6.1.2
STEAM TURBINES

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DEFINITIONS

"A steam turbine may be defined as a form of heat engine in which the energy of the steam is transformed into kinetic energy by means of expansion through nozzles, and the kinetic energy of the resulting jet is in turn converted into force doing work on rings of blading mounted on a rotating part."[1]

This definition may be restated:

"A steam turbine is a prime mover which converts the thermal energy of steam directly into mechanical energy of rotation."[2]

REASONS FOR USING STEAM TURBINES

Steam turbines are used to drive pumps for a variety of reasons:

1. The economical generation of steam often requires boiler steam pressures and temperatures that are considerably in excess of those at which the steam is utilized. Steam may also be used at two or more pressure levels in the same plant. Pressure reduction can be accomplished through valves, pressure-reducing stations, or use of a steam turbine.

   Pressure reduction by using a steam turbine and thereby developing power to drive a pump permits lower utility costs. The incremental increase in steam flow and consequently in fuel costs for the same lower pressure steam heating load is in most instances less than the cost of purchased power for a motor-driven pump.
2. A pump driven by a steam turbine may be operated over a wide speed range, utilizing the turbine governor system or a separately controlled valve in the turbine or in the steam line to the turbine. Operation at variable speeds is an inherent characteristic

of steam turbines and does not require the use of special speed-changing devices, as is the case with other prime movers.

The overall efficiency of the turbine and pump unit can be optimized by operating at reduced speeds and at the resultant reduced power ratings. Pump performance can be controlled by reducing the speed of the pump rather than throttling it. Although the turbine efficiency normally declines when operating at a reduced speed, the steam flow will still be less than when the pump is throttled.

Operation at reduced power but at constant speed is also permitted by the speed governor, which throttles the steam to the nozzles as the power is reduced. Efficiency may be improved by equipping the turbine with auxiliary steam valves that are closed for reduced power operation. Closing these valves reduces the available nozzle area and reduces the pressure drop across the governor valve.

When the turbine is operated with the auxiliary steam valve closed, the steam flow will approximate that for the same turbine designed for the reduced rating.

3. The use of a steam turbine driver permits the driven pump to operate essentially independent of the electric power or distribution system. The steam turbine is not affected by electric power stoppages or interruptions and is therefore ideal for critical pumping operations.

4. A turbine may be used as a secondary driver for a pump; it may also drive an independent standby or emergency pump. The particular plant design may not afford sufficient steam for the pump to be normally driven by the steam turbine. However, in the event of an electric power failure or power system disturbance, a steam turbine may be employed as a dual drive or to drive a separate pump to assure continued operation of the plant until the electric power system is again operable.

5. The steam turbine controls—governor system and overspeed trip system—are inherently sparkproof. Consequently, steam turbines can be readily applied to drive centrifugal pumps in a wide variety of hazardous atmospheres without entailing additional cost for explosionproof or sparkproof construction.

6. Steam turbines can normally be readily altered to accommodate an increase in rating for increased pump output or for new pump applications. This inherent flexibility of a steam turbine also permits it to be readily altered to accommodate changes in the initial steam pressure and temperature and in the exhaust steam pressure at which the turbine operates.

7. Steam turbines have a starting, or breakaway, torque of approximately 150 to 180% of the rated torque. Additional starting torque can be readily furnished by designing the turbine for the additional required steam flow—and without reducing the efficiency at the normal operating rating by using an auxiliary steam valve. The additional starting torque can often be obtained without increasing the turbine frame size.

8. Steam turbines can be used to drive all types of pumps.

9. Steam turbines are inherently self-limiting with respect to the power developed. Special protective devices do not have to be furnished to prevent damage to the turbine because of overload conditions. The maximum power that can be developed by a turbine is a function of the flow areas provided in the design of the nozzle ring and governor valve. Application of a load greater than that which can be developed by the turbine causes the turbine to slow down to a speed at which the torque generated by the turbine matches that required by the pump.

10. When the pump application requires the driver to be designed with excess power or to permit operation of the pump "at the end of the curve," the steam turbine can be designed for the corresponding rating without reducing the turbine efficiency when operating at the normal rating. Closing an auxiliary steam valve furnished for operation at the normal rating preserves efficiency because the turbine governor valve is not throttling to obtain the power rating.

11. With respect to the operation of the various types of pump drivers and their supporting systems, steam turbines afford minimum maintenance, low vibration, and a quiet installation.

TYPES OF STEAM TURBINES

Single-Stage Turbines A single-stage steam turbine is one in which the conversion of the kinetic energy to mechanical work occurs with a single expansion of the steam in the turbine—from inlet steam pressure to exhaust steam pressure.

A single-stage turbine may have one or more rows of rotating buckets that absorb the velocity energy of the steam resulting from the single expansion of the steam.

Single-stage turbines are available in wheel diameters of 9 to 28 in (22 to 71 cm). The overall efficiency of a turbine for a particular operating speed and steam conditions is normally dependent on the wheel diameter. The efficiency will generally increase with an increase in wheel size, and therefore the steam rate will be less (for the more usual speeds and steam conditions).

The larger-wheel-diameter steam turbines can be furnished with more nozzles to provide increased steam flow capacity and consequently greater power capabilities. The larger-wheel-diameter turbines are therefore furnished with larger steam connections, valves, shafts, bearings, and so on. Consequently the size of the turbine will generally increase with increases in power rating.

Multiple-Stage Turbines A multiple-stage turbine is one in which the conversion of the energy occurs with two or more expansions of the steam in the turbine. The number of stages (steam expansions) is a function of three basic parameters: thermodynamics, mechanical design, and cost. The thermodynamic considerations include the available energy and speed. The mechanical considerations include speed, steam pressure, steam temperature, and so on, most of which are material limits. Cost considerations include the number, type, and size of the stages; the number of governor-controlled valves; the cost of steam; and the number of years used as a basis for the cost evaluation.

The two factors generally used in selecting multistage turbines are initial cost and steam rate. Because these two factors are a function of the total number of stages, the application becomes a factor of stage selection. The initial cost increases with the number of stages, but the steam rate generally improves.

Multiple-stage turbines are normally used to drive pumps when the cost of steam or the available supply of steam requires turbine efficiencies greater than these available with a single-stage turbine, or when the steam flow required to develop the desired rating exceeds the capability of single-stage turbines.

Multiple-stage turbines can be furnished with a single or multiple governor valves. A single governor valve is often of the same design whether used in a single-stage or multiple-stage turbine and generally has the same maximum steam flow, pressure, and temperature parameters. Multiple valves are used when the parameters for a single valve are exceeded or to obtain improved efficiency, particularly at reduced power outputs.

