	...	746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s	...	...	...	...	...	0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499

* Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick C. Keyes, John Wiley & Sons, Inc., 1936. *u* values computed from data in reference.

and temperature. The first column lists the steam pressure in psia and corresponding saturation temperature. Across the top of the table is listed the temperature of the superheated steam. Under each temperature and opposite each pressure are listed the specific volume v, internal energy u, enthalpy h, and entropy s. In any of the tables the properties at values of pressure and temperature falling in between the listed values are found by interpolation. Interpolation will not be so satisfactory on these tables as on the originals from which they are abstracted. The original tables list many more values at smaller intervals and are therefore recommended for use.

Steam exhausted from condensing turbines usually consists of a mixture of saturated steam and saturated water. This mixture commonly termed "wet steam" must be measured according to its amount of water and steam content in order to determine its energy or enthalpy content. This is found by stating either per cent moisture or per cent quality. These terms are found as follows:

$$y = \text{per cent moisture} = \frac{\text{weight of saturated water in mixture, lb} \times 100}{\text{total weight of saturated water and steam, lb}} \tag{7}$$

$$x = \text{per cent quality} = \frac{\text{weight of saturated steam in mixture, lb} \times 100}{\text{total weight of saturated water and steam, lb}} \tag{8}$$

Per cent quality could just as well have been called per cent steam, but custom has fixed on the former term. Note that for any mixture

$$\text{Per cent quality} + \text{per cent moisture} = 100, \qquad \text{or} \qquad x + y = 100 \tag{9}$$

Knowing the make-up of the mixture by either of the foregoing percentages, it is possible to determine its volume and enthalpy with the aid of the steam tables as follows: When quality is given,

$$\begin{aligned} \text{Enthalpy of wet steam, Btu per lb} &= \frac{(100 - x)}{100} h_f + \frac{x}{100} h_g \\ &= h_f + \frac{x}{100} h_{fg} \end{aligned} \tag{10}$$

When moisture is given,

$$\begin{aligned} \text{Enthalpy of wet steam, Btu per lb} &= \frac{y}{100} h_f + \frac{(100 - y)}{100} h_g \\ &= h_g - \frac{y}{100} h_{fg} \end{aligned} \tag{11}$$

Similarly, to find the volume,

$$\begin{aligned} \text{Volume of wet steam, cu ft per lb} &= v_f + \frac{x}{100} v_{fg} \\ &= v_g - \frac{y}{100} v_{fg} \end{aligned} \tag{12}$$

Example 4: Steam having a temperature of 400 F is wet and has a quality of 80 per cent. Determine its enthalpy and specific volume.

Solution: From Table 5-1 at 400 F, $h_f = 375.0$ Btu per lb, $h_{fg} = 826.0$ Btu per lb, $h_g = 1{,}201.0$ Btu per lb, $v_f = 0.01864$ cu ft per lb, $v_{fg} = 1.8447$ cu ft per lb, $v_g = 1.8633$ cu ft per lb.

Enthalpy of wet steam $= h = 375.0 + {}^{80}\!/_{100}\ 826.0 = 1{,}035.8$ Btu per lb.

Specific volume of wet steam $= v = 0.0186 + {}^{80}\!/_{100}\ 1.8447 = 1.4944$ cu ft per lb.

This can also be solved by using the moisture, $y = 100 - 80 = 20$ per cent.

Mollier Chart. We have discussed the interchangeability of energy in the early part of this chapter, but there is an important limitation that should not be overlooked. While it is possible to convert all mechanical forms of energy to heat through impact and friction, it is possible to convert only *part of heat energy* to mechanical energy; *i.e.*, of any given amount of heat energy in steam only a minor part can be ordinarily converted to mechanical energy, the remainder stays in the form of heat, usually at a lower temperature and pressure than that of the initial amount. This fact is a part of the second law of thermodynamics.

Steam turbines take in steam at a given pressure and temperature, convert part of the steam energy into mechanical energy in the form of a rotating shaft, and then discharge the same quantity of steam at a lower pressure and temperature than at the throttle. To determine the maximum amount of steam energy that can be converted to mechanical work, a property called "entropy" is used. The most convenient way to demonstrate the use of entropy for this purpose is in the Mollier chart (Fig. 5-6).

Here entropy is represented by the vertical lines, enthalpy by the horizontal lines, and steam pressure, temperature, and per cent of moisture by the various sets of curved lines. Now, let's find the maximum amount of energy in steam at 100 psia, 650 F, that can be converted to mechanical work when the steam expands down to a pressure of 5 psia. First, find the enthalpy of steam at 100 psia and 650 F where the two curves cross. This value is 1,354 Btu per lb. From this intersection, follow down the vertical line until it intersects the 5-psia line. At this second point the steam enthalpy is 1,091 Btu per lb. At this latter condition the steam is wet with about 4.0 per cent moisture content. We started out with superheated steam and ended up the process with wet steam. The steam has given up $1{,}354 - 1{,}091 = 263$ Btu per lb. This is the maximum amount of mechanical work that the steam can develop when expanding to 5 psia even though it had initially 1,354 Btu per lb. To produce this maximum work the steam is said to expand at constant entropy.

If the steam expands down to 0.2 psia it still retains 911 Btu per lb and will have 16.2 per cent moisture. The work developed is then

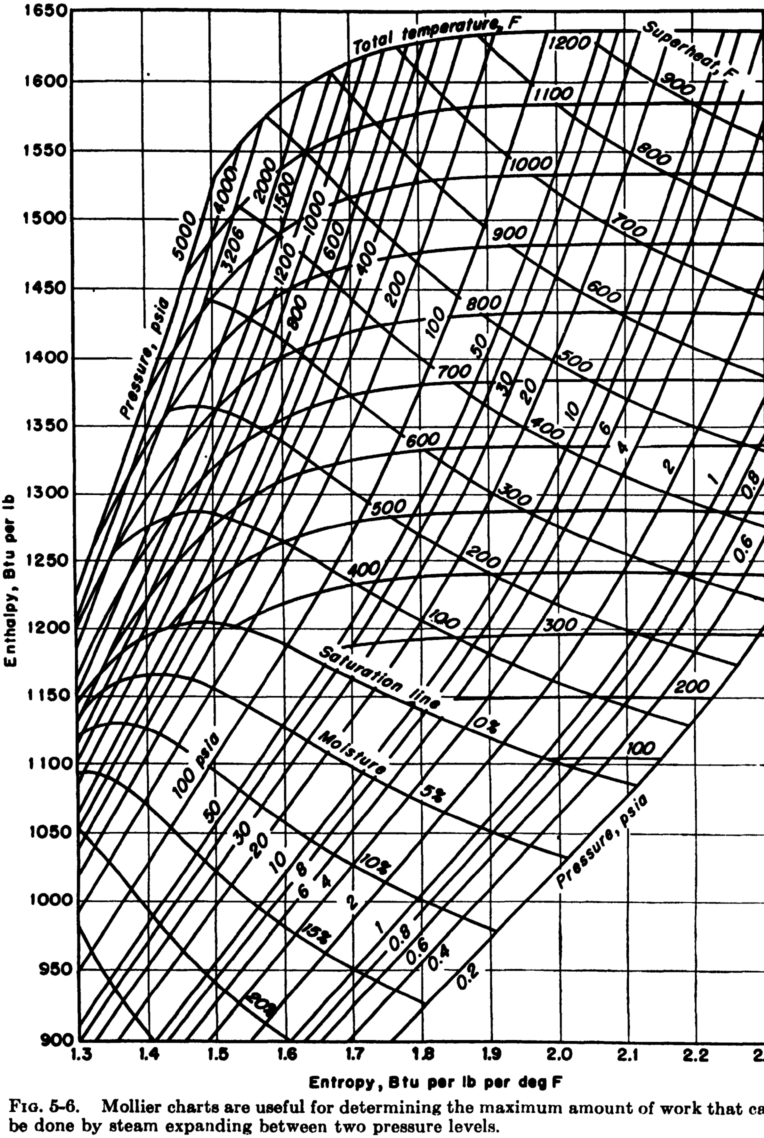


FIG. 5-6. Mollier charts are useful for determining the maximum amount of work that can be done by steam expanding between two pressure levels.

1,354 − 911 = 443 Btu per lb. It can be seen that only a minor part of the energy is *available* for conversion to work. What's more, this is the ideal amount of energy that can be converted; actually much less is converted to work, only about 70 to 85 per cent in most cases. The amount that is not converted to work is lost to friction and eddy-current effects, which will be explained later. This unconverted energy finally

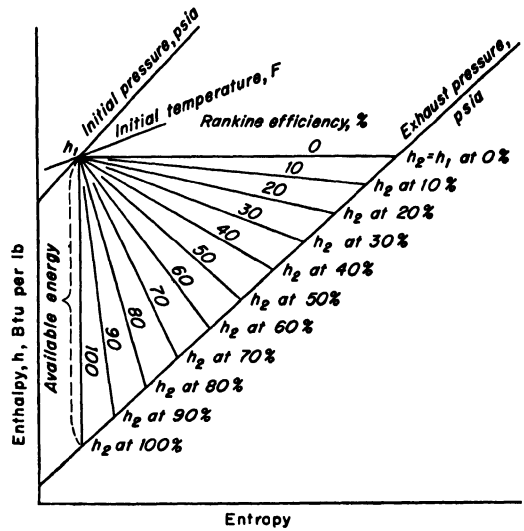


FIG. 5-7. Condition lines of steam at various Rankine efficiencies. At 100 per cent the line is vertical, at 0 per cent it is horizontal.

remains as heat in the steam. The ratio of actual heat converted to work to that ideally available is called the Rankine efficiency, *i.e.*,

$$\text{Rankine efficiency, per cent} = \frac{\text{actual energy converted, Btu per lb}}{\text{available energy, Btu per lb}} \times 100 \qquad (13)$$

The Mollier chart is useful for estimating changes in the condition of steam as it passes through a turbine. As just discussed, the ideal series of changes would be one that traces a vertical line on the chart at constant entropy. This would be a steam expansion process with 100 per cent Rankine efficiency. Actual steam processes, however, range less than 100 per cent down to 0 per cent. For the zero efficiency condition, the initial and final enthalpies are equal even though steam pressures and temperatures decrease during the expansion process; the steam has simply changed state without doing any work. This happens when steam is allowed to blow through a hole from an h-p region to one of lower pressure, and is called "throttling." Figure 5-7 shows the series of changes or path of the steam process for different efficiencies, starting from one initial state expanding to the same exhaust pressure. These lines are usually called *condition lines.* Note that all possible condition lines starting at a common point can lie in only one quadrant.

Example 5: Steam enters a turbine at 800 psia and 800 F and exhausts at a pressure of 1.0 psia and contains 10 per cent moisture. Find the Rankine efficiency.

From the Mollier chart (Fig. 5-6) the initial enthalpy is 1,398 Btu per lb. For a constant entropy expansion to 1.0 psia from this initial condition, the final enthalpy would be 891 Btu per lb (and moisture about 20.6 per cent). Then, available energy = 1,398 − 891 = 507 Btu per lb. Now, at 1.0 psia and 10 per cent moisture steam enthalpy is 1,003 Btu per lb. Then, actual energy converted = 1,398 − 1,003 = 395 Btu per lb. Hence,

$$\text{Rankine efficiency} = {}^{395}\!/_{507} = 0.779, \text{ or } 77.9 \text{ per cent}$$

Steam Flow. Just how does steam give up some of its energy in a turbine to produce mechanical shaft energy? This is done in two principal pieces of equipment built into every turbine: (1) the nozzle and (2) the blade. Figure 5-8 shows simplified versions of these elements.

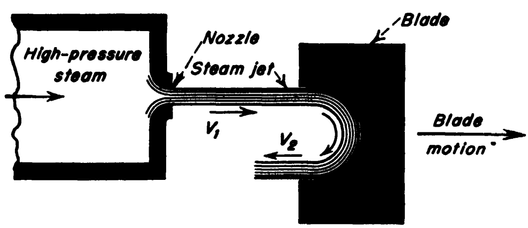


FIG. 5-8. In an ideal blade and nozzle arrangement, the steam jet travels in the same direction as the blade, when approaching; in the opposite direction, when leaving the blade.

The nozzle essentially consists of nothing more than a smooth-shaped hole in a wall. Its exact contour of cross section depends on the ratio of the pressure of the steam before it enters the nozzle and the pressure of the region into which the steam discharges. If the original pressure is less than twice the final pressure, the contour approximates that shown in the sketch, called a "convergent nozzle."

The h-p steam before reaching the nozzle opening has its molecules flying in all directions at high speed, colliding with each other and the walls of the pipe. Those molecules just happening to be near the entrance to the nozzle and traveling in the direction of the outlet speed out through the nozzle and form a smooth jet of steam issuing into the region of lower pressure due to the steam expansion in the nozzle..

The h-p steam as a whole moves with a relatively low speed toward the nozzle, but the jet has a high velocity. The jet's pressure is the same as that of the region in which the blade is located, and if a thermometer could be placed in the jet so that it would move with the jet at the same speed, it would show that the steam temperature had also dropped. We then realize that some of the heat energy of the steam has been used to produce mechanical kinetic energy in the form of the high-speed steam jet. This jet, just like a jet of water from a hose, could be directed against any object and produce a force that could perform work.

According to Eq. (2) (page 141) the kinetic energy depends on the speed of the jet. In turn the speed of the jet depends on the initial and final steam enthalpies determined by the initial steam conditions, the final pressure, and the Rankine efficiency of the nozzle, usually called the "nozzle efficiency."

$$\text{Jet velocity, ft per sec} = 223.7\sqrt{e_n\,(h_1 - h_2)} \tag{14}$$

where e_n = nozzle efficiency as a decimal (Rankine efficiency)
h_1 = enthalpy of steam before nozzle, Btu per lb
h_2 = enthalpy of steam after ideal expansion from initial conditions at constant entropy to final pressure, Btu per lb

Example 6: Steam at 200 psia and 500 F expands through a nozzle with an efficiency of 95 per cent to a pressure of 100 psia. Find the jet velocity and kinetic energy.

From the Mollier chart (Fig. 5-6), h_1 = 1,269 Btu per lb, h_2 = 1,204 Btu per lb. Then,

$$\text{Jet velocity} = 223.7\sqrt{0.95(1{,}269 - 1{,}204)} = 1{,}760 \text{ ft per sec}$$

$$\text{Kinetic energy} = \text{mass} \times \frac{\text{velocity}^2}{64.34}$$

$$= 1 \times \frac{1{,}760^2}{64.34} = 48{,}200 \text{ ft-lb}$$

Thus for each pound of steam brought to rest from the velocity of 1,760 ft per sec, 48,200 ft-lb of work can be done.

The power developed will depend on the rate of flow in pounds per minute. For instance, if the rate of flow in the above jet was 1,000 lb per min then the power developed according to Eqs. (4) and (5) is,

$$\text{Power} = 1{,}000 \times \frac{48{,}200}{33{,}000} = 1{,}460 \text{ hp}$$

$$= 1{,}460 \times 0.746 = 1{,}088 \text{ kw}$$

As shown by Eq. (14) jet velocity depends on the steam enthalpies as controlled by the inlet and discharge pressures of the steam before and after the nozzle. Rate of steam flow through a nozzle depends on the specific volumes of the steam (depending on steam pressures) and the cross-sectional area of the nozzle. Flow varies in a very interesting and unexpected manner, as shown in Fig. 5-8*a*.

Suppose that we take a simple nozzle with longitudinal section as shown at lower right of Fig. 5-8a (converging nozzle) and start off with equal inlet and discharge pressures, no steam flow will take place. As the discharge pressure decreases, a steam flow takes place from the higher to the lower pressure. As shown by the solid curve on the graph, the

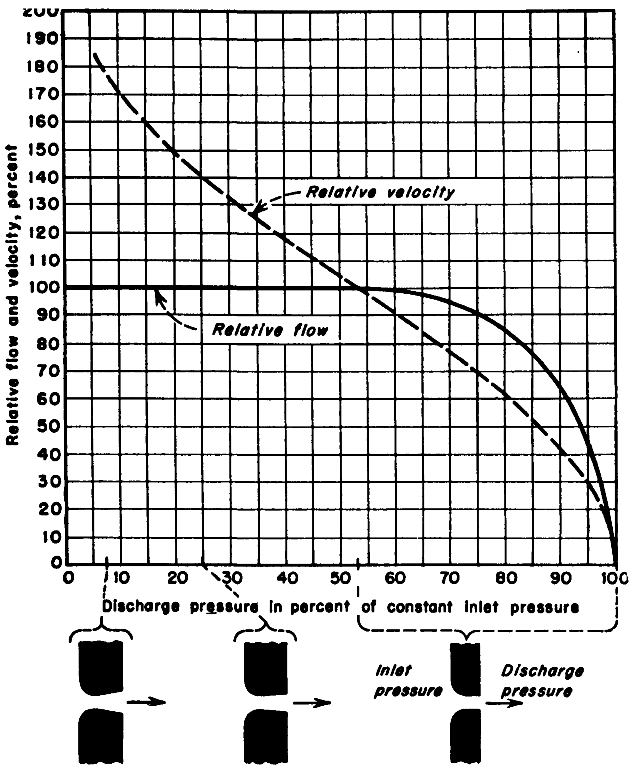


FIG. 5-8a. Nozzle cross section must be suitable for given inlet and discharge pressures. For a pressure ratio less than about 55 per cent, flow remains constant.

rate of flow increases rapidly at first but then begins to level off with lower back pressures until, at about a discharge pressure equaling anywhere from 53 to 58 per cent of the initial pressure, the flow remains constant regardless of how low the discharge pressure is reduced.

Now let's look at what happens to the steam velocity, which may be regarded as a measure of the energy output of the nozzle. The graph (Fig. 5-8a) shows that the velocity increases in a fairly uniform manner as the discharge pressure reduces, no matter how low. To utilize and develop the increasing velocities efficiently when the back pressure reduces below 53 to 58 per cent of the initial pressure, proper shaping of the nozzle has to be provided. A discharge section that flares out toward the nozzle exit has to be built into the nozzle. The exit section must be made larger as the discharge pressure decreases.

For exit pressures greater than 53 to 58 per cent of inlet pressure only converging nozzles need be used; for exit pressures less than this, converging-diverging nozzles must be employed. The section with the smallest cross-sectional area is known as the "throat" of the nozzle, and it's this throat area that controls the rate of steam flow, for given inlet and exit pressures.

If a converging-diverging nozzle is used for steam pressures different from those for which it is designed, the steam jet becomes adversely affected. If the discharge pressure increases above the design value, the steam will actually drop in pressure below that existing at exhaust, within the flare of the exit region of the nozzle, but rising to the pressure in the exhaust region as the jet leaves the nozzle. This sets up pulses and turbulence within the jet and lowers the Rankine efficiency of the nozzle.

For the converse condition where the back pressure drops below that for which the nozzle is designed, the steam will drop uniformly to the *design* pressure at the nozzle exit, then on leaving the nozzle the jet pressure will drop suddenly below the actual back pressure because of the abrupt expansion of the jet. The jet pressure fluctuates above and below the exhaust-region pressure and finally settles at that value. This instability of pressure again sets up turbulence, decreasing the nozzle efficiency.

The first condition may be described as overexpansion of the steam, and the second as underexpansion. Experiment shows that underexpansion tends to cause a smaller loss for equal percentage of departures of the nozzle exit area from that ideally required.

Nozzle shapes should ideally be round in cross section, because of the minimum surface involved for a given volume flow. This helps to cut down frictional losses. However, from a structural standpoint it often becomes less expensive to build nozzles with an approximate rectangular cross section; hence, most turbines are so built, as described in the chapter on constructional details, Chap. II.

It should be clear, now, that the nozzle functions to convert the steam heat energy into steam mechanical energy. What to do with the steam jet? If we let it just hit a flat-surfaced plate, the jet splatters, causes the molecules to interfere with each other, and results in internal frictional effects that waste part of the mechanical kinetic energy. But the jet would exert a force on this plate and could be made to do work.

Impulse Blading. A more efficient method is to direct the jet into a blade, as in Fig. 5-8. Here the jet enters one side of the semicircular groove smoothly and can be reversed in direction with little turbulence. This turning causes the jet to exert a force against the blade in the initial direction of the jet. Let's study what happens as the blade moves in respect to the jet.

First, assume that the blade is fastened in one place with zero speed; then the jet velocity V_2 leaving the blade equals the jet velocity entering, V_1. But since V_2 is in the opposite direction to V_1, we can say that $V_1 = -V_2$. Now the force developed on the blade can be found by

$$\text{Force, lb} = \frac{w(V_1 - V_2)}{32.17} \tag{15}$$

where w = steam flow rate of jet, lb per sec
V_1 = jet speed entering blade, ft per sec
V_2 = jet speed leaving blade, ft per sec

Proper account must be made of the directions of V_1 and V_2 in substituting. If they both have the same direction, they are both positive; if V_2 is opposite to V_1, it is negative.

Example 7: With the blade in locked position and a jet flow of 100 lb per sec, what force is developed with a jet speed of 500 ft per sec?

$$\text{Force} = \frac{100[500 - (-500)]}{32.17} = 3{,}116 \text{ lb}$$

However since the blade is locked and can't move, no work is done. Now suppose we let the blade move at the same speed as the jet, the extreme opposite condition of the locked-blade position. Obviously the jet can't catch the blade, no force can be developed, and the jet can't give up its kinetic energy to do work.

Now let's use a little imagination. When we let the blade move slowly compared to the jet, the velocity V_2 will become less than V_1 but still opposite in direction. Then, as we let the blade speed up but move more slowly than the jet, the velocity V_2 will become smaller till it reduces to zero. For further increases in blade speed direction of V_2 will be the same as the blade and V_1. Somewhere between zero and maximum blade velocity the magnitude of V_2 dropped from its maximum, with locked-blade position, to zero and then changed its direction to that of V_1 and the blade. Wherever V_2 becomes zero, it means that the jet has given up all its kinetic energy and transferred it to the blade, the ideal condition.

If you can't follow this line of reasoning, let's use vectors (Fig. 5-9). Here, the blade velocity is shown by vector V_b for speeds of zero, $\frac{1}{4}V_1$, $\frac{1}{2}V_1$, $\frac{3}{4}V_1$, and V_1. Now let V_1 be the jet speed leaving the nozzle and entering the blade at all five blade speeds shown. Now, when the blade moves, the speed of the jet relative to the blade is different from the jet speed relative to the nozzle or the earth (the nozzle isn't moving). The jet speed relative to the blade is $V_{1r} = V_1 - V_b$. Notice that the relative speed of the jet leaving the blade equals the relative speed entering the

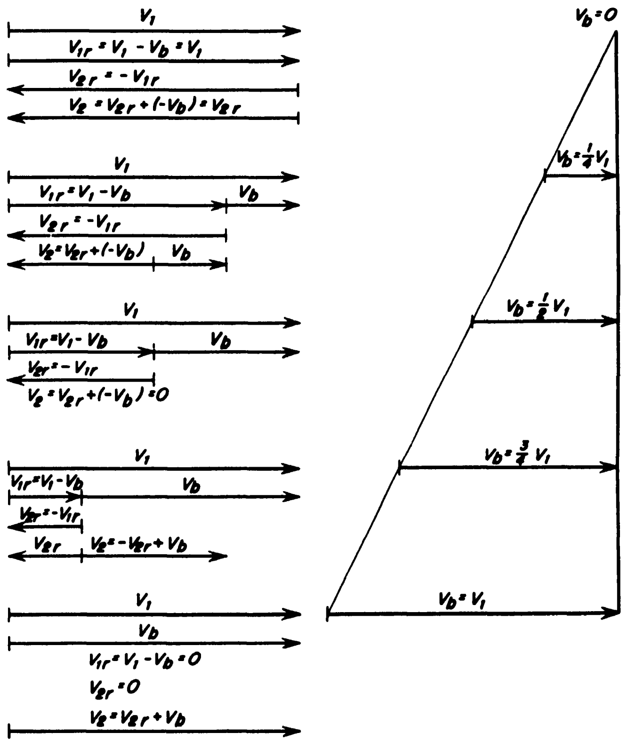


FIG. 5-9. Vectors provide an easy way to figure what relative velocity the blade and steam jet have for the ideal arrangement shown in Fig. 5-8.

blade, but the direction is opposite; then, $V_{2r} = -V_{1r}$. Now the relative jet leaving speed is the difference of the absolute leaving speed (relative to earth) and the blade speed, or, $V_{2r} = V_2 - V_b$. In using this last equation, proper account of direction must be made to determine the proper sign for V_b. If the direction of V_b is opposite that of V_2, then $V_{2r} = V_2 - (-V_b)$, or the relative velocity is the sum of the two values. As we use this relation in Fig. 5-9, $V_2 = V_{2r} + V_b$, again proper account must be taken of signs.

Here $V_2 = 0$ when $V_b = \frac{1}{2}V_1$; *i.e.*, when the blade speed equals one-half the initial jet speed, all the jet kinetic energy is given up in doing work on the blade. This is the ideal condition (see Fig. 1-6). Let's check on the amount of force developed for this condition.

Example 8: Similar to Example 7, let jet speed = 500 ft per sec, but the blade speed = ½ jet speed = 250 ft per sec. Then for a flow of 100 lb per sec, find the force developed, remembering that $V_2 = 0$ for this blade to jet speed ratio.

$$\text{Force} = \frac{100(500 - 0)}{32.17} = 1{,}558 \text{ lb}$$

We learned that power is the rate of doing work and can be written as,

$$\text{Power, hp} = \frac{\text{force, lb} \times \text{distance, ft}}{\text{time, min} \times 33{,}000} \tag{16}$$

But, $\dfrac{\text{distance}}{\text{time}}$ is a velocity or speed, so Eq. (16) can be rewritten as,

$$\text{Power, hp} = \frac{\text{force, lb} \times \text{velocity, ft per min}}{33{,}000} \tag{17}$$

Example 9: Find the power developed in the blade by the jet for the conditions outlined in Example 8

$$\text{Power} = \frac{\text{force} \times V_b}{33{,}000}$$

Now,

$$V_b = 250 \text{ ft per sec} = 250 \times 60 = 15{,}000 \text{ ft per min}$$

Then,

$$\text{Power} = \frac{1{,}558 \times 15{,}000}{33{,}000} = 708 \text{ hp}$$

Since the kinetic energy of the jet leaving the blade is zero, the jet power and blade power must be equal, or

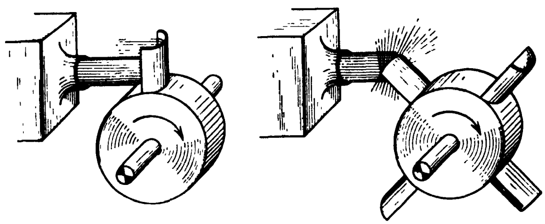
$$\text{Jet kinetic energy} = \frac{\text{velocity}^2}{64.34} = \frac{500^2}{64.34} = 3{,}890 \text{ ft-lb per lb}$$

$$\text{Jet power} = \frac{\text{kinetic energy} \times \text{flow}}{550} = \frac{3{,}890 \times 100}{550} = 708 \text{ hp}$$

Practical Blade Arrangements. So far we have talked about an extremely simple impulse blade and nozzle. This arrangement known as an "impulse stage" can always be identified by the steam pressure dropping through the nozzle but with the steam pressure of the jet remaining the same at the entrance and leaving edges of the blade. As the steam direction changes in the blade, there is, of course, a compression of the steam caused by centrifugal force. It is this momentary compression that produces the force acting on the blade.

Straight-line motion of the blade in Fig. 5-8 would be an ideal arrangement if it could be made to work. However, the most useful motion for any machine is rotation; the easiest way to transmit energy as work seems always to be with a rotating shaft.

This suggests mounting the blade on the perimeter of a wheel fixed to a shaft as in Fig. 5-10*a*. This will not be practical because of the intermittent action of the jet on the blade. For a large part of one revolution the jet would be blowing to waste. Mounting more than one blade on a wheel causes part of the jet to be deflected wastefully, as the blades move into best position (Fig. 5-10*b*). There's one modification of this arrangement that proves workable (Fig. 5-17), but one of the most serious limita-



(a) (b)

FIG. 5-10. (*a*) An ideal blade could be mounted on a wheel, but the jet would go to waste during part of the revolution. (*b*) Mounting more blades solves the problem, but the jet splashes wastefully as each blade moves into position.

tions comes from the relatively small capacity that can be developed with a machine using this nozzle-blade arrangement.

The vast majority of impulse turbines use the arrangement shown in Fig. 5-11. Here the nozzles are cut through the diaphragm at a very sharp angle, about 12 to 15 deg, from the line of motion of the blades. The blades are mounted on the periphery of a wheel in close order, filling the entire rim of the wheel. The blade working-surface shape differs little from the ideal, it being essentially an arc with straight steam entering and leaving edges, these edges not being parallel as in the ideal blade. The exact shape of the blade cross section results from the consideration of many factors, the principal ones being velocities of blade and speed.

The angle between the steam-jet and blade directions (Fig. 5-12) will be found to vary from about 10 to 30 deg, the angle chosen depending on such factors as compromising between a zero angle (the ideal) and the proper guidance of the steam jet into the blade passages. If the nozzle angle is made too small, the steam will not expand properly and will cause considerable turbulence and loss of energy. If the angle is too large, only a small part of the jet energy will be absorbed by the blading.

The blade must be shaped to allow smooth entry of the steam jet without shock. It must change the direction of the steam to create the working force, discharge the steam smoothly, and reduce its absolute velocity as near to zero as possible. To absorb all the jet's kinetic energy, the absolute leaving velocity would have to be zero, as mentioned before; but actually, since the steam flows continuously, there must be some velocity to move it. In Fig. 5-12 a typical vector analysis of blade and steam velocities is shown. The steam-jet velocity relative to the blade equals the vector difference of blade and absolute steam-jet speeds. Now, if the steam flows over the surface of the blade without friction, the leaving speed in feet per second will be the same as the entering speed, both relative to the blade. The directions will be different depending on the blade shape.

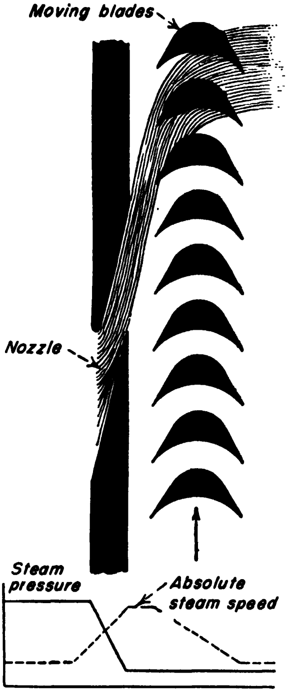


FIG. 5-11. In actual impulse-blade arrangement, steam enters the side of the blade after expanding through the nozzle. Steam pressure drops through the nozzle but stays constant through the blade. Velocity increases through the nozzle but drops through the blade.

The steam-jet absolute velocity is the vector sum of the steam speed relative to the blade and the blade absolute speed. Here the absolute steam velocity leaving the blade is almost at right angles to the direction of blade rotation. The right-angle direction of the leaving steam will be the optimum condition for this arrangement of blade and nozzle since the kinetic energy remaining in the steam jet will be a minimum for this angle. For a variable-speed turbine, the blade speed vector length will vary and cause all the other vectors to vary except the jet speed leaving the nozzle. For a turbine driving, say, an a-c generator, the blade speed will always be constant. As the load varies, the amount of steam in pounds per hour will vary and the jet speed leaving the nozzle will vary. This again will change these vector relations, and it will be evident that the nozzle and blade of a turbine stage are designed for one given condition only. Let's look at Fig. 5-13. Here the solid lines or vectors are the same as in Fig. 5-12, which we may call the "design condition." Now for a condition where the jet speed is only 75 per cent of the design speed (dash lines), the vector of jet speed relative to the blade is also shortened and the angle relative to blade motion increases. This means that the steam jet will hit the backs of the blades, at a glancing angle to be sure, but this will be enough to cause turbulence and friction within the steam jet, reducing the amount of kinetic energy that can be absorbed by the moving blade.

The relative direction of the steam leaving the blade is the same for all initial steam-jet velocities, because this is controlled by the blade

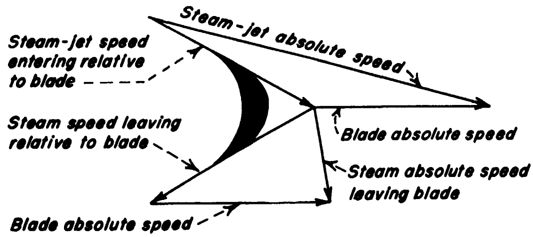


FIG. 5-12. Vector relations show that steam turns through an angle of almost 90 deg in passing through the blade and giving up part of its energy.

shape. The diagram assumes that the total values of the entering and leaving steam velocities relative to the blade are equal; actually, the turbulence caused at the entrance of the blade will make the leaving velocity

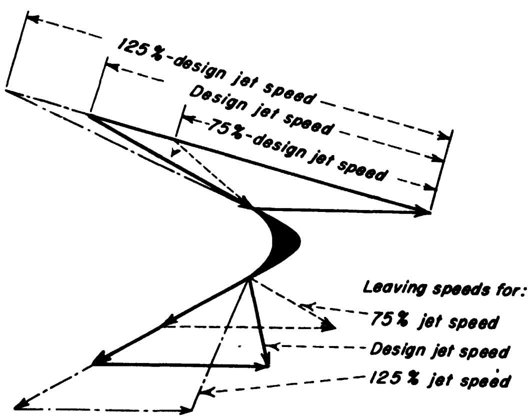


FIG. 5-13. Impulse-blade shape can be designed for only one jet speed at which it will operate with maximum efficiency. Vectors show variation in angles and velocities for different initial jet speeds and one given blade.

smaller than the entering velocity. However, neglecting this loss, notice the change in direction of the absolute steam velocity leaving the blade. Because of this the kinetic energy remaining in the steam is considerably larger than it should be and roughly equals the energy that would be left for the design condition. Hence the efficiency of the turbine stage drops appreciably below that at the design condition.

Now, examining the jet speed for 125 per cent of that of the design condition, we see that the steam does not enter the blade smoothly but impinges on the working surface abruptly, again setting up some turbulence. The absolute leaving velocity of the steam departs considerably from the ideal 90 deg, indicating an excess of kinetic energy over the optimum condition causing a drop in efficiency below that at the design condition.

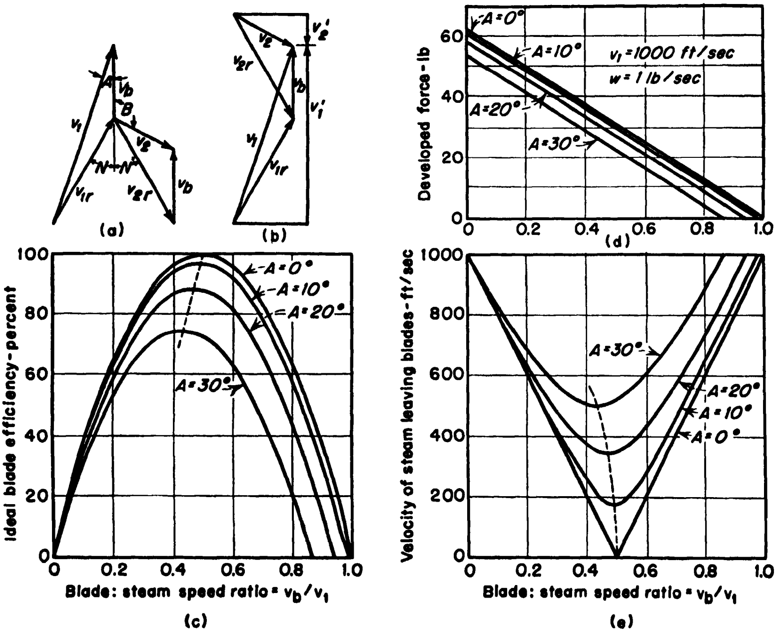


FIG. 5-14. Curves show how impulse-blade performance is affected by blade angles and relative blade and steam speeds.

There is obviously a limitation to what can be done with various arrangements of blade and nozzle angles and speeds. The behavior of ideal elements with a symmetrical blade (entering angle equaling the leaving angle) is summarized in Fig. 5-14* for a steam-jet velocity of 1,000 ft per sec and a flow of 1 lb per sec. These curves assume that the blade is properly shaped to receive the steam without shock for all ratios of blade to jet velocities.

When $A = 0$ deg, we have the ideal blade of Figs. 5-8 and 5-9. Notice that the developed force reduces from a maximum at zero blade speed to

* For derivation, see page 152, "Applied Energy Conversion," by Skrotzki and Vopat, McGraw-Hill Book Company, Inc., New York, 1945.

zero at jet speed, that the blade efficiency is highest of any angle arrangement and is 100 per cent at a blade speed exactly one-half the jet speed, and that the steam velocity leaving the blades is lowest of any other arrangement and zero at exactly one-half speed ratio, blade to jet. As the nozzle angle rotates in equal increments of 10 deg, the developed force and blade efficiency both drop with increasing rapidity and the velocity of the steam leaving the blade increases accordingly. The optimum ratio of blade speed to jet speed decreases from 0.5 with increasing nozzle angle.

Turbines for the generation of stationary power will be designed to operate most of the time at the optimum speed ratio. Since very few prime movers for this class of service always operate at full load, it will be found that the most efficient operating point will be at some part load, usually anywhere from about one-half to three-fourths of rated capacity, depending on the nature of the load curve.

Blade Compounding. To produce power at lowest cost, as much of the available energy in steam must be converted to work as possible. Looking at the Mollier chart (Fig. 5-6), we can see that increasing energy becomes available as the exhaust pressure decreases. Then according to Eq. (14) and Fig. 5-8a, the velocity of steam flowing through a nozzle steadily increases with lowering back pressure. For high initial pressures, the jet velocities become enormous. This requires the blade speed to be high for the proper absorption of the jet kinetic energy.

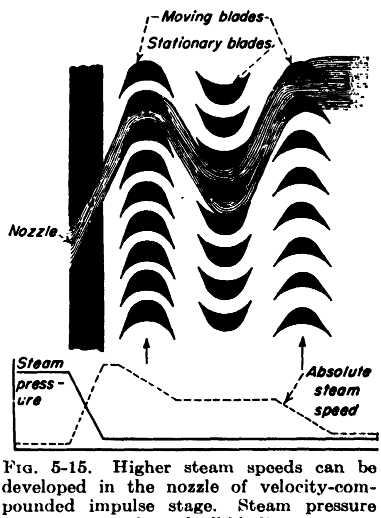


FIG. 5-15. Higher steam speeds can be developed in the nozzle of velocity-compounded impulse stage. Steam pressure stays constant through all blading.

Since the blades are mounted on the rim of a wheel, this in turn requires a high wheel perimeter speed and high turning speed. This can cause the centrifugal force acting on the wheel and blades to exceed the safe stresses that can be withstood and result in a literal explosion of the wheel. It becomes necessary then to develop methods that allow the use of maximum energy in the steam but keep the rotor speeds within mechanical limitations. These methods, called "compounding," are (1) velocity compounding and (2) pressure compounding.

In velocity compounding, the nozzle develops a high steam-jet velocity (Fig. 5-15), but the moving-blade speed is much less than half the jet speed. As shown in the diagram, a velocity-compounded stage consists of a set of nozzles followed by one row of moving blades, a second row of stationary blades, and a third row of moving blades. As in all impulse-blade stages, the steam pressure remains constant from the nozzle exit through all the blading to the stage exhaust. Steam-flow velocity increases in the nozzle, drops partly in the first row of moving blades, remains essentially constant in passing through the stationary blades, and then drops to a low value in passing through the last row of moving blades. Notice the increasing flatness of the blade shapes that the steam passes

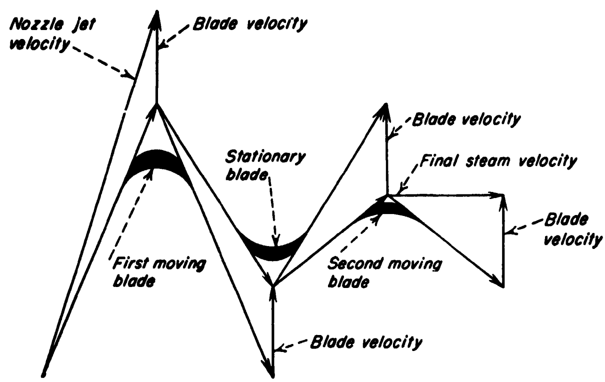


FIG. 5-16. Vectors show the reason for increasingly flatter shapes of blade cross sections in velocity-compounded impulse-turbine stage.

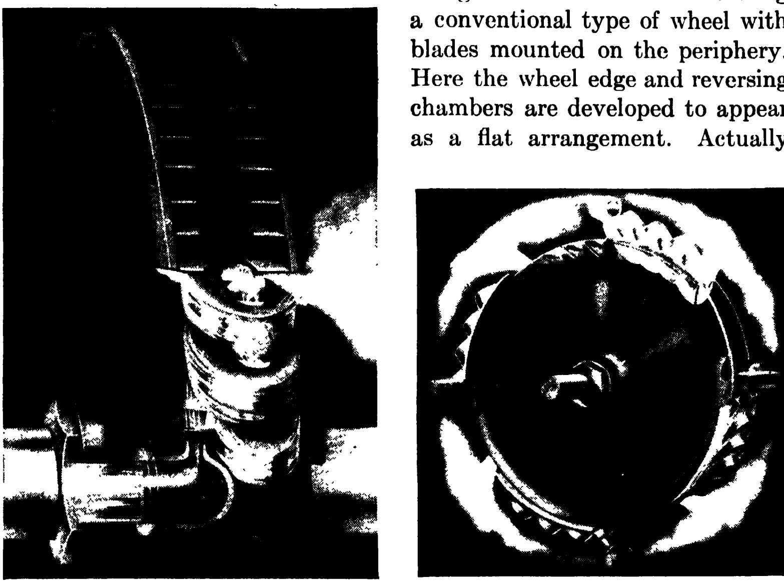
through successively. A study of the vector diagram (Fig. 5-16) shows the reason for the varying blade contours. Here the moving-blade velocities, which are identical for the two rows, are about one-fourth of the initial jet velocity.

The stationary blades serve only to redirect the steam into the succeeding row of moving blades; they do not accelerate the steam as nozzles do. The velocity relationships are such that the total change in steam direction becomes smaller for succeeding blade rows. This results in increasingly flatter blades.

A velocity-compounded turbine has the advantages of compactness and lower cost, but tends to operate at a lower Rankine or engine efficiency. In the majority of high-capacity turbines the first stage, being velocity-compounded, permits the use of a large steam-pressure drop at the front end, reducing the length of turbine rotor and casing subjected to h-p high-temperature steam.

Velocity compounding can also be done by passing the steam two or more times through a single row of moving blading. Small turbines utilize this principle in two different arrangements. Figure 5-17 shows one form where the blades are milled as semicircular chambers into the perimeter of the wheel. Steam jet leaving the nozzle enters the moving blades, makes a 180-deg turn, and enters a stationary blade or chamber immediately adjacent to the wheel. The steam makes a total of three reversals before exhausting into the casing surrounding the rotor.

Figure 5-18 shows a turbine using a conventional type of wheel with blades mounted on the periphery. Here the wheel edge and reversing chambers are developed to appear as a flat arrangement. Actually this takes up about half the diameter of the wheel. The steam makes three passes through the one set of blading before exhausting from the unit.



(*Courtesy of Terry Steam Turbine Co.*)

Fig. 5-17. One arrangement of velocity compounding using one blade row in a single-stage turbine, with specially shaped blades in wheel perimeter.

In pressure compounding the steam pressure reduction takes place in two or more steps. The group of equipment, nozzle and moving blades (Fig. 5-11), constitutes an impulse stage. Pressure compounding simply consists of placing two or more impulse stages in series (Fig. 5-19) so that h-p steam from the boiler enters the first nozzle and blading. The steam exhausting from the first-stage blading then enters the second-stage nozzles and passes through its blading. The steam subsequently enters the next stage, and so on until it exhausts at the final pressure. In this way the acceleration of the steam in each nozzle can be limited to values that can be readily handled by the blading.

Reaction Blading. When steam increases its velocity by flowing through a nozzle, an unbalanced pressure condition is set up. These unbalanced pressures exert a force that can do work. This situation is very aptly illustrated by a rubber balloon that has been blown up with air and then released to travel freely without closing the inlet. As long as any air under pressure remains in the balloon, it travels on a swift and erratic course.

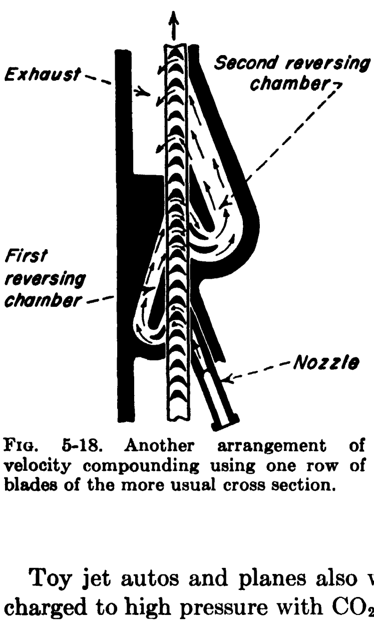


FIG. 5-18. Another arrangement of velocity compounding using one row of blades of the more usual cross section.

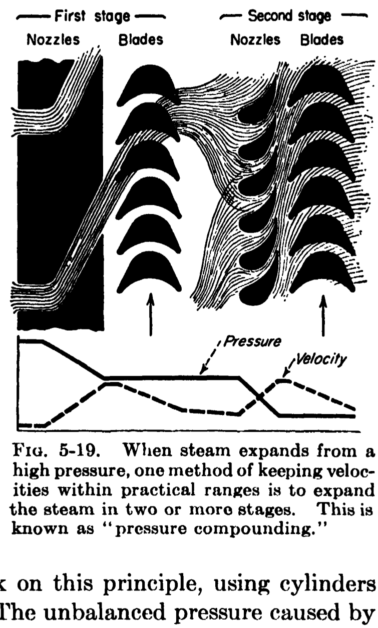


FIG. 5-19. When steam expands from a high pressure, one method of keeping velocities within practical ranges is to expand the steam in two or more stages. This is known as "pressure compounding."

Toy jet autos and planes also work on this principle, using cylinders charged to high pressure with CO_2. The unbalanced pressure caused by releasing the gas through a suitable opening develops a force that sends the toy traveling at a high speed. Recoil in a gun is another example of unbalanced pressures developing a force.

Figure 5-20 shows how the unbalanced force would be created in a cubical box fed by steam from a h-p line. When the box is filled with steam as in (*a*), pressures are equal on all sides and no net force tends to move the box. The steam pressure in the box balances the pressure in the steam inlet line. However, when a nozzle opening is made in the box (*b*), the steam will flow out and there can be no pressure on that area of the box wall removed for the nozzle opening. Then the pressure on the opposite wall will be greater by the amount acting on this area. For instance, suppose that the area of each wall equals 10 sq in. and that the steam pressure is 120 psi. Then each wall will have a total pressure or force of $120 \times 10 = 1{,}200$ lb tending to push it out. If the wall with the nozzle has 1 sq in. removed for the opening, the wall will only have a total pressure of $120 \times 9 = 1{,}080$ lb acting on it. Then the box will have a net force of $1{,}200 - 1{,}080 = 120$ lb pushing it to the left, as indicated by the heavy arrow.

Note that the box movement to the left *does not* result directly from the jet flow to the right; *i.e.*, the jet *does not* push on the box from which it leaves. Rather the jet flow and the force are both caused by steam pressure only. Because of this unbalanced force, all turbines must be

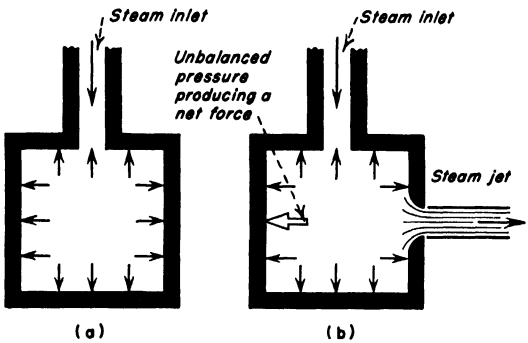


FIG. 5-20. (*a*) Steam under pressure in a closed chamber exerts equal forces in all directions. (*b*) A steam jet issuing from the chamber exerts an unbalanced reactive force in a direction opposite to that of the jet movement.

firmly anchored to their foundations. In the impulse turbine the force that can be developed by the steam jet is the only one utilized to do work. The force that can be developed by bringing the jet to rest, of course, equals the unbalanced force within the box or nozzle bowl.

This unbalanced force is commonly called a "reaction force." Isaac Newton many years ago discovered the law that stated for every action there must be an equal and opposite reaction. Thus the pressure that produces the steam flow also produces the reactive force. As far as creating mechanical work is concerned, it is immaterial whether the nozzle or the jet moves. If the steam connection in Fig. 5-20*b* were flexibly connected to the nozzle box or bowl, the box could be made to move relative to the earth and the jet issuing from the nozzle would be absolutely stationary relative to the earth. Figure 5-21 shows simplified versions of the use of such a jet in an impulse turbine or as a reaction turbine. In the latter case the steam could be led into the box through a hollow shaft. But this simple version of a reaction turbine does not prove practical for larger capacity units, and we find different designs for practical turbines. In general the external appearance of reaction blading looks similar to impulse-turbine blading; however, the cross section of the blading has a distinctive shape of its own.

Figure 5-22 shows the general velocity diagram of one stage of reaction blading. The stationary blading really makes up the stationary nozzles

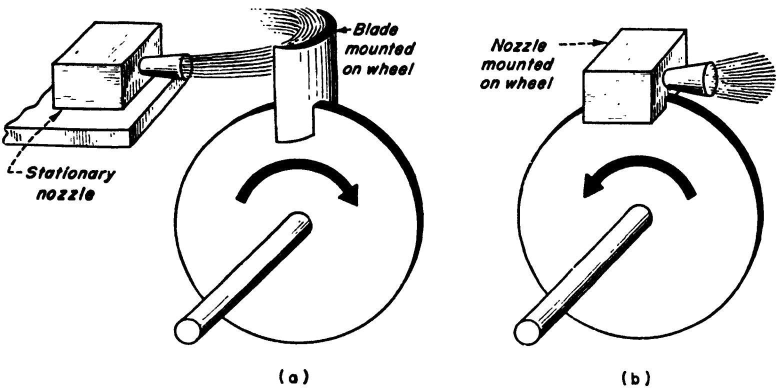


FIG. 5-21. Comparison of steam jet used in an impulse turbine (*a*) and in a reaction turbine (*b*).

having shapes similar to the nozzles of pressure-compounded impulse turbines for the later stages (see second-stage nozzles of Fig. 5-19). Steam pressure drops through both stationary and moving blades in reaction turbines, in contrast to impulse turbines in which pressure drop

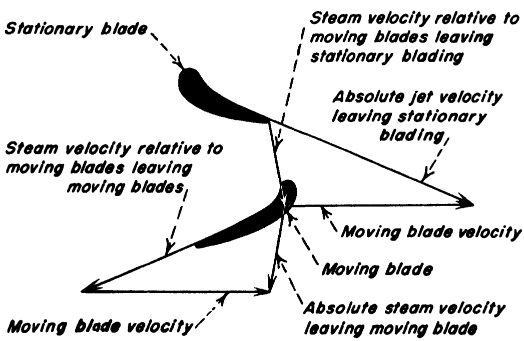
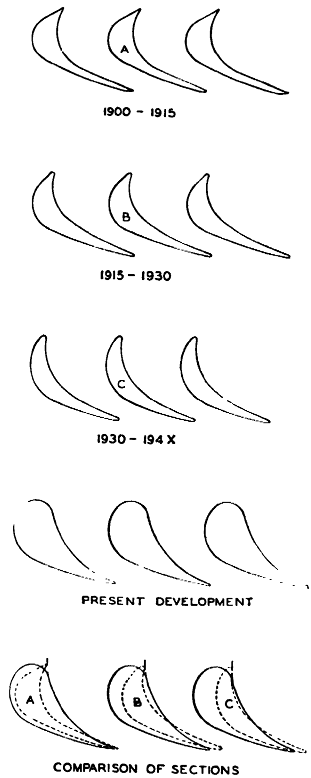


FIG. 5-22. Vector relations of steam and blade speed in a reaction-turbine stage.

takes place only in the stationary nozzles. The steam accelerates to a velocity somewhat greater than the blade velocity. This makes the steam velocity relative to the moving blade comparatively small, and the steam enters the moving blades at a direction almost at right angles to the direction of blade movement. The steam pressure drop through the moving blading accelerates the steam so that its leaving velocity is considerably greater than its entering velocity, both relative to the moving blade. This acceleration of course is accompanied by a reactive force which produces the blade motion.

When the blade-velocity vector is subtracted from the leaving-steam velocity (relative to blade), the absolute steam velocity leaving the moving blade is small and practically at right angles to the blade direction. As in impulse staging, small final leaving velocity of the steam is indicative of efficient energy conversion from heat in the steam to work on the turbine blading. Early designs of reaction blading had the entering edge of the blades tapered down to a knife-edge. Present designs use a rounded entering edge as this gives better part-load performance because of smoother steam entrance (see Fig. 5-23).



(Courtesy of Westinghouse Electric Corp.)
FIG. 5-23. Modern reaction-turbine blading has rounded streamlined inlet edge.

Comparing the impulse stage of Fig. 5-12 with the reaction stage of Fig. 5-22 it will be seen that for the same moving-blade speed the steam speed leaving the nozzle in the impulse turbine is much greater than the steam speed leaving the stationary blading in the reaction turbine. To satisfy this relationship it is apparent that the steam pressure drop through the reaction-turbine stationary blading must be less than for the impulse-turbine nozzle. However, the pressure drop through the moving blading in reaction turbines causes steam leakage between the blade tips and the casing. This loss is minimized both by reducing the clearance between blade tip and casing and by keeping the pressure drop as small as practicable. This results in the reaction turbine having a much greater number of stages than a comparable impulse turbine of the same capacity operating between the same pressures. The greater number of stages in a reaction turbine makes it more practical to mount them on a drum instead of on individual disks as in an impulse turbine. This construction causes a larger peripheral area open to steam leakage to appear between the stationary blades and the drum, but the smaller pressure drop across stages counteracts this factor. Owing to the lower pressure differentials, the steam velocity is lower and a lower optimum blade speed may be realized than for the impulse turbine. This is shown in Fig. 5-24, where the

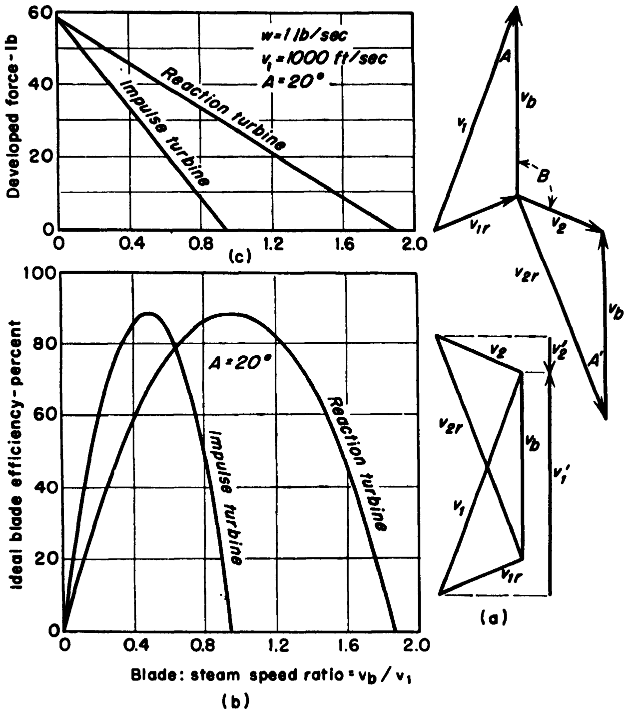


FIG. 5-24. Blade performances compared for impulse and reaction blading.

developed force and ideal blade efficiency are compared for reaction and impulse turbines.

Figure 5-25 shows the variation in steam pressure and in steam absolute velocity as it passes through two stages of reaction blading. The flowing steam fills all the blades in a reaction turbine. In a pure reaction turbine this means that the only means of controlling its output is to raise and lower the pressure of the steam entering the first-stage blading as a result of opening and closing the governor-controlled inlet valve. However, better steam control with more efficient utilization of steam energy at low loads can be secured by the partial admission arrangements used on the first stage of impulse turbines. In addition, by using a velocity-compounded impulse stage for the first one in a turbine, the length of the unit can also be appreciably shortened. Hence, it will be found that most reaction turbines use such a stage as the initial one. Such an arrangement is shown in Fig. 5-26, only two of many reaction stages being included.

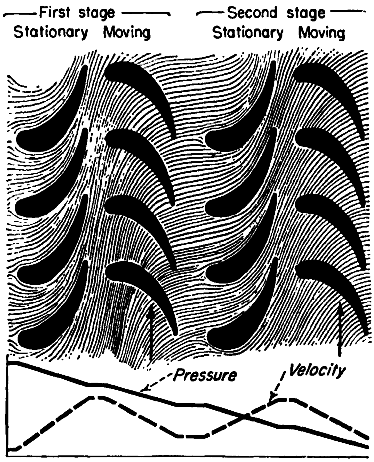


FIG. 5-25. Steam pressure drops in both stationary and moving blading of a reaction turbine.

Flow and Load Control. In the vast majority of cases the demand for energy varies over a considerable range. Turbines must be built to produce energy in the form of a rotating shaft in any amount that will be needed. Since 1 lb of steam will produce just so much energy, the obvious way to control output will be to regulate the rate of steam flow through the turbine.

Since the main steam pressure and the turbine exhaust pressure are usually held constant, the best way to control steam-flow rate is by varying the cross-sectional area at some point in the supply pipe through which the steam flows. A valve performs this function and would be inserted between the steam supply line and the nozzle bowl or steam chest of the turbine, as shown in Fig. 5-27*a*. This diagrammatic arrangement shows the principal elements of a simple single-stage impulse turbine. With constant steam pressures at supply and exhaust, Fig. 5-27*b* shows how steam flow varies with valve opening area. However, whenever the valve is only partly open, the steam pressure on the downstream side of the valve will be lower than the supply pressure. Figure 5-27*c* shows the variation of this pressure with valve opening area. This means that the pressure in the steam chest or at the nozzle inlet varies directly with steam flow (Fig. 5-27*d*).

A variation in pressure entering the nozzle causes a change in the amount of work that can be done by each pound of steam. Figure 5-27*e* shows what happens in a turbine with a 425-psia supply pressure and an atmospheric or 14.7-psia exhaust. When the valve is wide open, full, or 100 per cent, flow passes through the turbine, there usually is some pressure drop through the valve, in this case a 25-psi drop, and the steam expands from 400 to 14.7 psia in the nozzle. The work done per pound of steam is then the difference of the initial and exhaust enthalpies, at the beginning and end of the condition line shown as 100 per cent on the Mollier chart. When the valve opening is closed down so that only 75 per cent flow passes, the pressure after the valve drops to 310 psia. Notice that the steam enthalpy stays constant with this reduction in pressure. Dropping steam pressure, without making it do work, does not

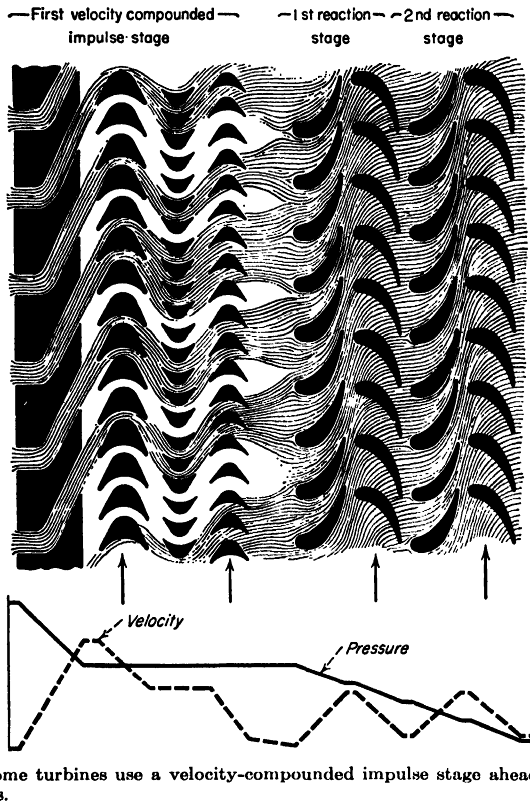


FIG. 5-26.—Some turbines use a velocity-compounded impulse stage ahead of a series of reaction stages.

change its energy content. Such a process is called "throttling," and the control valve often is called a "throttling valve."

When the steam expands in the nozzle from 310 to 14.7 psia, it follows a different condition line on the Mollier chart and the steam exhaust enthalpy is higher than at full 100 per cent flow. The work done by the steam is less per pound because of the shifting of the condition line to the right. Thus as the steam throttles to lower and lower pressures, the work done per pound becomes smaller. For a single-valved turbine of this type the theoretical maximum efficiency occurs at full steam flow because each pound of steam can do a maximum amount of work at this condition. However, the actual efficiency depends on the load for which the nozzle and blade are designed. If the turbine will operate mostly

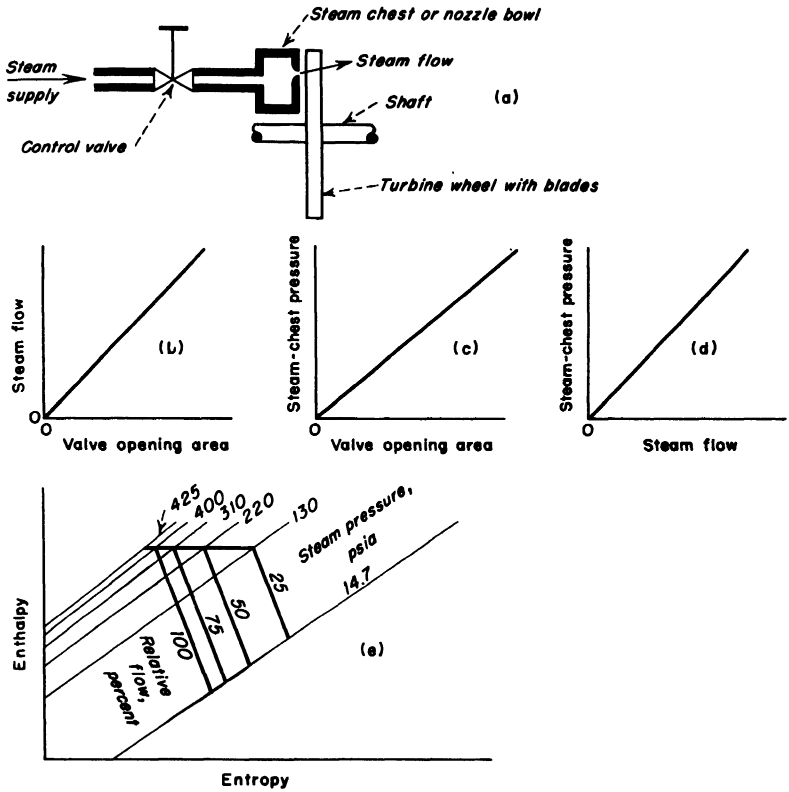


FIG. 5-27. In a turbine controlled by a single valve, steam pressure varies with the steam flow, and the condition line becomes shorter with smaller flow.

at about 75 per cent steam flow or load, the blade efficiency will be a maximum at this load. At higher loads, the steam enters the blade at too abrupt an angle (see Fig. 5-13), and the leaving velocity loss is higher than optimum because the blade cannot absorb all the steam velocity energy efficiently at the higher steam speed. These two effects generally overbalance the higher work available per pound of steam, with the result that the efficiency at maximum flow will be lower than at the design point.

One of the fundamental ways of defining turbine performance is by means of the input-ouput curve (Fig. 5-28), where the steam flow is plotted against the output. Steam flow usually is expressed in pounds per hour and the output either in horsepower or kilowatts. The curve shows that somewhere between 10 to 20 per cent of full load, steam flow is needed to operate the turbine at zero load, because of various losses. As load increases, the steam flow increases at almost a straight-line relation. This line is called a "Willans line." For many design and estimating purposes, the line is assumed to be straight, but in actual turbines it is slightly concave upward, as shown, for a single-valved turbine.

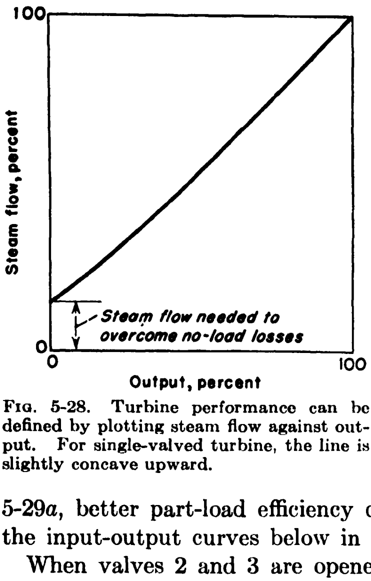


FIG. 5-28. Turbine performance can be defined by plotting steam flow against output. For single-valved turbine, the line is slightly concave upward.

