Process Machinery Drives

Heinz P. Bloch, P.E., B.S.M.E., M.S.M.E., *Consulting Engineer, Process Machinery Consulting; American Society of Mechanical Engineers, Vibration Institute; Registered Professional Engineer (New Jersey, Texas). (Section Editor)*

R. H. Daugherty, Ph.D., *Consulting Engineer, Research Center, Reliance Electric Company; member, Institute of Electrical and Electronics Engineers. (Electric Motors and Auxiliaries)**

Fred K. Geitner, P.Eng., B.S.M.E., M.S.M.E., *Consulting Engineer, Registered Professional Engineer (Ontario, Canada). (Reciprocating Engines; Steam Turbines)†*

Meherwan P. Boyce, P.E., Ph.D., *President, Boyce Engineering International; ASME Fellow; Registered Professional Engineer (Texas, Oklahoma). (Turbines)*

Judson S. Swearingen, Ph.D., *Retired President, Rotoflow Corporation. (Expansion Turbines)*

Eric Jenett, M.S.Ch.E., *Manager, Process Engineering, Brown & Root, Inc.; associate member, AIChE, Project Management Institute; Registered Professional Engineer (Texas). (Power Recovery from Liquid Streams)*

Michael M. Calistrat, B.S.M.E., M.S.M.E., *Owner, Michael Calistrat and Associates; member, American Society of Mechanical Engineers. (Mechanical Power-Transmission Equipment)*

ELECTRIC MOTORS AND AUXILIARIES

RECIPROCATING ENGINES

* Based largely on material originally compiled and contributed by Carl R. Olson, M.S.E.E.
† Based largely on material originally compiled and contributed by Frank L. Evans, Jr., B.S.M.E., L.L.B. (Reciprocating Engines), and H. Steen-Johnsen, M.S.M.E. (Steam Turbines).

STEAM TURBINES

GAS TURBINES

EXPANSION TURBINES

POWER RECOVERY FROM LIQUID STREAMS

MECHANICAL POWER TRANSMISSION

Nomenclature

Symbol	Definition	SI units	U.S. customary units
C	Constant		
C_1	Velocity of steam flow	m/s	ft/s
c	Clearance	m	ft
d	Bearing diameter	m	ft
E	Applied voltage	V	V
E	Modulus of elasticity	N/m^2	lbf/ft^2
e	Gross pump efficiency	Dimensionless	Dimensionless
e	Base of natural logarithm		
e	Efficiency	Dimensionless	Dimensionless
e_h	Hydraulic efficiency	Dimensionless	Dimensionless
F	Load		
F_f	Friction force	N	lbf
f	Friction coefficient		
f	Frequency	Hz	Hz
G, g	Gravitational constant	m/s^2	ft/s^2
H	Power	W	hp
H_g	Enthalpy change, ideal	J	Btu
H_i	Reversible enthalpy change	J/kg	Btu/lb
H_p	Total head (pump)	m	ft
H_t	Total head (turbine)	m	ft
h	Oil thickness	m	ft
Δh	Enthalpy change per mass	J/kg	Btu/lb
I	Line current	A	A
K_A	Constant		
K_L	Constant		
K_S	Constant		
K_T	Constant		
k	Constant		
Mol. wt.	Molecular weight	kg	lb
N	Force	N	lbf
N_c	Compressor efficiency	Dimensionless	Dimensionless
N_e	Expander efficiency	Dimensionless	Dimensionless
n	Speed	r/s	r/min
n_s	Specific speed (turbine or pump)		
P	Pressure	kPa	lbf/in^2
P	Mean bearing pressure	N/m^2	lbf/ft^2
P	Power	kW	kW
PD	Pitch diameter	m	ft
P_{max}	Contact pressure, maximum	N/m^2	lbf/ft^2
p	Number of poles		
Q	Quantity of heat added	J	Btu
Q_c	Quantity of heat removed	J	Btu
Q_p	Capacity, pump	m^3/h	ft^3/h
Q_t	Capacity, turbine	m^3/h	ft^3/h
R	Operating load	kg	lb
R	Gas constant	J/(mol·K)	Btu/(mol·°R)
R	Armature resistance	Ω	Ω
r	Radius	m	ft
S	Apparent power	kVA	kVA
S_{max}	Stress, maximum	N/m^2	lbf/ft^2
T	Tension	N	lbf
T	Torque	N·m	lbf·ft
T	Temperature	K	°R
T_a	Average temperature	K	°R
T_c	Centrifugal tension	N	lbf
t	Time	s	s
V	Applied voltage	V	V
V	Counterelectromotive force	V	V
V	Velocity	m/s	ft/min
W	Net work to compressor	J	Btu
W_c	Compressor work	J	Btu
W_{theor}	Theoretical work	J	Btu
WK^2	Inertia	kg·m^2	lb·ft^2
y	Power factor	Dimensionless	Dimensionless
Z	Viscosity	N·s/m^2	cP
Z_a	Average compressibility factor	Dimensionless	Dimensionless
	Greek symbols		
α_r	exit angle		
β	exit angle		
θ	Arc of contact		
φ	Belt density	kg/m^3	lb/in^3
μ	peripheral velocity	m/s	ft/min
ϕ	Magnetic-field flux	Wb	Wb
ω	relative velocity	m/s	ft/min

General References: Bartlett, *Steam Turbine Performance and Economics,* McGraw-Hill, New York, 1958. Baumeister, *Standard Handbook for Mechanical Engineers,* 7th ed., McGraw-Hill, New York, 1967. Bloch, Heinz P., *Practical Guide to Compressor Technology,* McGraw-Hill, New York, 1996. Bloch, Heinz P., *Practical Guide to Steam Turbine Technology,* McGraw-Hill, New York, 1996. Boyce, Meherwan W., *Gas Turbine Engineering Handbook,* Gulf Publishing Company, Houston, 1982. Calistrat, Michael M., *Flexible Couplings—Their Design, Selection, and Use,* Caroline Publishing, Houston, 1994. Collins and Canaday, *Expansion Machines for Low Temperature Processes,* Oxford, Fair Lawn, New Jersey, 1958. Csanaday, *Theory of Turbomachines,* McGraw-Hill, New York, 1964. Fink and Carroll, *Standard Handbook for Electrical Engineers,* 10th ed., McGraw-Hill, New York, 1968. Jennings and Rogers, *Gas Turbines: Analysis and Practice,* McGraw-Hill, New York, 1953. Katz et al., *Handbook of Natural Gas Engineering,* McGraw-Hill, New York, 1959. Rase and Barrow, *Project Engineering of Process Plants,* Wiley, New York, 1957. Salisbury, *Steam Turbines and Their Cycles,* Wiley, New York, 1950. Scott, *Cryogenic Engineering,* Van Nostrand, Princeton, New Jersey, 1959. Shepherd, *Principles of Turbomachinery,* Macmillan, New York, 1956. Stepanoff, *Centrifugal and Axial Flow Pumps,* 2d ed., Wiley, New York, 1957. Stodola, *Steam and Gas Turbines,* Peter Smith, New York, 1945.

ELECTRIC MOTORS AND AUXILIARIES

All electric motors operate on the same basic principle regardless of type or size. When a wire carries electric current in the presence of a magnetic field (at least partially perpendicular to the current), a force on the wire is produced perpendicular to both the current and the magnetic field. In a motor the magnetic field radiates either in toward or outward from the motor axis (shaft) across the air gap, which is the annular space between the rotor and stator. Current-carrying conductors parallel to the axis (shaft) then have a force on them tangent to the rotor circumference. The force on the wire opposes an equal force (or reaction) on the magnetic field. It makes no difference whether the magnetic field is created in the rotor or the stator; the net result is the same: the shaft rotates.

Within these basic principles there are many types of electric motors. Each has its own individual operating characteristics peculiarly suited to specific drive applications. Equations (29-1) through (29-9), presented in Table 29-1, describe the general operating characteristics of alternating-current motors. When several types are suitable, selection is based on initial installed cost and operating costs (including maintenance and consideration of reliability).

ALTERNATING-CURRENT MOTORS, CONSTANT SPEED

The majority of industrial drives are constant speed. Typical applications include:

Pumps
Compressors
Fans
Conveyors
Crushers and mills

TABLE 29-1 Useful Formulas for Alternating-Current Motors

Power output:

$$H = Tn/5250 \qquad (29\text{-}1)$$

$$P = 0.00173VIye \quad \text{(three-phase)} \qquad (29\text{-}2)$$

$$P = 0.001VIye \quad \text{(single-phase)} \qquad (29\text{-}3)$$

Power input:

$$P = 0.00173VIy \quad \text{(three-phase)} \qquad (29\text{-}4)$$

$$P = 0.001VIy \quad \text{(single-phase)} \qquad (29\text{-}5)$$

$$P = 0.746H/e \qquad (29\text{-}6a)$$

$$S = P/y = 0.746H/ye \qquad (29\text{-}6b)$$

Line current and power factor:

$$I = \frac{0.746H\ \text{(output)}}{0.00173Vye} \quad \text{(three-phase)} \qquad (29\text{-}7)$$

$$I = \frac{0.746H\ \text{(output)}}{0.001Vye} \quad \text{(single-phase)} \qquad (29\text{-}8)$$

$$y = \frac{P\ \text{(input)}}{S} \qquad (29\text{-}9)$$

where e = efficiency, decimal
H = power, hp
I = line current, A
n = speed, r/min
P = power, kW
S = apparent power, kVA
T = torque, lbf·ft
V = applied voltage, V
y = power factor, decimal

NOTE: To convert horsepower to watts, multiply by 746; to convert pound-force-feet to newton-meters, multiply by 1.356; and to convert revolutions per minute to radians per second, multiply by 0.1047.

Alternating-Current Squirrel-Cage Induction Motors These motors are by far the most common constant-speed drives. They are relatively simple in design and therefore both low in cost and highly reliable. Representative prices are shown in Fig. 29-1 for various speeds and horsepowers.

The typical three-phase squirrel-cage motor has stator windings which are connected to the power source. The rotor is a cylindrical magnetic structure mounted on the shaft with slots in the surface, parallel (or slightly skewed) to the shaft; either bars are inserted into these slots or molten metal is cast in place and connected by a short-circuiting end ring at both ends of the rotor. The name "squirrel-cage" derives from this rotor-bar construction. In operation, current passing through the stator winding creates a rotating magnetic field which cuts the rotor winding unless the rotor is turning in exact synchronism with the stator field. This cutting action induces a voltage, and hence a current, in the rotor which in turn reacts with the magnetic field to produce torque.

The typical medium-sized squirrel-cage motor is designed to operate at 2 to 3 percent slip (97 to 98 percent of synchronous speed). The synchronous speed is determined by the power-system frequency and the stator-winding configuration. If the stator is wound to produce one north and one south magnetic pole, it is a two-pole motor; there is always an even number of poles (2, 4, 6, 8, etc.). The synchronous speed is

$$n = 120\,f/p \qquad (29\text{-}10)$$

where n = speed, r/min
f = frequency, Hz (cycles/s)
p = number of poles

The actual operating speed will be slightly less by the amount of slip. Slip depends upon motor size and application. Typically, the larger the motor, the less slip; an ordinary 7460-W (10-hp) motor may have 2½ percent slip, whereas motors over 746 kW (1000 hp) may have less than ½ percent. High-slip motors (as much as 13 percent slip) are used for applications with high inertia and requiring high starting torque; typical applications are punch presses and some crushers. Typical speed versus torque curves for various National Electrical Manufacturers Association (NEMA) design motors up to 149.2 kW (200 hp) are shown in Fig. 29-2. Typical characteristics and applications for these motors are given in Table 29-2.

As noted in Fig. 29-1 and Table 29-2 motors of various efficiency ratings may be available for an application. The efficiency of all motors of the same design will not be identical because of normal variations inherent in materials, variations in manufacturing processes, and inaccuracies in the test methods and equipment used to determine the efficiency. To address these variations, two values of efficiency have been identified for polyphase induction motors. Nominal efficiency refers to the average efficiency of a large population of motors of duplicate design. Some of the motors will have efficiency lower than the nominal efficiency value, and some will have higher. Minimum efficiency refers to the lowest level of efficiency that any motor in the population might have as a result of the variations introduced by the materials, manufacturing processes, and testing. The nominal effi-

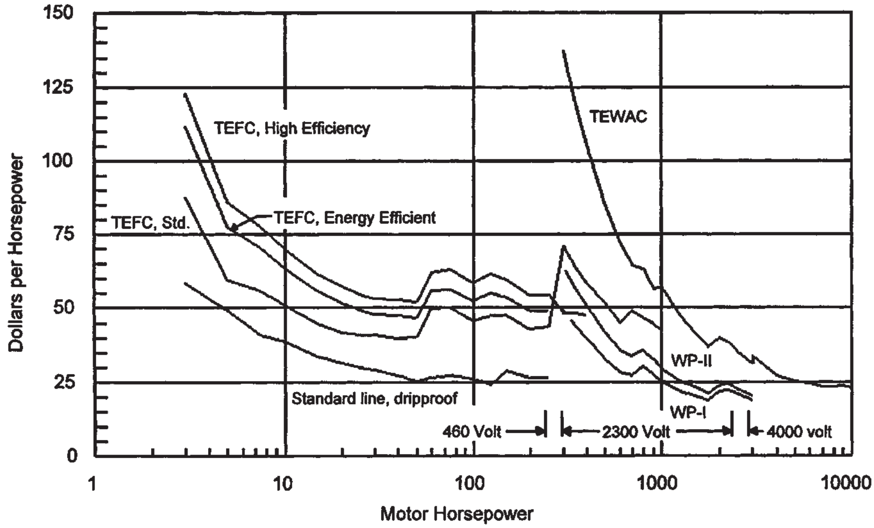


FIG. 29-1 Motor prices in dollars per horsepower for 1800 rev/min squirrel-cage induction motors from 3 to 10,000 hp. Dripproof and TEFC motors shown from 3 to 400 horsepower have 1.15 service factor; for other motors above 250 horsepower, the service factor is 1.0. The basis of these data is July, 1994. To convert dollars per horsepower to dollars per kilowatt, multiply by 1.340; to convert horsepower to kilowatts, multiply by 0.746.

ciency should be used when estimating the power required to supply a number of motors. The NEMA Standard MG-1, *Motors and Generators,* requires that all polyphase squirrel-cage integral horsepower motors, 1 to 500 horsepower, designated as Design A, B (and equivalent Design C ratings), and E be marked with the NEMA nominal efficiency value. A minimum level of efficiency is defined for each level of nominal efficiency in the NEMA Standard MG-1.

A motor purchaser or user can use the defined NEMA nominal efficiency to determine the relative economics of alternate motors. Common to the various methods one can use is consideration of the costs of energy and the motor, the annual hours of operation of the motor, and the motor efficiency. A simple payback analysis is used to determine the number of years required for the savings in energy cost resulting from the use of a more efficient motor to pay back the higher initial cost of the motor. The present worth life-cycle analysis considers both the time value of money and energy cost inflation to determine the present worth of savings for each motor being evaluated. The cash flow and payback analysis method considers motor cost premium, motor depreciation life, energy cost and energy cost inflation rate, corporate tax rate, tax credit, and the motor operating parameters.

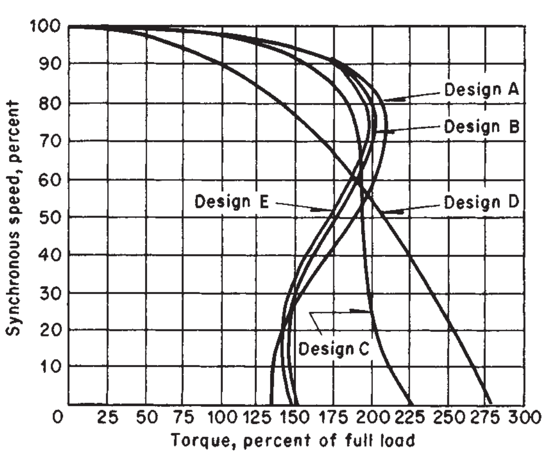


FIG. 29-2 Typical speed versus torque curves for various NEMA-design squirrel-cage induction motors. (See Table 29-2 for an explanation of design types.)

When making any economic analysis, care should be taken to be certain that the efficiency ratings of all motors being considered are on the same basis. While this should not be a problem for motors rated 1 to 500 horsepower as covered by the NEMA Standards for efficiency marking, it is common practice for several different test methods to be used when measuring the efficiency of motors rated over 500 horsepower. A particular test method may need to be selected by the test facility on the basis of available test equipment and power supply. All test methods that may be used to test any one motor will not necessarily give the same result for efficiency.

Further incentives to use energy-efficient motors are provided by various cost rebate programs offered by utilities based on horsepower rating and efficiency level. Another factor that will have a significant impact is the Energy Policy Act of 1992, in which the U.S. Congress established limits on the lowest level of nominal efficiency that certain classes of motors of standard design can have after 1997.

Control or **starting** of squirrel-cage induction motors normally consists of applying full voltage to the motor terminals. The speed-torque curves in Fig. 29-2 are based on full voltage throughout the speed range from start to run. The specific motor design determines the amount of starting current. However, if the motor is a typical standard (NEMA A or B) design, the starting current may be estimated at 6 to 6.5 times normal full-load current with full voltage applied. For NEMA Design E, it may be estimated at 8 to 9 times normal full-load current with full voltage applied. Particularly for large motors, this starting inrush current may cause an undesirable voltage dip which can shut down other equipment, temporarily dim lights, or even initiate malfunctions in sophisticated controls on the power system. For these conditions various alternatives exist.

1. *Reduced-voltage starting.* A reactor, resistor, or transformer is temporarily connected ahead of the motor during start to reduce the current inrush and limit voltage dip. This is accompanied by reduced starting torque. For reactor or resistor start, the torque decreases as the square of current; for transformer start, the torque decreases directly with line current. The reactor, resistor, or transformer can be adjusted to give a proper balance between torque and current.

TABLE 29-2 Characteristics and Typical Applications for Squirrel-Cage Induction Motors

	NEMA A and B	NEMA C	NEMA D	NEMA E
Starting torque	Normal	High	High	Normal
Running slip	Low	Low to medium	High	Low
Efficiency	Normal	Normal	Low	High
Applications	Pumps Compressors Fans Machine tools General use	Electrical stairways Pulverizers Conveyors	Punch presses Crushers	Pumps Compressors Fans Machine tools General use

2. *Star-delta starting.* A delta-connected motor is reconnected in Y form for starting, thus applying 57.7 percent voltage to each phase winding. This results in a developed torque of $(0.577)^2$, or only 33 percent. There is no means of adjustment; therefore, this method is useful only for loads requiring less than one-third of the motor's normal starting and accelerating torques.

3. *Part-winding starting.* This method employs a motor with two sets of windings, only one of which is energized during start. Torque and current are both roughly 50 percent. Two small contactors (starting switches) are used instead of one large one, and no reactors or transformers are required. The disadvantages are the fixed value of available torque and the harmonic disturbances from possible winding unbalance, causing deviations in the speed-torque curve and therefore possible failure to accelerate.

Braking and regeneration are possible with squirrel-cage motors. The direction of rotation is determined by the sequence or phase rotation of the power supply. If two leads on a three-phase motor are interchanged, the rotation reverses. If this occurs during operation, the motor will come to a rapid stop and reverse. Power is removed at standstill for effective braking. For estimating only, this plug-stop torque is approximately equal to starting torque. The braking time can be estimated by

$$t = WK^2\, n/308T \tag{29-11}$$

where t = time, s
WK^2 = inertia, lb·ft^2
n = running speed, r/min
T = torque, lbf·ft

To convert pound-square feet to kilogram-square meters, multiply by 0.0421; to convert revolutions per minute to radians per second, multiply by 0.1047; and to convert pound-force-feet to newton-meters, multiply by 1.356.

These estimates are frequently inaccurate because of second-order effects such as rotor saturation and harmonics. If the application is at all critical, the motor manufacturer should be consulted.

Regenerative braking occurs at speeds above synchronous-motor speed resulting from an overhauling load or from switching from high to low speed on multispeed motors. The action is similar to normal motor operation except that the slip is negative. The motor acts as an induction generator, delivering energy to the power source. If a power source such as a gas expander or a downhill conveyor is available, regenerative braking is an effective method of regulating speed, conserving energy, and starting the driven (driving) machine. An induction generator can deliver power to the source about equal to its rating as a motor. Regenerative braking can be used only on power systems capable of absorbing the generated energy and of supplying magnetizing excitation (reactive power) for the motor.

Direct-current dynamic braking utilizes direct current applied to the stator winding. Alternating-current power is first removed by opening the motor contactor or starter; direct current is then applied by a second contactor. The direct current produces a stationary magnetic flux, in contrast to the normal rotating ac field. The rotor bars cut this field, inducing currents which react with the dc flux to develop braking torque. Braking effort is easily varied by adjusting the amount of direct current. A desirable feature of this method for standard motors is the relatively soft braking effort at full-load speed, reducing impact; further, the braking effort typically increases as speed drops, reaching a maximum near zero speed. Braking torque at standstill is zero; however, maximum torque occurs at such low speed that static friction is usually sufficient to prevent coasting. Peak braking torques can be high; so shafts, gearing, couplings, etc., should be checked. Caution should be exercised because frequent starting and stopping cause excessive heating.

Synchronous Alternating-Current Motors These motors run in exact clock synchronism with the power system. For most modern power systems, these are truly constant-speed motors.

In the conventional synchronous motor a rotating magnetic field is developed by the stator currents as in induction motors. The rotor, however, is different, consisting typically of pairs of electromagnets (poles) spaced around the rotor periphery. The rotor field corresponds to the field produced by the ac stator having the same number of poles. The rotor or field coils are supplied with direct current; the magnetic field is therefore stationary with respect to the rotor structure. Torque is developed by the interaction of the rotor magnetic field and the stator current (in-phase component). Under no-load conditions and with appropriate dc field current, rotor and stator magnetic-field centers coincide. The voltage applied to the stator winding is balanced by an opposing voltage generated in the stator by the rotor field (induced), and no ac power current flows. As load is applied, the rotor tends to decelerate momentarily, causing a shift of rotor position with respect to the ac field. This shift produces a difference between the applied and induced voltages; the voltage difference causes current to flow; the current reacts with the rotor magnetic flux, producing torque.

Synchronous motors should not be started with the dc field applied. Instead they are started as induction motors; bars, acting like a squirrel-cage rotor, are embedded in the field-pole surface and connected by end rings at both ends of the rotor. These damper bars also serve to damp out oscillations under normal running conditions. When the motor is at approximately 95 percent speed (depending upon application and motor design), direct current is applied to the field and the motor pulls into step (synchronism). Because the damper bars do not affect the synchronous-speed characteristics, they are designed for starting performance. This provides flexibility in the accelerating characteristics to meet specific application requirements without affecting running efficiency and other synchronous-speed characteristics. The rotor design of a squirrel-cage motor, on the other hand, must be a compromise between starting and running performance. The dc field is usually shorted by a resistor during starting and contributes accelerating torque, particularly near synchronous speed.

Power-factor correction is an important feature of synchronous motors. Conventional synchronous-motor power factors are either 100 or 80 percent leading. Leading-power-factor machines are used frequently to correct for the lagging power factor of the remaining plant load (such as induction motors), preventing penalty charges on power bills. Even 100 percent power-factor motors can be operated leading at reduced loads. An advantage of synchronous motors over capacitors is their inherent tendency to regulate power-system voltage; as voltage drops, more leading reactive power is delivered to the power system, and, conversely, as voltage rises, less reactive power, in contrast to capacitors for which the reactive power decreases directly in proportion to the voltage drop squared. The amount of leading reactive power delivered to the system depends on dc field current, which is readily adjustable.

Field current is an important control element. It controls not only the power factor but also the pullout torque (the load at which the motor pulls out of synchronism). For example, field forcing can prevent pullout on anticipated high transient loads or voltage dips. Loads with known high transient torques are driven frequently with 80 percent power-factor synchronous motors. The needed additional field supplies both additional pullout torque and power-factor correction for the power system. When high pullout torque is required, the leading power-factor machine is often less expensive than a unity-power-factor motor with the same torque capability.

Direct-current field excitation is supplied by various means. A dc generator (exciter) is often used either directly coupled to the motor shaft, belt-driven off the motor shaft (seldom used), or driven by a separate small motor (exciter motor-generator set). Direct-coupled and belt-driven exciters are always associated with a single motor and are controlled by adjustment of the exciter field. Motor-generator-set exciters may supply one or more synchronous motor fields. For reliability, several motor-generator sets may be paralleled to supply multiple motor fields; in such a case the exciter voltage is usually fixed (e.g., 125 or 250 V), and the individual synchronous-motor fields are controlled by motor field rheostats (much larger than exciter field rheostats). Static (rectifier) exciters are also used for single-motor or multimotor excitation. Special rectifiers are required to avoid damage from surge voltages on pullout. These exciters, rotating or static, require brushes and slip rings to conduct direct current to the rotating field structure.

Another concept is **brushless excitation,** in which an ac generator (exciter) is directly coupled to or mounted on the motor shaft. The ac exciter has a stator field and an ac rotor armature which is directly connected to a static controllable rectifier on the motor rotor (or a shaft-mounted drum). Static control elements (to sense synchronizing speed, phase angle, etc.) are also rotor-mounted, as is the field discharge resistor. Changing the exciter field adjusts the motor field current without the necessity of brushes or slip rings. Brushless excitation is suitable for use in hazardous atmospheres, where conventional brush-type motors must have protective brush and slip-ring enclosures.

Because of the more complicated design and the necessity for a field power supply, synchronous motors are typically applied only in large-horsepower ratings (several hundred horsepower and larger); synchronous motors over 59,680 kW (80,000 hp) have been built. With their latitude in size and characteristics and their important inherent high power factor and efficiency, synchronous motors are applied to a wide variety of drives. Engine-type motors (without shaft or bearings) are used almost exclusively to drive large low-speed reciprocating compressors. Other typical applications include jordans, compressors, pumps, ball and rod mills, chippers, crushers, and grinders. Speeds as low as 80 r/min are practical; the top speed is limited by the rotor structure and is dependent on horsepower. The approximate limit for 1800 r/min is 2238 kW (3000 hp); for 1200 r/min it is 29,840 kW (40,000 hp).

Synchronous speeds are calculated by Eq. (29-10). Speeds above the limits given are obtained through step-up gears; large high-speed centrifugal compressors are examples. Two-pole (3600 r/min at 60 Hz) synchronous motors can be built but are uneconomical in comparison with geared drives.

ALTERNATING-CURRENT MOTORS, MULTISPEED

Squirrel-cage induction motors are inherently single-speed machines, but multispeed operation can be obtained by reconnecting the stator windings of motors designed for this purpose.

Two-Winding Motors These motors illustrate the simplest concept. The two separate stator windings (three-phase or two-phase only) are designed and wound for a different number of poles. For example, one winding may be four poles (1800 r/min at 60 Hz) and the other six poles (1200 r/min at 60 Hz). Only one winding is connected at a time. This method is used for speed ratios other than 2:1. Since the two windings are independent, a large number of speed combinations is possible. The two windings are not necessarily of equal capacity.

Two-winding motors may be built for constant torque, variable torque, or constant horsepower. Constant-horsepower motors are capable of handling the same horsepower at both speeds (i.e., higher torque at the low speed). Constant-torque motors can handle the same load torque at either speed (e.g., conveyor drives). Variable-torque motors are designed for loads in which load torque varies as the square of speed and horsepower varies as the cube of speed. Typical applications are as follows:

Variable torque	Constant torque	Constant horsepower
Fans	Conveyors	Machine tools
Centrifugal pumps	Feeders	
	Reciprocating compressors	

Single-Winding Consequent-Pole Motors These motors can be used when the low speed is one-half of the high speed. They are available as three-phase only. The specially designed winding is regrouped by external reconnection (motor control) to obtain the desired speed. A 2:1 speed ratio only is obtainable by this method; speeds such as 3600/1800, 1800/900, and 1200/600 are obtainable. Variable-torque, constant-torque, and constant-horsepower designs are available with torque characteristics as discussed under "Two-Winding Motors." The control for two-speed single-winding motors is more complicated than for the two-winding control.

Four-Speed, Two-Winding Squirrel-Cage Motors These motors are built by combining the preceding two methods. The stator winding is composed of two consequent-pole windings. Each winding gives two speeds with a relation of 2:1 to each other. The standard 60-Hz speed combinations are 1800/1200/900/600 r/min and 1200/900/600/450 r/min. The three torque-capability designs of variable torque, constant torque, and constant horsepower are also available in these four-speed motors.

Pole-Amplitude-Modulated Induction Motors These are single-winding squirrel-cage motors with any combination of poles or speeds (e.g., 8/10 poles or 900/720 r/min or as wide as 4/20 poles or 1800/360/min). They are smaller and lighter than equivalent two-winding machines. The entire winding works at both high and low speed, resulting in greater thermal capacity and higher efficiency. The basic principle is that one frequency acted on (modulated) by another produces new frequencies equal to the sum and difference of the two. Thus, a six-pole field modulated by a two-pole field produces a four- and an eight-pole field. The four-pole field can be eliminated by proper winding geometry. Such a motor runs at six-pole speed (1200 r/min, 60 Hz) when connected normally and at eight-pole speed (900 r/min) when half of the coils are reversed to produce the two-pole modulation.