Shaft Orientation Some steam turbines, particularly single-stage turbines, can be furnished with vertical downward shaft extensions. The application of such turbines can require considerable coordination between the pump and the turbine manufacturer to assure an adequate thrust bearing in the turbine, shaft length and details, mounting flange dimensions, and even shaft runout.

Vertical shaft pumps are frequently driven by horizontal turbines through a right-angle speed-reduction gear unit.

Direct-Connected and Geared Turbines Steam turbines can be directly connected to the pump shaft so the turbine operates at the pump speed or can drive the pump through a speed-reduction (and even speed-increase) gear unit, in order to permit the turbine to operate at a more efficient speed.

Turbine Stages The two types of turbine stages are impulse and reaction. The turbines discussed in this subsection employ impulse stages because steam turbines driving pumps normally have impulse-type stages.

In the ideal impulse stage, the steam expands only in the fixed nozzles and the kinetic energy is transferred to the rotating buckets as the steam impinges on the buckets while flowing through the passages between them. The steam pressure is constant, and the steam velocity relative to the bucket decreases in the bucket passages.

In a reaction stage, the steam expands in both the fixed nozzles and the rotating buckets. The kinetic energy is transferred to the rotating buckets by the expansion of the steam in the passages between the buckets. The steam pressure decreases as the steam velocity relative to the buckets increases in the bucket passages.

In an impulse stage, the steam can exert an axial force on the buckets as it flows through the blade passages. Although this force is usually referred to as a reaction, the use of the term does not imply a reaction-type stage.

The larger buckets used in the last stages of an impulse-type multistage turbine can be of a free-vortex design—twisted and tapered. Such a bucket is ideally subjected to a nearly pure impulse force at its root and a nearly pure reaction force at its tip, but, in reality, this bucket is a high reaction-design bucket compared with a normal impulse-stage bucket. A steam turbine stage with such a bucket design is still referred to as an impulse stage because the primary conversion of kinetic energy is by a reduction rather than an increase in relative steam velocity.

A reaction turbine has more stages than an impulse turbine for the same application because of the small amount of kinetic energy absorbed per stage, and requires a larger thrust bearing or a balancing piston because of the pressure drop across the moving blades. The small pressure drop per stage and the pressure drop across the moving blades require that the steam-leakage losses be minimized by elaborate sealing between the tips of the nozzle blades and the rotor, and the tips of the moving blades and the casing.

The small pressure differential across the rotating blades of an impulse stage results in smaller thrust bearings and no close blade-tip clearances. Consequently impulse turbines can be started more quickly without thermal-expansion damage, and their stage efficiencies remain relatively constant over the life of the turbine.

CONSTRUCTION DETAILS

Component Parts The main components of a single-stage steam turbine are shown in Figure 1.

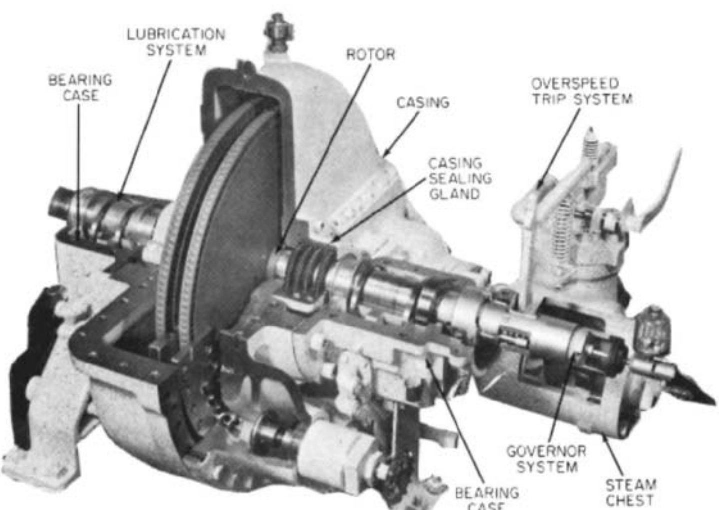

FIGURE 1 Component parts of steam turbines (Elliott)

Function and Operation—Single-Stage Turbine The *steam* chest and the *casing* contain the steam furnished to the turbine, being connected to the higher-pressure steam supply line and the lower-pressure steam exhaust line, respectively. The steam chest, which is connected to the casing, houses the governor valve and the overspeed trip valve. The casing contains the rotor and the nozzles through which the steam is expanded and directed against the rotating buckets.

The *rotor* consists of the shaft and disk assemblies with buckets. The shaft extends beyond the casing and through the bearing cases. One end of the shaft is used for coupling to the driven pump. The other end serves the speed governor and the overspeed trip systems.

The *bearing cases* support the rotor and the assembled casing and steam chest. The bearing cases contain the journal bearings and the rotating oil seals, which prevent outward oil leakage and the entrance of water, dust, and steam. The steam end bearing case also contains the rotor positioning bearing and the rotating components of the overspeed trip system. An extension of the steam end bearing housing encloses the rotating components of the speed governor system.

The *casing sealing glands* seal the casing and the shaft with spring-backed segmented carbon rings (supplemented by a spring-backed labyrinth section for the higher exhaust steam pressures).

The *governor system* commonly consists of spring-opposed rotating weights, a steam valve, and an interconnecting linkage or servomotor system. Changes in the turbine inlet and exhaust steam conditions, and the power required by the pump will cause the turbine speed to change. The change in speed results in a repositioning of the rotating governor weights and subsequently of the governor valve.

The *overspeed trip system* usually consists of a spring-loaded pin or weight mounted in the turbine shaft or on a collar, a quick-closing valve that is separate from the governor valve, and interconnecting linkage. The centrifugal force created by rotation of the pin in the turbine shaft exceeds the spring loading at a preset speed. The resultant movement of the trip pin causes knife edges in the linkage to separate and permit the spring-loaded trip valve to close.

The trip valve may be closed by disengaging the knife edges manually, by an electric or pneumatic signal, by low oil pressure, or by high turbine exhaust steam pressure.

The two usual types of lubrication systems are oil-ring and pressure. The *oil-ring* lubrication system employs an oil ring(s) that rotates on the shaft with the lower portion submerged in the oil contained in the bearing case. The rotating ring(s) transfers oil from the oil reservoir to the turbine shaft journal bearing and rotor-locating bearing. The oil in the bearing case reservoirs is cooled by water flowing in cooling water chambers or tubular heat exchangers.

A *pressure* lubrication system consists of an oil pump driven from the turbine shaft, an oil reservoir, a tubular oil cooler, an oil filter, and interconnecting piping. Oil is supplied to the bearing cases under pressure. The oil rings may be retained in this system to provide oil to the bearings during startup and shutdown when the operating speed and bearing design permit.

Typical sectional drawings are shown in Figures 2, 3, and 4.