The more steam that can be made to flow through an impulse wheel of a turbine, the more the work that can be produced. Thus, instead of having just one nozzle for a stage as in Fig. 5-27*a*, two nozzles could be provided as in Fig. 5-29*a*. In fact in large central-station turbines, the entire periphery of the wheel has nozzles preceding it that send steam through all blades simultaneously at full load. By providing individual valve controls on the nozzles in Fig. 5-29*a*, better part-load efficiency can be obtained. This is shown by the input-output curves below in Fig. 5-29*b*.

When valves 2 and 3 are opened wide and steam flow is controlled only by valve 1, the steam-flow curve will be the upper one in the graph. A different schedule of valve operation can be followed, however, that will produce better efficiency. Eliminating or opening valve 1 wide will do this, thus allowing valves 2 and 3 to control steam flow through the nozzles individually. For all loads of less than half capacity, only one nozzle is brought into operation, the valve of the other nozzle being shut tight. When more than half capacity is needed, one nozzle carries its full steam flow; *i.e.*, its valve is wide open, and the second nozzle controls the variation in steam flow through its valve. The curves show that at all loads less than full capacity, the turbine operates at better efficiency with individual nozzle control.

Figure 5-30 shows how the pressures in the nozzle bowls or steam chests vary for each of the methods of control. With the single-valve control the pressures in both nozzle bowls are equal at all loads. At zero output or load, the pressure is somewhat greater than the exhaust pressure from where it steadily increases to a value slightly less than the supply pressure at full load. With individual nozzle control, while the first nozzle-bowl pressure increases from a minimum at zero load to slightly less than supply pressure at 50 per cent load, the nozzle-bowl pressure of the

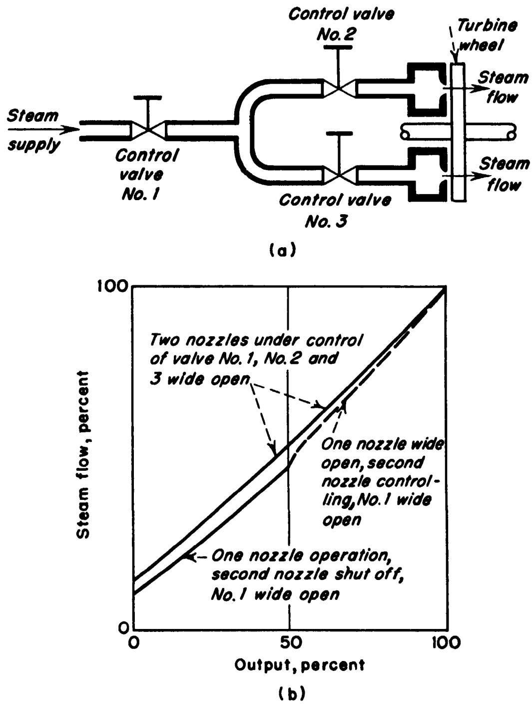


FIG. 5-29. Better part-load efficiency can be realized in a turbine by controlling the steam flow through two or more nozzles separately.

second nozzle remains equal to the stage pressure, which for a single-stage turbine equals the exhaust pressure. When the first nozzle valve is fully opened at 50 per cent load, the second nozzle valve opens to admit steam with resulting increasing bowl pressure until it also is fully open at 100 per cent load. While the second nozzle valve is in the process of opening, the first nozzle valve remains fully open and passes its maximum amount of steam through the turbine wheel from its nozzle.

Individual nozzle control proves more efficient because a greater portion of the total steam flow is expanding through a greater pressure range, allowing each pound of steam to develop a greater amount of work on the average.

Multistage-turbine pressures behave somewhat differently for throttling and nozzle group control. Figure 5-31*a* shows a five-stage turbine, the first stage with an impulse wheel, and the last four being reaction stages. A pressure-compounded five-stage impulse turbine will behave in the same way in regard to steam pressure as discussed in the following. In simple throttle control, valves 2 and 3 will be eliminated. Let's consider them as wide open at all times. Again the steam flow is proportional to the area opened in valve 1, and the pressure ahead of the first-stage nozzles will be proportional to the steam flow, as long as the supply pressure and the exhaust pressure remain constant. At any one flow the pressure successively drops in the turbine stages as the steam passes on to exhaust. The steam always fills all the space in each stage. As a result, it is found that the pressure ahead of the nozzles in each stage varies proportionally with the steam flow, as shown in Fig. 5-31*b*.

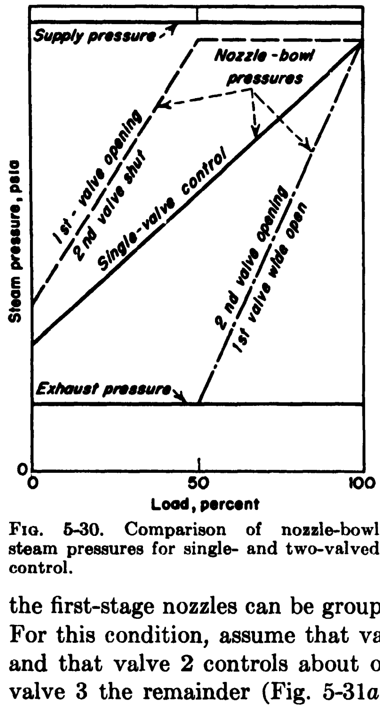


FIG. 5-30. Comparison of nozzle-bowl steam pressures for single- and two-valved control.

In order to reduce the throttling loss, as with the single-stage turbine, the first-stage nozzles can be grouped and controlled by separate valves. For this condition, assume that valve 1 remains wide open at all times and that valve 2 controls about one-half of the first-stage nozzles and valve 3 the remainder (Fig. 5-31*a*). Valve 2 first opens while 3 stays shut. As valve 2 opens, the second-stage steam pressure increases. The steam backs up into the nozzles controlled by valve 3 so that the pressure on the downstream side of valve 3 is the same as the second-stage pressure. When valve 2 has opened fully, the turbine is developing about half its rated output. Pressure drop usually occurs between the steam inlet to the chest and the nozzle bowl as indicated on both (*b*) and (*c*) of Fig. 5-31; and hence, the pressure immediately after valve 2 is slightly less than supply pressure when the valve is wide open. Valve 3 then opens, and the nozzle-bowl pressures of the group it controls increase. When it is wide open, the first-stage nozzle pressures are all equal, and the turbine will be passing the maximum amount of steam and developing its maximum possible output.

Other methods of control and valving are used which increase the output of turbines. One passes steam from the supply or steam chest into the second stage directly. This raises the pressure of all the succeeding stages so that they can pass more steam and develop greater output. However, this usually is done at a falling efficiency. Some designs have an internal by-pass valve which allows steam to pass from the first to the third stage.

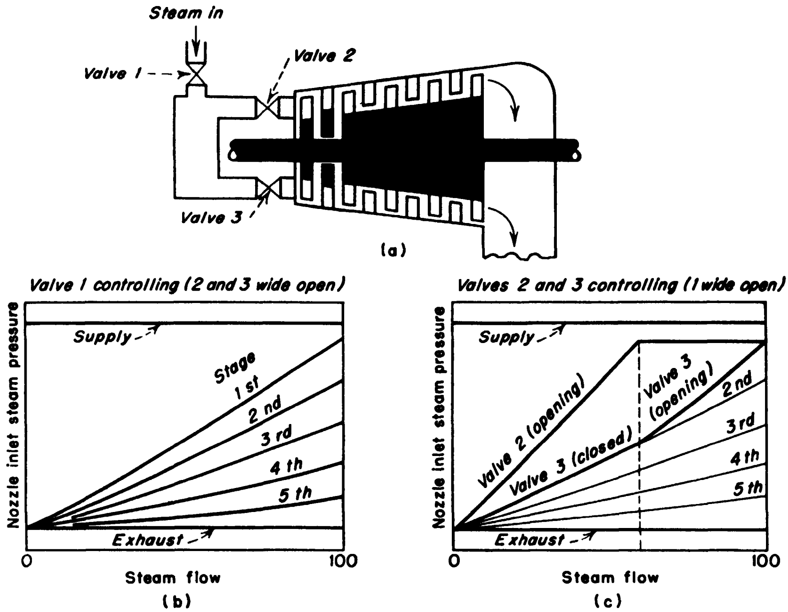


FIG. 5-31. In multistage turbines, the steam pressure increases in all stages proportionally to the steam flow through the turbine.

Turbine Losses. The most serious loss occurring in turbines has already been mentioned in connection with Fig. 5-7. This is the inability of steam to pass through nozzles and blades and fully develop the mechanical energy from the available heat energy or enthalpy. Friction between the steam and walls of the nozzles and internal friction of the steam molecules within the steam stream cause this loss. It results in only part of the energy appearing in mechanical form, the remainder stays within the steam as heat.

Other losses act to reduce the over-all efficiency of turbines. At part loads there is throttling, which has been discussed in the preceding section. Throttling further reduces the available energy of the steam.

Steam leakage causes some of the steam that might do useful work in the blading to follow some undesirable path other than the intended one.

Steam can leak along the shaft from the first stage toward atmosphere. Usually this flow is throttled by special packing described in Chap. 2, but this minimum flow may be led to a heater or to the condenser. In impulse turbines steam leaks between stages through the diaphragm packing. This flow is relatively small because the cross-sectional area of the clearance in the packing around the shaft is small. In reaction turbines there is leakage past the tips of both stationary and moving blades because of the pressure difference existing on both sides of the blades. The large cross-sectional area between moving blade tips and casing and between stationary blade tips and drum is compensated by the lower pressure differentials which keep leakage to tolerable values.

In impulse turbines the blades are mounted on disks. These disks revolve in chambers formed by the diaphragms carrying the nozzles for their own stage and the following one. These chambers are filled with steam, and the friction between revolving disk and the steam constitutes a windage loss. An impulse stage operating at partial admission, or an early stage in such a turbine with nozzles provided over only part of the blade periphery, will have blades idle during part of the revolution. At such times these blades churn the steam in the chamber and cause an appreciable loss.

Bearing losses carrying the turbine rotor are generally small but measurable. They heat the lubricating oil, the heat energy being dissipated in water-cooled oil coolers.

The larger turbines with water-sealed glands use an amount of work to make these effective. The water seals are really small centrifugal pumps which need work input to operate.

A small amount of energy is lost from the front high-temperature end of the turbine as radiation to the cooler surroundings. Hence the front end is always insulated. The exhaust end of condensing turbines, usually being only slightly above room temperature, does not need heat lagging.

The steam leaving the last stage of the turbine has a certain velocity which represents an amount of kinetic energy that cannot be imparted to the turbine shaft. The steam ultimately slows down and causes a rise in enthalpy at the exhaust. This energy is lost as far as turbine output is concerned and appears as heat in the steam.

Finally the steam leaving the last rows of blades has a pressure which must be higher than the pressure at the exhaust flange, to ensure the steam flowing out in the proper direction. The amount of pressure difference depends on the design of the exhaust hood and the area provided for steam flow. The larger the area, the less the pressure drop caused by the steam flow.

CHAPTER 6

PERFORMANCE

In Chap. 5 we discussed the way in which steam does its work inside a turbine. It should be evident that nozzles, blading, clearances, valves, etc., must be in proper adjustment to enable the turbine to utilize the steam to best advantage. If any of these features depart from the design condition because of wear, accident, or foreign material deposited within the turbine, more steam will be needed to do a given amount of work. Since fuel represents a large portion of the total cost in running a power plant, it becomes important to use the steam it generates to maximum advantage.

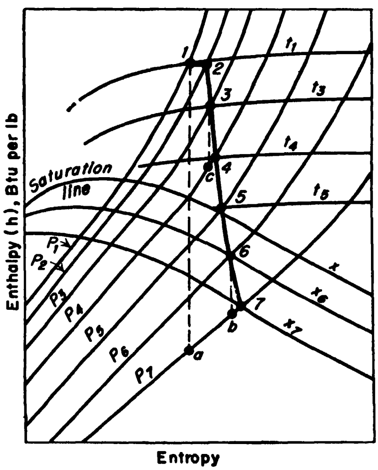


FIG. 6-1. Condition line of a five-stage turbine running at rated capacity. The last two stages operate with wet steam.

Condition Line. The most complete method of expressing a turbine's performance would be to plot its condition line on a Mollier chart, as indicated in Fig. 6-1, for given developed outputs. To do this adequately, steam pressure and temperatures in each stage would have to be known and, in the lower pressure stages where the steam is wet, the moisture or quality. Since it takes time and special instruments to gather all this data, it is not a practical method to use in following day-to-day operation, but it is invaluable in analyzing the condition of a turbine, especially where trouble is suspected to be developing.

Figure 6-1 shows a condition line for a five-stage impulse turbine operating at rated capacity. Point 1 is found from the pressure p_1 and the temperature t_1 of the steam supply. Point 2 is located by the nozzle-inlet pressure of the first stage, immediately after the governing valve. Since the steam throttles from the main to the nozzle inlet, the enthalpy remains constant; hence point 2 lies to the right of point 1 on the chart. Point 3 is located by the pressure and temperature of the steam in the first-stage blades. This is also the condition of the steam in the nozzle inlet of the second stage. Points 4 to 7 are, respectively, the steam condition in the second, third, fourth, and fifth stages. The fifth-stage pressure is also the exhaust pressure, point 7. To find points 6 and 7 some type of calorimeters must be used because the steam is wet, the temperature of the steam is the saturation temperature and will be of no help in finding the enthalpy of the steam. If the pressure at point 7 is considerably below atmospheric, the quality can usually be calculated only by special methods not described in this book.

If trouble should develop, the condition line determined by test will not lie in the same position as the designed value for the given load; it will generally be somewhat to the right of the correct position on the chart. If trouble should be developing in one intermediate stage only, the condition line is likely to show a jog to the right for the steam condition in that stage.

Efficiencies. As described in Chap. 5 the Rankine efficiency of a steam engine or turbine is the ratio of the actual work developed to the available energy. This ratio is given special names when applied to given sections of a turbine. When determining the performance of a single stage in a turbine, it is called the "stage efficiency." For the second stage of the turbine in Fig. 6-1 the stage efficiency is $\frac{h_3 - h_4}{h_3 - h_c}$; for the last stage, the stage efficiency is $\frac{h_6 - h_7}{h_6 - h_b}$.

When the efficiency of the turbine as a whole is determined, the Rankine efficiency is called the "internal efficiency" and for this unit will be $\frac{h_1 - h_7}{h_1 - h_a}$.

From the operator's standpoint the most useful efficiency expression takes into account all the losses; for instance, in the case of a turbine driving an electric generator, the Rankine efficiency is called the "engine efficiency," or

$$n = \frac{h_1 - h_7}{h_1 - h_a} \times e_t \times e_g$$

where e_t is the turbine mechanical efficiency and e_g the generator efficiency.

Heat Rate. The basic performance term is the "heat rate" which expresses the amount of energy input to the turbine for each unit of energy output. Its determination is fairly simple and provides a quick means of checking turbine performance. Let us see just what the *heat rate* means. To run a turbine we must supply steam from a steam generator or boiler, as in Fig. 6-2. A condensing steam turbine expands this steam, extracting part of the available energy and exhausting the steam in a wet condition to a condenser. The condensed steam or water is then returned to the boiler by the pump. Evidently, the boiler must add enough heat to this water to reevaporate it to steam so that it can again pass through the turbine and do work.

At first glance it may seem silly to condense the turbine exhaust, since all of this heat plus that converted to work must be replaced by the boiler. It might be thought that it would be more logical to recompress the cool exhaust steam and then reheat it to its original temperature. If that were done, more work would be needed for recompression to original pressure than the turbine produced by expanding the steam. All that

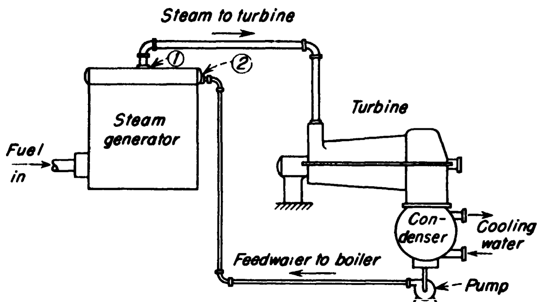


FIG. 6-2. The simplest turbine cycle indicates the method of computing turbine energy input. The turbine is charged with energy the boiler must supply between points 1 and 2.

would be done in the ideal case (if all the available energy in the steam were developed) is to pass this energy to the turbine shaft and then back to the steam to recompress it, with no work left over for useful purposes. It might be said that the "secret" of any heat-work cycle is to throw away the unavailable energy, the heat in the exhaust steam. For given conditions this heat cannot be converted into useful work by any means. In Fig. 6-2 the condenser throws away this unavailable energy; if it does its job properly, it removes only the enthalpy of vaporization and no more; *i.e.*, it returns the water to the boiler at a temperature equal to that of the exhaust steam.

With this conception of a simple steam-turbine cycle, it can be seen that the turbine needs all the heat in the steam less the amount it returns to the boiler in the water. In turn, the boiler must furnish this energy by reevaporating the water. Hence the input to the turbine per pound of steam is the difference in enthalpy at points 1 and 2 in Fig. 6-2. For a given output through the turbine shaft there will also be a certain steam flow through the turbine and boiler cycle. All these relationships can be expressed by a single figure or ratio, the heat rate.* Fundamentally,

* For a complete list of formulas for all turbine types see ASME Power Test Codes for Steam Turbines.

$$\text{Heat rate} = \frac{\text{input}}{\text{output}}$$
$$= \frac{\text{steam flow, lb per hr} \times (h_1 - h_f)}{\text{kilowatt output}}$$

where h_1 is the steam enthalpy entering turbine and h_f is the water enthalpy leaving the condenser (the condenser removing only vaporization energy). In other words, h_f is the enthalpy of saturated water at the pressure of the exhaust steam from the turbine.* The heat rate will be expressed in terms of Btu per kilowatt-hour.

For any one turbine if the inlet steam pressure and temperature and the exhaust pressure can be closely controlled to stay practically constant, the unit steam rate can be used as an index of any given turbine's performance. This is

$$\text{Unit steam rate} = \frac{\text{steam flow, lb per hr}}{\text{kilowatt output}}$$

in terms of pounds per kilowatt-hour.

Note that

$$\text{Heat rate} = \text{unit steam rate} \times (h_1 - h_f)$$

Another variation of heat rate is to express it as thermal efficiency. Since 1 kwhr equals 3,412 Btu,

$$\text{Thermal efficiency} = \frac{3{,}412}{\text{heat rate}} \times 100$$

where thermal efficiency is expressed in per cent. This really expresses the proportion of total energy input produced as useful work output.

Turbine Cycles. Turbines may be arranged in a number of ways in a heat cycle to produce shaft work. Calculation of heat-rate performance must charge the turbine with only the heat energy that it uses. Figure 6-3 shows four fundamental methods; there are others, of course, but these will serve to demonstrate how to handle all cases. In every case the steam flow to the throttle must be known as indicated by the heat-rate formula, so all that need be identified is the point and method of determining the steam enthalpies. In Fig. 6-3 all the values of h_1 are taken at the throttle. In (*a*) we have already discussed the cycle; and, as indicated in the figure, the condensate or feed-water enthalpy should be taken to correspond to the turbine exhaust pressure. In a regenerative feedwater heating turbine cycle (*b*), steam is tapped off at intermediate stages

* For the general case h_f will be the enthalpy of the water as returned to the boiler. Note that as heaters are introduced in the cycle, h_f is not taken at the condenser hot well.

to mix with the feed water in the contact heaters. Compared to a given throttle flow in a straight-condensing turbine, less output will be developed because all the steam does not flow entirely through the turbine, but the much higher feed-water temperature requires less heat input by the

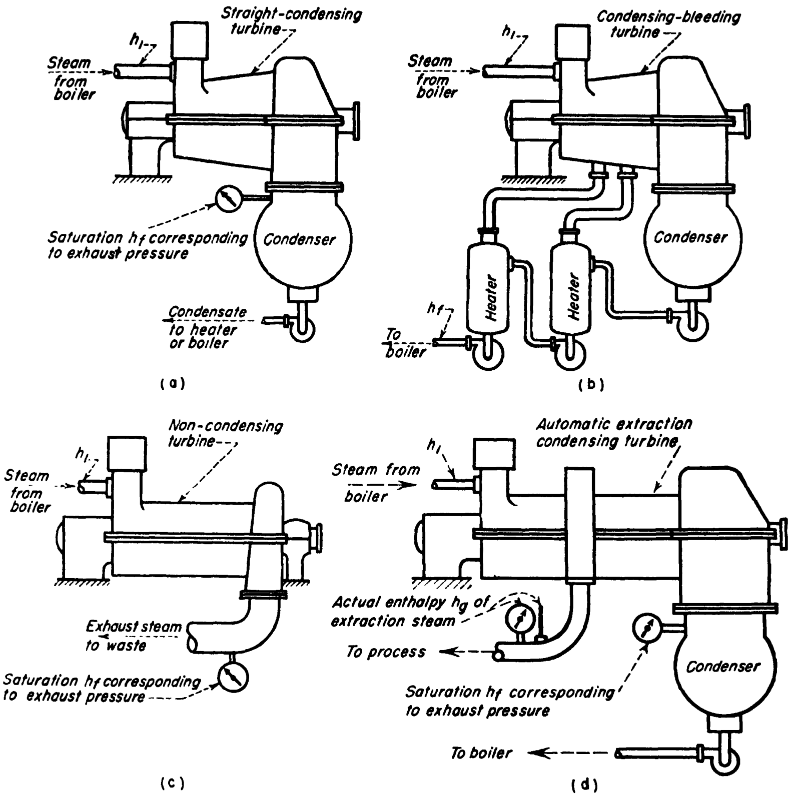


FIG. 6-3. Enthalpy-measuring points vary according to the type of turbine cycle.

boiler. It will be found that the heat rate is lower for a regenerative cycle because the hotter water overbalances the effect of less output. There are many varieties of regenerative cycles, using from one to eight heaters; some use closed heaters, others a combination of open and closed heaters. The arrangement selected in each case depends on a balancing of investment cost against the saving in fuel cost.

In Fig. 6-3c a noncondensing turbine wastes its exhaust to atmosphere. For this case it is customary to charge the turbine with all the heat above the liquid saturation enthalpy corresponding to the exhaust pressure. For any appreciable back pressure this heat will be considerably greater than the heat usually carried by the make-up entering the boiler. In figuring an over-all heat cycle, account must be taken of this difference. If the exhaust of a noncondensing turbine is employed for some useful purpose such as space heating or process work, then the actual exhaust enthalpy h_g is taken instead of h_f in the heat-rate formula. The heat rate will then be very close to 4,500 to 5,000 Btu per kwhr.

Figure 6-3*d* shows one arrangement of an automatic-extraction-condensing turbine. Because of the two different methods of using the steam leaving the turbine, a heat rate normally is not expressed for turbines of this type, though one can readily be calculated. Usually the performance is expressed as a family of curves of total hourly steam flow vs. kilowatt output. These will be discussed later in the chapter. The heat rate for this turbine would be calculated as

$$\text{Heat rate} = \frac{\text{exhaust flow} \times (h_1 - h_f) + \text{process flow} \times (h_1 - h_n)}{\text{kilowatt output}}$$

where the flows are in pounds per hour.

Pressure and Temperature. Knowing how to evalute the performance of a turbine in terms of heat rate, thermal efficiency, and steam rate, let's look at the why and wherefore of steam pressures and temperatures. The principal reason for using high steam pressures and temperatures is the improved efficiency of operation that can be realized in terms of lower fuel consumption. However, as pressures are increased, it becomes necessary to use heavier construction; and higher temperatures demand more expensive alloys in the materials of construction. These mean additional investment cost in equipment, and it takes competent engineering study to decide whether the additional first cost is warranted by the saving in fuel cost over the life of the turbine and the boiler.

The effects of pressures and temperatures on work per pound of steam flow, heat rate, thermal efficiency, and steam rate are shown in Figs. 6-4 to 6-7. These are calculated for an engine efficiency of 80 per cent with mechanical turbine and generator efficiencies of 100 per cent; this is the same as 80 per cent internal efficiency.

Figure 6-4 shows the number of Btu of work done by 1 lb of steam expanding from the given pressure and temperature through a turbine to 1 in. Hg back pressure. The curves show that for dry saturated steam the work developed is a maximum at about 1,430 psia and corresponding temperature of 590 F. For superheated steam at any one pressure the work per pound steadily increases as the temperature increases. When considering a constant temperature and varying the pressure, the relationship is not quite so simple. For instance, lets's examine the 700 F ordinate. The work increases as pressure rises from 100 psia to a maximum at about 1,100 psia. For further pressure increase the work per pound falls till at about 3,000 psia it is less than at 100 psia. On the other hand,

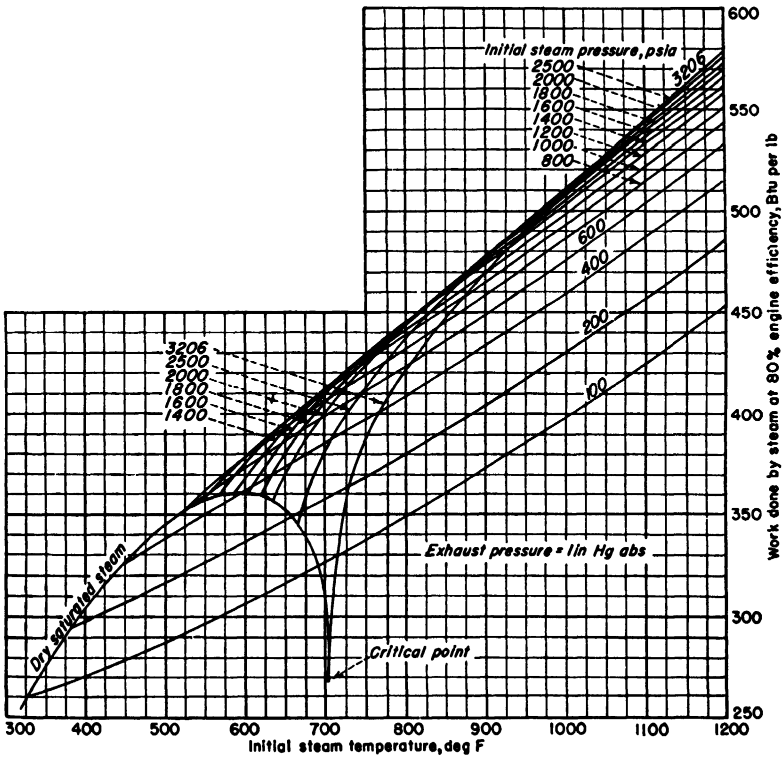


FIG. 6-4. Work done by 1 lb of steam at 80 per cent engine efficiency varies with initial steam pressures and temperatures for a constant 1 in. Hg abs back pressure.

at a constant temperature of 1200 F the work steadily increases from 100 to 3,206 psia.

However, do not confuse work per pound with thermal efficiency or heat rate. While the work varies with pressure and temperature, the amount of heat or enthalpy put into the steam also varies (see Fig. 5-5) but at a different rate. As a result the heat-rate variation with pressure and temperature behaves as shown in Fig. 6-5. Notice that at constant temperature there is much more marked improvement with increasing pressures, especially at the higher values, for the heat rate than for the work per pound. An interesting point is the decreasing rate of improvement per pound of pressure increase; for instance, at 1000 F the heat rate improves about 790 Btu per kwhr when raising the pressure 200 psi from 200 to 400 psia. On the other hand, when raising the pressure 706 psi

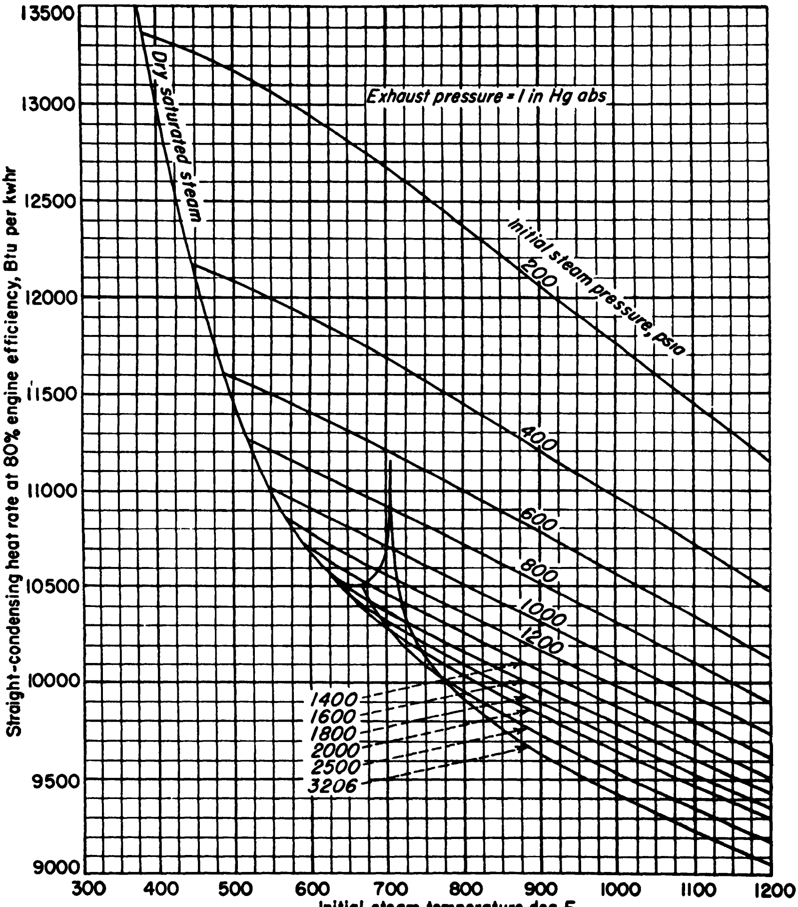


Fig. 6-5. Turbine heat rate at 80 per cent engine efficiency varies with initial steam pressure and temperature for a constant 1 in. Hg abs back pressure.

from 2,500 to 3,206 psia, the heat rate improves only about 100 Btu per kwhr. In general it can be said that increasing steam pressures and temperatures improves performance, with the exception of a small range of pressures near the critical.

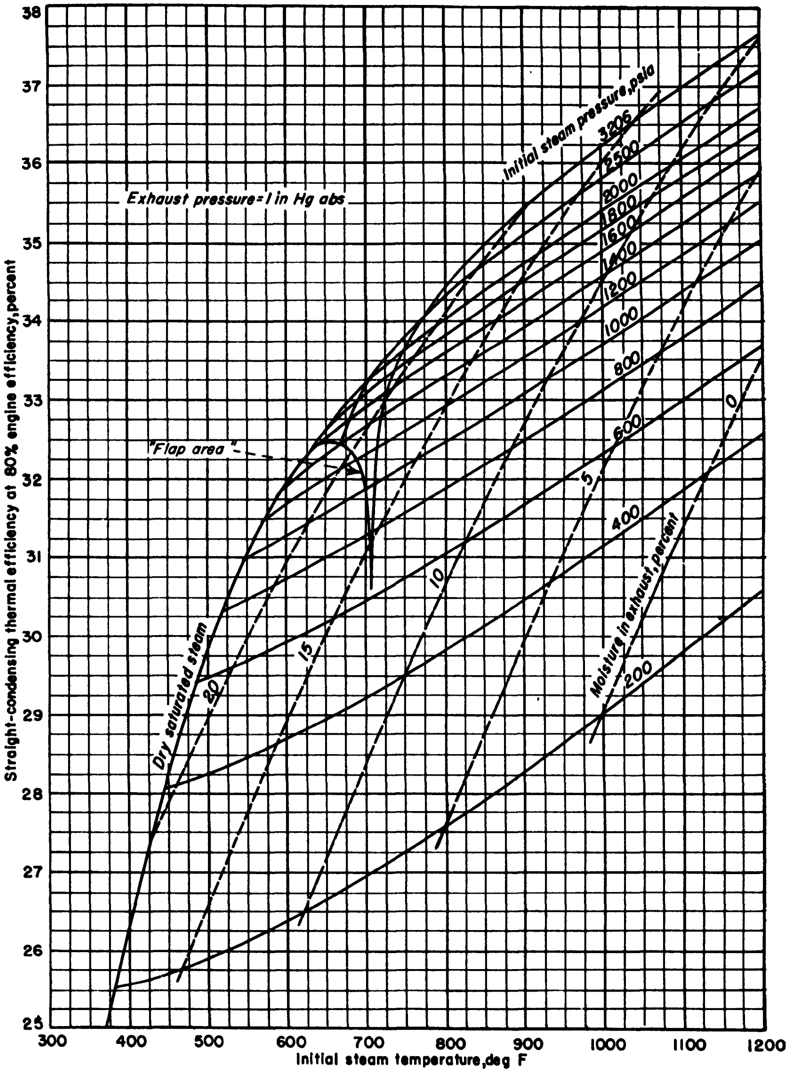


FIG. 6-6. Turbine thermal efficiency operating at 80 per cent engine efficiency varies with initial steam pressures and temperatures for constant back pressure of 1 in. Hg abs. The dashed lines show the corresponding moisture in exhaust steam.

Figure 6-6 gives essentially the same information as Fig. 6-5, except that it is expressed as a reciprocal, thermal efficiency. Additional curves of moisture drawn on this graph show the amount of water appearing in the steam exhausting from the turbine when expanding from given initial

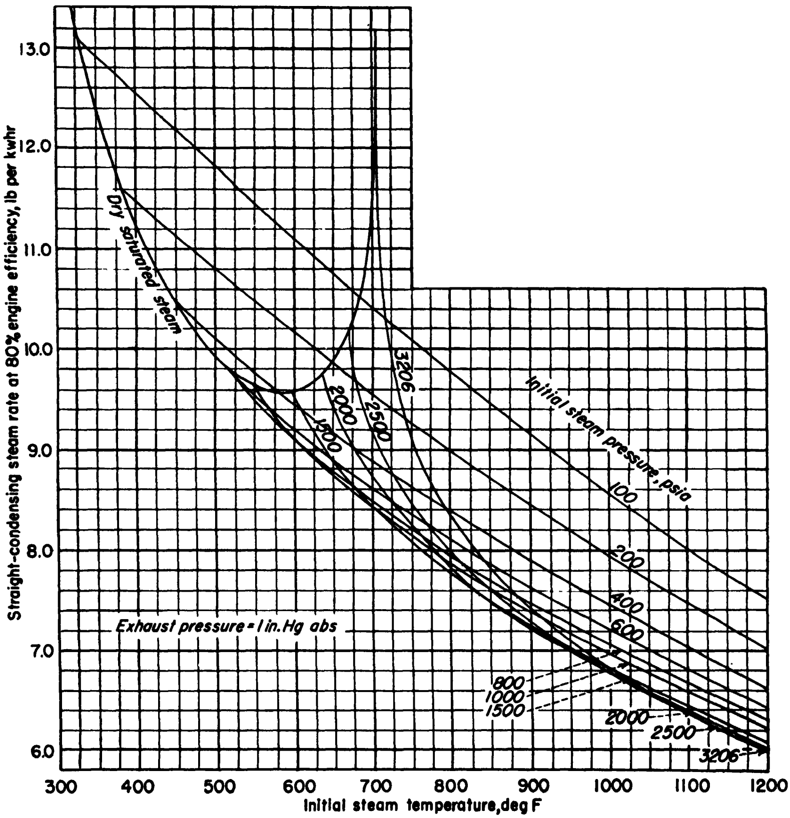


FIG. 6-7. Turbine steam rate operating at 80 per cent engine efficiency varies with initial steam pressure and temperature for constant 1 in. Hg abs back pressure.

pressures and temperatures. For example, at 750 F and 400 psia the moisture in exhaust will be 10 per cent. For all pressure and temperature combinations to the right of the 0 per cent moisture line the exhaust will be superheated steam. Because of the erosive action of moisture in steam, the water entrainment should be limited to less than 15 per cent. Higher moisture contents cause severe erosion of the last-stage turbine blading, resulting in high replacement and maintenance costs. Hence all steam conditions to the left of the 15 per cent moisture line are impracticable. The moisture curves do not apply to the "flap" area near the critical pressure and temperature; here the moisture will run about 45 per cent.

Figure 6-7 shows the variation in steam rate. One common error assumes that steam rates for turbines operating at different steam conditions are indicative of their relative thermal efficiencies or heat rates. This fallacy will be readily apparent by comparing the appearance of the curves of Fig. 6-7 with those of Fig. 6-5. When an idea of their relative efficiencies is desired, steam rates can be compared for two different turbines only when they are working under identical inlet and exhaust steam conditions and with identical cycle layouts.

Checking Turbine Performance. Once a turbine has been installed, constant vigilance is needed to keep it in efficient working condition. This requires collecting and logging sufficient data during operating periods to enable one to figure performance. It was shown in Figs. 5-28 and 5-29 how the total steam flow through a condensing steam turbine will vary according to the governing valve arrangement. All that is needed to plot such a curve for a turbine-generator set is a good flowmeter and watt-hour meter. This curve can usually be obtained from the manufacturer for specified steam pressure and temperature and a given exhaust pressure. Figure 6-8*a* shows a curve typical for a unit with three main control valves, the secondary and tertiary valves opening, respectively, at 60 and 80 per cent of the total maximum output. Note that this curve holds for given throttle and exhaust conditions. Any deviation in this respect will alter the performance and hence the curve. This aspect is discussed in more detail later.

A corresponding curve for total heat-input rate in Btu per hour can be readily calculated from the total steam-flow curve as shown in Fig. 6-8*b*. Heat-input rate equals steam flow times $(h_1 - h_f)$ where the steam flow is in pounds per hour. The heat-rate curve in (*c*) can then be computed from the curve in (*b*) by simply dividing the total heat input at any load by the load. The heat rates are very high at low loads; at zero load or output the heat rate is infinite. As load increases, the heat rate drops rapidly, and for this *particular* turbine becomes a minimum at 60 per cent of maximum output. For higher outputs the heat rate increases somewhat. This particular turbine has been designed for a load curve that has load values occurring most frequently at 60 per cent of the peak load.

Figure 6-8*d* shows the variation in thermal efficiency and corresponds to the other three curves. It is simply the reciprocal of the heat-rate curve in (*c*) multiplied by 3,412. Since the heat rate is lowest at 60 per cent load, it follows that the efficiency must be highest at the same load.

These curves can all be determined by test and, for new turbines, can be closely calculated by the designer. Complete details of turbine testing procedure are given in the ASME Steam Turbine Power Test Code.

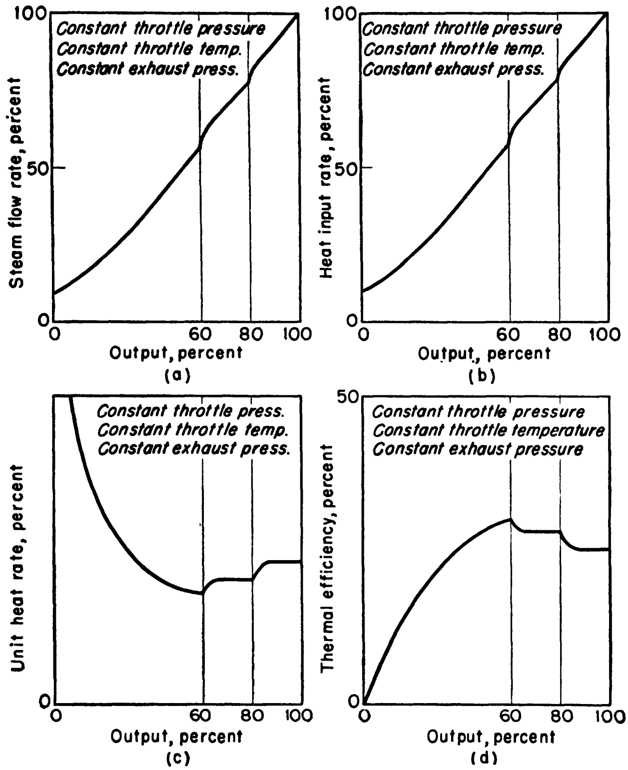


FIG. 6-8. Turbine performance factors vary with load and control-valve arrangement. This turbine has been designed for its most efficient operation at 60 per cent of rated capacity.

All types of turbines are covered in this test code, which should be followed closely to obtain correct results.

While turbines are designed for specific conditions of pressure and temperature at throttle and pressure at exhaust, these factors are very difficult to keep exactly constant during practical operation. As they vary, the performance of the turbine varies. Hence, to check on whether a turbine is performing during normal operation as it should, it becomes necessary to take proper account of variation in all these factors. Before discussing how this should be done, let's first study how performance is affected by each of these factors.

In the case of a condensing turbine, the most variable operating factor is the exhaust pressure as held by the condenser. More important, only a small variation in back pressure causes a marked change in economy performance. Figure 6-9 shows how condenser pressure varies with steam flow and inlet cooling-water temperature. As steam flow or load increases, the back pressure increases; *i.e.*, the vacuum decreases. As the cooling water becomes warmer, the back pressure increases. Not shown by this curve family is the fact that, as condenser tubes become fouled, the back pressure increases. Cooling-water flow through the condenser also affects the pressure held; as

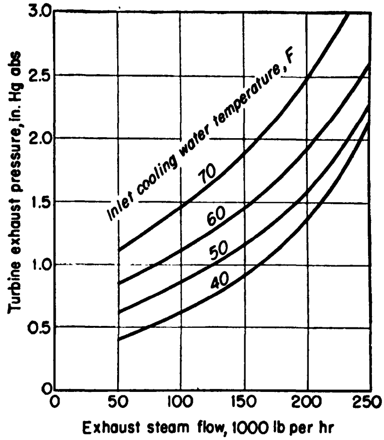


FIG. 6-9. Exhaust pressure maintained by condensers depends on the rate of steam flow entering, inlet cooling-water temperature, speed of cooling water passing through tubes, and cleanliness of heat-transfer surface.

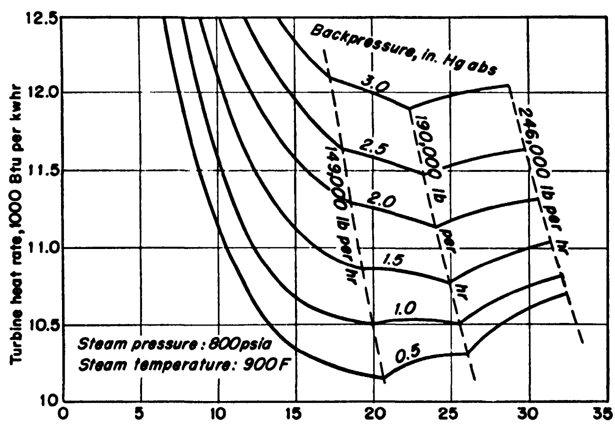


FIG. 6-10. Turbine heat rate varies with load and turbine exhaust pressure. The loads at which the valve groups start to open depend on back-pressure magnitude.

the flow decreases, the pressure increases. Many condensers are provided with two pumps each of which usually can be operated at two different speeds. An economy study must be made to find how best to operate pump speeds in respect to load, cooling-water temperature, and turbine heat rate.

With the condenser pressure affected by so many factors, the turbine heat rate is bound to vary. Extent of this variation is shown for a 30,000-kw unit in Fig. 6-10. This family assumes that steam throttle temperature and pressure remain constant at the specified values. As

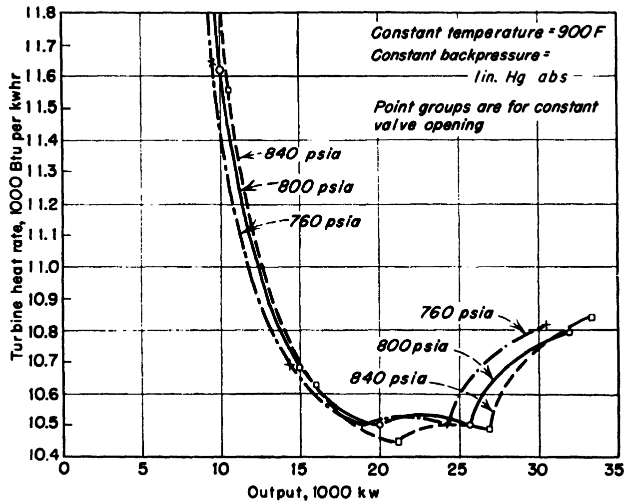


FIG. 6-11. Turbine heat rate varies with load and initial steam pressure. The loads at which the valve groups start to open depend also on initial pressure.

back pressure increases at any load, the heat rate rises rapidly. Looking back to Fig. 6-9 it will be apparent that a turbine will operate much more efficiently in the winter with the low cooling-water temperatures than during the summer. In Fig. 6-10 note that the loads at which valves start to open decrease as the back pressure rises. However, the steam-flow rate through maximum valve openings stays constant as the back pressure changes. The turbine may be regarded as an orifice. Hence, when the exhaust pressure is less than about half the inlet pressure, steam flow will stay constant, regardless of how low the exhaust pressure may drop (see Fig. 5-7*a*). The heat-rate change follows a more or less consistent pattern for a condensing turbine.

Figure 6-11 shows the effect of changing throttle pressure on the heat-rate curve of a turbine. The pattern of change does not appear so consistent as for back-pressure change. For partial primary valve opening at constant output, the heat rate improves with falling pressure. Maximum valve openings are reached at lower loads as the throttle pressure falls. On the secondary valve the effect of pressure change is mixed; on the tertiary valve, heat rate increases with falling pressure over most of the range. Fortunately, steam pressure is one of the easily controlled factors in plant operation, and any variations from design conditions are usually very small.

Figure 6-12 shows the effect of change in throttle temperature. As

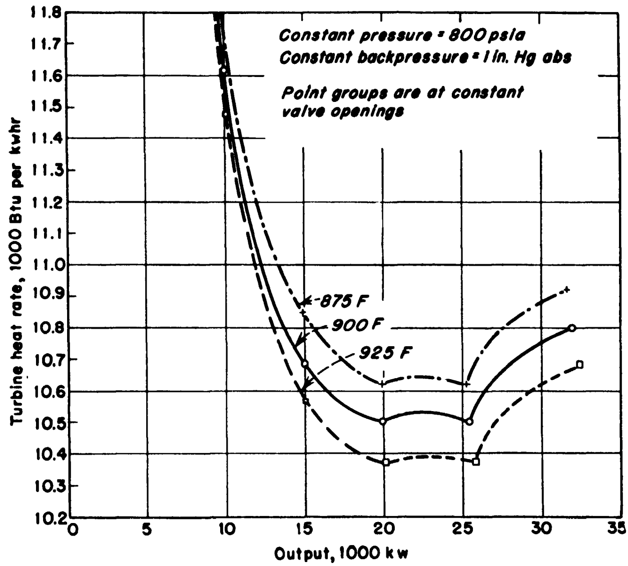


FIG. 6-12. Turbine heat rate varies with load and initial steam temperature.

temperature increases, the heat rate improves uniformly. For modern h-p high-temperature units, steam temperature can be very closely controlled over the upper half of the load range; for the lower load range, the temperature usually drops with the load. For the older units operating at temperatures below 850 F, steam temperature usually rises with load, except in cases where radiant and convection superheaters are used in the boiler, or where special controls keep the temperature nearly constant over the entire load range.

Average Heat Rate. In checking on a turbine's performance over a period of time, the easiest index to use is the average heat rate. This is easily figured as,

$$\text{Average heat rate} = \frac{\text{total steam } (h_1 - h_f)}{\text{total kwhr}}$$

Where the total steam is the total number of pounds that have passed through the throttle during the period as found by planimetering flow-meter charts, total kilowatt-hours are obtained from the integrating watt-hour meter for the same period, h_1 is the enthalpy of the steam corresponding to the average pressure and temperature for the period; h_f is the enthalpy corresponding to the average feed-water temperature.

Time	Load, kw	Back pressure, in. Hg abs	Throttle temperature, deg F	Throttle pressure, psia
Midnight	10,000	1.0	890	800
1 A.M.	8,000	0.9	885	800
2 A.M.	8,000	0.85	888	800
3 A.M.	7,000	0.70	880	800
4 A.M.	7,000	0.75	882	800
5 A.M.	9,000	0.95	890	800
6 A.M.	13,000	1.15	900	800
7 A.M.	17,000	1.40	900	800
8 A.M.	24,000	2.33	900	800
9 A.M.	24,000	2.14	900	800
10 A.M.	24,000	2.09	900	800
11 A.M.	24,000	2.11	900	800
Noon	15,000	1.20	900	800
1 P.M.	28,000	2.31	900	800
2 P.M.	28,000	2.28	900	800
3 P.M.	28,000	2.32	900	800
4 P.M.	28,000	2.33	900	800
5 P.M.	19,000	1.71	900	800
6 P.M.	20,000	1.75	900	800
7 P.M.	20,000	1.77	900	800
8 P.M.	19,000	1.73	900	800
9 P.M.	17,000	1.42	900	800
10 P.M.	13,000	1.23	898	800
11 P.M.	10,000	1.06	894	800

Using average pressure and temperatures for computing enthalpies usually proves sufficiently accurate. Some operators weight these factors against the amount of steam corresponding to each value of pressure and temperature, but many investigations have shown that weighted and arithmetical averages are so close that the additional time needed for weighting is not warranted.

Having the average heat rate figured, the question then arises, is it correct or not? By experience over a considerable time the operator may know whether that is so or not. But the figure is of value only when compared directly with what the turbine can do as expressed either by test curves or by the designer's guarantee curves. Such a comparison must be made by taking proper account of all the operating variables. To do this, curves such as Figs. 6-10 to 6-12 must be available for the unit. The comparison simply consists of figuring the average heat rate for the period from the curves. Here's how it's done. Let's assume that the table on page 198 is the logged operating data of the unit for one day. With a day-by-day check, the heat rate for each hour could be found by taking the data logged and going directly to the corresponding curve sheet for the test heat rate of the unit. However, if this check is made for weekly or monthly periods, this method consumes too much time. A shorter procedure first groups the data by loads to determine the number of hours at that load and average back pressures, temperatures, and pressures at each load as follows:

	Load, mw										
	7.0	8.0	9.0	10.0	13.0	15.0	17.0	19.0	20.0	24.0	28.0
Hours	2	2	1	2	2	1	2	2	2	4	4
Back pressure, in. Hg	0.70	0.90	0.95	1.00	1.15	1.20	1.40	1.71	1.75	2.33	2.31
	0.75	0.85		1.06	1.23		1.42	1.73	1.77	2.14	2.28
										2.09	2.32
										2.11	2.33
Average	0.73	0.87	0.95	1.03	1.19	1.20	1.41	1.72	1.76	2.17	2.31
Temperature	800	885	890	890	900	900	900	900	900	900	900
	882	888		894	898		900	900	900	900	900
										900	900
										900	900
Average	881	887	890	892	899	900	900	900	900	900	900

The pressure has been held constant at 800 psia, which will be the usual case for most plants.

Next set up a calculating table as shown on page 200.

$$\text{Calculated average heat rate} = \frac{4{,}761.1 \times 10^6}{420{,}000} = 11{,}330 \text{ Btu per kwhr}$$

The actual average heat rate should be within 1 or 2 per cent of this calculated heat rate. If the actual is considerably higher, there is a strong indication that some maladjustment has taken place within the turbine. In such a case, first check all computations to make sure that no arithmetical errors have been made (also check instrument accuracy); if not, then investigate by first checking stage pressures and then dismantling.

The calculating table is practically self-explanatory. Columns 1 to 4 are rearrangements of the averaged group data. Column 5 is obtained from the curves in Fig. 6-10 corresponding to the load and back pressure. The correction factor, column 6, is calculated with the aid of Fig. 6-12 and the average temperature at the load. As correction factors for temperature are rather small, an approximation will suffice for most uses.

(1) Load, mw	(2) Hours	(3) Back pressure, in. Hg	(4) Temp., deg F	(5) Heat rate	(6) Correction factor	(7) Mwhr	(8) Total heat input, 10^6 Btu
7.0	2	0.73	881	12,620	1.0059	14.0	177.7
8.0	2	0.87	887	12,250	1.0040	16.0	196.9
9.0	1	0.95	890	11,940	1.0031	9.0	107.8
10.0	2	1.03	892	11,640	1.0025	20.0	233.5
13.0	2	1.19	899	11,080	1.0003	26.0	288.0
15.0	1	1.20	900	10,860	......	15.0	162.8
17.0	2	1.41	900	10,730	......	34.0	365.0
19.0	2	1.72	900	11,050	......	38.0	420.0
20.0	2	1.76	900	11,060	......	40.0	442.4
24.0	4	2.17	900	11,250	......	96.0	1080.0
28.0	4	2.31	900	11,480	.....	112.0	1287.0
Totals	..	....	...	.....	....	420.0	4761.1

Here it will be found that at 10,000 kw, the lowest load on Fig. 6-12, the heat rate changes from 11,620 Btu per kwhr at 900 F to 11,710 Btu at 875 F. This a change of 90 Btu for 25 F. The correction is then $\frac{90}{11,620} = 0.00774$, or 0.774 per cent per 25 F. Applying this correction to the 7-mw or 7,000-kw load calculation, we find the average temperature was 881 F, a deviation of 19 F from the standard 900 F. Then the heat-rate correction factor is

$$1 + (1\%_{25} \times 0.00774) = 1.0059$$

Column 7 is the product of columns 1 and 2. Column 8 is the product of columns, 5, 6, and 7. The sum of column 7 is the total output of the day and should approximately check the integrated watt-hour meter reading for the day. The sum of column 8 is the total heat input for the day corresponding to calculated kwhr.

Regenerative-cycle Turbine. In a regenerative-cycle or bleeder turbine, such as in Fig. 6-3*b*, the steam is taken from intermediate stages of the turbine to mix with the feed water it heats. The heaters act as condensers of the bleed steam, and the amount of steam extracted is automatically balanced against the amount of feed water entering or flowing through the heater.

For instance, if steam and feed-water flows are in balance at one condition and if feed-water flow should increase through the heater, the steam would be condensed at an increased rate which would tend to lower the pressure in the heater. As a result of the tendency to reduce the heater pressure, bleed steam flow increases until the heater pressure is only slightly less than the stage pressure in the turbine. Since turbine stage pressures increase with turbine load, the heater pressures and amount of bleed steam flow will vary in the same manner. Hence, the feed-water temperature to the boiler increases with a rise in load. If closed heaters should scale up for any reason, the heat-transfer rate will drop and the feed-water temperature leaving the heater will be lower.

It should be evident that, when checking the performance of a regenerative- cycle turbine, the performance and condition of the heaters must also be accounted for. If a heater is taken out of service for overhaul while the turbine carries load, the heat rate of the unit will increase. The amount of change depends on which heater was inoperative. If the highest pressure heater goes out of operation, the heat-rate increase will be at its maximum from heater outage cause. When any of the other heaters are taken out, the next higher pressure heater partly makes up for the outage. This heater will increase its bleed steam flow because of the cooler feed water entering it and practically keep the outlet feed-water temperature constant; actually there may be a drop of a few degrees. However, despite the heating job being maintained, the over-all unit efficiency will be less, because the steam all being bled at a higher pressure to produce a given feed-water temperature rise does not do so much work in the turbine. For instance, if two heaters each are required to raise the feed-water temperature 60 deg each for a total of 120 deg and the lower pressure heater of the two goes out of service, the remaining one can practically maintain the required 120-deg rise. But, the additional higher pressure bleed that now does the first 60 deg of heating replaces the lower pressure bleed. Being extracted at a higher pressure, it can't do so much work in the turbine as the l-p bleed; hence the turbine output reduces for the same heat input. Naturally the turbine heat rate increases.

For those wishing to have a more detailed explanation of regenerative-cycle performance, Fig. 6-13 has been prepared. This turbine takes throttle steam at 800 psia and 900 F and expands it to a pressure of 1 in. Hg abs in the condenser. Steam is bled at three points to heat the feed water, returning to the boiler, in three closed heaters. The bleed steam is taken from intermediate stages in the turbine. Steam pressure in each heater will depend on the pressure in the turbine stage for the particular load. Thus at low turbine outputs or steam flows, the heater pressures will be lower than at higher turbine outputs and steam flows. In the

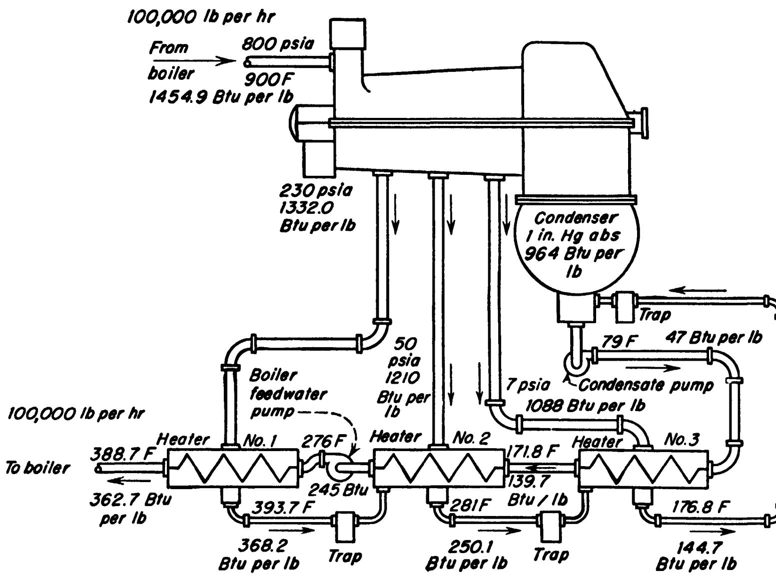


FIG. 6-13. Steam conditions in a three-heater regenerative cycle, assuming no pressure drops between bleed points and heaters, and zero radiation losses.

cycle shown in Fig. 6-13, heater 1 receives the highest pressure bleed steam. The bleed in giving up its heat to the feed water passing through the tubes in the heater (indicated by the zigzag line) condenses and collects in the hot well at the heater bottom. The trap automatically discharges this bleed condensate into the next lower pressure heater 2. The bleed condensate being at a higher temperature than the saturation temperature in heater 2 partly flashes into steam and gives up some of its energy to heating the feed water passing through heater 2. Most of the heating in heater 2 is done of course by the bleed steam from the extraction point 2 in the turbine. The condensate from heaters 1 and 2 then flows through the trap to the lowest pressure heater 3, where it again partly flashes to steam helping to heat the feed water in the heater. The major part of the heating job of course is done by No. 3 bleed steam from the turbine. The bleed condensate from all three heaters then traps to the condenser, where it is cooled to the same temperature as the main condensate. Total condensate leaving the condenser hot well then equals the steam entering the turbine throttle. The condensate pump forces this flow through heaters 3 and 2. From heater 2 the boiler feed-water pump forces the feed water through heater 1 and into the boiler. Let's make a few simple calculations to see how the turbine performs with a constant flow of 100,000 lb per hr when operating (1) on a straight-condensing cycle and (2) on a regenerative cycle with the three heaters. We'll make a few simplifying assumptions which will be pointed out as we go along, so that small considerations won't confuse the discussion. First thing we need to know will be the steam condition at throttle, exhaust, and each of the three bleed points. This can be obtained from the turbine condition line for the load, such as shown in Fig. 6-1. These are labeled directly on the diagram in Fig. 6-13. Pressures, temperatures, and enthalpies progressively decrease as the steam expands through the turbine. For our purpose we assume that the pressure of the steam in the heater is the same as at the bleed point. Actually there would be a few pounds drop. Also assumed is zero radiation loss. Notice that the feed-water temperature leaving each heater is less than the temperature of the bleed condensate. Theoretically, the feed-water temperature could be raised to the bleed temperature, but this would take a very large heater. It pays to use smaller heaters. These are assumed to have a 5-deg terminal difference; *i.e.*, the leaving feed-water temperature is 5 deg less than the bleed condensate saturation temperature.

We also assume that the turbine condition line remains constant whether operating straight condensing or bleeding. Actually there will be a small change in the lower stages because of the reduced flow through them, when bleeding steam for feed-water heating. First, let's calculate the turbine heat rate and thermal efficiency when operating straight condensing, all bleed points shut tight, and the feed water returning to the boiler at the temperature it attains in the main condenser hot well, 79 F in this case. The numbers used in the following can all be found on Fig. 6-13.

1. Work done on turbine blades by steam = 1,454.9 − 964.0 = 490.9 Btu per lb flow
2. Turbine energy input = 1,454.9 − 47.0 = 1,407.9 Btu per lb flow
3. Total work developed on turbine blades $= \dfrac{100{,}000 \times 490.9}{3{,}412}$ = 14,390 kw

4. Total turbine input $= 100{,}000 \times 1{,}407.9 = 140{,}790{,}000$ Btu per hr
5. Turbine heat rate $= \dfrac{140{,}790{,}000}{14{,}390} = 9{,}780$ Btu per kwhr
6. Thermal efficiency $= \dfrac{3{,}412}{9{,}780} \times 100 = 34.9$ per cent

Now let's calculate the performance when bleeding. To do this we start with the energy balance about heater 1 in order to find the bleed steam flows. With no losses by radiation or friction, the energy picked up by the feed water must equal the energy given up by the bleed steam. Bleed pressure, temperature, and enthalpy are fixed by the turbine condition line, the feed-water temperature leaving will be 5 deg less than the bleed condensate saturation temperature, and the feed-water temperature entering is fixed by the steam conditions in heater 2. If we designate bleed steam flow 1 by B_1 we can write for heater 1:

1. Heat picked up by feed water = heat given up by bleed steam
2. $100{,}000(362.7 - 245.0) = B_1\,(1{,}332.0 - 368.2)$

Since B_1 is the only unknown quantity, we can calculate it by the equation above.

3. $B_1 = \dfrac{100{,}000 \times 117.7}{963.8} = 12{,}220$ lb per hr

Thus in the turbine, steam from throttle to bleed point 1 flows at the rate of 100,000 lb per hr. But with 12,220 lb per hr leaving the turbine through bleed point 1 only $100{,}000 - 12{,}220 = 87{,}780$ lb per hr flows from bleed point 1 to 2.

Next let's find bleed 2 by a similar precedure:

4. Heat picked up by feed water = heat given up by bleed steam and condensate 1

The bleed condensate trapped from heater 1 gives up heat by cooling down to the saturation temperature of heater 2 which reduces the amount of steam that flows through bleed point 2.

5. $100{,}000(245.0 - 139.7) = B_2(1{,}210.0 - 250.1) + 12{,}220(368.2 - 250.1)$
6. $B_2 = \dfrac{(100{,}000 \times 105.3) - (12{,}220 \times 118.1)}{959.9} = 9{,}470$ lb per hr
7. Steam flow in turbine from bleed point 2 to 3 $= 87{,}780 - 9{,}470 = 78{,}310$ lb per hr
8. Total bleed condensate trapped to heater 3 $= 12{,}220 + 9{,}470 = 21{,}690$ lb per hr

Now let's solve for the third point bleed flow:

9. Heat to feed water = heat from bleed steam and bleed condensate 1 and 2
10. $100{,}000(139.7 - 47.0) = B_3(1088.0 - 144.7) + 21{,}690(250.1 - 144.7)$
11. $B_3 = \dfrac{(100{,}000 \times 92.7) - (21{,}690 \times 105.4)}{943.3} = 7{,}400$ lb per hr
12. Turbine steam flow from bleed point 3 to condenser = 78,310 − 7,400 = 70,910 lb per hr

Now we are ready to figure the amount of work done on the turbine blades by the steam in each section of the turbine:

			Btu per hr
13.	Throttle to bleed 1	= 100,000(1,454.9 − 1,332.0)	= 12,290,000
14.	Bleed 1 to 2	= 87,780(1,332.0 − 1,210.0)	= 10,700,000
15.	Bleed 2 to 3	= 78,310(1,210.0 − 1,088.0)	= 9,550,000
16.	Bleed 3 to condensate	= 70,910(1088.0 − 964.0)	= 8,790,000
17.	Total.		= 41,330,000

18. Total work done on blades $= \dfrac{41{,}330{,}000}{3{,}412} = 12{,}110$ kw

Notice now that the turbine returns feed water to the boiler at a high temperature, 388.7 F. Hence the boiler has only to add heat above this temperature to reevaporate the steam, an obvious saving in fuel and increase in efficiency. Then,

19. Total turbine input = 100,000(1,454.9 − 362.7) = 10,922,000 Btu per hr
20. Turbine heat rate $= \dfrac{10{,}922{,}000}{12{,}110} = 9{,}020$ Btu per kwhr
21. Thermal efficiency $= \dfrac{3{,}412}{9{,}020} \times 100 = 37.8$ per cent

When regenerating, the turbine output reduces from 14,410 to 12,110 kw for constant throttle flow, but the efficiency increases from 34.9 to 37.8 per cent. The reduction in output results from the diversion of steam for feed-water heating, and the efficiency increase is realized by the return of hotter feed water to the boiler. In other words, more energy circulates within the power-plant cycle. This contributes to increased efficiency. Note that we assumed 100 per cent turbine mechanical efficiency.