In the preceding discussion of multispeed ac motors note that only induction motors are considered. These have no discrete physical rotor poles, so that only the stator-pole configuration need be modified to change speed. To operate multispeed, a synchronous motor would require a distinct rotor structure for each speed. Thus multispeed is practical only for squirrel-cage induction motors.

ALTERNATING-CURRENT MOTORS, WOUND-ROTOR INDUCTION

Wound-rotor induction motors operate on the same principle as squirrel-cage motors. However, as the name implies, the rotor has windings rather than bars, and these windings are connected to shaft-mounted slip rings. Brushes riding on the slip rings are connected to external resistance or short-circuited. Wound-rotor motors have an additional dimension of flexibility in the **variability of external rotor resistance.** Rotor resistance affects the shape of the speed-torque curve; increasing resistance decreases the speed at which maximum torque occurs. Figure 29-3 illustrates this effect as resistance is increased from zero external resistance (at top) to a very high value (extreme left).

Reduced-speed operation reduces efficiency. **Efficiency** is approximately equal to speed expressed as a percentage of synchronous speed. Thus at 75 percent speed, about three-fourths of the motor

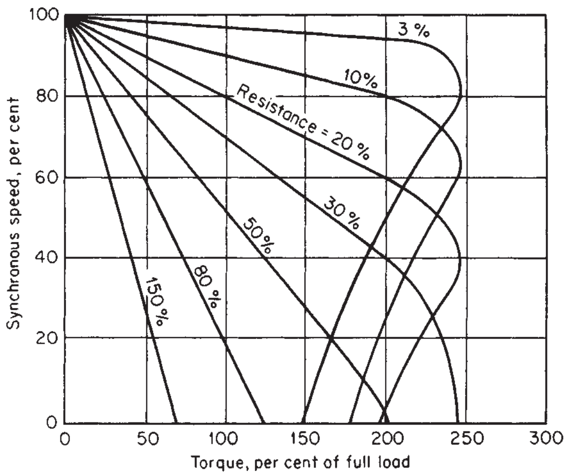


FIG. 29-3 Typical speed versus torque curves for a wound-rotor induction motor with varying amounts of external secondary (rotor) resistance. Resistance values are based on resistance at 100 percent torque and zero speed = 100 percent.

input goes to the load; the other quarter is dissipated in the rotor resistance. The external resistance is sized to get rid of this heat. Further, in accelerating a load to operating speed, the heat generated in the rotor resistance is equal to the energy required to accelerate the load (inertia plus friction). The rotor resistance (internal or external) must have capacity to store this heat, since accelerating time is typically too short to allow any significant heat dissipation.

These characteristics of wound-rotor motors determine their **scope of application.** They can accelerate very-high-inertia loads, such as crushers, by using a large external resistance to absorb the heat. Loads sensitive to shock-accelerating torques may also be accelerated softly by inserting a high starting rotor resistance; this effect is used, for example, to take up slack in gears. Wound rotors are also used to handle loads like punch presses and car crushers, for which extreme transient peak loads are supplied by the mechanical-system inertia, allowing the system to slow down during these peaks; permanent external rotor resistance provides this soft characteristic. Wound-rotor motors are also used to provide adjustable-speed drives for pumps, cranes, and other loads when precise speed regulation is not required. Reduced-speed losses are not very significant for pump loads; the percent efficiency is low at reduced speed, but the torque and load horsepower are dropping rapidly. If torque is proportional to speed squared, the maximum rotor-resistance losses never exceed 10.5 percent of the full-speed load (occurs at 70 percent speed and efficiency, 50 percent torque, 35 percent load horsepower, and 30 percent losses, thus 10.5 percent losses based on full load at full speed).

Control of wound-rotor motors, as discussed, can be effected by adjusting the external secondary (rotor) resistance either in steps or continuously by liquid rheostat (this method is seldom used). Commonly when secondary resistance is varied to adjust speed or torque or to control acceleration, multiple resistance steps are used. These steps may be switched manually (typically a drum switch) or electrically by contactor.

In addition to secondary resistance control, other devices such as reactors and thyristors (solid-state controllable rectifiers) are used to control wound-rotor motors. Fixed secondary reactors combined with resistors can provide very constant accelerating torque with a minimum number of accelerating steps. The change in slip frequency with speed continually changes the effective reactance and hence the value of resistance associated with the reactor. The secondary reactors, resistors, and contacts can be varied in design to provide the proper accelerating speed-torque curve for the protection of belt conveyors and similar loads.

Saturable reactors, which are adjustable by a small dc signal, have also been used for both primary (stator) and secondary (rotor) control. In the primary they control motor voltage and therefore torque. In combination with fixed secondary resistors and feedback from a tachometer, this system can be used for precise speed and torque control of cranes, hoists, etc. Even reversing can be accomplished by using two saturable reactors in each of two (of three) phases. Other combinations of fixed or saturable reactors in the primary and/or secondary, all combined with secondary resistors, provide a wide range of capabilities and flexibility for the wound-rotor motor.

Thyristors have been replacing saturable reactors; they are small, efficient, and easily controlled by a wide variety of control systems. A modern crane control drive uses fixed secondary resistors and two sets of primary thyristors (one set for hoist, one for lower). With tachometer feedback for speed sensing, the control for the motor provides speed regulation and torque limiting in both directions, all with static devices. A wide variety of control systems is possible; the control should be designed for the specific application.

DIRECT-CURRENT MOTORS

Direct-current motors are adjustable in speed over a wide range. Further, efficiency is high over the entire speed range, unlike wound-rotor motors, in which efficiency is roughly proportional to speed. This flexibility is attained at the expense of additional complexity and cost.

Direct-current motor fields are on the stator. The rotor is the armature. The magnetic field does not rotate like the field in ac machines. Current in the armature reacts with the stator field to produce torque.

The armature windings generate a voltage opposite to the applied voltage as they cut the magnetic field. The difference between the terminal voltage and the generated voltage (counterelectromotive force) applied to the armature resistance produces armature current. Torque is proportional to armature current and magnetic flux. Counterelectromotive force is almost equal to the applied voltage, so the speed can be changed by changing the applied voltage; the speed can also be changed by varying the field current.

$$E = V + IR \quad \text{or} \quad I = (E - V)/R \tag{29-12}$$

$$V = kn\phi \tag{29-13}$$

where E = applied voltage, V
V = counterelectromotive force (generated voltage), V
R = armature resistance, Ω
I = armature current, A
k = constant dependent on motor design
n = speed, r/min
ϕ = magnetic-field flux

Generated voltage is proportional to the magnetic-flux cut; so a motor must change speed to generate the same counterelectromotive force if the field current is changed.

Direct-current motors are connected in several ways. Shunt motors have armature and field connected in parallel. This connection provides almost constant speed regardless of the applied voltage. If the voltage drops, the counterelectromotive force also drops because of the reduction in field strength. Speed can be changed by varying the field current and/or armature voltage independently. Increasing armature voltage increases speed; increasing field current decreases speed.

Series motors have their armature and field windings connected in series; both carry the same current. The speed depends on both voltage and load. Torque is the product of armature current and magnetic field; both are dependent on current. For any specific load torque, the current is constant and speed is proportional to applied voltage. If, however, the load changes, the speed also changes. A 50 percent drop in load torque reduces the motor current to 70 percent, making the product of armature and field current 50 percent. The reduced field current decreases counterelectromotive force, and the motor speed increases by 40 percent. Because of this characteristic, series motors overspeed severely if unloaded. Series motors are suitable for constant-horsepower applications with a wide speed range, such as machine tools. They are also used for traction drives (e.g., shuttle cars and locomotives), some cranes, hoists, and elevators.

Compound-wound dc motors have both series and shunt fields. The addition of a small series field helps provide the proper amount of no-load to full-load speed regulation or droop. Shunt or compound-wound motors are applied widely to many adjustable-speed drives. They are important for drives requiring accurate speed regulation and adjustment.

A dc motor's inherent speed-torque curve can be varied widely by adjusting the relative amounts of shunt and series fields. The series field may also be connected to aid or buck the shunt field. The usual practice is to connect the series field so that it adds to the shunt field (cumulative compound), which gives a stable, drooping speed with increasing load.

The great flexibility of dc motors both through inherent design characteristics and through the way in which they are operated makes them ideally suited to adjustable-speed drives, particularly regulated drive systems.

ADJUSTABLE-SPEED DRIVES

One of the oldest adjustable-speed drives is the **Ward-Leonard system.** This consists of an ac to dc motor-generator set and a shunt or compound-wound dc motor. Speed is adjusted by changing the generator voltage. A functional equivalent of this drive uses an adjustable-voltage rectifier feeding a dc motor. This system has only one rotating machine in contrast to the three of a conventional Ward-Leonard system.

Modern static controllable rectifiers such as thyristors respond almost instantaneously to control signals and are adaptable as the power supply to the most critical regulated-drive systems. The sensing and regulating system can be designed to hold speed, speed differential, tension, torque, current, acceleration and deceleration, etc., or any required combination. For example, a drive may be speed-regulated with torque limit or speed-regulated with acceleration and/or deceleration regulated or limited.

Typical applications for adjustable-voltage, adjustable-speed dc drives include winders, paper machines and auxiliaries, blending systems, feeders, extruders, calendars, machine tools, range and slasher drives, cranes, hoists, shovels and draglines, and an almost unlimited variety of drives requiring the flexibility and efficiency possible with direct current.

Mechanical adjustable-speed drives are used when a high degree of regulation is not required. One drive consists of a constant-speed ac motor driving the load through V belts and variable-pitch pulleys. The speed range can be as high as 8:1. It is available up to 18.6 kW (25 hp). Speed adjustment is either manual or remote with a motor drive for the adjustment. Speed regulation from no load to full load is normally 3 to 6 percent. Efficiency is high over the entire speed range since there are no slip losses.

Electromagnetic drives are simple adjustable-speed ac drives with efficiency comparable with that of wound-rotor motors. This drive uses a magnetic slip coupling driven by a squirrel-cage motor. The slip is determined by the excitation current and the load. Efficiency is proportional to speed. Therefore, these drives are uneconomical for continuous low-speed high-torque operation but are ideal for fans and centrifugal pumps requiring little speed adjustment when torque decreases rapidly with speed and for controlling acceleration. Electromagnetic drives are functionally equivalent to hydraulic couplings.

RectiFlow adjustable-speed drives use both a wound-rotor motor and a dc motor connected to the same shaft. The rotor winding of the wound-rotor motor is connected to a rectifier. The dc rectifier output supplies the dc motor. Semiconductor-rectifier developments make this a practical, high-efficiency adjustable-speed drive for applications up to several hundred horsepower. Since the rotor losses of the wound-rotor motor are used to produce shaft power and are not dissipated in a resistor, the efficiency of these drives is high over the entire speed range. A typical 223.8-kW (300-hp) RectiFlow drive operating over a 3:1 speed range has an efficiency of more than 83 percent over its operating range. RectiFlow adjustable-speed drives are suitable for high-torque low-speed drives such as extruders, mixers, pumps, fans, and kilns.

A modification of this basic drive system uses solid-state rectifiers and thyristors to convert the wound-rotor, variable-frequency slip power first to direct current and then to line-frequency power (60 Hz in the United States). This in turn is fed back to the power system as useful energy.

Adjustable-frequency alternating current can be used with squirrel-cage or synchronous motors for adjustable-speed drives. A typical application is small synthetic-fiber spinning drives which require multiple motors operating at constant speed. When precise synchronism between drives is required, synchronous-reluctance motors are used. These are squirrel-cage motors with flats or grooves on the rotor which form magnetic poles because of the change in magnetic path as the rotor position moves with respect to the stator field. This causes the rotor to rotate in exact synchronism with the stator field for light loads. The power source for these drives is either an ac generator driven by any adjustable-speed drive or a static inverter.

It should be noted that even motors as large as 60 MW have been equipped with these drives.

MOTOR ENCLOSURES

Except for areas with fire or explosion hazards (hazardous areas), motor enclosures are designed to provide protection to the internal working parts. The development of improved insulating materials and finishes has affected the required degree of protection and consequently the design and classification of enclosures. Examples of several types of enclosures are shown in Fig. 29-4.

Open, dripproof is the standard enclosure for induction, high-speed synchronous, and industrial dc motors. This design is useful for most indoor and many outdoor applications. Dripproof construction provides good mechanical protection to the internal working parts of the motor and prevents the entrance of dropping liquids and heavy dirt particles. However, it does not protect against airborne moisture, dust, or corrosive fumes. **Guarded** machines have all openings protected to prevent objects more than 12.7 mm (½ in) in diameter from entering the motor. **Splashproof** motors are not affected by water or by solid particles striking or entering the enclosure at an angle less than 100° from the vertical.

Weather-protected, type I is the next degree of protection for larger motors. Such a motor is defined as "an open machine with its ventilating passages so constructed as to minimize the entrance of rain, snow, and airborne particles to the electric parts" (NEMA Standard MG-1, "Motors and Generators"). All openings are restricted against passage of a 19-mm- (¾-in-) diameter rod. Some modern insulation systems are completely satisfactory for most outdoor applications.

Weather-protected, type II motors are recommended for large sizes when a higher degree of protection and longer life are desired. They have extensive baffling of the ventilating system so that the air must turn at least three 90° corners before entering the active motor parts [maximum air velocity, 3.05 m/s (600 ft/min)]; thus, rain, snow, and dirt carried by driving winds are blown through the motor housing without entering the active parts.

Totally enclosed motors offer the greatest protection against moisture, corrosive vapors, dust, and dirt. Totally enclosed fan-cooled (TEFC) motors are the obvious choice rather than *weather-protected* below 186.5 kW (250 hp). Their internal and external ventilating air are kept separate; external air never gets inside except for the small amount that enters by breathing.

TEFC motors have both an internal fan for circulating air within the motor and an external fan for forcing the air through or over the motor frame or heat exchanger. Small motors [approximately 2.238 kW (3 hp) and below] do not require ventilating fans; these totally enclosed nonventilated motors are similar to TEFC with the fans omitted.

Separate forced ventilation is required for some applications (for example, adjustable-speed drives which operate at low speed); these must depend on an external ventilation. This classification includes *open externally ventilated* machines, *open pipe-ventilated* machines,

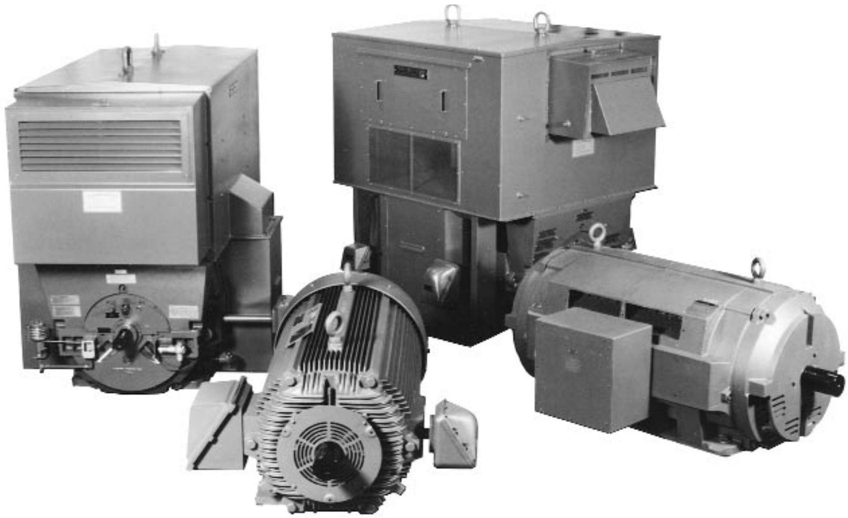


FIG. 29-4 Examples of open dripproof, totally enclosed fan-cooled, and weather-protected motor enclosures. (*Photo courtesy of Reliance Electric Company, Cleveland, Ohio.*)

and *totally enclosed pipe-ventilated* machines. In corrosive or hazardous areas, safe or clean air ventilates the motor.

Enclosed motors with air-to-water coolers cost much less than TEFC motors above 373 kW (500 hp); in large synchronous-motor ratings, they cost even less than weather-protected, type II. Large enclosed synchronous machines with coolers for mounting in the motor foundation are frequently supplied at lower cost than motors with integral-mounted coolers.

Fire or explosion hazards require special motor enclosures. Hazards include combustible gases and vapors such as gasoline; dust such as coal, flour, or metals that can explode when suspended in air; and fibers such as textile lint. The kind of motor enclosure used depends on the type of hazard, the type and size of motor, and the probability of a hazardous condition occurring. Some available enclosures are explosionproof motors, which can withstand an internal explosion; force-ventilated motors cooled with air from a safe location; and totally enclosed motors cooled by air-to-water heat exchangers and pressurized with safe air, instrument air, or inert gas.

MOTOR CONTROL

The basic functions of motor starters are:

1. Normal "start-stop" control of the motor.
2. Protection of the motor.
3. Protection of the electrical supply system in the event of a motor or motor-feeder short circuit. The fault must be cleared from the rest of the system to prevent further trouble.
4. Electrical isolation to provide accessibility for maintenance.
5. Provision for other control such as master sequence control, protective shutdown devices (e.g., bearing overtemperature, overtravel, pump high pressure, remote control, etc.).

Types of Starters

High Voltage and Low Voltage The electrical industry has standardized the distinction between high voltage and low voltage at 600 V. Below 600 V the common system voltages in use in the United States are 120, 208, 240, 480, and 600 V. Above 600 V, the standard nominal system voltages commonly in use are 2400, 4160, and 6900 or 7200 V. Higher voltages are available, but the motor cost is usually prohibitive.

For low-voltage starters below 600 volts, the same starters are used for any voltage, since there is only one insulation class.

For high-voltage motor-starting applications, there are several classes of insulation: 2500, 5000, 7500, and 15,000 V. The conventional control-type high-voltage motor starter is available for 2500- or 5000-V service. For voltages higher than this, switchgear must be used.

The construction of high-voltage starters employs much greater clearances and provides additional safety features such as grounded barriers between the high- and low-voltage sections of the starter. Extensive mechanical and electrical interlocking is also used for additional safety.

One of the major differences between high- and low-voltage starters is the amount of power handled. An approximate dividing line is 149.2 kW (200 hp). This, however, is not a fixed and rigid rule.

Line Starters and Combination Starters A line starter consists of a contactor (motor-starting switch) and motor-overload relays. Contactors are capable of carrying and interrupting normal motor-starting and -running currents; they are not, however, normally capable of interrupting short-circuit currents. They must be backed up by fuses or a circuit breaker for this function.

When a disconnect switch, circuit breaker, or set of fuses is included in the same enclosure as the contactor, the starter is then called a **combination starter.** In addition to the fault-current-interrupting function, the breaker or fuses serve as the disconnecting device. Figure 29-5 illustrates schematically combination starters of various types. The latch is arranged to open the disconnect before the door can be swung open. There are also provisions for padlocking the disconnect open with the door closed so that maintenance work on the motor may proceed in safety.

Manual and Magnetic Starters Manual motor starters are operated by hand. The simplest type of manual starter is a snap switch with no overload protection, used only for motors of 1.492 kW (2 hp) and smaller, usually single-phase motors with integral overload protection.

Magnetic motor starters are similar in function to manual starters except that they are solenoid-operated. They are available up to 3730 kW (5000 hp). One of the main advantages is the convenience of electrical operation. Start-stop push buttons can be located anywhere. When automatic or remote operation is needed, magnetic starters are essential.

Comparison of Switchgear and Contactor-Type Control Frequently switchgear is used for motor control, particularly for large high-voltage motors. Switchgear (Fig. 29-6) must be used for motors

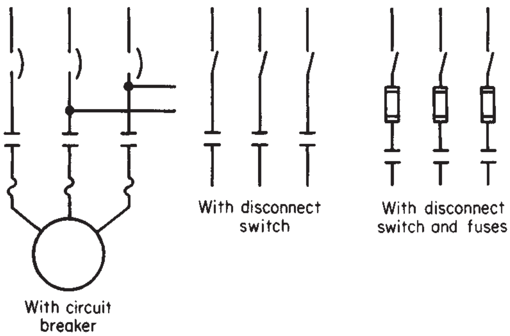


FIG. 29-5 Simplified schematic diagram of a combination line starter with a circuit breaker as the fault interrupter and disconnect. Alternative fuses and disconnect switch are shown as substitutes for the circuit breaker.

larger than 3357 kW (4500 hp)* at 4160 or 4600 V or 1865 kW (2500 hp)* at 2300 V and for all motors above 5000 V. Switchgear consists of circuit breakers and protective relaying. Circuit breakers are electrical switches designed primarily for their ability to interrupt short-circuit currents. This is one of the major differences from contactors, which are designed principally to handle starting and running currents. Contactors normally depend on a set of fuses or a circuit breaker to handle major faults (short circuits).

Contactors are designed for frequent operation. Circuit breakers are designed for far fewer operations and therefore are never used as motor starters when repetitive operation is required. A typical example of frequent operation is mine-hoist service, in which the motor must be reversed at the end of every hoisting or lowering operation; contactors would be used.

* 3730 kW (5000 hp) and 2051 kW (2750 hp) for unity power factor synchronous motors.

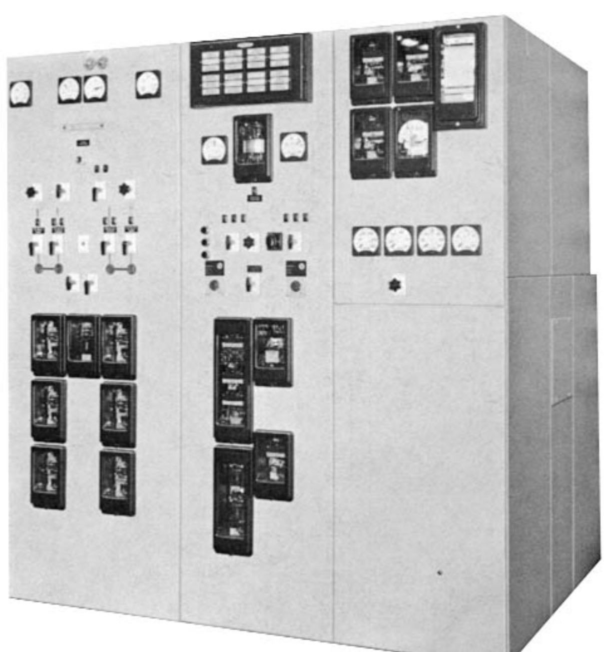


FIG. 29-6 Typical lineup metal-clad switchgear including motor starters and protective relaying.

High-voltage ac control-type (contactor) motor starters use fuses to provide short-circuit interrupting capacity. One disadvantage of fuses is that only one fuse may blow. This leaves single phase applied to the motor. Motors will continue to operate with single-phase power but can overheat even with less than rated current flowing. In contrast to contactors, circuit breakers are three-pole devices: a fault on one phase will trip all three, minimizing the single-phasing problem.

Centralized Control As mentioned previously, motor starters may be located either at the motor or at some remote point. Frequently they are grouped at a location convenient to the source of power. The feeders radiate from this point to the individual motor loads. A convenient method is the control-center modular structure for low-voltage control, into which are assembled motor starters and other control devices. The individual starters can be drawn out of the structure for rapid, easy maintenance and adjustment. With this construction it is easy to change starter size or add additional starters. All the starters are in one location, so that interwiring is simple and easy to check. Auxiliary relays, control transformers, and other special control devices can also be included. See Fig. 29-7.

Motor Protection Money spent for motor-protective devices can be compared to insurance, in which premiums depend on the protected value when the protected value is the cost of the motor, the cost of anticipated repairs, or the cost of downtime, lost production, and, in some cases, contingent damage to other equipment.

Overload Protection Overload relays for protecting motor insulation against excessive temperature are located either in the motor control or in the motor itself. The most common method is to use thermal overcurrent relays in the starter. These relays have heating characteristics similar to those of the motor which they are intended to protect. Either motor current or a current proportional to motor-line current passes through the relays so that relay heating is comparable to motor heating.

Standard thermal overcurrent relays located in the starter have some disadvantages. They cannot detect abnormal temperatures in the motor caused by blocked ventilation passages or high ambient temperature at the motor. They are also likely to trip out unnecessarily in locations where the control enclosure is at a higher temperature than the motor. Motors are normally ventilated with external air so that their ambient temperature is the ambient temperature of the surrounding air. However, control enclosures are not freely ventilated, so their internal temperature can become quite high if they are located in a sunny location. High-current relays are sometimes used to avoid this difficulty. This prevents the motor from being tripped out unnecessarily because of high ambients inside the control enclosure, but the motor will be improperly protected during cool weather and overcast days and at night. Ambient-temperature-compensated relays should be used in these situations.

Some overload-protection schemes measure motor-winding temperature directly; various methods are used. Small single-phase motors are available with built-in overload protection. A thermostat built into the motor senses motor-winding temperature directly.

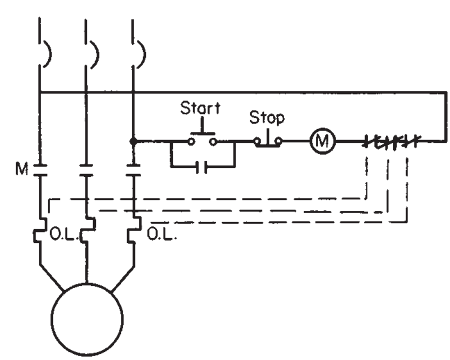


FIG. 29-7 Schematic diagram of a combination starter, showing a simple control scheme.

When the motor overheats, the thermostat opens, interrupting motor-line current. Pilot thermostats mounted on the windings of larger motors trip the motor starter rather than interrupt line current. This method gives good protection for sustained overloads, but because of the thermal time lag between the copper winding and the thermostat it may not provide adequate protection for stalled conditions or severe overloads.

Temperature detectors embedded in the motor winding give close, accurate indication of motor temperature. Both conventional resistance temperature detectors (RTD) and special thermistors (highly temperature-sensitive nonlinear resistors) are used. With appropriate auxiliaries these devices can indicate or record motor temperature, alarm, and/or shut down the motor.

Short-Circuit Protection Short circuits must be removed promptly to avoid severe damage at the fault and to avoid disturbances to the rest of the electrical system. Short-circuit protection should be set as low as possible so that tripping action is initiated quickly. Motor-starting inrush current sets a limit on how low short-circuit devices may be set. For squirrel-cage ac motors, instantaneous short-circuit tripping should be initiated at about 7 to 10 times full-load running current. This gives an adequate margin above the normal inrush of approximately 6 times full-load current. Modern low-voltage combination starters are available with adjustable instantaneous circuit breakers which can be set just above motor-starting current.

High-voltage contactor-type motor controls depend on power fuses for short-circuit protection. The fuses are coordinated with the overload relays to protect the motor circuit over the full range of fault conditions from overload conditions to solid maximum-current short circuits.

Locked-Rotor Protection Under locked-rotor (stalled) conditions the rotors of large synchronous and squirrel-cage motors are the most likely motor elements to be damaged by overheating. The rotor's heating is not related to stator heating during startup. Therefore for large motors it is common practice to use separate devices or characteristics to protect against running overloads and locked-rotor conditions if the overload and short-circuit protective devices cannot be coordinated to handle this condition for the specific motor characteristics.

Synchronous-motor rotor frequency can be detected because the rotor field circuit is available. Special control schemes have been devised which take into account both speed and induced rotor current in providing locked-rotor and accelerating protection.

Undervoltage Protection If a power outage occurs, it is necessary to remove motors from the line to prevent excessive starting current surges on the electrical system when voltage is reestablished. It is also unsafe to have drives starting indiscriminately when electrical service is reestablished. Conversely, it may be desirable to leave the motors connected during short voltage dips; this is time-delay undervoltage protection. Instantaneous undervoltage protection disconnects the motor as soon as the voltage drops appreciably. This is satisfactory if continuity of operation is relatively unimportant. It is inherent in low-voltage magnetic starters when a power loss drops out all contactors as soon as a voltage dip occurs. If time-delay undervoltage protection is desired for these controls, time-delay relays must be added to the standard control circuit. Because circuit breakers do not drop out on a voltage dip, undervoltage relays are necessary.

Reverse-Phase Protection Reverse-phase relays are used on some large motors to prevent their starting when the electrical-system phase rotation is reversed because of improper wiring or maintenance. They are also used as undervoltage and voltage-balance relays. Individual relays may be applied to each motor in place of the undervoltage relay, or one relay may be operated off a bus for several motors. Individual relays are more expensive but more reliable, particularly when motor circuits are changed frequently. This type of protection is normally supplied only on high-voltage switch-gear-type starters.

Phase-Current Balance Protection Three-phase ac motors will usually continue to operate on single phase. Single phasing is serious on large ac motors because of the severe rotor heating it causes. Single-phase conditions cannot be detected by measuring voltage; a running motor acts as a generator so that, even under single-phase conditions, motor terminal voltage is nearly normal. Current-balance relays give a positive indication of system current unbalance and single-phase operation. Normally one three-phase relay is used for each motor. The use of these relays is restricted to large motors [approximately 1119 kW (1500 hp) and larger] when the value of the equipment protected justifies the cost of this protection.

Adequate single-phase protection is provided on low-voltage ac motor starters by three overload relays, which are now standard. Rotor heating is not particularly a problem on smaller motors which have more thermal capacity, but it is important to protect the stator windings of these machines against burnout.

Differential Protection Differential protection is applied to detect internal motor faults quickly and limit damage. The cost of this protection is justified on large motors [1119 kW (1500 hp) and above], for which limiting the motor damage may save the cost of this additional protection many times over.