GOVERNORS AND CONTROLS

Governor systems are typically speed-sensitive control systems. The turbine speed is controlled by varying the steam flow through the turbine by positioning the governor valve. Variations in the power required by the pump and changes in steam inlet or exhaust conditions alter the speed of the turbine, causing the governor system to respond to correct the operating speed.

Control systems, unlike governor systems, are not directly speed-sensitive but respond to changes in pump or pump-system pressures and then reposition the turbine "governor" valve to maintain the preset pressure. Consequently, changes in turbine steam conditions or in the power required by the pump result in a repositioning of the turbine governor

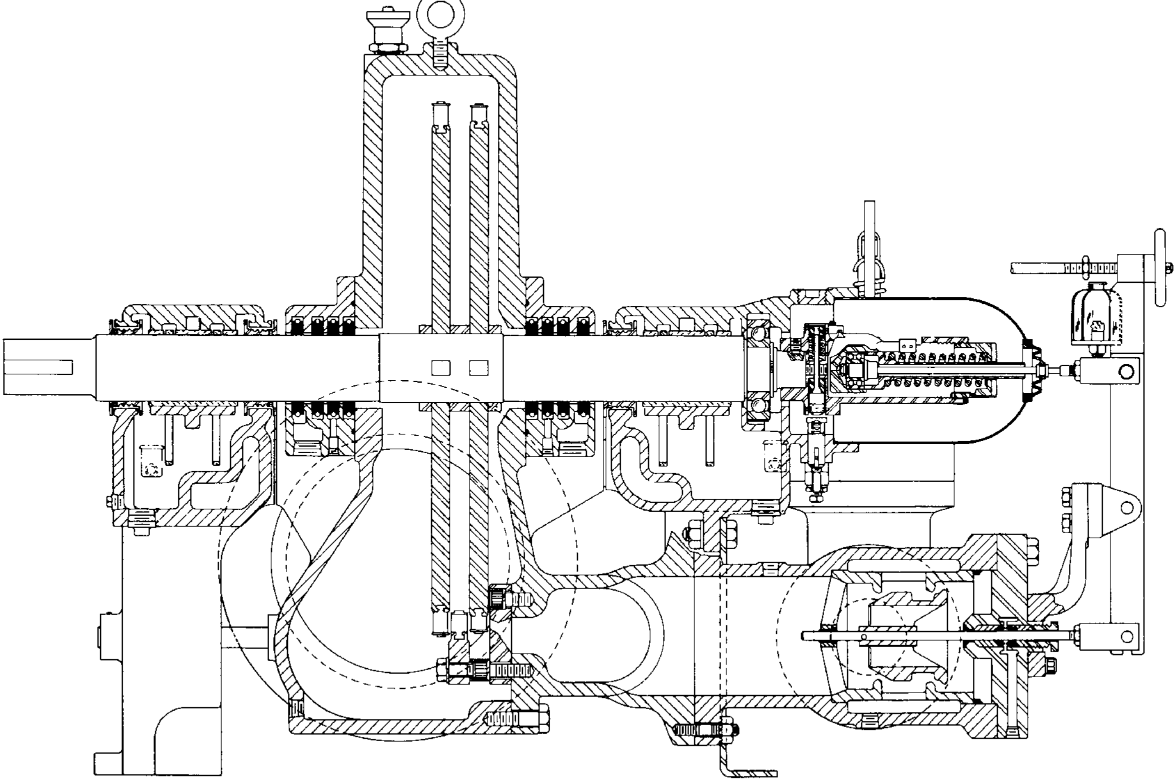

FIGURE 2 Section of single-stage turbine and governor system (Elliott)

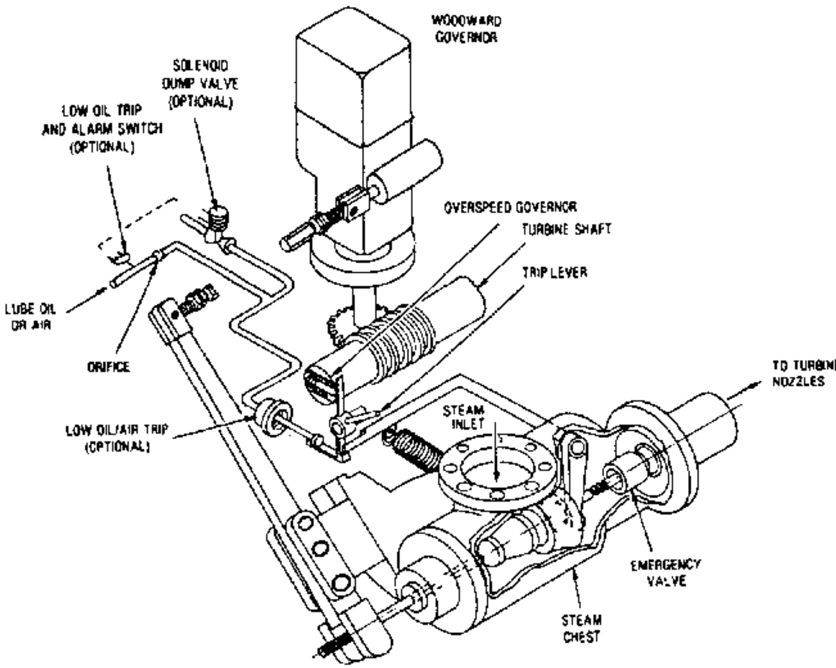

FIGURE 3 Typical turbine overspeed trip arrangement (Dresser-Rand).

valve or of a separate steam valve only after the pressure being sensed by the controller has changed.

Even when a control system is furnished, a speed governor is also normally furnished. The speed governor is set for a speed slightly higher than the desired operating speed in order to function as a pre-emergency governor; that is, prevent the turbine from reaching the trip speed when the controller causes the turbine to operate at a speed above rated speed.