Not shown in Fig. 6-13 are valves, by-passes, heater vent lines, generator air-cooling devices, air-ejector condensers, and the like that will be found in many practical plant cycles. These will be discussed in Chap. 8.

The efficiency of the cycle in Fig. 6-13 could be increased slightly by pumping all the bleed condensate leaving heater 3 into the feed-water line after heater 3. However, this requires investment in a pump which will cost more initally than a trap and requires energy input to operate. Only an engineering study for a given installation can decide which scheme is more economical over the life of the installation.

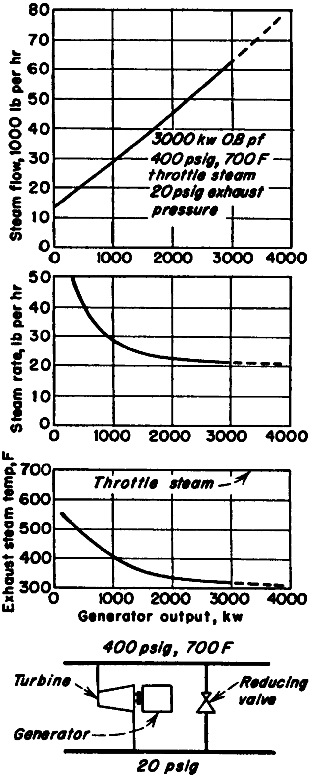


FIG. 6-14. Performance curves of a simple noncondensing turbine. The maximum steam temperature that can be tolerated by a turbine at its exhaust nozzle determines the minimum constant load that can be carried.

Combined Steam and Electric Supply. Many factories require simultaneous supplies of l-p steam and electric energy. A simple back-pressure turbine fits into such a situation neatly, producing electricity under the most economical conditions possible. At the bottom of Fig. 6-14 a back-pressure turbo-generator takes steam from a boiler header at 400 psig, 700 F, exhausting it to a l-p header at 20 psig. If the turbine exhaust is the only l-p steam supply, then a simple back-pressure governor controls steam flow through the turbine so that the lower pressure stays constant with variable steam demand. For such an arrangement the a-c generator must run parallel with other speed-governed units to hold the frequency constant. Electric output will obviously vary directly with steam flow. Whenever steam demand by the l-p header exceeds turbine capacity or the steam flow required for the concurrent electric demand, the reducing valve opens and passes the required flow. When the electric demand exceeds that which can be developed by the turbine for the concurrent steam flow passing through it, the difference must be made up by some other generator or source of supply.

The upper graph in Fig. 6-14 shows the variation in steam flow with kilowatt output for a back-pressure unit rated 3,000 kw at 0.8 pf taking steam at 400 psig, 700 F, and exhausting it at 20 psig. The dotted portion indicates the relation up to maximum possible output with 1.0 pf on the generator for a maximum capability of 3,750 kw. The corresponding steam-rate curve is shown in the middle graph. As the load decreases, the enthalpy of the exhaust steam increases because of the increased throttling by the control valves (see Fig. 5-27*e*). In this case it means that the exhaust temperature increases as the load decreases, as shown on the lowest graph, Fig. 6-14. If the higher temperature at low electric outputs is undesirable, the exhaust steam must be desuperheated by the introduction of a water spray or by passing it through a cooler. For some conditions the exhaust steam may be wet at all loads; then the enthalpy will vary with the load, but the temperature remains constant.

The electric output is produced under the most economical conditions possible because all the energy in the exhaust becomes available for use in the process consuming the l-p steam. Hence the turbine is charged only with the energy it removes from the steam flowing through it. The turbine heat rate can then be figured as

$$\text{Heat rate} = \frac{3{,}412}{\text{turbine mechanical efficiency} \times \text{generator efficiency}}$$

The turbine mechanical efficiency is the ratio of the energy produced at the shaft to the energy developed on the blades by the steam, the figure being generally well over 95 per cent for the larger turbines. Generator efficiency is the ratio of kilowatt output at the terminals to the shaft energy input. It varies about a value of 95 per cent. For instance, if the turbine mechanical efficiency is 97 per cent and the generator efficiency 95 per cent, then the turbine heat rate for a back-pressure unit is $= \dfrac{3{,}412}{0.97 \times 0.95} = 3{,}700$ Btu per kwhr. From an over-all plant standpoint, if the steam is generated in a boiler operating at 80 per cent efficiency, these kilowatt are produced at a heat rate of $\dfrac{3{,}700}{0.8} = 4{,}630$ Btu per kwhr. Compare this with heat rates for condensing turbines in Fig. 6-5, and the superior performance of a back-pressure turbine becomes obvious. However, as mentioned before, the back-pressure turbine can be used only where there are simultaneous demands for electricity and steam.

Where demands for electricity exceed what can be supplied by a back-pressure turbine, a condensing turbine can be used to supply the difference. Such a hookup can be arranged as shown in Fig. 6-15. Here the condensing turbine takes its steam from the exhaust of the back-pressure unit. Generators of both turbines supply a common electric demand. The main requirement of this arrangement is that the l-p process steam pressure be held constant. This is accomplished by proper manipulation of the control valves of both turbines.

For instance, suppose that no process steam is required momentarily and the electric demand requires a steam flow, A, through both turbines as shown in the graph of Fig. 6-15. As can be seen, the back-pressure turbine control valve throttles the steam to hold the indicated stage pressures (nozzle-inlet steam pressures) and more importantly the exhaust or

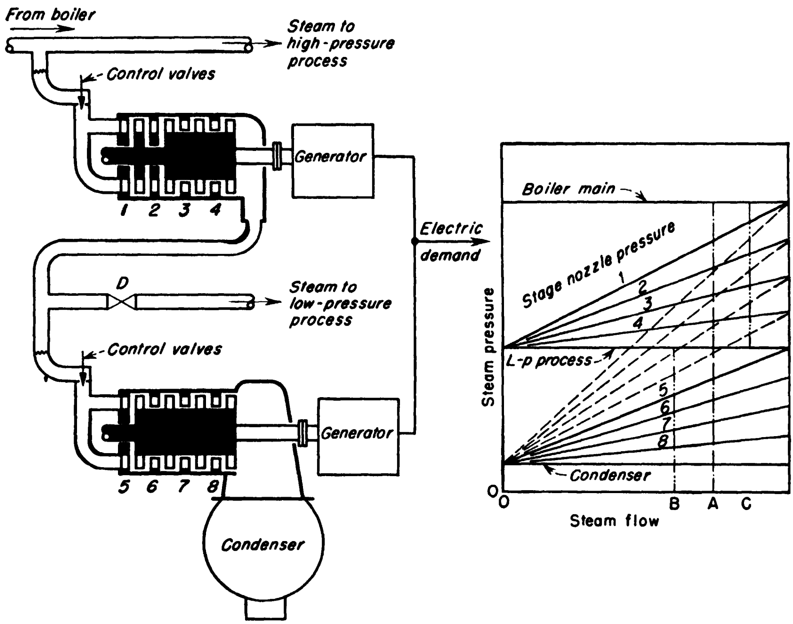


FIG. 6-15. When l-p process steam is needed, it can be provided by tapping off from noncondensing-turbine exhaust which also supplies the condensing turbine. Stage-pressure variations with steam flow for each turbine are shown in the graph. By closing valve D and opening the condensing-turbine valves wide, the turbines operate as a cross-compound unit; stage pressures then vary as indicated by the dotted lines.

l-p process pressure. Simultaneously the control valve of the condensing unit throttles the l-p steam entering to hold the stage pressures indicated.

Now suppose that a demand for l-p steam arises equal to a value of the flows $C - B$ shown on the graph, and the electric demand remains constant. This results in increasing the flow through the back-pressure turbine from A to C and reducing the flow through the condensing turbine from A to B. The former now produces a larger portion of the total electric demand, and the latter correspondingly less. To do this the back-pressure turbine control valve has opened wider, raising its stage pressures (but holding the exhaust pressure constant) and the condensing-turbine control valves have correspondingly closed, lowering the stage pressures.

One way of looking at the control-valve action is to regard the back-pressure turbine valves as controlling steam flow for both electric generation and l-p steam production, the condensing-turbine control valve serves to hold the l-p pressure. In other words, the latter turbine's control valves "back up" the steam flow, thereby maintaining l-p steam pressure. However, the control valve must simultaneously be responsive to rotor speed change as affected by variation in electric demand. If the first turbine's control valve is responsive only to l-p pressure (its exhaust pressure), it will simply pass steam through the turbine to conform with the

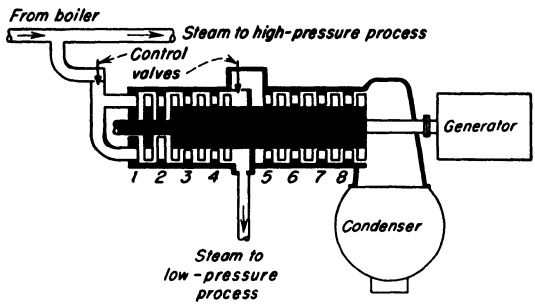


FIG. 6-16. Instead of using two separate units as in Fig. 6-15, l-p steam can be supplied by single automatic-extraction condensing unit. Stage pressures behave as shown in Fig. 6-15.

l-p demand and automatically generate the corresponding amount of electric energy. This will affect the speed of the condensing turbine; for example, when an increasing demand for l-p steam increases the generation of the first turbine and the electric demand stays constant, the turbine speed increases, automatically increasing the speed of the condensing turbine. The speed governor of the latter then closes down its control valves, reducing its generator output until the two generator outputs match the demand, and the speed of both machines returns to normal.

An interesting side light on the hookup of Fig. 6-15 is that it can be operated as a cross-compound unit if l-p process steam is not needed. By closing valve D and opening the condensing-turbine control valves wide, the stage pressures of the first turbine will behave as shown by the dotted lines in the graph. The control valves of the first turbine then do all the throttling to control steam flow through the unit, and the crossover (connection between turbines) pressure varies with load; *i.e.*, it equals the fifth-stage pressure, which is the first stage of the condensing turbine.

Automatic-extraction Turbines. Instead of having two separate turbine-generator units as in Fig. 6-15, a more economical unit can be built as in Fig. 6-16 with one turbine and one generator to perform exactly the same functions. Such a unit has been mentioned previously—automatic-extraction turbine. Control valve actions and stage pressures behave exactly the same as for separate rotating elements—with the added advantage that only one rotor has to be supervised. The diagrammatic unit shown is a single automatic-extraction turbine and may be regarded as a form of bleeding turbine with the special provision that the bleed pressure remains constant at all loads. There are also turbines with two constant-pressure extraction points called "double automatic-extraction turbines." Control valves must be provided at each of the extraction points, as in the single automatic-extraction turbine. As mentioned in Chap. 1, this class of turbines can have either back-pressure or condensing exhaust.

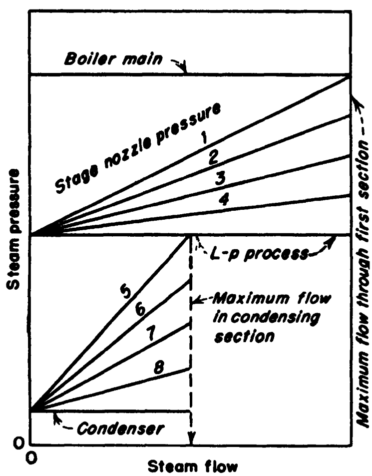


FIG. 6-17. In some designs of automatic-extraction turbines, maximum flow through the exhaust or condensing section is less than through the first section. Stage nozzle pressures then vary with steam flow as shown above.

The stage-pressure graph in Fig. 6-15 showed the relations for a design accommodating the same maximum steam flow throughout the entire unit. Actually any desired proportion of flows can be provided that a particular application needs. Often the exhaust section need accommodate only a much smaller flow of steam than the first section to produce the maximum amount of electric generation needed. For such conditions the stage-pressure relations may appear as in Fig. 6-17, where the condensing section can pass only one-half the maximum amount passing through the first section.

To check turbine operation and condition, the stage-pressure graphs afford one method, but a more common method uses a plot of steam flows vs. kilowatt output, as shown in Fig. 6-18. This is a basic input-output curve as used for other simpler forms of turbines, but because of the added controllable extraction flow the performance is expressed as a family of curves. Any one kilowatt output can be produced by a large range of throttle and extraction flows. For instance, the 3,000-kw output can be produced by passing 33,000 lb per hr steam flow through the entire turbine from throttle to condenser; by passing 65,000 lb per hr through the throttle, extracting 40,000 lb per hr, and passing the remaining 25,000 lb per hr to the condenser; or, by passing maximum throttle flow of 110,000 lb per hr, extracting 96,000 lb per hr, and passing remaining 14,000 lb per

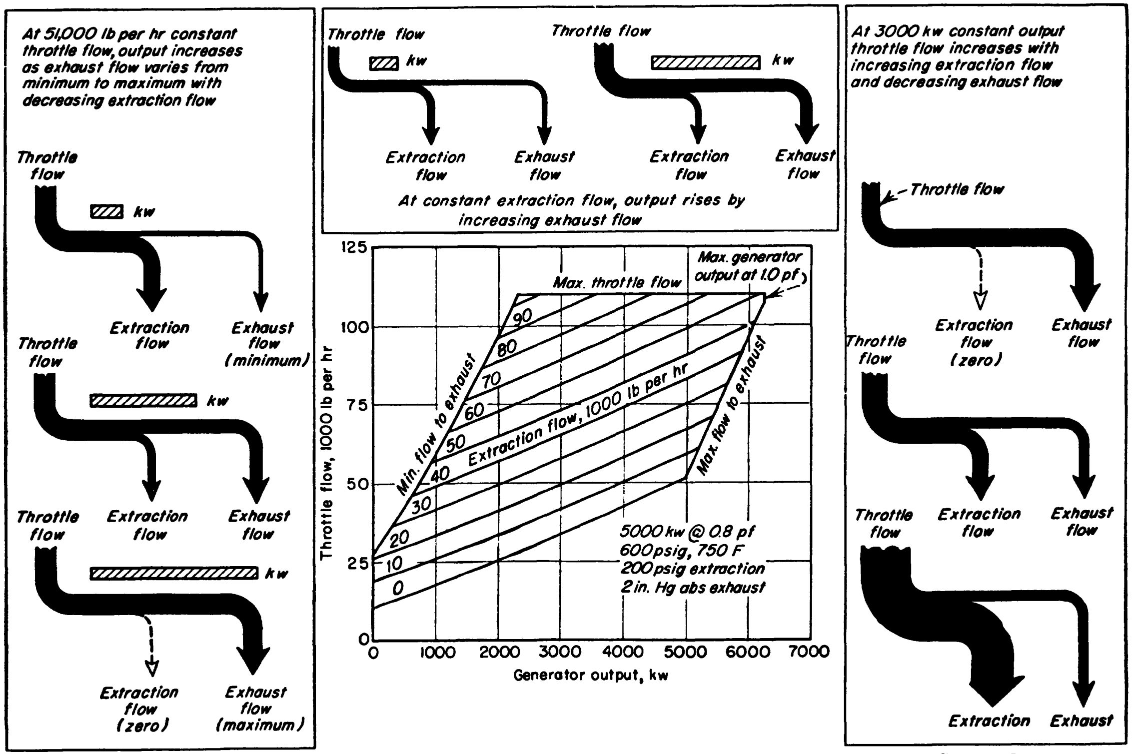


FIG. 6-18. Throttle-flow performance curve for 5,000-kw single automatic-extraction condensing turbine. Stream-flow diagrams show variation in throttle, extraction, and exhaust flows for different conditions of output.

hr to the condenser. These flows are indicated by the stream charts at the right in the figure.

Any one throttle flow can produce a wide range of kilowatt output, depending on how the flow is divided between extraction and exhaust. The flow charts at the left indicate the steam flows for 51,000-lb-per-hr constant throttle flow with diminishing extraction flow and correspondingly increasing exhaust flow. The cross-hatched bar shows the variation in kilowatt output.

The flow charts at the top of the figure show how turbine kilowatt output is varied with constant extraction flow.

On the graph are shown certain limits that come into play. At the top, one limit appears in the maximum amount of steam that the control valves can pass, because of the area available for steam passage. A little thought will show that the left end of the constant-extraction flow curves must end at least where the throttle flow equals the extraction flow. Actually for this turbine it will be seen that they end at a throttle flow 6,000 lb per hr greater than the extraction flow. The limit is marked "minimum flow to exhaust." The 6,000 lb per hr is the steam flow needed to *cool* the blading of the turbine's exhaust section. If this section of the rotor ran without the steam flowing through, the fanning action of the blades on the idle steam would generate enough heat by friction to melt the blades. The steam flow prevents this heating. The flow is so small that it does not contribute much to supplying the no-load losses of this part of the turbine.

At the right there is a limit marked "maximum flow to exhaust." In this region the exhaust section is carrying all the steam it can without an increase in pressure at the extraction point. As the graph shows, the only way kilowatt output can be increased (without pressure rise) is by raising the extraction flow. Study of Figs. 6-15 and 6-17 will show that flow can be raised by increasing the stage pressures and extraction pressure. Ordinarily this is not desirable. Another limit that appears is the maximum capacity of the generator at 1.0 pf. This will be seen at the upper right corner of the curve family.

For purposes of checking actual operating performance these curves can be used in pretty much the same fashion as outlined on page 200 with appropriate corrections for pressure and temperature variations. Instead of checking heat rate the objective would be to compare the actual kilowatt-hours produced with that which should be produced according to the design curves. Again spot checks could be made, but it would prove more practical to run such a check over a week or month. The validity of these checks depends on logging enough readings according to the frequency with which load conditions change.

For a condensing automatic-extraction turbine the entire curve family will shift as the condenser vacuum varies; hence curve families should be prepared for about every 0.2 in. Hg interval over the range of exhaust pressures encountered. This will allow easy interpolation in picking off values.

Mixed-pressure Turbine. In some industries where clean l-p steam discharges from the processes, opportunities exist for producing cheap

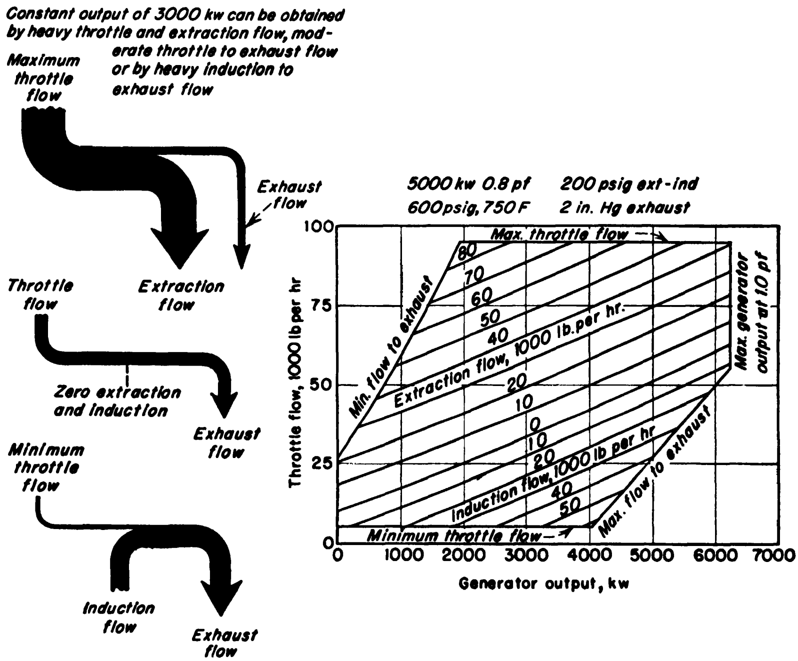


FIG. 6-19. Throttle-flow performance curve for 5,000-kw single automatic-extraction-induction (mixed pressure) condensing turbine. Stream-flow diagrams show how the same output can be produced by different flow combinations.

electrical energy by putting this steam to work in a mixed-pressure turbine. A mixed-pressure turbine can take in h-p steam from a boiler and also l-p steam at an intermediate stage. Basically a mixed-pressure turbine has the same physical arrangement as an automatic-extraction turbine, the difference lying in mode of operation, l-p steam being inducted into the turbine at the stage opening (instead of extracted) and joining the steam entering at the throttle on its way to the turbine exhaust. It is obvious that a mixed-pressure turbine, properly valved, can act also as an automatic-extraction turbine.

Performance chart for a turbine acting in both fashions is given in Fig. 6-19. Constant induction flow lines appear below and parallel to the constant extraction lines. Any constant kilowatt output can be produced by three different combinations of flow as shown by the streamlines at the left: (1) concurrent flow to extraction and exhaust, (2) flow to exhaust only, and (3) concurrent inflow through throttle and induction point.

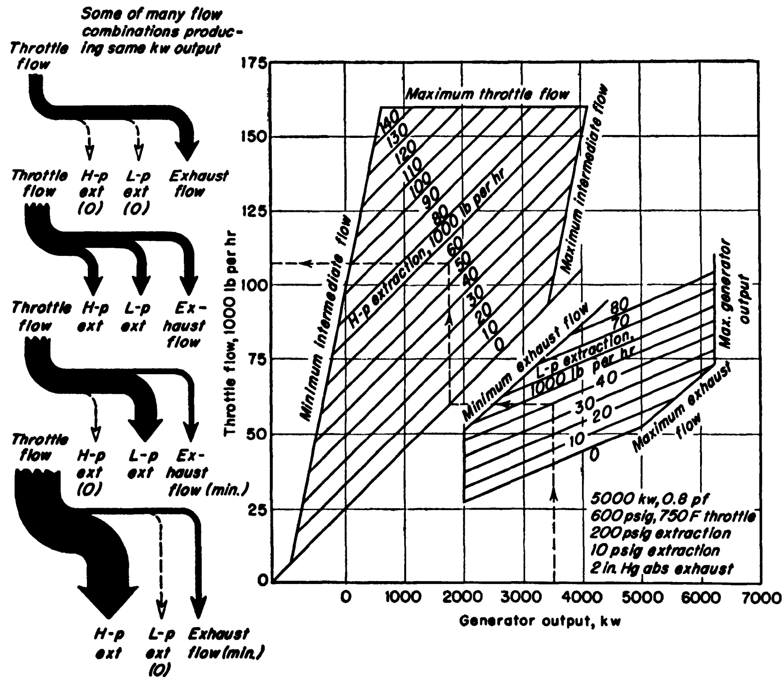


FIG. 6-20. Throttle-flow performance curves for double automatic-extraction condensing turbine. Stream-flow diagrams show a few of many combinations producing the same kilowatt output.

An additional limit exists in a mixed-pressure turbine as shown on the bottom line of the graph on Fig. 6-19. This is the minimum throttle flow. The first section of the turbine must now be protected also from overheating with cooling steam when all the power-producing steam enters at the induction opening.

Double Automatic Extraction. Input-output graphs for double automatic-extraction turbines must have two families of curves as shown in Fig. 6-20. One family (left) refers to various h-p extraction flows, the second family to l-p extraction flows. To read the chart follow the dashed line.* Two additional limits appear on this chart: minimum and maximum flows to the intermediate section of the turbine (following the h-p extraction point). The relative widths of the flow lines illustrated in Figs. 6-18 to 6-20 depend of course on the individual pressures at throttle, extraction points, and exhausts; *i.e.*, as exhaust pressures and extraction pressures increase with a given throttle pressure, the steam flow must increase to maintain a constant output.

ESTIMATING NEW TURBINE PERFORMANCE

Whenever load grows on a power plant, plans must be laid for installing additional turbine capacity. Before calling in manufacturer's salesman, the plant engineer often wishes to gain some idea of what turbines at different steam conditions can be expected to do. A way of making thumb-nail estimates of performance is explained in the following discussion. It is fundamentally based on Figs. 6-4 to 6-7. These curves are based on an engine efficiency of 80 per cent and a back pressure of 1 in. Hg abs.

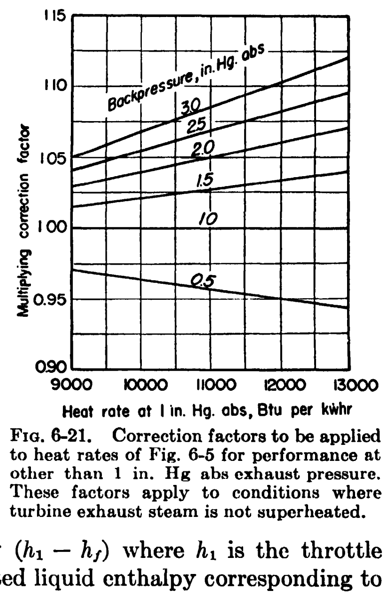


FIG. 6-21. Correction factors to be applied to heat rates of Fig. 6-5 for performance at other than 1 in. Hg abs exhaust pressure. These factors apply to conditions where turbine exhaust steam is not superheated.

If heat rates are wanted at some other back pressure than 1 in. Hg abs, use Fig. 6-21 to find the correction factor. Find the heat rate from Fig. 6-5 at 1 in., and multiply by the correction factor from Fig. 6-21. Steam rate for the new back pressure can then be found by dividing the corrected heat rate by $(h_1 - h_f)$ where h_1 is the throttle steam enthalpy and h_f is the saturated liquid enthalpy corresponding to the new back pressure.

Turbine engine efficiencies vary with initial steam conditions, capacity, and shaft speed. To correct for these conditions the factors in Figs. 6-22 and 6-23 may be applied to the heat rates obtained from Fig. 6-5. They are multipliers as demonstrated in Example 1 for straight condensing turbines. For large units, turbine mechanical efficiencies may be taken as 99 per cent and generator efficiencies as about 98 per cent. The corrected

* The dashed line in Fig. 6-20 shows that 3,500 kw can be produced by 40,000 lb per hr l-p extraction, 60,000 lb per hr h-p extraction, and 107,500 lb per hr throttle flow. Then flow to exhaust is 107,500 − 40,000 − 60,000 = 7,500 lb per hr.

heat rates so computed are for the best operating point (often the rated load). Make allowances for part-load operation, rough estimates of which can be made by assuming the hourly heat input to the turbine at no load equal to about 9 per cent of the heat input at rated load. By

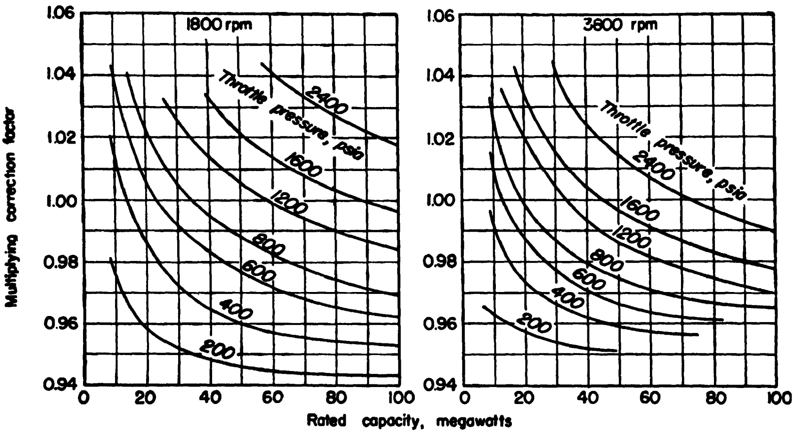


FIG. 6-22. Correction factors to be applied to heat rates of Fig. 6-5 for effects of capacity, rotor speed, and throttle pressure of actual turbines. Turbine mechanical and generator efficiencies are not included in these figures.

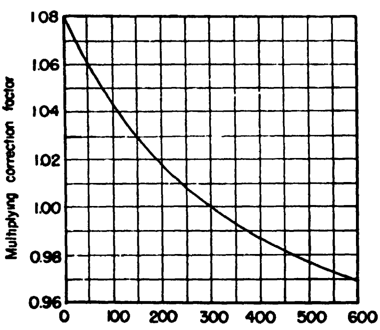


FIG. 6-23. Superheat correction factor to be applied to the heat rates of Fig. 6-5 to adjust for a change in engine efficiency.

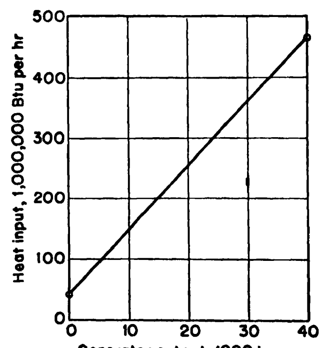


FIG. 6-24. Plot to estimate roughly the part-load heat rates of turbines.

plotting these two points on a graph, as shown in Fig. 6-24, and drawing a straight line between them, part-load inputs and heat rates can be roughly calculated.

Example 1. *Straight Condensing Turbine Performance Estimate:* Find the heat and steam rate of a 40,000-kw turbine designed for 800 psia, 800 F, 3 in. Hg abs back pressure, and shaft speed of 3,600 rpm.

1. From Fig. 6-5, heat rate = 10,710 Btu per kwhr
2. From Fig. 6-21, correction factor for 3-in. back pressure = 1.08
3. From Fig. 6-22, correction factor for capacity, pressure, and speed = 0.98
4. From Fig. 6-5, superheat at 800 F, 800 psia = 800 − 518 = 282 F
5. From Fig. 6-23, correction factor for superheat = 1.003
6. Turbine mechanical efficiency = 0.99
7. Generator efficiency = 0.98
8. Full-load heat rate $= \dfrac{10{,}710 \times 1.08 \times 0.98 \times 1.003}{0.99 \times 0.98}$

   = 11,730; say, 11,700 Btu per kwhr
9. From steam tables: h_1 at 800 psia, 800 F = 1,399 Btu per lb

   h_f at 3 in. Hg abs = 83 Btu per lb
10. Full-load steam rate $= \dfrac{11{,}700}{1{,}399 - 83} = 8.89$ lb per kwhr
11. Approximate heat rate at 20,000-kw load:
    a. Full-load heat input = 11,700 × 40,000 = 468 × 10^6 Btu per hr
    b. No-load heat input = 468 × 10^6 × 0.09 = 42 × 10^6 Btu per hr
    c. From plot of these two points on graph Fig. 6-24, with connecting straight line, input at 20,000 kw = 255 × 10^6 Btu per hr
    d. Heat rate at 20,000 kw $= \dfrac{255 \times 10^6}{20{,}000} = 12{,}750$ Btu per kwhr*

Regenerative-cycle Turbine. Because of the economy realized by bleeding steam from the main turbine for feed-water heating, the regenerative cycle is most frequently applied in the present-day central station. Heat rate of this cycle depends on the initial steam conditions, back pressure, feed-water temperature, and the number of heaters. To estimate the heat rate of a turbine operating on this cycle, all the foregoing curves for straight-condensing calculations are used, and in addition those in Figs. 6-25 to 6-27. Example 2 demonstrates the method of figuring the regenerative-cycle heat rate:

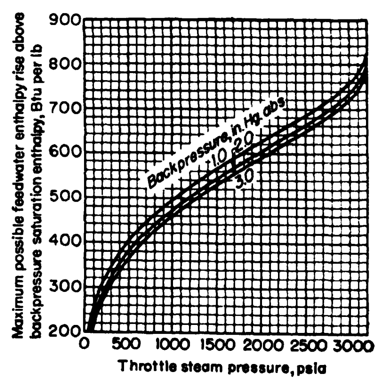


FIG. 6-25. Maximum possible feed-water enthalpy rise above back-pressure saturation enthalpy in turbine cycle for various throttle and exhaust pressures.

Example 2. *Regenerative-cycle Turbine Performance Estimate:* Find the steam and heat rates of a 30,000-kw turbine designed for 900 psia, 900 F, 2 in. Hg abs back pressure, 1,800 rpm, steam bled at three pressures, final feed-water temperature of 350 F.

1. From Fig. 6-5, straight-condensing heat rate = 10,400 Btu per kwhr
2. From Fig. 6-21, correction factor for 2-in. back pressure = 1.044

* Part-load heat rates are dependent upon the turbine-valve arrangement and this type of estimate may be in considerable error.

3. From Fig. 6-22, correction factor for capacity, pressure, and speed = 1.01
4. From Fig. 6-5, superheat at 900 F, 900 psia = 900 − 532 = 368 F
5. From Fig. 6-23, correction factor for superheat = 0.99
6. Turbine mechanical efficiency = 0.99

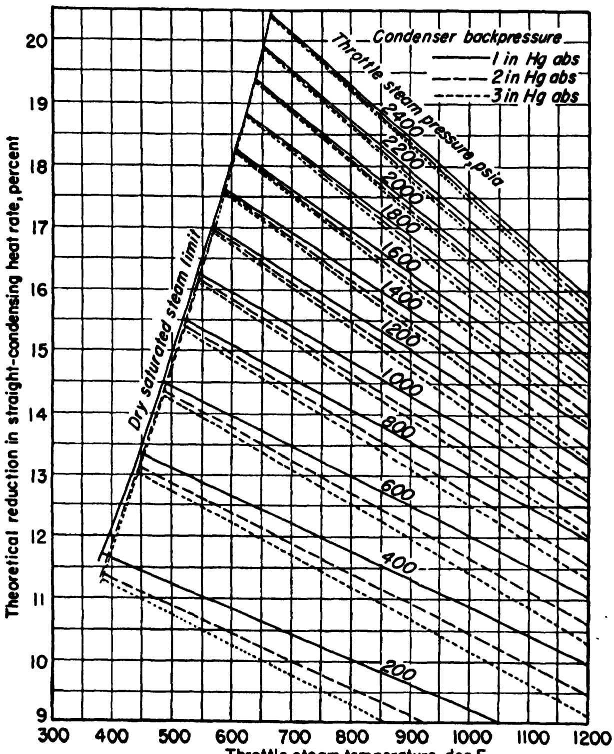


FIG. 6-26. Improvement in the turbine heat rate by ideal regenerative heating compared to straight-condensing performance, as given in Fig. 6-5.

7. Generator efficiency = 0.98
8. Full-load straight-condensing heat rate $= \dfrac{10{,}400 \times 1.044 \times 1.01 \times 0.99}{0.99 \times 0.98}$

$= 11{,}190$ Btu per kwhr
9. From steam tables, h_1 at 900 psia, 900 F = 1452 Btu per lb
   h_f at 2 in. Hg abs = 69 Btu per lb
   h_f at 350 F = 322 Btu per lb
10. From Fig. 6-25, maximum possible feed-water enthalpy rise = 458 Btu per lb
11. Actual enthalpy rise = 322 − 69 = 253 Btu per lb
12. Per cent possible rise in feed-water enthalpy $= \dfrac{253}{458} \times 100 = 55.2$

13. From Fig. 6-26, theoretical reduction in straight-condensing heat rate = 13.6 per cent
14. From Fig. 6-27, correction for number of heaters and actual enthalpy rise = 0.682
15. Correction factor for actual regenerative cycle $= 1 - \frac{13.6}{100} \times 0.682 = 0.907$

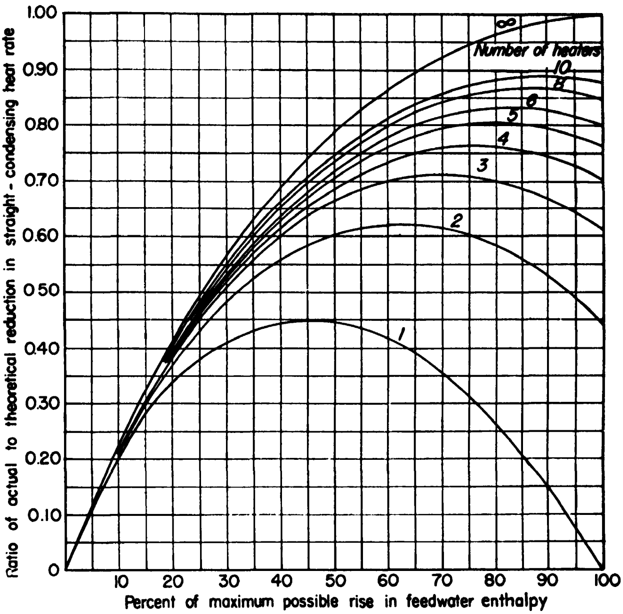


FIG. 6-27. Correction factors for the number of heaters and actual feed-water enthalpy rise in regenerative cycle. Based on paper by J. K. Salisbury (see Bibliography).

16. Full-load regenerative-cycle heat rate = 11,190 × 0.907 = 10,140 Btu per kwhr
17. Full-load regenerative-cycle steam rate $= \frac{10{,}140}{1452 - 322} = 8.98$ lb per kwhr

Part-load heat rates may be estimated roughly by the same methods as outlined in step 11 of Example 1.

Figure 6-25 shows the maximum increase in enthalpy through which the condensate may be raised in preparing it as feed water for different condenser pressures. In other words, it gives the energy needed to raise the liquid enthalpy from saturation in the condenser to saturation at the boiler pressure. The ideal regenerative cycle would theoretically heat the feed water in this fashion, but it would need an infinite number of heaters to do this to obtain maximum possible efficiency. Figure 6-28 indicates how a turbine would have to be arranged with its heaters. Each heater would raise feed-water temperature by a very small part of 1 deg. Bleed steam would have to be tapped off the turbine after each very small drop in pressure. Obviously such an arrangement can never be realized, but the idea is useful because it represents an ultimate condition.

By using this ideal condition the heat rate of a straight condensing turbine can be reduced a certain amount theoretically. The amount of reduction depends on the pressure and temperature of the steam entering the turbine and the pressure of the steam in the condenser. These relations are all given in Fig. 6-26. At a constant temperature greater

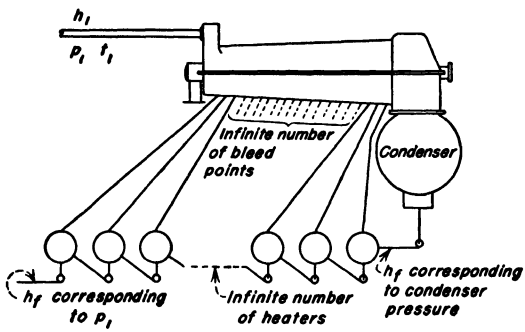


FIG. 6-28. The ideal regenerative cycle has an infinite number of bleed points and heaters. Each heater causes an infinitesimal rise in feed-water temperature.

economies by regenerative heating are realized with increasing pressures, whereas at constant pressure the gain decreases with increasing temperature.

Practically, only a fixed number of heaters can be used, so a gain less than the theoretical amount will be realized. The exact proportion of realizable gain, given in Fig. 6-27, depends on the amount of possible feed-water enthalpy rise effected and the number of heaters used. As the curves show, the efficiency gain starts at zero for a given number of heaters, rises to a maximum, and then drops. Let's look at the case for one heater to gain a better idea of why this relation holds. Figure 6-29*a* shows the one-heater cycle arranged so that it does not raise the feed-water temperature. The bleed steam is tapped off at the turbine exhaust, but since exhaust steam and condensate are at the same temperature, no feed-water temperature rise occurs. As the bleed point on the turbine for the heater is moved toward the throttle (away from the exhaust), the bleed temperature is higher than condensate temperature; therefore, it can heat the feed water. This will result in increasing the cycle efficiency, or lowering the cycle heat rate.

Now let's go to the other extreme and hook the heater at the steam inlet to the turbine, Fig. 6-29*b*. Here the steam can heat the feed water to the saturation temperature corresponding to the boiler pressure and hence effect a 100 per cent possible rise in feed-water temperature. Live heating steam, however, does no work in the turbine and hence does not contribute to increasing the cycle efficiency, or decreasing the turbine heat

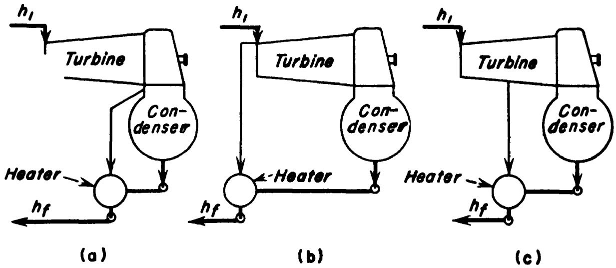


FIG. 6-29. Single-heater regenerative cycle: (*a*) zero temperature rise of feed water; (*b*) maximum possible temperature rise of feed water; (*c*) most efficient arrangement has heater raising feed temperature through 46 per cent of the possible range.

rate. As the bleed point is moved down along the turbine, however, this heating steam does a certain amount of work and hence increases over-all efficiency.

By moving the bleed point from either end of the turbine toward the center (Fig. 6-29*c*), increasing heat rates are realized. The curve in Fig.

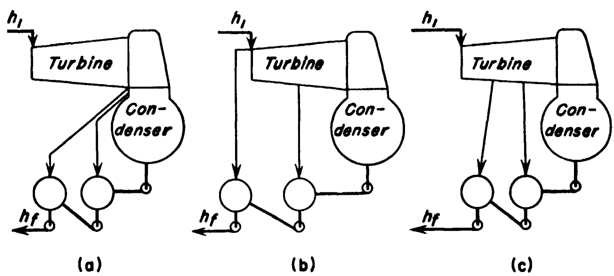


FIG. 6-30. Two-heater regenerative cycle: (*a*) zero temperature rise; (*b*) maximum possible temperature rise; (*c*) most efficient arrangement causes equal temperature increments in heaters for a total rise of about two-thirds of the possible range.

6-27 shows that the optimum point occurs when about 46 per cent of the total possible feed-water enthalpy rise is effected.

The limits and optimum point for two heaters are shown in Fig. 6-30. Hooking both heaters into the turbine exhaust effects zero feed-water temperature rise as in (*a*). Hooking the h-p heater into the turbine steam inlet and the l-p heater in the optimum position for a one-heater arrangement as in (*b*) effects a 100 per cent possible feed-water temperature rise.

But the optimum condition occurs when feed-water enthalpy rise is about 64 per cent of possible, and each heater is tapped in to the "third" points on the turbine as in (*c*) so that each heater raises feed temperature through approximately equal increments.

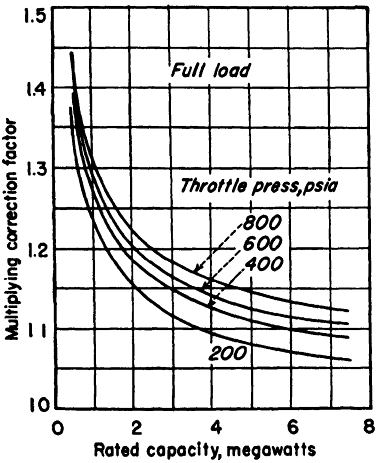


FIG. 6-31. Correction factor for pressure and capacity to be applied to heat rates of Fig. 6-5 for 60-cycle turbines. Based on publications by L. E. Newman (see Bibliography).

Smaller condensing turbines of the 500- to 7,500-kw range do not operate at the same efficiencies as the larger units. Figures 6-31 to 6-33 give the correction factors to estimate the heat rates of the smaller turbines, using Fig. 6-5 as a basis. Figure 6-31 gives the correction factors for capacity and pressure (these factors also include the turbine mechanical efficiency and the generator efficiency). Figure 6-32 gives the correction factors for throttle superheat, and Fig. 6-33 the correction factors for condenser back pressure. These curves are applied in a different order as indicated in Example 3. Figures 6-25 to 6-27 are used for the small turbines, as well as the large, in figuring regenerative-cycle performance.

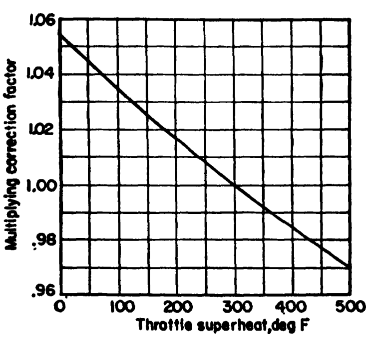


FIG. 6-32. Correction factor for superheat to be applied to the heat rates of Fig. 6-5 for actual 60-cycle turbines. Based on L. E. Newman's publications (see Bibliography).

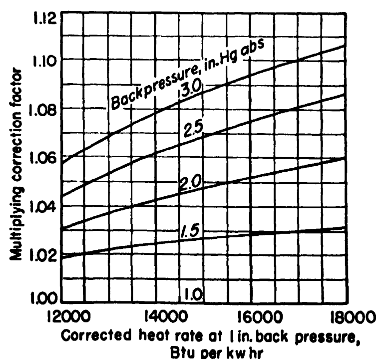


FIG. 6-33. Correction factors for back pressure to be applied to turbine heat rates calculated at 1 in. Hg abs exhaust pressure. Based on L. E. Newman's publications (see Bibliography).

Example 3. *Small Regenerative-cycle Turbine Performance Estimate:* **Find the steam and heat rates of a 1,000-kw turbine designed for 600 psia, 800 F, 2 in Hg abs back pressure, steam bled at two pressures, final feed-water temperature of 300 F.**

1. From Fig. 6-5, straight-condensing heat rate = 11,000 Btu per kwhr
2. From Fig. 6-31, correction factor for capacity and pressure = 1.29
3. From Fig. 6-5, superheat at 600 psia, 800 F = 800 − 486 = 314 F
4. From Fig. 6-32, correction factor for superheat = 0.997
5. Straight-condensing heat rate at 1-in. back pressure = 11,000 × 1.29 × 0.997 = 14,150 Btu per kwhr
6. From Fig. 6-33 correction factor to 2-in. back pressure = 1.043
7. Straight-condensing heat rate at 2 in. = 14,150 × 1.043 = 14,760 Btu per kwhr
8. From steam tables: h_1 at 600 psia, 800 F = 1,408 Btu per lb
   h_f at 2 in. Hg abs = 69 Btu per lb
   h_f at 300 F = 270 Btu per lb
9. From Fig. 6-25, maximum possible feed-water enthalpy rise = 403 Btu per lb
10. Actual enthalpy rise = 270 − 69 = 201 Btu per lb
11. Per cent possible rise in feed-water enthalpy $= \frac{201}{403} \times 100 = 49.9$
12. From Fig. 6-26, theoretical reduction in straight-condensing heat rate = 12.7 per cent
13. From Fig. 6-27, correction for number of heaters and actual feed-water enthalpy rise = 0.602
14. Correction factor for actual regenerative cycle $= 1 - \frac{12.7}{100} \times 0.602 = 0.923$
15. Full-load regenerative-cycle heat rate = 14,760 × 0.923 = 13,620 Btu per kwhr
16. Full-load regenerative-cycle steam rate $= \frac{13{,}620}{1{,}408 - 270} = 11.97$ lb per kwhr

Figure 6-31 indicates the marked decrease in efficiency for the small turbines. This results from the increasing proportion that the fixed and leakage losses assume of the total energy input.

Noncondensing Units. Noncondensing turbines may be designed to exhaust at any pressure above atmospheric. Common practice states the performance in terms of steam rate. The ideal steam rate can be easily found by using a full-scale Mollier chart (Fig. 5-6). Correction factors in Fig. 6-34 then adjust the ideal steam rates to actual at full and half loads for units operating at 50- and 60-cycle speeds as demonstrated in Example 4.

Example 4. *Noncondensing Turbine Performance Estimate:* **Find the steam rates of a 2,000-kw turbine designed for throttle steam at 400 psia and 600 F, to exhaust at 100 psia.**

1. From Mollier chart find h_1 = 1,306.9 Btu per lb
2. At isentropic expansion from h_1, find h_2 = 1,176.7 Btu per lb (100 psia)
3. Ideal steam rate $= \frac{3{,}413}{1{,}306.9 - 1{,}176.7} = 26.2$ lb per kwhr
4. From Fig. 6-34, correction factors for capacity and pressures: full load = 1.47
   half load = 1.72

5. Steam rates: full load = 26.2 × 1.47 = 38.5 lb per kwhr
   half load = 26.2 × 1.72 = 45.1 lb per kwhr
6. Steam flows: full load = 2,000 × 38.5 = 77,000 lb per hr
   half load = 1,000 × 45.1 = 45,100 lb per hr
7. By plotting steam flows vs. kilowatt load and drawing a straight line through points, estimates of steam rates from one-quarter to full load can be made as shown in Fig. 6-35.

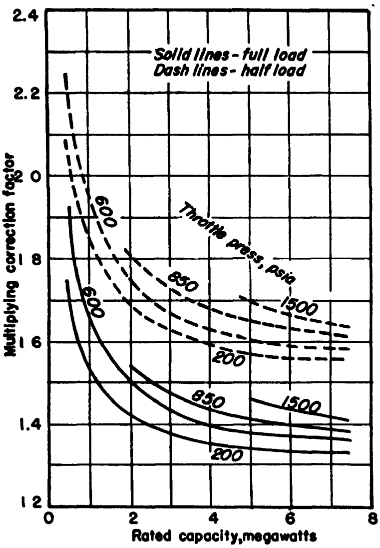


FIG. 6-34. Correction factor for capacity and pressure. Apply to the theoretical steam rate for noncondensing turbines to find the actual steam rate. Based on publications by L. E. Newman (see Bibliography).

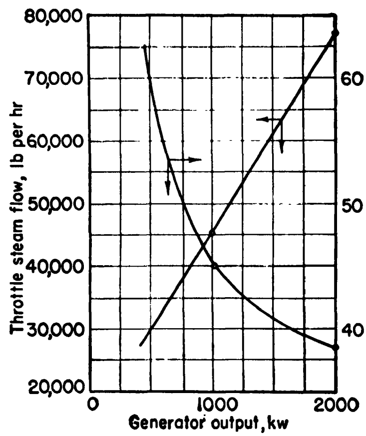


FIG. 6-35. Example of estimated performance curves for 2,000-kw turbine expanding steam from 400 psia, 600 F, to 100 psia.

All the foregoing methods of computing performance data are only approximate and suitable for quick thumbnail estimates. Usually the answers are within 5 per cent of the guarantees that manufacturers will give; special cases probably deviate further. If a prospective buyer's estimate from these data shows the turbine to be below profitable possibilities by a small margin, it pays to ask the turbine manufacturer for more exact performance data to make a more thorough study before drawing final conclusions.

Automatic-extraction Units. Calculating the performance of automatic-extraction turbines involves joint consideration of engine efficiency variation of the different sections. By developing certain constants and tables, a quick method of estimating performance of these turbines has been evolved by L. E. Newman and published in the January, February, March, and April, 1945, issues of *Power Plant Engineering* under the title Modern Extraction Turbines. The General Electric Co. now issues this series of articles as *Bulletin* GET-1294.

Turbine Performance Corrections. Turbines should be operated at their design conditions of throttle steam pressure and temperature and exhaust pressure. However, at times circumstances force the operator to run his equipment at other than design conditions. To know whether the turbine is operating properly at the new conditions its expected performance must be calculated or its actual performance at the new conditions must be corrected to what it would be at standard or design conditions. Manufacturers can supply correction curves that permit making either calculation.

To calculate performance at new conditions, the correction factors are applied as multipliers to guaranteed performance at design conditions. They are applied as divisors to actual performance at new conditions to obtain equivalent performance at design conditions, which can then be compared to guaranteed values. Correction curves may be presented in any one of several different forms, and care must be taken to apply them in proper fashion.

A change in steam temperature and pressures alters the amount of energy that can be converted to work, affects the turbine engine efficiency, and controls the amount of steam that can be passed through the turbines. In calculating performance change, some fixed value must be taken as the reference point; the most convenient is a fixed control-valve opening. For turbines with exhaust pressure less than half the throttle pressure, at any fixed valve opening the steam flow is directly proportional to the absolute throttle pressure and inversely proportional to the square root of the *absolute* throttle temperature. Absolute throttle temperature equals the Fahrenheit temperature plus 460. Change in exhaust pressure within limits does not affect steam flow.

For turbines with exhaust pressure greater than half throttle pressure, steam flow will vary in a different fashion with pressure and temperature change and will also be affected by an exhaust pressure change. Correction factors for these turbines should be obtained from the manufacturer or from an article by E. E. Harris listed in the Bibliography.

With the new steam flow determined for changed steam conditions, the effect of altered energy availability and engine efficiency on kilowatt output is then computed with the aid of given correction curves. Figures 6-36 to 6-38 are examples of some *average* correction curves that may be used for condensing turbines; their application is demonstrated in Example 5. For any specific turbine, correction curves applicable to the

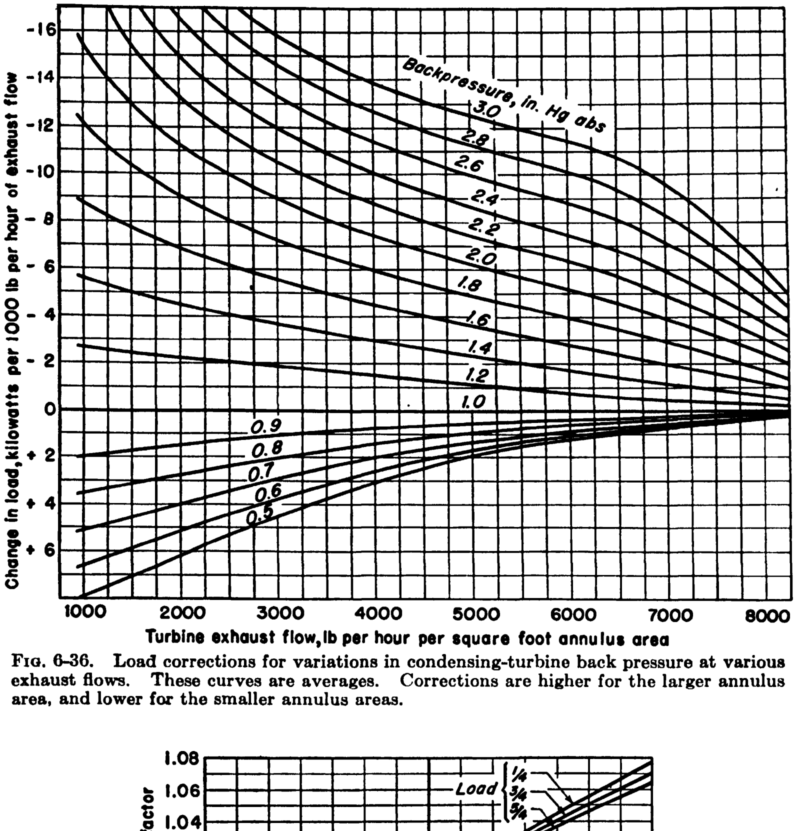


FIG. 6-36. Load corrections for variations in condensing-turbine back pressure at various exhaust flows. These curves are averages. Corrections are higher for the larger annulus area, and lower for the smaller annulus areas.

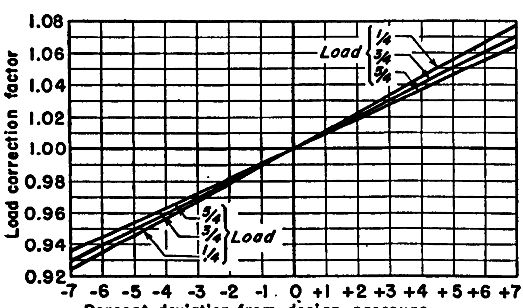


FIG. 6-37. Load-correction factors for changes in throttle steam pressure from design values.

particular machine are preferable to the foregoing average curves. Correction curves can be obtained from the builder or by test.

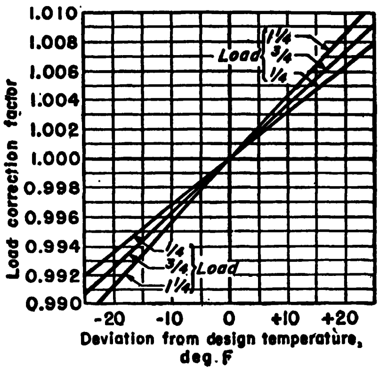


FIG. 6-38. Load-correction factors for changes in throttle steam temperature from design values.

Example 5. Correcting Expected Turbine Performance at Standard Conditions to New Conditions: A 35,000-kw condensing turbine is designed to operate at 600 psig, 850 F throttle conditions and 1.0 in. Hg abs back pressure. The annulus area of the exhaust stage is 35.4 sq ft. If the expected steam rate of 30,000 kw is 8.00 lb per kwhr, find the output and steam rate for the same control valve opening at throttle conditions of 585 psig and 830 F, and 1.6 in. Hg abs back pressure.

1. Steam flow at 30,000 kw = 30,000 × 8.00 = 240,000 lb per hr
2. Corrected steam flow at new conditions and constant control valve opening =

$$240{,}000 \times \frac{585 + 14.7}{600 + 14.7} \times \sqrt{\frac{850 + 460}{830 + 460}} = 240{,}000 \times 0.975 \times 1.008 = 236{,}000 \text{ lb per hr}$$

3. Exhaust flow $= \dfrac{240{,}000}{35.4} = 6{,}780$ lb per hr per sq ft annulus area
4. From Fig. 6-36, change in load at 1.6 in. Hg abs and 6,780 lb per hr per sq ft = −2.2 kw per 1,000 lb per hr of exhaust flow
5. Load correction factor for change in back pressure

$$= \frac{\left[30{,}000 + \left(\dfrac{-2.2 \times 240{,}000}{1{,}000}\right)\right]}{30{,}000} = 0.982$$

6. Change in throttle pressure $= \dfrac{585 - 600}{600 + 14.7} = -0.0244$, or -2.44 per cent
7. From Fig. 6-37, load correction factor for change in throttle pressure = 0.974
8. Change in throttle temperature = 830 − 850 = −20 F
9. From Fig. 6-38, load correction factor for change in throttle temperature = 0.992
10. Total load correction factor = 0.982 × 0.974 × 0.992 = 0.949
11. Corrected load at new conditions = 30,000 × 0.949 = 28,500 kw
12. Corrected steam rate at 585 psig and 830 F throttle conditions, and 1.6 in. Hg abs back pressure $= \dfrac{236{,}000}{28{,}500} = 8.28$ lb per kwhr

Having computed corrected steam flow and corrected output, the ratio of the two will give the corrected steam rate. By computing a sufficient number of points from the original guarantee or test-performance curves of a turbine, a new performance curve at the new steam conditions can be plotted.

In regenerative cycles there are many other conditions that must be corrected for as they depart from standard, to know the true turbine performance. Some of these are make-up water flow, heater temperature terminal differences, main condenser subcooling, and change in stage and heater pressures.

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CHAPTER 7

TURBINE GOVERNORS

The commercial practicality of both steam and gas turbines is due among other things to the governing systems designed for their control. Control is required for power production, and in the case of the steam turbine when it supplies process-steam demands. Turbine governing is of two broad types: *speed* and *functional*. In speed governing the steam flow varies with load change so that speed remains essentially constant. Speed governors can control speed within 0.1 per cent or less over the entire load range and find extensive use in electrical power generation. In functional governing, steam is also controlled but the reasons are varied. Maintaining exhaust- or extraction-steam pressure constant and the controlled use of waste steam are duties of functional governors.

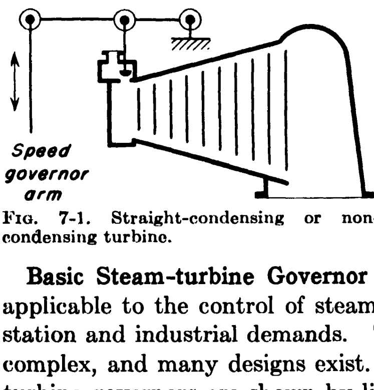


FIG. 7-1. Straight-condensing or noncondensing turbine.

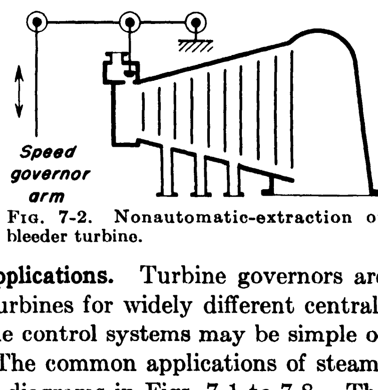


FIG. 7-2. Nonautomatic-extraction or bleeder turbine.

Basic Steam-turbine Governor Applications. Turbine governors are applicable to the control of steam turbines for widely different central-station and industrial demands. The control systems may be simple or complex, and many designs exist. The common applications of steam-turbine governors are shown by line diagrams in Figs. 7-1 to 7-8. The illustrations show the manner of control but do not attempt to show the specific design of the governor itself. In all cases it is to be understood that the arm of the speed governor is caused to move by the action of a primary speed governor (not shown) linked in most cases to a servomechanism (not shown). The performance and characteristics of these types are as follows:

Figure 7-1, *Straight-condensing or noncondensing:* The governor responds to speed change induced by a load change. In turn the steam valves are opened or closed so that the steam flow is changed to correspond to the load. The load and exhaust flow are variable, and inlet steam depends upon the load.

Figure 7-2, *Nonautomatic-extraction or bleeder:* The governor functions like that in Fig. 7-1. Load is variable. Exhaust flow depends upon inlet steam and the amount of bleeder steam; inlet steam depends upon the load.

Figure 7-3, *Inlet-pressure regulation:* Most commonly applied to condensing turbines. The governor maintains steam-chest pressure constant and in so doing permits the turbine to use all excess waste steam from other sources up to its capacity. The turbine operates in parallel with other units which are speed-governed and so obtains its speed control. The speed governor on the unit shown functions only when the speed varies beyond a given amount. The inlet steam flow is variable; the exhaust flow and load depend on inlet steam.

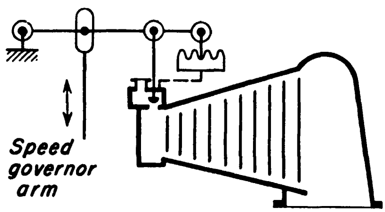


FIG. 7-3. Straight-condensing or noncondensing-waste heat or inlet-pressure-regulation turbine.

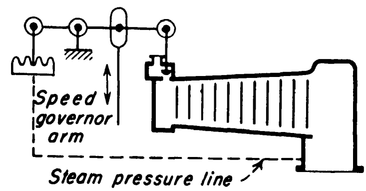


FIG. 7-4. Noncondensing back-pressure-regulation turbine.

Figure 7-4, *Back-pressure regulation:* The governor functions like that in Fig. 7-3, except that back pressure is held constant. This unit must also operate in parallel with other speed-governed turbines to obtain its speed control while its own speed governor acts as a safety control. The exhaust flow is variable; inlet steam and load depend on exhaust flow.

Figure 7-5, *Automatic-extraction or induction* (*single-stage*)*:* This turbine operates to supply a power load and to supply process steam at a controlled pressure or, conversely, to use an available steam supply by mixing it with the main steam flow through the unit. Governing is a combination of a speed governor and a pressure regulator. Speed change reacts either to raise or to lower both valves with no effect on extraction steam pressure. A variation in steam flow at the extraction point causes a pivoting of the governor arm to the valves. This causes the valves to have opposite motions. Hence load is kept constant, but a variation in extraction or induction steam is provided for. With this unit the load and extraction or induction steam flow are variable; inlet and exhaust flow depend upon both load and process steam.

Figure 7-6, *Automatic-extraction or induction* (*two-stage*)*:* With this system the speed governor meets a load change by moving two three-arm levers to adjust the position of all valves and change the through-steam flow. Pressure-responsive governors maintain constant extraction pressure with a steam demand change by moving the governor and extraction

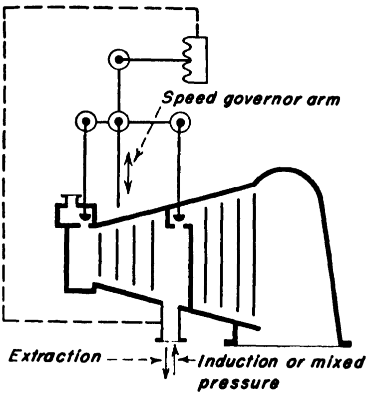


FIG. 7-5. Automatic-extraction (single-stage or induction turbine.

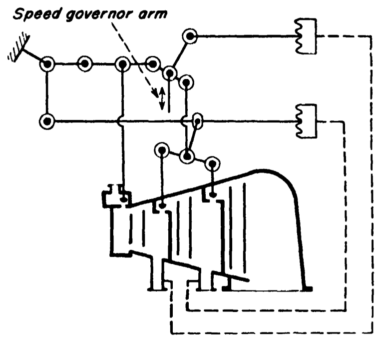


FIG. 7-6. Automatic-extraction (two-stage) or induction turbine.

valves in opposite ways to alter the ratio between through and extraction flows. Load- and pressure-controlled extraction flows are variable; inlet and exhaust steam flow depend upon the former.

Figure 7-7, *Automatic-extraction with inlet-pressure regulation:* Governor gear as indicated permits a turbine to use any waste steam to the amount available and at the same time to supply a controlled lower pressure extraction steam flow. The unit must be operated with others in parallel for speed control. Load and exhaust flow depend on inlet and extraction steam while the latter two flows are independently variable although they are pressure-controlled.

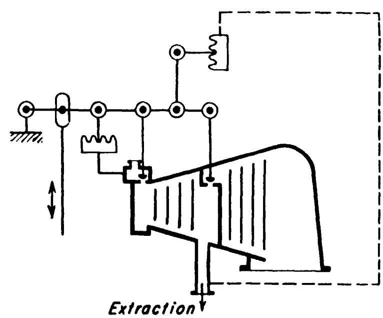


FIG. 7-7. Automatic-extraction (single-stage) turbine with inlet-pressure regulation.