Motor differential protection is one of the most sensitive forms of large-motor protection available. Figure 29-8 illustrates the basic principles involved. All six leads (both ends of all three windings) are brought out to terminals. The electric current entering each winding and the current leaving that winding pass through the same current transformer in opposite directions. If everything is normal, these currents are equal and no current is induced to the current transformer winding. If a phase-to-phase (winding-to-winding) or winding-to-ground short circuit occurs, the currents do not balance, current is induced in the current-transformer winding, and the differential relay operates instantaneously, shutting down the motor. Because of its sensitivity and speed, this system limits motor damage, minimizing repair costs and downtime.

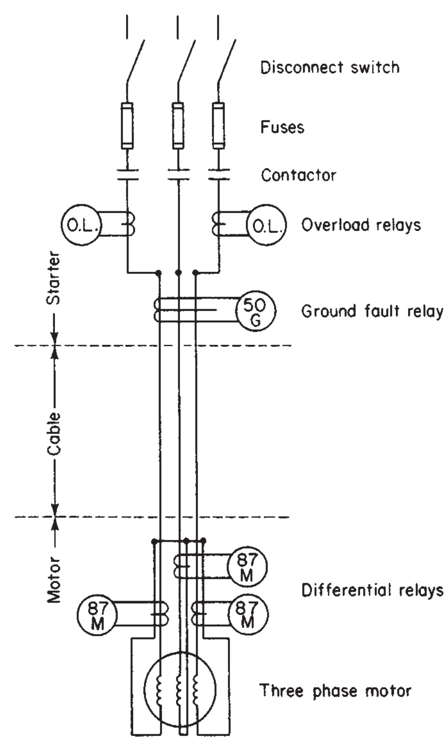


FIG. 29-8 Typical high-voltage ac motor starter illustrating several protective schemes: fuses, overload relays, ground-fault relays, and differential relays with the associated current transformer that act as fault-current sensors. In practice, the differential protection current transformers are located at the motor, but the relays are part of the starter.

Ground-Fault Protection High-voltage motors (2300 V and above) should be protected with ground-fault relays if the power source is grounded (see Fig. 29-8). This scheme includes a large-diameter current transformer (CT) encircling all three motor leads. Short-circuit current to ground flows through the CT to ground and returns to the power source external to the CT; this unbalance induces current in the CT and ground relay to shut down the motor. With this protection only two overload relays and two line CTs (rather than the standard three) are required, so the additional protection is very economical. It cannot be used, however, unless the power source is grounded.

Both differential and ground relaying detect ground faults. Ground-fault protection is located at the starter and protects the cable and the motor; differential CTs are located at the motor and protect the motor only. Economic priorities indicate ground-fault protection first, adding differential protection when justified by potential savings in downtime and repair costs.

Surge Protection High-voltage motors should be equipped with surge-protection apparatus consisting of a set of three lightning arresters and three surge capacitors. Potentially damaging voltage surges or spikes can be generated on the power system by switching operations, certain faults, or lightning. The surge capacitors slope off these steep front voltage spikes, and the lightning arresters limit the peak voltage; both functions are essential for adequate protection. Surge protection should be located at each motor's terminals for maximum protection, although in many instances one set of surge equipment is connected to the electrical bus serving several motors.

Special Control

Reduced-Voltage Starting Reduced-voltage starting is used to reduce system voltage dip. Voltage dips must be limited; otherwise, they may drop other motors off the line, cause synchronous motors on the system to pull out of step, or cause objectionable lamp flicker.

Resistor and reactor starting are the simplest methods of reduced-voltage starting. These systems require two contactors or breakers and a set of reactors or resistors, in contrast to the single-contactor full-voltage starter. The starting contactor closes first, connecting power to the motor terminals through the reactors. The impedance in the circuit reduces the motor terminal voltage and the starting current. As the motor approaches full speed, the running contactor closes, shorting out or bypassing the reactors, applying line voltage to the motor terminals. Starting current is reduced in proportion to the reduction in motor voltage. However, torque is proportional to the square of motor voltage, so starting torque is reduced far more than starting current.

If a greater reduction in line current is required for starting ac motors than is possible with reactor starting, autotransformers may be used. Because of transformer action, the reduction in motor-starting torque is directly proportional to the reduction in line current. Table 29-3 compares reactor and autotransformer starting with respect to line current and torque. Other, less commonly used methods of reduced-voltage starting include part-winding starting and star-delta starters.

TABLE 29-3 Effects of Reduced Voltage Starting*

Starter type	Motor voltage	Motor current	Line or source current	Motor torque	Source voltage dip
Design	100	100	100	100	0
Actual full voltage	80	80	80	64	20
Reactor:					
0.8 tap	67	67	67	45	17
0.65 tap	56	56	56	31	14
0.5 tap	44	44	44	20	11
Autotransformer:					
0.8 tap	69	69	55	48	14
0.65 tap	59	59	38	35	10
0.5 tap	47	47	24	22	6

*Values shown are in percent of design or normal starting values and are calculated for an arbitrary hypothetical power source whose voltage would dip by 20 percent if full-voltage starting were used.

Synchronous-Motor Starters Except for the addition of the synchronous-motor field-application panel, control schemes are identical for both synchronous and induction motors. Excitation is not applied to synchronous motors until they reach approximately 95 percent speed. Field current should be applied when the field poles are in proper space relationship to the stator's rotating magnetic field. Both speed and position are indicated by the ac voltage generated in the field winding. The frequency is directly proportional to slip and therefore indicates speed; the magnitude and polarity of the generated wave indicate position relative to the armature field. When the proper speed and position are detected, field current is applied. When reduced-voltage starting is employed, the ac starting sequence is completed before the application of field current.

Multispeed Alternating-Current Starters Multispeed induction motors are either two-winding motors, single-winding motors with consequent-pole connection or pole-amplitude-modulated motors (see subsection "Alternating-Current Motors, Multispeed"). The starters for two-winding, two-speed motors are quite simple; they consist of two standard single-pole, single-speed starters in the same enclosure with appropriate mechanical and electrical interlocks so that the two contactors cannot be closed simultaneously.

Two-speed, single-winding motors, either consequent-pole or pole-amplitude-modulated, require a three-pole and a five-pole contactor mechanically and electrically interlocked. Three- and four-speed, two-winding motors require a combination of two-speed, single-winding and two-speed, two-winding starters. Further modifications are possible by making these multispeed-control-reversing.

Secondary Control of Wound-Rotor Motors Wound-rotor motors may be effectively reduced-voltage-started or have their speed controlled by using external secondary resistance. The addition of resistance into the secondary circuit of a wound-rotor motor reduces the starting current and affects the speed under load conditions.

When external secondary resistance is used for improved starting characteristics, short-time-rated resistors are employed. As the motor accelerates, steps of resistance are cut out on a time or current basis to give the desired accelerating torque and current characteristics.

When external secondary resistance is used for speed adjustment, the resistors may be either infinitely adjustable (e.g., liquid rheostats) or adjustable in steps (if fine speed adjustment is not required).

Direct-Current Motor Control Control for dc motors runs the gamut from simple manual line starters to elaborate regulating systems. Only the starting problems are considered here since variable-speed drives and regulating systems are discussed elsewhere.

The major differences between ac and dc starters are necessitated by the commutation limitation of dc motors, which is the ability of the individual commutator segments to interrupt their share of armature current as each segment moves away from the brushes. Normally 250 to 275 percent of rated current can be commutated safely. Since motor-starting current is limited only by armature resistance, line starting can be used only for very small [approximately 1492-W (2-hp)] dc motors. Otherwise, the commutator would flash over and destroy the motor. External resistance to limit the current must be used in starting to prevent this.

Manual rheostats can be used in series with the motor armature for the current-limiting function. If the rheostat has ample thermal capacity, it can also be used to vary speed. If this system is used, interlocks should be included to prevent closing of the contactor unless maximum resistance is in the circuit.

Magnetic starters short out the starting resistance in one or several steps based on time, current, or speed. The number of steps depends on the size of the motor and the application. Current-limit acceleration is used frequently for high-inertia drives which require a long accelerating time. Motor current is sensed by a current relay which actuates the shorting contactors in sequence as the current drops. Time-limit acceleration is more common. The motor accelerates in a definite time by shorting out the starting resistor steps in timed sequence.

RECIPROCATING ENGINES

STEAM ENGINES

The advent of electric motors, steam turbines, and other drivers has relegated the steam engine to a minor position as an industrial driver. It does have the advantages of reliability and operating characteristics that are not obtainable with other drivers but also the disadvantage of bulkiness and oily exhaust steam.

In the simple **nonexpanding engine** as used with direct-acting reciprocating pumps, steam is admitted over the entire stroke and does not expand in the cylinder, resulting in relatively low efficiency. Control is simple, and pump speed is regulated by steam throttling. By proper selection of the steam and pump piston sizes, these pumps can deliver high shutoff pressures, which can be used to overcome temporary blockages of pipe lines or for other situations requiring high pressure of short duration.

The higher-efficiency **expanding steam engines** use cutoff valves to limit steam admission to the cylinder during the initial part of the stroke, and the expansion occurs during the remainder of the stroke. Larger engines use several cylinders in series to achieve full expansion. Although this type of engine can be controlled with throttling valves, the preferred method is to change the cutoff point, thus eliminating throttle-valve losses and permitting change in output from zero to maximum design power. Thus almost complete expansion is achieved at part loads and overloads, resulting in efficient operation for the full range of loadings.

Although this type of engine is efficient, it is limited by its inability to utilize low-vacuum exhaust and/or high steam pressures and temperatures commonly used by steam turbines. With low steam pressures [2068 kPa (300 psig)] and low vacuums [88 kPa (26 in Hg)] a steam engine will have a better efficiency than a steam turbine of the same rated power. For each cutoff setting an engine will develop the same torque at all speeds, with steam consumption and power output directly proportional to speed. All other drivers, except certain dc electric motors, require the same input for constant torque at varying speeds.

Changing the cutoff-point setting will allow an engine designed for one gas to operate efficiently on any other gas (limited only by corrosiveness, fouling, etc.). This characteristic favors the use of reciprocating engines when it is necessary to expand gases efficiently in process applications in which the composition is variable. A further advantage is that an engine is the only driver with essentially zero gas consumption at zero speed while developing full torque and maintaining full process pressures. A combination of an expansion engine driving an oil brake provides a high-efficiency refrigeration effect over a wide range of process conditions, with the process controllers throttling the oil flow in the oil brake.

The **uniflow design** reduces cylinder condensation and also allows greater expansion ratios per cylinder (see Figs. 29-9 and 29-10). Steam is admitted during the start of the power stroke and after cutoff is expanded to a pressure slightly higher than the exhaust pressure. At the end of the stroke the piston uncovers the exhaust ports of the cylinder with partial steam and discharges into the exhaust system. The steam remaining in the cylinder is compressed during the return piston stroke, maintaining higher average cylinder temperatures in order to reduce steam condensation during admission.

To reduce friction and cylinder wear oil is injected into the cylinders of engines, and to maintain lubricity cylinders of engines on low-temperature services are warmed. Oil causes foaming in boilers and can contaminate low-temperature process streams. Therefore, in steam plants oil is usually removed from the condensate. By using carbon or plastic rings similar to those used in oil-free reciprocating compressors, oil for lubrication can be omitted. But these rings are not as reliable as lubricated cast-iron piston rings.

INTERNAL-COMBUSTION ENGINES

Internal-combustion engines range in size from small portable gasoline engines to over 14,914 kW (20,000 hp) diesels for ship propulsion. They are usually designed for particular industrial applications and to meet specific objectives as to weight per horsepower, reliability, and operating conditions.

All internal-combustion engines fall into two main types, namely, four-cycle and two-cycle engines. These engines may be further classified as (1) gasoline or gas engines (Otto cycle), in which a spark plug is used to ignite a premixed fuel-air mixture; (2) diesel engines (diesel cycle), in which high-pressure compression raises the air temperature to the ignition temperature of the injected fuel oil; (3) dual-fuel or gas-diesel engines, in which the fuel is a combination of gas and oil in any desired ratio, provided that at least 5 percent oil is used at all times; and (4) trifuel engines, which can operate as dual-fuel or as straight gas engines by replacing the oil-injection system with a spark plug for ignition.

Design Characteristics Internal-combustion engines involve consideration of the following design features: (1) **Compression ratio,** an increase of which usually increases engine efficiency but also results in higher average cycle temperatures and therefore hotter cylinders, piston heads, and rings which increases the difficulty of piston lubrication. (2) **Piston speed,** which is a major overall criterion of engine design since power rating is proportional to piston speed. Reciprocating forces and lubrication problems increase with piston speed. (3) **Brake mean effective pressure** (bmep), which is an overall measure of the output of an engine frame; bmep increases with compression ratio and degree of supercharging, and, in general, high values are associated with modern high-efficiency industrial engines. It is also the overall criterion for bearing loadings and average piston-head and cylinder temperature. (4) **Engine rating,** which is proportional to the product of piston speed and bmep. Acceptable values of piston speed and bmep are more dependent on industrial usage than on the type of engine. (5) **Supercharging,** which connotes means for increasing the inlet manifold air pressure above ambient

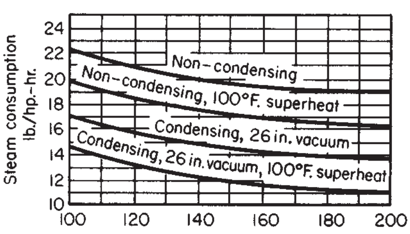


FIG. 29-9 Steam consumption of a 400-hp uniflow engine. To convert pounds per horsepower-hour to kilograms per kilowatthour, multiply by 0.6084; to convert pounds-force per square inch to kilopascals, multiply by 6.89.

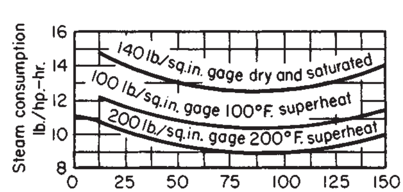


FIG. 29-10 Steam consumption of a uniflow engine with 27.5-in vacuum. To convert pounds per horsepower-hour to kilograms per kilowatthour, multiply by 0.6084; to convert pounds-force per square inch to kilopascals, multiply by 6.89.

pressure. (6) **Turbocharging,** which applies to the expansion of the hot exhaust gases through a turbine to drive the supercharging compressor. Since power output is proportional to inlet manifold air pressure, supercharging provides increased power output and usually higher efficiency. Highly supercharged engines may require an air cooler between the compressor and inlet manifold.

Engine size is usually in terms of power rating (horsepower) rather than the physical size of the engine frame. Frame size is determined by the diameter of the cylinder bore, length of stroke, and number of cylinders. The power rating of a given frame varies with industrial practice and usage; accordingly, automobile engines are designed to develop between 0.54 and 0.95 hp/in^3 of piston displacement, while industrial diesels are limited to 0.045 to 0.60 hp/in^3.

Engine rating reflects industrial practice. Automobile engine rating is the peak horsepower developed on a test stand, whereas industrial engine rating is usually in terms of continuous load.

Industrial engines are made in a wide variety of frame sizes, each offered at several power and speed ratings. The same engine frame might drive either a 900-kW generator at 900 r/min or a 600-kW generator at 600 r/min by using lighter pistons and special parts for the higher-speed application. An engine originally designed for 746 kW (1000 hp) may be uprated to 820 kW (1100 hp) by higher compression heads and further to 984 kW (1320 hp) by supercharging. Thus one frame can cover a power range from 447 kW (600 hp) (60 percent speed) to 984 kW (1320 hp). An engine frame can also be designed for different output by varying the number of cylinders; for example, a 10-cylinder 2610-kW (3500-hp) engine will develop 1566 kW (2100 hp) and 2088 kW (2800 hp) at the same efficiency with six and eight cylinders respectively. This enables a manufacturer to use the same basic design, tools, and fixtures for a variety of ratings.

Maximum rating is the horsepower capability of the engine that can be demonstrated within 5 percent at the factory corrected to standard conditions. Standard conditions in this case are at sea level with barometric pressure of 760 mm (29.92 in) Hg and a temperature of 15° C (59° F). **Intermittent** is the horsepower and speed capability of the engine which can be utilized for approximately one hour, followed by about an hour of operation at or below the continuous rating. Many engine operators choose to run their engines continuously in this rating area, trading shorter maintenance periods for increased earnings. **Continuous** is the horsepower and speed capability of the engine which can be utilized without interruption or load cycling. This can extend for months or years of operation if the engine is equipped for on line lube oil and filter changes. **Intermittent** and **continuous** ratings are at standard conditions of 746 mm (29.38 in) Hg and 30° C (86° F); turbocharged engine ratings are applicable to at least 760 m (2500 ft) elevation above sea level without derating.

It is important that atmospheric conditions at the installation site be specified, since engines operate with very little excess combustion air and consequently maximum power output is proportional to air density.

Operating Characteristics The operating characteristics of internal-combustion engines are basically the same regardless of fuel used or whether the engine is two- or four-cycle. To vary the speed and/or load-carrying capacity, which can range from zero to full torque for all speeds within the operating range, it is necessary only to vary the fuel input. Large engines use a governor to control the fuel-input rate and maintain constant speed under variations of load. With auxiliary instrumentation the governor can be used as a controller of power or process. Starting, stopping, and operation from a remote location are made possible by instrumentation that will also shut down an engine in the event of loss of cooling-water or lubricating-oil flow.

Starting is accomplished by rotating the engine at a speed sufficient to achieve ignition and self-sustained operation. Small engines are started with electric motors or smaller hand-starting engines, and large engines are provided with special valving whereby some of the engine cylinders can be operated as air motors, utilizing high-pressure air to rotate the engine. Starting motors are usually sized at 5 to 10 percent of the engine rating. Starting-air requirements are 0.014 m^3 (0.5 ft^3) (free air)/hp stored at a pressure of 1723 kPa (250 psig) to 2068 kPa (300 psig). Usually three starts per successful firing are assumed for sizing the starting air compressor and air-storage vessel.

Operating characteristics of engines are also influenced by **service requirements.** Automotive engines must develop maximum torque at lower speeds for hill climbing; marine engines are required to develop full torque only when the propeller is at full speed; generator drivers run at a single speed and therefore require maximum possible efficiency at all loads for this speed; and reciprocating compressors at constant pressure require full torque over the entire operating-speed range, thus presenting an engine-instability problem at reduced speeds. The engine must be selected to meet these service requirements.

Supercharging is employed primarily to improve engine efficiency or power output, but part-load performance and rate of response to load changes depend on the type of supercharging system used. Some systems result in supercharged engines having the same torque-speed characteristics and rate of response to load changes as nonsupercharged engines, while other systems result in poorer or better response and in different speed-torque characteristics.

Maintenance and Reliability Preventive maintenance requires that all engines be shut down at periodic intervals for inspection and repair. For properly maintained heavy-duty engines availability is over 97 percent, with maintenance costs of $2.50 to $5 per horsepower-year and lubricating-oil consumption of 1 to 2 gal/hp·year.* While this represents a high degree of reliability, outages of heavy-duty engines are more frequent than those of electric motors or steam turbines.

Satisfactory engine performance is assured by maintaining a minimum log that allows comparison of present characteristics with past records of (1) cylinder exhaust, cooling water, and supercharger exhaust temperature for similar loadings, (2) running sounds and smokiness of exhaust, and (3) engine efficiency and losses.

The minimum safety devices for an industrial engine are a governor and separate overspeed and low-oil-pressure trips. Engines operating in nonsupervised areas should be arranged to shut down in the event of cooling-water or lubricating-oil failures and for excessive exhaust or jacket-water temperatures. Engines operating in supervised areas should be provided with instruments for the operator to check performance.

Fuel Characteristics Fuels used in industrial engines of the internal-combustion type are usually derivatives of petroleum or else natural or manufactured gases. Alcohols and mixtures of gasoline and alcohol or benzol can also be used. A gas engine will operate satisfactorily on any gas which is free of dust, noncorrosive (i.e., less than 0.6 grains/ft^3), does not detonate, does not preignite during compression stroke, and produces enough heat on burning to develop power.

In general, the fuel must have a heat capacity of over 600 Btu/ft^3. Gasoline engines require, in addition, that the fuel will vaporize in the carburetor. Diesels will burn any fuel that can be injected, provided that it will burn under controlled conditions, possesses sufficient lubricity to lubricate the injection plungers, will supply enough heat, and is grit-free, containing less than 3 percent sulfur, 70 ppm vanadium, and 125 ppm vanadium pentoxide. Most diesel engines use either No. 2 or No. 5 fuel oil. The latter must be heated to a viscosity of 50 to 70 SSU [121°C (250°F) approximately] for proper injector lubrication and injection characteristics.

Gaseous fuels containing fractions whose ignition temperature is lower than that of methane may require the use of low-compression heads and a resulting derating of the gas engine.

The method of reporting fuel consumption varies among different industries and also among countries. Trade associations usually have recommended procedures. Thus the Diesel Engine Manufacturers

* For further details of industrial operating costs see *Annual Oil and Gas Engine Power Costs,* published by the American Society of Mechanical Engineers. Engine manufacturers will supply recommended schedules for preventive maintenance.

Association (United States) calculates efficiencies based on the lower heating value (LHV) for gas fuels and the higher heating value for oil fuels. It is general practice to report gas-engine performance in terms of British thermal units per horsepower-hour (LHV) and oil-engine performance in terms of pounds of fuel consumed per horsepower-hour. For electric power plants, fuel consumption is reported in terms of kilowatts. Auxiliaries included with engine-efficiency calculations vary with industry practice.

Fuel Economy and Heat Recovery The high overall efficiencies obtained by modern high-compression supercharged diesels or gas engines can be approached only by very large high-pressure reheat steam plants or by very complicated gas-turbine cycles. The following efficiencies (LHV) based on methane fuel for gas engines and oil fuel for diesel engines can be used for estimating fuel consumption: 63 kW (85 hp) to 298 kW (400 hp), 28 percent; 328 kW (440 hp) to 597 kW (800 hp), 32 percent; 597 kW (800 hp) to 2237 kW (3000 hp), 36 percent; and over 2461 kW (3300 hp), 41 percent.

Instead of working with efficiencies (Eff.) the term **brake-specific fuel consumption** (BSFC) is frequently used with liquid fuel engines. BSFC is expressed in g/kWh (lb/Hp·h) and assumes equal fuel or heating value (LHV) when comparing different engine performances. Using the relationship BSFC = 1/LHV × Eff and LHV = 43 MJ/kg (18,400 Btu/lb) for gasolines and diesel fuels, efficiencies can be calculated from Eff = 84/BSFC (Eff = 0.14/BSFC).

Plant fuel costs chargeable to power production can be reduced if heat losses can be utilized to provide process or other heating. Engine heat losses as percentages of heat input (LHV) are (1) lubricating-oil cooling, 5 to 7 percent available at 82° C (165° F); (2) jacket cooling, 17 to 30 percent available as 82° C water or with vapor-phase cooling as 103 kPa/ (15 psig) steam; (3) engine exhaust gases, 26 to 30 percent available with approximately half of this at sufficient temperature to generate 689.5 kPa/ (100 psig) steam in waste-heat boilers. Table 29-4 gives some heat-balance data for gas engines.

Vapor-phase cooling reduces the cost of the cooling system, increases heat recovery, and may result in improved engine efficiency.

At full speed fuel consumption decreases linearly with reduction in load, becoming at zero load almost one-third to one-fourth of full-load consumption and with no potential for heat recovery in the exhaust. Jacket-water heat losses decrease by 20 to 40 percent at zero load. Fuel consumption at rated torque is almost proportional to speed. At half speed exhaust recovery decreases to less than half, and jacket-water cooling becomes more than half of full-torque values.

TABLE 29-4 Approximate Heat Balance for Gas Engines—5965 kW (8000 bhp)

	Two-cycle			Four-cycle		
	U.S. units	SI units		U.S. units	SI units	
	Btu/BHP·hr	kJ/kWh	%	Btu/BHP·hr	kJ/kWh	%
Heat Input (LHV)	6500	9196		6200	8772	
Useful Work	2545	3601	39.2	2545	3601	41.0
Cooling System	940	1330	14.5	1000	1415	16.1
Lubricating Oil System	292	413	4.5	300	424	4.8
Intercooler System	530	750	8.2	250	500	4.0
Exhaust	1540	2179	23.7	1685	2384	27.2
Radiation	650	920	10.0	200	283	3.2
Combustion	—	—	—	220	311	3.5

Emission Control Internal cumbustion engines must meet national, state, and regional exhaust emission regulations. On an international scope, many countries have introduced emission regulations on stationary engines. These regulations not only vary from one location to the next, but they also tend to become more restrictive as time goes by. Further, as fuel costs rise, the economics of various emission control techniques are changing. Current emission control measures are given in Table 29-5.

Installation and Costs An engine installation includes auxiliary equipment necessary for operation, such as lubrication pumps and attendant storage, filtering and cooling equipment; jacket-water pumps with expansion tank and cooler; starting-air tanks and compressors; inlet-air piping and screens; exhaust-air piping and silencers or waste-heat boilers; a fuel system, which in the case of diesels would include a day tank, filters, pumps, heaters, and a main storage tank; ignition system or fuel-injection plungers; and cooling towers or radiators. Diesel engines operating on heavy residual fuel oils would have two-fuel systems, a heavy-fuel-oil system for normal operation and a light-fuel-oil system for starting and stopping.

Pipe-line and marine installations are frequently arranged so that the engine drives all its auxiliaries from the crankshaft by means of chains and V belts. But process-plant practice is to have all the auxiliaries independently driven, using standby pumps to minimize engine downtime.

Foundations should be designed to control **vibrating motion** resulting from reciprocating masses. Engine manufacturers will recommend the size of the foundation, but usually their recommendations do not take soil properties into account and are based on making the combined engine and foundation weight sufficiently large to limit vibration. When possible, foundations should be separate from the building structure. In many cases vibration of the engine will cause no damage; nevertheless, it is good practice to reduce it whenever possible. Even though vibration does not increase any forces in the engine, it can loosen pipe joints, nuts, etc.

Torsional vibration can also be a problem and results from pressure variations in the cylinders which can produce cyclic torques with harmonics ranging from half speed to 10 or 12 times running speed.

Table 29-6 gives costs for engine installation, which can be prorated to make preliminary estimates of installation costs.

To avoid operating difficulties, the torsional critical frequencies of the combined engine and driven equipment should be calculated or measured to assure that operating speeds are removed from these criticals or that vibration dampers are provided or that the equipment is designed for the resulting cyclic stresses.

The costs of both engines and auxiliaries are reasonably consistent on the basis of dollars per horsepower as long as essential details are

TABLE 29-5 Current Emission Reduction Measures

Type of engine	Emissions	Emission reduction measures
Gas-Otto-Engine 20 kW (27 hp) to 2,000 kW (2,685 hp)	NOx CO HC	Air/fuel ratio = 1- and 3-way catalytic converter Stratified charge technology (lean burn) and enhanced turbocharging Catalytic converter plus NH_3 injection
Diesel Engine 5 kW (7 hp) to 4000 kW (5400 hp)	NOx CO HC PM*	Small engines: Prechamber Midrange: Adjustment for maximum performance and fuel economy Large engines: Exhaust filter and catalytic converter with NH_3 injection
Diesel-Gas-Engine 300 kW (400 hp) to 8,000 kW (10,740 hp)	NOx CO HC PM*	Catalytic converter Catalytic converter with NH_3 injection

*Particulate matter (soot).

TABLE 29-6 Comparative Installation Costs of Integral-Engine Compressors for Pipeline Stations and Process Plants*

	Pipeline station	Process plant
A. Land and improvements	$ 459,200	$ 67,900
B. Structures	1,036,800	553,600
C. Testing	40,000	40,000
D. Equipment	8,183,000	6,053,750
Subtotal	9,719,000	6,715,250
Add 10% for overhead and undistributed field costs	971,900	671,525
Subtotal	10,690,900	7,386,775
Add 5% for contingencies	534,545	369,339
Total	11,225,445	7,756,114
Cost per horsepower	802	554

*10 units are assumed for a total of 10, 4 MW (14,000 hp). Basis year is 1993.

TABLE 29-7 Typical Cost of Engine-Driven Equipment (1994 basis)

	$/hp
Integral engine compressors:	
Uninstalled and without auxiliaries	334
Installed cost with auxiliaries and cooling water	541
Installed costs of large units with cooling water supplied from process	532
Diesel or gas-engine generators:	
Uninstalled	282
Installed	544

the same. Published figures on **installed engine costs** are often misleading, since with supercharging more power output can be obtained from the same size of engine, which also reduces cooling-water and foundation requirements. Pipe-line compressor costs frequently include piping, buildings, etc.; and in some process plants, cooling water which has been used and charged against process operation can be reused for engine cooling at no cost. It is obvious that general cost predictions must be used with caution unless their detailed basis is known. However, as preliminary figures, Table 29-7 may prove useful (1994 basis):

STEAM TURBINES

Steam turbines are divided into two broad categories: those used for generating **electric power** and general-purpose units used for driving pumps, compressors, etc., and frequently called **mechanical-drive** turbines.

Figure 29-11 illustrates in general the relationship of capability versus speed. At 1800 and 3600 r/min are the turboelectric generator drives with capability limits above the top of the chart. The majority of mechanical-drive applications are within the shaded area; capabilities above the solid line are special and unusual.