Governor systems are defined by their performance as follows:[2]

Class of governor system	Speed range, % (as specified)	Maximum speed regulation, %	Maximum speed variation, %, ±	Maximum speed rise, %
A	10–65	10	0.75	13
B	10–80	6	0.50	7
C	10–80	4	0.25	7
D	10–90	0.50	0.25	7

Speed range is the percentage below rated speed for which the governor speed setting may be adjusted. For example, a turbine with 4000 rpm rated speed and a governor system having a 30% range can be operated at a minimum speed of 2800 rpm:

$$4000 - \frac{30 \times 4000}{100} = 2800$$

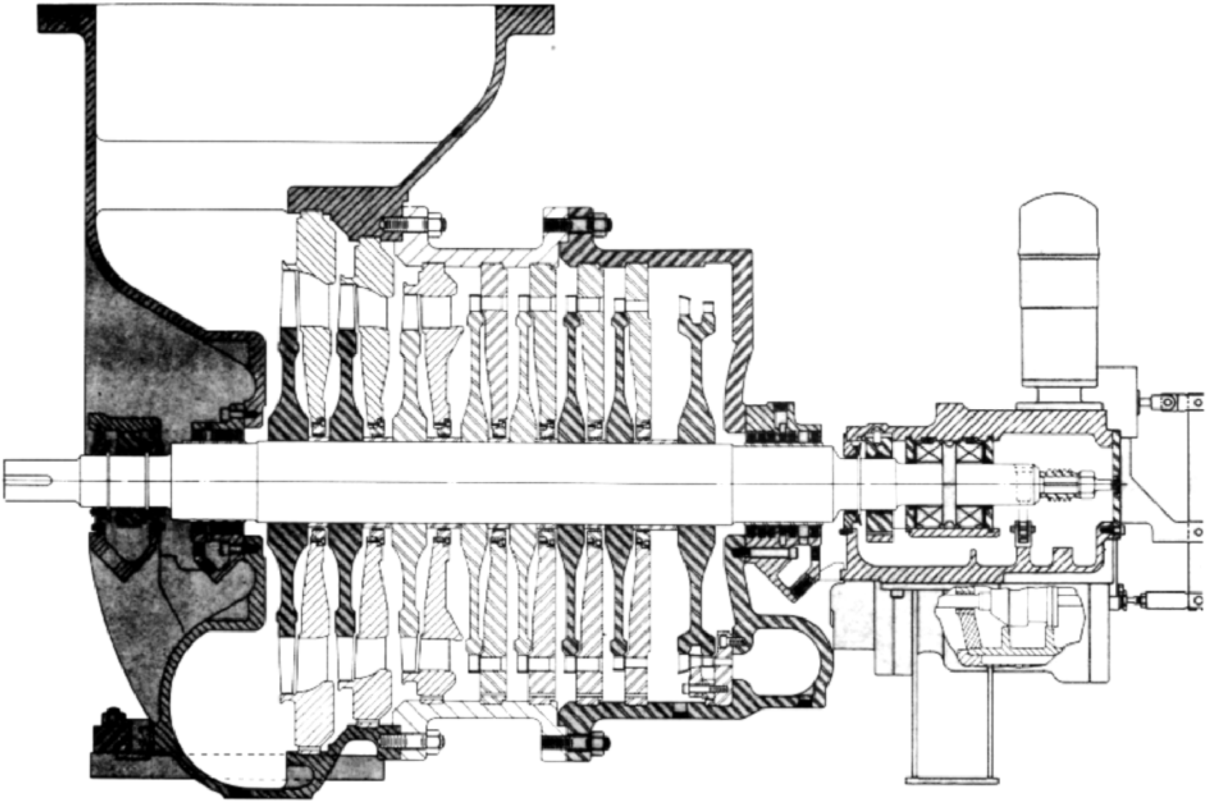

FIGURE 4 Section of multistage turbine (Elliot)

If the speed range had been specified as plus 5% and minus 25%, for example, the maximum and minimum speeds would be 4200 and 3000 rpm.

Steady-state speed regulation is the change in speed required for the governor system to close the governor valve when the load is gradually reduced from rated load to no load with turbine steam conditions constant. Regulation is always expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{No-load speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \%\ \textit{regulation}$$

Therefore

$$\text{No-load speed} = \text{rated speed}\left(1 + \frac{\%\ \text{regulation}}{100}\right)$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A speed governor system would have 4400 rpm maximum no-load speed:

$$4000 \times \left(1 + \frac{10}{100}\right) = 4400\ \text{rpm}$$

Consequently, whenever the turbine is developing less than rated power, the operating speed will be greater than rated speed, as required for the governor system to reposition the governor valve.

Speed variation, expressed as a percentage, is the total magnitude of the fluctuations from the set speed permitted by the governor system when the turbine is normally operating at rated speed, power, and steam conditions. This is the "insensitivity" of the governor system. The speed variation equation is

$$\frac{\text{Maximum speed} - \text{minimum speed}}{\text{Rated speed} \times 2} \times 100 = \pm\%\ \textit{speed variation}$$

A turbine with 4000 rpm rated speed and NEMA A governor system would have ±30 rpm maximum speed variation:

$$4000 \times \frac{\pm 0.75}{100} = \pm 30\ \text{rpm}$$

Maximum speed rise represents the momentary increase in speed when the load is suddenly reduced from rated power to no-load power with the pump still coupled to the turbine shaft. Shortly after the sudden loss of load, the governor system will cause the turbine speed to be reduced to the no-load speed.

Maximum speed rise is also expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{Maximum speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \%\ \text{speed rise}$$

Therefore

$$\text{Maximum speed} = \text{rated speed} + \frac{\text{rated speed} \times \%\ \text{speed rise}}{100}$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A governor will have a maximum speed rise to

$$4000 + \frac{4000 \times 13}{100} = 4520\ \text{rpm}$$

The setting of the *overspeed trip* is a function of the maximum speed rise of the governor system—it must be higher than the maximum speed rise. The recommended settings are

Class of governor system	Overspeed trip setting, (% of rated speed)
A	115
B	110
C	110
D	110

The overspeed trip system for a turbine with 4000 rpm rated speed and equipped with a NEMA A governor is set for operation at

$$4000 \times \frac{115}{100} = 4600 \text{ rpm}$$

The speed-sensitive portion of the speed governor system is usually a set of spring-loaded rotating weights. Movement of the weights caused by a change in turbine speed positions the governor valve through a suitable linkage.

The speed-sensitive element can also be other devices that are speed-responsive, such as a positive displacement oil pump, electric generator, or a magnetic impulse signal generator.

The rotating-weight governor system is a direct-acting type and is classified as a NEMA A governor. *Direct-acting* designates a governor system in which the speed-sensitive element also provides the power for positioning the governor valve.

The NEMA B, C, and D governor systems have speed-sensitive elements that position the governor valve through a relay or servomotor system instead of actuating the valve directly. The speed-sensitive element can therefore be more precise and sensitive, as required for the improved governor system performance.

Electronic governors with pneumatic/hydraulic actuators are also available that offer more control, flexibility, and speed adjustment as well as other operating modes.

THEORY

A steam turbine develops mechanical work by converting to work the available heat energy in the steam expansion. Heat and mechanical work, being two forms of energy, can be converted from one to the other.

The heat energy is converted in two steps. The steam expands in nozzles and discharges at a high velocity, converting the available heat energy to velocity (kinetic) energy. The high-velocity steam strikes moving blades, converting the velocity energy to work. Because the total heat energy available in the steam is converted to velocity (kinetic) energy, the magnitude of the steam velocity is dependent upon the available energy.