Figure 7-8, *Automatic-extraction with back-pressure regulation:* This governor is similar to that in Fig. 7-7 except that back pressure rather than inlet pressure is controlled together with extraction steam. Here the exhaust and extraction flows are variable while inlet steam flow and load depend on these former steam flows.

The governors outlined above cover their principal uses in steam-turbine control. Modifications of them are used in further applications to such cases as simultaneous control of inlet pressure, back pressure, and extraction pressures although these are rarely met in practice. Designs of governors are today possible for almost any form of operating need.

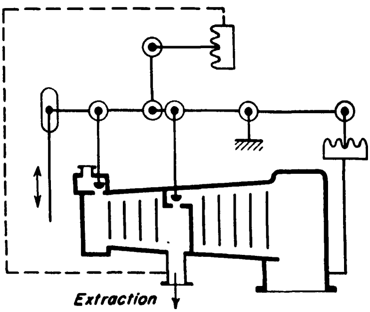


FIG. 7-8. Automatic-extraction (single-stage) turbine with back-pressure regulation.

Basic Gas-turbine Governor Applications. The control of gas-turbine output is essentially simple although the present development of stationary units has not led to a wide number of methods. At present the common methods are as indicated below:

Figure 7-9, *Single-shaft, open-cycle, variable-temperature:* Here fuel control is employed with a consequent variable inlet temperature at constant air-flow conditions. High excess air controls the outlet gas temperature. This arrangement is simple but gives poor part-load performance.

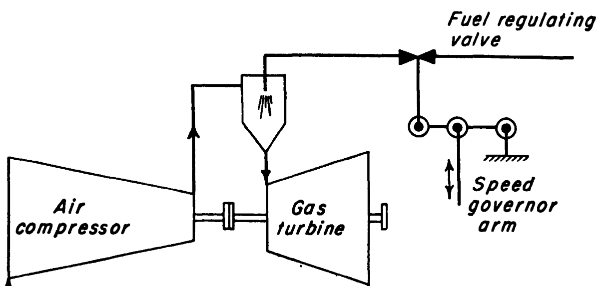


FIG. 7-9. Single-shaft, open-cycle, variable-temperature gas turbine.

Figure 7-10, *Two-shaft, open-cycle, constant-temperature:* Here fuel control by a governor is also employed for a turbine and compressor on one shaft which operates at varying speed for variable air flow. The governor responds to speed changes of the load turbine on another shaft while temperature to the load turbine is independently maintained. The arrangement gives good part-load characteristics.

Figure 7-11, *Single-shaft, closed-cycle, constant-temperature:* In the closed cycle, varying load demands are met by adjusting the working medium density. Air bleeds from the system to a l-p accumulator on appreciable load decrease and enters it from a h-p accumulator on load increase.

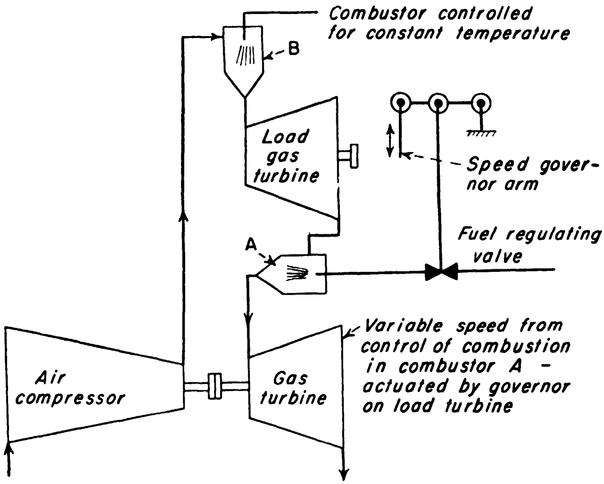


FIG. 7-10. Two-shaft, open-cycle, constant-temperature gas turbine.

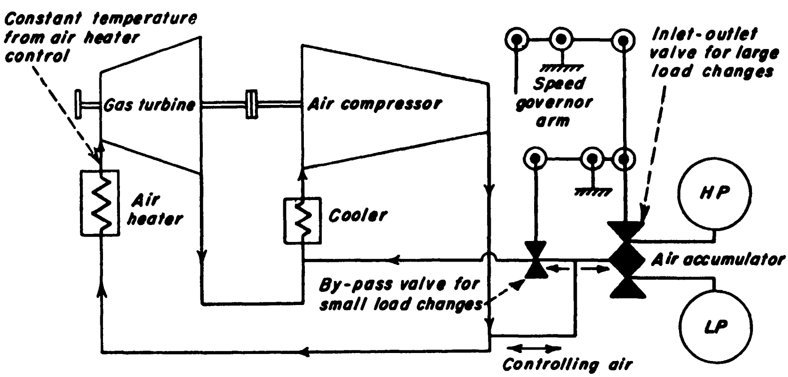


FIG. 7-11. Single-shaft, closed-cycle, constant-temperature gas turbine.

These bleeding operations are operated by a speed-control governor which for small load changes, however, simply by-passes air as indicated in the figure.

Governor Elements. Governor systems for turbines can be constructed from certain basic mechanical elements to satisfy practically all conditions of load and process-steam flow demanded in power plants. The specific designs of these components differ for various companies, and the completed governing systems are of varying complexities for the different applications illustrated in Figs. 7-1 to 7-11. Structurally, steam-turbine governors can be divided into three general classes:

Hydraulic
Mechanical
Electrical

In many governors a combination of these features is used in the design of the completed system.

The elements comprising governing systems are well outlined by a set of terms and definitions which form Sec. II of Recommended Specifications for Speed Governing of Steam Turbines, prepared by a joint AIEE-ASME committee. They are given below slightly abridged.

Speed-governing system: The speed-governing system includes the speed governor, the speed-control mechanism, and the governor-controlled valves.

Speed governor: The speed governor includes only those elements which are directly responsive to speed and which position or influence the action of other elements of the speed-governing system.

Speed-control mechanism: The speed-control mechanism includes all equipment such as relays, servomotors, pressure or power-amplifying devices, levers, and linkages between the speed governor and the governor-controlled valves.

Governor-controlled valves: The governor-controlled valves include those valves which control the energy input to the turbine and which are normally actuated by the speed governor through the medium of the speed-control mechanism.

Speed changer: The speed changer adjusts the speed-governing system to change the speed or power output of the turbine while the turbine is in operation.

Load limit: The load limit acts on the speed-governing system to prevent the governor-controlled valves from opening beyond the position for which the device is set.

Steady-state speed regulation: For straight condensing and noncondensing turbines and nonautomatic extraction turbines, the steady-state speed regulation is the change in sustained speed, expressed in per cent of rated speed, when the power output of the turbine is gradually reduced from rated power output to zero power output with identical settings of all adjustments of the speed-governing system. Thus,

$$R_s = \frac{N_0 - N}{N_r} \times 100$$

where R_s = steady-state speed regulation, per cent
N_0 = speed at zero power output, rpm
N = speed at rated power output, rpm
N_r = rated speed, rpm

For automatic-extraction turbines and mixed-pressure turbines, the steady-state speed regulation, expressed in per cent of rated speed, is the change in sustained speed corresponding to the load change from rated load to no load with normal rated pressures in the controlled extraction stages, with zero extraction flows, and with identical settings of all adjustments of the speed-governing and pressure-regulating systems.

It is the speed change expressed in per cent of rated speed when, with normal extraction pressures and zero extraction flows, the load is reduced from rated load to the minimum load at which these extraction conditions are permitted, times the ratio of rated load to this load change. Thus,

$$R_s = \frac{N_m - N}{N_r} \times \frac{P_r}{P_r - P_m} \times 100$$

where R_s = steady-state speed regulation, per cent
N = speed at rated power output with rated extraction pressures and zero extraction flows, rpm
N_m = speed at minimum load at which rated extraction pressures with zero extraction flows can be obtained, rpm
N_r = rated speed, rpm
P_r = rated load, kw
P_m = minimum power output at which rated extraction pressures with zero extraction flows is permitted, kw

For the purposes of these specifications "rated power output" is the turbine nameplate rating in kilowatts.

The steady-state speed regulation may vary with different settings of the speed changer. For the purposes of these specifications the percentage regulation is based upon the speed changer set to give rated speed with rated power output.

Steady-state incremental speed regulation: The steady-state incremental speed regulation at a given steady-state speed and power output is the rate of change of the steady-state speed with respect to the power output. It is the slope of the tangent to the steady-state speed-power output curve at the point of power output under consideration. It is expressed in per cent of rated speed when the difference in sustained speed (expressed in per cent of rated speed) for any two points on the tangent is divided by the corresponding difference in power output (expressed as a fraction of the rated power output).

For the purposes of these specifications the several points of power output at which the values of steady-state incremental speed regulation are derived is based upon the rated speed obtaining at each point of power output.

Speed regulation (general): Speed regulation is considered positive when speed increases with decrease in power output.

All definitions concerning speed regulation are based upon zero dead band.

Dead band: Dead band is the total magnitude of the sustained speed change within which there is no resulting measurable change in the position of the governor-controlled valves. Dead band measures the insensitivity of the speed-governing system and is expressed in per cent of rated speed.

For a given position of the governor-controlled valves, there may be both an increase and a decrease in speed before there is a change in the position of the valves. In this case the total magnitude of the sustained speed change is the sum of these speed changes.

Stability of the speed-governing system: Stability is the capability of the speed-governing system to position the governor-controlled valves so that a sustained oscillation of turbine speed or energy input to the turbine is not produced by the speed-governing system during operation under sustained load demand or following a change to a new sustained load demand. A small forced high-frequency oscillation introduced to overcome mechanism friction is excluded.

Overspeed: Overspeed is the maximum increase in speed from a stated speed following a sudden reduction of turbine power output. It is expressed in per cent of rated speed.

Steam-pressure regulating system: The steam-pressure regulating system includes the pressure regulator, the pressure-control mechanism, and the pressure-controlled valves.

Pressure regulator: The pressure regulator includes only those elements which are directly responsive to the controlled steam pressure and which position or influence the action of other elements of the steam-pressure regulating system.

Pressure-control mechanism: The pressure-control mechanism includes all the equipment such as relays, servomotors, pressure or power-amplifying devices, levers, and linkages between the pressure regulator and the pressure-controlled valves.

Pressure-controlled valves: The pressure-controlled valves include those valves which control the flow of steam through the turbine, normally actuated by the pressure regulator through the medium of the pressure-control mechanism. These valves may also be actuated by the speed governor through the medium of the pressure-control mechanism. The governor-controlled valves may also be pressure-controlled.

Pressure changer: The pressure changer adjusts the steam-pressure regulating system to change the pressure (or flow) of the steam under control, while the turbine is in operation.

Steady-state pressure regulation: Steady-state pressure regulation is the change in sustained pressure of the steam under control (expressed in per cent of rated pressure) when the controlled steam flow is gradually reduced from rated flow to zero flow, with identical settings of all adjustments of the steam-pressure regulating system and with the turbine operating at rated speed.

For the purpose of these specifications, "rated flow" is (1) for automatic-extraction and mixed-pressure turbines, the specified maximum extraction or induction steam flow; (2) for noncondensing turbines, the throttle flow corresponding to the rated power output of the turbine; (3) for initial pressure-controlled turbines, the throttle flow corresponding to the rated power output of the turbine.

Stability of steam-pressure regulating system: Stability is the capability of the steam-pressure regulating system to position the pressure-controlled valves so that "sustained oscillations" of the controlled steam pressure, or of the energy input to the turbine, are not produced by the steam-pressure regulating system during operation under sustained flow demands or following a change to another value of sustained flow demand.

Compensated control system: A compensated control system is one which is provided with interconnections between the control mechanisms of both the speed-governing and the steam-pressure regulating systems to permit simultaneous action of both systems in response to a change in either speed or controlled steam pressure.

Centrifugal or Flyball Type of Primary Speed Governor. The simplest and most familiar speed governor is the direct-acting flyball arrangement of Fig. 7-12. In the usual form, flyballs or other shaped weights are connected directly or through gears to the turbine shaft and rotated at a rate proportional to turbine speed. Arms connect the weights to a sliding collar, acted against or restrained from moving by a spring. A lever connects the collar to a governing valve controlling the steam flow. As the weights rotate, centrifugal force tends to move the collar against the spring force. This motion in turn changes the position of the steam valve. Hence, for the simple governor, there will be a different speed corresponding to each load. Constant speed can be had if a speed changer, discussed later, is added to the governing system. This form of centrifugal governor shown in Fig. 7-12 operates with appreciable movement of the collar and is a form of "travel governor." A given speed change causes a different amount of movement of the weights depending upon whether the balls are in a high or a low position. The valve movement is not proportional to speed change for this type of governor which is undesirable. It means that, when plotted, the steam-flow and turbine-speed relationship is not a straight line. This means that for a given setting the per cent of regulation of the governor varies at different turbine speeds.

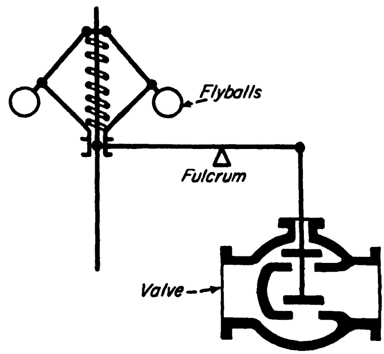


FIG. 7-12. Simple direct-acting flyball governor.

Another form of centrifugal governor (Fig. 7-13) is called a "strap governor." The sliding collar of the first type is replaced by a metal strap. The strap governor has a very small movement caused only by the elastic deformation of the strap metal under the centrifugal action of the weights. In this way the valve travel at different governor positions is very closely the same for similar speed changes.

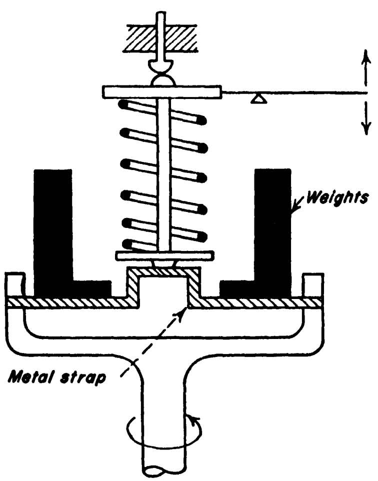


FIG. 7-13. Centrifugal strap primary governor.

The centrifugal governor alone has limitations in that (1) change in speed accompanies a load change, (2) it is comparatively insensitive, and (3) it must be built in large sizes (inherently insensitive) to obtain large operating forces. Its size is therefore limited. It is, however, suitable to operate directly the steam valves for many smaller turbines, as those of the mechanical-drive type. Figure 7-14 shows such an application.

In many installations ball bearings at joints and mechanical oscillators are used with the centrifugal type to vibrate the linkages of the governor. This reduces the effect of friction which is deterimental to governor action as it allows a speed change of the turbine during which no governor motion occurs. This is referred to as the "dead band" of a governor.

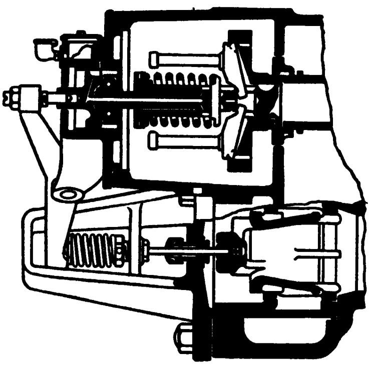
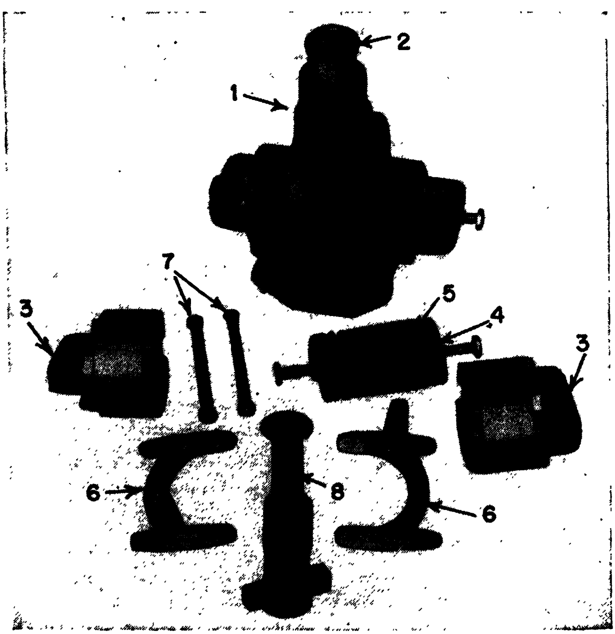


FIG. 7-14. Direct-acting flyball type of governor.

The assembly and parts of a centrifugal spring type of speed governor are shown in Fig. 7-15. The shape, proportion, and parts of the primary mechanical governor as designed by different manufacturers naturally vary from the one illustrated. The essential parts of a set of rotating weights and a restraining spring are common to governors of this type.



(*Courtesy of Allis-Chalmers Manufacturing Co.*)
FIG. 7-15. Spring-type speed governor.

Fluid-pressure Type of Primary Speed Governor. In place of a mechanical-speed-sensitive governor, a fluid type as shown in Fig. 7-16 is used by many companies. It consists of a governing oil pump driven by the turbine shaft. As the speed of the turbine varies, the discharge oil pressure changes. The governor is sensitive since the oil pressure varies as the square of the turbine speed. Suction effect is overcome by supplying the pump with constant-pressure oil. Other factors that may affect operation, such as oil temperature changes, are minimized so their effect is negligible. The oil can be used directly, as shown, to position the turbine steam valve. Figure 7-17 indicates an application of a fluid governor directly connected to the steam valve.

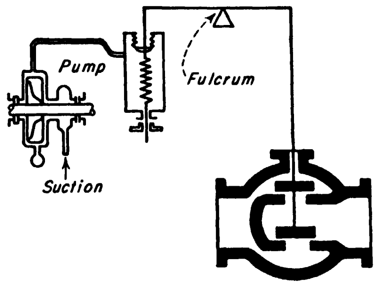


FIG. 7-16. Simple direct-acting fluid governor.

The fluid pump, as shown in Fig. 7-18, is one of several commercial forms. Oil at high pressure from the main oil-pump discharge flows through the orifice shown to a series of radial nozzles leading to a hollow chamber in the shaft. Drain openings lead from this chamber. When the turbine is at rest, oil from the orifice flows into the hollow chamber and out of the drains, and no pressure is applied to the governor transformer. As the turbine shaft rotates, centrifugal force in the nozzles opposes the oil flow, and a pressure is developed in the passageway leading from the orifice to the governor transformer. At normal speeds this pressure is always less than the orifice pressure, and a small amount of oil always flows from the orifice to the hollow chamber and out of the drains. Speed changes cause a change in the centrifugal force and hence in the passageway pressure to the transformer or relay. This pressure change is used to cause the turbine valves to open or close for load changes.

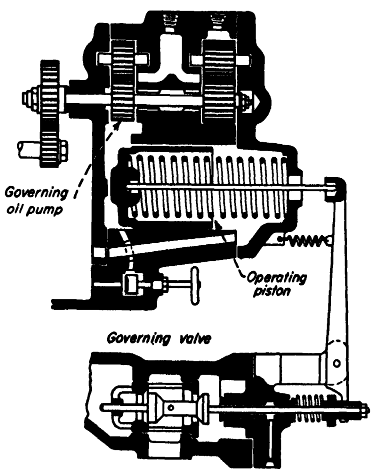
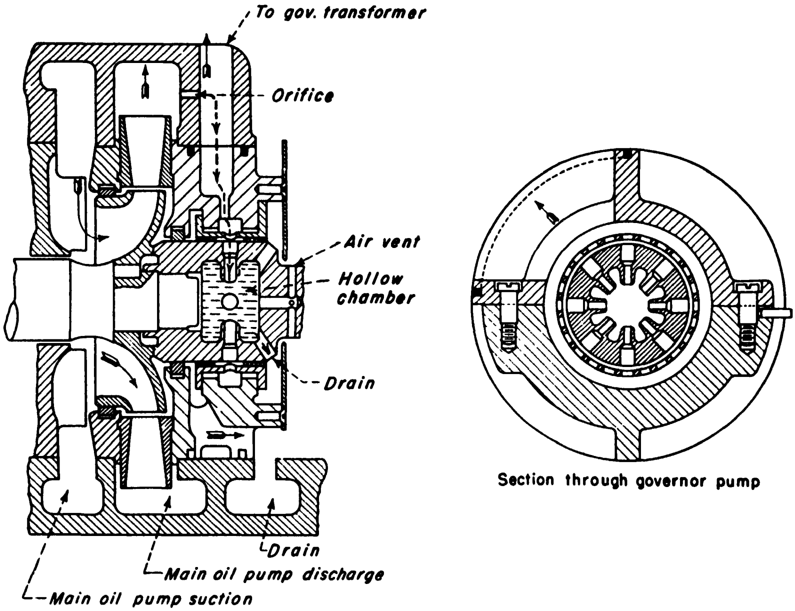


FIG. 7-17. Direct-acting fluid type of governor.

Primary-pressure Governors. A governing system including features other than speed control is necessary for extraction, admission, inlet, or exhaust pressure regulation. The primary type of governor is commonly a spring-balanced bellows which is responsive to pressure changes. Movement of the bellows is amplified and in some manner relayed to the mechanism of the governing system. Designs are such that action is

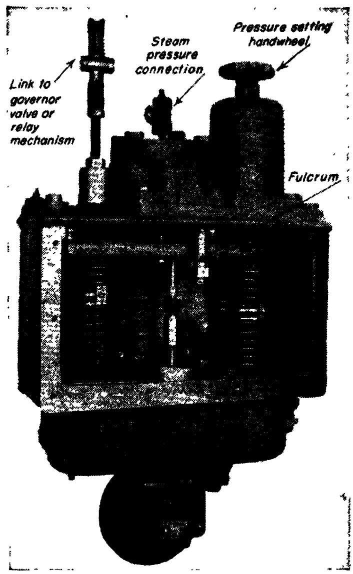


(*Courtesy of Westinghouse Electric Corp.*)

FIG. 7-18. Typical fluid governor pump.

obtained by small movements of the bellows. This small movement produces very small stress in the metal of the bellows—which is conducive to long-time operation.

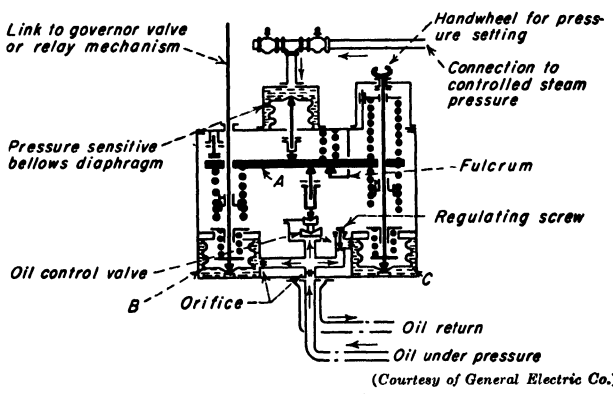
A typical example of a primary-pressure governor is shown in Fig. 7-19. As such, a connection is made to the region of pressure control (*viz.*, the exhaust hood for back-pressure control). A system of a sensitive bellows diaphragm and amplifying linkages is used to actuate a link leading to the steam valve or governor relay. The operation is more evident from the schematic diagram of Fig. 7-20. A steam pressure change reacts on the pressure-sensitive bellows diaphragm. This in turn moves lever *A* about its fulcrum and varies the setting of the oil-control valve. As this valve is closed, oil under pressure acts on bellows *B*. This moves the link to the governor valve or relay and changes the compression in the spring on the link. The movement of bellows *B* is paralleled, but more slowly, by bellows *C* which is also actuated by oil pressure through a regulating screw as shown. Its movement offsets the effect of the compression change in the spring on the governor link stem. Then as bellows *C* responds, the steam pressure will return to its correct value.



(*Courtesy of General Electric Co.*)

FIG. 7-19. Typical primary-pressure governor.

Governor Amplifying and Control Mechanisms. A direct-acting governor although simple in construction must, to reduce friction, be built in small sizes if it is to be at all accurate. A governor must do work against this internal friction and also do work to move the steam valves. The force necessary to move valves may have to be very large. For example, a 5-in. valve opening against an unbalanced pressure of 400 psi requires a force at the start of about 7,800 lb. The double requirement of sensitivity and development of large forces is met by the use of various systems of relays, servomotors, and pressure- or power-amplifying devices.



(*Courtesy of General Electric Co.*)

FIG. 7-20. Schematic diagram of primary-pressure governor.

A common construction of such governor is shown in Fig. 7-21. This relay governor obtains amplification by using the governor force to move a light and relatively frictionless pilot valve. This valve controls the flow of h-p oil to a piston. The piston in turn is powered by the oil and can then operate the governor valves and other mechanisms and develop any desired force.

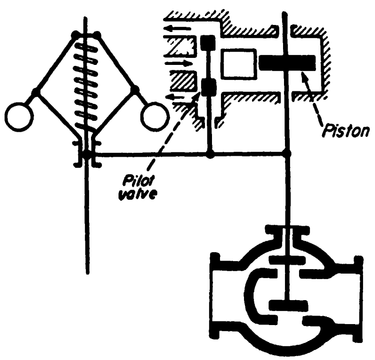


FIG. 7-21. Centrifugal governing mechanism with hydraulic-power amplifier.

A fundamental requirement with all these hydraulic-powered pistons, also called "servomotors," is that the motion of the piston must be stopped when it has accomplished a change in the steam valve position. Some form of restoring mechanism is part of these systems. Two common constructions are shown in Fig. 7-22. In both, motion from the primary governor with speed change moves the pilot valve. The power piston movement initiated by this pilot valve acts as shown to reset the pilot valve in its mid-position and so cause the power piston to come to rest at some new valve position.

In larger units the linkage between the governor proper and the hydraulic cylinder or servomoter, or between the servomotor and the admission

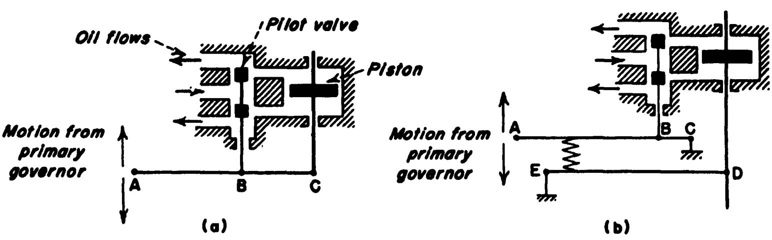


FIG. 7-22. Typical restoring mechanisms.

valves, may be extensive. This condition tends to increase the work that the governor must do as these links require supports which increase the friction. In these cases to maintain sensitivity a double relaying system is employed, as in Fig. 7-23, which consists in having a hydraulic piston actuated by a pilot valve. This piston in turn moves a pilot valve controlling the main power piston. Steam valves as shown are operated by this last piston. The restoring mechanism feature is used for both pilot valves. Figure 7-24 shows the double-relaying principle employed with a somewhat different design. A short linkage connects the governor to the first-stage amplifier, which in turn actuates the second stage. The system shown uses a cup valve controlling the oil pressure under the first-stage operating piston by regulating h-p oil leakoff.

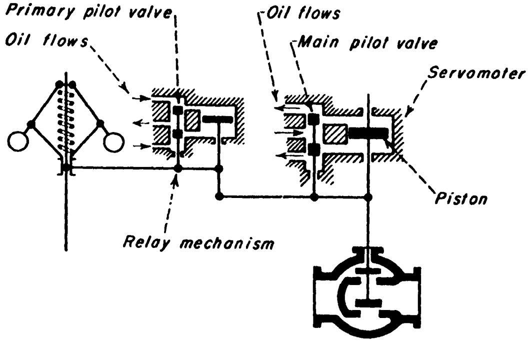


FIG. 7-23. Centrifugal governing mechanism with double hydraulic-power amplifier using pilot valves.

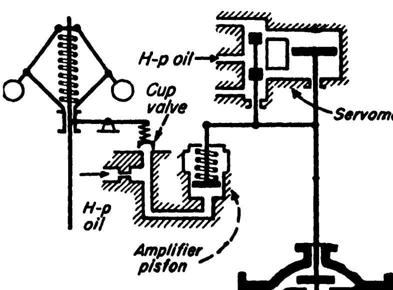


FIG. 7-24. Centrifugal governing mechanism with double hydraulic-power amplifier using cup and pilot valves.

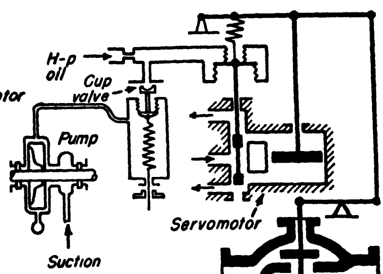
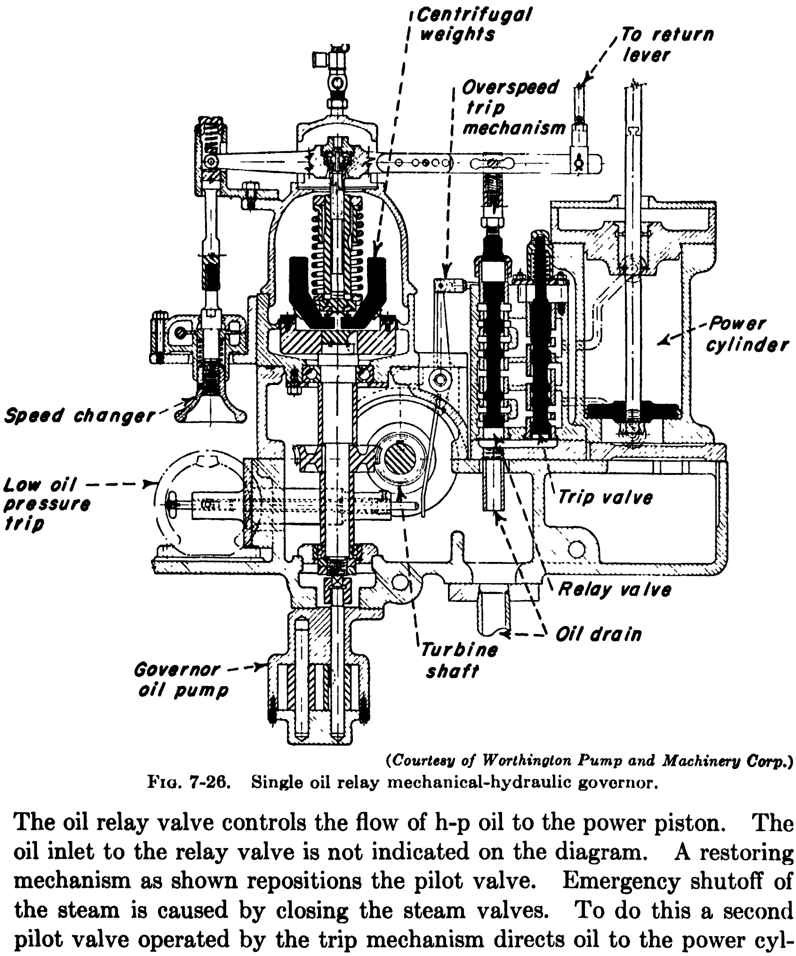


FIG. 7-25. Shaft-driven impeller governor with power amplifiers.

The primary governor in the cases so far shown has been of the centrifugal weight type. The oil impeller or fluid pressure type is also used in conjunction with a relay and amplification system. In Fig. 7-25 this form of primary governor is shown with a double amplifier using the cup valve for the first and the pilot valve for the second stage of amplification.

Relay and Transformer Types of Governors. The principles employed in designing amplifying devices for governors have been shown in the preceding paragraph. Some of these principles, as such or in some combination, are used by the various turbine manufacturers in the governors for their turbines. Several such applications are shown here. In Fig. 7-26 a single relay governor is indicated. The main oil relay valve is actuated as the centrifugal weights shift their position with speed change.

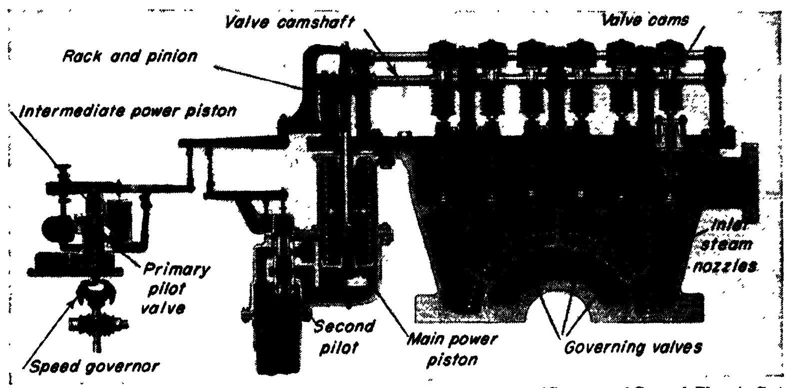


(*Courtesy of Worthington Pump and Machinery Corp.*)

FIG. 7-26. Single oil relay mechanical-hydraulic governor.

The oil relay valve controls the flow of h-p oil to the power piston. The oil inlet to the relay valve is not indicated on the diagram. A restoring mechanism as shown repositions the pilot valve. Emergency shutoff of the steam is caused by closing the steam valves. To do this a second pilot valve operated by the trip mechanism directs oil to the power cylinder. A speed changer varying the spring tension is employed. The trip mechanism responds to overspeed and to low oil pressure.

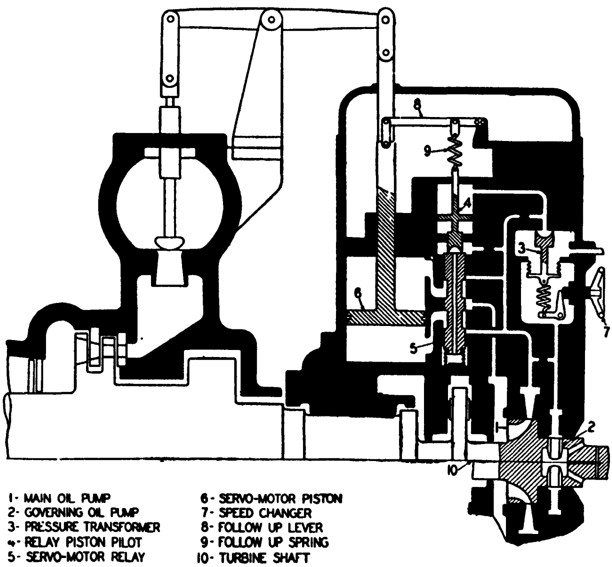
In Fig. 7-27 a double-relay governor is shown. Here the speed governor also actuates the primary pilot valve. This governs the movement of an intermediate power piston which operates downward under spring action as oil pressure beneath it is relieved, or upward as oil pressure increases. Either motion causes the second oil pilot to move and also restores the first pilot by moving the cylinder of the first pilot. The second pilot controls the movement of the main power piston which in moving will alter the steam valves and restore the second pilot. The main power piston moves a rack and pinion in turn. The pinion rotates a valve camshaft, and each cam in turn opens or closes a steam valve.



(Courtesy of General Electric Co.)

FIG. 7-27. Double relay mechanical-hydraulic speed governor.

A completely hydraulic governor is shown in Fig. 7-28. The governing system consists of two main parts: (1) the governor oil pump, pressure transformer, and speed changer, (2) the servomotor relay and power piston operating the governor valves. The governor operation for a load decrease, which means an increase of turbine speed, follows. The governing oil-pump discharge pressure will increase and this will compress the bellows and raise the oil cup (3). This chokes off oil flow from the main oil pump through the line above the cup, and the oil pressure in the line increases. This increased secondary oil pressure acts on (3) and stops its motion and also reacts through to small piston (4), causing it to move downward. Its motion is stopped by the tension in (9). The cup valve attached to (4) moves downward, and the oil pressure above relay (5) increases. This increased pressure moves (5) downward. The pilot then allows oil to flow under the power piston and releases oil from above it. Suitable linkages close the steam valves. The movement upward of the power piston through (8) restores (4), and (5) is therefore restored to neutral because of the spring beneath it. The above operation is reversed on load increase. The speed regulation depends on the constant of the spring (9) and the ratio of the lengths of the arms of lever (8). Moving the fulcrum point adjusts the regulation. Speed adjustment is obtained by operation of the handwheel (7). The entire governor and servomotor are mounted in a housing as shown on the thrust-bearing pedestal which contains also the main and oil governor pumps.



(*Courtesy of Westinghouse Electric Corp.*)

FIG. 7-28. Pressure transformer relay hydraulic governor.

Speed Changers. The turbine governor can be designed to maintain constant speed at all steady turbine loads within the capacity of the unit, or it can be made to give slightly decreasing speeds as the load increases. The former type is referred to as an "isochronous" governor, or one having zero speed droop. In this type, however, a momentary departure of the speed from normal will occur each time a load change occurs, but the speed very quickly returns to normal. The latter type is a nonisochronous governor or one with speed droop. Either type has its applications as will be later discussed. Governors with speed droop are commonly designed with a speed changer which can be manually or mechanically adjusted after a load change to return the unit to normal speed. Isochronous governors also have their operating point adjustable to set for different speeds. Automatic adjustment of governors by some form of remote control necessitates the installation of additional electrical control equipment.

Figure 7-29 shows a speed changer using a secondary spring, manually adjusted, which acts to change the governor spring force. This restores the original speed after a load change. With this type, the governor

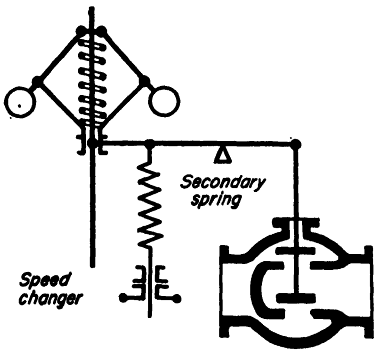


FIG. 7-29. Speed changer using secondary spring.

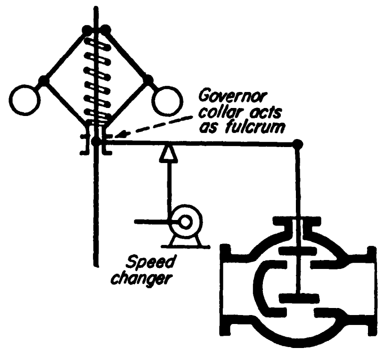


FIG. 7-30. Speed changer using collar as fulcrum.

weights are moved as the speed is adjusted and as it affects the governor performance. The effect is that, as the speed changer is adjusted, the governor speed droop (speed change with load change) is altered. Figure 7-30 shows a design in which the governor sliding collar acts as a fulcrum when the link is moved by the speed changer. This scheme, although somewhat more intricate, gives a better governor characteristic in that the speed-travel relationship is the same at all loads since the position of the governor does not change.

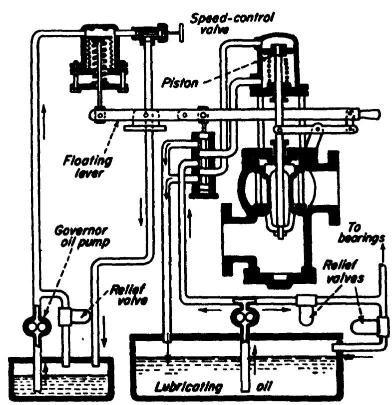


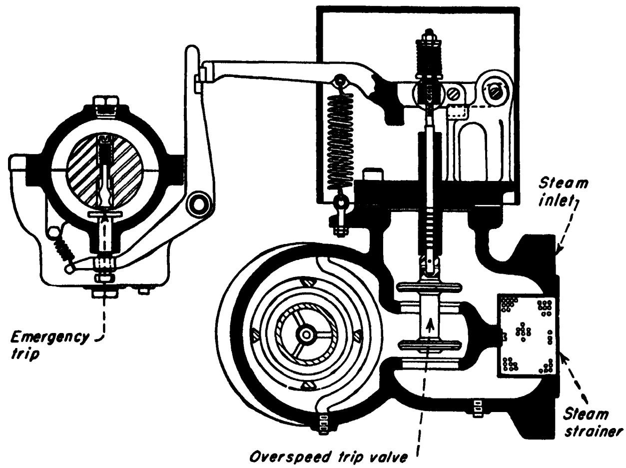
FIG. 7-31. Speed changer in fluid-pressure governor.

Speed variation is obtained with the all-fluid governor by the principle shown in Fig. 7-31. The discharge pressure of the governor fluid pump is controlled by adjustment of the speed-control valve. To raise the speed, the valve is opened, thereby reducing the pressure over the controlling bellows piston. Similarily closing the valve lowers the speed.

Governor Protective Auxiliaries. In addition to operating speed governors like those described, turbines are fitted with various forms of protective or emergency governors. They are commonly applied for the following conditions:

Overspeed
Low oil pressure
Rotor end travel
Low vacuum
Reverse current

The principle and use for a smaller turbine of the overspeed trip are shown in Fig. 7-32. In most designs a pin or weight in the main shaft,

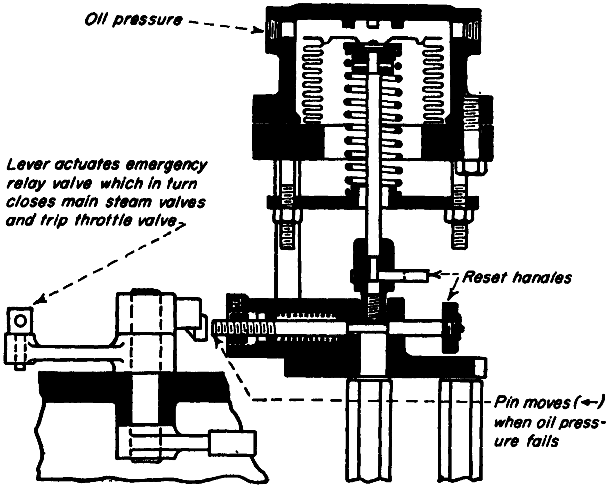


(*Courtesy of Elliott Co.*)

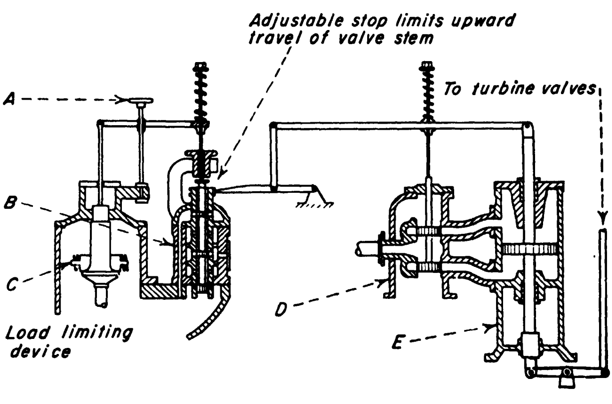
FIG. 7-32. Cross section of overspeed governor and valve.

or in a unit attached to it, is acted on by centrifugal force. When the shaft speed exceeds a desired safe level, say 10 per cent overspeed, the pin or weight is thrown against a latching device. This releases a spring which closes either the governing valve or a special emergency stop valve. In some designs the overspeed trip is incorporated with the operating governor. Figure 7-33 shows an emergency valve for protection when lubricating oil pressure fails. Oil pressure normally acts against the spring-resisted bellows which upon loss of pressure acts to close the turbine valves. Various designs of protective governors for all the conditions mentioned above are used by turbine builders.

Load-limiting Devices. This feature of a governor as indicated in Fig. 7-34 is applied principally to turbines used in central stations. By some form of automatic limit of the steam-admission valves, the steam flow and consequent load can be limited to some arbitrary amount. A pos-



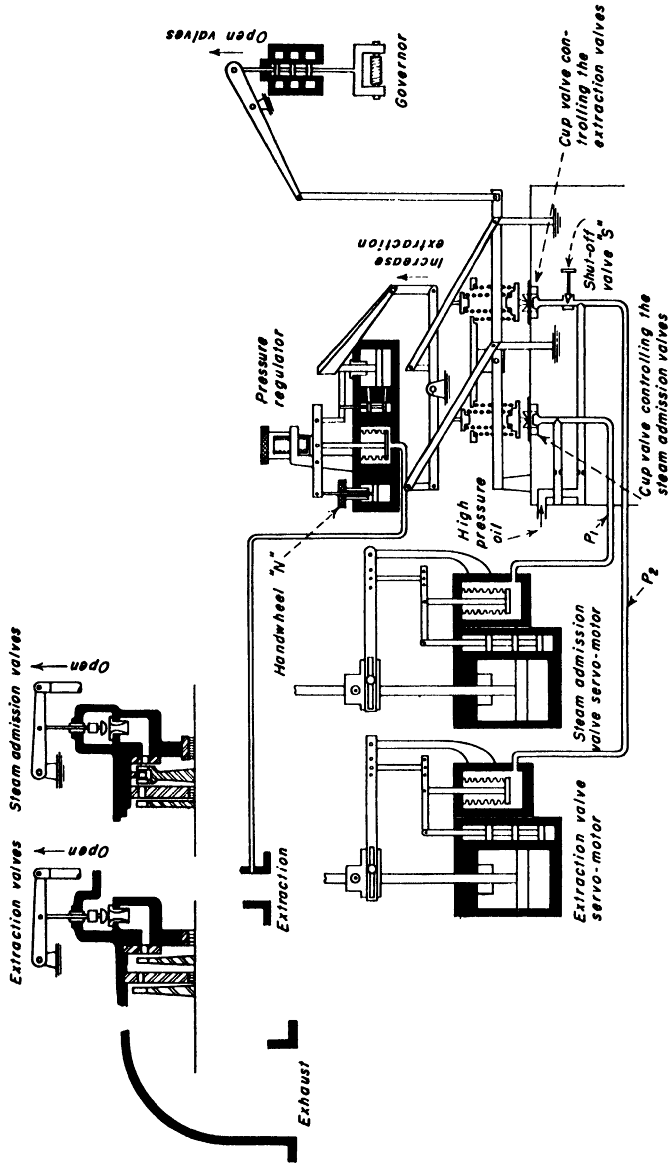
(Courtesy of Worthington Pump and Machinery Corp.)
FIG. 7-33. Low oil pressure trip mechanism.



(Courtesy of Allis-Chalmers Manufacturing Co.)
FIG. 7-34. Governor gear showing load-limiting device.

sible application would be where two boilers normally furnish steam to one turbine, and only one boiler operates. The load limit could be set to use only the steam flow which one boiler can provide.

Extraction and Induction Governing Systems. Turbines used in industrial applications which supply extracted steam for process work must



(Courtesy of Elliott Co.)

FIG. 7-35. Outline of extraction or induction control elements.

have a governor system designed for both speed control and control of the extraction steam pressure. Similar controls are required if the turbine is of the induction type. Such systems must provide for three possible conditions: a change in load only, a change in extraction (or induction) steam only, and a possible change in both load and steam flow.

The elements of such a control system for one stage of extraction (or induction) are shown in Fig. 7-35. Here, with the load remaining constant, a drop in extraction pressure will make the pressure regulator tip

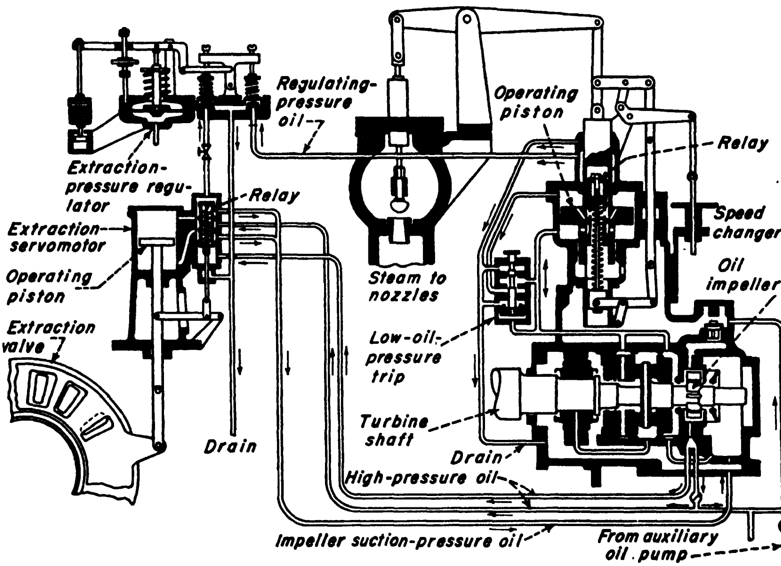
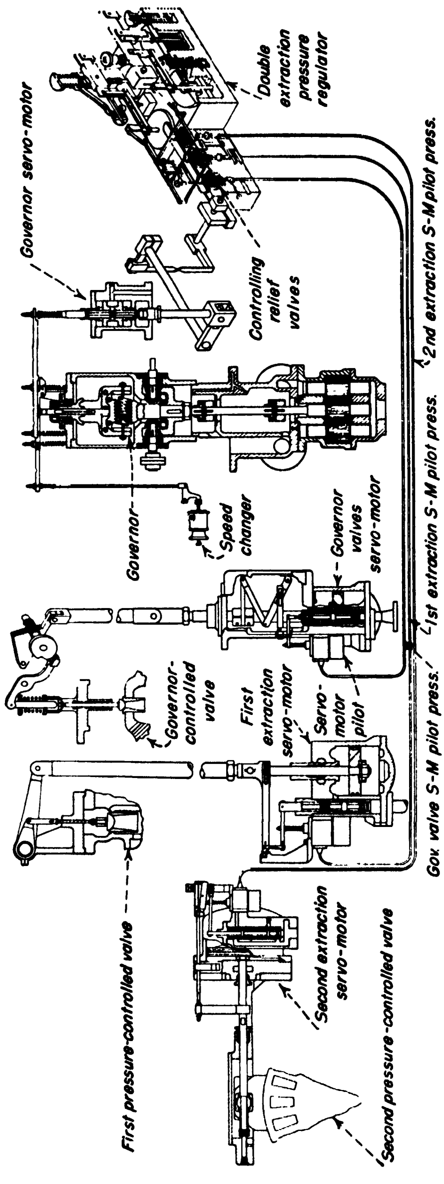


FIG. 7-36. Single-extraction hydraulic system.

the cup-valve loading lever, causing a decrease in spring load on the extraction cup valve which reduces pressure P_2, and causing an increase in spring load on the cup valve controlling the steam-admission valves which increases pressure P_1. The drop in pressure P_2 causes the extraction valves to travel in the closing direction enough to restore the desired extraction pressure at the same time that the increase in pressure P_1 opens the steam-admission valves by the amount necessary to maintain the load and extraction flow. Both the steam admission and extraction valves respond to the action of the governor when the load changes, or the load is lost and the unit tends to overspeed. The governor acts upon both cup valves and, for an increase in load, will increase both pressures P_1 and P_2 so that both sets of valves open sufficiently to carry the increased steam flow. For straight condensing operation, the pressure regulator is



(Courtesy of Elliott Co.)

FIG. 7-37. Double-extraction governing system.

made inoperative by screwing down on the handwheel N. The extraction valve is opened wide by closing the shutoff valve S to complete the changeover.

Figure 7-36 illustrates another system of extraction control. Here the impeller speed governor meets load changes by adjusting the pressure on the governor and servomotor relays and hence by moving both governor and extraction valves to change the through flow of steam. A change in steam demand acts on the pressure governor diaphragm, and this changes the pressure above the extraction and governor relays. The valves move in opposite ways to change the relation between extraction and through flow.

Governing systems controlling both load and steam pressure at two extraction points are parts of many industrial turbines. Figure 7-37 shows the line diagram of such a system. This system combines hydraulic and mechanical connections for a control arrangement. It simultaneously controls the speed and two extraction pressures and is designed so that variations in one of these variables is corrected by valve movements without affecting the other two.

The system has the following primary elements as shown in Fig. 7-37:

1. Speed governor
2. Extraction pressure regulator interconnected with the speed governor
3. Three servomotors operating governor and pressure valves
4. Servomotor pilot valves

The valve servomotors are operated by oil from the main turbine oil pump and oil for the control system comes from the same pump, but it is filtered through a fine-mesh filter. Control is due principally to the action of the double-extraction pressure regulator. As shown, it has two regulators which react to extraction steam pressure changes and three spring-loaded relief valves. These relief valves act as variable orifices. They control oil pressure to sensitive bellows. The bellows operate a small piston, which in turn moves the oil pilot valve. Then the respective servomotor moves and so repositions the steam governor or pressure-controlled extraction valves. The system as indicated is built so that the relief valves, pressure regulators, and the turbine speed governor are all interconnected. The relief valves controlling oil pressure to the pilots are designed with two sets of springs: one loaded by the governor and one by the pressure regulators. In this way the oil pressure to the pilot valve bellows depends on both the speed governor and the extraction pressure regulator.

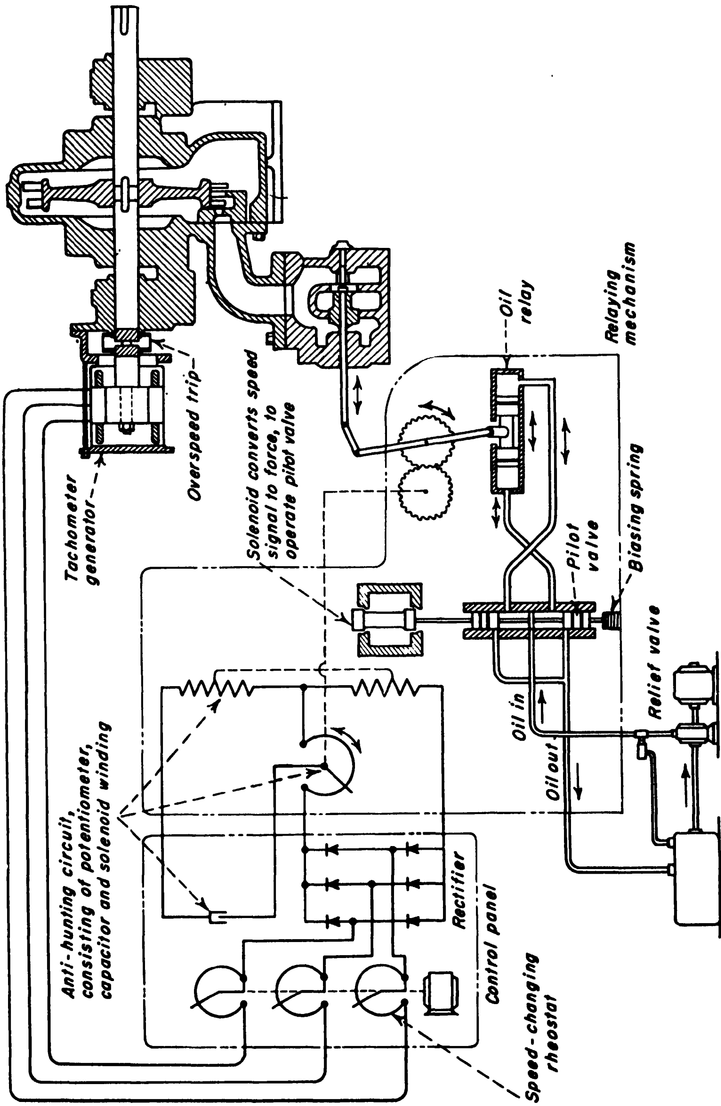
In the case of a load change with no change in the extraction steam flow, the operation is as follows: The centrifugal weights of the speed governor react through a governor servomotor as shown. This unit transmits the same motion with increased force to all the relief valves through the set of springs loaded by the governor. These relief valves then, through bellows, pilot piston, pilot, and servomotor, cause both the main inlet and the extraction valves to change their position a given amount. Hence, load change is provided for with no change in extraction pressure.

In the case of a change in the extraction steam flow from the first stage but no load or second-stage extraction change the operation is as follows: Change in extraction flow causes a change in the stage pressure. The extraction pressure regulator now responds and changes the spring loadings on the three relief valves in such a way as to cause the governor-controlled valves to change slightly and the two pressure-controlled valves to change a slight amount. The result is that steam flow to the turbine is changed and restores the first extraction pressure to normal. This affects the steam flow from the first to the lower pressure extraction stage, and hence its valve must also be repositioned as indicated above to keep that extraction-stage pressure constant. The load remains constant since the power lost by decreased steam flow to the condenser equals the increased power developed by the added steam flow through the governor-controlled valves.

The action of the system for a change in second-stage extraction is very similar to the above. In this case both the main governor-controlled steam valves and the first extraction pressure-controlled valves open and the second extraction pressure-controlled valves close. In operation, the governor is continually reacting to the variations in load and extraction pressure changes and regulates to fulfill these changing demands. Extraction pressures are maintained constant up to the point where the extracted steam is greater than that needed to carry the existing load. The speed governor controls in this case, and extraction pressure will drop until the load again increases or an outside reducing valve is cut in to provide make-up steam.

Electrical Governors. Although the electrical turbine governor has been developed and is applied to units of capacities up to several thousand horsepower, the mechanical and hydraulic form is most extensively used. Its future seems to be a matter of further development and application. At present it is applied primarily to turbines of about 750 to 1,000 hp driving paper and textile machinery. It is a governor which can operate over speed ranges as large as 30 to 1, and at any speed the changer setting is practically isochronous.

The schematic diagram of a typical governor is shown in Fig. 7-38. The primary governor is an a-c tachometer generator directly connected to the turbine shaft. Speed change is accompanied by a change in the



(Courtesy of General Electric Co.)

FIG. 7-38. Schematic diagram of governing system.

output voltage. As shown, the current flows through a rheostat which acts as a speed changer and then through a rectifier to a solenoid coil. When operating at normal speed, the solenoid coil is balanced by a spring. An oil pilot valve is actuated by the solenoid and leaves its mid-position on speed change. The pilot valve upon displacement from mid-position controls the operation of an oil relay which in turn positions the turbine-inlet steam valve. Return to normal speed resets the pilot valve. A second coil in series with a condenser and potentiometer is part of the solenoid coil. The potentiometer is arranged as shown to be mechanically moved by the oil relay. This movement causes a voltage change, and a temporary current flow is produced in the second coil. This acts as a stabilizing circuit by producing an antihunting field opposing the field of the main coil of the solenoid. Speed is varied by operating panel-mounted push buttons either at the governor or from remote stations.

Centrifugal Weight Governor Characteristics. The performance of a centrifugal weight governor with auxiliary spring speed changer is indicated in Fig. 7-39 and applies to a governor as in Fig. 7-29. The curves of Fig. 7-39 show the essential characteristics of this governor. In order to understand them, consider the forces acting on the collar. Centrifugal force depends on the diameter of the circle of rotation of the weight and on the speed. Greater diameter or higher speed means more force. The spring force depends on dimensions of the spring and how much it is compressed. Assume a constant speed, say of 98 per cent of rated speed, and a spring of known dimensions. With weights at "in" position, a certain centrifugal force results, and this will just balance the spring force when the sliding collar is all the way down and the valve is wide open. Still holding speed at 98 per cent, move the collar all the way up and close the valve. The spring, being compressed fully, now exerts greater force, but the weights also exert greater force, because they swing in a wider circle. Again the forces balance. Similar balances will result at points between these extremes, and it is possible to plot centrifugal force against governor travel, obtaining a sloping line like that labeled 98 per cent in Fig. 7-39. The shape of this curve depends on governor design and may deviate appreciably from a straight line.

At any speed above 98 per cent the weights exert greater centrifugal force at each position, and a stronger spring is needed to balance these forces. Figure 7-39 shows force-travel curves for speeds from 98 to 102 per cent. Obviously it is impractical to use a different spring for each speed as assumed for illustration purposes here. An actual governor would have a spring of such dimensions that its force would just balance centrifugal force at 98 per cent speed and the weights-in position and at 102 per cent speed and the weights-out position. Lines X and Y of Fig.

7-39 show limits of governor travel, and line *A* is the force line of the actual spring.

With this spring in place, the governor opens the valve far enough to carry full load at 98 per cent speed and closes it enough to hold speed to 102 per cent at no load. This speed change of 4 per cent is the governor's regulation or speed droop. To change this characteristic requires changing the spring. For example, to improve regulation to 3 per cent, the new spring must match centrifugal force at 98 per cent as before, but need match force at only 101 per cent less than before. Since travel and hence

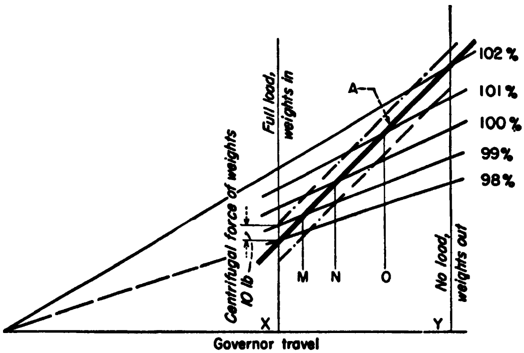


FIG. 7-39. Curves showing the relation of centrifugal force and travel of weights at constant speeds and using an auxiliary spring (Fig. 7-29) for speed adjustment.

spring compression stay the same, this means a lighter spring. Hence, the use of the auxiliary spring speed changer, as in Fig. 7-29, produces a different regulation for each of its settings, as illustrated in Fig. 7-39. The effect of the speed changer in Fig. 7-39 is to shift curve *A* parallel to itself, *e.g.*, to the dot-and-dash lines indicated.

Thus far, friction has been neglected. To see its effect, assume that difference in centrifugal force at weights-in position between 98 and 99 per cent speed is 10 lb. Now if, because of friction, it takes 10 lb over and above spring force to move the collar, speed must increase from 98 to 99 per cent after a load change before the valve starts to close. As load drops farther, governor positions at various speeds are shown along the dotted line above *A*. Note that at the weights-out position, only about two-thirds of 1 per cent speed change produces the needed 10-lb force. Thus, in speaking of the power of a governor, defined as the change in governor force for 1 per cent speed change without motion, it is necessary to refer always to position.

If the process with friction described above is reversed and load is applied to the unit, governor positions for various speeds are shown by the dotted line below A in Fig. 7-39. The vertical distance between lines above and below A represents the dead band of the governor, the speed change in which no governor motion occurs. This is a measure of the governor's sensitivity, which may be defined as the percentage of speed change required to produce a corrective movement.

The governor shown in Fig. 7-29 is one wherein the weights have to be moved to change the governing valves to a different steam-flow position while the turbine speed is constant. This is called a "travel governor." Both speed regulations are affected by the speed-changer position, and the speed-governor travel curves are distorted and not straight lines as

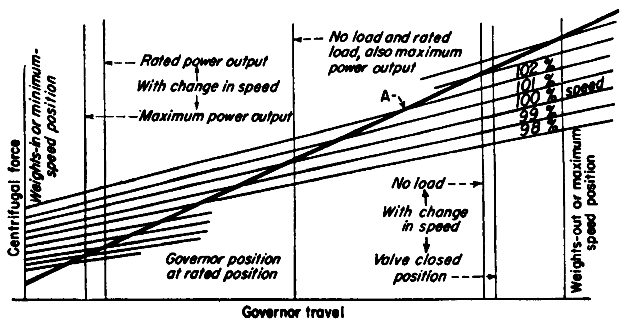


FIG. 7-40. Centrifugal force and travel curves for governor using the speed changer of Fig. 7-30.

in Fig. 7-39. The effect of a speed changer (Fig. 7-30) is shown on the centrifugal force-travel curves in Fig. 7-40. This changer keeps the same governor position with the same speed but changes the relation of the governor position to that of the governing valves and is called a "non-travel unit." Figure 7-40 shows the governing characteristics of this system. It has the advantage of having the same speed-travel relation at any load, as the position of the governor is constant for any given speed. The governor acts to open the valves fully from no-load position due to speed change without changing the speed-changer setting. It must also be able to drop full load and close the valves from full-load rated speed position. This takes a considerable amount of governor travel as indicated. The speed changes above and below rated speed also necessitate governor travel. This governor demands accurate travel setting to obtain the required speed range.

Turbine Speed Regulation. The over-all effect of a speed-responsive governor, either direct acting or through relay mechanisms, is commonly indicated by the speed regulation curve, as shown in Fig. 7-41. Two types of operation are possible as indicated by curves A and B. A turbine governor designed and set to give a characteristic as at A is called an "isochronous" governor; *i.e.*, the governor will maintain constant speed for all values of steady load within the capacity of the turbine. A load change with a governor of this type will produce a momentary change in speed, but the speed quickly returns to normal. A single turbine operating alone and not expected to be paralleled with another may operate with a governor whose characteristic is fairly isochronous, consistent with governor hunting. Hunting occurs with an oversensitive governor. A

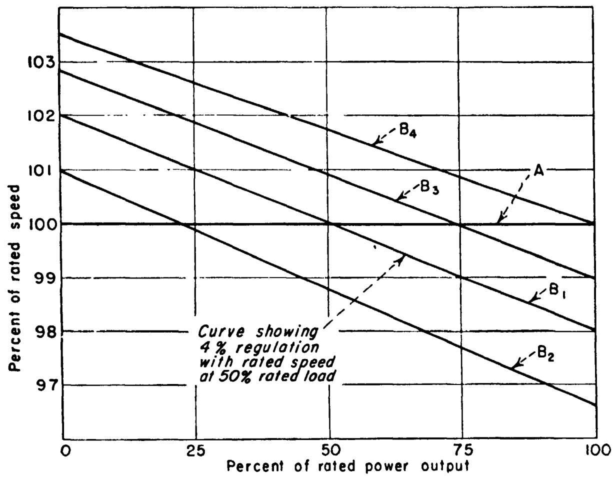


FIG. 7-41. Typical turbine regulation curves.

speed changer will, when used, produce constant-speed operation at other values of the rated speed or will have the effect of shifting curve A up or down but still remaining horizontal.