Inlet-steam pressure is usually in the range of 1723 kPa (250 psig) at zero superheat to 5860 kPa (850 psig) at 482° C (900° F). Some turbines have been built to operate at 35 kPa (5 psig) with zero superheat from a process exhaust. Pressures of 10,342, 12,410, and 16,547 kPa/(1500, 1800, and 2400 psig) are common for large turbine generators, and some operate at supercritical pressures of 24,131 kPa/(3500 psig) and 34,474 kPa (5000 psig). Power plants that generate steam with nuclear reactors generate saturated steam in the range of 1379 to 6895 kPa (200 to 1000 psig). Early units were 125 MW, but currently 250 to 1500 MW are the most common size. These units have multiple casings and 1.32-m-(52-in-) long blades in 1800-r/min exhaust stages.

TYPES OF STEAM TURBINES

Straight Condensing Turbine All the steam enters the turbine at one pressure, and all the steam leaves the turbine exhaust at a pressure below atmosphere.

Straight Noncondensing Turbine All the steam enters the turbine at one pressure, and all the steam leaves the turbine exhaust at a pressure equal to or greater than atmosphere.

Nonautomatic-Extraction Turbine, Condensing or Noncondensing Steam is extracted from one or more stages, but without means for controlling the pressures of the extracted steam.

Automatic-Extraction Turbine, Condensing or Noncondensing Steam is extracted from one or more stages with means for controlling the pressures of the extracted steam.

Automatic-Extraction-Induction Turbine, Condensing or Noncondensing Steam is extracted from or inducted into one or more stages with means for controlling the pressures of the extraction and/or induction steam.

Mixed-Pressure Turbine, Condensing or Noncondensing Steam enters the turbine at two or more pressures through separate inlet openings with means for controlling the inlet-steam pressures.

Reheat Turbine After the steam has expanded through several stages, it leaves the turbine and passes through a section of the boiler, where superheat is added. The superheated steam is then returned to the turbine for further expansion.

STAGE AND VALVE OPTIONS

The **single-stage, single-valve** turbine is the simplest turbine, and it sees the most varied application. There is a single governor valve in a steam chest, operated directly from a mechanical flyball governor. After passing through the valve, steam is expanded through the nozzles, where it gains velocity and momentum for driving the wheels by impulse action against the blades. The shaft is sealed by carbon rings. The sleeve bearings are ring-oiled. The majority of applications are below 1119 kW (1500 hp) and at speeds below 5500 r/min with 4137 kPa (600 psig) or lower steam pressure. By certain changes higher limits such as 1492 kW (2000 hp), 10,000 r/min, and 8274 kPa (1200 psig) can be made available.

The **multistage, single-valve turbine** is widely used for driving compressors and pumps in the range from 1119 to 4474 kW (1500 to 6000 hp). Figure 29-12 shows a section of a turbine of this type. The inlet end of this multistage turbine retains the general arrangement of bearing case, governor, and steam chest as used on the single-stage, single-valve turbine. The casing is extended to contain the added stages, and the last blade row and the exhaust opening are large in order to contain the volume of the exhaust steam at the low condensing pressure, which may be 6.9 or 13.8 kPa(a) (1 or 2 psia).

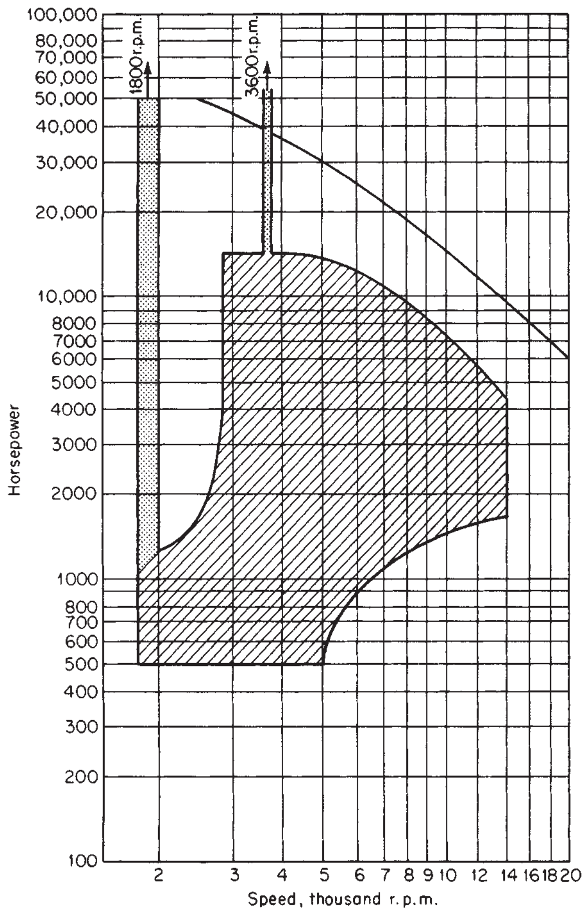


FIG. 29-11 Steam-turbine capability versus speed. To convert horsepower to kilowatts, multiply by 0.7457.

TYPES OF BLADES AND STAGING

Figure 29-13 illustrates a turbine stage in which steam at pressure p_1 enters the nozzle or stationary blade and expands to a lower pressure p_2, leaving the nozzle at a velocity of C_1. The rotating row is moving at a velocity μ so that steam enters the rotating row at a relative velocity w_1 and leaves the rotating row at a relative velocity w_2. The pressure on the exit side of the rotating row is p_3.

Depending upon the relationship between the pressures p_1, p_2, and p_3, the stage is classified as either impulse or reaction.

For an **impulse** (Rateau or Curtis) **stage,** p_2 is equal to p_3 or only slightly higher, and w_2 is slightly less than w_1 as a result of friction loss in the blade passage. The exit area a_r of a rotating row is 50 to 74 percent larger than the exit area a_s of the stationary row in order to pass the same quantity of higher-specific-volume steam.

The work done in the stage, which is the push of the steam on the blades, is the change in momentum of the steam as it alters direction from C_1 to C_2, using the peripheral projection of the velocities. Therefore, it is desirable for the angles α and β to be very small.

For a **reaction stage** the exit area a_r is reduced by reducing the angle β. This will increase the pressure p_2, possibly to midway between p_1 and p_3. The exit velocity w_2 is now greater than the blade entrance velocity w_1 because of the pressure drop from p_2 to p_3 through the blade passage. The reaction in a stage is expressed as a percentage of stage available energy.

Because of the smaller blade angle the reaction stage is more efficient than the impulse stage, but it requires more stages for the same pressure drop. This will increase the losses and the leakage, and the choice is generally close to a standoff.

A **Curtis stage** is an impulse stage that makes use of two rotating rows to absorb the energy in C_1 with a stationary reversing row between them. This comes about when C_1 is high compared with the wheel peripheral velocity μ, so that the exit velocity C_2 still has a lot of energy left in it. This energy is dissipated by adding a stationary row to reverse the flow and then putting it through an additional rotating row. It is universal practice for single-stage turbines when the ratio p_1/p_3 is in the range of 2 to 2.5 or greater. The two-row impulse (Curtis) stage is not as efficient as the single-row stage, owing to the higher losses encountered with the high steam velocities and repeated turning of the stream.

PERFORMANCE AND EFFICIENCY

The energy available in the steam is expressed in British thermal units per pound, or enthalpy. The velocity of the steam flow through the nozzle is calculated from

$$C_1 = 223.7\sqrt{\Delta h} \tag{29-14}$$

where C = velocity, ft/s, and Δh = enthalpy drop from p_1 to p_2, Btu/lb. This is the same formula that is used for the spouting velocity of liquid in terms of head by introducing the mechanical equivalent of heat.

For calculation purposes the **efficiency of individual turbine stages** is plotted as efficiency versus velocity ratio μ/C_1, where μ equals wheel peripheral velocity. Figure 29-14 shows the relation between the three principal types of stages; each one has a peak efficiency at a certain ratio of μ/C_1. For the same revolution per minute, of course, the two-row stage takes the highest value of C_1 and the highest enthalpy drop. The reaction stage takes the lowest. The single-row impulse (Rateau) stage is in the middle. The reaction stage is denoted as 70 percent, because 30 percent of the enthalpy is allowed for expansion in the stationary row and 70 percent in the rotating row.

Steam Rate Enthalpy data can be obtained from Mollier diagrams or from steam tables (see Sec. 2), from which the **theoretical steam rate** can be calculated. For example, a throttle inlet condition of 4137 kPa (600 psig) and 399° C (750° F) gives an enthalpy of 3.2 MJ/kg (1380 Btu/lb), and if the end point is at 348 kPa (50 psig), then adiabatic expansion is to 2.69 MJ/kg (1157 Btu/lb). This gives 0.52 MJ/kg (223 Btu/lb) available, and the theoretical steam rate is calculated from the Btu equivalent per kilowatthour or horsepower-hour:

$$2544/223 = 11.4 \text{ lb steam/(hp·h)}$$

Theoretical-steam-rate tables are available as separate publications. Table 29-8 covers some common conditions.

The **actual steam rate** is obtained by dividing the theoretical steam rate by the turbine efficiency, which includes thermodynamic and mechanical losses. Alternatively, internal efficiency can be used, and mechanical losses applied in a second step.

Efficiency varies over a wide range, dependent upon the number of stages in the turbine. If steam conditions are assumed to be 4 137 kPa (600 psig) at 399° C (750° F) inlet and 13.8 kPa (2 psia) exhaust, Table 29-8 shows a theoretical steam rate of 3.47 kg/kWh (7.65 lb/kWh). With efficiencies that might be experienced for the given steam conditions with single-stage, five-stage, seven-stage, and nine-stage turbines, the actual steam rates would be as in Table 29-9.

From this table note that efficiency increases with the number of stages and that the increased number of stages corresponds to larger horsepower values. For each stage, as characterized by diameter and speed, there is a Btu drop that gives the best efficiency provided there is enough steam to fill the stage so that it operates with minimum friction and windage loss. Thus the nine-stage turbine is fine for 7.46 MW (10,000 hp), but it would not show up as well as a five-stage turbine at 0.746 MW (1000 hp) because the losses would increase with the light flow.

From the velocity diagram in Fig. 29-13 it is apparent that an increase in wheel peripheral velocity μ permits an increase in nozzle exit velocity C_1 without increasing C_2. Accordingly, a high-speed tur-

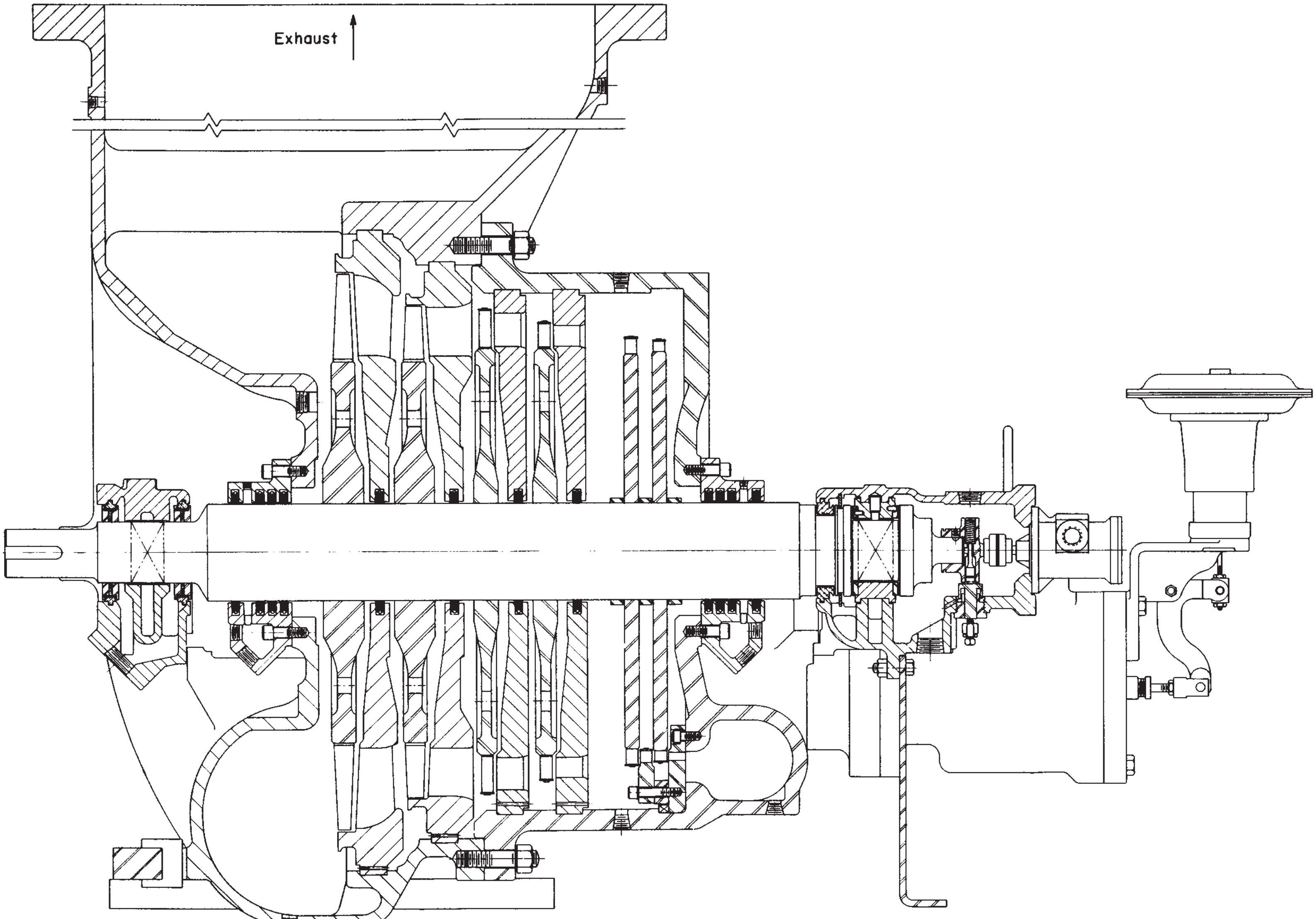


FIG. 29-12 Single-valve, multistage steam turbine. (*Elliott.*)

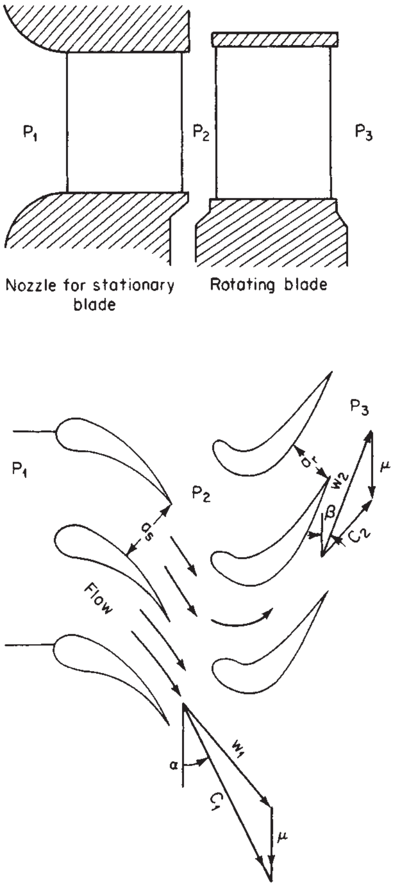


FIG. 29-13 Basic mechanics of a turbine stage.

bine can use more Btu per stage and will have fewer stages than a slow-speed turbine.

The leaving velocity C_2 is a measure of the unused energy. For best efficiency C_2 should have no radial component; C_2 should be straight axial. For all stages except the last one, C_2 represents a carryover to the next stage. For the last stage, C_2 is the velocity into the exhaust hood and is referred to as the leaving loss or exhaust loss.

The curves in Figs. 29-15 and 29-16 can be used for estimating **steam rates of single-stage turbines** by proceeding according to the following example. For steam conditions of 2760 kPa (400 psig) and 399° C (750° F) inlet and 5170 kPa (75 psig) exhaust, Table 29-8 gives theoretical steam rate as 20.59 lb/kWh. If the turbine is 300 hp and 4000 r/min, enter the top of Fig. 29-15 at 4000 and for a trial stop at 18-in-base diameter of turbine blading, then drop to TSR 20.59, and the base steam rate is 37 lb/hp·h. Next enter the top of Fig. 29-16 at 4000 r/min, and intersect 75 psig, then drop to 18-in diameter and find 8-hp loss. Total horsepower then is 308.5, and steam required is (308.5) (37.0) = 11,400 lb/h. By reading values for other diameters a selection table can be prepared, shown as Table 29-10.

From this table the most efficient unit can be selected and balanced against price. It is apparent that for 300 hp the 28-in diameter achieves no gain over the 22-in diameter because of the increase in horsepower loss. For 22 in versus 18 in the gain is small and may be offset by the higher price.

Steam rates for multistage turbines depend upon many more variables than do single-stage turbines and require extensive computation. Depending upon the type of turbine, single-valve or multivalve, general-purpose or generator-drive, condensing or noncondensing, with or without extraction, the manufacturers have shortcut procedures for estimating performance in their bulletins for the different types. As a general approximation the curve in Fig. 29-17 may be used, if one keeps in mind that an actual turbine may be several points above or below the curves, depending upon the use of optimum staging for efficiency or a compromise to meet price. Speed is also important; high speed at 373 kW (500 hp) may have high losses, while 7460 kW (10,000 hp) at 12,000 r/min may be above the curve. And steam pressure affects performance. The most efficient turbine is one in which speed, pressure, and steam flow combine to fill the blade path so that there are no partial-admission stages. For partial admission the nozzles do not fill the entire 360° arc because there is not enough steam for that many nozzles. Those portions of the blades which are spinning outside of the nozzle arc create friction and windage.

TURBINE CONTROL

A turbine may be speed-, pressure-, or process-controlled. Some of the terms used are defined as follows:

Speed-governing system includes the speed governor, the speed changer, the servomotor that moves the valves, and the governor-controlled valves.

Speed governor includes only those elements which are directly responsive to speed and position the other elements.

The **speed changer** is a device by means of which the set point may be varied.

Steady-state regulation is the change in sustained speed or pressure (expressed as a percentage of rated) when power or flow output is gradually reduced from rated value to zero.

TABLE 29-8 Theoretical Steam Rates for Steam Turbines at Some Common Conditions, lb/kWh

	Inlet conditions							
Exhaust pressure	150 lb/in^2 gage, 366°F, saturated	200 lb/in^2 gage, 388°F, saturated	250 lb/in^2 gage, 500°F, 94°F superheat	400 lb/in^2 gage, 750°F, 302°F superheat	600 lb/in^2 gage, 750°F, 261°F superheat	600 lb/in^2 gage, 825°F, 336°F superheat	850 lb/in^2 gage, 825°F, 298°F superheat	850 lb/in^2 gage, 900°F, 373°F superheat
2 in. Hg.	10.52	10.01	9.07	7.37	7.09	6.77	6.58	6.28
4 in. Hg	11.76	11.12	10.00	7.99	7.65	7.28	7.06	6.73
0 lb./sq. in. gage	19.37	17.51	15.16	11.20	10.40	9.82	9.31	8.81
10 lb./sq. in. gage	23.96	21.09	17.90	12.72	11.64	10.96	10.29	9.71
30 lb./sq. in. gage	33.6	28.05	22.94	15.23	13.62	12.75	11.80	11.07
50 lb./sq. in. gage	46.0	36.0	28.20	17.57	15.36	14.31	13.07	12.21
60 lb./sq. in. gage	53.9	40.4	31.10	18.75	16.19	15.05	13.66	12.74
70 lb./sq. in. gage	63.5	45.6	34.1	19.96	17.00	15.79	14.22	13.25
75 lb./sq. in. gage	69.3	48.5	35.8	20.59	17.40	16.17	14.50	13.51

NOTE: To convert pounds-force per square inch to kilopascals, multiply by 6.8948; to convert pounds per kilowatthour to kilograms per kilowatthour, multiply by 0.4536; °C = 5/9(°F − 32).

TABLE 29-9 Typical Stage Efficiencies for Steam Turbines at 600 psi Inlet Pressure and 750°F Inlet Temperature

Turbine design	Turbine hp	Internal efficiency, %	Exhaust enthalpy, Btu/lb	Δh,* Btu/lb	Steam rate
Single-stage	500	30	1245	135	7.65/0.30 = 25.5 lb/kw·hr
5-stage	1,000	55	1135	245	7.65/0.55 = 13.9 lb/kw·hr
7-stage	4,000	65	1090	290	7.65/0.65 = 11.75 lb/kw·hr
9-stage	10,000	75	1020	360	7.65/0.75 = 10.02 lb/kw·hr

NOTE: To convert horsepower to kilowatts, multiply by 0.7457; to convert British thermal units per pound to kilojoules per kilogram, multiply by 2.33.

*Based on inlet enthalpy = 1380 Btu/lb.

Speed variation is the total variation in speed from the set point and includes both dead band and oscillation.

Proportional-action governor is a governor with inherent regulation and a continuous linear relation between the input (speed change) and the output of the final control element, the governing valve.

Proportional-action governor with reset is a governor with inherent regulation so that the momentary output is proportional to input change, and subsequently a reset action initiated by the output acts on the speed changer or its equivalent to make the settled regulation less than the inherent regulation.

Isochronous governor is a floating-action governor that controls for constant speed. It is equipped with a dashpot or buffer to give momentary regulation for a speed-input change.

Control-System Components The three principal elements of a control system are the sensing device which measures the error as the deviation from the set point, means for transmission and amplification of the error signal, and the control output device in the form of a servo-operated valve. In the case of the **direct-acting flyball governor** (Fig. 29-18) these three elements are combined in the flyball element and the linkage that connects to the valve.

The centrifugal force of the weights is continually compared against the set point as established by the governor spring which opposes the force from the governor weights. For an increase in load the speed drops, and the weight force is reduced, which allows the spring to push the governor spindle to the left. The lever then pivots, and the valve moves to the right and opens to increase steam flow and torque. The feedback in the control is the resultant increase in speed to match the set point of the governor spring again and eliminate the error. This governor corresponds to NEMA Class A, 10 percent speed regulation, that is, 10 percent speed rise from full load to no load, and the governor weights must move out to close the valve.

The force required to position the valve and the specified regulation puts a practical limit on the direct-acting governor. Beyond this limit a servo is required. The servo requirement has three levels. The lowest level is for a single valve of the balanced type in Fig. 29-18 which may be controlled with less than 889.6 N (200 lb) force. The next level is a single valve with single seat of the venturi type; partially balanced it requires 3558.4 N (800 lbf) to 4448 N (1000 lbf) force. The top level is the multivalve bar-lift or the cam-lift valve gear, in which 8- and 10-in and larger oil servos are used in developing several thousand pounds-force.

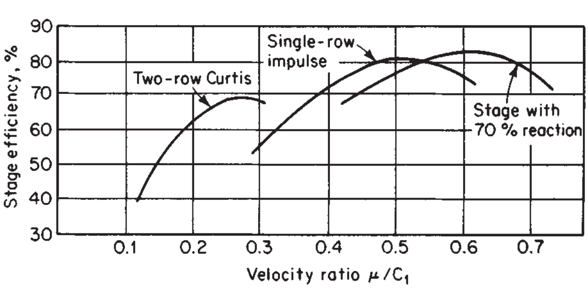


FIG. 29-14 Stage efficiency for different types of stages.

Speed-Control Systems The most common sensing element is mechanical; some systems are hydraulic or electronic. For valve positioner they all have a hydraulic servo as first choice, with an occasional choice of pneumatic for lighter loads.

Figures 29-19 and 29-20 illustrate two different **mechanical-hydraulic systems.** Figure 29-19 is a bar-lift steam chest with a heavy-duty hydraulic servo. The speed-sensing element is a flyball assembly attached to a rotating pilot. This rotating pilot sends a control-pressure signal that is proportional to speed to a bellows on the servo. A change in control pressure initiated through the rotating pilot by either speed or speed changer deflects the bellows and servo pilot valve. The servopiston position is proportional to the control pressure.

Figure 29-20 has a spring-return servo that finds application on single-valve turbines with a moderate valve force. For a speed change the rotating pilot sends an error signal through the dashpot to the servopiston. This is a dashpot-type isochronous governor; the error signal sees an instantaneous regulation due to the springs in the dashpot, but after the pressures have equalized through the needle valve, there is no resultant force change on the governor pilot. Control pressure is not proportional to speed; so the governor has zero regulation, also referred to as isochronous control. The setting of the needle valve determines the time required for the system to equalize after a disturbance. This may be several seconds.

Electrohydraulic speed control is in use for turbogenerators and mechanical drive applications because of accurate speed control and easy adaptation to computer operation and such remote control as automatic starting and loading. These same characteristics make it suitable in process plants. Figure 29-21 indicates the elements of electrohydraulic control. A pickup is mounted adjacent to a tooth wheel on the shaft. As the teeth move past the pickup, each tooth generates a small emf pulse. These pulses constitute a digital input to the amplifier. The amplifier performs three functions. The digital input is integrated to a dc level proportional to speed, it is amplified, and the amplified voltage level is matched against a set-point circuit. The differential output current is used to drive an electrohydraulic converter. The converter controls the pilot valve on a standard steam-chest servo. The converter shown puts out a control pressure proportional to current, and the pressure is applied to the bellows. Current electronic governor systems position the servo pilot valve directly.

Remote speed control or **process speed control** takes one of two forms. The remote signal may position the valve directly by acting on the valve stem or on the servo pilot. This action is independent of the governor. For this form of operation the governor is set for maximum operating speed and will take over in an emergency. The operation is referred to as preemergency. In the second type the remote signal acts on the speed changer or its equivalent to adjust the set point. The governor is now always in the circuit, and the unit is always speed-responsive.

The speed control operates the governing valve to maintain steam flow commensurate with load demand while holding speed essentially constant. For sudden load changes there will be a short-time overshoot, and a special case is the instantaneous loss of load, load dump at full load. The usual specification states that the overshoot on load dump must not exceed 9 to 10 percent of rated speed. The settled speed rise will of course be equal to the regulation, 4 or 6 percent for a NEMA Class C or B governor and less than 1 percent for Class D.

Speed governors are classified as shown in Table 29-11.

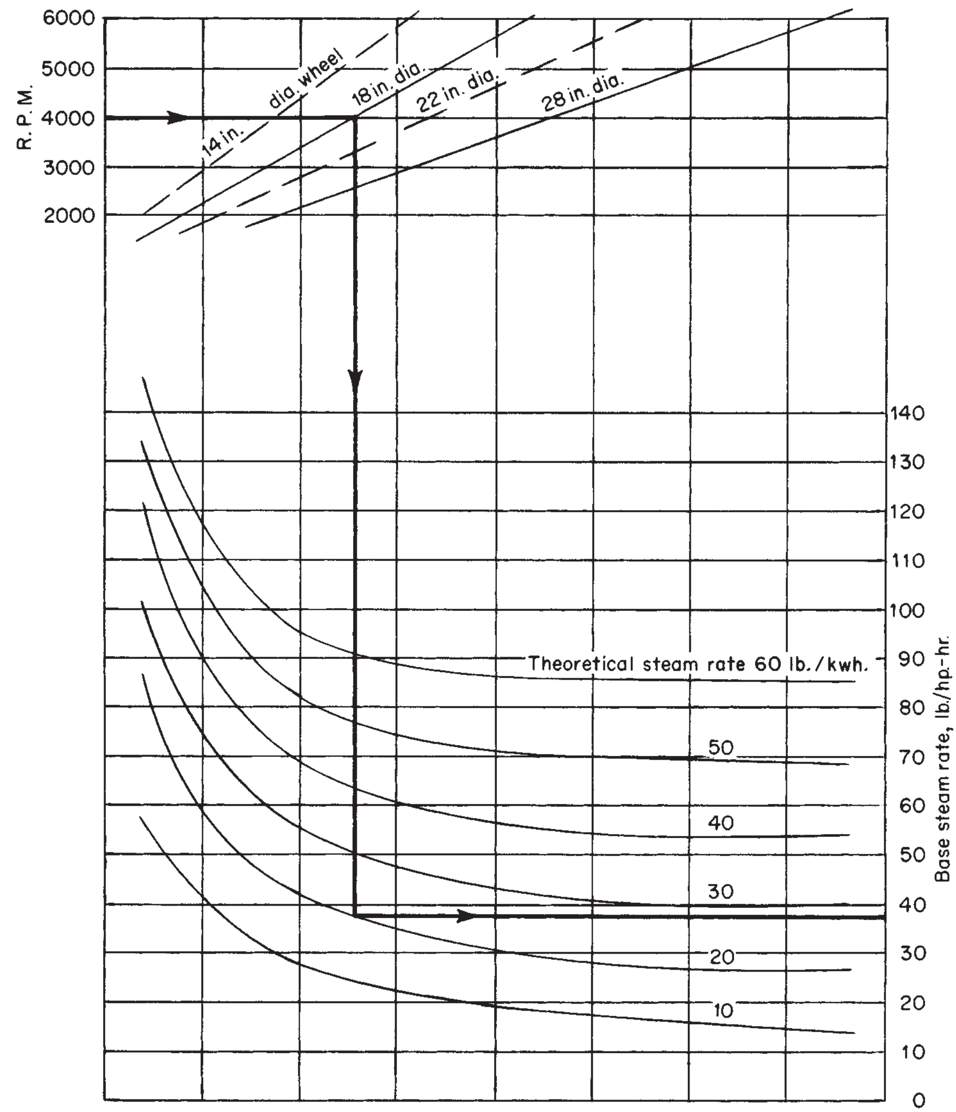


FIG. 29-15 Approximate steam rate for single-stage turbines. To convert pounds per kilowatthour to kilograms per kilowatthour, multiply by 0.4537; to convert inches to meters, multiply by 0.0254; and to convert pounds per horsepower-hour to kilograms per kilowatthour, multiply by 0.6084.