The mechanical work that is developed in the turbine by the high-velocity steam striking the buckets is a function of the speed of the buckets. Maximum work occurs when the bucket velocity is approximately one-half the steam jet velocity for an impulse stage and one-fourth the steam jet velocity for a velocity-compounded impulse stage. Although the steam jet velocity is fixed by the available heat energy, the bucket velocity is fixed by the speed of the turbine and the diameter of the turbine wheel on which the buckets are mounted. The work developed, or the efficiency of the turbine, ignoring losses in the turbine, is therefore determined by the size of the turbine and the turbine (pump) speed for a fixed amount of available heat energy.

The most common single-stage turbine is the velocity-compounded (Curtis) type. The complete expansion from inlet to exhaust pressure occurs in one step. The Curtis stage,

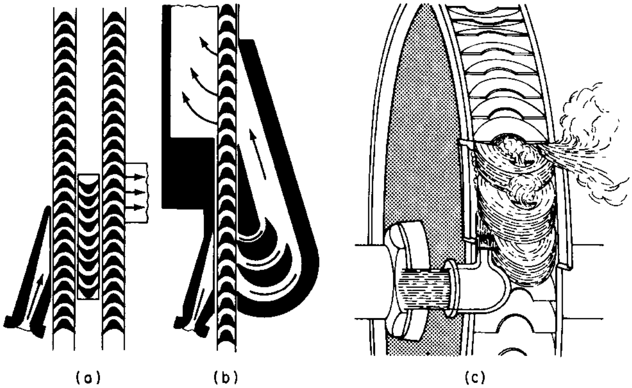

FIGURE 5 Velocity-compounded stages: (a) Curtis stage; steam flows once through moving buckets; (b) re-entry stage; steam flows twice through moving buckets; (c) re-entry stage; steam flows three times through moving buckets (reprinted with permission from *Power*, June 1962)

with two rows of rotating buckets, and two re-entry-type velocity-compounded stages are illustrated in Figure 5.

Single-stage turbines are available with a wide range of efficiencies because they are manufactured with a variety of wheel diameters: 9 to 28 in (22 to 71 cm).

Multistage turbines are manufactured with a more limited variety of wheel sizes. The efficiency of multistage turbines is varied primarily by varying the number of stages. When the total available energy of the steam results in a steam velocity greater than twice the bucket velocity (using convenient wheel sizes), a multistage turbine will be more efficient. In a multistage turbine, the total steam expansion is divided among the various impulse stages to produce the desired steam velocity for each row of buckets.

A steam turbine is normally evaluated using *steam rate*—the amount of steam required by the turbine to produce the specified power per hour at the specified speed—rather than *efficiency.* The steam rate is a direct function of the turbine efficiency.

The steam consumption can be expressed either as steam rate, pounds of steam per horsepower-hour (kilograms per kilowatt-hour) or as steam flow, pounds of steam per hour (kilograms per hour). The higher the efficiency, the lower the steam rate or steam flow, and vice versa.

The total available energy of the steam is that available from an isentropic expansion. For given initial steam pressure and temperature and exhaust pressure, the available energy in British thermal units per pound (kilojoules per kilogram) of steam can be obtained from the tables or the Mollier chart in Reference 3.

The available energy can be converted to power units and expressed as the theoretical steam rate—pounds per horsepower-hour or pounds per kilowatt-hour (kilograms per kilowatt-hour). The theoretical steam rate is the steam rate for a 100% efficient turbine and therefore can be used more conveniently than energy in British thermal units per pound (kilojoules per kilogram) for the calculation of turbine steam rates. Theoretical steam rates can be obtained directly from theoretical steam rate tables (ASME) or from the polar Mollier chart (Elliott Company).

The actual steam rate for a turbine is greater than the theoretical steam rate because of the losses that occur in the turbine when the available energy is converted to mechanical work and because of the ratio of the steam velocity to bucket velocity. The energy remaining in the steam exhausting from the turbine is greater than that after an isentropic expansion, as illustrated in Mollier diagram shown in Figure 6, where

1 = energy in steam at initial steam pressure and temperature

2 = energy in steam at exhaust pressure for an isentropic expansion

3 = actual energy in steam at exhaust pressure

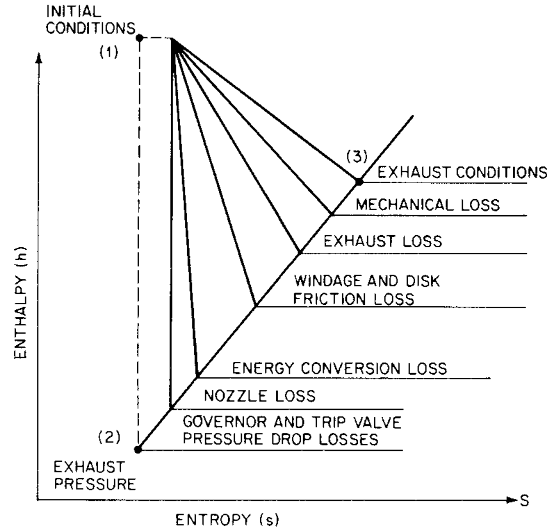

FIGURE 6 Mollier diagram with energy losses

The efficiency of the turbine is

$$\frac{h_1 - h_3}{h_1 - h_2} \quad \text{or} \quad \frac{\text{theoretical steam rate}}{\text{actual steam rate}}$$

The governor and trip valve pressure drop losses are a function of the sizes of these two valves and the steam flow. The governor valve pressure drop will vary more than the trip valve pressure drop because of changes in valve position with power required by the driven pump, speed variations, and changes in inlet and exhaust steam conditions.

The nozzle loss is due to friction in the nozzles as the steam expands. The efficiency of the nozzles is a function of the ratio of the actual and ideal exit steam velocities squared. The efficiency is usually between 95 and 99%.

The windage and disk friction losses are due to the friction between the stream and the disks and the blades fanning the steam. This loss varies inversely with the specific volume of the steam, increases with exhaust pressure, and increases with the diameter of the wheel and the length of the blades.

The use of a larger-diameter wheel may increase the efficiency, but the windage and disk friction losses will reduce the improvement and may even cause a net loss in overall efficiency.

The exhaust losses represent the kinetic energy remaining in the steam as a result of the velocity of the steam leaving the bucket and the pressure drop in the steam as it passes out the exhaust connection.

The energy conversion loss is due to the nonideal conversion of the steam velocity energy to mechanical work in the buckets as a function of the steam velocity and bucket velocity, plus nonideal nozzle and bucket angles, friction in the system, and so on.