When two or more turbines are either mechanically connected to the same shaft or are driving synchronous a-c generators feeding to a common line, the governor must have some speed droop. Such governor action is shown in the B curves. Consider B_1 for example. The turbine would decrease in speed as load is added to the unit. The curve B_1 results from one setting of the speed changer. If the speed changer is reset, the effect on the regulation curves is to produce such curves as B_2, B_3, or B_4 for each new speed-changer setting. The B curves are not exactly parallel to one another and are not necessarily straight lines, as shown, but may be somewhat irregular and curved. Curve B_1 happens to go through rated speed at 50 per cent load although operation at any other condition of rated speed and load is possible.

Referring to curve B_1, the over-all speed regulation for straight-condensing or noncondensing or nonautomatic-extraction turbines is given as

$$R_s = \frac{N_0 - N}{N_r} \times 100$$
$$= \frac{102 - 98}{100} \times 100 = 4 \text{ per cent}$$

Speed regulation for automatic-extraction turbines is calculated as given in the section on governor terms and definitions, page 235. Speed regulation

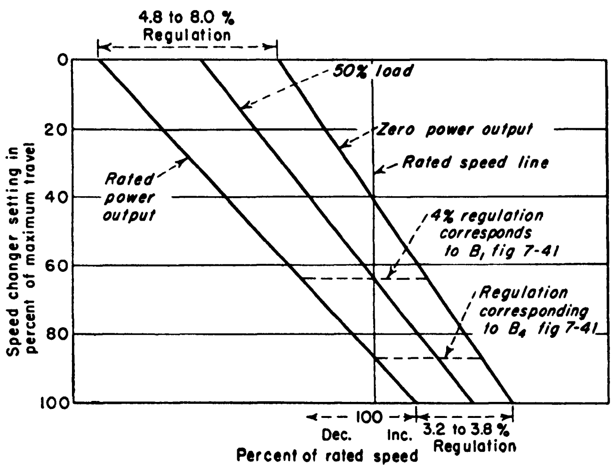


FIG. 7-42. Typical curves for steady-state speed regulation and speed-changer setting.

as shown by the B curves other than B_1 are, however, not identical to this one. Governors can have practically uniform values of the steady-state speed regulation for different speed-changer settings or B curves going, say, from about 3.8 to 4.8 per cent, whereas a wide variation as from about 3.2 to 8.0 per cent may occur for turbines. This effect is indicated qualitatively in Fig. 7-42, which shows a range of speed regulations corresponding to each speed-changer setting. Turbines are in general specified to have a steady-state speed regulation of not less than about 2 and not more than 4 per cent for electrical power generation, when the speed changer is set to give rated speed with rated power output. Complete equipment and performance specifications for governors of turbines 500 kw and larger and driving electric generators are given in the Recommended Specifications for Speed Governing of Steam Turbines prepared by the AIEE-ASME Joint Committee.

Load Division. Speed droop on one or several interconnected turbines in parallel produces different speeds as the load changes. Figure 7-43 shows the speed regulation curve at a given speed-changer setting of two turbines rated at 7,500 and 10,000 kw, with each having a rated speed of 3,600 rpm. If one or the other were operated alone, speed would change as shown with load, provided the speed changer was not readjusted. Hence, to keep rated or any other speed, the speed changer must be manually or automatically adjusted. As shown, however, with both turbines interconnected, rated speed would occur only at a total load of

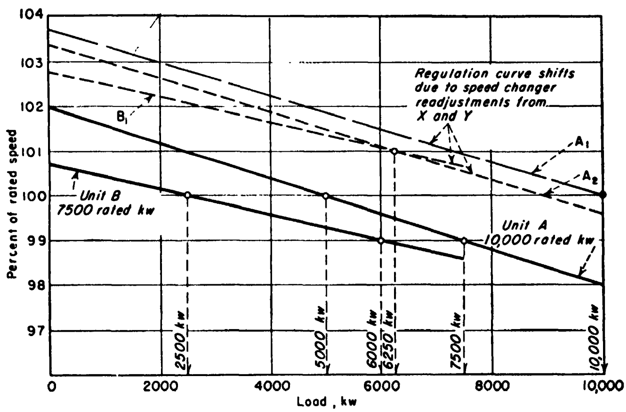


FIG. 7-43. Regulation curves as applied to load division.

2,500 + 5,000 = 7,500 kw. Should the load be increased to, say, 13,500 kw, unit *A* would take 7,500 kw and unit *B* 6,000 kw with a drop in speed to $0.99 \times 3{,}600 = 3{,}564$ rpm. Conversely, a dropping load increases speed.

The speed changer of unit *A* or *B*, or of both, can be reset. In this manner speed can be returned to normal and a load divided in any desired manner between the two units. For example, if unit *A* is readjusted to A_1 with no change in *B*, the rated speed will result and unit *A* will deliver 10,000 kw and unit *B* will deliver 2,500 kw, which together equal a load of 12,500 kw.

This load of 12,500 kw could also be divided equally between the two turbines by adjusting *A* to A_2 and *B* to B_1. Equal division is not necessarily the best method of dividing a load. It is seen that, by a relative adjustment of the speed changer, any load can be divided in some arbitrary manner between the turbines within their capacity range.

Load division between two identical units *A* and *B* with similar 4 per cent droop governors is shown in Fig. 7-44. As load is added or removed without change of the speed adjuster, each unit follows the same speed load line, and the load will be equally divided at all times. Thus at 50 per cent of the full load on each, the speed as shown is rated speed; at 75 per cent of the load, the speed is 99 per cent of rated speed.

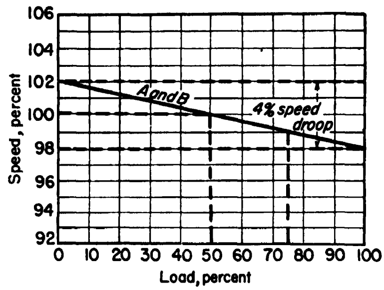


FIG. 7-44. Speed-regulation curves of two identical units.

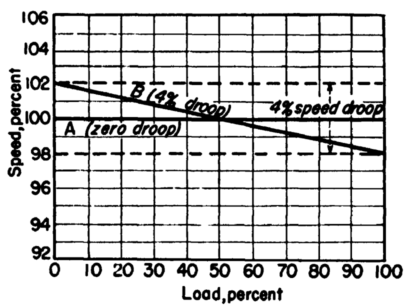


FIG. 7-45. Speed-regulation curves with and without speed droop.

A combination of units, one with speed droop and the other without speed droop (isochronous) operating in parallel, is shown in Fig. 7-45. The speed changer happens to be set so that each unit is carrying 50 per cent of the load. Any change in total load will be taken up or dropped by unit A within its entire load range while unit B will remain at its initial setting. For rapid load changes and equally sensitive governors, unit B may tend to react, but they finally settle down with B at its original value and A will have absorbed the change within its limits.

CHAPTER 8

STEAM-TURBINE POWER CYCLE AUXILIARIES

Steam turbines find application in power plants of two general forms: the central-station or utility plant and the industrial or equivalent type of plant. The central station is the plant which is designed and operated to generate electrical power, and to this end highly efficient equipment and complex plant layouts are generally economically justified. The industrial plant is usually tailored to the particular plant it serves and is used to generate electrical or mechanical power and also to supply steam for some heating processes used in production. Much of the equipment in these plants is piped to receive steam from the turbines. The following pages will show briefly the nature of this equipment so that the turbine will be better understood as an integral part of a power plant.

Central-station Layout. The type of equipment and the manner in which it is connected to form a station layout differ among various central stations. While all layouts are based on a few fundamental theoretical cycles, the practical cycles differ widely and any one layout is peculiar to a certain plant. There are nevertheless certain fundamental similarities between utility stations, and Fig. 8-1 is shown in that sense. Turbines of various types as indicated at the top of the diagram may be used in conjunction with similar layouts. Nonautomatic-extraction or bleeder turbines are employed as they produce the highly efficient cycles. The related equipment consists principally of a condenser and a series of feed-water heaters in series which successively heat the water to higher temperatures by using bled steam from the turbines. As shown, there are four principal heaters. A prime difference between this layout and that of other central stations is in the number of feed-water stages used. Some have five or more but most stations have four or fewer feed-water heating stages.

Various pumps are indicated and, in this layout, are driven by electric motors. In some applications steam turbines or a combination of turbine and motor drives may be employed. This does not materially affect the basic layout except that steam lines to the turbines and from the turbine exhausts leading to the heaters are added. The layout as in Fig. 8-1 is not a piping diagram but shows merely the main flow paths through the major equipment. Valves, by-pass lines, tanks, and several minor but essential other equipment are not shown.

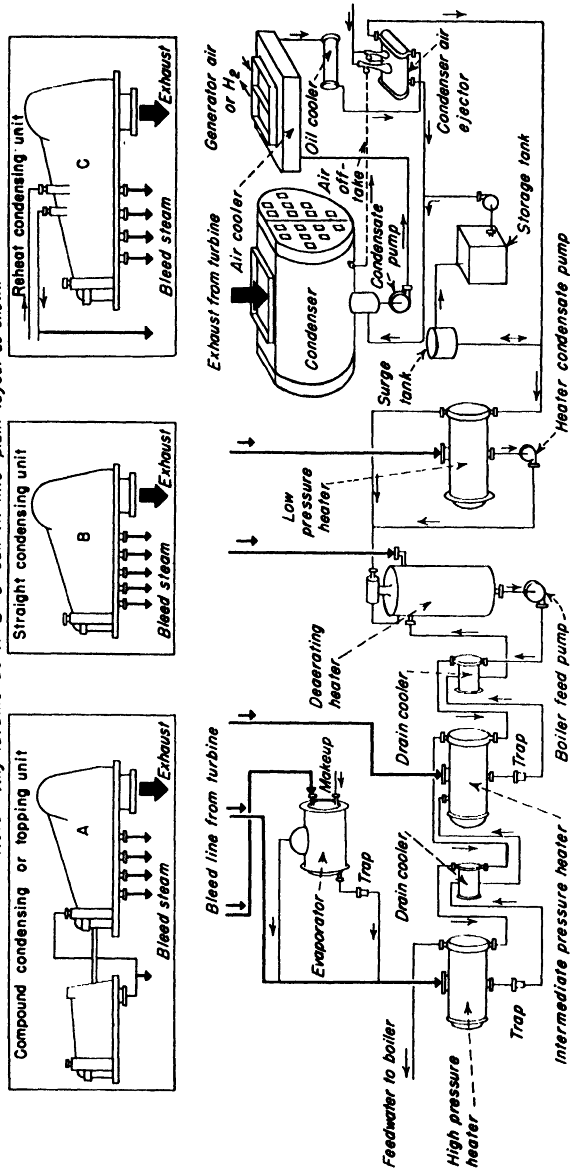


FIG. 8-1. Modern central-station layout to show typical equipment related to turbines.

The principal steam flow in such a plant is from the boiler to the turbine steam chest. Minor flows occur to the condenser air ejectors and to turbine auxiliary drives if used. The steam passes through the turbine to the exhaust hood and then exhausts to the condenser. At various points (five for the particular cycle shown) steam is extracted and used to heat feed water in four heaters and in an evaporator to provide make-up water for water losses that occur in all cycles. Steam losses as from glands and seals are led to the deaerating heater and returned to the cycle.

Principal water or condensate flow in such a plant starts from the condenser hot well from which it is removed by the condensate pump. The water at a low temperature (about 80 F) can be used in an air or hydrogen cooler.

This is a unit which may be used with large electrical generators. In it air or hydrogen is circulated and cooled by giving up heat to the condensate. The air or hydrogen is in a closed system and circulates through the generator to cool this unit and thereby itself heats up—which requires the air or hydrogen cooler. The cooler eliminates the dust that is deposited in generator windings if an open system without an air cooler is used. Hydrogen has advantages over air, the principal one being the reduction of generator windage losses since hydrogen is much less dense than air.

The condensate then flows through the turbine oil cooler and the condensers of the air ejector. The air ejector uses steam jets to withdraw air out of the main condenser and so keep turbine exhaust pressure low. The steam used in these jets is condensed by the main condensate and gives up its heat and so improves cycle efficiency. The condensate flow from the condensed-steam jets is returned to the main condenser hot well to join the main condensate flow.

The main condensate then flows to the closed-type l-p heaters. This type of heater is common at this point since bleed steam pressure is generally less than atmospheric, and the main condensate is above atmospheric pressure. As shown, a pump removes heater condensate although a trap may be used discharging to the main condenser.

The deaerating heater is commonly next in line and receives the main condensate. Aside from bled steam from the main turbine, this heater may receive trapped bleed condensate from the higher pressure heaters and turbine leakoff steam and steam generated in the evaporator which in the figure shown happens to be part of the h-p heater drips.

The boiler feed-water pump taking its suction from the deaerating heater then discharges through the remaining heaters and drain coolers. Closed heaters are again used since the feed-water pressure is higher than the steam pressure at the turbine bleed points. Condensate found in these heaters is trapped to lower pressure regions as shown through small closed heaters known as "drain coolers."

The surge and storage tanks are common to power cycles and are useful when the station load changes. The amount of water in the boiler depends on the rate at which steam is being formed; therefore, water must be taken out or put into the cycle with load change. The surge tank is set above the main condensate pump. When the load drops, more water must be put into the cycle, and it comes from the surge tank. Too much withdrawal to a low level will cause the storage-tank pump to operate and add water to the main condensate hot well. For load rise, water backs up in the surge tank and spills over to the storage tank. The evaporator is usually operated only when the water in the storage tank gets too low.

The diagram shows how the equipment described above is integrated into a cycle. The steam boiler is not shown* but is, of course, of major importance. The emphasis here is rather on that equipment which receives exhaust or bled steam from the turbines and whose arrangement and number are the principal bases for forming various hookups of central or utility stations.

Industrial Stations. The operation of industrial plants of any type requires energy supplies which are in the form of either mechanical or electrical power or steam for heating. The power is used to operate mechanical equipment such as pumps, fans, lathes, presses needed in production. On the other hand, manufacturing processes require heat supplies in many instances, *e.g.*, in drying, space heating, evaporating, and processing methods. Steam is a very desirable heating medium since, if it is wet or saturated, it will give up heat at constant temperature, and fire hazards from direct heating are eliminated. The temperature depends on the pressure; therefore, if the pressure is held constant, temperature will not change. If an industrial station is near a central plant from which it may get power and steam, it might be economical to buy requirements from it. However any fair- or large-sized factory using both process steam and power is often served most economically by its own plant.

The extraction steam turbine or at times other types offer an ideal method to obtain both power and steam at controlled pressures. The most desirable situation is one where both power and steam demands are in phase with each other. The layouts of industrial plants differ widely because of the varying demands and sizes of different industries. In fact

* See Chaps. 2 and 3 of "Applied Energy Conversion," by Skrotzki and Vopat, McGraw-Hill Book Company, Inc., New York, 1945.

no one layout can be used as in the case of central stations to give an idea of the usual hookup. Figure 8-2 is used as a composite diagram to show various industrial turbines and some of the common unit processes requiring steam for heating. One or more extraction, mixed- or back-pressure turbines may be used as the prime movers. They are coupled to generators for electric power, but in smaller sizes they may be used for direct mechanical drive of equipment, *e.g.*, in driving fans or blowers. These turbines are supplied with h-p steam. Many applications have process steam supplies at two lower pressures, as indicated by Fig. 8-2. More or less may, of course, be used. In addition to the turbines, other plant equipment may need h-p steam—such as a forging hammer, or steam-engine driven equipment, as indicated in Fig. 8-2. These may also exhaust to the process lines.

Extraction steam is shown feeding the process lines—although, for some applications if process steam from other exhausts is in excess, a mixed-pressure turbine might be most effective in the plant layout. The diagram emphasizes the use of steam for the industrial processes and does not show the entire plant. Boilers are of course required to generate steam and, even as in central stations, feed-water heaters of different designs and number are also supplied with extracted or exhaust steam. There is shown a condenser which is used with industrial turbines when they are operated to drive generators and supply electric power not synchronized with steam demand. When all the process steam needs equal the steam needed for power purposes, the plant is most efficient. In this case the turbine section between the l-p extraction stage and the condenser is not used. However if a small amount of "cooling" steam were not allowed to flow through to the condenser during this condition, the steam trapped in this turbine section would be churned up by the turbine blades and the friction would heat them excessively.

The term "industrial plant" usually refers to those power plants used in manufacturing. Turbines, however, are also used in power plants associated with hospitals, schools, hotels, prisons, and a variety of other institutions. Plant layouts are tailored as in any industrial plants since, in addition to electrical power, steam is needed for space heating and many other operations.

Condensers. Turbines become more efficient when their exhaust pressure decreases. This condition is brought about by having turbines exhaust into condensers which in most cases are fastened directly to the turbine exhaust hood. The low pressure, or "vacuum" as it is commonly called, is produced by the action of large quantities of circulating water which pass through the condenser and condense the entering steam.

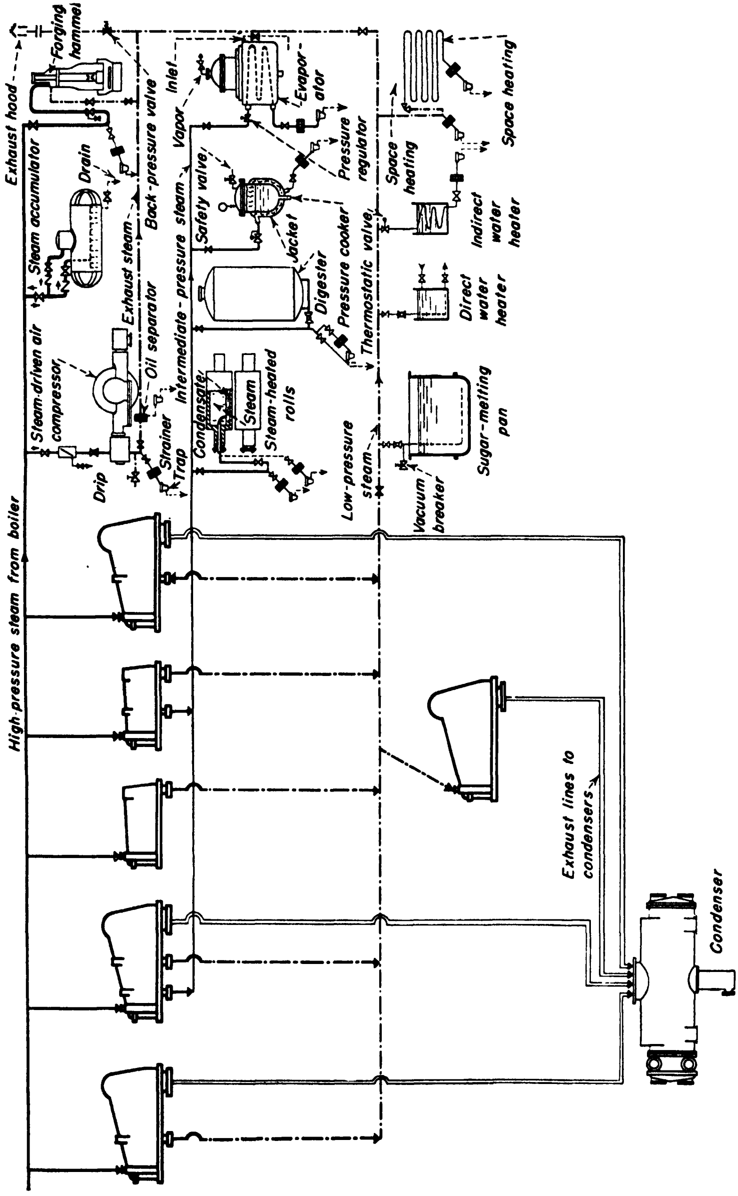


Fig. 8-2. Composite diagram illustrating industrial turbine applications.

Condensers are of two basic types:

Direct-contact or jet
Surface

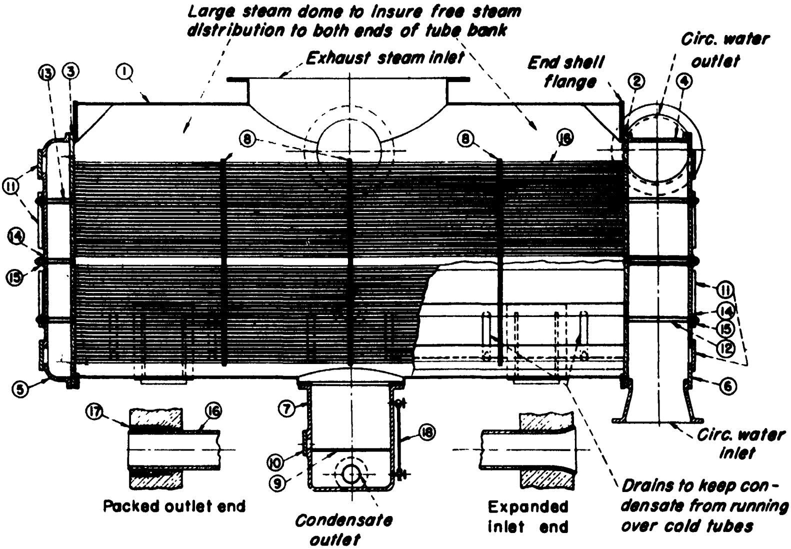
In the jet condenser the circulating water and steam mix physically. The effect is to condense the steam while the circulating water rises slightly in temperature. Inasmuch as steam is steadily exhausted to the condenser and water steadily supplied, the condition produced within the condenser is that of a low constant pressure. For such operation there must be also a constant removal of condensed steam together with the water in which it is mixed. This type of condenser can use fairly dirty condensing water, is cheaper, and in general requires less maintenance and cooling water than the surface condenser. In the surface condenser, the turbine exhaust steam is also condensed by water cooling the steam, although the two do not mix and are removed separately. The surface condenser has advantages in that the condensate can be used for feed water, and a low condenser pressure is more economically obtained.

Surface Condensers. The surface condenser is used in power plants to produce the low back pressures or high vacuums for turbine exhaust. Compared to the direct-contact type of condenser it has the advantage of being able to recover the condensate for reuse in the cycle and in general can produce the low pressures more economically. The former feature holds to a minimum the make-up water needed and its consequent treatment before being introduced into the cycle. Surface condensers are built in a variety of forms, but the medium- and large-sized condensers can be fairly well illustrated in a diagrammatic way as in Fig. 8-3, which shows a vertical cross section. The external appearance of this condenser is shown in Fig. 8-4. The principal elements of a surface condenser are

Tubes
Tube sheets
Tube supports
Water boxes
Shell
Hot well
Steam, air, water connections

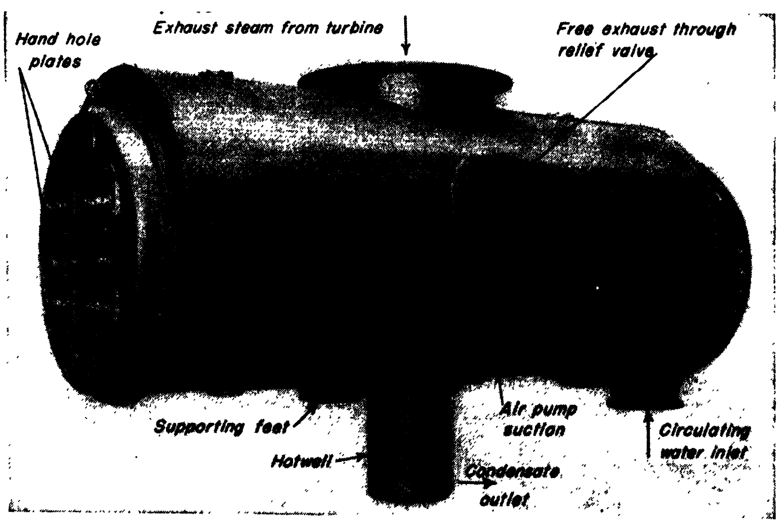
Surface condensers are classified also on the following bases:

Horizontal or vertical
Single- or double-pass
Shell cross section (cylindrical, oval, heart)
Air-cooling (external or internal)



(Courtesy of Worthington Pump and Machinery Corp.)

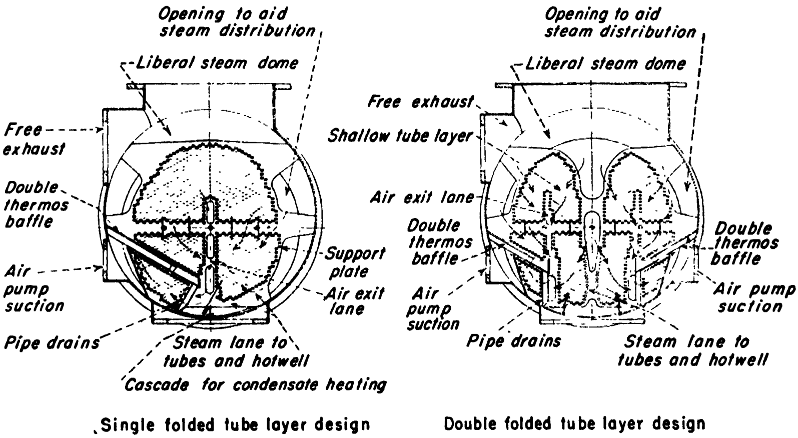
FIG. 8-3. Typical surface condenser elements and terminology.



(Courtesy of Worthington Pump and Machinery Corp.)

FIG. 8-4. External appearance of welded steel shell surface condenser.

The condenser tubes comprise the cooling surface and consist of straight lengths of seamless drawn tubing. Usual sizes are ¾-, ⅞-, or 1-in. tubes, some 10 to 30 ft in length. Tube material is usually some copper alloy such as Admiralty and Muntz metal, although some alloys of aluminum-brass, copper-nickel, and copper-nickel-tin have been developed to obtain tubes with better corrosion and erosion resistance. The tubes extend between tube sheets drilled to receive the tube ends. They may be expanded into the sheet or fastened by use of packing or ferrules as shown in Fig. 8-3.



(*Courtesy of Worthington Pump and Machinery Corp.*)

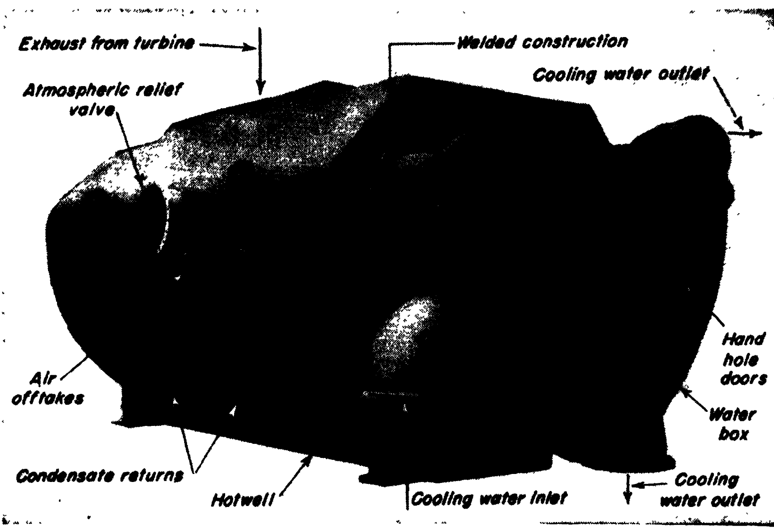
FIG. 8-5. Surface condenser cross sections.

Tube sheets may be drilled in a variety of patterns. The tube arrangement has a decided effect on condenser performance, and various layouts have become identified with certain companies. Figure 8-5 shows two arrangements used with the condenser of Fig. 8-1. Passageways or lanes are provided through and around the outside of the tube nests. These lanes provide an effective flow path for the steam from inlet to the exit lane and then to the air cooler. As shown here, the air-cooling section on this condenser is internal, and the air offtake removes the air-vapor mixture after passing the lower cooler tubes.

Tubes are supported by the support plates. These resemble the tube sheets and are drilled and chamfered for a loose fit of the tubes. Openings through the support plates provide for longitudinal steam distribution. Water boxes commonly of cast iron are used at both ends. One end is designed for an inlet and outlet nozzle for the circulating water. This water box is divided horizontally and makes possible the two-pass flow of circulating water. Inlet of water is always at the bottom to assure a full flow of water in the condenser. On large condensers this water box and that at the other end may also have a vertical dividing plate and two inlet and outlet circulating water connections. With the vertical division, one-half of the condenser tubes may be cleaned while the condenser is operating on the other half. Access to the tubes is obtained through manholes or handholes provided on the outer flat side of the water boxes. These openings are used when tubes are replaced or when they are cleaned. Cleaning is done in several ways. Wire brushing or shooting of special rubber plugs through the tubes is common.

Condenser shells were formerly cast, but the common construction today is of welded steel sections. In the particular condenser shown in Figs. 8-3 and 8-4 the shell consists of a cylinder of arc-welded steel plate to which heavy steel plate end flanges are welded. The opening in the end flange is eccentric. This locates the tubes and water boxes below the center line of the shell and provides a liberal steam space for steam distribution longitudinally and around the tube nest. Many condensers do not, however, have a cylindrical shell. Internal reinforcing is used for required rigidity. In large capacities the shells may be sectionalized and bolted together in the field. In condensers where the tubes are expanded into both headers, expansion is provided for by having an expansion joint in the shell. All condensers are fitted with hot wells. In Fig. 8-4 a cast-iron or steel hot well with water-gauge glass is bolted to a flanged opening on the bottom of the shell. It is desirable to have the water in the hot well at exhaust-steam temperature as any subcooling of the condensed steam in dripping over the tubes represents a waste of heat to the circulating water. This cooling inevitably occurs to some extent and is counteracted by constructing the hot well to act as a condensate heater. As shown in Fig. 8-5, the principle is to have full exhaust pressure and steam in the hot well and permit the condensate which has collected to cascade through this atmosphere of steam as it falls into the hot well. In so doing the condensate is reheated reasonably close to steam temperature. Condensers are commonly provided with outlets for an automatic exhaust relief valve, as indicated in Fig. 8-4. The purpose is to allow the engine or turbine to exhaust automatically to the atmosphere in case of loss of vacuum due to accident to the condensing apparatus. The valve is generally water-sealed to prevent air leakage into the condenser. These valves are also provided with attachments to hold them permanently open when necessary.

Surface condensers are often constructed with a noncircular cross section as in Fig. 8-6 to accommodate the condensing steam flow. The particular unit shown is for a turbine of 15,000-kw capacity and has 15,000 sq ft of surface area of the tubes. This is a ratio of 1:1, which is a rough relationship holding between turbine capacity and condenser size for lower initial steam pressure range. The unit shown has the usual connections for auxiliaries as indicated. This particular unit has a circulating water flow of 18,000 gpm. The water box is divided in thirds. Water comes in the middle box and makes a pass to the far one. The water returns through the top and bottom bank to the two outlets. The illustration shows the usual constructions of cast water boxes and welded plates to form the shell used in modern condensers.



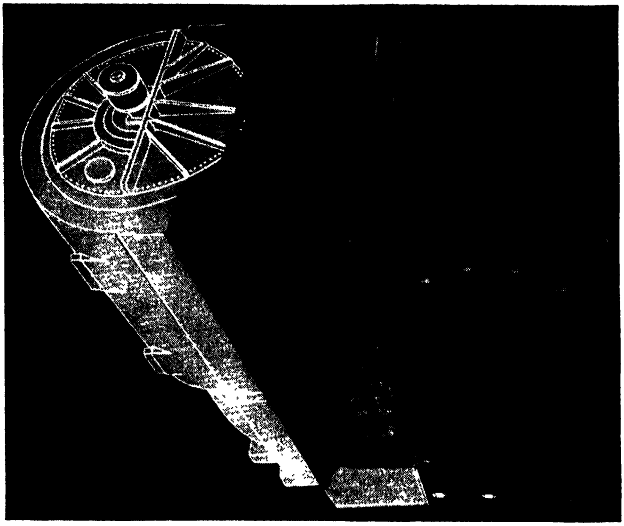
(Courtesy of Ingersoll-Rand Co.)

FIG. 8-6. Surface condenser showing welded shell construction.

In the attempt to reduce high initial capital investment in steam-turbine installations, a design of condenser has been evolved as shown in Fig. 8-7. A separate turbine foundation is eliminated by having a rectangular condenser act as the foundation for the turbogenerator. This design was based to some extent on experience with maritime installations. Savings in space and foundation costs are obtained. The condenser is of two-pass circulating-water flow design having a surface of 7,150 sq ft of ⅞-in. O.D. tubes 24 ft long. All auxiliaries, such as air ejector, generator oil coolers, and storage tanks, are mounted on the side of the condenser.

The turbine exhaust is connected to the condenser neck by either a welded joint (in the large size) or a bolted flange (in the smaller sizes). An integral steel expansion joint is placed immediately below this connec-

(Courtesy of Ingersoll-Rand and General Electric Co.)



(Courtesy of Foster Wheeler Corp. and Burns & Roe.)

FIG. 8-7a. Condenser-turbine generator, integral unit. Outside view of installation.

tion. This joint is a V-shaped expansion device which will take both transverse and up-and-down motion. The condenser completely supports the turbine and its generator. The cross section in Fig. 8-7*b* is not exactly that of the unit shown at left in *a* but it is typical of the rectangular condenser.

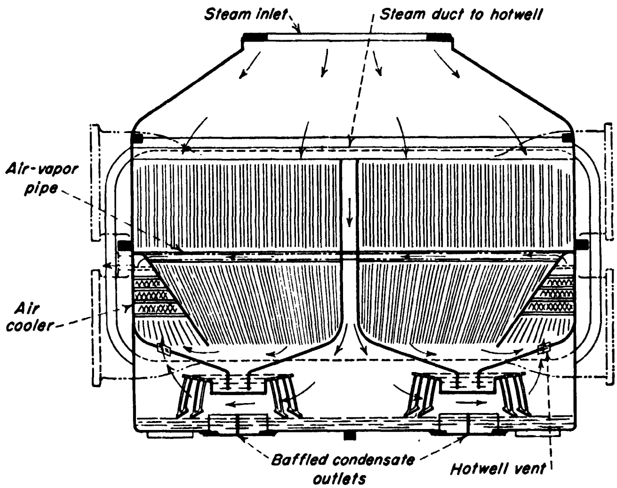


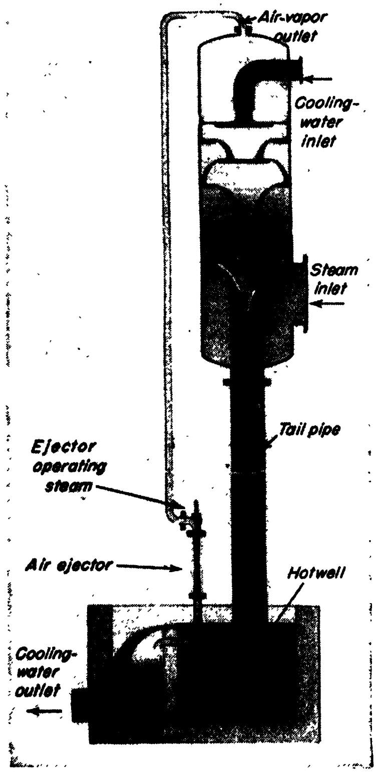
FIG. 8-7*b*. Typical cross section of rectangular-type condenser.

Jet Condensers. Jet condensers are built in three general designs:

Barometric
Low-level
Ejector

The barometric condenser, as in Fig. 8-8, is frequently used with industrial turbines. It is essentially a design of head and tail pipe some 30 ft in height. Mixing occurs in the head, and water and condensate stand at a height in the tail pipe depending on exhaust and atmospheric pressures. Continued inflow of water and steam causes an equal amount to leave the tail pipe. The condenser may be used with "air"-removal equipment to keep a lower pressure in the head. If a sufficient natural difference in inlet and discharge levels exists (approximately 10 ft), the condenser is self-acting, *i.e.*, water will flow in and out without any pumps. If this is not possible, a pump may be used on condenser inlet water. The piping is somewhat involved but not excessively so.

The low-level jet condenser, as in Fig. 8-9, is in effect a barometric condenser with mixing action in the head. The elimination of a tail pipe permits installing the condenser directly below the turbine. A pump removes water, and air-removal equipment may also be used. Air



(*Courtesy of Ingersoll-Rand Co.*)
FIG. 8-8. Typical barometric condenser.

removal is from the steam space, and inlet water flows in naturally because of the lowered pressure. A vacuum-breaking valve opens a line leading to the steam space from the atmosphere if the water level in the condenser rises too high. This is necessary since the water may back up and rise into the turbine if for any reason discharge from the condenser fails.

Another form of condenser, as in Fig. 8-10, is the ejector type. In this unit high-velocity jets of water condense the steam, and the resulting mixture is discharged through a diffusing tube to the atmosphere. These condensers act so that air is also removed because of this jet action. A vacuum breaker again eliminates the possibility of water backing up into the turbine, for the diffusing tube generally has a submerged outlet. Inlet water is supplied from a pump under some pressure to obtain the jet action of the water. Jet condensers are used in industrial plants generally for small turbine capacities.

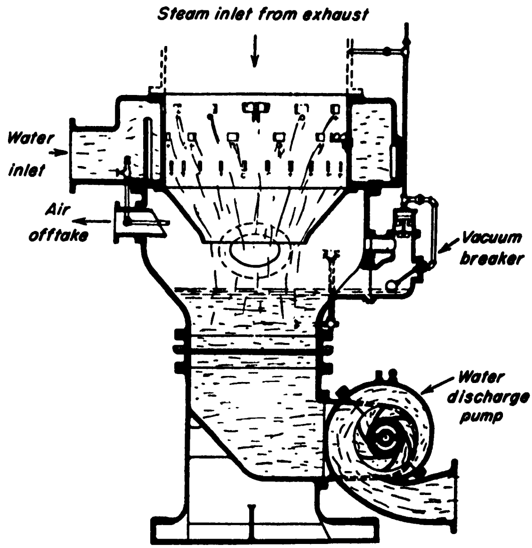


FIG. 8-9. Low-level jet condenser.

Condenser Air-removal Equipment. In the steam space of a condenser, either of the jet or surface type, there is always a certain amount of air and other noncondensables which unavoidably gets in. This air may come from turbine gland leakage, porous exhaust hoods, drains, flexible condenser joints, air in the circulating water of jet condensers, or any air originally in the steam. Air leakage may run from about 1.5 to 6 cfm for surface condensers in good condition. This air is in general not removed with the condensate and circulating water, and tends to concentrate in the condenser and so increase the turbine exhaust pressure. The air is usually diffused throughout the steam in the steam space. It is, therefore, removed by some form of pumping equipment which has its suction in the steam space and draws out a mixture of air and steam. A number of different types of pumps have been used, but the most common and most widely used equipment today is the steam-jet air ejector which has been standard equipment for condensers since it supplanted the reciprocating and hydraulic types of vacuum pump. In these units high-velocity steam jets entrain the air and vapor mixture led from a condenser by an air offtake pipe. The steam jets picking up this mixture at condenser pressure have enough power to discharge it to atmospheric pressure

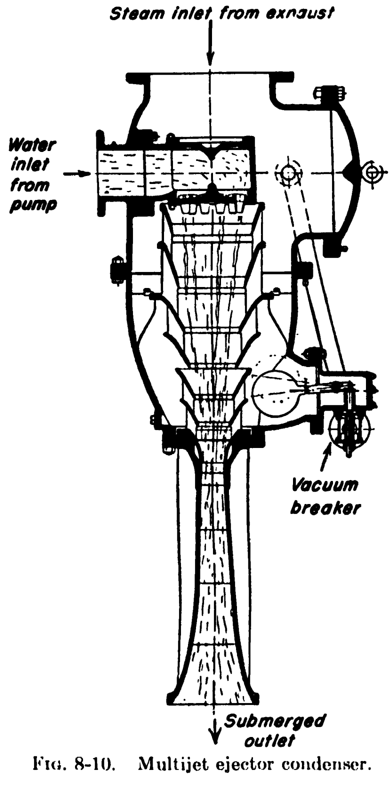
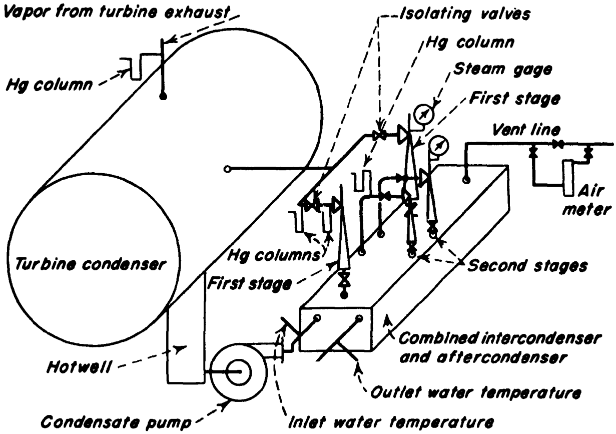


FIG. 8-10. Multijet ejector condenser.

through diffusing tubes. Figure 8-11 illustrates diagrammatically the hookup of an ejector to a surface condenser. Steam at about 100 to 300 psi is commonly used for its operation. A two-stage ejector is generally required for vacua greater than 26 in. Hg. One stage suffices for lesser vacua. Circulating make-up or condensate from the hot well through intermediate and aftercondensers recovers the heat in the steam from the jets. Parallel jets are used for added capacity. The multijet unit lends itself to more efficient part-load operation by stopping some of the jets and allowing the remainder to work at best efficiency.

The first-stage ejectors discharge to the intercondenser while the second-stage jets discharge to the aftercondenser or to an open heater. A vent line is required from the aftercondenser through which the air dis-



(*Courtesy of Power*)

FIG. 8-11. Condenser hookup with two-stage steam-jet air ejector.

charge to the atmosphere may be metered. When used with jet condensers, the intercoolers are also often of the jet type.

To improve the operation of the air ejector, the mixture is passed through a special air-cooling section integral with the condenser, as in Fig. 8-12, or external to the main condenser. This section uses the coldest water available to reduce the mixture temperature leaving the main condenser to a low value. The lower the mixture temperature, the greater will the ratio of air to vapor be in any volume taken from the condenser air offtake.

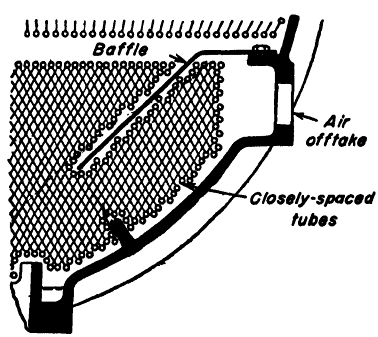
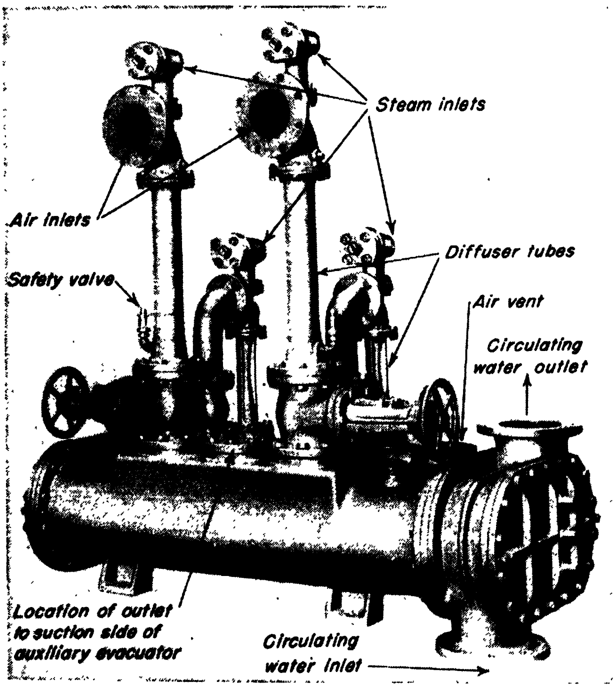


FIG. 8-12. Condenser air cooler.

A typical two-stage ejector is shown in Fig. 8-13. It is a double-element two-stage ejector consisting of two first-stage and two second-stage ejectors. Each of the elements is commonly designed for normal capacity so that one may be used for stand-by. The type of unit shown is used with surface condensers but may also be used with the barometric or other direct-contact type of condenser. The stages are arranged in series with inter- and aftercondensers, which are in a common shell. Valves are part of the unit to permit independent operation of either element. Steam nozzles are made of steel or monel metal to withstand the action of superheated steam. The body and diffuser sections are made of hard-grained cast iron. Deflecting baffles are used under the

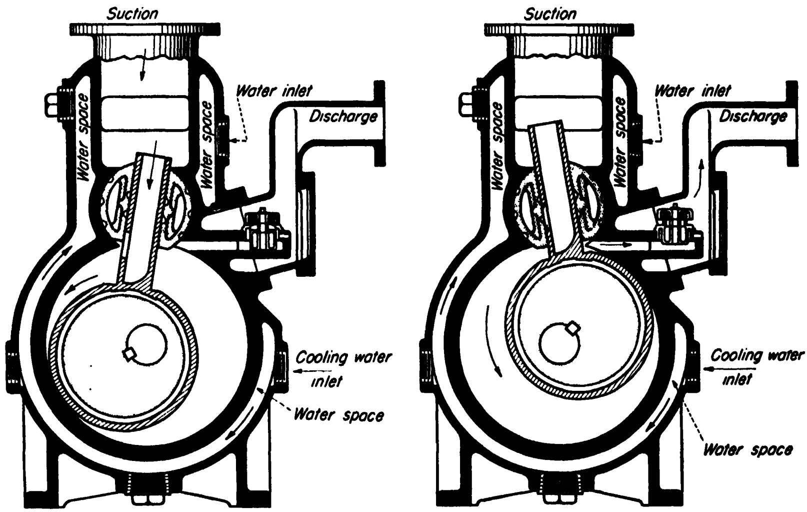


(*Courtesy of Worthington Pump and Machinery Corp.*)

FIG. 8-13. Double-element two-stage condensing steam-jet ejector with inter- and aftercondenser.

discharge of each stage to protect the tubes from direct impingement of the vapor. Condenser tubes are of Admiralty metal flared at the inlet and packed at the outlet end in threaded boxes with metallic packing and ferrules. Tube sheets are of Muntz metal. A loop seal or drain trap handles condensate from the intercondenser, and a drain trap is used for the aftercondenser. A hogging jet or auxiliary single-stage noncondensing ejector (not shown in Fig. 8-13) may be used for removing large quantities of air from the system at low vacuum. This evacuator is usually cut out at about 20 in. of vacuum, and the main ejector operates from there on. Main condenser condensate is used for circulating water in the inter- and aftercondensers. At light loads condensate recirculation keeps the flow through the ejector condensers at a required minimum.

The steam-jet air ejector has advantages over rotary pumps in (1) lower first cost, (2) no moving parts, (3) greater air-removal capacity. However, several large surface condensers have recently been equipped with rotary-type vacuum pumps in preference to air ejectors. Reasons for this, particularly with the trend to h-p and high-temperature units,

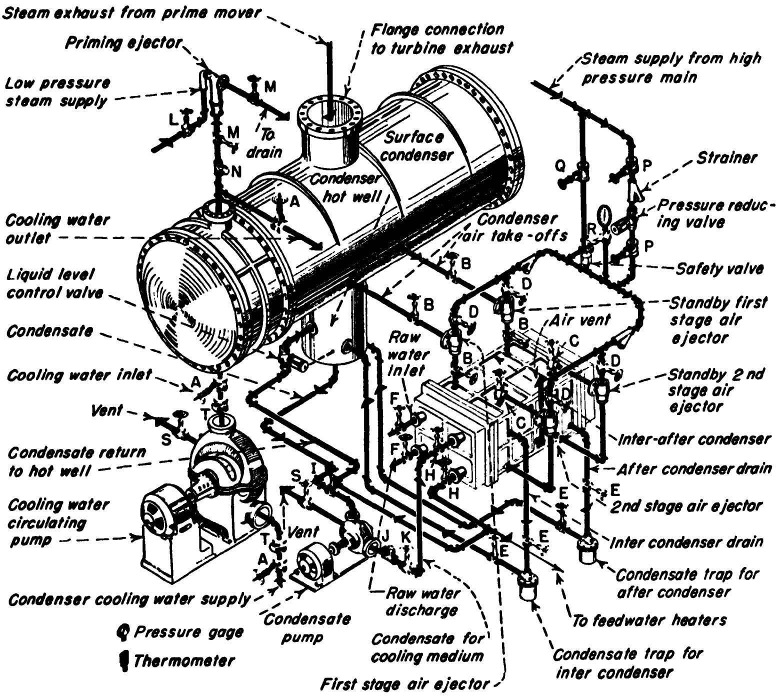


(*Courtesy of Kinney Manufacturing Co.*)

FIG. 8-14. Rotary-type vacuum pump.

are (1) the sometimes excessive maintenance of pressure-reducing devices, since steam jets commonly operate at reduced pressures, and (2) the contaminations which are believed to be due to returning air-ejector drains to the fluid cycle in power plants. A typical modern rotary pump is shown in Fig. 8-14. This pump can be used for pumping down when starting up and for normal air removal. Steam-jet ejectors often have extra-large-capacity jets to assist in initial air removal for starting up. The pump shown has an eccentric rotor which, operating in the casing, serves to take in and exhaust the air-vapor mixture from the condenser. A viscous fluid maintains the seal between rotor and casing between which no metal-to-metal contact exists. Air inflow is through a plunger and port, and discharge through an outlet valve. In condenser service, discharge is through a baffling tank and whirl chamber to remove moisture and sealing fluid. The rotary pump has been applied in combination with a steam jet wherein the steam jets are the primary units and the rotary pump acts as the secondary stage. The pump so used can also function during the starting operation for initial pumping down.

Condenser Auxiliary Layout. A condenser installation consists of the condenser proper and auxiliaries associated with the water flows and those



(*Courtesy of Jenkins Bros.*)

FIG. 8-15. Surface condenser with two-stage steam-jet air ejector.

with the air removal which affect the vacuum. Figure 8-15 shows in considerable detail the assembly of a surface-condenser installation. In effect the same type of design is used for jet condensers except as some of the water pumps may not be used. The condensate is removed from the bottom of the hot well by a condensate pump and acts as a cooling medium through the air-ejector condensers from which it goes to other heaters for further heating. At times other heat exchangers may be placed before the ejector condensers.

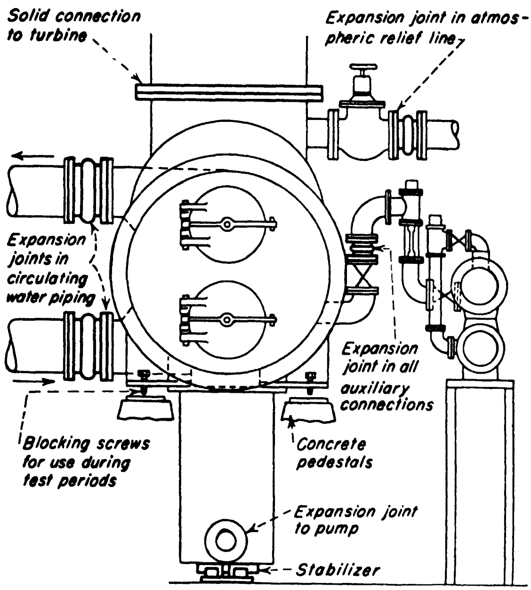
Centrifugal-type pumps are common for large surface condensers although reciprocating pumps may be used for smaller sizes and for jet condensers. Plentiful supplies of circulating water are necessary, and plants are located generally where access to river, ocean, or bay water is possible. About 100 lb of such water is required for each pound of steam condensed. Circulating water flows into the plant through a large intake tunnel and through screens to remove any debris. If water is not readily available, cooling towers may be used for water supply to the inlet of the circulating pumps. These are most often centrifugal pumps which discharge to the bottom of the condenser water box, and the water after circulation leaves at the top. When running full, the intake piping and outlet piping dip under water, and the entire system is, in effect, a siphon which reduces the power needed for the circulating-water pumps. A priming ejector as shown is used to evacuate the water side of the condenser for ease in starting the circulating-water flow. Surface condensers are commonly fitted with atmospheric relief valves which are by-pass valves that permit exhaust steam to be discharged to the atmosphere instead of to the condenser.

Turbine and Condenser Supports. Condensers are placed as close as possible to turbine exhaust nozzles to maintain the lowest possible pressure on the turbine exhaust. In attaching condensers to turbines two main problems arise. The weight of the condenser and water content tends to pull down and distort the turbine casing, and any condenser movement from temperature changes tends to set up stresses in the circulating condensate piping. Common methods for installing condensers are as follows:

1. Hung from turbine
2. Rigidly mounted on foundation with flexible connection to turbine
3. Rigidly fastened to turbine and mounted on a spring foundation

The hung method is used on small- and medium-sized condensers while the other methods find use on all sizes. The particular method used depends upon such factors as turbine size, exhaust shape, basement depth, foundation construction, offset nozzle, or side steam inlet. If all three methods are feasible, the selection depends upon relative costs. Figure 8-16 shows a typical hung condenser. The condenser is rigidly connected to the turbine which bears the full weight. Expansion joints for all lines connecting to the condenser are required. The condenser design is such that the inlet from the turbine is above the center of gravity of the condenser to prevent unbalanced loading. A condenser's steam space is at times filled with water for leakage test purposes. Blocking screws are used at such times to bear the added load. Vibration problems with this type of mounting are offset by stabilizers. As shown, they may be in the form of blocks welded to the hot well. These butt against stationary blocks and allow up-and-down motion but check sidewise vibrating motions.

The flexible connection, as in Fig. 8-17, is usually more desirable. It eliminates the need of flexible connections in attached piping but, in terms of cost, is generally offset by the cost of the one large connection. It has disadvantages in that it is difficult to fit to rectangular or odd-shaped inlet openings and may leak some air into the condenser. Then



(*Courtesy of Power.*)

FIG. 8-16. Hung condenser and piping connections.

also, since there is a vacuum in the condenser, there is a collapsing action between the turbine exhaust nozzle and the condenser. This action tends to close the expansion joint and causes an upward pull on the condenser and a downward one on the turbine exhaust which necessitates stronger turbine supports. Flexible connections permit offset mounting of condensers and are adaptable to side-exhaust turbines.

The spring-mounted condensers are used on most medium- and large-sized units. There is a rigid connection between turbine exhaust and condenser inlet, as in Fig. 8-18. The condenser can move to accommodate temperature expansion, and the springs keep the loading on the turbine exhaust nozzle within limits. This mounting is very suitable to rectangular or odd-shaped exhausts. The springs as to type, number, and size

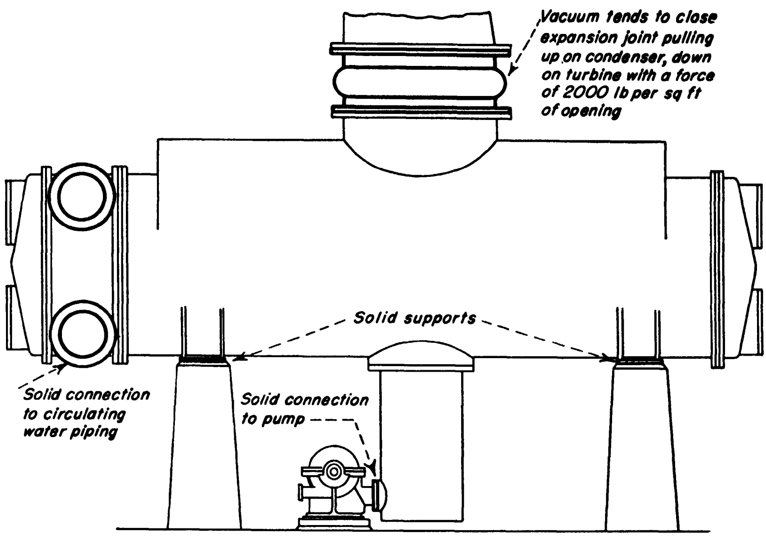


FIG. 8-17. Condenser with flexible inlet connection.

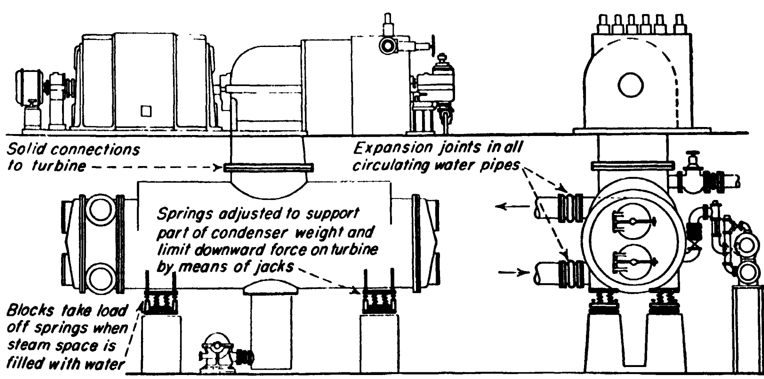


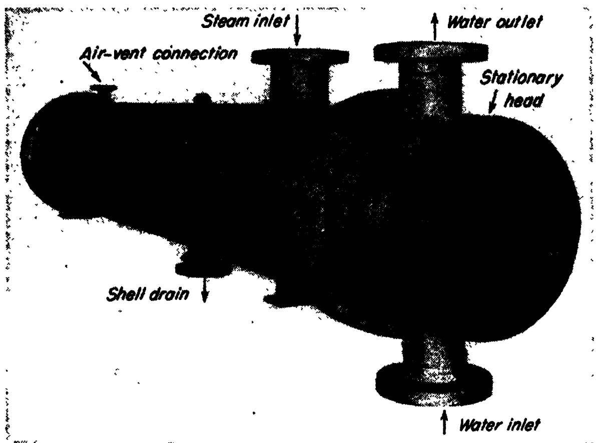
FIG. 8-18. Spring-supported condenser with rigid inlet connection.

are based on condenser and water weight, amount of expansion, and allowable turbine load. The spring design accommodates unbalanced loading and prevents distortion of the turbine exhaust. Condensers having this type of support must have expansion joints in all piping connections. At the spring supports, there are also blocks to take the load when condenser steam space is water-filled for leakage tests and jacks to adjust the spring load.

Closed and Open Feed-water Heaters. Feed-water heating has advantages in that it (1) improves efficiency and so reduces the amount of fuel needed, (2) lengthens boiler life, and (3) reduces maintenance. "Closed" and "open" are the two main classes of heaters. They are also referred to as "surface" and "contact" types. Closed or open heaters use steam that might otherwise be wasted or bled from turbines for the heating medium. Closed heaters transmit the heat through tube walls so that the steam and water do not come into physical contact. They avoid all contamination, *e.g.*, by lubricating oil from engine exhausts, and they permit the heat transfer when water pressure is greater than that of the steam. This latter feature is important in regenerative-cycle turbine plants where the feed-water or other pumps discharge water at high pressures which is to be heated subsequently in several stages, as indicated in Fig. 8-1.

There are innumerable forms of closed heaters, generally divided into steam-tube and water-tube types. Most feed-water heaters are of the latter type. Structually they are also classified as straight-tube or bent-tube (spiral coils or U tubes) heaters and as single or multipass. The design of heaters must take into account the following factors: (1) shell and joint tightness and strength, (2) shell and tube expansion, (3) tube cleaning and replacement, (4) venting of noncondensable gases, (5) heat transfer (condensation, desuperheating, and subcooling of heating steam). Both closed and open heaters may be piped for "through" or "induction" operation as far as heating steam flow is concerned. The former is used chiefly in noncondensing plants where a large excess of heating steam is available. In these heaters all the steam passes to the heater, and some condenses while the remainder passes to process lines. The heater must be amply designed for the steam flow involved. In the induction heater no steam leaves the heater except for venting, and condensation of the steam draws in only as much as is needed to heat the feed water.

The usual closed heater consists of a cylindrical shell of steel or cast iron, depending on the heater pressure. The tubes are of steel or copper alloy expanded into tube sheets. A typical bent-tube design is shown in Fig. 8-19 and a straight-tube design in Fig. 8-20. The straight-tube design has straight tubes rolled into sheets at each end. The end opposite the water connections or stationary head has a floating head construction to take care of expansion strains by being free to move in the shell. The unit shown in Fig. 8-20 is of vertical design but is also used horizontally as in Fig. 8-21. This latter illustration is an external view of a typical heater as it appears before it is covered with insulation. The U-tube-design heater is simple, and the absence of a floating head eliminates one joint. They are generally of smaller diameter than floating-head types and often preferable when wide temperature differences exist. The unit

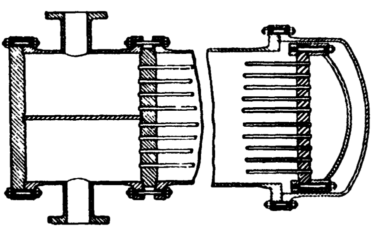


(*Courtesy of Griscom-Russell Co.*)

FIG. 8-21. Typical horizontal feed-water heater with insulation removed.

shown in Fig. 8-19 is mounted horizontally but may also be used vertically with proper connections.

High-pressure applications require heaters with special designs to contain the h-p water. This means the use of heavy heads and heavy bolting or some ingenious locking design as in Figs. 8-19 and 8-20. The design shown uses a head cover notched like an artillery breech-block but without screw thread. The head is inserted and turned the width of one projection for locking against the force of the water pressure. Water tightness is secured by the use of a ring gasket. This is held in place by two concentric rings and bolts which make any leakage visible. A similar construction may also be used on the floating head. Lower water pressures are accommodated by simpler construction as in Fig. 8-22.

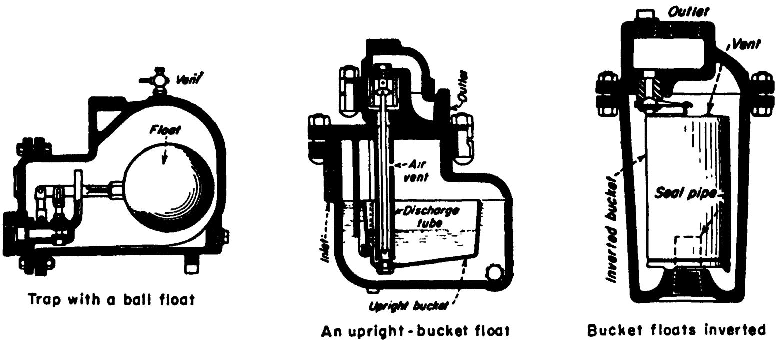


(*Courtesy of Griscom-Russell Co.*)

FIG. 8-22. Head constructions for low (below 300 psi) pressures.

A closed heater may be designed to heat feed water in several ways. The simplest form condenses steam in the shell and heats the water to a temperature somewhat below steam temperature. This type of heater, The noncondensable gases pass to the atmosphere through the vent condenser. Deaerated water enters a storage space after leaving the scrubber section. Cold water enters the heater through a float-controlled valve as in the tray-type unit.

Traps. The use of steam in power plants commonly results in the formation of condensate which must be removed from equipment. The condensate may be formed because of undesirable heat losses such as those occurring in steam piping, or it may result when steam is used as a heating medium, *e.g.*, in feed-water heaters. In either case it is essential that this



(*Courtesy of Power.*)

FIG. 8-27. Mechanical traps with float and bucket mechanisms.

condensate be constantly removed to permit operation of equipment. Some form of pump is usually necessary if the condensate is to be discharged to a region of higher pressure. When it is possible and desirable to discharge the condensate to a lower pressure region, no power is required, and some simple equipment in the form of a trap is frequently used. The trap acts essentially like a valve which opens to let condensate pass but closes the passage to steam. The removal is accomplished by the pressure applied from within the equipment being drained. Traps may also serve an additional purpose in removing air which may be accumulating in the equipment.