The **trip valve** is provided as a second line of defense in case of overspeed. The trip valve is frequently equipped with a trip-actuating solenoid which can be operated by push button, by low oil pressure, or by some other process upset. When the speed control functions as described above, the trip will not be actuated by load dump.

Extraction-Pressure Control An extraction turbine equipped with a regulator so that the extraction pressure will be automatically controlled is provided with two sets of steam-chest valves as shown in the schematic in Fig. 29-22. Each of the two sets of steam-chest valves is operated by a servomotor. The throttle flow is illustrated as the total of two flow streams: *A*, which travels the length of the turbine and leaves through the exhaust opening; and *B*, which leaves through the extraction opening. The shaft output is the sum of the power generated by the two streams. If the process demand increases, flow *B* increases and develops more power. For constant output, flow *A* must be reduced, and this is the function of the three-arm linkage: to open the governing valves and close the extraction valves, which will increase throttle flow ($A + B$) and decrease condenser flow *A* for more extraction flow at constant load. For a reduction the opposite happens.

For an increase or decrease in load the governor moves the three-arm link parallel to itself, and both sets of valves move in the same direction.

SELECTING A TURBINE

The **major variables** that affect turbine selection are as follows:

1. Horsepower and speed of the driven machine
2. Steam pressure and temperature available or to be decided
3. Steam needed for process, so that a back-pressure turbine should be considered
4. Steam cost and value of turbine efficiency, so that consideration can be given to stage and valve options
5. Use of speed-reducing or speed-increasing gears
6. Extraction for feedwater heating

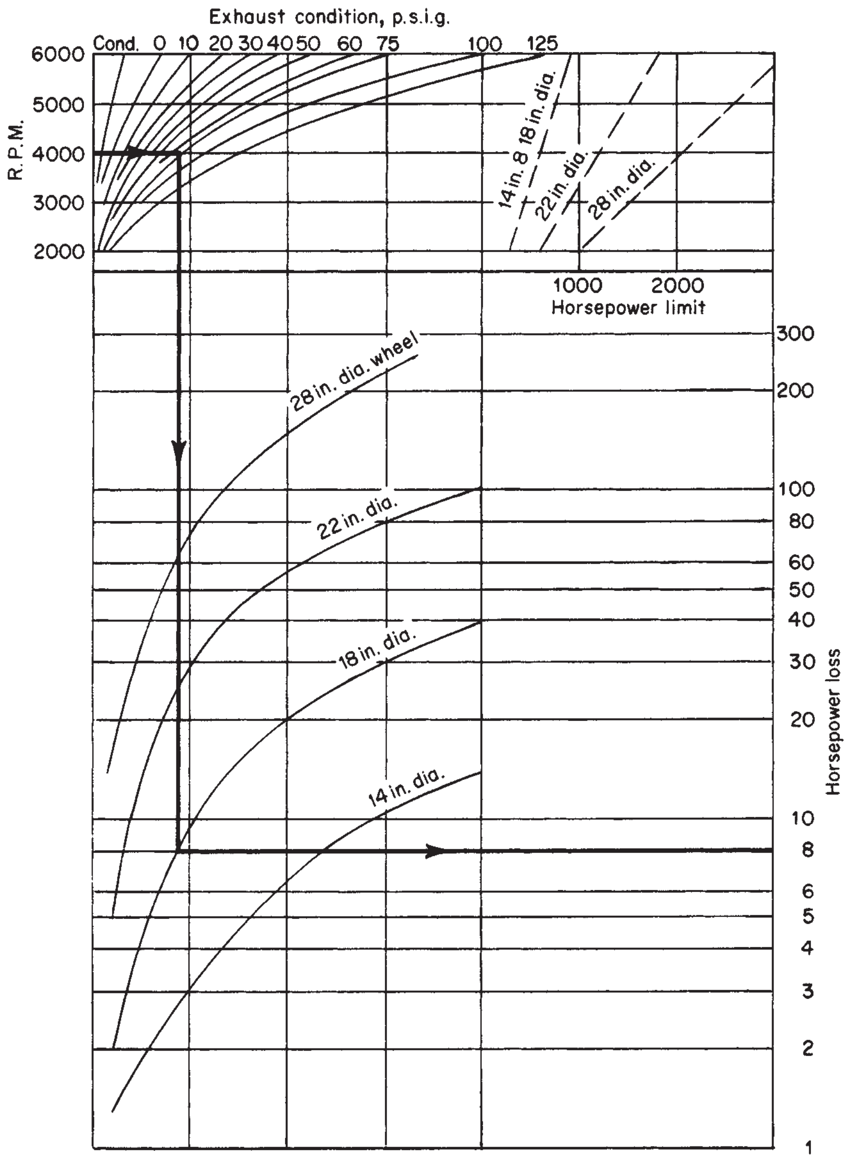


FIG. 29-16 Approximate horsepower loss for single-stage turbines. To convert horsepower to kilowatts, multiply by 0.7457; to convert inches to meters, multiply by 0.0254; and to convert pounds per square inch gauge to kilopascals, multiply by 6.895.

7. Condensing turbine with extraction for process
8. Control system, speed control, pressure control, and process control, so that consideration can be given to remote control, speed or pressure variation that can be tolerated, and system response speed
9. Safety features such as overspeed trip, low-oil trip, remote-solenoid trip, vibration monitor, or other special monitoring of temperature, temperature changes, and casing and rotor expansion
10. Price range from the minimum single-stage turbine to the most efficient multistage turbine

The **initial step in selection** could logically be to make an estimate of the steam flow at various steam pressures by using Fig. 29-17 for a rough estimate of efficiency. Unfortunately, there are no rigid standards for steam pressure and temperature as electrical-voltage steps are fixed, and many engineers pick a pressure and a temperature that look good to them. In general, however, manufacturers prefer working with the standards proposed by a joint ASME-IEEE committee. The values are 2760 kPa (400 psig) and 399° C (750° F), 4.140 kPa (600 psig) and 441° C (825° F), 5 860 kPa (850 psig) and 482° C (900° F), and 8.620 kpa (1250 psig) and 510° C (950° F) or 538° C (1000° F). The values fall on line *A* in Fig. 29-23. For a 1.5-inHg absolute exhaust pressure, line *A* corresponds to 9 percent moisture. Operating to the left of this state line (78 to 80 percent efficiency), last-stage moisture increases rapidly, which means more erosion; also less heat is available. Moving to the right of this line, the temperature lines become quite flat, the pressure drops at constant temperature, and heat available is reduced. It should be noted that 538° C (1000° F) is a good upper limit for steam turbines without a sharp increase in cost because of special materials. Experience has indicated that main-

TABLE 29-10 Typical Steam-Turbine Selection Table

Wheel diameter, in	Base steam rate, lb/hp·hr	Power loss, hp	Total power, hp	Steam required, lb/hr	Steam rate, lb/hp·hr
14	44.5	3.0	303	13,500	44.6
18	37.0	8.5	308.5	11,400	38.1
22	33.0	26.0	326.0	10,750	36.9
28	29.5	64.0	364.0	10,750	36.9

NOTE: To convert pounds per horsepower-hour to kilograms per kilowatthour, multiply by 0.6084; to convert horsepower to kilowatts, multiply by 0.7457.

tenance and initial cost exceed the gain in performance with temperatures in excess of the range of 538 to 566° C (1000 to 1050° F).

Example 1: Selection of Vacuum If a turbine is to be operated with exhaust to a condenser vacuum that will give 3 inHg absolute in the summer and 1 inHg absolute in the winter, what vacuum should be specified?

The turbine for 3 inHg will have shorter exhaust blades and a smaller exhaust opening and will lose 16 Btu/lb when operated at 1 inHg. It will also have a high pressure drop over the last-stage blading. Sonic velocity and shock waves will emanate from the last-stage blading, inducing blade loading and oscillation that may lead to fatigue failure. For operation at 1 inHg the last stage should operate with a diffuser which will limit the annulus discharge area of the stage. A turbine designed for 1 inHg will have a lot of windage in the last stage because the blade annulus cannot be filled by the higher-density steam.

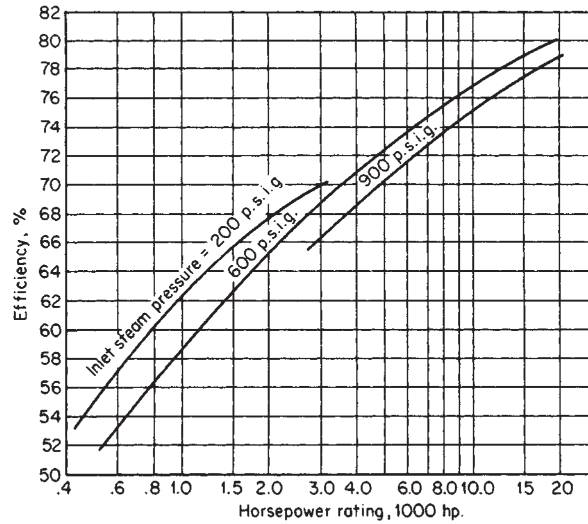


FIG. 29-17 Approximate efficiency for multistage turbines. To convert horsepower to kilowatts, multiply by 0.7457.

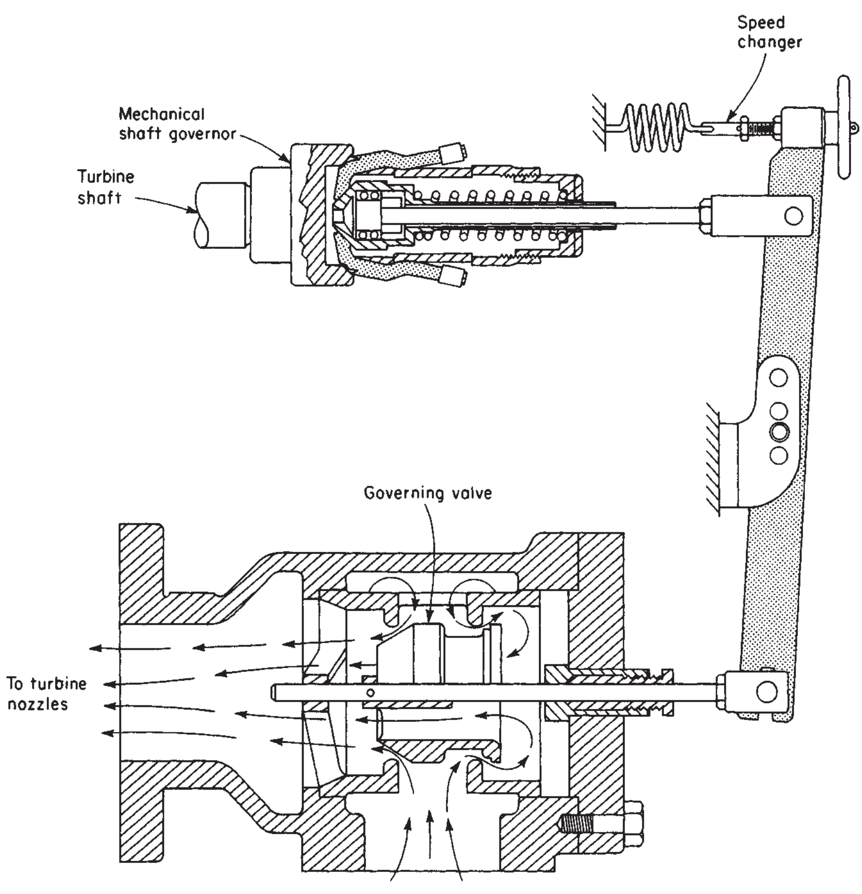


FIG. 29-18 Direct-acting flyball governor.

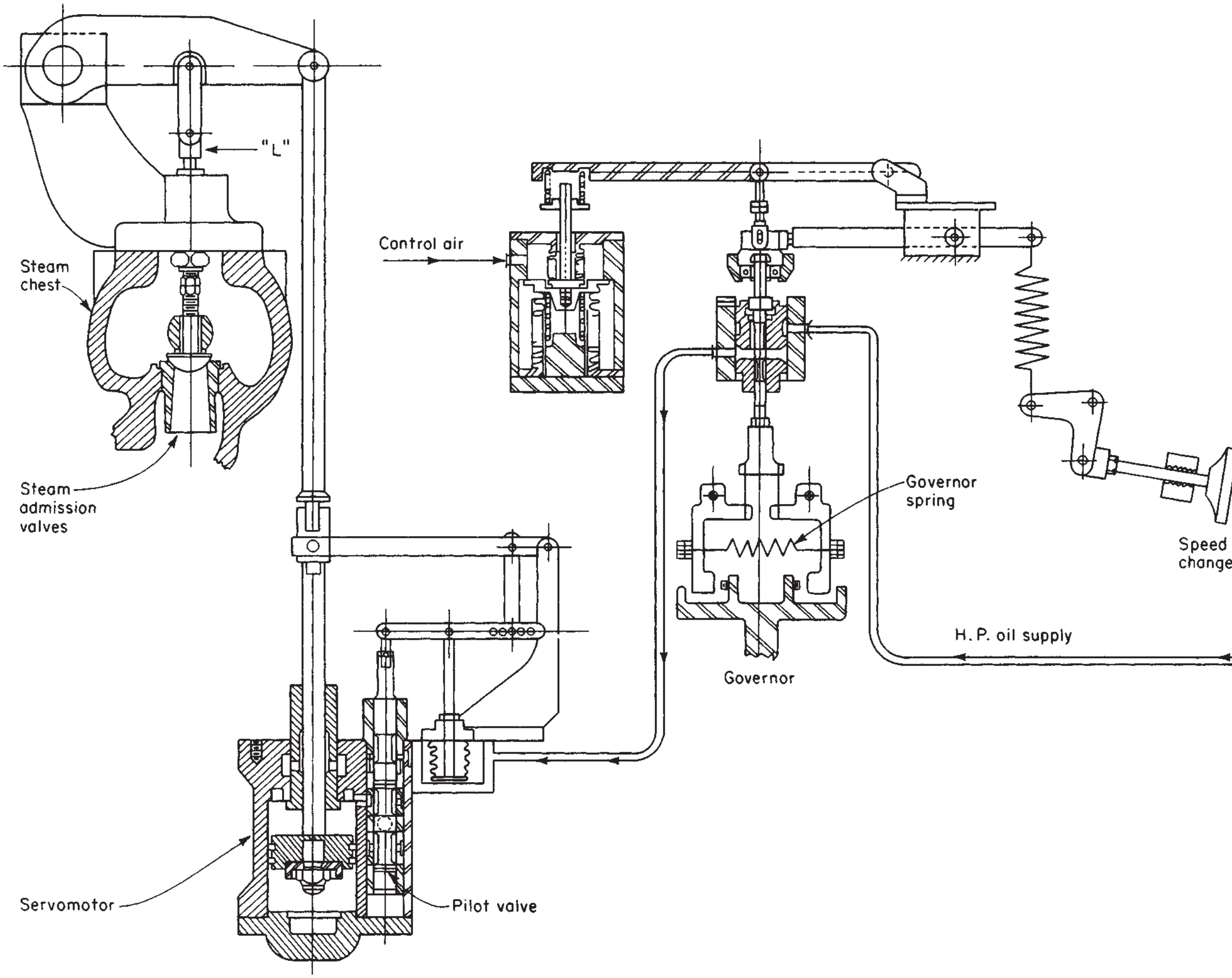


FIG. 29-19 Mechanical-hydraulic speed control: proportional.

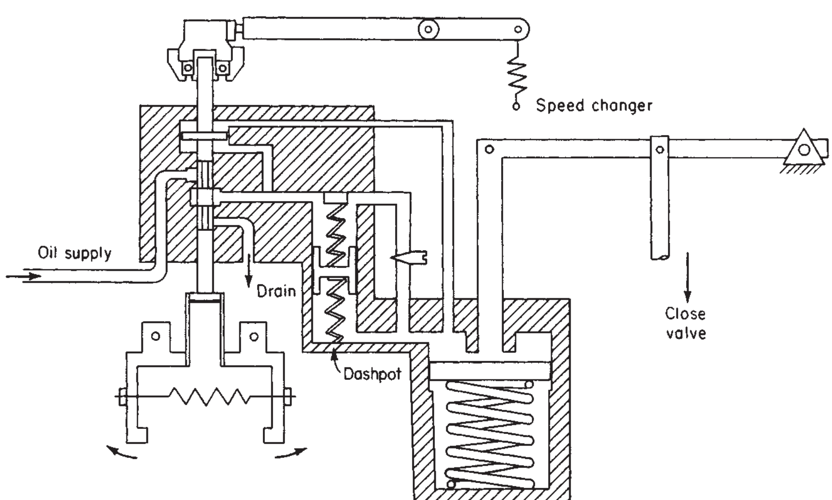


FIG. 29-20 Mechanical-hydraulic speed control: isochronous.

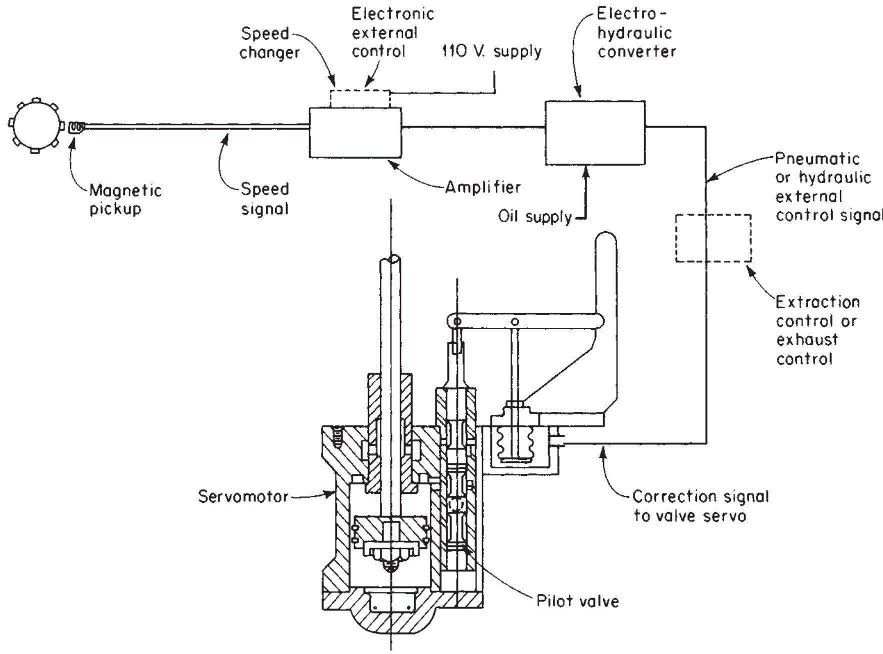


FIG. 29-21 Electrohydraulic speed control.

The best answer is an exhaust designed for 2 inHg and equipped with exhaust-blade diffuser. This will protect at 1 inHg and give some pressure recovery at 3 inHg by virtue of its venturi action.

When an **extraction-condensing turbine** is decided upon, it may be specified in three different ways, depending upon process steam and power demand. Referring to Fig. 29-24, the usual purchase is a unit in which rated capability can be carried either straight-condensing or total-extraction. The zero-extraction line terminates at *A*, and the total-extraction line terminates at *B*.

If the process-steam demand is high and steady, then the exhaust size can be reduced, because not much condensing capacity is required. The choice would be to save cost by a smaller exhaust which would terminate the zero-extraction line at *C*, while the total-extraction line would extend to *B* for rated capability.

TABLE 29-11 Classification of Speed Governors

Class of governor	Adjustable speed range	Maximum steady-state speed regulation	Maximum speed variables, plus or minus	Maximum speed rise	Trip speed (percent above rated speed)
A	10	10	0.75	13	15
	20	10	0.75	13	15
	30	10	0.75	13	15
	50	10	0.75	13	15
	65	10	0.75	13	15
B	10	6	0.50	7	10
	20	6	0.50	7	10
	30	6	0.50	7	10
	50	6	0.50	7	10
	65	6	0.50	7	10
	80	6	0.50	7	10
C	10	4	0.25	7	10
	20	4	0.25	7	10
	30	4	0.25	7	10
	50	4	0.25	7	10
	65	4	0.25	7	10
	80	4	0.25	7	10
D	30	0.50	0.25	7	10
	50	0.50	0.25	7	10
	65	0.50	0.25	7	10
	80	0.50	0.25	7	10
	85	0.50	0.25	7	10
	90	0.50	0.25	7	10

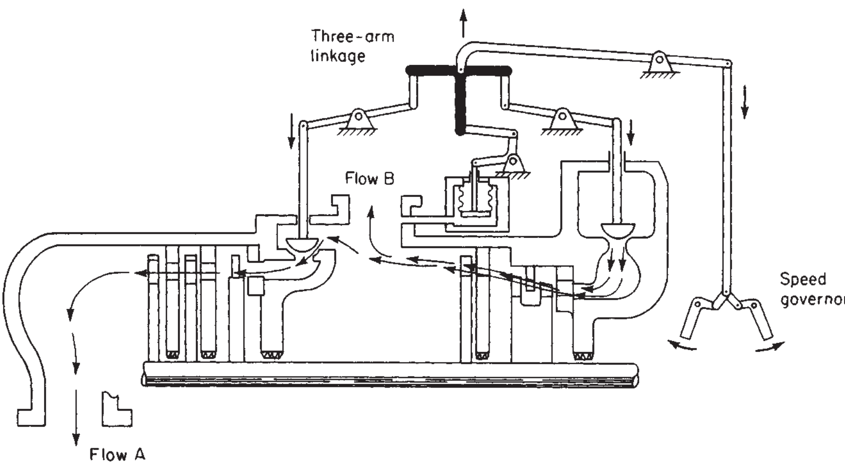


FIG. 29-22 Three-arm lever/mechanism for extraction-turbine-pressure control.

If the process demand is light and high-extraction flow is not required, then most of the power will be from the condensing flow. The choice would be to save cost by a smaller inlet and steam chest, which would terminate the total-extraction line at *D* and the zero-extraction line at *A*.

TEST AND MECHANICAL PERFORMANCE

When testing to establish the **thermodynamic performance** of a steam turbine, the ASME Performance Test Code 6 should be followed as closely as possible. The effect of deviations from code procedure should be carefully evaluated. The flow measurement is particularly critical, and Performance Test Code 19 gives details of flow nozzles and orifices. The test requirements should be carefully studied when the piping is designed to ensure that a meaningful test can be conducted.

Mechanical performance is generally checked by a running test at the factory before shipment and again when the turbine is installed on the site. The following is an enumeration of items that may provide smooth and vibration-free operation. The rotor must be in dynamic balance, and at-speed balancing is the most effective. The bearings must be in line, and the seal clearance must be correct. This alignment must be maintained when both cold and hot and during the transient from cold to hot. Disturbance of alignment may originate by unequal heat expansion of supports, from pipe expansion, or from binding and lack of freedom to expand. Excessive pipe expansion and high forces on the turbine have caused many vibration problems. Flexible couplings that do not flex also cause problems. This may result in torque lock or in unloading of bearings, and driver and driven machine bearings may be shifted out of line. Then there is the question of resonance and critical speed. The foundation enters into this relationship, and slender columns may contribute problems of resonance. If there is a gear in the lineup, it should be checked carefully for bearing load, alignment, and resonance frequencies that may be multiples of the gear ratio. Also, bearing-oil supply must be adequate.

From this it can be seen that **vibration** is the universal manifestation that something is wrong. Therefore, many units are equipped with instruments that continuously monitor vibration. Numerous new instruments for vibration analysis have become available. Frequency can be accurately determined and compared with computations, and by means of oscilloscopes the waveform and its harmonic components can be analyzed. Such equipment is a great help in diagnosing a source of trouble.

OPERATING PROBLEMS

While most turbines have a 10-year-availability record in the range of 95 to 99 percent, troubles may develop in any number of places. The most common are vibration, cycling governor, sticking valve stems, leaky packing, temperature bow, erosion of blading, loss of power, and bearing problems.

The causes of **vibration** have already been discussed. An increase in vibration over a period of time is generally caused by loss of alignment, settling of the foundation, or sticking of some expansion feature such as a pipe or a pedestal. Other causes are wear in the teeth of a flexible coupling, an internal rub in the unit, loss of bearing oil, and bearing wear. (Startup vibration is discussed later under temperature bow.)

If a **governor** starts to **cycle** after operating for some time, this is generally the result of wear which causes dead band or sticking. Also, all pilot valves should be inspected for the effects of dirt in the oil.

Sticking of valve stems is common if solids are present in the steam. The steam must be without solids. (Note comments later under loss of power.) It is important that units operating on a steady load for long periods be checked for sticking stems at regular intervals. The records show that in several cases deposits have caused the stem of both the governor valve and the trip valve to stick when there was a loss of load. The effect of the loss of load was destructive overspeed.

Wear and increased **leakage from the glands** are common. Carbon rings may need replacement after 1 or 2 years of operation. A unit with labyrinth packing may never need packing replacement. This depends upon operation. If a unit is started quickly with a temperature bow in the shaft, the result is a rub in the labyrinth, and then all packing may need replacement.

It is important to understand the reasons for a **temperature bow.** When a turbine is shut down and starts to cool, the lower half, particularly on a condensing unit, will cool faster than the top half. After the rotor has stopped turning, the temperature difference increases, and in 20 min there may be a 28 to 83° C (50 to 150° F) difference between top and bottom. Both the casing and the shaft bow up because of this temperature difference in the vertical plane. If the throttle is opened now and the bowed shaft starts to turn in the bowed casing, the packing may wipe out in a few revolutions, and at 200 r/min and up there will be a heavy thumping. The packing rub serves to increase the temperature bow by heating the high side of the shaft.

Other causes of a temperature bow are leaky valves and damaged sleeves. If steam is leaking into a stopped turbine from either an exhaust valve or a stop valve that is leaking, the upper half will be

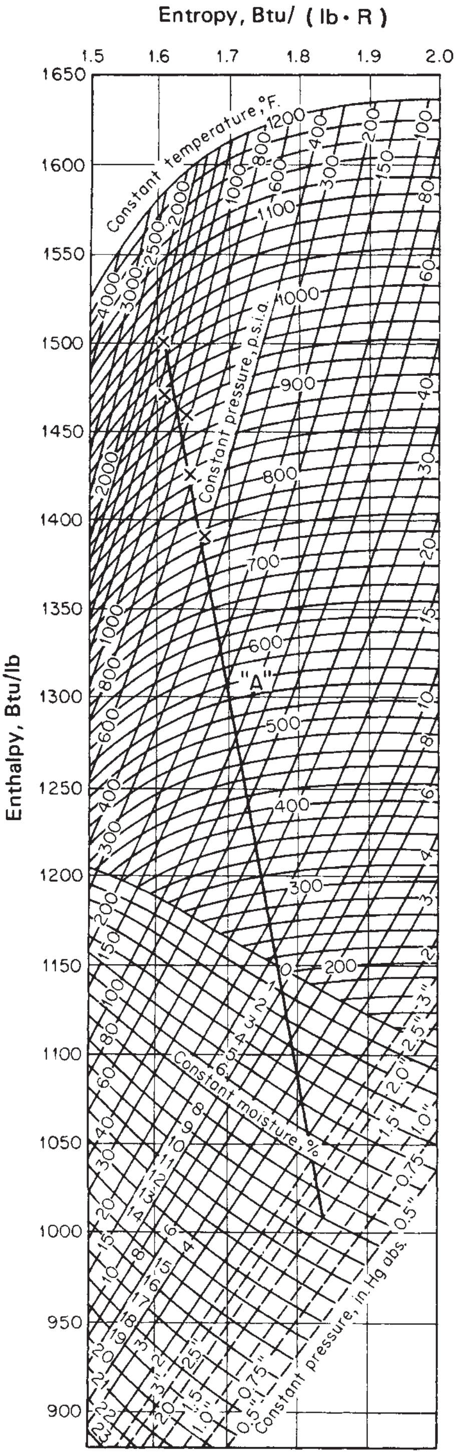


FIG. 29-23 Mollier diagram showing ASME-IEEE steam-turbine standards. To convert British thermal units per pound to kilojoules per kilogram, multiply by 2.328; to convert British thermal units per pound-degrees Rankine to joules per gram-Kelvin, multiply by 4.19; and to convert pounds-force per square inch to kilopascals, multiply by 6.89.

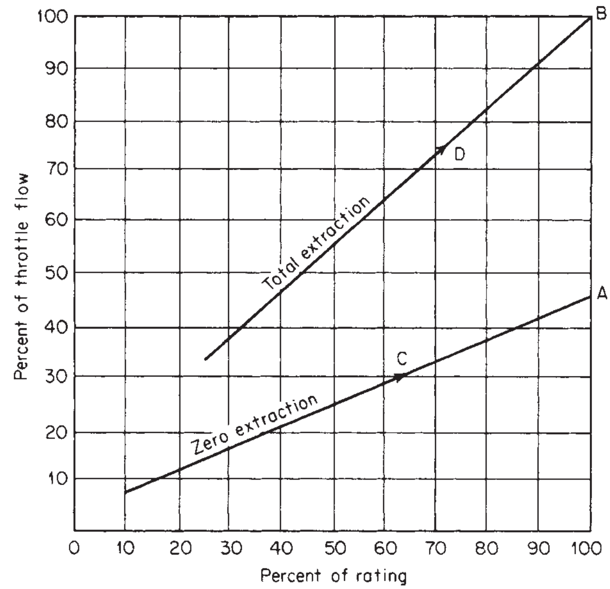


FIG. 29-24 Characteristic of extraction-condensing steam turbines.

warmer than the lower half, and it bows. If a shaft sleeve does not contact uniformly, there will be a transient difference in heat transfer to the shaft, and a bow will result. Also, turning sealing steam on before the shaft rotates may cause a bow.