The performance that can be expected from a single-stage Curtis-type turbine may be obtained from Figures 7, 8, and 9, and Table 1 after determining the theoretical steam rate.

$$\text{Steam rate} = \frac{\text{base steam rate}}{\text{superheat correction factor}} \times \frac{\text{power + power loss}}{\text{power}}$$

To obtain superheat, subtract temperature given in Table 1 from total initial temperature.

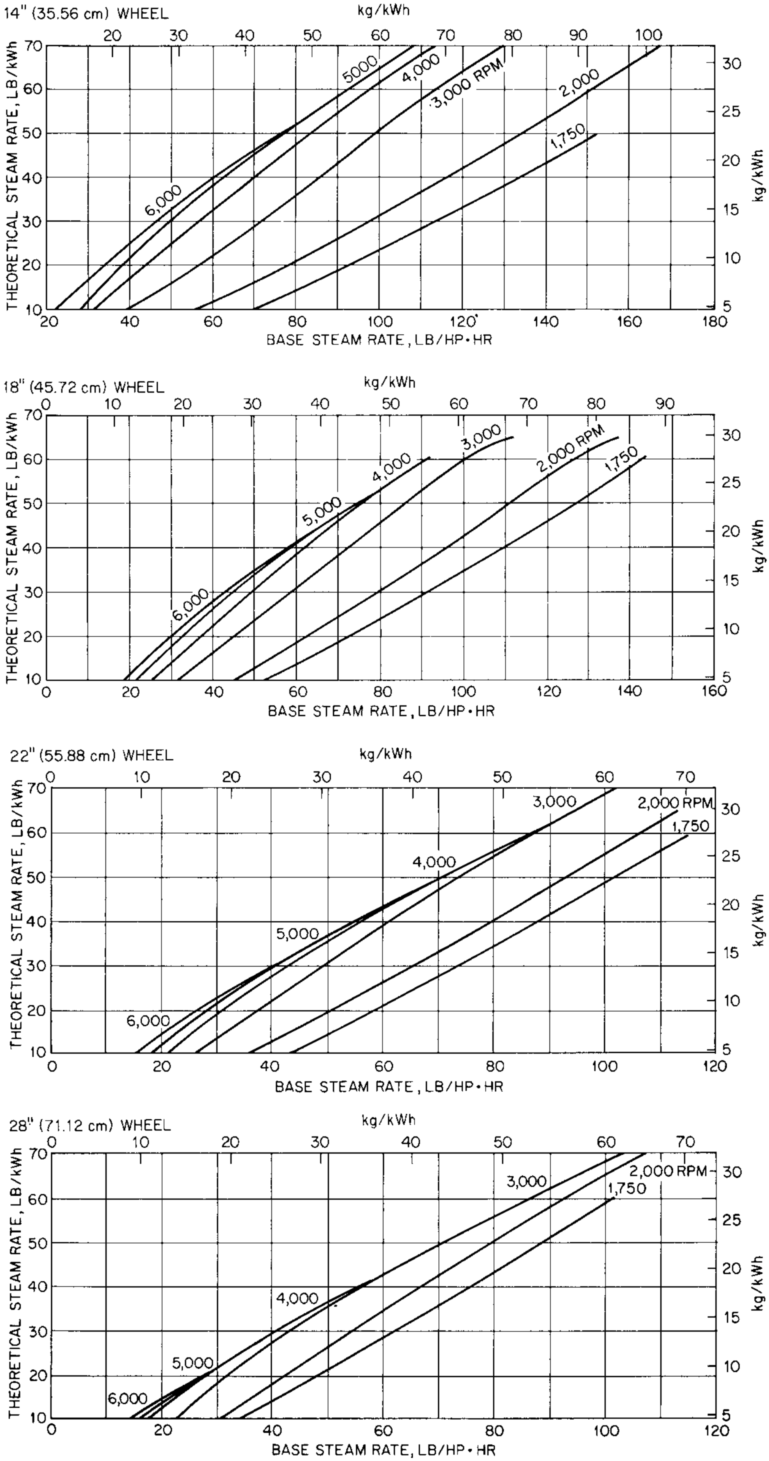

FIGURE 7 Base steam rates (Elliott)