Steam traps of many different designs exist and in general are of a mechanical, thermostatic, or labyrinth type. Figure 8-27 illustrates a typical set of mechanical trap designs. In the ball-float trap, condensate entering by gravity causes the ball to float and open a valve to the discharge line. This trap often acts as a continuous discharge if the condensate level positions the float according to inflow. The jet cock is used for purging air. In the inverted-bucket trap, as condensate continues to enter, it overflows into the bucket which eventually sinks and the water in the bucket is discharged through a valve in the outlet. The air vent permits discharge of air when the trap operates. As condensate enters the inverted-bucket trap, the bucket is sealed and any air trapped in the bucket is forced out the vent while the bucket floats. Continued condensate inflow causes the bucket to sink and open the discharge valve until steam enters and makes the bucket buoyant by filling it. Air entering with the condensate passes through the bucket vent and pockets around the discharge valve. Air is, therefore, purged during each cycle. Figure 8-28 shows a typical thermostatic and orifice trap. In the former

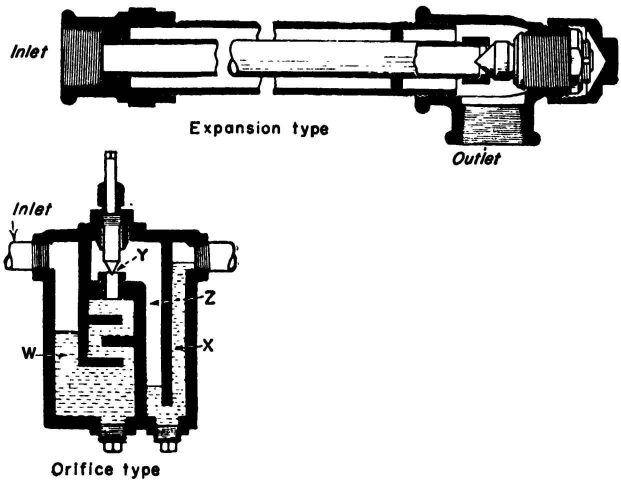
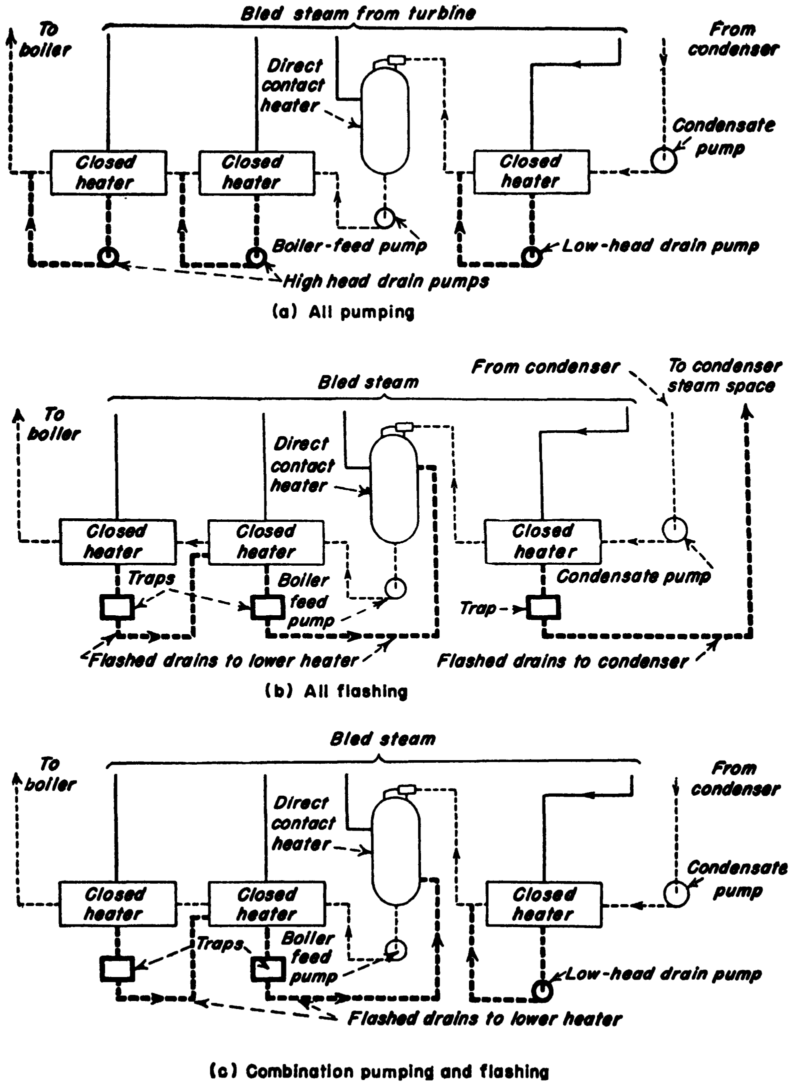


FIG. 8-28. Expansion and orifice traps.

as long as cool condensate or air enters the trap, the valve is open and discharge occurs. Steam entering causes the metal tube to expand and close the flow. These traps discharge fairly continuously. The orifice trap has a receiving chamber W leading to a chamber X through an adjustable orifice Y. Condensate at low temperatures flows past the baffles out the orifice. As condensate temperature increases, it flashes in chamber Z and chokes off the discharge. Air entering W will depress the water level below the sealing baffle and pass up through the orifice.

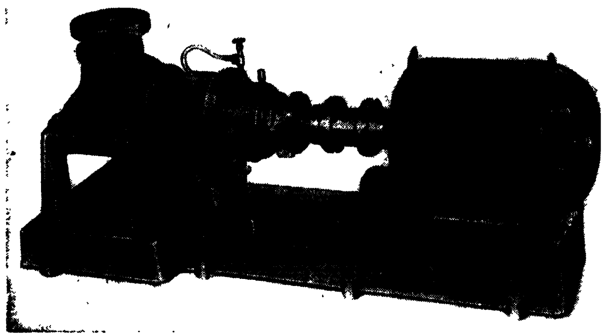
Heater Drainage Systems. The condensate formed in closed heaters, or "heater drains" as they are commonly called, is returned to the feed-water cycle. This saves on make-up water and a certain amount of heat as the drains are close to heater steam temperatures. Two principal methods are used in power plants to handle this condensate: (1) pumping into the feed-water line and (2) flashing to a lower pressure heater. Pumping of all drains, as in Fig. 8-29a, requires power-driven equipment.



(Courtesy of *Power*.)

FIG. 8-29. Heater drainage arrangements.

Pumps that will handle h-p and high-temperature liquid are required for the h-p heaters, as in Fig. 8-30, and are usually motor-driven centrifugal units. Technically, this is the most efficient way to handle drains, but it may not be the most effective method from the over-all economic viewpoint. Flashing of all drains, as in Fig. 8-29*b*, is the simplest mechanical arrangement and requires no power-driven equipment. It is done by the use of some form of trap in each heater drain line which discharges to the steam space of some other heater or condenser. While any heater drain



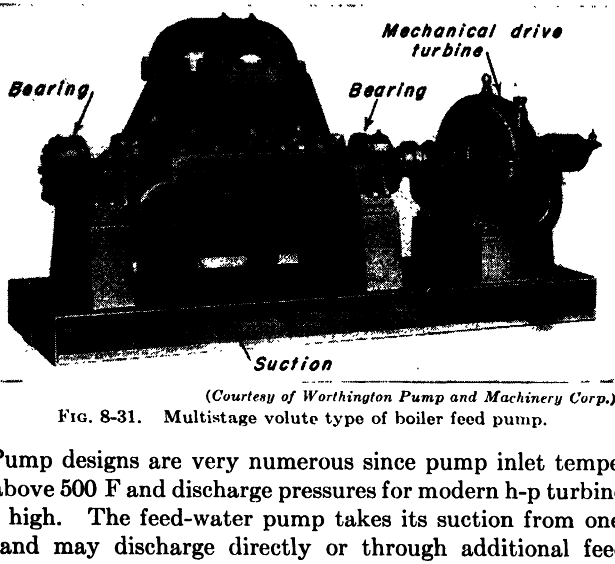
(*Courtesy of Worthington Pump and Machinery Corp.*)

FIG. 8-30. Typical heater drain centrifugal pump.

can be flashed to any lower pressure region, the usual commercial application is as in Fig. 8-29*b*; *i.e.*, the drains are flashed to the next lower pressure heater rather than all being discharged to, say, the condenser or lowest pressure heater. This method of successive flashing permits heat recovery at higher temperatures than if all were flashed to the lowest pressure heater. Practically, troubles arise in the form of damage to traps by wire drawing and by erosion of piping due to high-velocity discharge from the traps.

In commercial power plants the best economical arrangement generally works out to be some combination of the two methods, as in Fig. 8-29*c*. As shown, the h-p drains flash ultimately to the open heater, and the lowest pressure closed-heater drains are pumped into the feed line. This method requires a simple drain pump that does not operate at high pressures and temperatures. Should there be no open heater in a plant, the h-p heaters would have their drains flashed to the l-p closed heater. A comparison of heat balances indicating thermal performance will generally show only about 1 per cent better performance for the pumping system compared to the flashing system, as in Fig. 8-29*a* and *b*. The combination in Fig. 8-29*c* is about ¼ per cent poorer than the use of all pumps for heater drainage and $\frac{3}{4}$ per cent better than all flashing. Practically, the conbination system has advantages in that little attention is required for the traps or drainers and that no h-p drain pumps are required. In practice the exact arrangements of heater drains may vary from those indicated, depending on the number of stages of heating and the economics of the particular hookup, but the diagrams do indicate the general pattern of cycle arrangements.

Boiler Feed Pumps. Feed water for boilers is supplied by mechanical pumps except in some l-p heating installations or where injectors may be



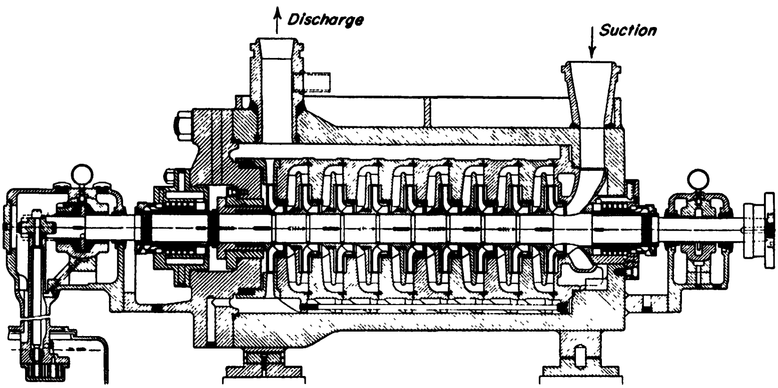
(*Courtesy of Worthington Pump and Machinery Corp.*)
FIG. 8-31. Multistage volute type of boiler feed pump.

used. Pump designs are very numerous since pump inlet temperatures may be above 500 F and discharge pressures for modern h-p turbine plants are very high. The feed-water pump takes its suction from one of the heaters and may discharge directly or through additional feed-water heaters to the boiler. The boiler feed pumps are of two general types:

Reciprocating
Centrifugal

The reciprocating pump is of either the piston or the plunger type and may be motor-driven or direct-connected to a steam piston, giving the common simplex, duplex, or triplex pumps. Centrifugal pumps (because of the high discharge pressures) are commonly of the multistage type and may be of either the volute or the diffuser type. Centrifugal pumps are common in steam-turbine power plants, and two representative pumps are shown in Figs. 8-31 and 8-32. The former indicates the external appearance of a boiler feed pump for pressures up to 1,200 psi and is shown as a unit driven by a turbine. It is a balanced multistage volute type built in two- to six-stage units with the casing horizontally split. Axial balance is achieved by having an equal number of impellers facing in opposite directions. Radial balance is obtained by having the volutes staggered. Balancing in this direction helps to reduce vibration, stuffing-box troubles, and wear on internal clearances.

Figure 8-32 is typical of a h-p pump. The pump casing is made of forged steel with a solid type of case. This pump is so designed and constructed that the rotor may be removed or assembled outside and inserted



(*Courtesy of Ingersoll-Rand Co.*)

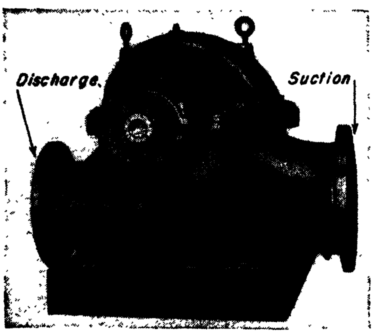
FIG. 8-32. Multistage solid-case type of boiler feed pump.

into the body of the pump without disturbing the suction and discharge piping connections.

Circulating Pumps. A turbine condenser requires about 100 lb of circulating water to condense 1 lb of exhaust steam. This large water flow is necessary since, in absorbing the heat given up by the exhaust steam, only a small temperature rise of the condensing water can be allowed or the condenser pressure would rise. The supply of this water is an important factor in plant location and plant extensions. The water for condensers usually comes from oceans, bays, lakes, or rivers. When these sources are not available, water may be recirculated and cooled by cooling towers so that the heat finally discharges to the atmosphere. Part of all condenser installations is a system for providing this water flow. It will in general consist of

Intake works
Trash screens
Intake tunnels
Circulating-water pumps
Condenser water boxes and tubes
Discharge tunnels

The circulating-water pumps provide the means for causing the water to flow into and through the condenser. The type most commonly used is the horizontal double-suction volute centrifugal pump, as shown in Figs. 8-33 and 8-34. The pump is generally located on the basement floor above the water supply. The one shown has a horizontal suction although vertical suction is also commonly used. Depending on design, these pumps may range from a maximum capacity of 2,200 to 6,750 gpm, a maximum head of 129 to 373 ft, and a maximum speed of 1,450 to 1,750 rpm. The casing is of cast iron split on the horizontal center line, and the suction and discharge nozzles are cast integral with the lower half. The impeller may be inspected without breaking the water connections. The balanced impeller is of bronze, the shaft of steel, and the guide rings of bronze also. Ball bearings of the double-row type located on each side of the casing are used. One outboard bearing is of the angular contact type for both radial and thrust loading. As seen in Fig. 8-34 it is water-jacketed, cooled by the water being pumped. Water boxes are shown with bronze glands. A water seal prevents air leakage and is supplied with sealing water from the pump discharge. The coupling is of the flexible type with a rubber bushing and pins for power transmission. The pump has a horsepower characteristic such that no overloading of the motor can occur between shutoff and full capacity. When located above the source of water supply, it must be primed before starting.

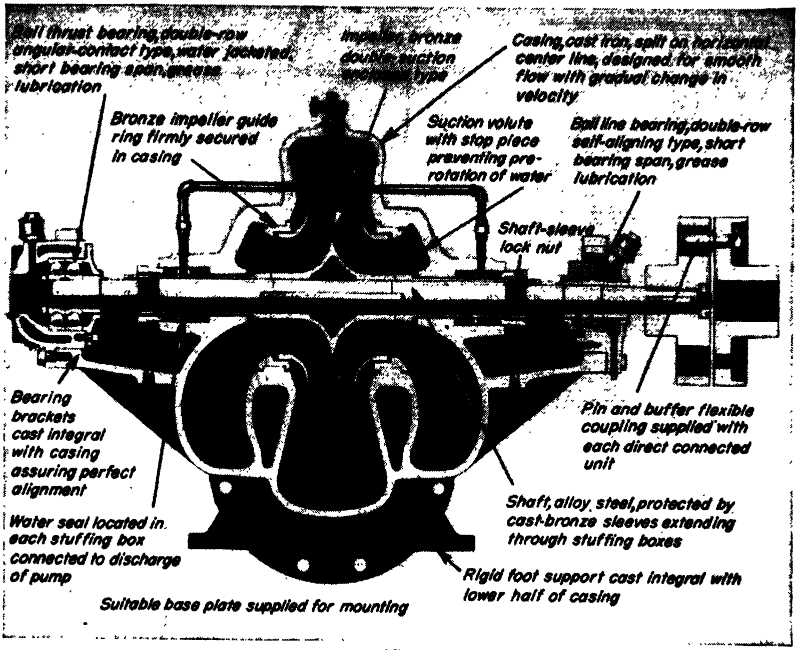


(Courtesy of Worthington Pump and Machinery Corp.)

FIG. 8-33. Typical volute type of centrifugal circulating pump.

Circulating pumps may also be built with impellers of the axial flow or propeller type. These impellers are used for vertical pump installations as are also some centrifugal types of impeller. Vertical pumps may be installed in a pit so that the pump is below the minimum water level. This is called a "dry-pit" installation in contrast to a wet-pit. In the latter, the pit is filled with water, and the impeller on its vertical shaft is submerged in the water. The vertical pump when below water level has the advantage of functioning without initial priming.

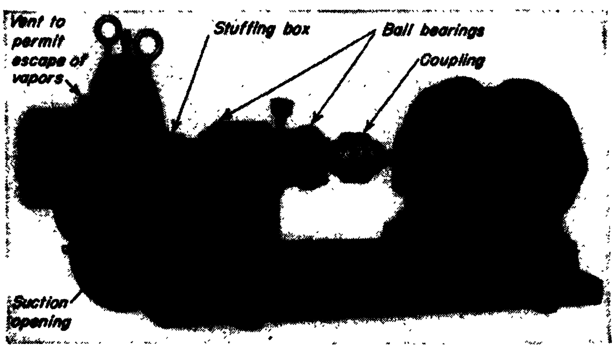
Hot-well or Condensate Pumps. Condensate pumps remove the condensate from the hot wells of surface condensers. The low pressure or vacuum at the level of the condensate in a condenser makes the pumps operate with a high suction lift. In addition, the water from the hot well, near its boiling point, gives rise to the problem of some of the water flashing into vapor. The pumps must also operate over a wide range in capacity as a result of load changes on the steam turbine and are often oversize to handle possible abnormal flow conditions. The condensate pumps are designed to have glands that will prevent air leakage into the condensate



(*Courtesy of Worthington Pump and Machinery Corp.*)

FIG. 8-34. Section of volute type of single-stage circulating pump shown in Fig. 8-33.

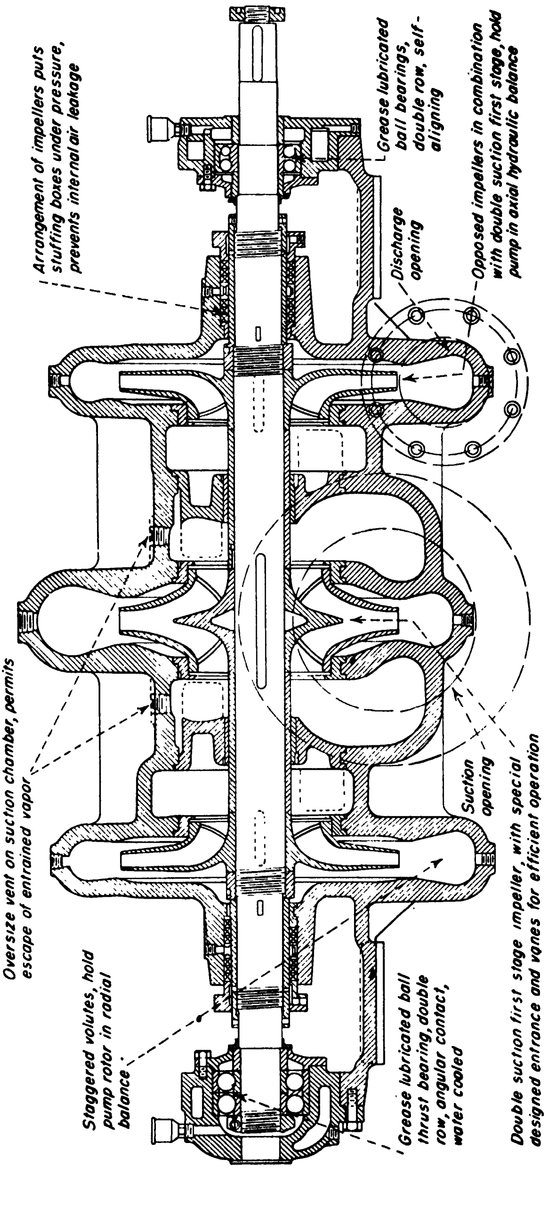
and are at times installed in duplicate to assure continuous operation in case of failure of one of them. The pumps are placed at a lower elevation than the hot-well outlet to have the advantage of a static head of water. They have very large suction connections and suction chamber so that the water will flow with very little loss of head. Condensate pumps may discharge to atmospheric pressure, *e.g.*, in an open heater. They often have to discharge at higher pressures if such equipment as oil coolers, air-ejector condensers, air coolers, and a l-p heater are in the hookup before the open heater, as in Fig. 8-1. Condensate pumps are most often of the centrifugal type in modern power plants, although reciprocating pumps are used in some small condensers. A motor-driven single-stage centrifugal pump is shown in Fig. 8-35. The casing of cast iron splits horizontally for ready access and removal of the rotor. A vent connecting with the suction volute permits escape of entrained vapor. The impeller is of bronze and is of the single-suction enclosed type. The discharge side of the impeller is adjacent to the stuffing box so that the latter is under discharge pressure. In this way leakage of air into the pump is prevented. The shaft is of steel supported by one single-row line bearing and by one single-row thrust bearing of the ball type. The stuffing box is pressure-sealed during operation.



(*Courtesy of Worthington Pump and Machinery Corp.*)
FIG. 8-35. Single-stage condensate pump.

Oil Coolers. Lubricating systems of turbines require means for cooling the lube oil. In circulation-type systems cooling is accomplished by a separate oil cooler, as in Fig. 8-37. The oil returning from the bearings is circulated through the shell side around the tubes. Transverse baffles direct the flow. The cooling water flows through the tubes arranged in one or two passes. The cooling water may be raw water or condensate from the condensers. In the latter case the heat loss picked up by the cooling oil in the bearings and other parts returns to the system, bettering the over-all operation of the plant.

In construction, the oil-cooler shell is generally bored to remove the possibility of grit and core sand being picked up by the lube oil. One end of the cooler is fitted with a conventional head and cover plate which may be removed for tube inspection. Two-pass coolers on the tube side permit cleaning and examining the tubes without breaking water connections by removing both head covers. In a single-pass type, water connections coming out the ends may make this impossible. The other end of the cooler is of a floating-head construction to permit expansion and contraction of the tubes. This head is packed, and any leak becomes



(Courtesy of Worthington Pump and Machinery Corp.)

FIG. 8-36. Three-stage centrifugal condensate pump.

ratio varies with pressure ratio, and (*e*) how the turbine exhaust-gas and compressor-outlet air temperature varies with the pressure ratio. Solid lines are for 1500 F inlet gas temperature to the turbine and dashed lines for 1000 F. Inlet air temperature is 60 F, and miscellaneous losses are 0 Btu.

Ideally, for either temperature, the cycle thermal efficiency increases with rising pressure ratio; surprisingly enough, the efficiency is slightly lower for a higher temperature at any given pressure ratio. However, when machine efficiencies fall below 100 per cent, the higher inlet temperature definitely gives better efficiency. For instance, with 1000 F gas and 80 per cent machine efficiency, thermal efficiency rises to a maximum value of about 11 per cent at a pressure ratio of 4. At higher and lower pressure ratios, the thermal efficiency is lower. With 1500 F gas and 80 per cent machine efficiencies, maximum thermal efficiency has a top value of about 19 per cent at a pressure ratio of 8. These curves show that cycle operation also depends on high machine efficiencies; for values much less than 70 per cent, the cycle can't even be made to operate! The turbine doesn't develop enough power to operate the compressor.

The big improvement in performance with higher turbine-inlet temperatures for given machine efficiencies is obvious from a study of the curves. Gas-turbine development waited upon the invention of compressors with a high enough efficiency and upon the production of metals that could withstand high temperatures for a reasonable length of time. At the present time, actual machine efficiencies run about 85 per cent and temperatures from 1100 to 1500 F. To realize maximum thermal efficiencies, temperatures and pressure ratios must be properly selected for the available machine efficiencies.

Figure 9-3*c* shows that for the ideal plant (100 per cent machine efficiency) the air rate drops rapidly as the pressure ratio increases above unity, reaches a minimum, and then slowly rises. For lower machine efficiencies and inlet gas temperatures, the air rate is higher; for given values of both, the air rate has a minimum value at a certain value of pressure ratio. These minimum values do not coincide with the maximum values of thermal efficiency in respect to pressure ratio. Air rates give an idea of the relative physical size of plants; the higher the air rate, the larger the compressor, turbine, furnace, and connecting ducts to accommodate the flows.

Figure 9-3*d* shows that, for the ideal plant, the work ratio falls as the pressure ratio increases above unity; *i.e.*, the compressor takes a greater proportion of the mechanical work developed by the turbine. Despite this, remember that the thermal efficiency for a given machine efficiency

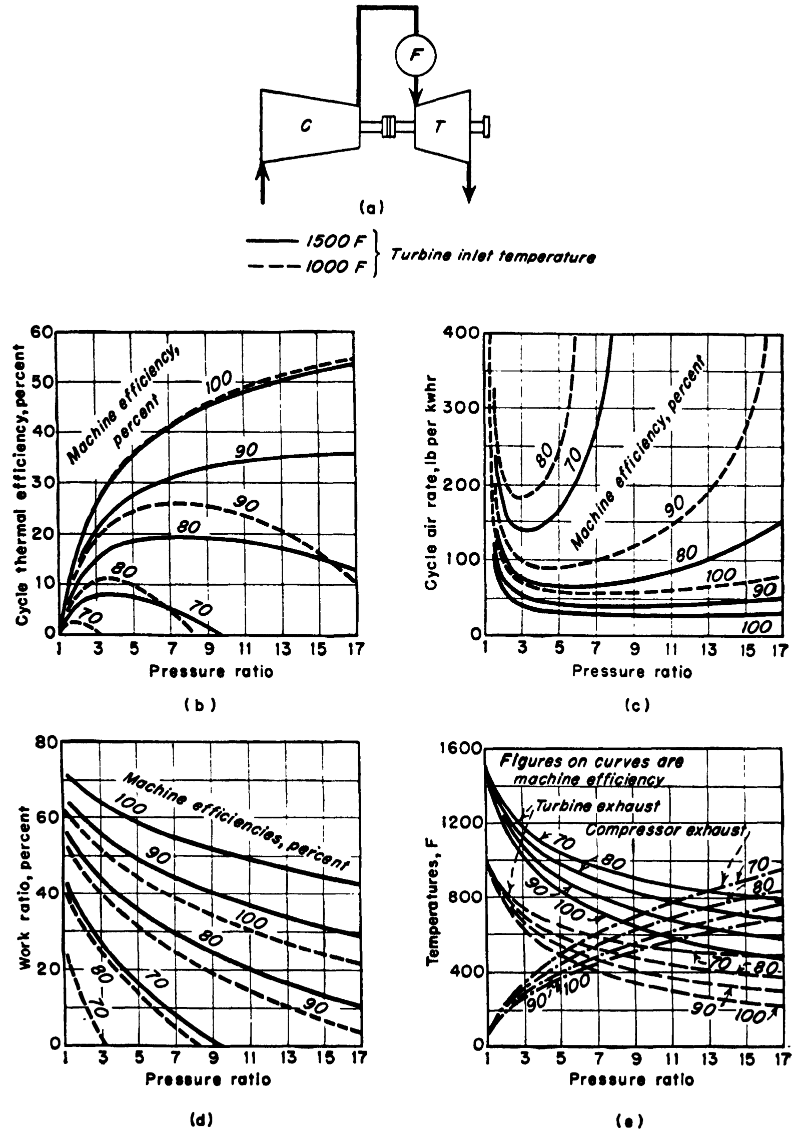


FIG. 9-3. Theoretical performance of a simple-cycle gas-turbine plant with zero miscellaneous losses and inlet air to compressor at 60 F. Dashed lines are for 1000 F turbine-inlet temperature; the solid lines for 1500 F. (*b*) Thermal efficiency vs. pressure ratio. (*c*) Air rate vs. pressure ratio. (*d*) Work ratio vs. pressure ratio. (*e*) Cycle temperatures vs. pressure ratio.

increases up to a certain pressure ratio. At any given pressure ratio, the work ratio falls rapidly with machine efficiency, but for any machine efficiency the work ratio improves with higher inlet gas temperatures to the turbine. Note the extremely low work ratios at low machine efficiencies.

To appreciate better the effects of machine efficiency and turbine inlet gas temperature on the four performance factors, Fig. 9-4 has been prepared. Variation of the maximum or optimum thermal efficiency at each machine efficiency with gas inlet temperature is shown in (*a*). Rising temperature improves thermal efficiency markedly. The air rates, work ratios, and pressure ratios corresponding to the thermal efficiencies of (*a*) are shown, respectively, in (*b*), (*c*), and (*d*). The curves show that air rates drop very rapidly with rising temperature, work ratios improve with temperature increase, and corresponding pressure ratios also rise with temperature.

Figure 9-4 shows that a simple-cycle gas turbine can achieve high efficiency only by two means: (1) improving machine efficiency and (2) raising turbine-inlet temperature. Naturally, pressure ratios must be properly selected to coincide with these factors. Designers are constantly striving to improve machine efficiencies which are, perhaps, the most critical factors in performance. For instance, look at the optimum efficiencies for 1000 F gas temperature in Fig. 9-4*a*. The following table lists the corresponding thermal and machine efficiencies and corresponding *increases* in thermal and machine efficiencies:

Machine efficiency, per cent	Thermal efficiency, per cent	Increment in machine efficiency, per cent	Increment in thermal efficiency, per cent
70	2.5		
		10	8.5
80	11		
		10	14.5
90	25.5		
		10	39
100	64.5		

As machine efficiencies are pushed higher in percentage points, the relative gain in thermal efficiency becomes greater. Machine efficiencies now are in the 85 to 90 per cent range for clean new equipment, and it becomes increasingly difficult to realize the next higher value. Friction and turbulence in fluid flow are difficult to minimize and impossible to eliminate.

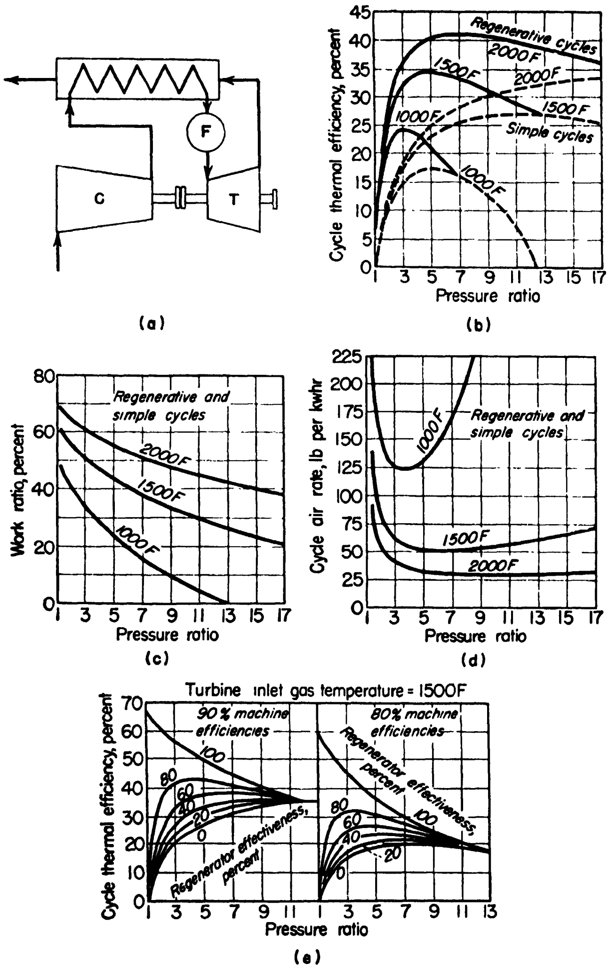


Fig. 9-5. Regenerative-cycle performance for 85 per cent machine efficiencies and 70 per cent regenerator effectiveness. (*b*) Thermal efficiency, (*c*) work ratio, (*d*) air rate, (*e*) thermal efficiency at different machine efficiencies and effectiveness.

the turbine, the exhaust temperature is higher than compressor outlet temperatures for pressure ratios less than 11.

Perhaps you see the light now! With exhaust temperatures higher than compressed-air temperatures entering the furnace, we can use the exhaust to preheat the air before burning. The hotter the air, the less fuel we have to burn to heat it and the higher will be the plant thermal efficiency. To do this we use a heat exchanger, called a "regenerator." The compressed air flows through the tubes and the exhaust gas around the tube exteriors. The circuit hookup appears in Fig. 9-5*a*. With sufficient heat-transfer surface the exhaust can be cooled down to the air temperature leaving the compressor, and the air can be heated to the gas temperature leaving the turbine. For this ultimate condition the regenerator is said to be 100 per cent effective. Regenerator effectiveness is defined as the ratio of actual air enthalpy rise through the regenerator to the maximum possible enthalpy rise for the given cycle conditions. Naturally the efficiency of the cycle depends on the regenerator effectiveness. Figure 9-5*b* compares the thermal efficiencies of regenerative cycles and simple cycles for 1000, 1500, and 2000 F inlet gas temperatures, 85 per cent machine efficiencies, 60 F inlet air temperature, and 70 per cent regenerator effectiveness. Regenerative-cycle optimum thermal efficiencies are considerably higher than simple-cycle ones and occur at lower pressure ratios. Air rates are essentially the same for regenerative and simple cycles as shown in (*d*). The same is true about the work ratios as shown in (*c*).

Variation of cycle thermal efficiency for the regenerative arrangement with machine efficiencies, regenerator effectiveness, and pressure ratio are shown in Fig. 9-5*e*. As regenerator effectiveness increases, the optimum thermal efficiency occurs at successively lower pressure ratios. The curves for zero effectiveness are those for the simple cycle.

Figure 9-6*a* shows how the optimum thermal efficiencies of regenerative and simple cycles compare for different inlet temperatures, 60 F compressor-inlet temperature, 85 per cent machine efficiencies, and 70 per cent regenerator effectiveness. As expected, the former operate at higher thermal efficiencies for all turbine-inlet temperatures. Air rates, work ratios, and pressure ratios corresponding to the efficiencies in (*a*) are given, respectively, in (*b*), (*c*), and (*d*). Though simple- and regenerative-cycle air rates are not exactly equal, they are so close that in the scale plotted that they may be represented by a single common curve. Regenerative-cycle work ratios are higher than those for the simple cycle because pressure ratios are lower.

Providing a regenerator means spending money for an extra piece of equipment, an economic study must be made for each particular application of this plant to determine whether the saving in fuel warrants the added investment in the regenerator.

Intercooling. Many are familiar with multistage air compressors using intercoolers. A common machine is the two-stage compressor in which air taken from atmosphere has its pressure partly raised in a

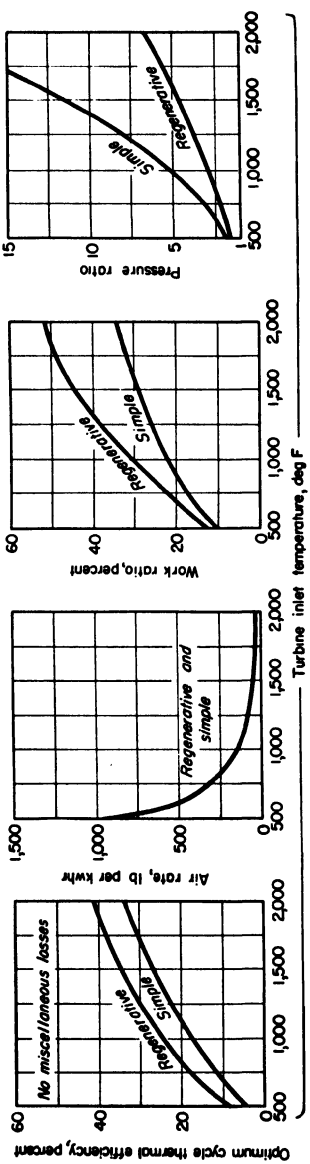


FIG. 9-6. Comparison of simple- and regenerative-cycle performance variations for 60 F compressor-inlet temperature, 85 per cent machine efficiencies, and 70 per cent regenerator effectiveness. Air rate, work ratio, and pressure ratio correspond to the optimum efficiencies shown in (*a*).

l-p cylinder, the air then flows through a heat exchanger, called an "intercooler," in which the air flows around tubes carrying cooling water. After usually being cooled to about its original temperature, the air then has its pressure raised to the final level in a h-p cylinder.

Intercooling the air reduces the amount of work needed to drive the compressor. Exactly the same hookup can be used with rotary compressors in gas-turbine plants as shown in Fig. 9-7*a*. The first circuit is for a simple cycle with a single-stage compressor, the second for a two-stage compressor with one intercooler, and the third for three-stage compression with two intercoolers. The curves of Fig. 9-7 are drawn for 60 F inlet air temperature, 85 per cent machine efficiencies, 1500 F turbine-inlet temperature. The thermal efficiency curves in (*b*) show that two-stage compression has an optimum efficiency about 2 per cent higher than that of the simple cycle. Three-stage intercooled compression gains less than 1 per cent over the two-stage. Curves in (*c*) show that the air rates improve for each added stage of compression, but with decreasing increment gains. The work ratio shows considerable improvement with intercooling. The improvement in this factor is perhaps of more practical significance than the small gain in thermal efficiency. With higher work ratios, a smaller turbine will be needed for given values of net plant output.

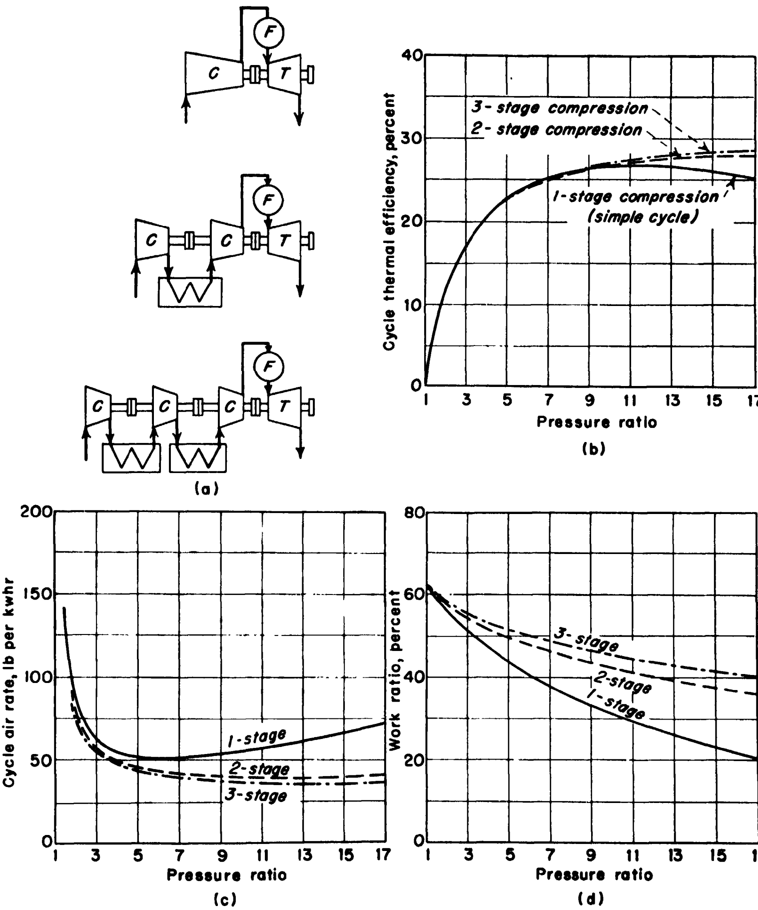


FIG. 9-7. Intercooled-cycle performance for 1500 F turbine-inlet temperature, 85 per cent machine efficiencies, 60 F compressor-inlet temperature. (*b*) Thermal efficiency, (*c*) air rate, (*d*) work ratio. Intercooling reduces air temperature to 60 F.

Reheating. Partly expanding the high-temperature gas in the turbine and reheating before finally expanding to atmospheric pressure will produce improvement in performance factors, as shown in Fig. 9-8. In the figure one-stage heating refers to the simple cycle, two-stage heating is with one stage of reheating, three-stage heating has two stages of reheating, as shown in (*a*). Thermal efficiency curves in (*b*) show only about ½ per cent improvement in optimum values for added stages of heating.

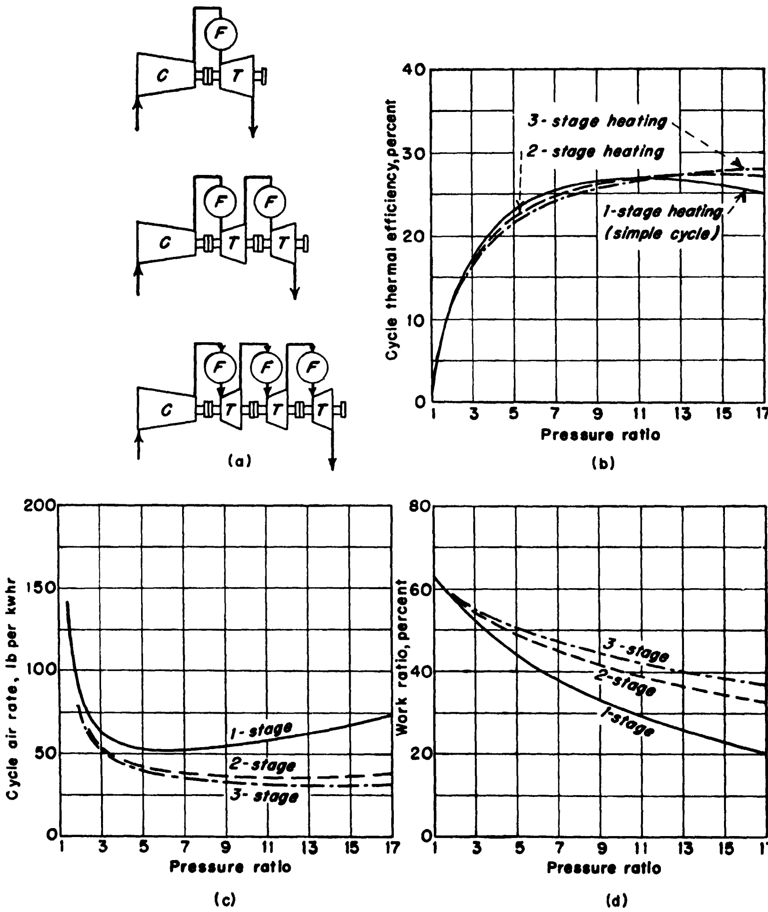


FIG. 9-8. Reheating-cycle performance for 1500 F turbine-inlet temperature, 85 per cent machine efficiencies, 60 F compressor-inlet temperature. (*b*) Thermal efficiency, (*c*) air rate, (*d*) work ratio. Reheating raises the gas temperature to 1500 F.

Larger gains are realized in air rate and work ratio as for intercooling with compression.

Combining Heat-conserving Equipment. Knowing what regeneration, intercooling, and reheating can do individually, it is evident that their advantages can be combined to achieve an even greater improvement than each can develop by itself. In Fig. 9-9*a* several possible arrangements are shown. At the left are combinations of intercooling and reheating, and on the right are combinations of all three, regeneration, intercooling, and reheating. Curves in (*b*) show that the thermal efficiencies for the cycles without the regenerator increase noticeably (at higher pressure ratios) as compression and reheating stages are added

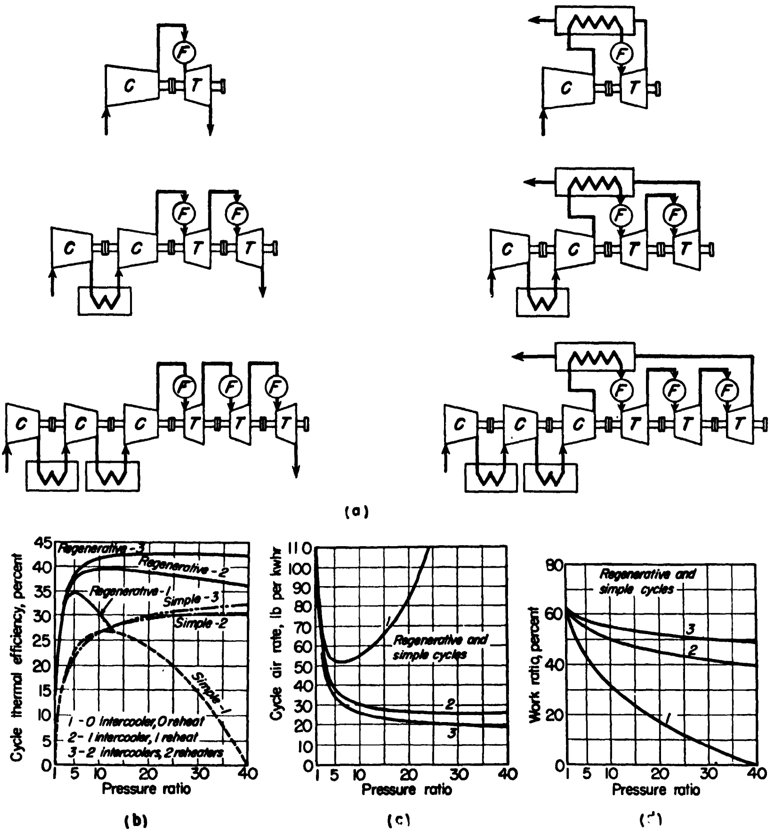


FIG. 9-9. Combined intercooling-reheating and intercooling-reheating-regenerative cycle performances for 1500 F turbine-inlet temperature, 85 per cent machine efficiencies, 70 per cent regenerator effectiveness, 60 F compressor-inlet temperature. (*b*) Thermal efficiency, (*c*) air rate, (*d*) work ratio.

simultaneously. The largest gains however are realized (at relatively lower pressure ratios) when a regenerator functions in the cycle.

These curves are drawn for 1500 F turbine-inlet temperature, 60 F compressor-inlet temperature, 85 per cent machine efficiencies, and a 70 per cent regenerator effectiveness. Curves in Fig. 9-9*c* show the improvement in air rate as stages of intercooling and reheating are added simultaneously. A regenerator of constant effectiveness has no influence, theoretically, on cycle air rate.

Curves in (*d*) show that a marked improvement is realized in work ratios by simultaneous intercooling and reheating. Again the regenerator has no appreciable effect on this factor.

In this brief discussion of gas-turbine cycles, we can see now that there is a wide choice of arrangement of equipment. To make the right selection takes a thorough study of many conditions. As the hookup becomes more complicated, it becomes more expensive to build; however, the increased thermal efficiencies mean a saving in fuel cost. So it becomes necessary to find out whether the increased investment cost can be profitably carried by the saving in fuel expense.

Compressor-inlet Temperature. Before we leave the subject of cycle performances, let's examine the effect of air-temperature variation entering the compressor. Figure 9-10*a* shows that, for both simple and regenerative open cycles at various turbine-inlet gas temperatures and appropriate pressure ratios, the thermal efficiency increases with drop in incoming air temperature. A gas-turbine plant operating in the cold north will perform much more efficiently than if it were moved to the tropics. In the milder latitudes, a gas turbine will be more efficient in the winter than in the summer.

The improvement in thermal efficiency with falling air temperature is accompanied by decreasing air rates as shown in (*b*) and by rising work ratio as shown in (*c*). Perhaps the most startling aspect of air temperature is its influence on the capacity of a gas-turbine plant, as shown in Fig. 9-10*d*. Any machine designed for a certain capacity at a given air temperature will gain in capacity with lower temperatures and vice versa. These curves are based on constant compression efficiency for all temperatures. Since any compressor usually falls off in efficiency when operating at other than design conditions, capacity changes will be less than shown and curves for an actual machine will not be quite so steep, but the effect will be quite marked. Air rate and the changing air density with temperature both influence the total gas-turbine plant capacity.

All the curves shown in Figs. 9-3 to 9-10 are computed on the air standard; *i.e.*, it was assumed that all air and combustion gas passing through the units acted the same as air. This is another reason why actual machine performances are lower than these more easily figured air-standard curves.

Practical Turbine Performance. Having studied the theoretical variation in gas-turbine plants, let's look at some practical design layouts.

Figure 9-11*a* shows an arrangement of a simple-cycle gas-turbine plant that can be built for capacities up to 10,000 kw. The curves of Fig. 9-11*b* show its performance over the load range for constant-speed operation to drive an a-c generator. The unit is designed to operate with 1100 F gas entering the turbine at full load.

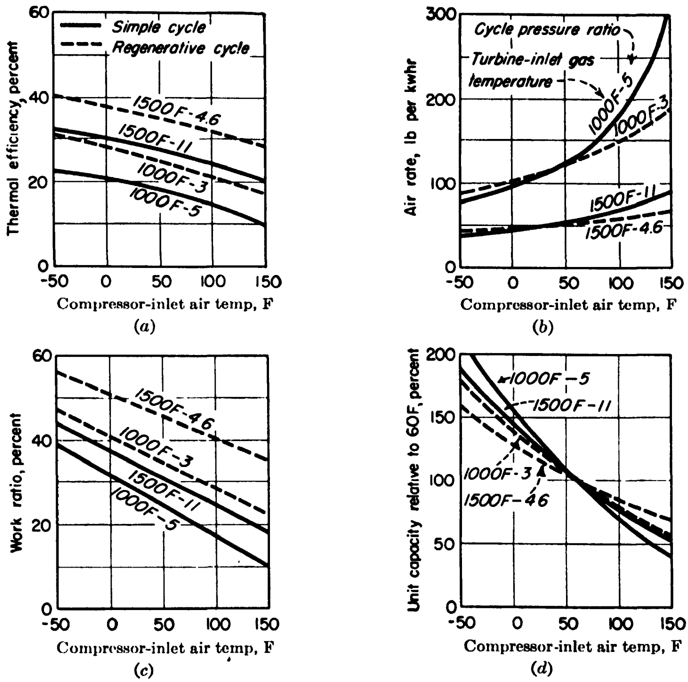
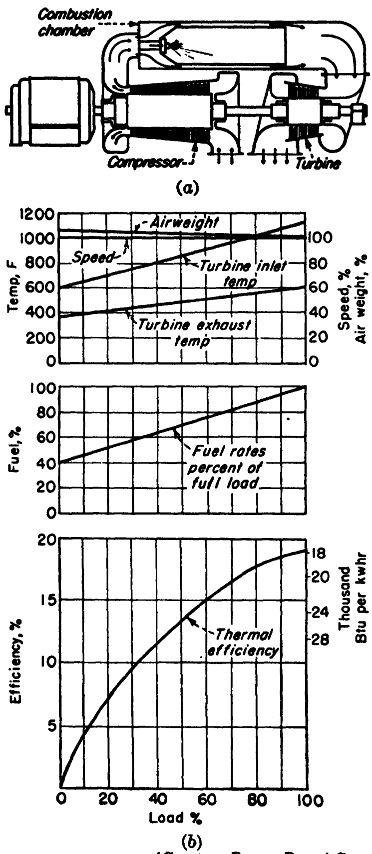


FIG. 9-10. Performance variation of simple- and regenerative-cycle plants with change in compressor-inlet air temperature.

How is the load controlled? You may have noticed that no valves are supplied anywhere in the air circuit; so air flow cannot be controlled that way. Throttling control is undesirable in gas turbines because pressure differences are generally small, and any additional drop caused by a valve introduces a tremendous energy loss. Since the load demands constant speed, the compressor will also run at that speed; hence the rate of air flow will be substantially constant at all loads.

The only recourse open is to vary the available energy per pound of air entering the turbine by varying the temperature. This is done by changing the amount of fuel burned in the combustion chamber. It should be obvious that the fuel input varies directly with load. The second chart in Fig. 9-11*b* shows that fuel input at zero load is 40 per cent of full-load input and varies directly with load between these limits. At zero load, turbine-inlet temperature drops to 600 F. Because of the lowered temperatures at less than full load, the thermal efficiency drops off quite rapidly. The speed governor on this type of unit (see Fig. 7-9) controls the fuel feed to the combustion chamber. The arrangement is relatively simple.



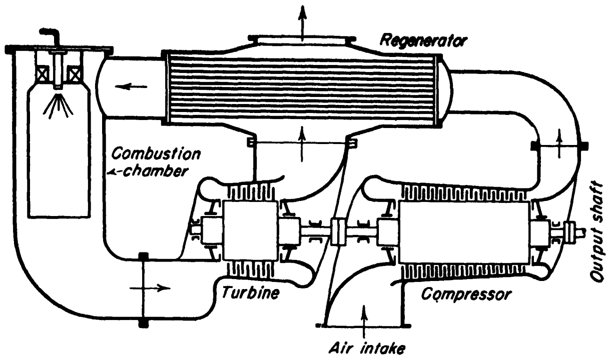
(*Courtesy Brown Boveri Corp.*)

FIG. 9-11. Actual performance curves of a simple-cycle gas-turbine plant as affected by load variation.

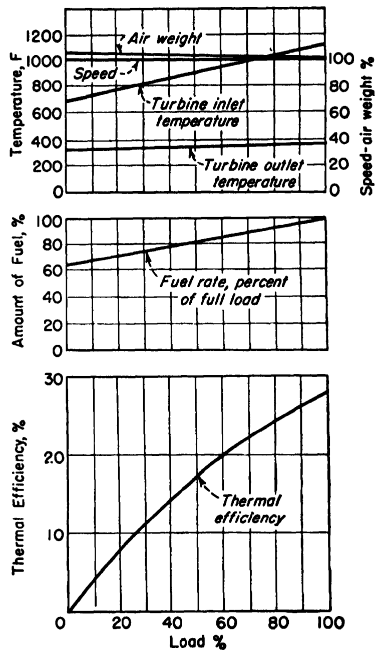
Figure 9-12*a* and *b* show a regenerative-cycle arrangement and its performance. While full-load thermal efficiency is higher because of the heat salvaged in the regenerator, the single-shaft arrangement requires dropping temperature at part load, and again we find part-load efficiency dropping rapidly. Notice that the no-load fuel input is about 65 per cent of the full-load rate. Turbine outlet temperature is that of the exhaust gas leaving the regenerator.

If a great deal of part-load operation is required, the single-shaft machines are not particularly suitable. The maximum available energy should be maintained at all loads by keeping turbine-inlet temperature up, and the air-gas flow should be varied with load. This can be done as shown in Fig. 9-13*a* by splitting the turbine into two separate units. One drives the load, or generator, at constant speed, and the other the compressor at variable speed.

The performance curves show that the turbine-inlet temperatures stay practically constant and that compressor-set speed varies from 40 per cent at no load to 100 per cent at full load causing a change in pressure ratio of comparable magnitude. The no-load fuel rate is only about



(a)



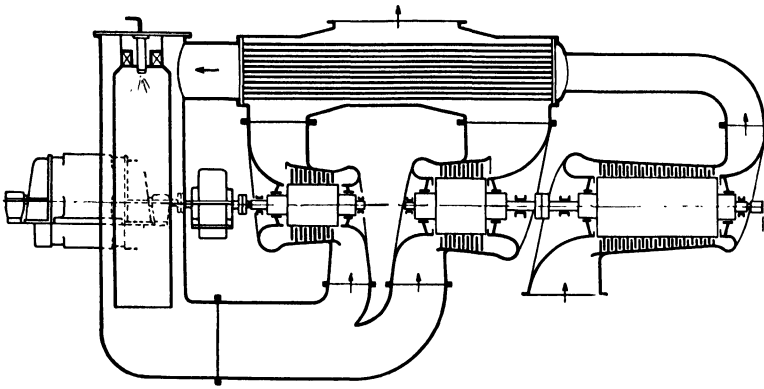
(b)

(Courtesy Brown Boveri Corp.)

FIG. 9-12. Actual performance curves of a regenerative-cycle gas-turbine plant as affected by load variation.

15 per cent of the full-load input with the result that thermal efficiency varies only by about 3 per cent from half to full load.

Let's trace through a sequence of events to see how the load-control system works. Fuel input is again controlled by the speed governor on the generator set. Let's assume that the unit is operating at full load and that the load drops somewhat. Falling generator load will momentarily speed up the load set and cause the governor to cut down on the fuel feed to the combustion chamber. Decreased fuel flow to the combustion chamber with constant air flow from the compressor will cause the turbine-inlet temperatures to drop momentarily. Falling temperature reduces available energy, and both turbines will then drop somewhat in speed. Falling compressor-set speed reduces the air flow until the original turbine-inlet temperature is approximately restored and the load-turbine speed also. The balance



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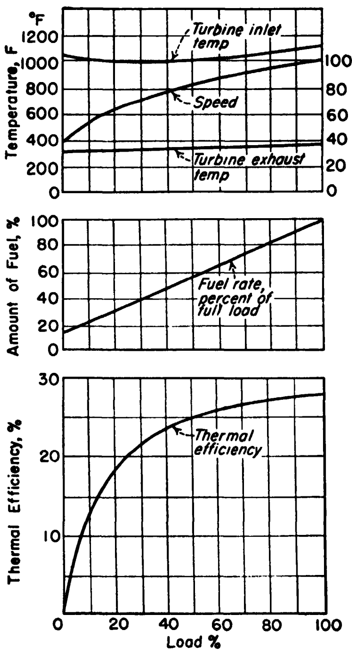

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(*Courtesy Brown Boveri Corp.*)

FIG. 9-13. Actual performance curves of a two-shaft regenerative-cycle gas-turbine plant as affected by load variation.

point of all these influences set off by the speed-control governor on the generator set depends on the design constants. As the curves show, the inlet temperature falls slightly from 100 to 50 per cent load, but the maintenance of a generally high level of temperature improves part-load efficiency remarkably, despite falling pressure ratio.

Another two-shaft arrangement using two combustion chambers is shown in Fig. 9-14. This hookup allows somewhat finer control of speed regulation, but it has been applied only on an experimental set for laboratory investigations.

Figure 9-15 shows the hookup of a two-shaft unit using three stages of compression with intercooling, one stage of reheat, and a regenerator. The h-p turbine drives compressor No. 3 and the load, and the l-p turbine the first two compressors. The h-p turbine runs at constant speed for all loads, and the l-p turbine speed varies with load. The curves show how the turbine-inlet temperatures are varied under control of the h-p and l-p combustion chambers. Thermal efficiency is excellent and relatively flat down to half load.

To increase load from zero to full, the l-p turbine accelerates from 40 to 100 per cent speed in about 20 sec minimum time. The water to intercoolers totals about ¼ gpm per kw (about 20 per cent of equivalent steam-plant needs).

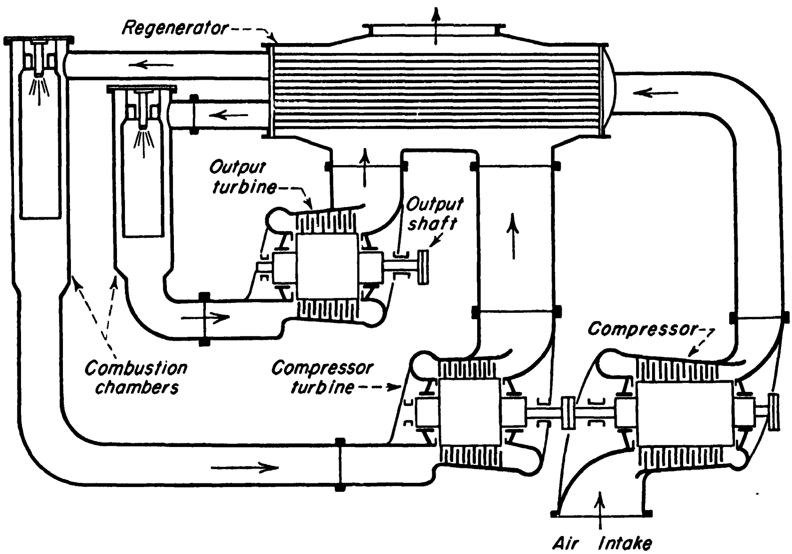
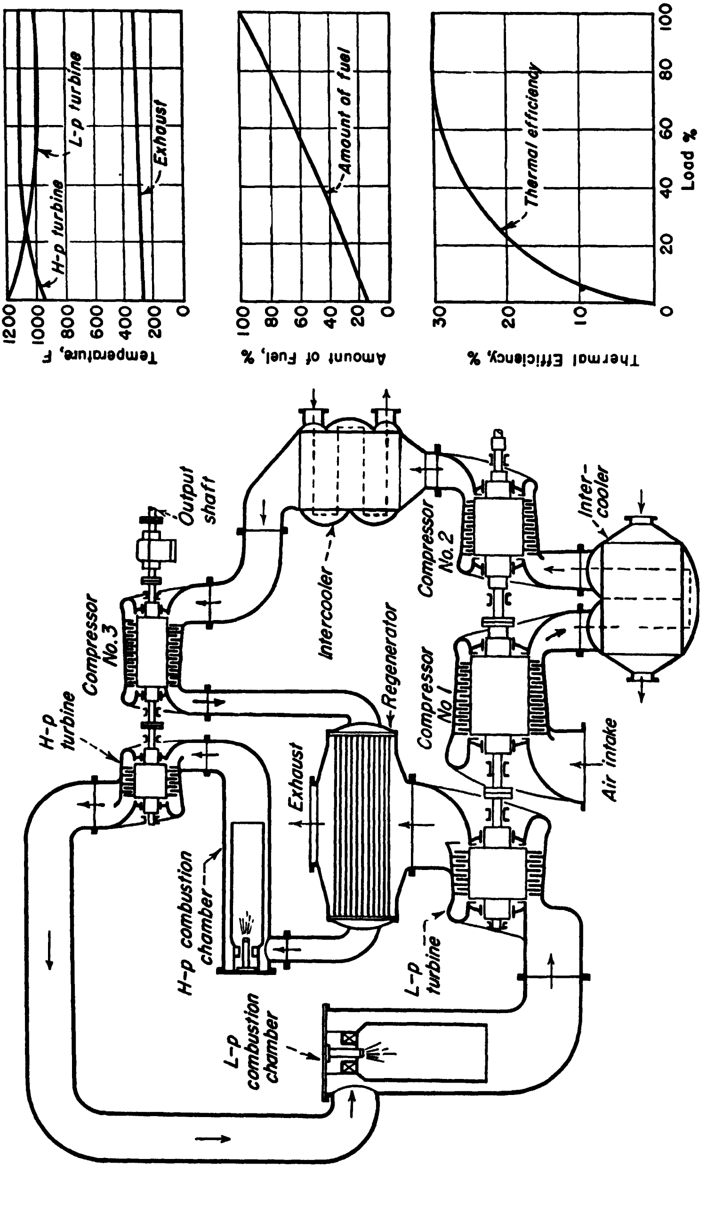


FIG. 9-14. Experimental regenerative-cycle arrangement with two shafts and parallel combustion chambers.

The largest gas-turbine unit that has been built to date, is the 27,000-kw unit for installation at Beznau, Switzerland, by the Brown Boveri Corp. Its arrangement is shown in Fig. 9-16, using two shafts, intercooling and reheating. To develop such large output a very high rate of air flow must be handled. Accordingly the unit features two compressors in parallel for the first stage of compression, and a double-flow turbine for the l-p element. This unit will operate at about 34 per cent full-load thermal efficiency with an air inlet temperature of 41 F and a turbine-inlet temperature of 1100 F.

Practical Plants. Except for the specialized application to the Houdry refining process, which produces high-test gasoline, application of gas turbines for stationary power production is brand new in the United States. Let's look at some practical gas-turbine units that are offered on a commercial basis.

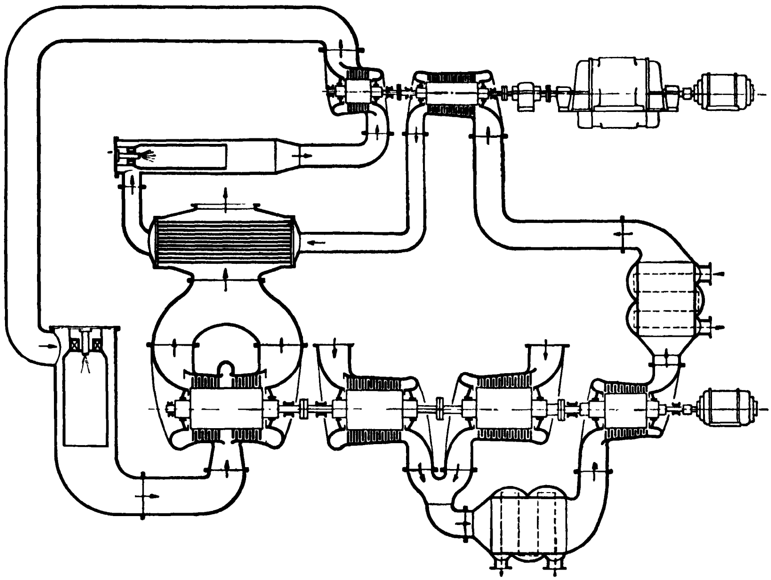


(Courtesy Brown Boveri Corp.)

FIG. 9-15. Performance of two-shaft, intercooled, reheat, regenerative cycle.

General Electric Co. Figure 9-17 shows a 1400-F 3,500-kw gas turbine on test at the factory. This open simple-cycle unit will be shortly entering into commercial stationary operation and is now undergoing tests in a locomotive at this writing. Figure 9-18 shows a cross section through the unit.

When running at 6700 rpm with 1300 F inlet temperature, the plant produces 3,580 kw for its design conditions of 80 F inlet air and com-



(*Courtesy Brown Boveri Corp.*)
FIG. 9-16. Arrangement of 27,000-kw plant built by Brown Boveri, using double-flow elements on l-p compressor and l-p turbine.

pressor-inlet pressure corresponding to 1500 ft elevation. Over-all thermal efficiency at full load, based on higher heating value is about 16 per cent. At 1400 F design temperature, the unit produces 4,470 kw at about 17 per cent thermal efficiency. The 15-stage axial compressor pumps about 70,000 cu ft per min of air through the unit at about a pressure ratio of 6.

Separate wheels bolted together make up the compressor rotor. Small gaps between them at the rim eliminate as far as possible any rotor bending effects caused by rubbing or local heating. The compressor casing is made up of four sections bolted together at horizontal and vertical joints. An extraction point provides cooling air for the turbine wheel.