Leaky valves are also a cause of **erosion.** Most turbine erosion-corrosion problems come from damage that takes place when the unit is not running. A slight steam leak into the turbine will let the steam condense inside the turbine, and salt from the boiler water will settle on the inside surfaces and cause pitting, even of the stainless blading. There must be two valves with a drain between them, i.e., a block valve on the header and an open drain in the line before it reaches the closed trip-throttle valve.

In a turbine that is running, erosion-corrosion is pretty much confined to units that are operating on saturated steam with inadequate boiler-water treatment. This type of erosion takes place behind the nozzle ring and around the diaphragms where they fit in the casing.

Loss of power is another item generally tied to water treatment. With dissolved salts in the steam these salts stay in solution while the steam is superheated. After the steam has expanded through several stages and become saturated, the salts condense out with the moisture. Silica and other salt deposits build up on the blading and the nozzles. The stage pressures increase, and the load drops. The thrust load increases, and the thrust bearing may fail. Depending upon the nature of the salts, it is possible to have corrosion associated with the deposits or corrosion only in the region where the steam changes from superheated to steam with moisture.

Turbine bearings show very little sign of wear as long as there is an adequate oil film. Wiping of the bearing is generally traced to dirt in the oil or a restriction in the oil supply. Thus filtration should be adequate and retain particles that may exceed the oil-film thickness. A checkerboard cracking of the babbitt is sometimes observed. This may have either of two origins. The shaft transports a lot of heat from the steam parts into the journal, and the oil flow in the bearing must be sufficient to lubricate and to remove this heat. Otherwise, the heat is conducted into the babbitt and the babbitt will soften and crack. This cracking may take place if oil flow is stopped too soon when a unit is shut down. The second origin for cracking of the bearing surface is pounding of the journal caused by shaft bow or oil whirl.

GAS TURBINES

The gas turbine is a power plant that produces a great amount of energy for its size and weight. The gas turbine has found increasing service in the past 15 years in the petrochemical industry and utilities throughout the world. It is the power source of the aircraft industry. In this section we deal with land-based gas turbines.

The gas turbine in a combined cycle mode will be the power source through the year 2020 for most countries throughout the world. Most of the new power plants of the 1990s are and will continue to be run by gas turbines with steam turbines in a combined cycle mode. Its compactness, low weight, and multiple fuel application also make it a natural power plant for offshore platforms. Today there are gas turbines that run on natural gas, diesel fuel, naphtha, methane, crude, low-Btu gases, vaporized fuel oils, and even waste. In the 1950s and 1960s, the gas turbine was perceived as a relatively inefficient power source when compared to other power sources. Its efficiencies, compactness, and light weight made it attractive for certain applications. The limiting factor for most gas turbines has been the turbine inlet temperature. With new schemes of air cooling and breakthroughs in blade metallurgy, higher turbine temperatures have been achieved.

The new gas turbines have fired inlet temperatures as high as 2300° F (1260° C) with efficiencies as high as 42–45 percent. Pressure ratios have increased from 5:1 in the 1950s to as high as 30:1 in some of the new turbines of the 1990s. Gas turbines are classified into two major categories:

1. Industrial heavy-duty gas turbines
2. Aeroderivative gas turbines

INDUSTRIAL HEAVY-DUTY GAS TURBINES

The gas turbine was designed shortly after World War II and introduced to the market in the early 1950s. The early heavy-duty gas turbine design was largely an extension of steam turbine design. Restrictions of weight and space were not important factors for these ground-based units, so the design characteristics included heavy-wall casings split on horizontal centerlines, hydrodynamic (tilting pad) bearings, large-diameter combustors, thick airfoil sections for blades and stators, and large frontal areas. The overall pressure ratio of these units varied from 5:1 for the earlier units to 30:1 for the units in the 1990s. Turbine inlet temperatures have been increased and run as high as 2300° F (1260° C) on some of these units. Projected temperatures approach 3000° F (1649° C) and, if achieved, would make the gas turbine even more efficient. The industrial heavy-duty gas turbines most widely used employ axial-flow compressors and turbines. In most U.S. designs combustors are can-annular combustors. Single-stage side combustors are used in European designs. The combustors used in industrial gas turbines have heavy walls and are very durable.

AERODERIVATIVE GAS TURBINES

The aeroderivative gas turbine, as its name implies, is composed of an aeroengine that produces high temperature and high pressure gas that is then put through a power turbine to produce the energy required. The aeroengine is the gas generator. It is usually an engine developed for an aircraft that is modified by the addition of compression stages to produce the high pressure and high temperature required. The gas generator serves to raise combustion gas products for the power turbine to a pressure of about 45–75 psig (3–5 Bar) and temperatures between 900° F (482° C) and 1200° F (649° C). The gas generator is very lightweight compared to the industrial gas turbine. The gas generator is characterized by light casing walls, blades with high aspect ratio (blade length/blade chord), roller bearings, annular combustors, and light weight.

The power turbine is free (i.e., the power turbine is not physically coupled to the gas generator but is closely coupled to the gas generator by a transition duct that transports the gas from the gas generator to the power turbine). The power turbine is an industrial-type turbine in design characterized by heavy wall casings, hydrodynamic (tilting pad) bearings, and thick airfoil sections. The rotative power produced in the gas turbine is then available for mechanical coupling to the driven equipment.

MAJOR GAS TURBINE COMPONENTS

The gas turbine in the simple cycle mode consists of a compressor (axial or centrifugal) that compresses the air, a combustor that heats the air at constant pressure and a turbine that expands the high pressure and high temperature combustion gases and produces power to run the compressor and through a mechanical coupling to the driven equipment. The power required to compress the gases varies from about 40–60 percent of the total power produced by the turbine.

Compressors A compressor is a device that pressurizes the air in a gas turbine. It transfers energy by dynamic means from a rotating member to the continuously flowing air. The two types of compressors used in gas turbines are axial and centrifugal. Figures 29-25 and 29-26 depict typical industrial gas turbines.

The axial flow compressor is used in over 95 percent of the gas turbines. An axial-flow compressor in a gas turbine compresses the working air by first accelerating the air and then diffusing it to obtain a pressure increase. The air is accelerated by a row of rotating airfoils or blades (the rotor) and diffused by a row of stationary blades (the stator). The diffusion in the stator converts the velocity increase gained in the rotor to a pressure increase. One rotor and one stator make up a stage in a compressor. A compressor usually consists of several stages: a typical compressor contains about 15–17 stages. One additional row of fixed blades (inlet guide vanes) is frequently used at the compressor inlet to ensure that air enters the first-stage blading at the desired angle. In addition to the stators, an additional diffuser at the exit of the compressor further diffuses the air and controls its velocity when entering the combustors.

In an axial compressor air passes from one stage to the next with each stage raising the pressure slightly. By producing low pressure increases on the order of 1.1:1 to 1.4:1, very high efficiencies (85–90 percent) can be obtained. The use of multiple stages permits overall pressure increases up to 30:1. The axial-flow compressor has a very narrow operating range. The operating range is defined as the range between surge and choke. *Surge* is when the flow in a compressor reverses itself; this phenomenon is very destructive. *Choke,* sometimes called "stone wall," is the point where the compressor flow has reached a maximum; this is accomplished by great loss in efficiency.

In the centrifugal or mixed-flow compressor, the air enters the compressor in an axial direction and exits in a radial direction into a diffuser. This combination of rotor (or impeller) and diffuser comprises a single stage. The air initially enters a centrifugal compressor at the inducer. The inducer, usually an integral part of the impeller, is very much like an axial-flow compressor rotor. Many European designs keep the inducer separate. The air then goes through a 90° turn and exits into a diffuser, which usually consists of a vaneless space followed by a vaned diffuser.

From the exit of the diffuser, the air enters a scroll or collector. The pressure ratio per stage in a centrifugal compressor can vary from about 1.5:1 to 9:1 on production units. Some experimental units have obtained pressure ratios of more than 12:1 for a single stage. The centrifugal compressor is slightly less efficient (78–83 percent) than the axial-flow compressor but has a higher stability. A higher stability means that its operating range (surge-to-choke margin) is greater; however, like the axial-flow unit, this range is reduced as the pressure ratio is increased.

Regenerators Regenerators are used in gas turbines to increase the turbine efficiency. They are placed between the compressor and the combustor. Heavy-duty regenerators are designed for applications in large gas turbines in the 5000–100,000-hp range. The use of regenerators in conjunction with industrial gas turbines substantially increases cycle efficiency and provides an impetus to energy management by reducing fuel consumption up to 30 percent. In most present-day regenerative gas turbines, ambient air enters the inlet fil-

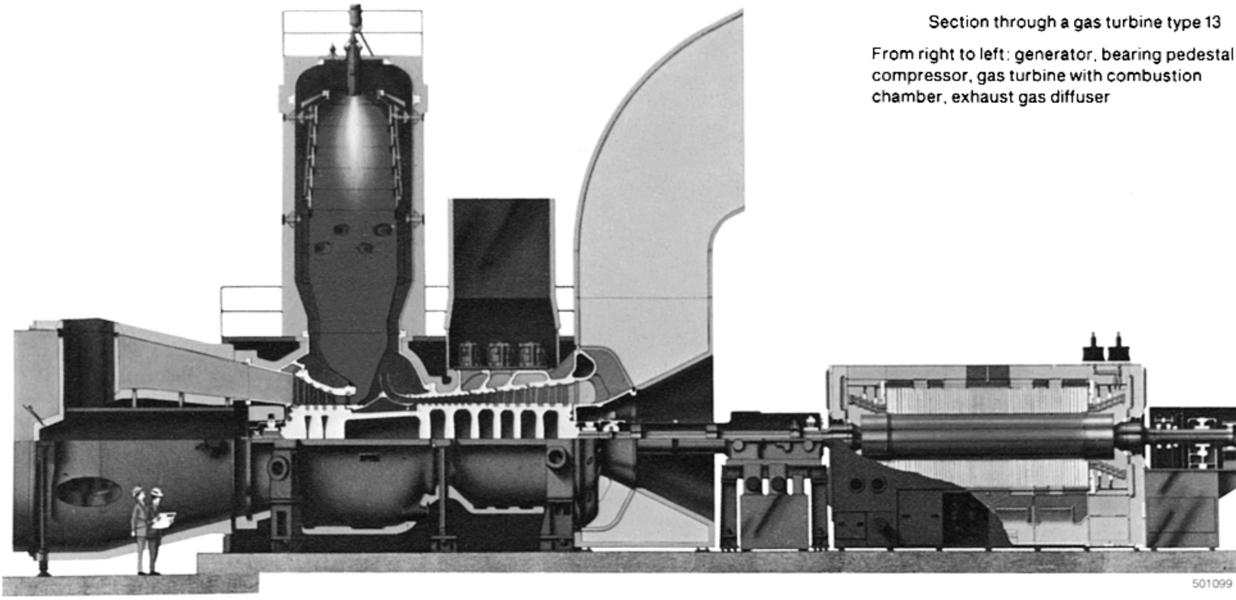
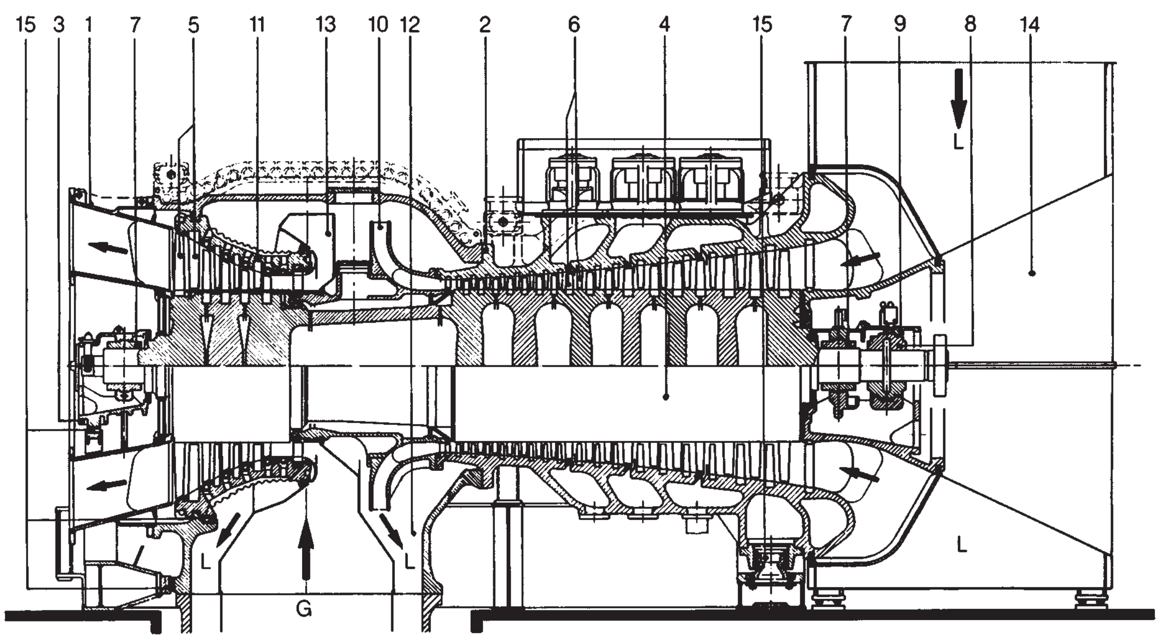


FIG. 29-25 Section through a Brown-Boveri gas turbine (with permission of Asea-Brown Boveri).



1 = Turbine housing
2 = Compressor housing
3 = Bearing body
4 = Shaft
5 = Turbine rotor and stator blades
6 = Compressor rotor and stator blades
7 = Journal bearings
8 = Thrust bearings
9 = Thrust bearing cover
10 = Diffuser
11 = Blade carrier
12 = Cooling-air admission
13 = Internal casing
14 = Air intake
15 = Position key
L = Air
G = Hot gas

FIG. 29-26 Principal components of a Brown-Boveri gas turbine (with permission of Asea-Brown Boveri).

ter and is compressed. The air is then piped to the regenerator, which heats the air to about 900° F (482° C). The heated air then enters the combustor, where it is further heated before entering the turbine. After the gas has undergone expansion in the turbine, it is about 1000° F (538° C)–1100° F (593° C) and essentially at ambient pressure. The gas is ducted through the regenerator where the waste heat is transferred to the incoming air. The gas is then discharged into the ambient air through the exhaust stack. In effect, the heat that would otherwise be lost is transferred to the air, decreasing the amount of fuel that must be consumed to operate the turbine. For a 30,000-hp turbine, the regenerator heats 10 million pounds (453,000 kg) of air per day.

Combustors All gas turbine combustors perform the same function: They increase the temperature of the high-pressure gas at constant pressure. The gas turbine combustor uses very little of its air (10 percent) in the combustion process. The rest of the air is used for cooling and mixing. The air from the compressor must be diffused before it enters the combustor. The velocity leaving the compressor is about 400–500 ft/sec (130–164 m/sec), and the velocity in the combustor must be maintained at about 10–30 ft/sec (3–10 m/sec). Even at these low velocities, care must be taken to avoid the flame to be carried downstream. To ensure this, a baffle creates an eddy region that stabilizes the flame and produces continuous ignition. The loss of pressure in a combustor is a major problem, since it affects both the fuel consumption and power output. Total pressure loss is in the range of 2–8 percent; this loss is the same as the decrease in compressor efficiency.

Despite the many design differences, all gas turbine combustion chambers have three features: (1) a recirculation zone, (2) a burning zone with a recirculation zone that extends to the dilution region, and (3) a dilution zone. The function of the recirculation zone is to evaporate, burn in part, and prepare the fuel for rapid combustion within the remainder of the burning zone. Ideally, at the end of the burning zone, all fuel should be burnt so that the function of the dilution zone is solely to mix the hot gas with the dilution air. The mixture leaving the chamber should have a temperature and velocity distribution acceptable to the turbine nozzles. Generally, the addition of dilution air is so abrupt that if combustion is not complete at the end of the burning zone, chilling occurs, which prevents completion. However, there is evidence with some chambers that if the burning zone is run overrich, some combustion does occur within the dilution region. Combustor inlet temperature depends on engine pressure ratio, load and engine type, and whether the turbine is regenerative or nonregenerative. Nonregenerative inlet temperatures vary from 250° F (121° C) to 1000° F (593° C), while regenerative inlet temperatures range from 700° F (371° C) to 1200° F (649° C). Combustor outlet temperatures range from 1500° (815° C) to 3000° F (1649° C) for large turbines. Combustor pressures for a full-load operation vary from 45 psia (310 kPa) for small engines to as much as 450 psia (3100 kPa) in complex engines. Fuel rates vary with load, and fuel atomizers may be required for flow ranges as great as 100:1. However, the variation in the fuel-to-air ratio between idle and full-load conditions usually does not vary. At lightoff and during acceleration, a much higher fuel-to-air ratio is needed because of the higher temperature rise. On deceleration, the conditions may be appreciably leaner. Thus, a combustor that can operate over a wide range of mixtures without danger of blowouts simplifies the control system.

Combustor performance is measured by efficiency, the pressure decrease encountered in the combustor, and the evenness of the outlet temperature profile. Combustion efficiency is a measure of combustion completeness. Combustion completeness affects fuel consumption directly, since the heating value of any unburned fuel is not used to increase the turbine inlet temperature. The uniformity of the combustor outlet profile affects the useful level of turbine inlet temperature, since the average gas temperature is limited by the peak gas temperature. This uniformity assures adequate nozzle life, which depends on operating temperature. The average inlet temperature to the turbine affects both fuel consumption and power output. A large combustor outlet gradient will work to reduce average gas temperature and consequently reduce power output and efficiency. Combustors in a gas turbine can be arranged in many different ways. These arrangements can be classified into three main categories:

1. Tubular (side combustors)
2. Can-annular combustors
3. Annular combustors

Tubular (Side Combustors) Tubular or single-can designs are preferred by many European industrial gas turbine designers. These large single combustors offer the advantage of simplicity of design and long life because of low heat-release rates. These combustors are sometimes very large. They can range in size from small units of about 6 inches (152 mm) in diameter to 1-foot (300-mm) combustors that are over 10 feet (3 m) in diameter and 30–40 feet (10–13 m) high. These large combustors use special tiles as liners. Any liner damage can be easily corrected by replacing the damaged tiles. The tubular combustors can be designed as "straight-through" or "reverse-flow" designs. Most large single-can combustors are of the reverse-flow design. In this design, the air enters the turbine through the annulus between the combustor can and the hot gas pipe. The air then passes between the liner and the combustor can and enters the combustion region at various points of entry. About 10 percent of the air enters the combustion zone, about 30–40 percent of the air is used for cooling purposes, and the rest is used in the dilution zone. Reverse-flow designs reduce the combustor lengths as compared to the straight-through flow designs.

Larger tubular, or single-can, units usually have more than one nozzle. In many cases a ring of nozzles is placed in the primary zone area. The radial and circumferential distribution of the temperature to the turbine nozzles is not as even as in tubo-annular combustors. In some cases, high stresses are exerted on the turbine casing leading to casing cracks.

Can-Annular Combustors Can-annular combustors are the most common type of combustors used in gas turbines. The industrial gas turbines designed by U.S. companies use the tubo-annular or can-annular type. The advantage to these types of combustors is the ease of maintenance. They also have a better temperature distribution than the side single-can combustor and can be of the straight-through or reverse-flow design. As with the single-can combustor, most of these combustors are of the reverse-flow design in industrial turbines.

In most aircraft engines, the can-annular combustors are of the straight-through flow type. The straight-through flow-type tubo-annular combustor requires a much smaller frontal area than the reverse-flow-type can-annular combustor. The can-annular combustor also requires more cooling air flow than a single or annular combustor because the surface area of the can-annular combustor is much greater. The amount of cooling air is not much of a problem in turbines using high-Btu gas, but for low-Btu gas turbines, the amount of air required in the primary zone is increased from 10 percent to as high as 35 percent of the total air, thus reducing the amount of air available for cooling purposes.

Higher temperatures also require more cooling and, as temperatures increase, the single can or annular combustor design becomes more attractive. The tubo-annular combustor has a more even combustion because each can has its own nozzle and a smaller combustion zone, resulting in a much more even flow. Development of a can-annular combustor is usually less expensive, since only one needs to be tested instead of an entire unit as in an annular or single-can combustor. Therefore, the fuel and air requirements can be as low as 8–10% of the total requirements.

Annular Combustors Annular combustors are used mainly in aircraft-type gas turbines where frontal area is important. There are smaller sized industrial turbines that also have annular combustors. This type of combustor is usually a straight-through flow type. The combustor outside radius is the same as the compressor casing, thus producing a streamline design. The annular combustor mentioned earlier requires less cooling air than the tubo-annular combustor, so it is growing in importance for high-temperature applications. On the other hand, the annular combustor is much harder to get to for maintenance and tends to produce a less favorable radial and circumferential profile as compared to the can-annular combustors. The annular combustors are also used in some newer industrial gas turbine applications. The higher temperatures and low-Btu gases will foster more use of annular-type combustors in the future.

Turbines The two types of turbine geometries used in gas turbines are the axial-flow and the radial-inflow type. The axial-flow turbine is used in more than 95 percent of all applications in a gas turbine.

Radial-Inflow Turbine The radial-inflow turbine, or inward-flow radial turbine, has been in use for many years. Basically a centrifugal compressor with reversed-flow and opposite rotation, the inward-flow radial turbine is used for smaller loads and over a smaller operational range than the axial turbine. Radial-inflow turbines are only now beginning to be used because little was know about them heretofore. Axial turbines have enjoyed tremendous interest due to their low frontal area, making them suited to the aircraft industry. However, the axial machine is much longer than the radial machine, making it unsuited for certain vehicular and helicopter applications. Radial turbines are used in turbochargers and in some types of expanders.

The inward-flow radial turbine has many components similar to a centrifugal compressor. The mixed-flow turbine is almost identical to a centrifugal compressor—except its components have different functions. The scroll is used to distribute the gas uniformly around the periphery of the turbine. The nozzles, used to accelerate the flow toward the impeller tip, are usually straight vanes with no airfoil design. The vortex is a vaneless space and allows an equalization of the pressures. The flow enters the rotor radially at the tip with no appreciable axial velocity and exits the rotor through the exducer axially with little radial velocity. These turbines are used because of lower production costs, in part because the nozzle blading does not require any camber or airfoil design. They are also more robust, but due to cooling restrictions are used for much lower turbine inlet temperatures.

Axial-Flow Turbine The axial-flow turbine is very widely used in gas turbines (95 percent). These axial flow turbines, like their counterparts, the axial-flow compressors, have flow that enters and leaves in an axial direction. Axial-flow turbines are the most widely employed turbines using a compressible fluid. Axial-flow turbines power most gas turbine units—except the smaller horsepower turbines—and they are more efficient than radial-inflow turbines in most operational ranges. Axial-flow turbine efficiencies range from 88–92 percent. The axial-flow turbine is also used in steam turbine design; however, there are some significant differences between the axial-flow turbine design for a gas turbine and the design for a steam turbine. Steam turbine development preceded the gas turbine by many years. Thus, the axial-flow turbine used in gas turbines is an outgrowth of steam turbine technology. In recent years, the trend towards high turbine inlet temperatures in gas turbines has required various cooling schemes and improved materials.

There are two types of axial turbines:

1. Impulse type
2. Reaction type

An impulse-type turbine experiences its entire enthalphy drop in the nozzle, thus having a very high velocity entering the rotor. The velocity entering the rotor is about twice the velocity of the wheel. The reaction type turbine divides the enthalphy drop in the nozzle and in the rotor. Thus, for example, a 50 percent reaction turbine has a velocity leaving the nozzle equal to the wheel speed and produces about ½ the work of a similar size impulse turbine at about 2–3 percentage points higher efficiency than the impulse turbine (0 percent reaction turbine). The effect on the efficiency and ratio of the wheel speed to inlet velocity is shown in Fig. 29-27 for an impulse turbine and 50 percent reaction turbine.

Impulse Turbine The impulse turbine is the simplest type of turbine. It consists of a group of nozzles followed by a row of blades. The gas is expanded in the nozzle, converting the high thermal energy into kinetic energy. This conversion can be represented by the following relationship:

$$V = \sqrt{2\Delta h} \tag{29-15}$$

The high-velocity gas impinges on the blade where a large portion of the kinetic energy of the moving gas stream is converted into turbine shaft work. Figure 29-28 shows a diagram of a single-stage impulse turbine. The static pressure decreases in the nozzle with a corresponding increase in the absolute velocity. The absolute velocity is then reduced in the rotor, but the static pressure and the relative velocity remain constant. To get the maximum energy transfer, the blades must rotate at about one-half the velocity of the gas jet velocity. By definition, the impulse turbine has a degree of reaction equal to zero. This degree of reaction means that the entire enthalphy drop is taken in the nozzle, and the exit velocity from the nozzle is very high. Since there is no change in enthalphy in the rotor, the relative velocity entering the rotor equals the relative velocity exiting from the rotor blade. For the maximum utilization, the absolute exit velocity must be axial.

The Reaction Turbine The axial-flow reaction turbine is the most widely used turbine. In a reaction turbine, both the nozzles and blades act as expanding nozzles. Therefore, the static pressure decreases in both the fixed and moving blades. The fixed blades act as nozzles and direct the flow to the moving blades at a velocity slightly higher than the moving-blade velocity. In the reaction turbine, the velocities are usually much lower, and the entering blade relative velocities are nearly axial. Figure 29-29 shows a schematic view of a reaction turbine. In most designs, the reaction of the turbine blade varies from hub to shroud. The impulse turbine is a reaction turbine with a reaction of zero ($R = 0$). The utilization factor that is a ratio of the ideal work to the energy supplied for a fixed nozzle angle will increase as the reaction approaches 100 percent. For $R = 1$, the utilization factor does not reach

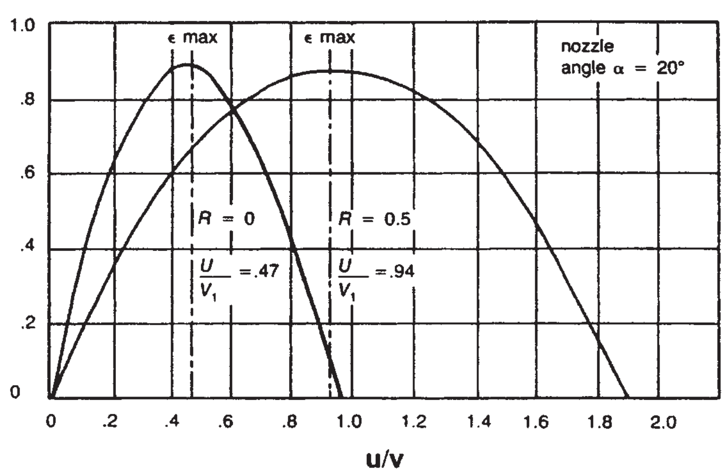


FIG. 29-27 Variation of utilization factor with U/V_1 for $R = 0$ and $R = 0.5$. (*From* Principles of Turbomachinery *by Dennis G. Shepherd, Copyright 1956 by Macmillan Publishing Co., Inc.*)

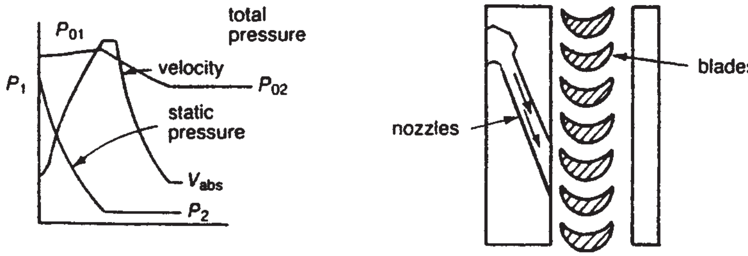


FIG. 29-28 View of a single-stage impulse turbine with velocity and pressure distribution.

unity but reaches some maximum finite value. The 100 percent reaction turbine is not practical because of the high rotor speed necessary for a good utilization factor. For reaction less than zero, the rotor has a diffusing action. Diffusing action in the rotor is undesirable, since it leads to flow losses. The 50 percent reaction turbine has been used widely and has special significance. The velocity diagram for a 50 percent reaction is symmetrical and, for the maximum utilization factor, the exit velocity must be axial. The pressure and velocity distributions in a reaction-type turbine are also shown in Fig. 29-29.

The work produced in an impulse turbine with a single stage running at the same blade speed is twice that of a reaction turbine. Hence, the cost of a reaction turbine for the same amount of work is much higher, since it requires more stages. It is a common practice to design multistage turbines with impulse stages in the first few stages to maximize the work and pressure decrease and to follow it with 50 percent reaction turbines. The reaction turbine has a higher efficiency due to blade suction effects. This type of combination leads to an excellent compromise, since otherwise an all-impulse turbine would have a low efficiency, and an all-reaction turbine would have many more stages.