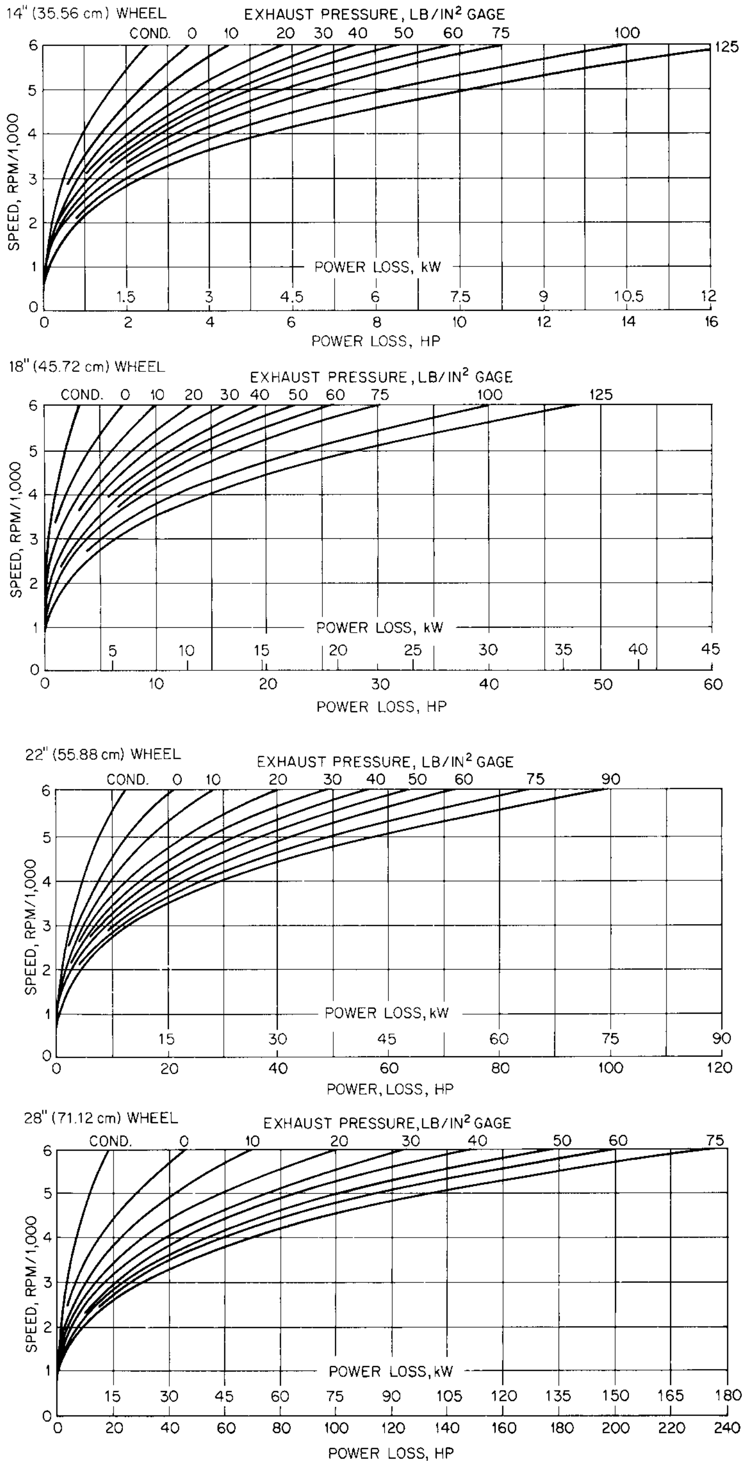

FIGURE 8 Power loss (lb/in² × 6.895 = kPa) (Elliott)

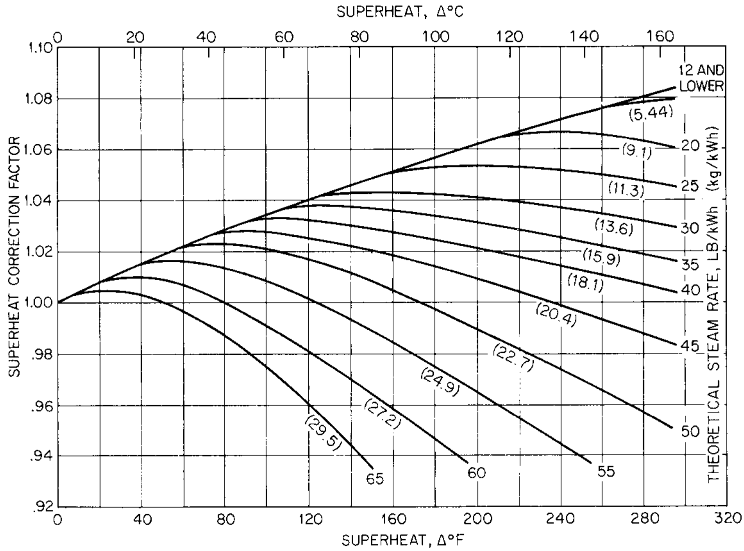

FIGURE 9 Superheat correction factor (Elliott)

SAMPLE CALCULATION

STEAM CONDITIONS 250 lb/in^2 (1724 kPa) gage inlet, 575°F (302°C) inlet, and 50 lb/in^2 (345 kPa) gage exhaust.

DESIGN CONDITIONS turbine to develop 500 hp (373 kW) at 4000 rpm.

1. Theoretical steam rate = 26.07 lb/kWh (11.82 kg/kWh)
2. Base steam rate for 28-in (71.12-cm) wheel (Figure 7) = 36 lb/hp · h (22 kg/kWh)
3. Power loss for 28-in (71.12-cm) wheel (Figure 8) = 55 hp (41 kW)
4. Temperature of dry and saturated inlet steam (Table 1) = 406°F (208°C)
5. Superheat (Table 1) = 575 – 406 = 169Δ°F (302 – 208 = 94Δ°C)
6. Superheat corrections factor (Figure 9) = 1052
7. Steam rate =

In USCS units $$\frac{36}{1.052} \times \frac{500 + 55}{500} = 38.0 \text{ lb/hp·h}$$

In SI units $$\frac{21.89}{1.052} \times \frac{373 + 41}{373} = 23.1 \text{ kg/kWh}$$

The ability of a particular size turbine to develop the required power is determined primarily by

1. The flow capacity of the inlet and exhaust connection from Figures 10 and 11, where steam flow = power × steam rate
2. The flow capacity of the nozzles available in a particular turbine (the number and size of the nozzles vary considerably with each design of turbine manufactured, and thus a meaningful plot cannot be included here).

TABLE 1 Temperature of dry and saturated steam

Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F
0	213	150	366	300	422	450	460
5	228	155	368	305	423	455	461
10	240	160	371	310	425	460	462
15	250	165	373	315	426	465	463
20	259	170	375	320	428	470	464
25	267	175	378	325	429	475	465
30	274	180	380	330	431	480	466
35	281	185	382	335	432	485	467
40	287	190	384	340	433	490	468
45	293	195	386	345	434	495	469
50	298	200	388	350	436	500	470
55	303	205	390	355	437	510	472
60	308	210	392	360	438	520	474
65	312	215	394	365	440	530	476
70	316	220	396	370	441	540	478
75	320	225	397	375	442	550	480
80	328	230	399	380	444	560	482
85	328	235	401	385	445	570	483
90	331	240	403	390	446	580	485
95	335	245	404	395	447	590	487
100	338	250	406	400	448	600	489
105	341	255	408	405	449	610	491
110	344	260	410	410	451	620	492
115	347	265	411	415	452	630	494
120	350	270	413	420	453	640	496
125	353	275	414	425	454	650	497
130	356	280	416	430	455	660	499
135	358	285	417	435	456	670	501
140	361	290	419	440	457	680	502
145	364	295	420	445	458	690	504

To obtain superheat, subtract temperature given above from total initial temperature. Pressure in kPa = 6.895 × lb/in². Temperature in °C = (°F − 32) ÷ 1.8.

EVALUATION OF COSTS

The economical selection of a turbine considers the initial cost of the turbine and the operating costs. A lower-steam-rate (more efficient) turbine generally has a higher initial cost than a turbine with a higher steam rate. The operating costs are the cost of the steam for the number of years upon which the evaluation is to be based:

$$\text{Total cost} = \text{initial cost} + \left(\text{power} \times \begin{matrix}\text{steam}\\ \text{rate}\end{matrix} \times \begin{matrix}\text{steam}\\ \text{cost}\end{matrix} \times \begin{matrix}\text{operating hours}\\ \text{per year}\end{matrix} \times \begin{matrix}\text{number}\\ \text{of years}\end{matrix}\right)$$

The cost of installation—foundation, steam piping, and cooling water service (also electric service if required)—is not normally considered unless there are significant differences between the turbines: single-stage versus multistage, direct-connected versus turbine and gear, vertical turbine versus horizontal turbine with a right-angle gear, and so on.