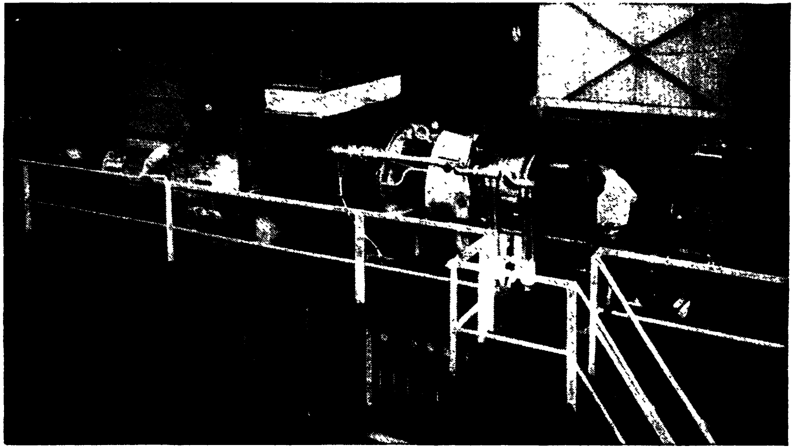


FIG. 9-17. 3,500-kw simple-cycle gas-turbine plant built by General Electric Co. on test stand. Left to right; generators, turbine-exhaust hood, turbine, combustion chambers, compressor, compressor-inlet hood, auxiliaries.

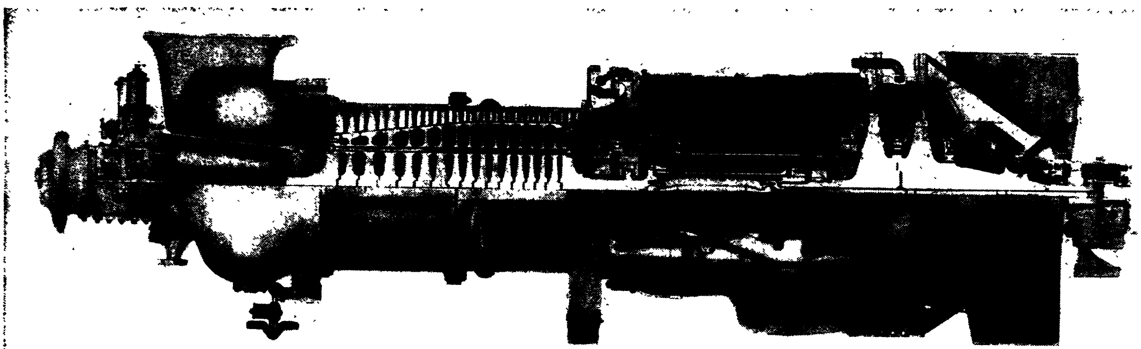


FIG. 9-18. Cross section of General Electric Co. 3,500-kw gas-turbine unit. Left to right; auxiliaries on shaft end, compressor inlet, 15-stage axial-flow compressor, six combustion chambers, two-stage impulse turbine, turbine-exhaust hood.

Six all-metal combustion chambers have an outer casing of carbon steel and inner liners of stainless material. Air enters the inner liner through annular slits between sections, through radial holes, and through louvers. Air-atomizing fuel nozzles obtain their air from a small compressor driven by the auxiliary assembly at the unit's compressor end. Spark plugs in two of the six chambers provide initial ignition. The others fire through cross-ignition tubes connecting the chambers.

The turbine has only two stages because (1) design is simplified, (2) the unit becomes more compact, (3) the first-stage bucket temperature is

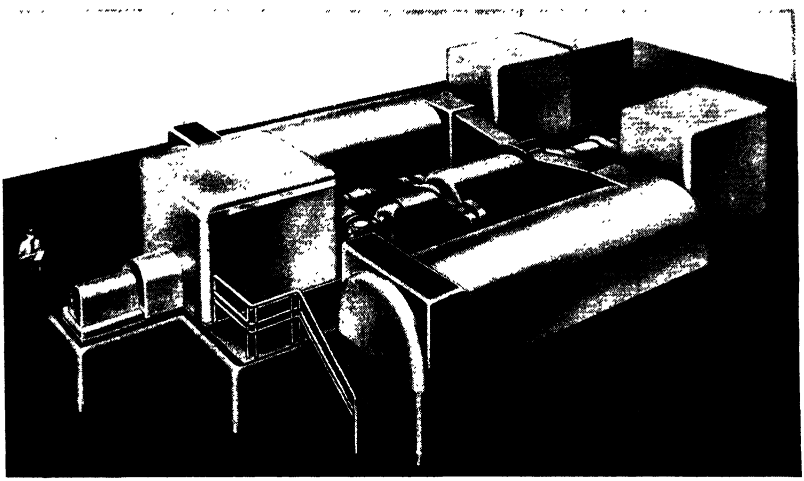


FIG. 9-19. Artist's conception of General Electric Co. 5,000-kw gas-turbine plant with intercoolers and regenerators. For element identification, see Fig. 9-20.

considerably lower than for a multistage design. The calculated bucket temperature for this arrangement is only 1160 F, whereas for an eight reaction-stage design the bucket temperature would be about 1360 F.

Both first- and second-stage wheels are of similar construction and bolted together. They are cooled by a small flow of air bled into the space between them and their adjacent stationary walls. Pipes carry this air from the compressor across the gas path to the packing. The gas turbine discharges to atmosphere through a diffusing exhaust hood. Ribs between the aft bearing and the turbine casing pass through hollow streamlined struts in the exhaust hood, which shield the ribs from the hot gas. A small fan on the turbine shaft draws cooling air over the turbine stator and down through hollow struts.

Auxiliaries include a lubricating system having a tank, two oil coolers, a motor-driven auxiliary lubricating pump, and one main pump driven directly from the main shaft, oil filter, and pressure switch assembly. Other items are fuel-oil pump, atomizing-air compressor, fuel regulator, overspeed governor, tachometer generator, starting motor, low-speed turning motor, ignition transformers, supervisory instruments, and electrical controls for the generator driven by the gas turbine.

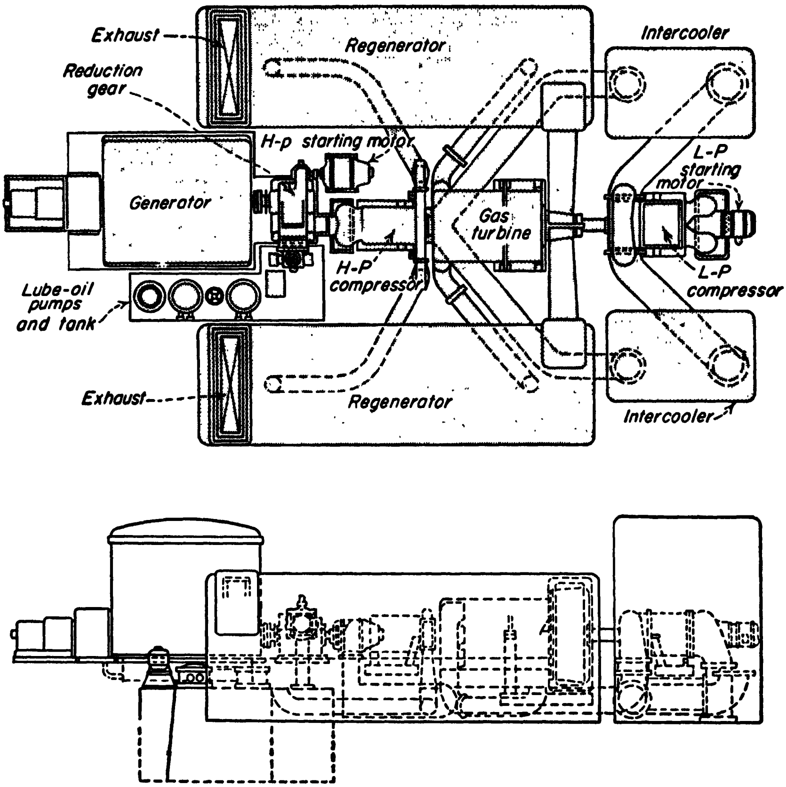


FIG. 9-20. Layout of General Electric Co. 5,000-kw high-efficiency gas-turbine plant to operate at 1500 F turbine-inlet temperature.

Figures 9-19 and 9-20 show another gas-turbine unit that will be going into commercial operation within a year or two of this writing. This 5,000-kw plant is designed for 1500 F inlet temperature, two shafts, intercooling and regeneration. The expected over-all thermal efficiency is 26.5 per cent. From the standpoint of size, the largest units of equipment are the regenerator and intercooler. Each is divided into two elements connected in parallel. As the perspective view and plans show, this allows a symmetrical arrangement laid out on a single floor level.

The plant has a h-p turbine driving the h-p compressor and generator at constant speed; the l-p turbine drives the l-p compressor at variable speed. Figure 9-21 shows the cross section of the rotating elements. At the left is the 11-stage h-p compressor operating at 3.1 pressure ratio with a speed of 8,694 rpm. Next are the six all-metal combustion chambers discharging the heated air into the two-stage h-p turbine, which is overhung from its bearing.

Exhaust gas from the h-p turbine enters the single-stage l-p turbine

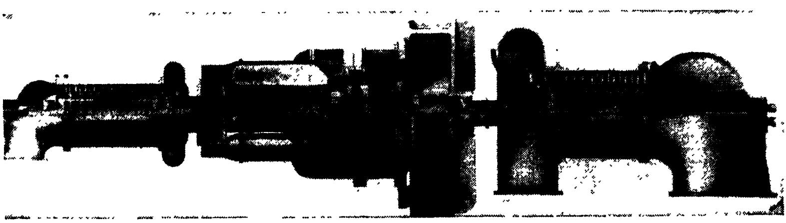


FIG. 9-21. Cross section of General Electric Co. 5,000-kw gas turbine and compressor. Left to right: h-p compressor, six combustion chambers, two-stage h-p turbine exhausting into single-stage l-p turbine, turbine-exhaust hood, l-p compressor.

directly. This turbine wheel is also overhung from its bearing to realize the closely stacked arrangement in what virtually is a single turbine casing. The last element at the right is the nine-stage l-p compressor driven by the l-p turbine. At full load this compressor develops a pressure ratio of 3.0 with a speed of 7,200 rpm.

About 30 gpm of cooling water is used to cool the turbine stators and rotors. Water flows through passages in the main circumferential rings and in the l-p turbine rotor-support struts. The rotors are cooled by radiant heat transfer to stationary cooling pads adjacent to the turbine wheels. Solid ring construction of the turbine stator minimizes distortion. All h-p turbine stator elements, except the l-p turbine nozzles, are segmented to allow differential expansion circumferentially between the hot austenitic and cool carbon-steel parts. The exhaust diffuser in the form of an airfoil grid gives a high degree of pressure recovery in a short axial distance. It also helps turn the exhaust gas radially outward into the exhaust ducts.

Features of the compressor construction are the movable guide vanes for the air entering the first stage of the compressors. By adjusting the angle for air-flow rate and rotor speed, appreciable improvement of plant efficiency is realized over the part-load range.

Plant output depends directly on fuel-rate input as controlled by a fuel regulator. The regulator performs the following duties:

1. *Speed governing:* Electrical signals from the tachometer generator on the load shaft are translated into control oil pressure determining the stroke of the fuel pump.
2. *Turbine temperature limitation:* Prevents turbine-inlet temperature from exceeding safe level. Turbine-exhaust temperature is also controlled to prevent excessive heating of the rear end.
3. *Rate of temperature change:* Limits the temperature-change rate during sudden load variations.

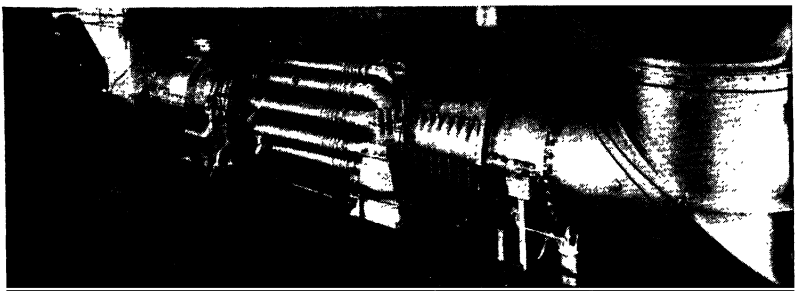


FIG. 9-22. Westinghouse Electric Corp. 2,000-hp 1350 F gas-turbine plant on test bed. Left to right: generators, reduction gear, compressor-inlet hood, compressor, 12 combustion chambers, turbine, turbine-exhaust diffuser, exhaust hood.

4. *Low-speed fuel control:* Regulates fuel during starting when the speed governor is inoperative.
5. *Constant-load control:* An adjustable fuel stop on the regulator will maintain constant output on the unit regardless of system load or frequency changes.
6. *Compressor guide-vane adjustment:* Sets these vanes for best efficiency.

A 100-hp motor connects through an overrunning clutch to an auxiliary drive shaft on the main reduction gear of the h-p turbine shaft. The starting motor supplies starting torque up to a shaft speed of 5,650 rpm for the turbine. At this speed the motor disconnects by the overrunning clutch, and the combustors being ignited can then accelerate the turbine to full speed.

While the motor on the h-p shaft will probably start the entire unit, starting time will be materially reduced by a 40-hp motor connected to the l-p shaft, also through an overrunning clutch. If tests show that the main starting motor does not provide breakaway torque, a turning gear will be provided.

Westinghouse Electric Corp. Figure 9-22 shows a 2,000-hp simple open-cycle gas-turbine unit designed for operation with 1350 F turbine-inlet temperature. From left to right are d-c generators, compressor air-inlet hood, compressor, 12 combustion chambers. Surrounded by the combustion chambers is a hollow jackshaft connecting the compressor and turbine. The turbine casing, identified by the long vertical bolts, is followed by the turbine-exhaust diffuser and the exhaust stack. The turbine develops about 6,000 hp, of which 4,000 is absorbed by the

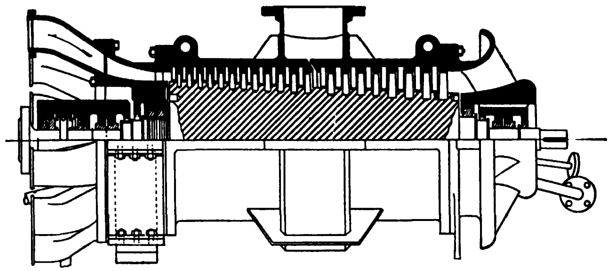


FIG. 9-23. Cross section of Westinghouse compressor of 2,000-hp gas-turbine plant. Inlet at the right, exhaust to combustion chamber inlets at the left.

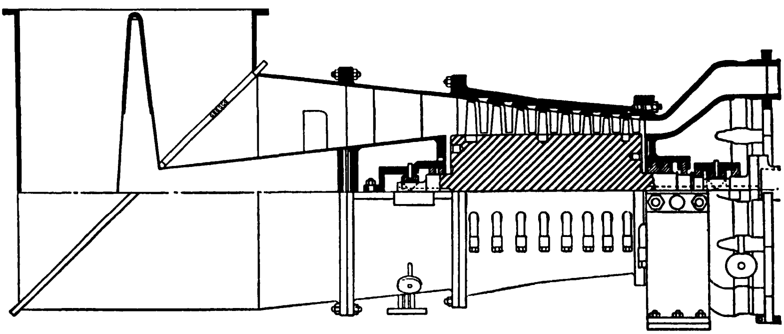


FIG. 9-24. Cross section of Westinghouse gas turbine for 2,000-hp plant. Inlet at the right; eight-stage turbine, exhaust diffuser, exhaust hood at the left.

compressor and the remainder drives the load. Full-load speed equals 9,200 rpm. The unit was initially designed for locomotive application but can be used equally as well for stationary-power generation.

The axial flow compressor (Fig. 9-23) handles 25,000 cfm of air at a pressure ratio of 5. It contains 20 stages of nonsymmetric blading with nearly all pressure rise occurring in the moving blading. Labyrinth-type air seals enclose each end of the compressor shaft. By solid connection between turbine and compressor the greater part of the thrust is balanced, eliminating the need for a dummy. A standard-type thrust bearing takes up any unbalance remaining.

The turbine (Fig. 9-24) has eight stages designed for equal heat drop through stationary and moving blades at the mean diameter. Labyrinth shaft seals and pressure-lubricated sleeve-type journal bearings are used. The turbine is designed to withstand instantaneous temperature changes from 700 to 1350 F. Mechanical hinges position the turbine and allow for rapid thermal changes while still maintaining proper alignment with the rest of the equipment mounted on the bedplate.

The 12 cell-type combustors (Fig. 9-25) have casings made of carbon-steel pipe with bellows-type joints welded to one end and tapered flanges welded to both ends. The flame tube sections made of Inconel are spot-welded together. Air-atomizing fuel nozzles maintain high burner

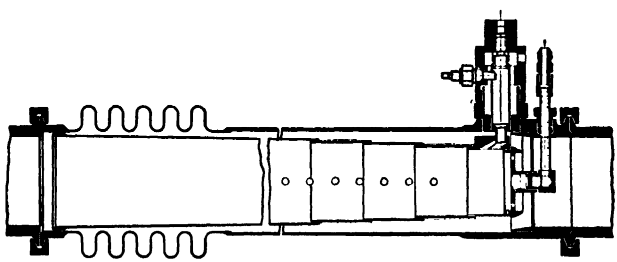


FIG. 9-25. Cross section of one combustion chamber of Westinghouse 2,000-hp gas-turbine plant. Inlet at the right; fuel nozzle, flame tube concentric in shell, discharge to turbine at the left.

efficiency over a wide load range when burning bunker C fuel oil. Heat release rates are about 6 million Btu per cu ft per hr at full load.

An acetylene igniter starts burning of the fuel during start-ups. A primary combustion zone in the flame tube admits only part of the air flow and creates flow reversal. In other parts of the combustor, secondary air surrounds the flame tube and mixes with the combustion gas after flowing through the axial holes in the flame tube.

Starting tests show that 80-kw cranking power brings the unit up to speed in 1 min and 50-kw in about 1.5 min.

Allis-Chalmers Manufacturing Co. At this writing the latest design offered by this company is shown in Fig. 9-26. This single-shaft regenerative-cycle unit develops 7,500 kw at 3,600 rpm with 1300 F turbine-inlet temperature, pressure ratio of 5, compressor-inlet temperature of 80 F, and pressure corresponding to 1,000-ft altitude. Expected over-all efficiencies based on fuel higher heating value with a 50 per cent effective regenerator is 20.5 per cent for oil firing and 19.8 per cent for gas firing.

The six-stage reaction turbine develops 24,000 kw at its coupling under full-load conditions. Six equal-diameter disk forgings welded together form the blade-carrying portion of the spindle. Stub ends welded to each end of the disk assembly form the bearing and coupling ends of the spindle. Serrated axial grooves across the circumferential faces of each disk hold the roots of the turbine blades.

Horizontally split rings carry the stationary blades in grooves. Radial dowels hold the cylinder rings concentric with the shaft and allow differential expansion without disturbing the relative position of rings and spindle.

All turbine blades are warped and tapered to a considerable degree. No shrouds are used. Sharpening the blade tips reduces the effect of accidental blade rubs.

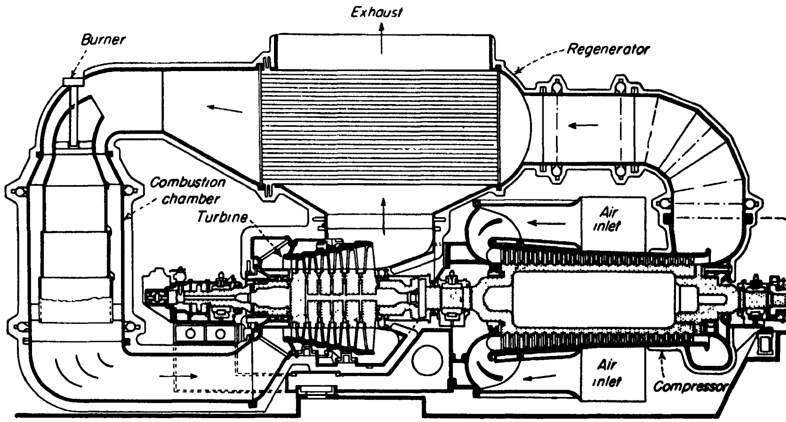


FIG. 9-26. 7,500-kw regenerative-cycle gas-turbine plant for operation at 1300 F as proposed by Allis-Chalmers Manufacturing Co. Single combustion chamber at the left, six-stage reaction turbine driving 20-stage axial-flow compressor, regenerator above the turbine exhaust.

Figure 9-27 shows the gland arrangement at the h-p end of the turbine. The gland performs three functions: (1) prevents the hot gas from escaping between spindle and casing, (2) cools the highest temperature portion of the spindle, (3) avoids high metal temperatures in the bearing. Air from the compressor discharge (at higher pressure than turbine-inlet gas because of regenerator and combustor pressure drops) enters at A, a belt around the glands as shown at B. The air then flows through the circumferential opening C after which part enters the turbine gas path through the labyrinth under the first-stage stationary blading. Cooling of this part of the spindle by the air will extend the life of the high-temperature parts. The remainder of the air flowing through C discharges through the gland labyrinth packing along the surface of the turbine spindle and out through D to the suction of a small exhauster fan.

The l-p labyrinth gland is of similar construction, but only sufficient air flow from the compressor outlet to prevent hot gas from escaping into the turbine room and against the l-p bearing is needed. A Kingsbury thrust bearing takes up any unbalance in the shaft.

The axial-flow compressor located between turbine and generator allows convenient arrangement of the regenerator above the unit. Com-

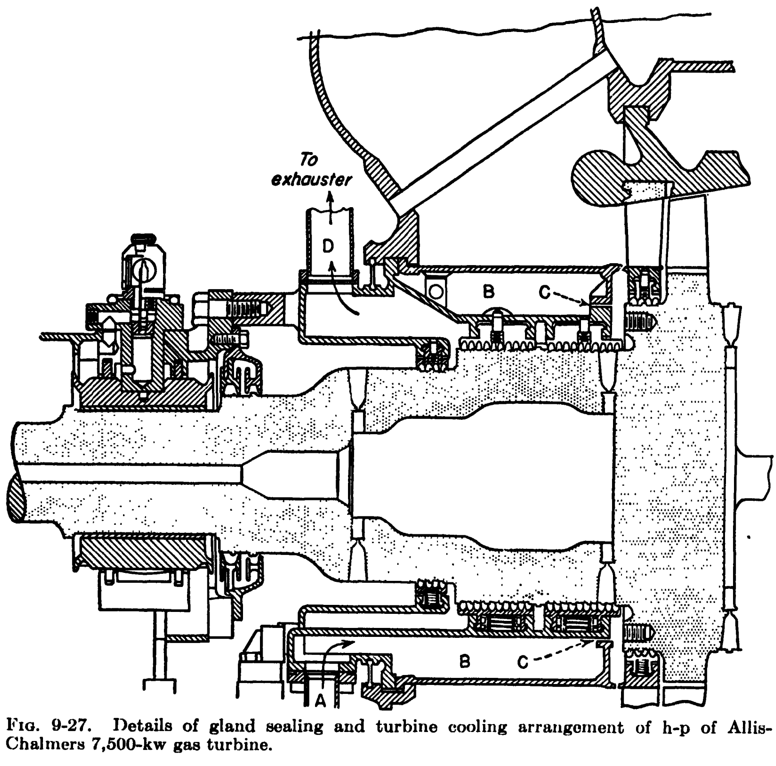


FIG. 9-27. Details of gland sealing and turbine cooling arrangement of h-p of Allis-Chalmers 7,500-kw gas turbine.

pression efficiency is expected to exceed 85 per cent. A single forging makes up the rotor cylinder and integral stub end at the l-p end. The h-p stub end bolts to the rotor body. The rotor body holds the moving blades in circumferential grooves. Both moving and stationary blades of the 20-stage compressor have integral spacers and may be precision cast with accurately machined roots for fitting in the grooves.

The unit will be fitted with all needed accessories such as lubrication system, speed governor, overspeed governor, turning gear, and supervisory instruments. In the regenerator the h-p air flows through the tubes on its way from the compressor to the furnace, while the exhaust gas from the turbine passes over the tubes on its way up to the stack. The tubes are rolled into each tube sheet at the ends. Differential expansion between tubes and shell is taken up by a steel expansion joint at one end of the shell.

The combustion chamber consists of a single unit with an inner firing tube made up of sections so that the excess air can mix with the combustion products by entering through annular slots. Turning vanes at the combustor exit help reduce friction and turbulence losses in the hot gas flowing to the turbine.

De Laval Steam Turbine Co. This company has under construction a 3,000-hp unit shown in Fig. 9-28. The full-load output will be developed with a turbine-inlet temperature of 1300 F, pressure ratio of 6, 68-F inlet air temperature, and 14.7-psia atmospheric pressure. Starting at the left in the figure are the auxiliary housing, compressor inlet, three-stage mixed-flow compressor, eight combustors, five-stage h-p turbine, and three-stage l-p turbine. All components except the last constitute a power-gas generator needed to produce the hot pressure gas which runs the l-p power-output turbine.

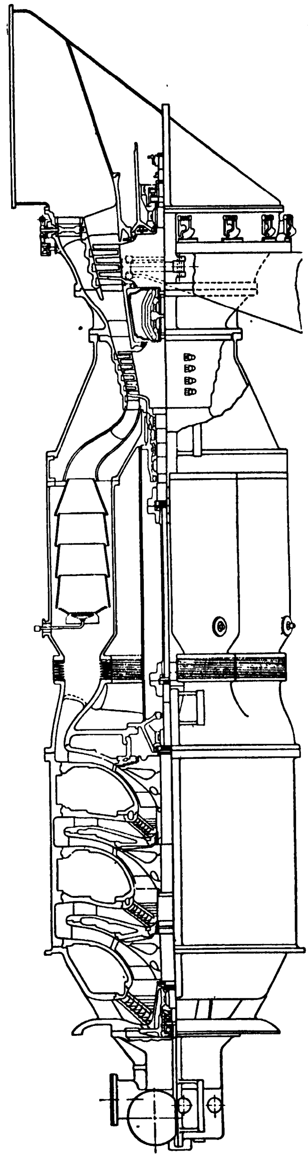


FIG. 9-28. Cross section of 3,000-hp gas-turbine plant under construction by De Laval Steam Turbine Co. Left to right: auxiliaries, three-stage mixed-flow compressor, eight combustion chambers, five-stage reaction turbine exhausting into three-stage l-p reaction turbine driving load on a separate shaft. L-p turbine surrounded by annular by-pass, with by-pass valves at the right discharging into the turbine-exhaust hood.

The three-stage compressor delivers 44,500 cfm of air at 7,900 rpm. Each of the three impellers is made from forgings so that the machined vanes and hubs form an integral piece. Air enters the compressor radially and turns axially to enter the first impeller. The impeller discharges the air in a generally radial direction and causes it to enter a vaned diffuser which directs it radially outward. Following the diffusion, the air flows inward radially to repeat the process in the second and third impellers. Labyrinth seals are used throughout. Sealing air

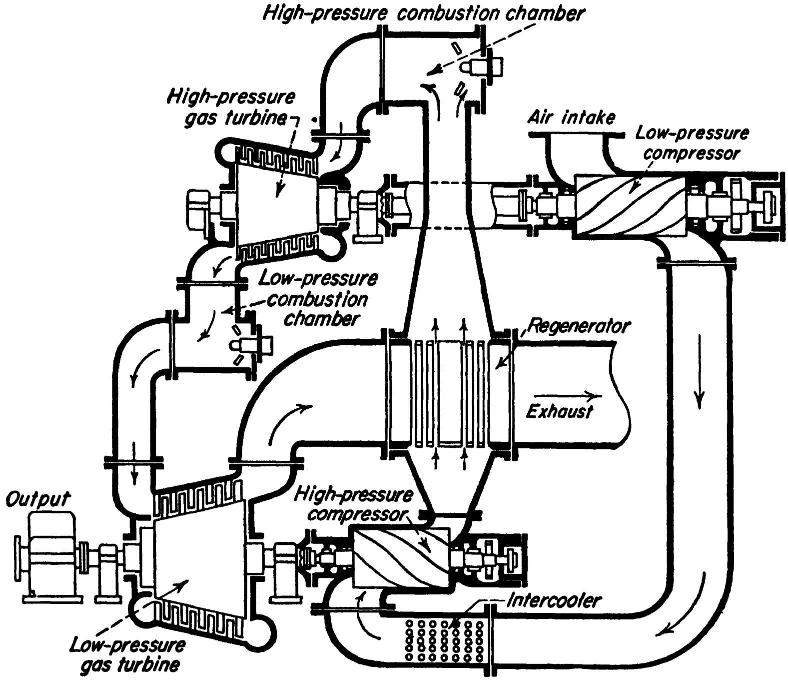


FIG. 9-29. Flow diagram of Elliott 3,000-hp gas-turbine plant. Reaction turbines drive Lysholm compressors and load. Cycle has intercooling, reheating, and regeneration; uses elbow combustion chambers fired with diesel fuel.

supplied to the labyrinths between the first-stage impeller and the auxiliary housing prevents lubricating oil from being drawn into the main air stream.

The h-p five-stage reaction turbine drives the compressor and auxiliaries, and may supply a small amount of power output above the main output. Blades are attached to the rotor by a fir-tree type of shank fitted into slots machined axially in the rotor. Two sleeve-type journal bearings carry the rotor, and the compressor thrust bearing also positions the turbine rotor. Exhaust from the h-p turbine enters the l-p turbine directly as shown in the figure.

The three-stage l-p power-output turbine is supported on its own two bearings and is positioned by a Kingsbury thrust bearing. It develops 3,000 hp at 4,200 rpm with the compressor set operating at design conditions and the by-pass valves fully closed. To increase operating flexibility at no-load conditions, especially for transportation applications, a

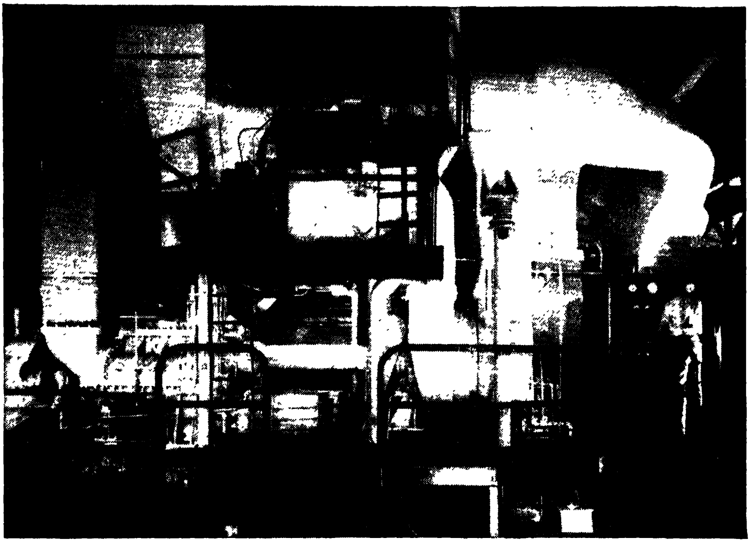


Fig. 9-30. General view of Elliott experimental 3,000-hp marine gas-turbine plant. Compressor at the lower left, driving turbine at lower center with combustion chamber above on the platform.

by-pass is placed around the power turbine. The by-pass can be readily identified in the cross section. The valves are at the outlet end of the annular channel. By-pass opening affects the idling performance by lowering the back pressure of the h-p turbine. This increases the available energy in that element and hence reduces the fuel consumption to keep the compressor turning over with zero plant output.

Elliott Co. In 1944, this company completed the 3,000-hp experimental gas-turbine plant, shown in Figs. 9-29 and 9-30. This plant had a two-shaft arrangement with the h-p turbine operating at 1230 F and the l-p turbine at 1207 F. The h-p turbine drove the l-p compressor and the l-p turbine the h-p compressor and the load. A regenerator, intercooler, and reheat combustion chamber were used in the plant. The over-all pressure ratio produced by the two compressors was 6.5.

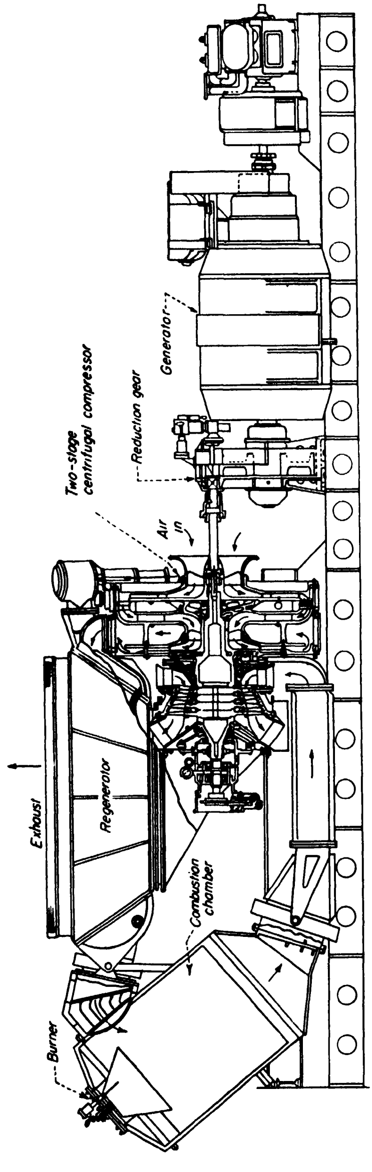


FIG. 9-31a. Cross section of Elliott 3,750-hp gas-turbine plant for locomotive drive. Left to right: combustion chamber, regenerator (above), four-stage reaction turbine driving two-stage centrifugal compressor and load, reduction gear, generator.

The Lysholm compressors, described in more detail later, were positive-displacement compressors consisting of two intermeshing rotors that took in separate charges of air and raised their pressure by reducing their volume before discharging them to the higher pressure region. Thus the pressure ratio was largely independent of rotative speed. Also no pumping limits had to be guarded against as in other types of rotary compressors which will be discussed under compressors.

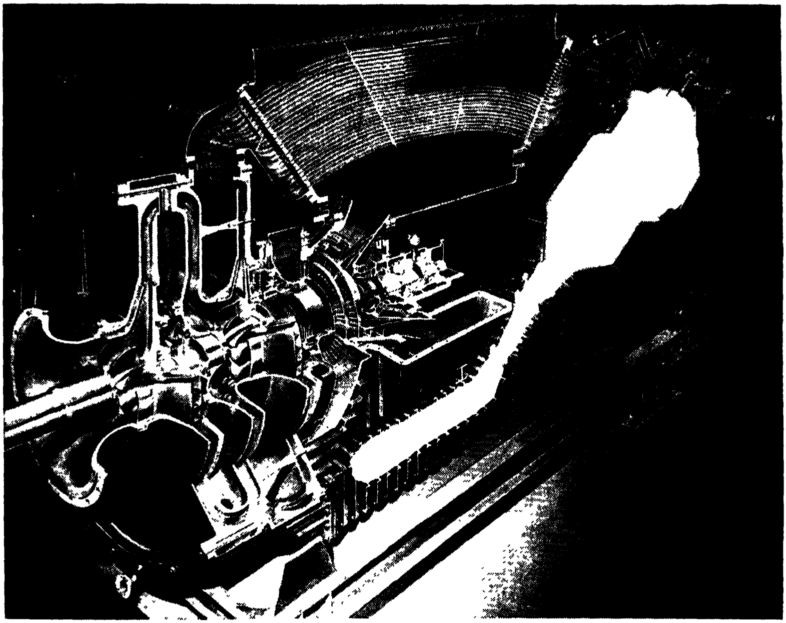


FIG. 9-31*b*. Cutaway view of Elliott gas-turbine plant for locomotive drive.

Another feature of this plant was the combustion chamber designed to burn diesel fuel. Both furnaces were quite small and were built around the fact that, as air in a duct made a sharp turn, two vortexes would be set up. Fuel was injected into them by a modified diesel-engine fuel pump.

Figures 9-31*a* and *b* show a design of regenerative-cycle gas-turbine plant designed specifically for locomotive application, but also suitable for stationary power generation. Air enters the two-stage centrifugal compressor from the right (in *a*), discharges into an annular collecting chamber connecting through an elbow with the regenerator. Flowing through the regenerator the air then enters the single combustion chamber. The large cross-sectional area ensures complete combustion in the oil-fired combustor before the gas enters the duct leading to the turbine inlet. Passing through the four-stage reaction turbine, with shrouded blades, the gas then enters the regenerator from beneath and exhausts to atmosphere through the top.

The 50 per cent effective regenerator uses a tubular design. The oblique disposition of the combustor makes the fuel nozzles accessible. A very compact arrangement results from combining the turbine and compressor rotors into a single assembly. The turbine shaft extends to

FIG. 9-32. 4,000-kw simple-cycle Brown Boveri gas-turbine unit for Neuchâtel, Switzerland, on test bed. First gas turbine to go into commercial service. Left to right: starting motor, generator, compressor, combustion chambers (above), gas turbine.

a bearing between the compressor stages and supports the two impellers. Opposing the h-p ends of the turbine and compressor at the mid-point of the rotor minimizes sealing and leakage problems.

The plant is designed to produce rated output of 3,750 hp at 5,910 rpm with 1275 F turbine-inlet temperature and a pressure ratio of 3.5. By raising the pressure ratio to 3.9 net, output can be increased to 4,000 hp.

Brown Boveri Corp. This Swiss company was the first to develop gas turbines to the point where they were technically workable. At this time they have probably built more stationary gas-turbine plants than any other group. Figures 9-11 to 9-16 are diagrammatic cycles of plants that they have built and proposed. Figure 9-32 is their well-known 4,000-kw underground stand-by plant for Neuchâtel, Switzerland, operating at about 1025 F with a pressure ratio of 4. It is a simple open-cycle unit. The combustion chambers are located above the rotating elements.

Figure 9-33 shows a 4,000-kw simple-cycle unit on the test bed at the Brown Boveri shops. Test results gave an over-all thermal efficiency, based on lower fuel heating value, of 19.5 per cent at 1020 F turbine-inlet temperature. This compares with guaranteed thermal efficiency of 18 per cent at 1100 F. During tests the set was started from cold shutdown and brought to full load in a period of 10 min. Despite the 4.6 pressure ratio the compressor starts without surging. With a large

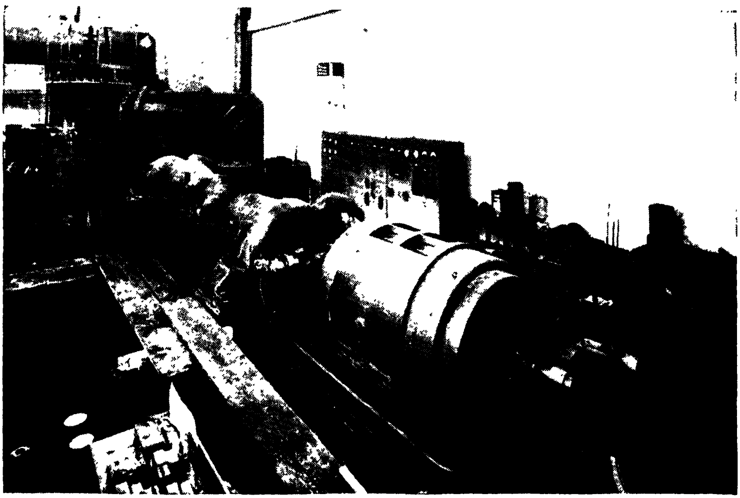


FIG. 9-33. 4,000-kw simple-cycle Brown Boveri gas-turbine unit on test bed. Left to right: combustion chamber, gas turbine, compressor, generator, starting motor.

enough alternator the unit can develop 5,000 kw when the inlet temperature is 1100 F.

Maschinenfabrik Oerlikon. Figures 9-34 and 9-35 show a 1,000-kw experimental gas-turbine plant operating in Zurich, Switzerland. It can operate as a single-shaft or two-shaft machine with the h-p turbine driving the generator at 3,000 rpm and the l-p turbine driving the compressor at 4,500 rpm. Inserting a gear between the two shafts converts the unit to single-shaft operation. Gas temperatures and performance have not been announced, but the pressure ratio is 4, developed by an intercooled three-stage centrifugal compressor of unusual design. Most centrifugal compressors have single diffuser outlets, as shown at the left in Fig. 9-36. By using four diffuser discharges for each stage, as shown at the right, turbulence and friction losses are remarkably reduced and result in a large increase in compression efficiency.

When the same impeller is used, compression efficiency rises from

76 per cent with a single diffuser discharge to 88 per cent when running with four diffuser discharges in the casing. Multistaging these compressors with four diffuser outlets makes a compact unit including intercoolers (Fig. 9-37). Each intercooler takes care of one diffuser discharge

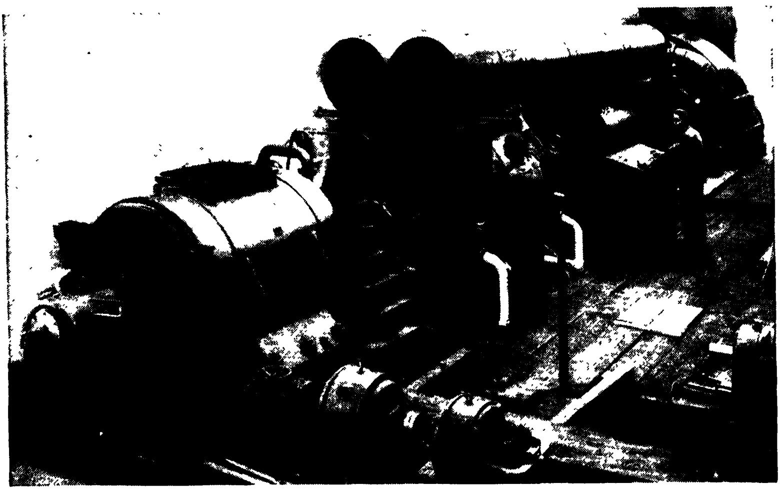


FIG. 9-34. 1,000-kw experimental Oerlikon gas-turbine plant. Compare with Fig. 9-35 for equipment identification.

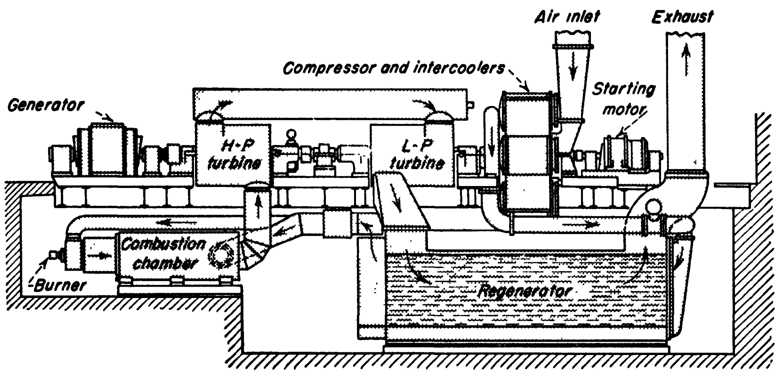


FIG. 9-35. Cross section of 1,000-kw Oerlikon gas-turbine plant.

from each of the first and second stages. The cooling-water tubes extend the entire length of each exchanger which is divided in half longitudinally to accommodate the air of differing pressures. A collecting pipe leads the cooled air from each stage into the suction of the succeeding stage.

This unit features a novel burner arrangement. The fuel-injection system needs only a few pounds excess pressure to inject fuel oil into the combustion chamber. The fuel nozzles have large cross-sectional areas that have proved to be practically nonclogging.

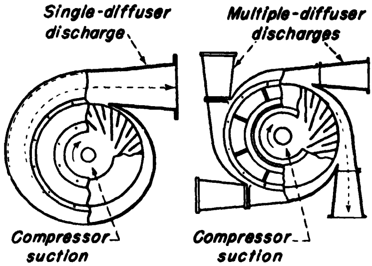


FIG. 9-36. Centrifugal compressor of Oerlikon unit has four diffuser discharges compared with one usually supplied on most designs. Arrangement markedly raises the compression efficiency.

Boeing Airplane Co. For many years most designers felt that gas turbines would not be suitable for rated capacities under 1,000 hp. However, this company has forseen the advantages of a small gas turbine for small power applications such as auxiliary drives and industrial uses. Figure 9-38 shows a 160-hp experimental unit that has completed extensive laboratory tests. The composite drawing shows an elevation cross section at the top half and a plan external view for the bottom half. The planes of the two halves are rotated 90 deg in respect to each other.

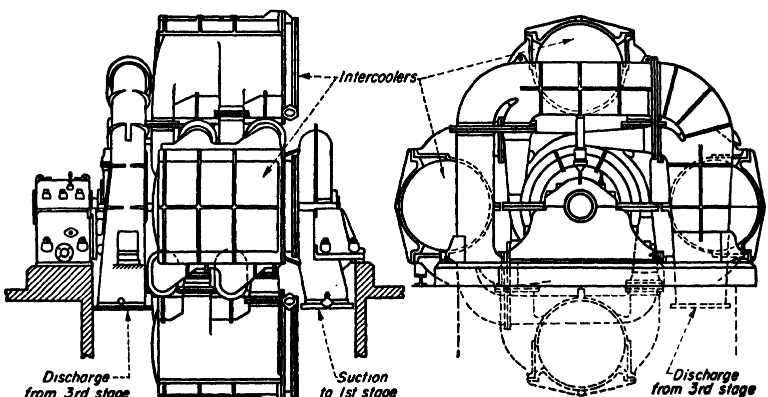
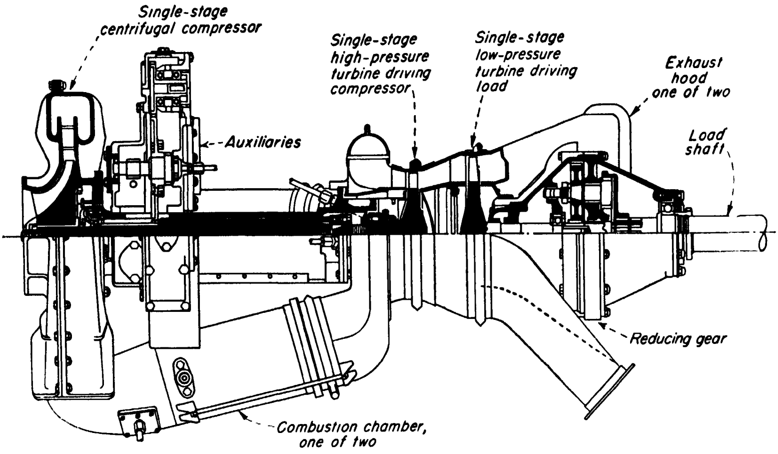


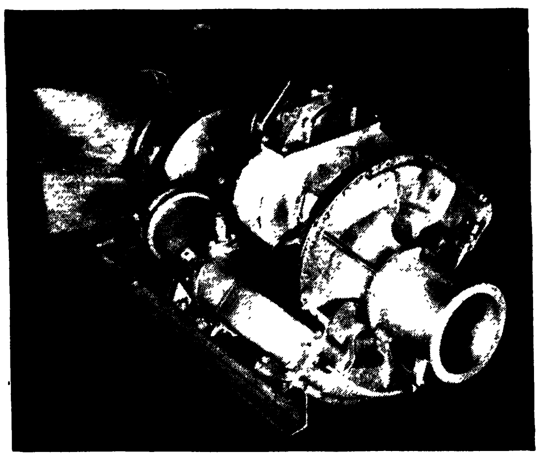
FIG. 9-37. Elevation views of three-stage centrifugal compressor and intercoolers of Oerlikon gas-turbine plant.

This extremely compact unit features a single-stage centrifugal compressor driven by a single-stage h-p impulse turbine at a full-load speed of 36,000 rpm. The compressor discharges through two diffuser outlets. Each outlet feeds a combustion chamber. The two combustors exhaust into a common chamber ahead of the h-p turbine nozzles.



(Top half is sectional elevation, lower half is plan view)

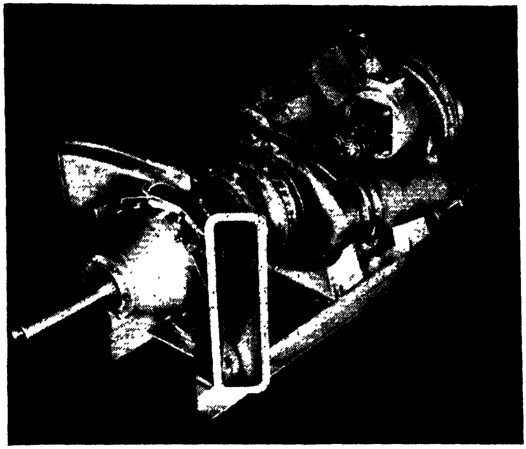
(a)



(b)

FIG. 9-38. 160-hp experimental turbine developed by the Boeing Airplane Co. Single-stage h-p turbine drives single-stage centrifugal compressor feeding two combustion chambers. Exhaust from the h-p turbine enters a single-stage l-p turbine driving the load.

The h-p turbine exhausts directly into the l-p load-driving single-stage impulse turbine turning it at full-load speed of 25,000 rpm. A reducing gear lowers the load-shaft speed to 2,500 rpm. The split-turbine arrangement allows variable-speed operation of the compressor for improved



(c)
FIG. 9-38. (Continued)

part-load efficiency. The unit is designed for operation at 1500 F turbine-inlet temperature with a pressure ratio of 3:1. Over-all efficiencies of about 15 per cent are expected for commercial models of this unit.

PLANT COMPONENTS

Turbines. Gas turbines utilize the familiar schemes of impulse and reaction blading. Developed cross sections of gas-turbine blading look much like those of steam turbines. Differences in construction grow out of the physical variations of a gas and vapor and from the differences in

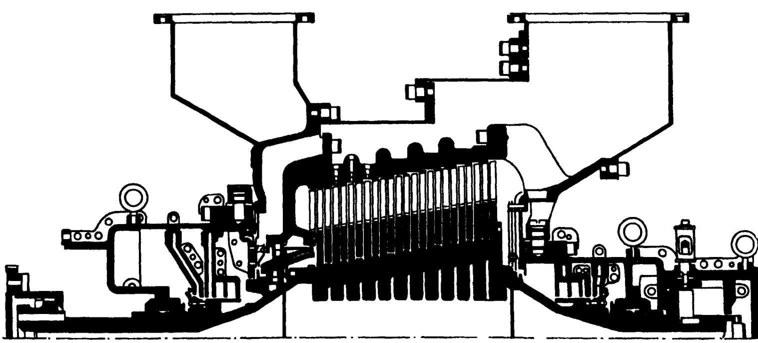


FIG. 9-39. Turbine of experimental 2,500-hp Elliott gas-turbine plant operates at 1250 F, 95.5 psia, 3,720 rpm. Hollow rotor is fabricated from rolled-plate disks and carries 12 reaction stages.

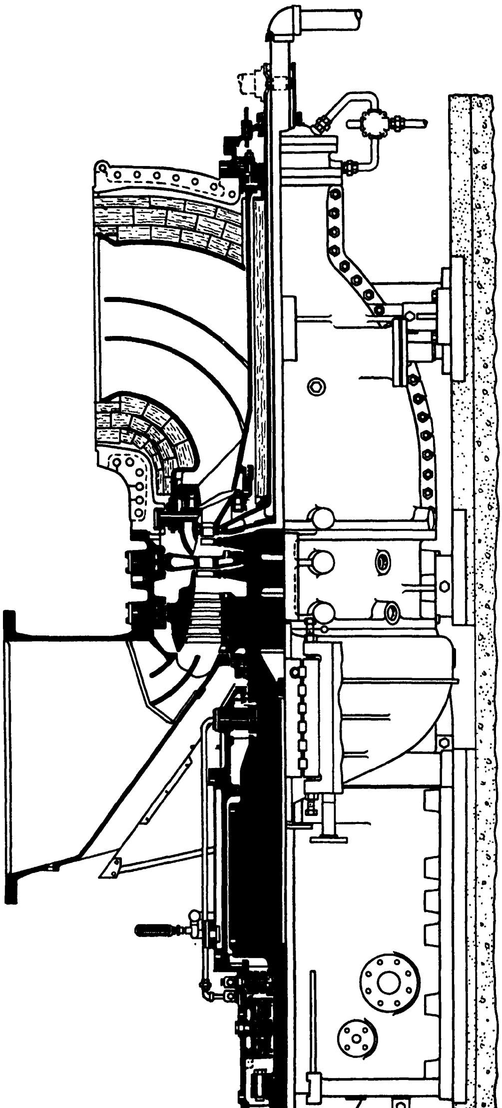


FIG. 9-40a. Turbine of 3,500-hp Allis-Chalmers plant at U.S. Naval Experiment Station operates at 5,200 rpm, with gas at 1500 F, 56.5 psia.

operating conditions. In contrast to steam turbines, gas units have relatively few stages, because they operate on smaller pressure drops. Larger volume flows lead to wider passages and taller blades without extreme differences in blade height. Normally no control valves and partial admission arrangements are necessary.

Blades have been made by forging, by machining from rolled bar stock and shapes, and by precision casting. Machining offers simplified metallurgical control and cost savings but may require compromising on blade shape. Practically any shape can be made by forging, but since many turbine blades must operate at temperatures not much below forging temperatures, this method of manufacture has limitations. With the development of improved metallurgical control, precision casting may offer a solution for extremely high temperature applications.

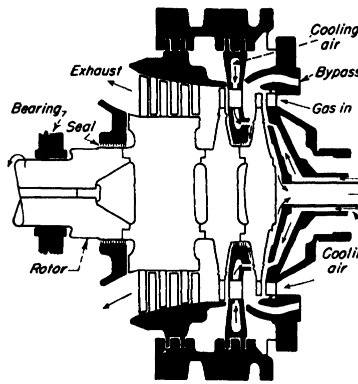


FIG. 9-40b. Detail showing two impulse and three reaction stages, starting by-pass, air cooling for both sides of the first-stage wheel of turbine in Fig. 9-40a.

Turbine rotors and disks present a serious problem. Large forgings of heat-resisting alloys cannot yet be fabricated dependably. This leads to the built-up rotor comprising disks welded or bolted together as shown in Figs. 9-18, 9-21, 9-26, 9-31, and 9-39. The Elliott turbine shown in the latter figure has a hollow rotor made up of cross-rolled plates. The first disk is shrunk on the forged stub shaft and welded, each succeeding disk being installed in the same manner.

Turbine casings can be fabricated without too much difficulty by forged and cast-alloy sections welded together. A high expansion rate with rising temperature characterizes many of the heat-resisting alloys; hence turbines must accommodate expansions up to ⅛ in. per ft. The turbine in Fig. 9-39 has its various components supported on radial pins so that correct centering is maintained while allowing complete freedom for uniform radial expansion. To provide for axial expansions of about ½ in., a linkage supports one of the main bearings.

By overhanging the turbine rotor from a shaft as in the Allis-Chalmers turbine of Fig. 9-40a, the gland sealing problem at the h-p high-temperature end of the casing is eliminated. Figure 9-27 details one of the schemes used in sealing the turbine and simultaneously cooling the high-temperature parts. The turbines of Figs. 9-18 and 9-21 also use cooling schemes in order to gain advantages of high-temperature operation. The unit in Fig. 9-40*a* has two rows of impulse blading followed by three rows of reaction blading. Large temperature drops can be effected in the impulse nozzles by allowing a relatively large pressure drop. Thus the gas temperature at the moving blading will be considerably below the nozzle-inlet temperature. In addition to this feature, the first-stage blading and disks are cooled as detailed in Fig. 9-40*b*. For this experimental turbine using 1500 F gas, air at slightly above first-stage pressure is brought in from a separate air pump through the annular space of a double wall pipe, right. From here the air flows through a series of holes just under the first-stage nozzles and impinges on the first-stage wheel below the blades. The air then flows over the disk surface radially inward and exhausts through the center pipe. A positive-pressure blower circulates the cooling air which is cooled for recirculation by external heat exchangers.

From the same source, cooling air enters a belt surrounding the second-stage nozzles, flows through hollow nozzle vanes into an annular space below, and then discharges against the downstream side of the first-stage disk. This air flow, which just about equals leakage past the diaphragm labyrinth seal, joins the gas stream and passes through the remainder of the turbine to exhaust.

Air cooling and other unusual features mark the turbine and compressor unit built by De Laval shown in Fig. 9-41. Figure 9-42 shows the two mixed-flow compressor rotors at the left and the Birman wheel at the right comprising the turbine rotor. As the cross section shows, gas from the combustion chamber flows through stationary nozzles distributed around the entire wheel periphery and enters the wheel tangentially. While flowing through the rotor, the gas makes a 90-deg turn and exhausts in an axial direction. Gas flow through this turbine can be controlled by providing the nozzles with movable vanes (as in the gate of a hydraulic turbine). Nozzle openings can in this way be varied from full open to closed, controlling both the direction of gas entering the wheel and the rate of flow.

Cooling air for the turbine wheel bleeds from the compressor discharge and is cooled before entering the chamber surrounding the turbine bearing adjacent to the wheel. From here air flows through channels in the wheel hub and leaves at the exhaust end after cooling the hub. The concentric tube keeps the cooling air separated from the turbine exhaust gas so that the air serves as combustion air in the following reheating combustion chamber.

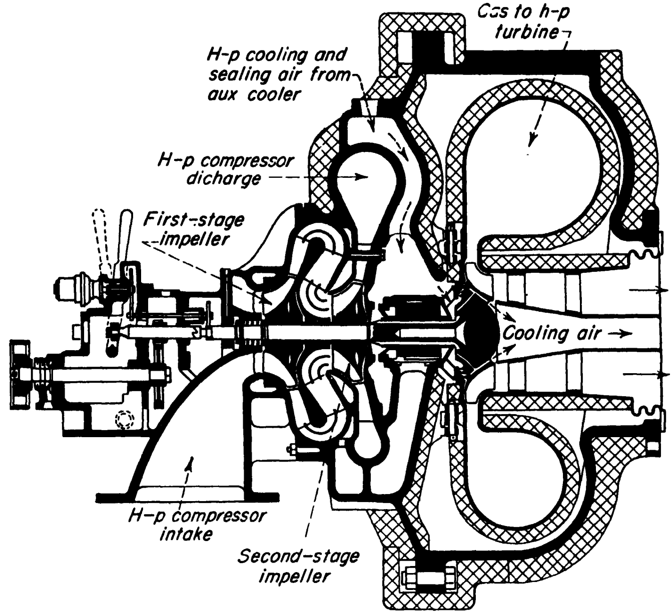


FIG. 9-41. Cross section of the De Laval high-speed two-stage mixed-flow compressor and Birman-wheel turbine for operation at 1500 F. Cooling air flowing through the hub of the turbine wheel supports combustion in the reheating furnace following the turbine.

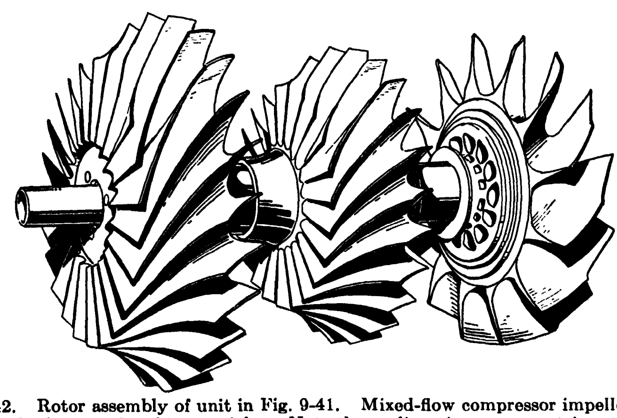


FIG. 9-42. Rotor assembly of unit in Fig. 9-41. Mixed-flow compressor impellers at left, Birman-wheel rotor of turbine at right. Note the cooling-air passage entries at the hub.

Compressors. The heart of the gas-turbine plant is the compressor. It was this component that delayed practical realization of gas turbines. While many means of compressing air were known for years, it was only lately that *efficient* methods needed for the cycle were developed. The large flow rates involved ruled out using the relatively efficient reciprocating forms of compressors. Three basic forms of rotary compressors have been successfully applied: (1) centrifugal, (2) axial-flow, (3) Lysholm.

To understand the behavior of centrifugal compressors let's first study the conditions controlling performance of a close cousin, the centrifugal blower or fan, used for ventilation and air supply to boilers, two of its many applications. Figure 9-43 shows the elevation of a steel-plate fan feeding into a duct which can be partly or completely closed by a vertical damper near its discharge. The rotor of the fan looks like a paddle wheel, flat blades supported between two rotating end rings. Air enters the center of the rings, flows between the blades, and discharges into the duct. The air caught between the blades is thrown out radially by the centrifugal force. The air pressure is raised and its velocity increased by this action. Removal of the air from between the blades tends to form a vacuum, in turn causing air from the wheel center to enter the blades. In this way a continuous flow through the fan is established. In most of these fans the total rise in pressure is usually only a fraction of 1 psig. The pressure rise depends on the flow and speed of the fan rotor. This relationship is shown in the graph of Fig. 9-43. At a constant fan speed of 100, the pressure developed when the system is closed will be a relative value of 100 as shown by point A corresponding to the system condition A, the discharge duct being tightly closed and no flow taking place.

By keeping constant fan speed and partly opening the damper as in B, air flow will be established, but the duct pressure between fan outlet and damper will be somewhat less, as shown by point B. Still maintaining constant fan speed and opening the damper wide as in C, the air flow will increase to a maximum, but pressure rise will be practically zero, as shown by point C. Thus for constant fan speed there can be only one pressure for any given air-flow rate.

However, this pressure-flow relation changes with fan speed as shown by the curves marked 90 and 80 relative fan speed. Both maximum shutoff pressure rise and maximum air flow or fan delivery decrease with lower fan speeds.

Now, most air-flow systems are constant in their cross-sectional areas and general layout. To vary the air-flow rate through such a system the pressure must be varied. The curve OH in Fig. 9-43 shows this relationship for a duct system. The usual relation shows that about four times the pressure rise is needed to double the flow, or nine times the pressure rise to triple the flow, and so on. The fan can deliver a flow rate at the point D when it operates at a speed of 100. To deliver

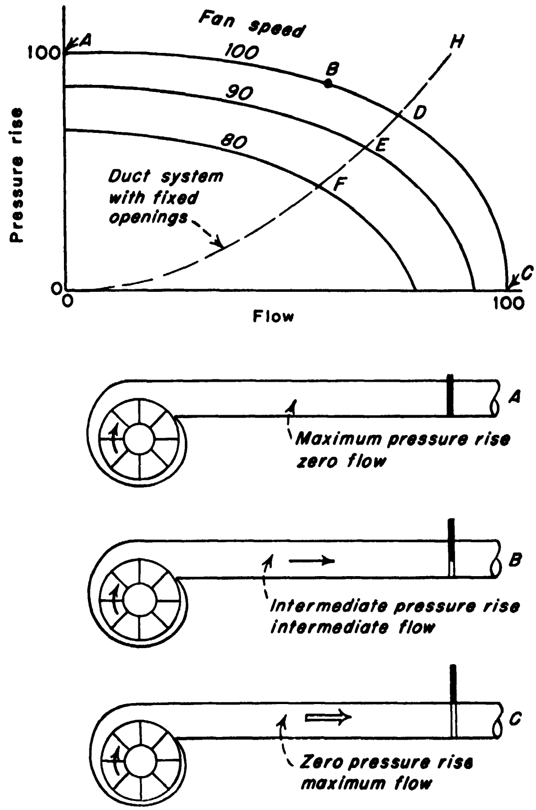


FIG. 9-43. Pressure-flow relations for a common steel-plate (flat-bladed) blower developing pressures below 1 psig.

a lower flow rate through this system the fan speed need only be reduced as shown by points E and F.

A three-stage centrifugal compressor similar to designs used on gas-turbine applications is shown in Fig. 9-44. The casing inlet to the first-stage wheel center can be seen at the left. Air flows through the wheel and into the diffuser surrounding the entire wheel periphery. The diffuser changes part of the increased air velocity to additional pressure rise by slowing it down slightly. After this diffusion the air is turned

180 deg to enter the center of the second-stage wheel. The process repeats through the second- and third-stage wheels. The wheels in this

FIG. 9-44. Three-stage centrifugal compressor with shrouded impellers. Note the backward curves of the vanes, diffusers at the wheel periphery, entries at the wheel hubs.

compressor are of the completely shrouded type. Another type used mostly in aircraft gas turbines is shown in Fig. 9-45. Designed for high-speed work, this wheel is built identically on both sides of the center disk. Air enters axially near the rotor center and is thrown out radially from the disk edge. The rotor turns within a closely fitting stationary casing.

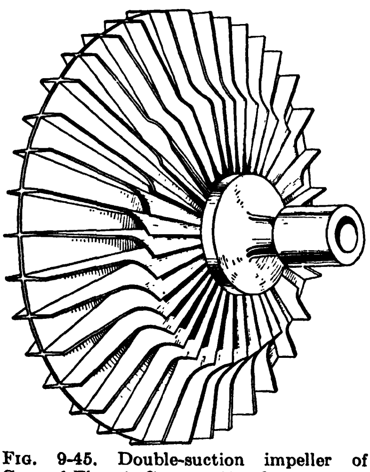


FIG. 9-45. Double-suction impeller of General Electric Co. centrifugal compressor for high-speed jet engine. Note the unshrouded design.