Turbine-Blade Cooling The turbine inlet temperatures of gas turbines have increased considerably over the past years and will continue to do so. This trend has been made possible by advancement in materials and technology, and the use of advanced turbine blade-cooling techniques. The blade metal temperature must be kept below 1400° F (760° C) to avoid hot corrosion problems. To achieve this cooling air is bled from the compressor and is directed to the stator, the rotor, and other parts of the turbine rotor and casing to provide adequate cooling. The effect of the coolant on the aerodynamic, and thermodynamics depends on the type of cooling involved, the temperature of the coolant compared to the mainstream temperature, the location and direction of coolant injection, and the amount of coolant.

In high-temperature gas turbines, cooling systems need to be designed for turbine blades, vanes, endwalls, shroud, and other components to meet metal temperature limits. Figure 29-30 shows the various types of air-cooling schemes used. The concepts underlying the following five basic air-cooling schemes are:

1. Convection cooling
2. Impingement cooling
3. Film cooling
4. Transpiration cooling
5. Water cooling

Until the late 1960s, convection cooling was the primary means of cooling gas turbine blades; some film cooling was occasionally employed in critical regions. However, in the early 1970s, other advanced cooling schemes were considered due to the greater cooling requirements for engines under development, and in the 1990s, these cooling schemes have been implemented. It should be noted that if more than 6–8 percent of the air is used in cooling, then the effect of the higher temperature becomes negated.

Convection Cooling This form of cooling is achieved by designing the cooling air to flow inside the turbine blade or vane and remove heat through the walls. Usually, the air flow is radial, making multiple passes through a serpentine passage from the hub to the blade tip. Convection cooling is the most widely used cooling concept in present-day gas turbines.

Impingement Cooling In this high-intensity form of convection cooling, the cooling air is blasted on the inner surface of the airfoil by high-velocity air jets, permitting an increased amount of heat to be transferred to the cooling air from the metal surface. This cooling method can be restricted to desired sections of the airfoil to maintain even temperatures over the entire surface. For instance, the leading edge of a blade needs to be cooled more than the midchord section or trailing edge, so the gas is impinged on that surface.

Film Cooling This type of cooling is achieved by allowing the working air to form an insulating layer between the hot gas stream and the walls of the blade. This film of cooling air protects an airfoil in the same way combustor liners are protected from hot gases at very high temperatures.

Transpiration Cooling Cooling by this method requires the coolant flow to pass through the porous wall of the blade material. The heat transfer is directly between the coolant and the hot gas. Transpiration cooling is effective at very high temperatures, since it covers the entire blade with coolant flow. This method has been used rarely due to high costs.

Water Cooling Water is passed through a number of tubes embedded in the blade. The water is emitted from the blade tips as steam to provide excellent cooling. This method keeps blade metal temperatures below 1000° F (538° C); however, a full application of this method is not expected until the year 2000.

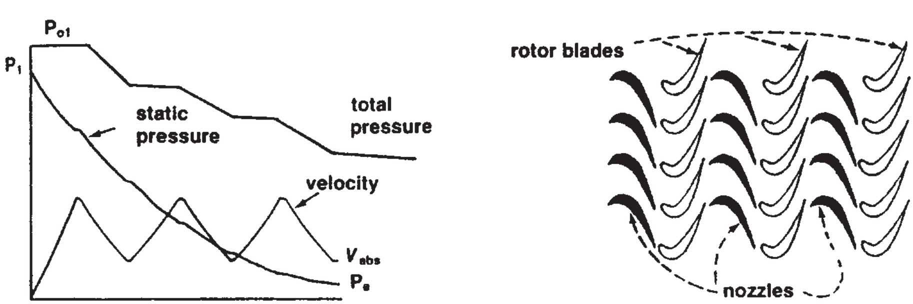


FIG. 29-29 Velocity and pressure distribution in a three-stage reaction turbine.

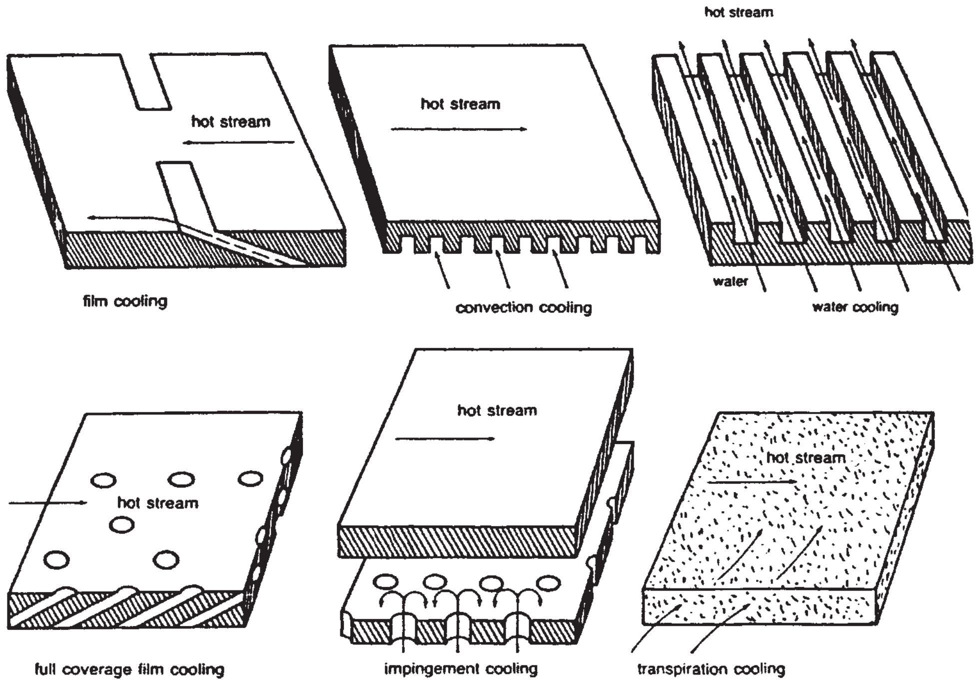


FIG. 29-30 Various suggested cooling schemes.

The incorporation of the above blade cooling concepts into actual blade designs is very important. The most frequently used blade cooling designs are:

1. Convection and impingement cooling
2. Film and convection cooling

It should be noted that in a blade the highest temperatures are encountered at the trailing edge and the highest stress points at about ⅓ the height from the base at the trailing edge.

Convection and Impingement Cooling Strut Insert Design The strut insert design has a midchord section that is convection-cooled through horizontal fins, and a leading edge that is impingement cooled. The coolant is discharged through a split trailing edge.

The air flow up the central cavity formed by the strut insert and through holes at the leading edge of the insert to impingement cool the blade leading edge. The air then circulates through horizontal fins between the shell and strut and discharges through slots in the trailing edge. The temperature distribution for this design is shown in Fig. 29-31. The stresses in the strut insert are higher than those in the shell, and the stresses on the pressure side of the shell are higher than those on the suction side. Considerably more creep strain takes place toward the trailing edge than the leading edge. The creep strain distribution at the hub section is unbalanced. This unbalance can be improved by a more uniform wall temperature distribution.

Film and Convection Cooling Design This type of blade design has a midchord region that is convection cooled, and the leading edges which are both convection and film cooled. The cooling air is injected through the blade base into two central and one leading edge cavity. The air then circulates up and down a series of vertical passages. At the leading edge, the air passes through a series of small holes in the wall of the adjacent vertical passages and then impinges on the inside surface of the leading edge and passes through film-cooling holes. The trailing edge is convection cooled by air discharging through slots.

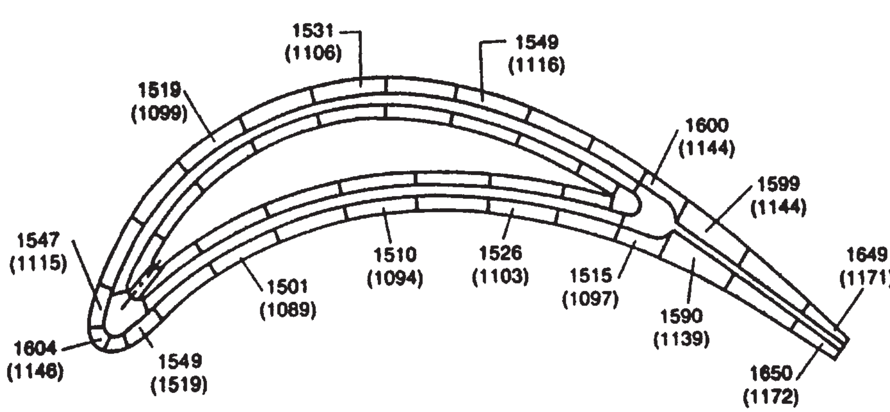


FIG. 29-31 Temperature distribution for strut insert design, °F (cooled).

The temperature distributions for film and convection cooling design are shown in Fig. 29-32. From the cooling distribution diagram, the hottest section can be seen to be the trailing edge. The web, which is the most highly stressed blade part, is also the coolest part of the blade.

Major Cycles The major application of most gas turbines is in an open cycle in which air is the working medium. The gas turbine can either be a single-shaft unit or a multiple-shaft unit. The single-shaft unit is one in which the compressor, the gas generator turbine, and the power turbine are on a single shaft. The single-shaft units are used in most electrical generating services where constant speed application is the norm. Multiple shaft units can be two or three shaft units. In a two-shaft unit, the high-pressure turbine (gas generator turbine) is driving the air compressor, and the low-pressure turbine, which is on a separate shaft and only aerodynamically coupled, produces the output power. A three-shaft unit is used in designs with very high compressor-pressure ratios. The high pressure requires two compressors: a low-pressure compressor and a high-pressure compressor. The high-pressure compressor is driven by the high-pressure turbine; the low-pressure compressor is driven by the intermediate-pressure turbine; and the low-pressure turbine drives the output shaft. Multiple shaft turbines are used often in mechanical drives where the driven equipment needs to be operated over a wide speed range.

The advantage of a multiple shaft unit over a wide output speed range is that the compressor and the compressor turbines can be maintained at relatively constant speed (75–100 percent of design), while the power turbine (low pressure turbine) can be operated over a very wide speed range without a great loss in thermal efficiency for the entire gas turbine. Variable area nozzles are sometimes used to control the low-pressure turbine; the high pressure turbine is controlled by change in the fuel flow, which affects the turbine firing temperature.

The Simple Cycle The simple cycle, or the Brayton cycle, is the most common type of cycle being used in the gas-turbine field today. The overall efficiency of a cycle can be improved by increasing the pressure ratio or the turbine inlet temperature (firing temperature). Today's simple-cycle turbines have pressure ratios as high as 17:1 and firing temperatures of about 2300° F (1260° C). In a simple cycle, there is an optimum pressure ratio for a turbine-firing temperature giving the highest efficiency. The efficiency of the various components such as the compressor, combustor, and turbine affects the overall thermal efficiency; however, even if these components were 100 percent efficient, the thermal cycle efficiency would only approach that of a Carnot cycle, which is the most efficient cycle between any two temperatures. Figure 29-33 shows the effect of pressure ratio and turbine inlet temperature on power and efficiency of simple cycle gas turbines.

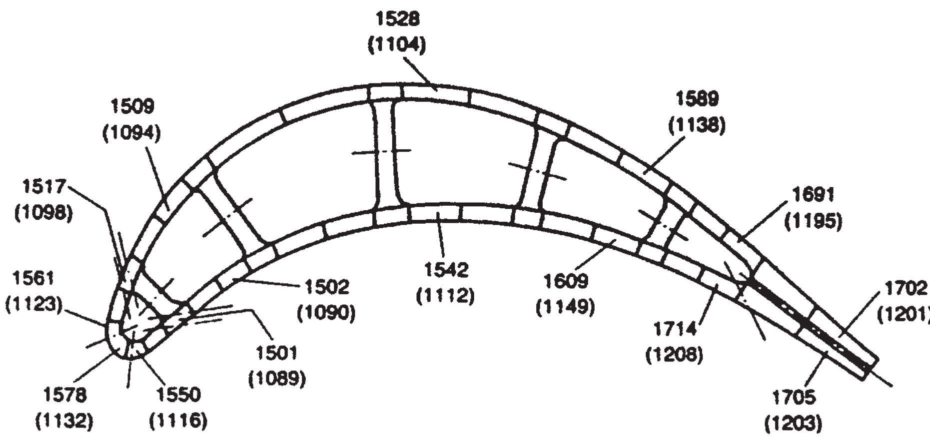


FIG. 29-32 Temperature distribution for film convection-cooled design, °F (cooled).

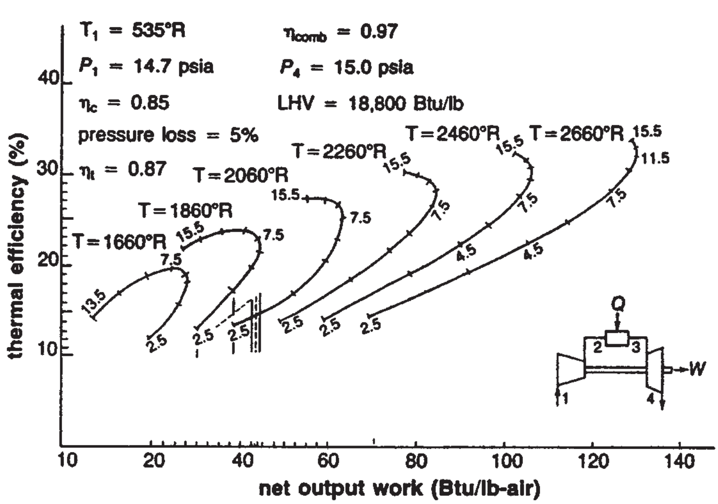


FIG. 29-33 Performance map showing the effect of pressure ratio and turbine inlet temperature on a simple cycle.

The Regenerative Cycle The regenerative cycle is becoming prominent in these days of tight fuel reserves and high fuel costs. The amount of fuel needed can be reduced by the use of a regenerator in which the hot turbine exhaust gas is used to preheat the air between the compressor and the combustion chamber. The regenerator increases the temperature of the air entering the burner, thus reducing the fuel-to-air ratio and increasing the thermal efficiency. For a regenerator assumed to have an effectiveness of 80 percent, the efficiency of the regenerative cycle is about 40 percent higher than its counterpart in the simple cycle, as seen in Fig. 29-34. The work output per pound of air is about the same or slightly less than that experienced with the simple cycle. The point of maximum efficiency in the regenerative cycle occurs at a lower pressure ratio than that of the simple cycle, but the optimum pressure ratio for the maximum work is the same in the two cycles.

The Reheat Cycle The regenerative cycle improves the efficiency of a gas turbine but does not provide any added work per pound of air flow. To achieve this latter goal, the concept of the reheat cycle must be utilized. The reheat cycle utilized in the 1990s has pressure ratios of as high as 30:1 with turbine inlet temperatures of about 2100° F (1150° C). The reheat is done between the power turbine and the compressor trains. The reheat cycle, as shown in Fig. 29-35, consists of a two-stage turbine with a combustion chamber before each stage. The assumption is made that the high-pressure turbine is only to drive the compressor and that the gas leaving this turbine is then reheated to the same temperature as in the first combustor before entering the low-pressure or power turbine.

The Intercooled Regenerative Reheat Cycle The Carnot cycle is the optimum cycle between two temperatures, and all cycles try to approach this optimum. Maximum thermal efficiency is achieved by approaching the isothermal compression and expansion of the Carnot cycle or by intercooling in compression and reheating in the expansion process. The intercooled regenerative reheat cycle approaches this optimum cycle in a practical fashion. This cycle achieves the maximum efficiency and work output of any of the cycles described to this point. With the insertion of an intercooler in the compressor, the pressure ratio for maximum efficiency moves to a much higher ratio, as indicated in Fig. 29-36.

The Steam Injection Cycle Steam injection has been used in reciprocating engines and gas turbines for a number of years. This cycle may be an answer to the present concern with pollution and higher efficiency. Corrosion problems are the major hurdle in such a system. The concept is simple and straightforward: Steam is injected into the compressor discharge air and increases the mass flow rate

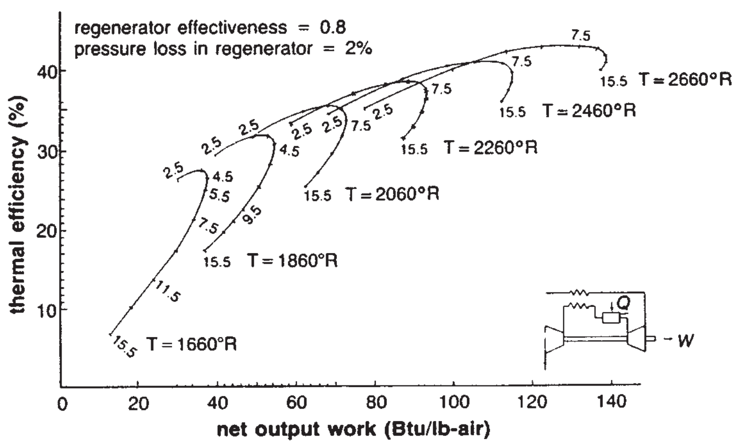


FIG. 29-34 Performance map showing the effect of pressure ratio and turbine inlet temperature on a regenerative cycle.

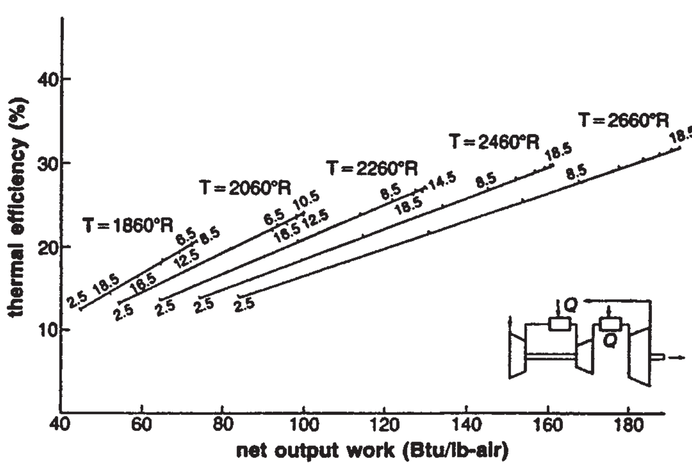


FIG. 29-35 Performance map showing the effect of pressure ratio and turbine inlet temperature on a split-shaft reheating cycle.

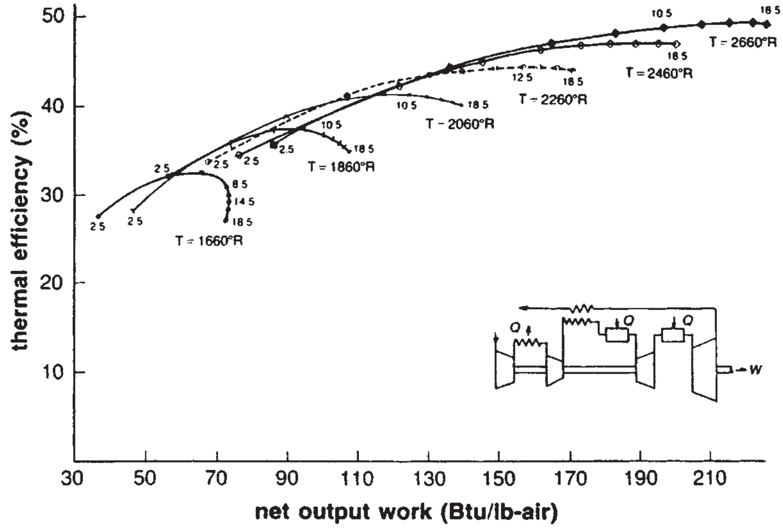


FIG. 29-36 Performance map showing the effect of pressure ratio and turbine inlet temperature on an intercooled regenerative reheat split-shaft cycle.

through the turbine. The steam being injected downstream from the compressor does not increase the work required to drive the compressor. The steam used in this process is generated by the turbine exhaust gas. Typically, water at 14.7 psi (101 kPa) and 80° F (27° C) enters the regenerator, where it is brought up to 60 psi (413 kPa) above the compressor discharge and the same temperature as the compressor discharge air. The steam is injected after the compression but far upstream of the burner to create a proper mixture.

For NO_x control only, steam is injected into the combustor directly to help reduce the primary zone temperature in the combustor. The amount of steam injected is in a ratio of 1:1 with the fuel. In this cycle, the steam is injected upstream of the combustor and can be as much as 5–8 percent by weight of the air flow. This cycle leads to an increase in output work and a slight increase in overall efficiency. Corrosion problems due to steam injection have been for the most part overcome with new high temperature coatings. Figure 29-37 shows the increase in efficiency and work output for various steam flow rates at a fixed turbine inlet temperature. Figure 29-38 shows the effect of 5 percent steam injection at various turbine firing temperatures.

The Combined (Brayton-Rankine) Cycle The 1990s has seen the rebirth of the combined cycle, the combination of gas turbine technologies with the steam turbine. This has been a major shift for the utility industry, which was heavily steam-turbine-oriented with the use of the gas turbine for peaking power. In this combined cycle, the hot gases from the turbine exhaust are used in a heat recovery steam generator or in some cases in a supplementary fired boiler to produce superheated steam.

The combined cycle work is equal to the sum of the net gas turbine work and the steam turbine work. About one-third to one-half of the design output is available as energy in the exhaust gases. The exhaust

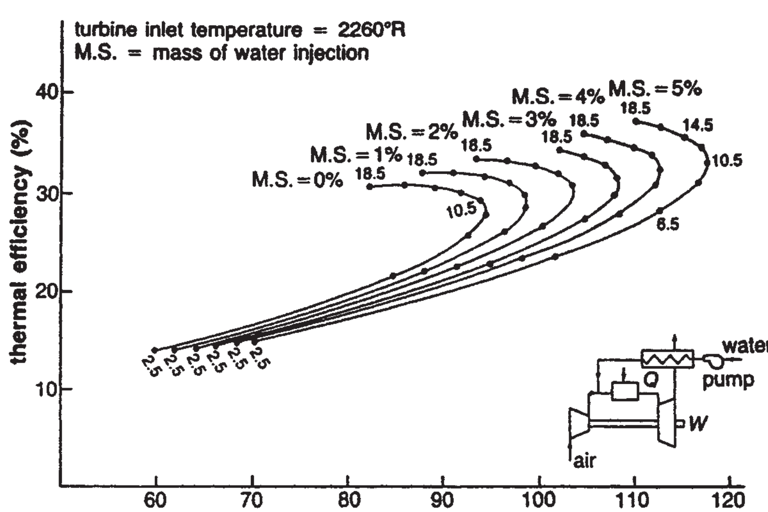


FIG. 29-37 Performance map showing the effect of pressure ratio and steam flow rate on a steam injection cycle.

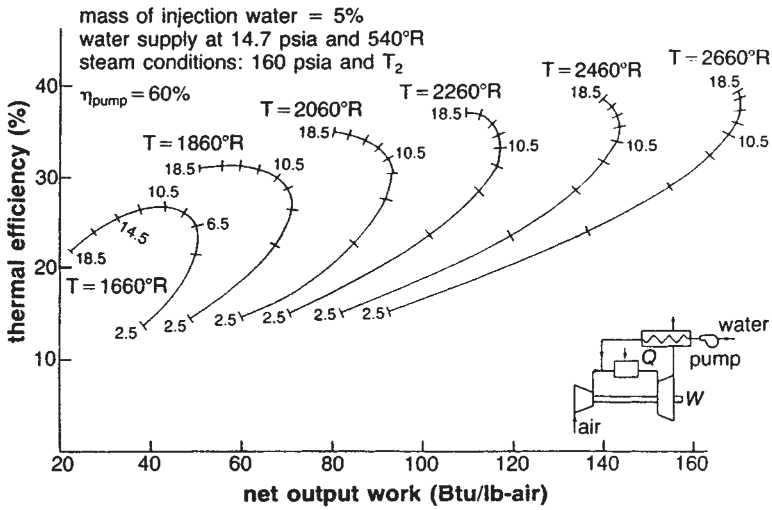


FIG. 29-38 Performance map showing the effect of pressure ratio and turbine inlet temperature on a fixed steam rate in a steam injection cycle.

gas from the gas turbine is used to provide heat to the recovery boiler. Thus, this heat must be credited to the overall cycle. This makes the combined cycle the highest practical efficiency cycle today. Figure 29-39 shows the effect of this cycle on the overall plant efficiency. To reduce the NO_x effect in the gas turbine, steam is injected in the combustor at a ratio of 1:1 with the fuel.

A comparison of the effect of the various cycles on the overall thermal efficiency is shown in Fig. 29-40. The most effective cycle is the Brayton-Rankine (combined) cycle. This cycle has tremendous potential in power plants and in the process industries where steam turbines are in use in many areas. The initial cost of the combined cycle is between $800–$1200 per kW while that of a simple cycle is about $300–$600 per kW. Repowering of existing steam plants by adding gas turbines can improve the overall plant efficiency of an existing steam turbine plant by as much as 3 to 4 percentage points.

Typically an inlet pressure decrease of one inch of water column reduces the power output by 0.4 percent and increases the heat rate by 0.125 percent. Similarly, an exhaust pressure increase of one inch of water reduces the power output by 0.15 percent and the heat rate by 0.125 percent.

TURBINE OPERATION CHARACTERISTICS

The gas turbine is a high-volume air machine. The compressor air power required is usually between 50–70 percent of the total power produced by the turbine. Thus, the ambient temperature affects the output of the gas turbine. On hot days, the gas turbine produces less output than on cold days. In dry climates, the use of evaporative cooling in the gas turbine decreases the effective inlet temperature and increases the power output of the unit.

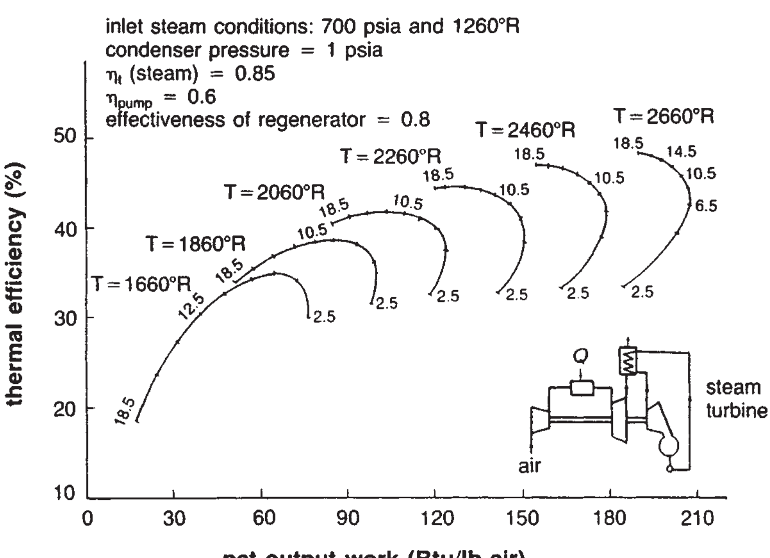


FIG. 29-39 Performance map showing the effect of pressure ratio and turbine inlet temperature on a Brayton-Rankine cycle.

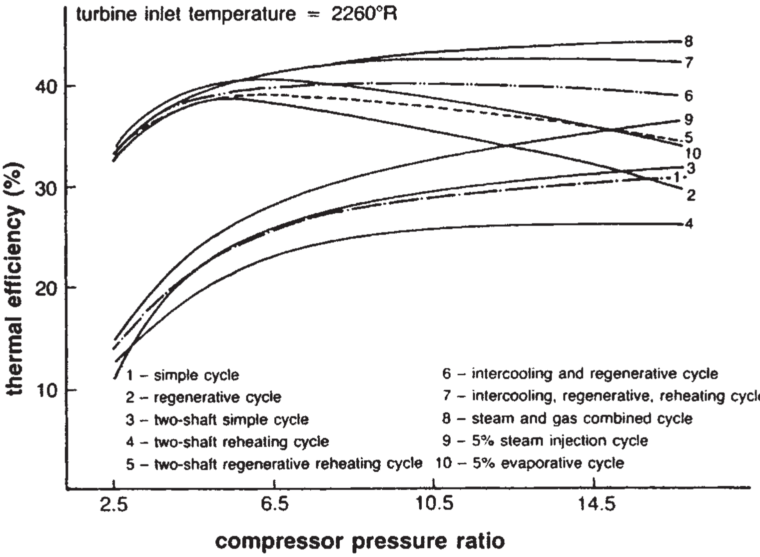


FIG. 29-40 Comparison of thermal efficiency of various cycles.

The effect on the performance of the inlet and exit conditions of the gas turbine is predominant. A increase in the inlet temperature by 5° F (2.8° C) will reduce the design power output by 2 %, and a reduction in the inlet pressure by 1 psi (6.9 kPa) would reduce the output by approximately 3 percent. Also, a 1 percent change in compressor efficiency reduces the overall thermal efficiency by ½ percent and reduces the power output by 2 percent. A 1 percent change in the turbine efficiency will produce a change of about 3 percent in turbine power output and a 0.75 percent change in the overall thermal efficiency. It is therefore very important to maintain the compressor in a very clean state. To do this, water solvent on-line cleaning is carried out on the compressor. Water jet nozzles are placed at the periphery of the compressor inlet. This type of cleaning is effective only on the first few stages. Coating the compressor blades has been found to improve efficiency. Inlet and outlet ducting should be designed with minimum losses; however, good filtration should not be compromised.