The economical selection of a steam turbine versus an electric motor, diesel engine, gas engine, gas turbine, and so on, must include an evaluation of all installation costs, costs of

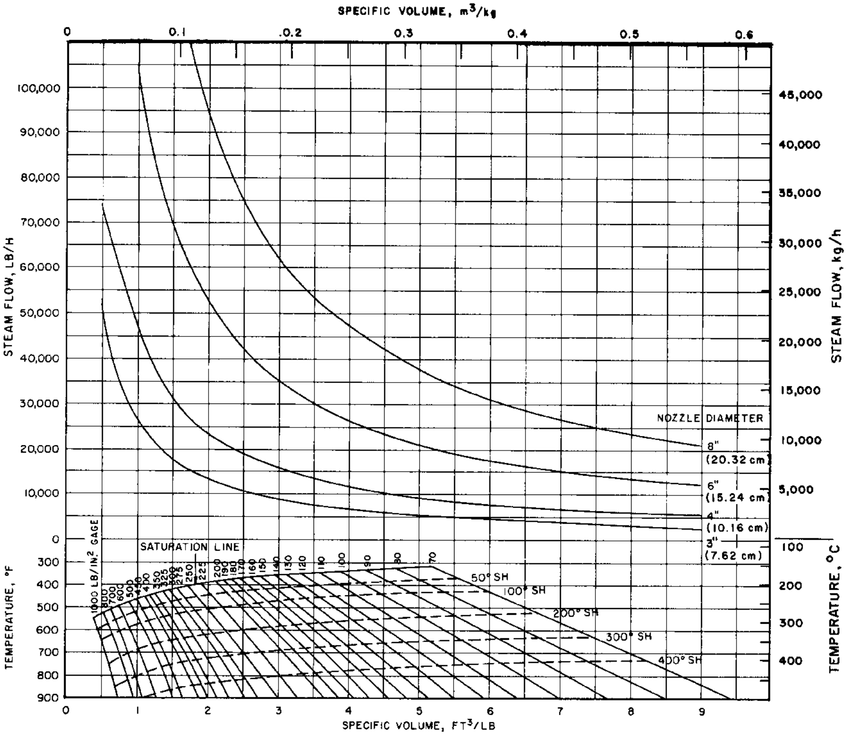

FIGURE 10 Nominal inlet flow capacity. Read inlet nozzle size required to pass maximum flow, based on 150-ft/s (45.7-m/s) steam velocity (lb/in^2 × 6.895 = kPa; °F$_{SH}$ × 0.555 = °C$_{SH}$)

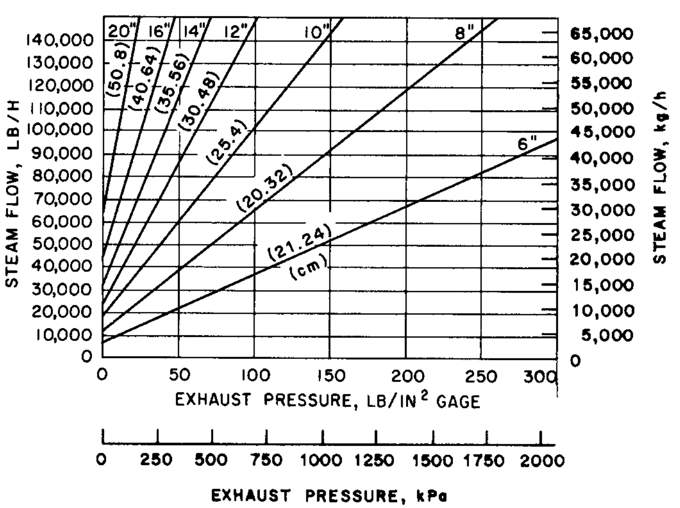

FIGURE 11 Nominal exhaust flow capacity. Noncondensing exhaust nozzles, based on 200-ft/s (61 m/s) steam velocity

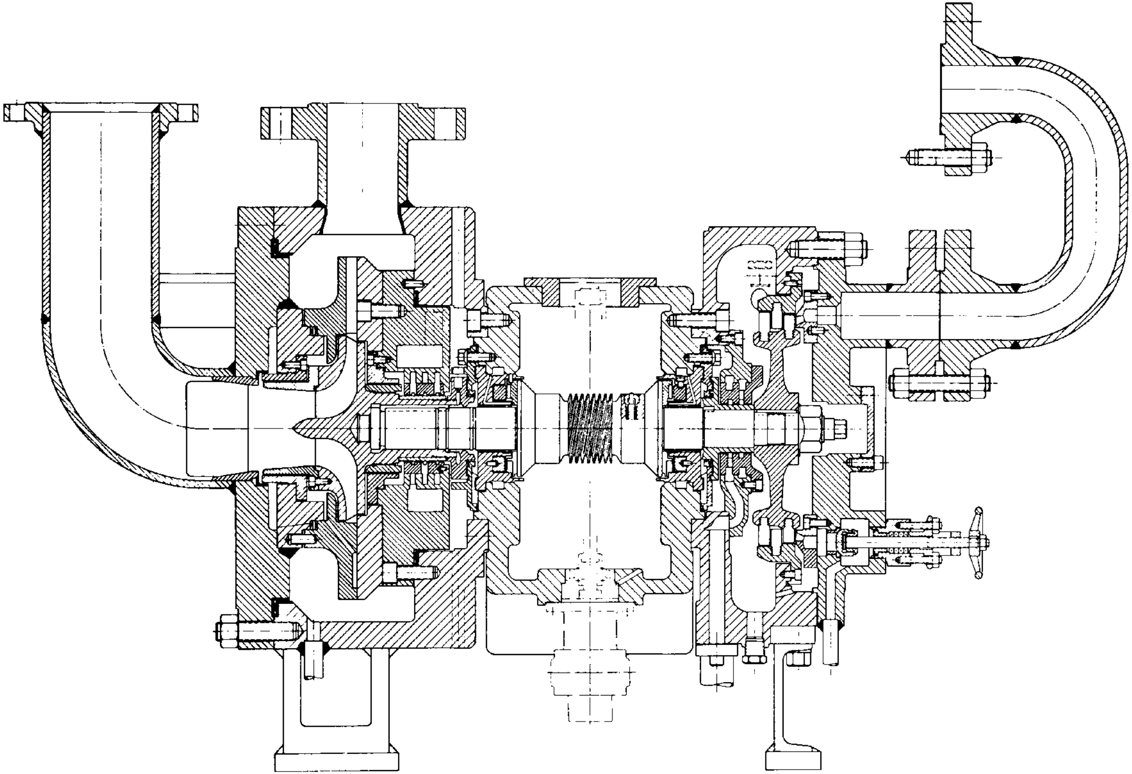

FIGURE 12 High-speed centrifugal pump and single-stage steam turbine mounted on a common shaft. Package is designed for boiler feed in marine, industrial, and process steam plants (Flowserve Corporation).

supporting equipment (starters, switch gear, fuel-supply system, and so on), operating costs, and the initial costs of the various drivers being considered.

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