Figure 9-46 shows an imaginary instantaneous snapshot of a centrifugal compressor rotor running at full speed with air flowing through the rotor. Again there is that queer impression of air flowing through the backward-curved blades of the wheel. One must realize that both wheel and air are moving. The streamlines show how the air actually moves relative to the earth. The blades however are acting on the air and changing its direction of flow from radial to a more tangential direction before it leaves the wheel. In addition to having its direction changed, the air is being compressed (called the "reaction" effect), causing the desired pressure rise. The velocity of the air is also increased as shown by the vector diagrams for the entrances and exits of the blades in Fig. 9-46.

The increased velocity energy of the air leaving the impeller can be converted to pressure increase by arranging a proper diffuser passage at

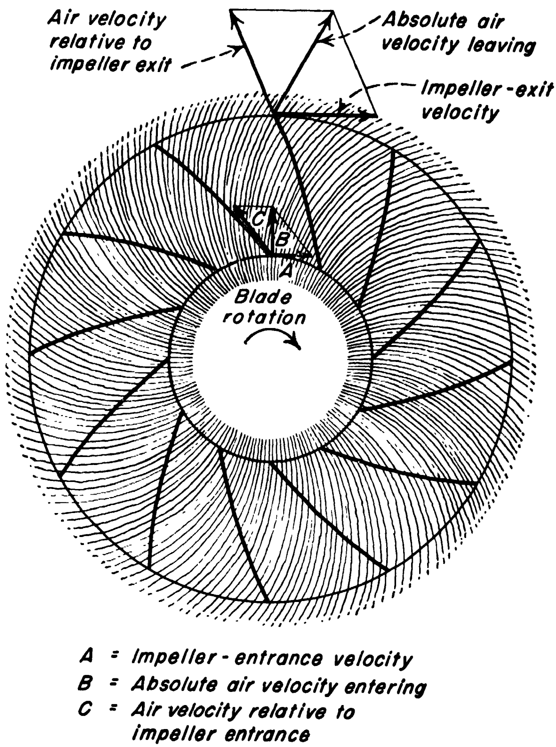


Fig. 9-46. Velocity relations of air flowing through centrifugal-compressor impeller. Flow lines indicate the path of air flow in relation to *casing*. Note that both impeller blades and air are moving simultaneously.

the outlet periphery of the impeller. This is shown diagrammatically in Fig. 9-47. By allowing the high-velocity air to enter the diffuser, which is nothing more than a reversed nozzle, the air speed drops as it passes through the diverging section of the diffuser. In effect the air catches up with itself and compresses with resulting pressure increase.

This action is the reverse of what takes place in a nozzle. Referring again to Fig. 5-7*a*, you will recall that in a nozzle h-p steam or air expands to a low pressure and leaves the nozzle at high velocity.

The normal arrangement of the diffuser casing about the wheel or impeller was shown in Fig. 9-36. Most have been built with just a single diffuser outlet. As mentioned in connection with this figure, the single diffuser-outlet centrifugal compressor has a relatively low efficiency, (1) because of the rather tortuous path the air must follow in a multistage compressor causing high friction losses and (2) because diffusion

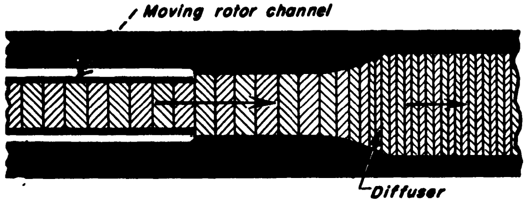


FIG. 9-47. Air leaving the periphery of a centrifugal compressor-impeller enters the diffuser which slows it down and compresses it. Action is the reverse of that taking place in a nozzle.

compression processes are relatively low in efficiency. This has led to successful development of the axial-flow compressor.

At first glance the axial-flow compressor of Fig. 9-48 looks very much like a reaction turbine. In general arrangement this is true, but the blade design is quite different. To understand how the axial flow com-

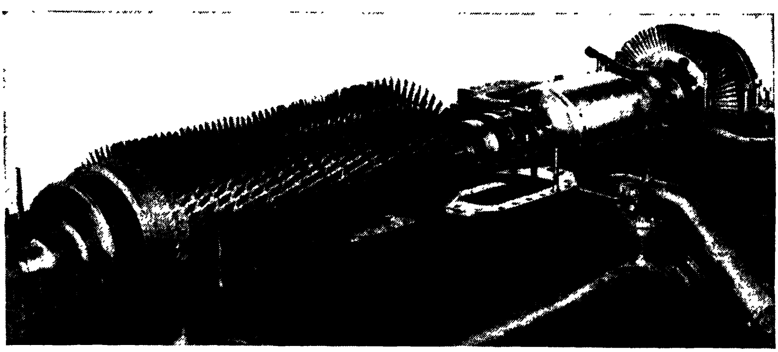


FIG. 9-48. Blading of an axial-flow compressor resembles a reaction turbine in general arrangement. Blade sections and dispositions differ from the turbine because of reverse action, compression instead of expansion of fluid.

pressor works, let's refer to Fig. 1-4. Here the steam jet impinging on the blades on the wheel exerted a force which turned the wheel. Now let's remove the jet and supply power by a motor to the shaft, turning it in the direction of the arrow. What happens to the air surrounding the blades?

If you cannot try it experimentally, a little thought will show that the right-hand edge of the blades will "bite" into the air and scoop it along in a direction generally parallel to the shaft, or in an axial direction. Air flow will be speeded up and, by passing it through some kind of diffuser, we could compress air. If the blades are properly shaped, we shall also be compressing air between the moving blades. Such blades are sections of propellers called "airfoils" as in Fig. 9-49. Here are two stages of an axial-flow compressor and the resultant flow of air relative to the earth. The stationary blades form diffusing passages which slow down the air flow slightly and compress it to a higher pressure. The moving blades increase the absolute air velocity (relative to earth), but the vector diagrams show that air velocity relative to the moving blades decreases as it passes through them; hence, air pressure also increases. The air-pressure graph shows that pressure increases are equal for stationary and moving blades. The vector diagrams below are, respectively, for the leading edge and the trailing edge of the moving blades. Note the slight change in direction that the air undergoes as it passes between the blading. This blading arrangement is known as symmetrical staging, the pressure ratio per stage is of the order of 1:15 to 1.2. High blade and air speeds characterize this arrangement. The air follows a generally helical path through the compressor casing. Similar to a reaction turbine, air fills the entire blade passage.

Figure 9-50 shows two stages of nonsymmetrical blading. The graph shows that most of the pressure rise occurs in the moving blading. The air generally follows a straighter path axially through the compressor. Diffusing action of the stationary blading is relatively small. Pressure ratio per stage runs about 1.1. Blade and air speeds are less than in the symmetrical-bladed compressor. Because of the lesser pressure rise per stage, more stages will be needed to produce the same over-all pressure ratio.

Figure 9-51 shows two stages of vortex blading. Here air pressure drops through stationary blades, actually making them nozzles that slightly accelerate the air on its way through. All pressure rise takes place in the moving blades. Pressure ratio per stage runs around 1.05. Air and blade speeds reach their lowest values in this arrangement. The vortex-bladed compressor requires the greatest number of stages to develop a given over-all pressure ratio.

Before considering the third type of compressor let's compare the operating characteristics of the centrifugal and axial-flow compressors in Fig. 9-52. The first thing noticeable is that the constant-speed pressure curves do not go back to the zero flow condition as for the fan in Fig. 9-43. The curves end at a curve called "pumping limit." This line is also called "surge line," "limit of stability," or "stall line." At

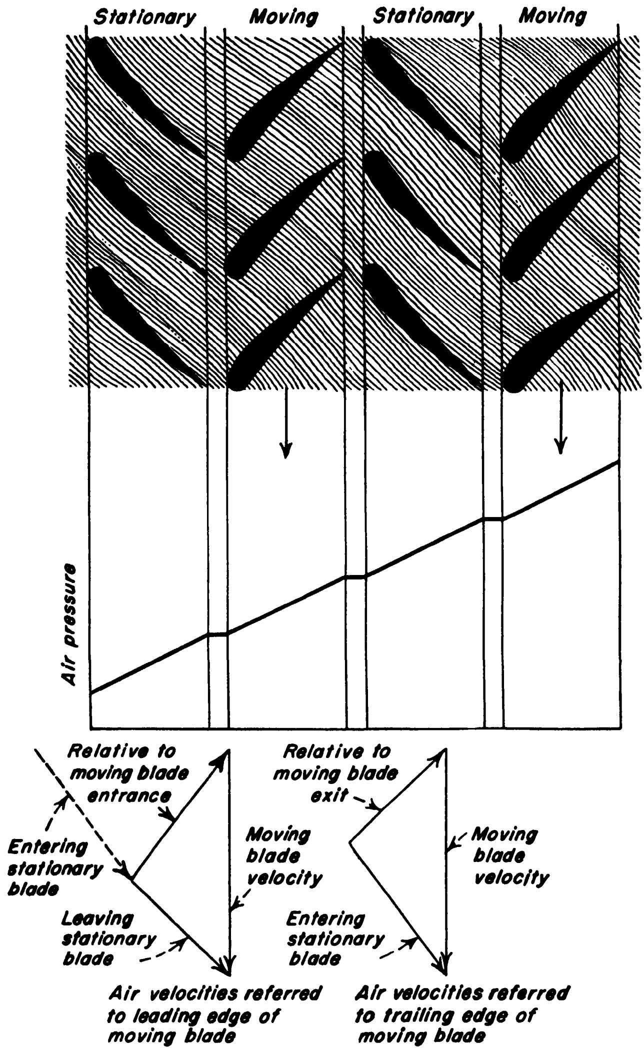


FIG. 9-49. In symmetrical blading of an axial-flow compressor, the pressure rise is equal in moving and stationary blading. Air velocities are slightly reduced in each as their directions are turned through only a few degrees, causing desired pressure increase. Air travels helically around the shaft.

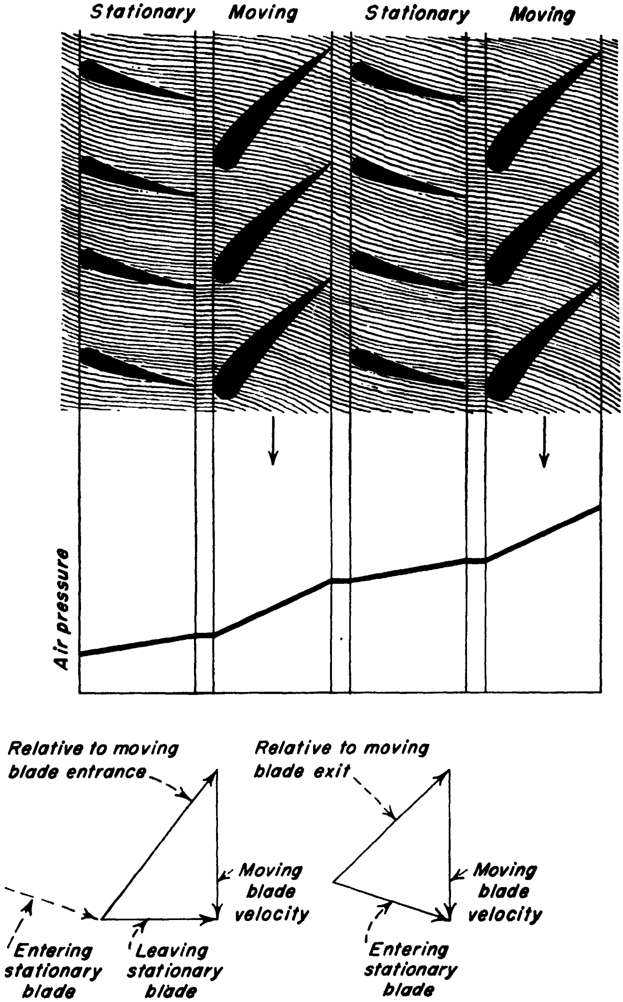


FIG. 9-50. In nonsymmetrical blading of an axial-flow compressor, the pressure rise is greater in moving blading. The helical air path is almost parallel to the rotor shaft.

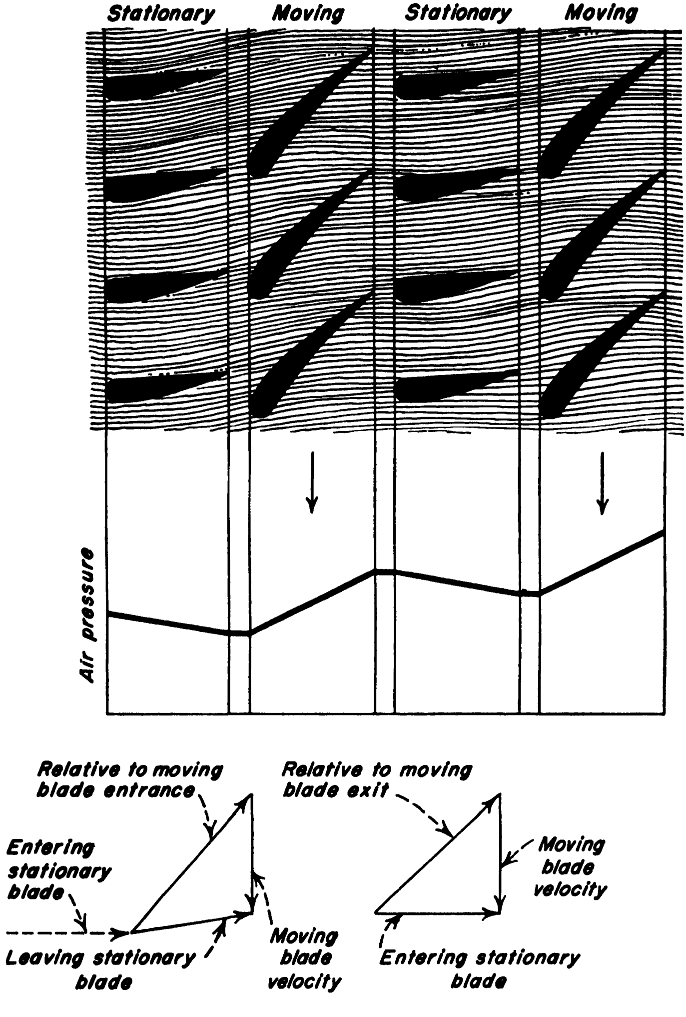


FIG. 9-51. In the vortex staging of an axial-flow compressor, the pressure drops through stationary blading but rises in moving blades. Absolute air velocity accelerates through stationary blading.

this limit, flow becomes unsteady in the rotor blading and tends to circulate within the blading. In the case of the axial-flow compressor, air tends to be thrown radially outward in the moving blading. This causes a sudden collapse of discharge pressure because of stoppage and reversal of flow. With constant speed of the rotating element, the pressure will immediately build up again and reestablish flow momentarily, only to collapse again, and so on. This operation at high speeds causes

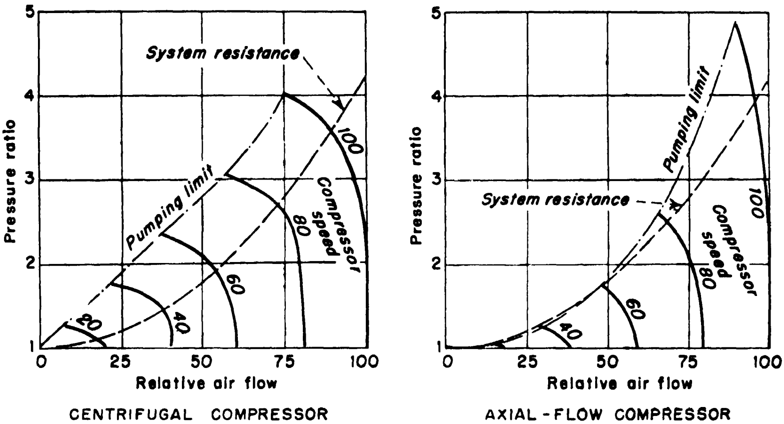


FIG. 9-52. Comparison of centrifugal and axial-flow compressor pressure-flow relations. As the flow reduces at constant rotor speed, a pumping limit is reached in each where flow becomes erratic with pressure rapidly falling and building up. Pumping limit occurs at lower flows in the centrifugal than in the axial-flow unit.

considerable noise and vibrations which can cause disastrous results. At lower speeds, operation at the pumping limit can be tolerated.

Normally a gas-turbine plant will have a constant system resistance curve, as shown in Fig. 9-52. In the case of the centrifugal compressor there is not much difficulty in designing the system to stay well away from the pumping limit. In the case of the axial-flow unit, careful proportioning is needed to fit the compressor into the plant, because of the much narrower range of operation at a given speed. If the system resistance cannot be kept below the pumping limit, then it usually can be compromised so as to cross the pumping limit at a value that would be encountered only during transient starting and stopping conditions. At these lower conditions the unstable operation is mild and can be safely withstood by the equipment. With the exception of the Oerlikon centrifugal compressor using the multidiffuser discharge, efficiencies of centrifugal units run lower than axial-flow compressors.

The third type of compressor, the Lysholm, is a rotary lobe machine

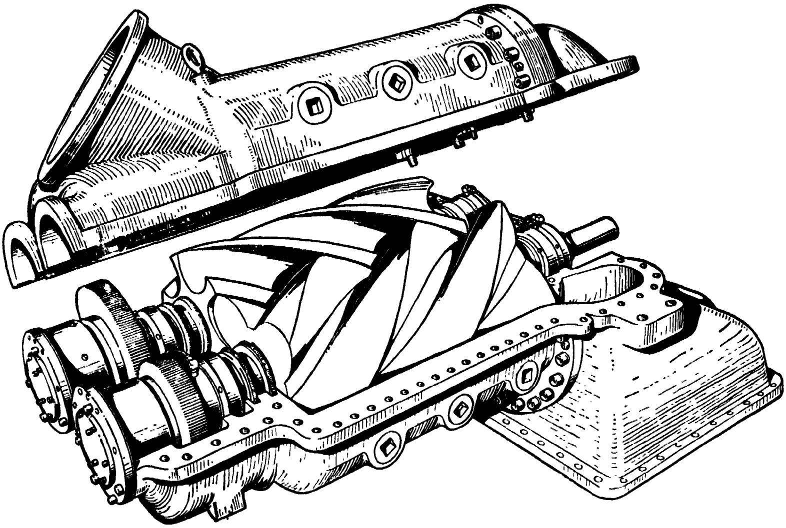


FIG. 9-53. Lysholm compressor with cover lifted. Gear-driven lobed rotors mesh without touching, raising the air pressure by reducing the volume. The intake hood is at the lower right, discharge at the upper left.

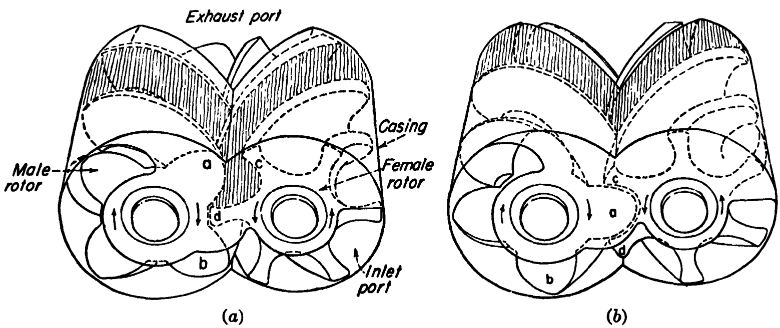


FIG. 9-54. (a) Shaded areas show two volumes of air trapped by each rotor as the lobes start to mesh. (b) As the lobes start to mesh with continued rotation, air volume reduces, raising its pressure. Volumes are just beginning to exhaust through the discharge at the rear.

which compresses air by reducing its volume before discharging. Figure 9-53 shows a Lysholm compressor with the top half of the casing lifted. The lobes mesh accurately without touching, and drive gears, at the left, maintain their relative positions. Air enters the inlet hood at the right and discharges through the nozzle at the upper left. Figure 9-54 shows the operating principle. As rotors unmesh in the lower part of the casing, spaces formed between lobes and casing open up into the inlet port and allow air to enter and fill the spaces. When the front ends of spaces move past the cutoff point on the inlet port, these air volumes are trapped. Figure 9-54*a* shows in the two shaded portions the two volumes of air trapped by each rotor, with male lobe (*a*) just starting to mesh with female rotor lobes (*c*) and (*d*). Further rotation

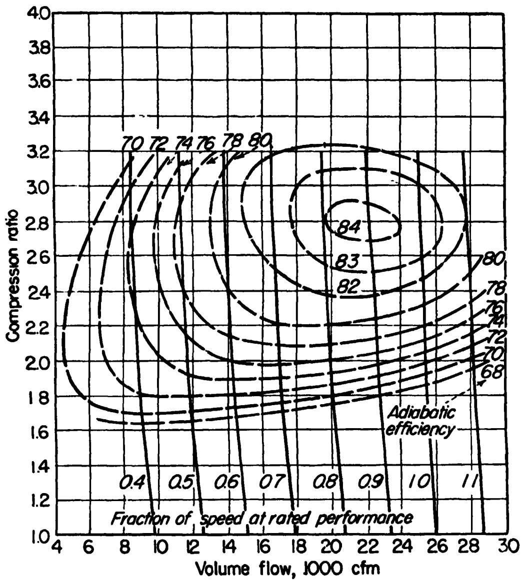


FIG. 9-55. Free-air volume flow through the Lysholm compressor is almost independent of back pressure and depends entirely on the speed of the rotors.

with forward ends of the lobes fully meshed is shown in Fig. 9-54*b*. The merging and consequent reduction in total volume of the two air masses caused by the lobe meshing are evident. In this position the shaded spaces are just opening to the discharge port. Air will be completely discharged against the higher pressure from the rotor spaces by their continued meshing to the back end.

The Lysholm displays the most stable operating characteristics, as shown in Fig. 9-55. In contrast to the others, in which pressure is a unique function of speed and the capacity range at any particular speed is limited, the Lysholm can deliver volume in proportion to speed and at any particular speed can work against any pressure up to the design limit.

Disadvantages of the Lysholm compressor are its low speed compared with an economically designed turbine for driving. This type of compressor also takes up more space than a centrifugal or axial-flow unit.

Combustion Chambers. Extremely high rates of heat release mark gas-turbine combustion chambers which, for open-cycle plants burning liquid or gaseous fuels, are relatively simple in construction. Figure 9-56 is typical of those designs using single furnaces. Since maximum usable gas temperatures range from 1100 to 1500 F, compared to 3000 to 4000 F

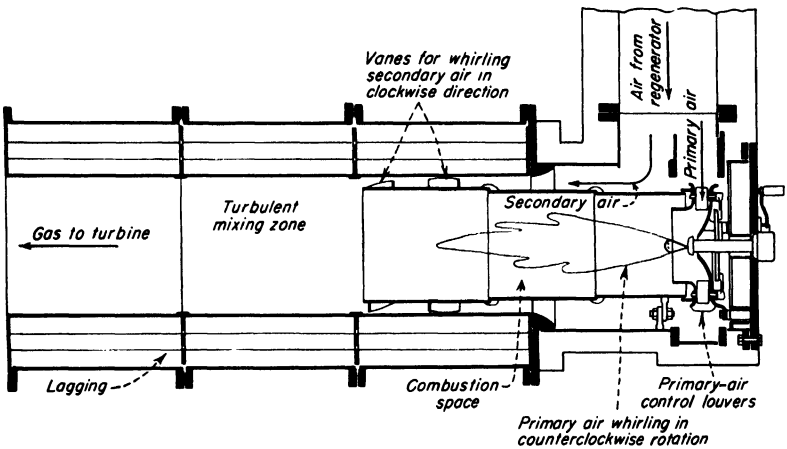


FIG. 9-56. Double-shell combustion chamber of Allis-Chalmers 1500 F unit. Only combustion air enters the flame tube. The flow of excess air cools the metal sections.

in the combustion zone, large quantities of excess air are needed. The double-shell design takes advantage of this excess air to maintain metal parts below critical temperature levels and avoid the need for refractory construction.

Incoming air divides into two streams, combustion air entering the flame tube behind the burner nozzle through a register with movable vanes controlled from the outside. In this way only enough air for burning enters the flame area, and ignition troubles from large quantities of air quenching the flame are avoided. The largest portion of the air flows through the annular space between the flame tube and the combustion-chamber wall and cools the flame tube. Vanes in the annular space give the air a spin opposite in rotation to that of combustion products promoting turbulence and good mixing.

A mechanical atomizing nozzle injects fuel oil continuously. Flow rate can be varied through a wide range in response to load demands. Heat releases of 2,500,000 Btu per hr per cu ft have been realized.

Figure 9-57 shows a comparison of the old and the new type of mechanical spray nozzles for injecting fuel oil into combustion chambers. The improved atomization is self-evident.

Figure 9-58 shows the combustion chamber used in the Elliott plant of Fig. 9-30. Air enters at the bottom, part entering the ignition cone

FIG. 9-57. Comparison of old and new atomizing nozzles for fuel oil used in gas-turbine combustion chambers.

through the slot as primary air. The main body of air makes a right-angle turn and forms two vortexes into which the fuel oil is injected. The comparatively small flame has an estimated temperature of 3000 F. Full-load heat release equals 2,500,000 Btu per hr per cu ft.

Injection equipment similar to that used in diesel engines, operating at a frequency of 160 pulses per second to produce a continuous flame, gives a turndown range of between 20 and 30 to 1 for part-load operation.

Multiple combustion chambers used in the plants of Fig. 9-17, 9-21, 9-22, and 9-25 follow the same principle as that of Fig. 9-56 in having a central flame tube cooled by surrounding excess air. In these, though, the excess air enters the flame tube through openings downstream of the burning zone. The annular space is closed on the downstream end. All mixing takes place in the exit section or end of the flame tube.

The fuel requirements of gas-turbine plants vary widely. Most open-cycle plants will be able to burn bunker C fuel oil, although some prefer diesel oil. Gas will always be an ideal fuel.

Coal availability and its lower price in most regions dictate the advisability of burning this fuel in gas turbines to widen their application opportunities. However, since combustion products pass through the turbine in open-cycle plants, this presents some problems. Coal will have to be burned at much higher rates than has been conventional in boiler furnaces, a top of about 35,000 Btu per cu ft per hr. Coal ash will have to be prevented from passing through the turbine where it will probably block gas passages and erode blading.

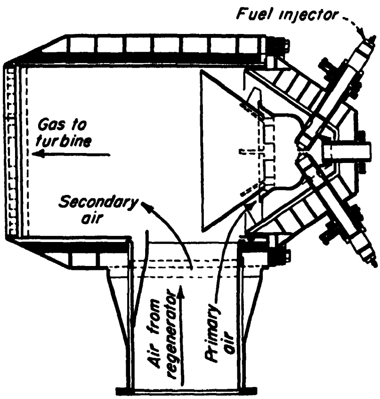


FIG. 9-58. In the Elliott elbow, combustion-chamber fuel is injected intermittently at about 160 pulses per second. A sharp change in air-flow direction produces vortex action for mixing of the combustion products and excess air.

Research is now being devoted to burning pulverized coal in the combustion chamber and passing the combustion products and ashes through mechanical separators to remove the ash. Other means being investigated are the superfine pulverization of coal with the thought that fine ash will not have detrimental effects on turbine parts. Also being investigated is the gasification of coal under pressure to produce fuel gas for the turbine. So far, these efforts have apparently not progressed beyond the laboratory stage.

Regenerators. Gas-turbine plant heat exchangers include intercoolers and regenerators. The former usually do not differ greatly from normal practice as used with compressed air and so are not described here. On the other hand the regenerator has unique problems to overcome, gas-to-gas heat transfer with relatively small temperature drops, and a strict limit on allowable pressure drop tends to require a bulky unit if any high degree of heat recovery is desired.

Figures 9-59 to 9-61 show typical regenerators. In the first one, air makes a single pass through tubes arranged in eight groups. Turbine exhaust makes three passes across tube exteriors with turns planned for minimum pressure loss.

Figure 9-60 shows a regenerator for a closed-cycle plant (discussed later) in which only clean air passes over both interior and exterior tube surfaces. This allows the use of small-diameter tubes in bundles connected to inlet and discharge manifolds. Both air streams make a

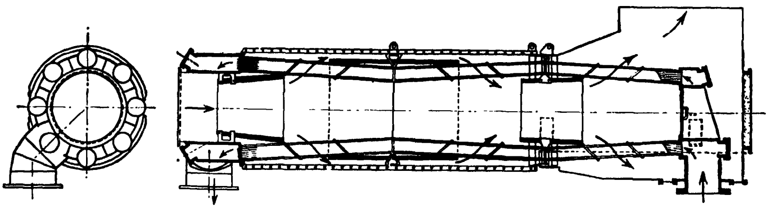


FIG. 9-59. Gas-turbine plant regenerator designed for exhaust gas to make three passes over the tube exteriors while compressed air makes a single pass through the tube interiors.

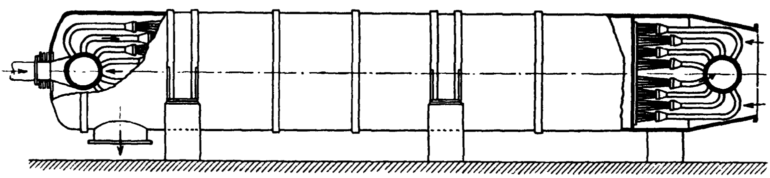


FIG. 9-60. Regenerators for closed-cycle plants are characterized by very small tube diameters which may be used because of clean and dense air. Shells must be strong to withstand the high air pressure.

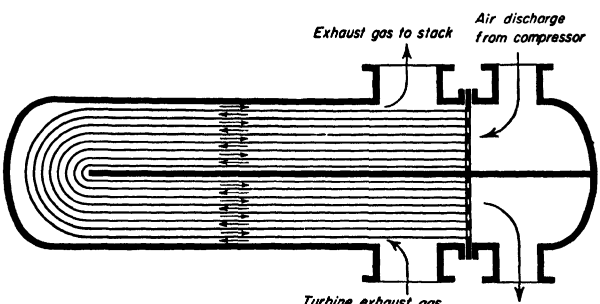


FIG. 9-61. A regenerator designed for single-pass counterflow arrangement. U-tube utilization shortens the over-all length for the given tube surface.

single pass. Regenerator shell must be designed for higher pressures as will be explained later.

A regenerator built to shorten over-all length by using U tubes is shown schematically in Fig. 9-61. Compressed air flows inside tubes, and turbine exhaust outside is confined in the shell. In effect both gases make single passes.

All these regenerators are arranged for counterflow. Figures 9-12, 9-13, 9-26, and 9-31 illustrate regenerators built for cross-flow arrangement in which the compressed air flows through the tube interiors, and turbine exhaust flows around the tube exteriors in a direction perpendicular to their length.

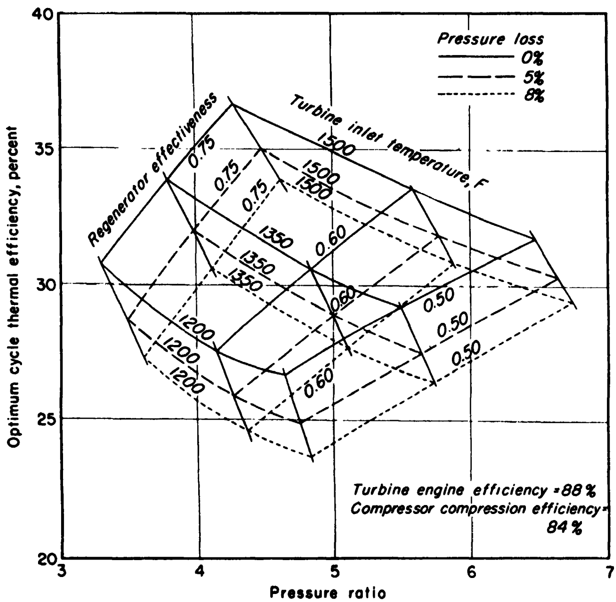


FIG. 9-62. Effect of pressure loss on optimum over-all thermal efficiency of gas-turbine plants in terms of pressure ratio, turbine temperature, and regenerator effectiveness as calculated by Dr. Eric Lype, Institute of Gas Technology.

Since all gas-turbine cycles are sensitive to pressure losses, heat exchangers, piping, and furnaces must be designed to keep pressure drops to an economical minimum. Figure 9-62 shows the effect of pressure loss on optimum cycle thermal efficiency. As pressure losses increase, thermal efficiency decreases and the optimum pressure ratio increases.

Auxiliary Equipment. Because large air and gas volumes must be handled, gas-turbine-plant piping is large in comparison to other equipment. To conserve costly alloys, h-p high-temperature piping often takes the form of double-wall construction. A thin inner tube of alloy steel serves to guide the hot gas. This tube is surrounded by insulation held in a conventional steel pipe strong enough to withstand gas pressure. Insulation reduces temperature so that outer piping performing pressure-resisting duty is not overheated.

Piping joints and supports must be designed so that relatively large thermal expansions can occur without causing any misalignment of equipment.

Gas turbines require some form of starting device, usually a motor. Its capacity need only be about 5 per cent of plant rating. The starting motor usually brings the compressor and turbine up to about one-third normal speed, when furnace can be lit off (Fig. 9-63). Turbine output is then enough to drive the compressor and bring the unit up to speed by increasing the fuel-flow rate.

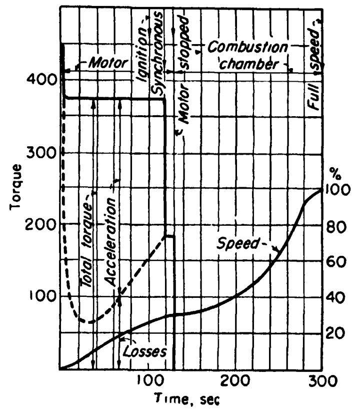


FIG. 9-63. Motor torque requirements in terms of starting time for a gas-turbine plant. From about 30 per cent rated speed, the turbine can accelerate itself.

Some of the gas turbines are being fitted with turning gears to prevent kinking the shaft during cooling-off and shutdown periods.

Control and lubricating devices must also be provided, these usually include main and auxiliary lubricating pumps, filters, tanks, oil coolers. Other important items are speed and overspeed governors, fuel pumps and nozzles, ignition devices, pressure, temperature, and flow instruments, generator electrical controls, and complementary equipment. For intercoolers and oil coolers there will have to be necessary piping systems and pumps for water.

Closed Cycles. Closed-cycle gas-turbine plants have several advantages over open-cycle plants that we have been discussing. Figure 9-64 compares the open-cycle circuit with that for a closed cycle, both are of the simple cycle variety without regeneration, reheating, or intercooling. In the open cycle, combustion products flow through the turbine; in the closed cycle, instead of a combustion chamber, there is an air heater. A separate furnace (as in a boiler) confines the fire and transmits the heat through tubes to warm the working air as it leaves the compressor under pressure. Thus only clean air passes through the turbine and eliminates any fouling dangers. As this air leaves the turbine, it could be exhausted to atmosphere and fresh air from atmosphere introduced into the compressor. However, by confining the working air in a closed circuit, another advantage can be realized. The entire circuit can be pressurized; *i.e.*, if the turbine exhaust and compressor inlet pressures are kept at 100 psia and the cycle works at a pressure ratio of 10, then the air pressure in the heater would be 1,000 psia. By using this denser air, the physical size of equipment reduces remarkably and heat-transfer rates in the air heater, intercoolers, and regenerators improve considerably.

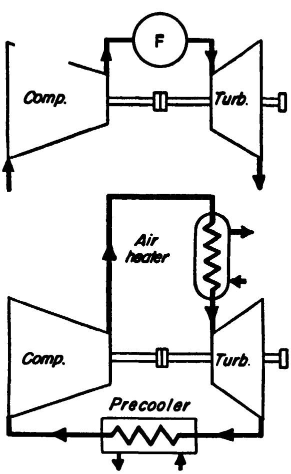


FIG. 9-64. Comparison of open and closed simple-cycle gas-turbine circuits. Closed cycle needs additional equipment in form of precooler, *i.e.*, an additional heat exchanger. The air heater form is more complex than that of the combustion chamber. The closed cycle can operate at higher air pressures.

The closed arrangement requires a precooler between the turbine exhaust and the compressor inlet to get rid of the unavailable energy by cooling the working air to the lowest possible temperature. Hence the closed cycle needs cooling water. On the other hand the closed cycle can burn any type of fuel, coal, oil, gas, wastes, etc.

Plant output is changed by raising or lowering the working medium density. This is done by adding or removing air mass from the circuit. With less pounds of air circulating, less work will be done, but temperatures can be kept at a maximum which gives good part-load thermal efficiency. With essentially constant-volume flow conditions, turbine and compressor blading can be designed for best-point performance rather than average.

Disadvantages of the closed cycle are the more complex air heater and the precooler that are needed. In addition, practically all of the heat-transfer surface in the heater works at high temperatures. This calls for the use of expensive alloys. The simpler boiler metals cannot be used to so great an extent in an air heater.

Figure 9-65 shows the circuit of an experimental 2,000-kw closed-cycle plant built by Escher-Wyss of Switzerland. At 1268 F with a pressure ratio of about 4 (85 to 343 psia), this plant showed an efficiency of 31.5 per cent at full load and about 25 per cent at one-quarter load. This plant uses a regenerator and three stages of compression with two steps of intercooling. The h-p turbine drives the three compressors, and the l-p turbine the load.

Figure 9-66 shows the design of the h-p turbine for a 25,000-kw plant working on 850 to 85 psia pressure ratio. The element features a double

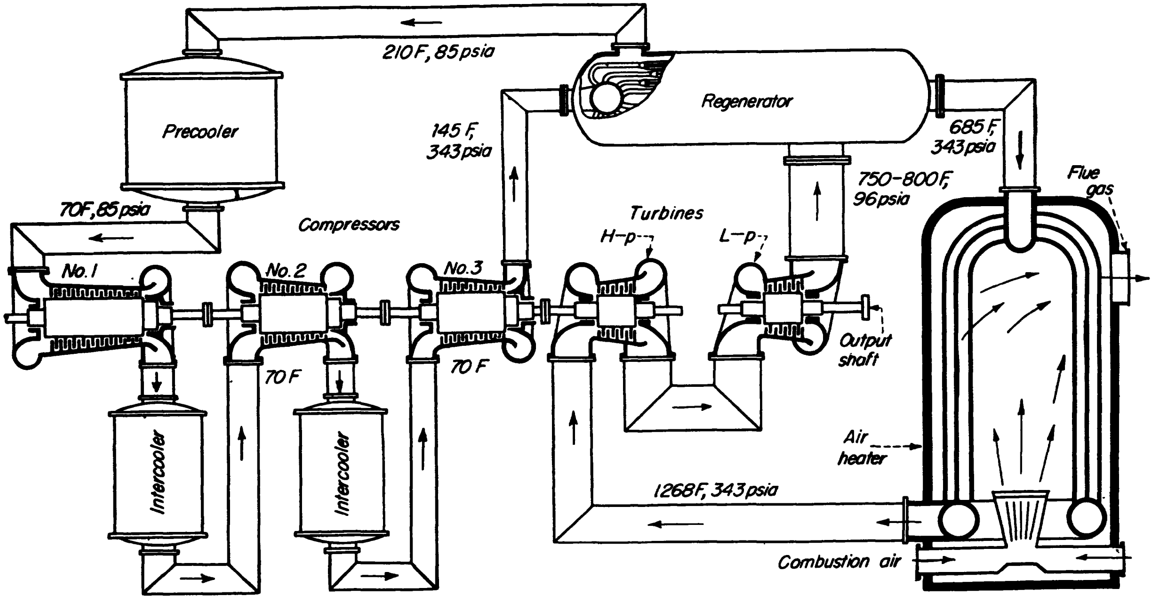


FIG. 9-65. Experimental 2,000-kw closed-cycle plant of Escher-Wyss uses three compressors with intercooling, regenerator, and two turbines without reheating. Note the air heater and precooler.

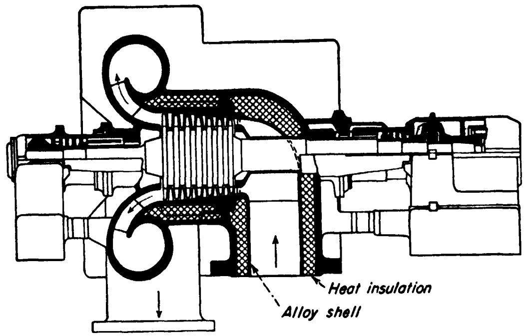


FIG. 9-66. High-pressure turbine design for 25,000-kw 850- to 85-psia closed-cycle plant features a double casing. The rotor diameter is only 24 in.

casing construction. The length of the rotor between bearings is about 10 ft, and its diameter is only 24 in.

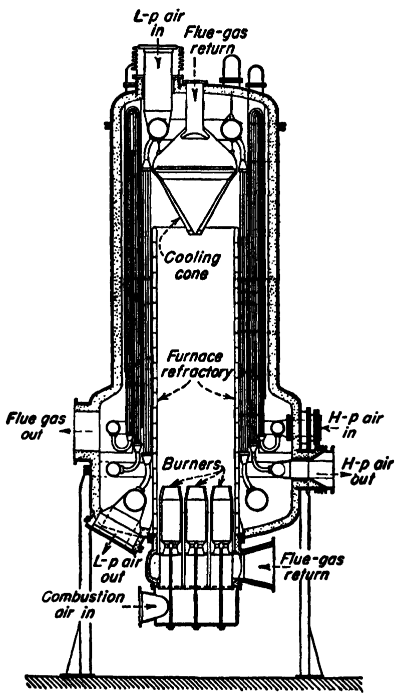


FIG. 9-67. Oil-burning two-stage supercharged air heater for closed-cycle plant of 25,000-kw capacity.

Most air-heater designs bear a superficial resemblance to modern boilers, but there are no drums, and small-diameter thin-wall tubes are used. Despite the high air-density conditions, heat-transfer conditions are poorer in air heaters than in boilers. However, an ingenious heating-surface layout reduces tube-wall temperatures in some areas and substantially decreases the amount of surface that must be built of alloy steels. Air-heater flue gas may be used to preheat combustion air, but this causes more severe tube conditions because of the higher temperatures.

Air-heater layout depends on the nature of the fuel. Figure 9-67 shows a supercharged oil-fired air heater, with reheat circuit included, for a 25,000-kw plant. Tube diameters vary from ¾ to 1½ in.

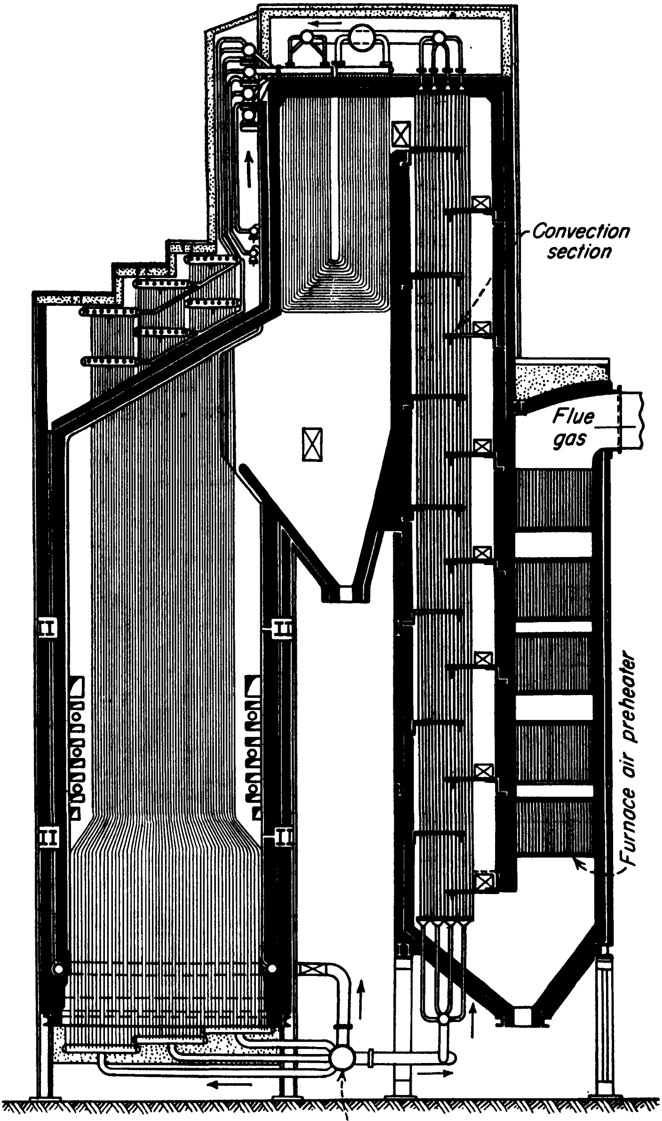


FIG. 9-68a. Pulverized-coal-fired design for closed-cycle plant has combustion-air preheater.

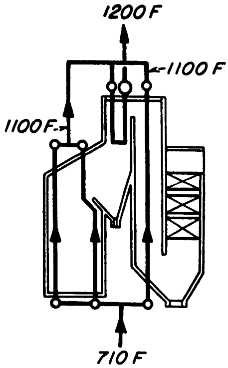


FIG. 9-68b. Circuit diagram of air flow through the heater (Fig. 9-68a).

The air heater in Fig. 9-68 burns pulverized coal at atmospheric pressure to supply a 12,000-kw plant. The small diagram shows air-flow circuits through the heater and illustrates the way in which heating-surface layout using counterflow and parallel-flow transfer is used to reduce the amount of surface made of expensive alloys. Returning part of the flue gas to the furnace holds the temperature entering the convection section to 1850 F. Flue gas preheats the combustion air.

Semiclosed Cycles. In order to gain partial advantages of smaller equipment and to reduce air-heater costs, there have been several proposed semiclosed cycles. Figures 9-69 and 9-70

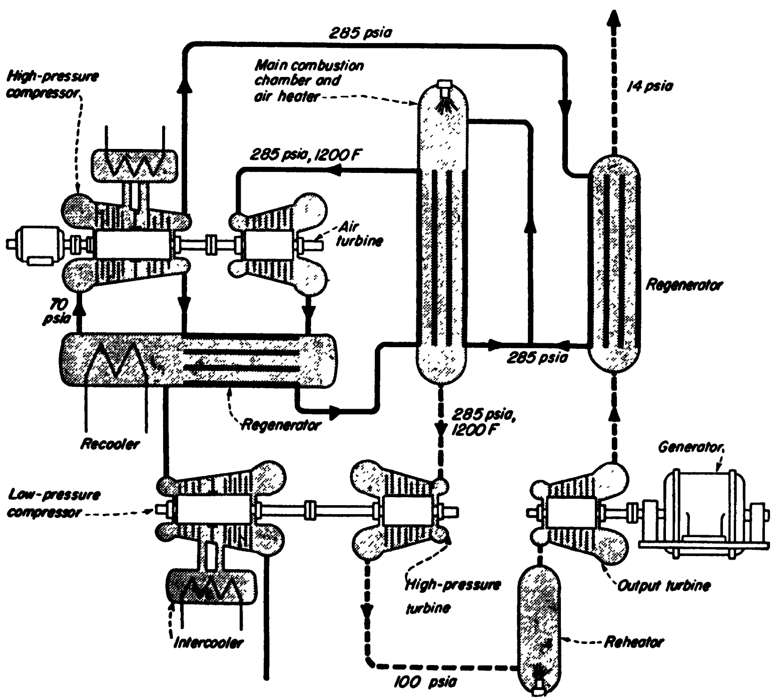


FIG. 9-69. Circuit diagram of semiclosed-cycle plant being built by Sulzer Brothers for 20,000-kw rating. The plant consists of two circuits, one handling only clean air (solid lines) and the other combustion gases (dashed lines).

show such a 20,000-kw plant being built by Sulzer Brothers of Switzerland for a Swiss utility system. Operating on 285-psia top pressure and 1200 F, the efficiency is expected to exceed 30 per cent when heavy fuel oil is burned.

The closed portion of the cycle handles clean air only. This consists of a two-cylinder intercooled compressor driven by an air turbine, regenerator, and a precooler (here called a "recooler") which restores the

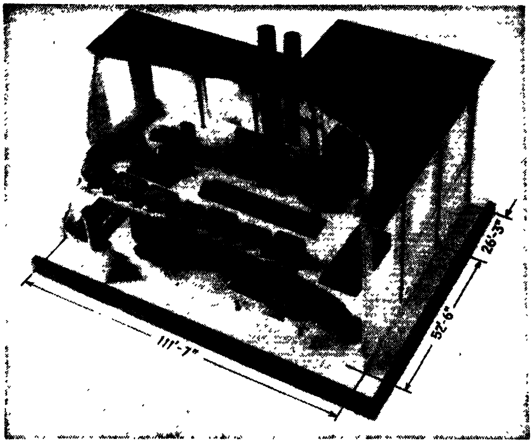


FIG. 9-70. Model of 20,000-kw semiclosed-cycle plant. Compressors and driving turbines in the foreground, turbine-generator and upper part of combustion chambers in the background, coolers and regenerators at ground level.

circulating air to its original temperature. Air from this circuit supports combustion in the air heater, raises the closed-circuit air temperature indirectly, and supplies combustion products to two gas turbines, one driving the compressor pumping atmospheric air into the closed cycle and the other driving a generator or load. Gas turbines are in series with a reheater between.

The air-heater and combustion-chamber combination, in which fuel is burned in the working air, operates with equal pressure inside and outside the tubes, eliminating stresses in tube metal working at high temperature. High pressure on the gas side increases the heat-transfer rate The net effect is a considerable saving in air-heater space requirements.

Free-piston Compressor Cycle. This type of plant is undergoing development both here and abroad. A schematic layout for a plant using three free-piston compressors, auxiliary charging set, and main gas turbine is shown in Fig. 9-71. Let's follow the air and gas flow to see how it works. Atmospheric air is picked up by the charging compressor

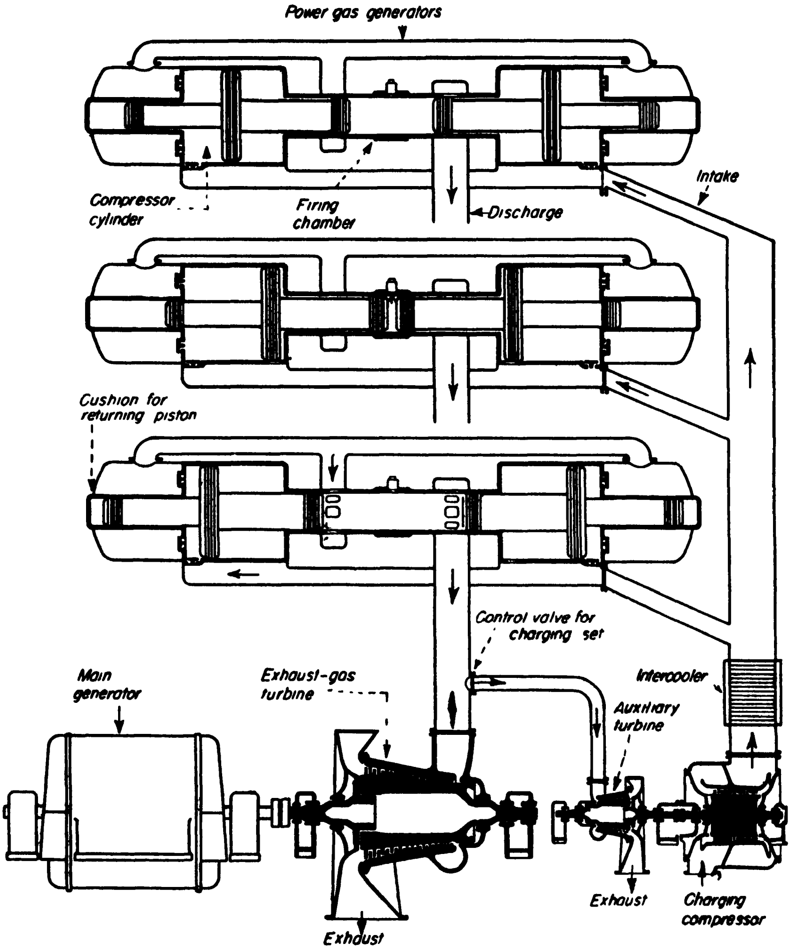


FIG. 9-71. In 7,000-hp gas-cycle unit built by Sulzer Brothers, three gas generators supply one output turbine. A precompressor driven by an auxiliary gas turbine handles l-p large-volume compression efficiently and increases the plant capacity.

and led to the intake valves of the reciprocating compressors. On the suction strokes, this air enters the compressor cylinders, has its pressure raised, and is then discharged into the lines connecting the compressor discharge valves and the firing chamber or diesel cylinder intake ports. When the pistons have traveled to their extreme outer positions, the compressed air enters the cylinder and scavenges out the products of combustion of the previous charge.

The pistons of the firing chamber then approach each other, further compressing the air. At the innermost position of the pistons, fuel is injected into this highly compressed air which immediately ignites because of the high temperature of the air. The expanding fuel and air, because of the pressure produced by burning, push the diesel pistons apart, thereby compressing air charges entrapped in the air-compressor cylinders, then at the outermost position the combustion gases still under considerable pressure and temperature (75 psia, 930 F) discharge through the right-hand cylinder ports and enter the gas turbines. The exhaust-gas turbine produces the net output of the cycle. The charging turbine-

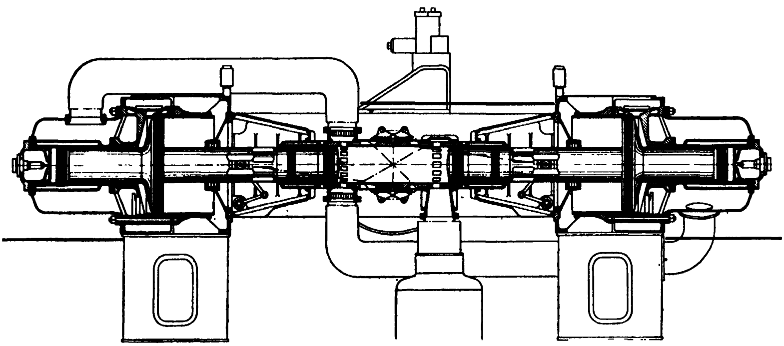


FIG. 9-72. Power-gas generator (free-piston compressor) of Sulzer Brothers design comprises diesel and air piston assemblies in opposed arrangement. A crank interconnection actuates and times the fuel injection.

compressor set serves to increase the capacity of the plant by raising the weight of each charge of air entering the compressor cylinders.

This cycle promises to develop the highest efficiency of all because the burning gases at 3500 to 4000 F do work in compressing the air. The higher the temperature of the *working* gas, the more efficient the cycle can be. Since this high temperature occurs only intermittently, as in an internal-combustion engine, the average temperature of the diesel cylinder is of an order under which the metal can live for a reasonable length of time.

Details of a free-piston compressor as built by Sulzer Brothers is shown in Fig. 9-72. Appearance of one of the compressor-diesel-piston assemblies can be seen in Fig. 9-73. Diesel piston assemblies reciprocate in opposite directions, actuating air-compressor and bouncing or cushioning pistons. Burning fuel supplies the motive energy for the compressor (and total cycle). At working stroke termination, expansion of the cushioning air and the clearance air in the compressor cylinder returns the pistons to innermost position. All compressor cylinder output goes

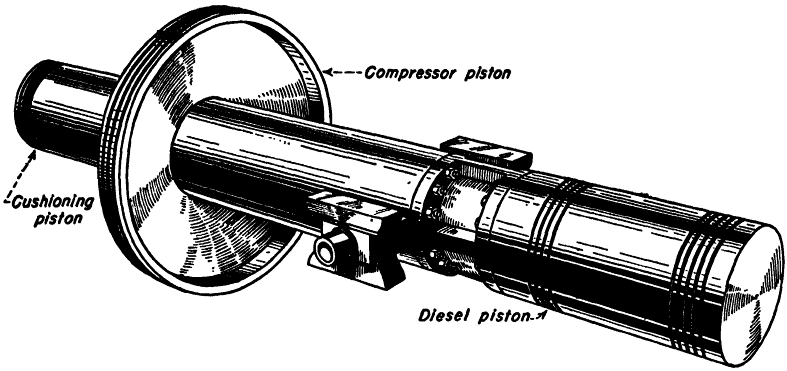


FIG. 9-73. Compressor and engine piston assembly of free-piston compressor.

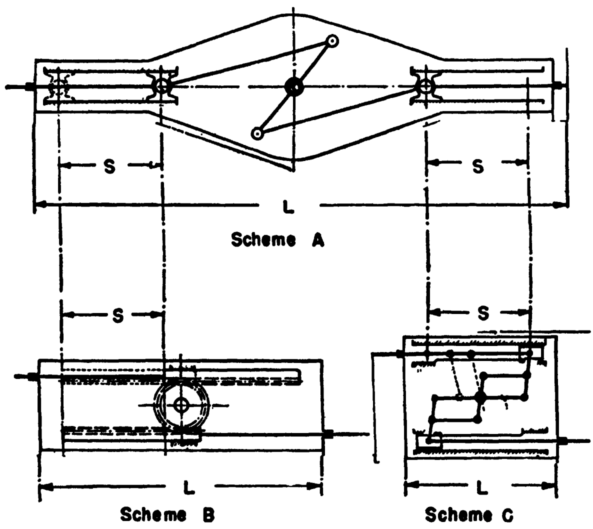


FIG. 9-74. Arrangements of synchronizing piston movements and fuel injection. *A* utilizes a crank for piston stroke *S* needing total length *L*. Scheme *B* uses rack and pinion shortens the length needed. Scheme *C* utilizing pantograph requires the shortest length of the three.

to scavenge and supercharge the diesel cylinders. The piping between the compressor and the diesel cylinders acts as a receiver to smooth out pulsations. Pressures of diesel-cylinder inlet air and discharge gas are almost equal.

The pistons are crank-interconnected as shown in scheme *A* of Fig. 9-74.

Incidental deviations from symmetrical positions should be prevented. The crank rods serve to take up secondary disturbing forces. Gas forces are inherently balanced, only slight differences being due to variations in valve tightness, port areas and lengths, and friction. These synchronizing cranks balance out these controlling factors. Another useful

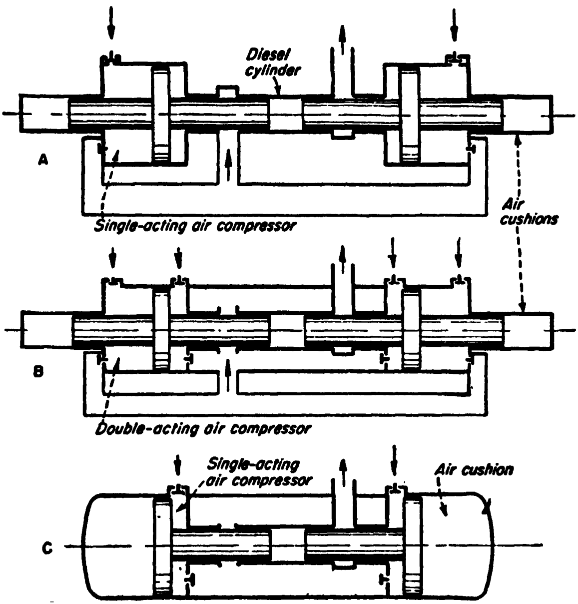


FIG. 9-75. Three possible schemes for free-piston compressor operation. A is a single-acting compressor with small-volume air cushions; B, a double-acting compressor with small-volume air cushions; C, a single-acting compressor with large-volume air cushion doing compression work on the return stroke.

function is the timing of the fuel injection at the proper point of the stroke.

Scheme B uses a rack and gear arrangement to synchronize the pistons and actuate the fuel injection system, while scheme C has the virtue of requiring the least length L for a given stroke S by using a pantograph arrangement.

Three possible arrangements of a free piston compressor are shown in Fig. 9-75. The Sulzer unit is shown in A, a double-acting arrangement is shown in B, and a scheme using large cushioning-air chambers is shown in C. In the last, the working air is compressed on the return stroke by expansion of the cushioning air. This scheme is used in a free-piston unit designed and built by Pescara of France and shown in Fig. 9-76.

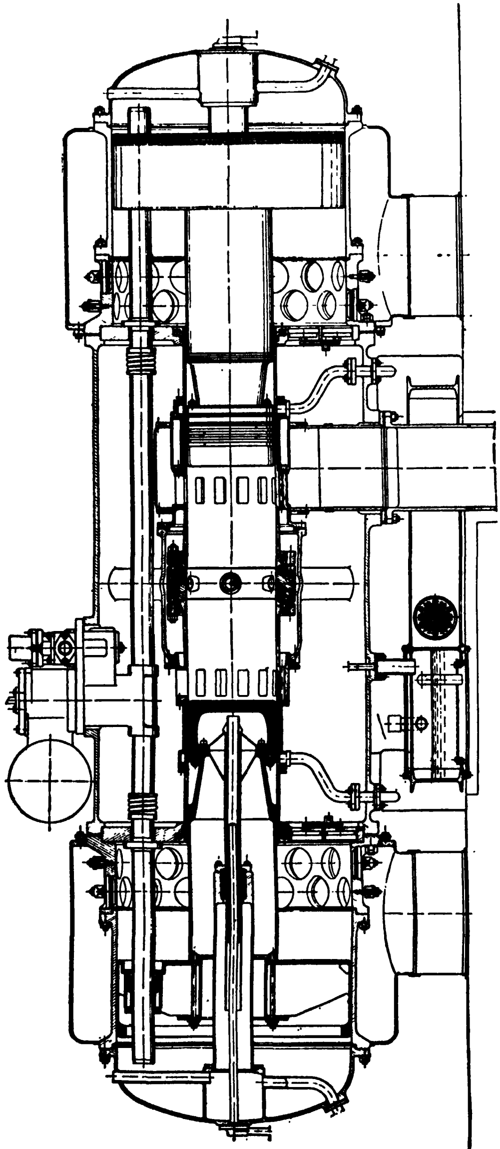


FIG. 9-76. Free-piston compressor as designed by Pescara of France, using the basic scheme *C*, shown in Fig. 9-75.

This permits making the machine shorter, and only a low pressure is needed for cushioning because of the large piston area. The main advantage lies in the total compressor work accumulating during the outward stroke, producing a large force acting backward. The acceleration of inward motion is high and the number of strokes increased.

In Fig. 9-76 the supporting center piece is formed by a barrel-shaped casing of welded steel, having the same diameter as the compressor cylinders at each end. In the central casing is the watercooled engine cylinder. The remaining annular space being only partly taken up by the chamber for the synchronizing rods, it is sealed to serve as a receiver for the scavenging and supercharging air. Air from compressor cylinders discharges into receiver space through pressure valves in the compressor covers.

The sliding surfaces of the engine and compressor cylinders carry the piston weight. A guiding pipe through the entire length of the machine provides an open passage between the two air-cushion chambers, equalizing the air pressure and, at the same time, preventing any rotational movement of the pistons. The arrangement for cooling the engine pistons needs only a single stuffing box at each end. The air-cushion space reaches deep into the interior of the engine piston.

APPENDIX

Preferred Standards for Large 3600-Rpm 3-Phase 60-Cycle Condensing Steam Turbine-Generators

	Air-Cooled Generator	Hydrogen-Cooled Generator Rated for 0.5 Psig Hydrogen Pressure				
Turbine-Generator rating, kw	11,500	15,000	20,000	30,000	40,000	60,000
Turbine capability, kw	12,650	16,500	22,000	33,000	44,000	66,000
Generator rating, kva	13,529	17,647	23,529	35,294	47,058	70,588
power factor	0.85	0.85	0.85	0.85	0.85	0.85
short-circuit ratio	0.8	0.8	0.8	0.8	0.8	0.8
Throttle pressure, psig	600	850	850	850	850 or 1250	850 or 1250
Throttle temperature, F	825	900	900	900	900 or 950	900 or 950
Number of extraction openings	4	4	4	5	5	5
Saturation temperatures at openings at "turbine-generator rating" with all extraction openings in service, F 1st	175	175	175	175	175	175
2d	235	235	235	235	235	235
3d	285	285	285	285	285	285
4th	350	350	350	350	350	350
5th	...	...	...	410	410	410
Exhaust pressure, inches Hg abs	1.5	1.5	1.5	1.5	1.5	1.5
			Hydrogen-Cooled Generator Operated at 15 Psig Hydrogen Pressure			
Generator capability at 0.85 power factor, kva			27,058	40,588	54,117	81,176

A tolerance of plus or minus 10 F shall apply to above saturation temperatures. (Tolerances shall be unilateral so as not to reduce the spread in temperature between adjacent extraction openings.)

The "turbine capability" is guaranteed continuous output at generator terminals when the turbine is clean and operating under specified throttle steam pressure and temperature and 2-½ in. Hg abs. exhaust pressure, with full extraction from all extraction openings.

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