The gas turbine is usually started by an auxiliary drive such as a steam turbine, diesel engine, turboexpander, or electric motor. These drives bring the turbine up to a speed of between 1200–2000 rpm, at which time fuel is injected and the turbine speed is increased rapidly as the turbine firing temperature increases. The power requirement for the starter is about 5–10 percent of the power rating of the unit. Most turbine control systems have an acceleration monitor that shuts down the turbine if an acceleration rate is not maintained. *This is very essential if combustion in the turbine nozzles and blades is to be avoided.* To avoid compressor surges during startup and shutdown, a series of bleed valves in the compressor are sequentially closed at various speeds during the startup or are opened at various speeds during the shutdown. These bleed valves also are used for directing cooling air to the various regions of the turbine. Bleed valves are located in the early stages (i.e., 5th or 6th stage) and in the latter stages (i.e., 11th or 12th stage) of an axial flow compressor.

Multiple-shaft turbines require a little more starting power than single-shaft turbines. The low pressure turbine reaches breakaway torque at about 50–60 percent of the design speed of the gas generator section. In cases of aborted starts, the gas turbine must be fully purged before another start is attempted; otherwise, the gas trapped in the turbines could explode. In many new gas turbines, the first 5–7 stages of the compressor have variable stators that change angles at various settings. These move with speed and thus reduce the losses encountered in the compressor section.

In the case of the steam-injected cycle, steam must be injected after the turbine has been brought up to full speed; otherwise, compressor surges could occur. Major temperature excursions during startups must also be avoided to prevent degradation of the life of the turbine.

Life Cycle The gas turbine life and especially hot section life are significantly influenced by the following parameters:

1. Blade material and cooling flow (blade metal temperature)
2. Type of fuel
3. Number of starts and full load trips

BLADE MATERIALS

Turbine life is very sensitive to blade metal temperature and blade material. It is essential for prevention of hot corrosion to keep this temperature below 1400° F (760° C). This temperature is a function of cooling flow to the blades, and the firing temperature. The effect of cooling flow blockage can be catastrophic for these blades. The life of the blades, especially on rotating elements, is a function of blade stresses and metal temperature. The Larson Miller curve shown in Fig. 29-39 for typical turbine blade alloys shows the logarithmic relationship between these parameters. While widely used to describe an alloy's stress rupture characteristic over a side temperature life and stress range, it is also very useful in comparing the elevated temperature capabilities of many alloys. Turbine blade alloys, which usually have large quantities of Ni, Cr, and Co, tend to indicate low durability at operating temperatures. This results in surface notches initiated by erosion or corrosion, after which cracks are propagated rapidly. This often leads to high cycle and low stress failure.

Turbine blade coatings can extend the life of blades by nearly 70–80 percent. Coating prevents corrosion from attacking the base metal. Most present-day coatings are diffusion-packed-type coatings. They usually consist of a thin uniform layer of a precious metal (platinum) electroplated onto the blade surface. This procedure is followed by pack diffusion steps to deposit layers of aluminum and chromium, resulting in a coating that has an outer skin of an extremely corrosion-resistant, platinum-aluminum intermetallic composition. The gas turbines of the 1990s are all coated, especially in the hot section area. Coatings are also being applied to the compressor blades and have been found to be very effective.

TYPES OF FUEL

The life of a gas turbine depends heavily on the type of fuel used. An inherent fuel flexibility is the gas turbine's major advantage. Gaseous fuels traditionally include natural gas, process gas, and low-Btu gas

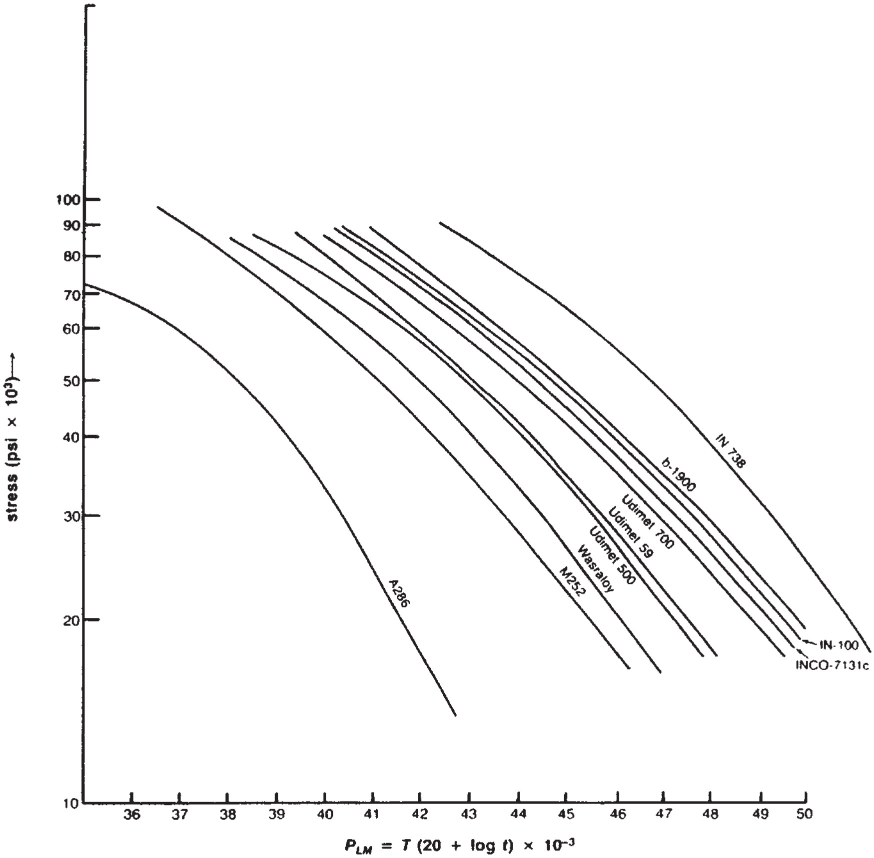


FIG. 29-41 Larson-Miller parameters for turbine blade alloys.

(coal gas or water gas). Natural gas is the benchmark against which performance of a gas turbine is compared, since it is a clean fuel that promotes long machine life.

Liquid fuels vary from light volatile fuels such as naphtha through kerosene to the heavy viscous residuals. True distillate fuel such as #2 distillate oil is a good fuel; however, because trace elements such as vanadium, sodium, potassium, lead, and calcium are found in the fuel, the fuel has to be treated. The corrosive effect of sodium and vanadium is very detrimental to the life of a turbine. Vanadium originates as a metallic compound in crude oil and is concentrated by the distillation process into heavy oil fractions. Sodium compounds are usually present in the form of salt water, which results from salty wells, transport over seawater, or mist ingestion in an ocean environment. Fuel treatments are costly and do not remove all traces of the metal. Sodium is usually removed by water washing and letting the sodium dissolve into the water. It is then separated by a centrifuge. Vanadium is counteracted by the addition of magnesium which causes a friable deposit. A magnesium/vanadium ratio of 3:1 reduces corrosion by a factor of six.

The typical amounts of sodium and vanadium in the fuel should be less than 1 ppm. Figure 29-42 shows the effect of sodium and vanadium on the life of the blade and on the combustor life. Figure 29-43 shows the reduction in firing temperature required to maintain design life (hrs) of a typical turbine (IN718) blade due to sodium and vanadium in the fuel.

In general terms, the life of a combustor might be reduced by about 30 percent through use of a distillate fuel and by 80 percent through the use of residual fuel. The first stage turbine nozzle life can be reduced by 20 percent through use of a distillate fuel and by about 65 percent when certain residual fuels are used.

Number of Starts and Full-Load Trips Temperature differentials developed during starting and stopping of the turbine produce thermal stresses. The cycling of these thermal stresses causes thermal fatigue. Thermal fatigue is a low cycle event and is similar to creep rupture failure. The analysis of thermal fatigue is essentially a problem in heat transfer and is affected by properties such as modulus of elasticity, coefficient of thermal expansion, and thermal conductivity. The most important metallurgical factors are ductility and toughness.

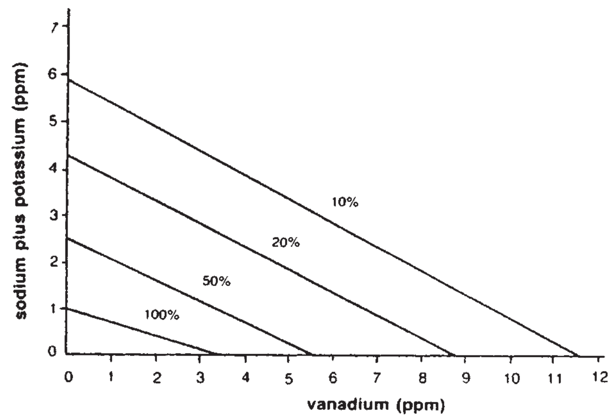


FIG. 29-42 Effect of sodium, potassium, and vanadium on combustor life.

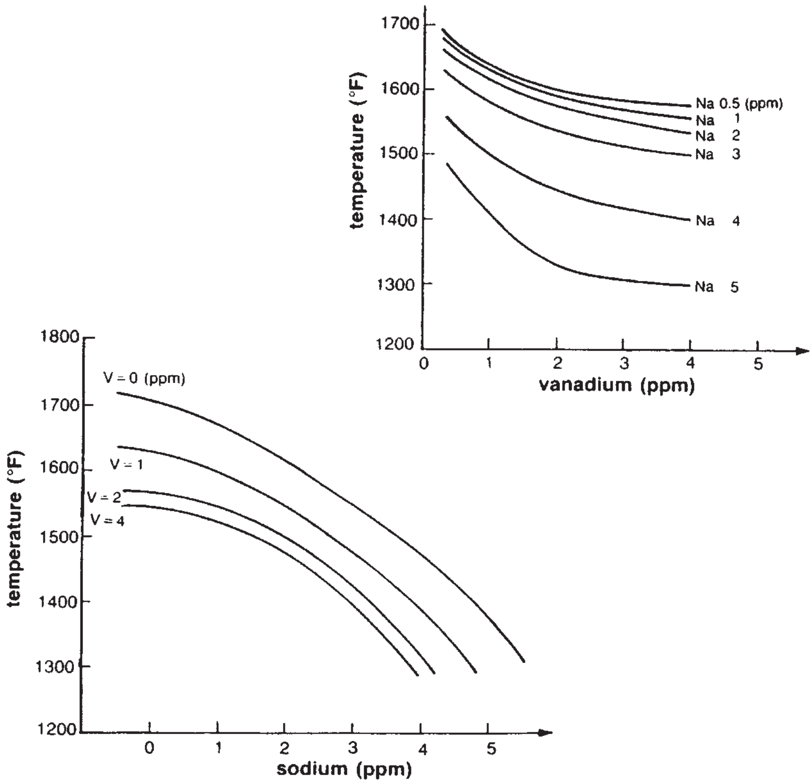


FIG. 29-43 Firing temperature reduction needed to offset IN 718 corrosion by sodium and vanadium.

Highly ductile materials tend to be more resistant to thermal fatigue and also seem more resistant to crack initiation and propagation.

The operating schedule of a gas turbine produces low-frequency thermal fatigue. The number of starts per hours of operating time directly affects the life of the hot sections (combustor, turbine nozzles, and blades). The life reduction effect of the number of starts on a combustor liner could be as high as 230 hours/start and on the turbine nozzles as high as 180 hours/start. The effect of full load trips can be nearly 2–3 times as great!

The life of a gas turbine depends on the above detailed operational characteristic. It is interesting to note that, for a gas turbine life of 25 years, the life cycle costs can be distributed as 5–10 percent on initial cost, 10–20 percent on maintenance costs and 70–85 percent on cost of fuel. Gas turbines will be very widely used in the 21st century in combined cycle applications as the power source for the world. These combined cycle plants will have efficiencies in the high fifties and will cost between \$1000 and \$1200 per kW, using 1994 as a monetary benchmark.

EXPANSION TURBINES

Fundamentally, an expansion turbine is a device for converting the pressure energy of a gas or vapor stream into mechanical work as the gas or vapor expands through the turbine. The mechanical work so produced, however, is generally a by-product, the primary objective of the turboexpander being to chill the process gas. Turboexpanders are in wide use in the cryogenic field to produce the refrigeration required for the separation and liquefaction of gases.

By common usage, the terms “turboexpanders” and “expansion turbines” specifically exclude steam turbines and combustion gas turbines, which are covered elsewhere in Sec. 29.

Any work developed by the turboexpander is at the expense of the enthalpy of the process stream, and the latter is correspondingly cooled. A low inlet temperature means a correspondingly lower outlet temperature, and the lower the temperature range, the more effective the expansion process becomes.

FUNCTIONAL DESCRIPTION

The turboexpander in combination with a compressor and a heat exchanger functions as a heat pump and is analyzed as follows: In Fig. 29-44 consider the compressor and aftercooler as an isothermal compressor operating at T_2 with an efficiency E_c, and assume the working fluid to be a perfect gas. Further, consider the removal of a quantity of heat Q_e by the turboexpander at an average low temperature T_1. This requires that it deliver shaft work equal to Q_e. Now, make the reasonable assumption that one-tenth of the temperature drop in the expander is used for the temperature difference in the heat exchanger. If the expander efficiency is N_e and this efficiency is multiplied by 0.9 to include the effect of the temperature difference in the heat exchanger, the needed ideal enthalpy drop across the expander is

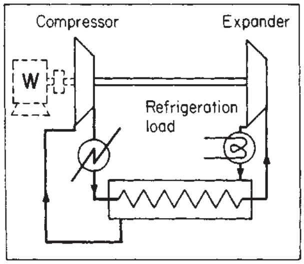


FIG. 29-44 Turboexpander system functioning as a refrigeration machine.

$$H_e = Q_e/0.9N_e \tag{29-16}$$

The theoretical required (isothermal) compression work in the compressor, which is assumed to operate isothermally at T_2, is

$$(Q_e/0.9N_e)(T_2/T_1) \tag{29-17}$$

The actual compressor work W_c is this latter quantity, divided by the compressor isothermal efficiency N_c; thus,

$$W_c = (Q_e/0.9N_eN_c)(T_2/T_1) \tag{29-18}$$

Mechanical work equal to $Q_e/0.9$ is returned by the expander to the compressor, so the net work to the compressor is

$$W = W_c - \frac{Q_e}{0.9} = \left(\frac{Q_e}{0.9}\frac{T_2}{N_eN_cT_1}\right) - \frac{Q_e}{0.9}$$

$$W = \left(\frac{Q_e}{0.9}\frac{T_2}{N_eN_cT_1}\right) - 1 = \frac{Q_e}{0.9}\left(\frac{T_2 - N_eN_cT_1}{N_eN_cT_1}\right) \tag{29-19}$$

The second-law theoretical work is

$$W_{\text{theor}} = Q_e\frac{T_2 - T_1}{T_1} \tag{29-20}$$

Hence, the second-law efficiency of the expander-heat-exchanger-compressor system is

$$\frac{W_{\text{theor}}}{W} = \frac{Q_e\dfrac{T_2 - T_1}{T_1}}{\dfrac{Q_e}{0.9}\left(\dfrac{T_2 - N_eN_cT_1}{N_eN_cT_1}\right)} = \frac{0.9(T_2 - T_1)N_eN_c}{T_2 - N_eN_cT_1} \tag{29-21}$$

A plot of this efficiency in which commonly available equipment is assumed is shown by the expander curve in Fig. 29-45.

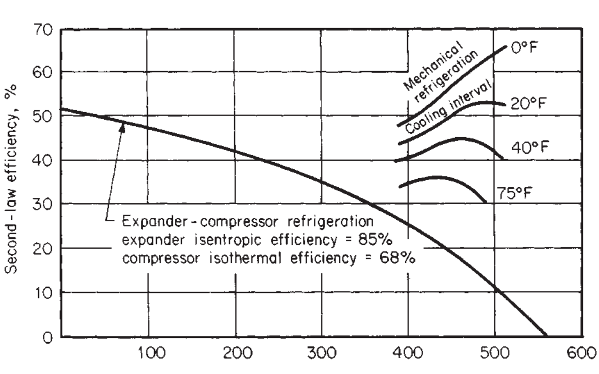


FIG. 29-45 Mechanical versus turboexpander refrigeration. K = 5/9°R; °C = 5/9(°F − 32).

The family of short curves in Fig. 29-45 shows the power efficiency of conventional refrigeration systems. The curves for the latter are taken from the *Engineering Data Book*, Gas Processors Suppliers Association, Tulsa, Oklahoma. The data refer to the evaporator temperature as the point at which refrigeration is removed. If the refrigeration is used to cool a stream over a temperature interval, the efficiency is obviously somewhat less. The short curves in Fig. 29-45 are for several refrigeration-temperature intervals. A comparison of these curves with the expander curve shows that the refrigeration power requirement by expansion compares favorably with mechanical refrigeration below 360° R (−100° F). The expander efficiency is favored by lower temperature at which heat is to be removed.

Another conclusion that can be drawn from Fig. 29-45 is that if the process can justify the complexity, it is more efficient, powerwise, to use conventional means rather than expanders to absorb heat at moderate temperatures in the range of ambient to 360° R, although for expediency expanders are frequently used in any case.

SPECIAL CHARACTERISTICS

An example of a typical turboexpander is shown in Fig. 29-46. Radial-flow turbines are normally single-stage and have combination impulse-reaction blades, and the rotor resembles a centrifugal-pump impeller. The gas is jetted tangentially into the outer periphery of the rotor and flows radially inward to the "eye," from which the gas is jetted backward by the angle of the rotor blades so that it leaves the rotor without spin and flows axially away.

Radial-flow turbines have been developed primarily for the production of low temperatures, but they also may be used as power-recovery devices.

The characteristics of these machines include the following:

1. High efficiency: 75 to 88 percent
2. Operation usually at a very low temperature
3. Operation often on small or moderate streams, dictating a rather high rotating speed
4. Supports having low heat conductivity
5. Effective shaft seals to conserve the process stream
6. Heavy-duty construction resistant to abuse
7. High reliability

Commonly established operating limitations for turboexpanders without special design features are an enthalpy drop of 93 to 116.3 kJ/kg (40 to 50 Btu/lb) per stage of expansion and a rotor-tip speed of 304.8 m/s (1000 ft/s). Commercial turboexpanders are available for inlet pressure up to 20.68 MPa (3000 lbf/in^2) and inlet temperatures from near absolute zero to 538° C (1000° F). The permissible liquid condensation in the expanding stream varies with discharge pressure; it may be 50 weight percent or higher in the discharge, provided the turboexpander has been specially designed to handle condensation.

RADIAL INFLOW DESIGN

The radial reaction design has been selected for turboexpanders primarily because it attains the highest efficiency of all turbine designs. However, it has several additional features which favor this application:

1. In the single-stage configuration, it is usual to have the rotor on the end of the shaft (overhung); this provides a convenient opportunity for thermally insulating the cold turbine portion and is an ideal arrangement for axial discharge.
2. This design permits variable primary nozzles, which enable the attainment of high efficiency over a wide flow range.
3. It applies lower axial thrust to the shaft than a single-stage axial reaction turbine.
4. It has shaft bearings on only one side of the expander rotor, and heat-barrier insulation between the warm, lubricated bearings and the cold turbine is convenient to arrange.
5. Its speed is reasonably acceptable for suitable loading devices.

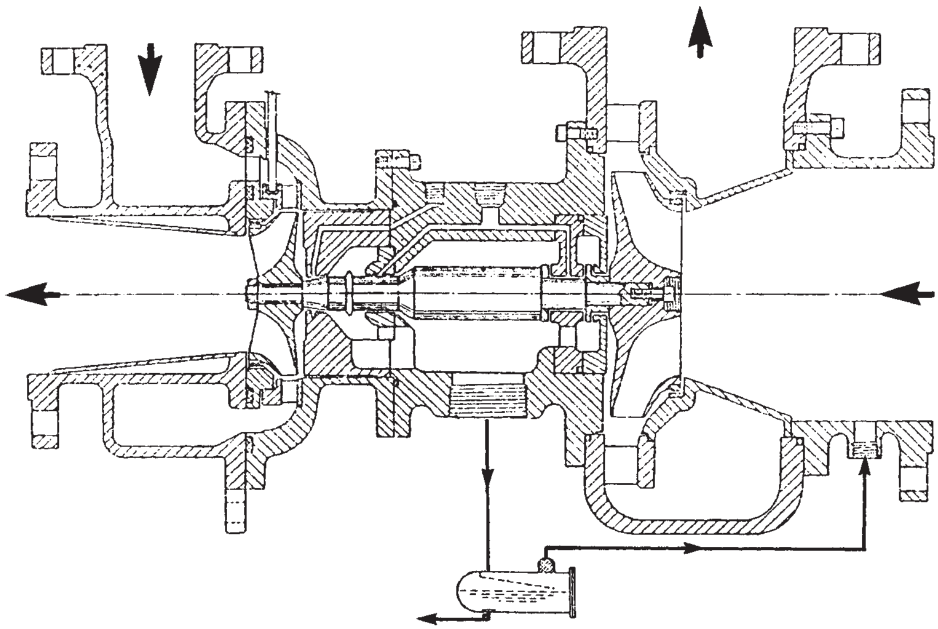


FIG. 29-46 Typical radial-flow turboexpander.

EFFICIENCY

Efficiency for a turboexpander is calculated on the basis of isentropic rather than polytropic expansion even though its efficiency is not 100 percent. This is done because the losses are largely introduced at the discharge of the machine in the form of seal leakages and disk friction which heats the gas leaking past the seals and in exducer losses. (The exducer acts to convert the axial-velocity energy from the rotor to pressure energy.)

BEARINGS

Radial Bearings Antifriction bearings are largely unsuitable for these applications, chiefly because of the special attention and maintenance which they demand.

Virtually all expanders have lubricated sleeve bearings or tilting-shoe bearings. The advantage of sleeve bearings is that they can be designed to support the shaft sufficiently rigidly that the oil-film critical (first critical) can be designed to be safely above the design running speed. This is desirable because it eliminates all shaft vibrations in the full operating range; more important, if the critical were below the design speed, then at design speed the shaft assembly would be rotating about its center of gravity. In turboexpanders, there frequently are ice deposits or other reasons for rotor imbalance, and such imbalance would cause gyration and damage to the shaft seals.

In arrangements in which the shaft is running with its first critical (oil-film critical) below the design speed, then not far above design speed is the oil swirl (half-speed gyration). Substitution of tilting-shoe bearings for sleeve bearings moves the oil swirl to a higher speed, but it does not prevent the rotor from rotating about its center of gravity. Tilting-shoe bearings do not lubricate well at this high rubbing speed and small shaft diameter.

The sleeve bearing has the further advantage that it functions as a lubricated, pressurized shaft seal to contain the pressure on the process gas (see subsection "Shaft Seals").

Thrust Bearings Turboexpanders often have process upsets or ice plugging or the like, which can cause serious thrust-bearing load variations. In applications above 506.6 to 1013.2 kPa (75–148 psi), the best available thrust bearing usually is insufficient to protect against such high thrust loads. Various indications, such as the differential pressure across the rotor and thrust-bearing temperatures, are available to protect the unit at least to some extent.

Thrust-bearing-load meters for protection against excessive loading are available on thrust bearings, and automatic thrust control, which functions by controlling the pressure behind a balancing drum, also is available.

SEALS

Shaft Seals Mechanical shaft seals generally are not acceptable in turboexpanders for the same general reason that antifriction bearings are not: they require periodic replacement and careful attention. In their stead, close-clearance labyrinth-type seals are generally used. Such a labyrinth seal generally has an injection point intermediate from its two ends into which a suitable buffer gas is injected to prevent the escape of the process gas; instead, buffer gas escapes. If the escaping seal gas is inexpensive and nontoxic, it may be allowed to leak to the atmosphere. However, it is possible to enclose the expander housing to the bearing housing and allow the journal bearing, acting as an oil seal, to contain the outleaking seal gas and collect it in a float-operated drainer for suitable disposal or reuse.

In refrigeration applications in which the refrigerant must be completely conserved, the expander housing and bearing housing can be hermetically sealed through a speed-reducing gearbox. The low-speed shaft of the gearbox is sealed with a low-speed mechanical seal. Then any refrigerant which leaks out of the labyrinth seal is totally contained in the gearbox and in the closed lubrication system for complete collection and reuse.

Rotor Seals To balance the thrust on the rotor, usually there are one or two labyrinth-type seals on the rotor. These seals often are damaged if there is dust in the incoming fluid or gas, and wear on the backside seal causes serious upsets in thrust-bearing loads. Provisions are available for collecting and disposing of the dust which tends to accumulate in the seal so as to protect the seal from serious erosion.

VARIABLE NOZZLES

The pressurized process stream is guided radially into the rotor by the primary nozzles, which are a series of vanes forming nozzles jetting the gas tangentially and inwardly into the rotor (see Fig. 29-47). These

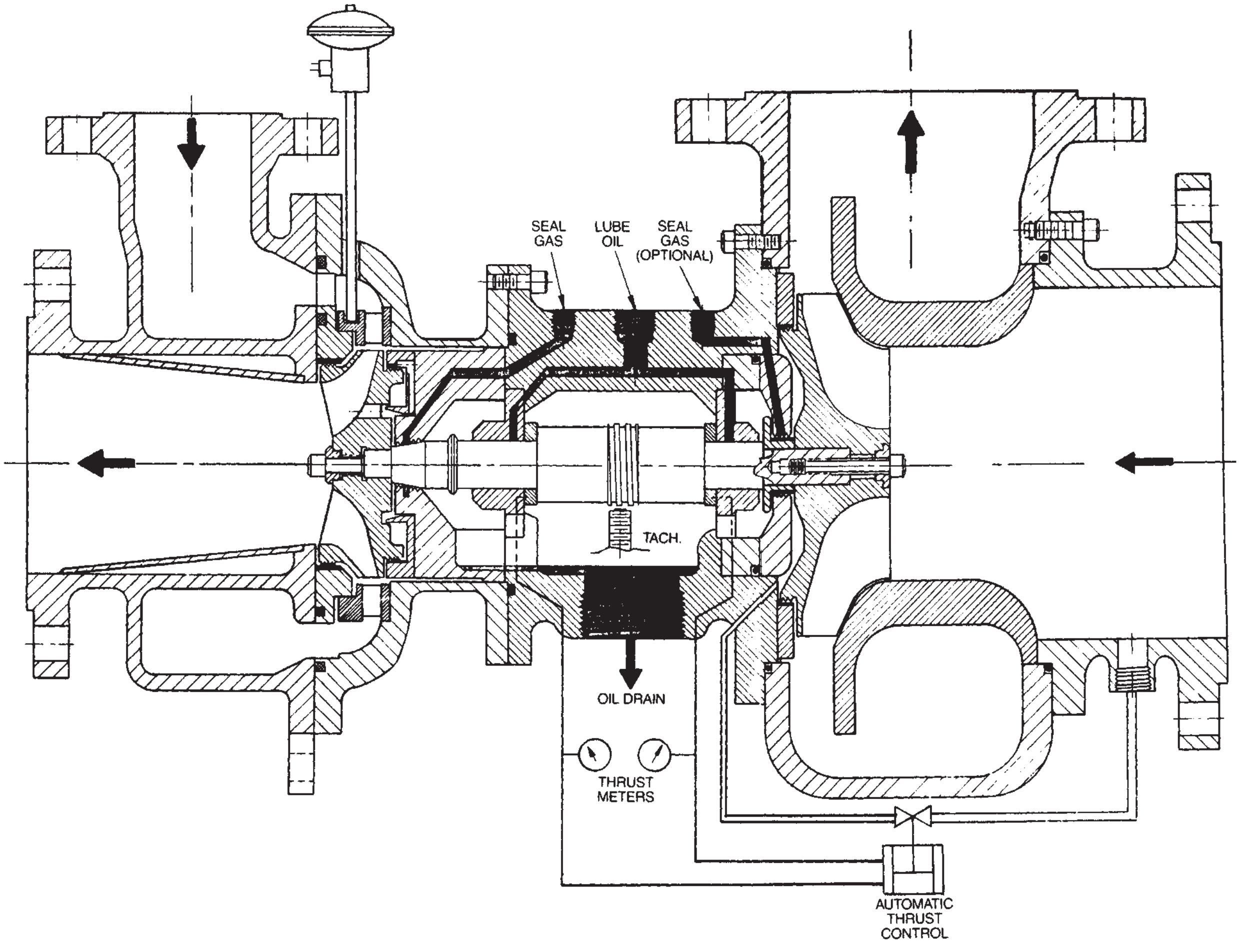


FIG. 29-46 (*Continued*) Typical radial-flow turboexpander.

nozzle vanes are clamped between two flat rings and usually are pivoted so that they can be rotated in unison to open or close the spaces between them in order to vary the nozzle-throat areas. This is quite an important function because it can be used to vary the flow widely through the expander without wasteful throttling: all the expansion energy in the nozzles is recovered in the rotor. The variable nozzles, from a control standpoint, act just as a throttle valve would act in controlling the flow (but without throttling loss), and conventional flow-control instrumentation can be used to operate them.

ROTOR RESONANCE

Variable nozzles produce a series of jets of gas entering the rotor, and these impulses add up to form a frequency equal to the blade-passing frequency: the number of revolutions per second multiplied by the number of nozzle vanes, which is of the order of thousands of cycles per second. Frequently the rotor will resonate at this frequency, and if it does, it will be fatigued and crack and break up; thus these frequencies must be avoided, and the manufacturer should be asked to supply information to the customer on this subject.

CONDENSING STREAMS

It is advantageous to have the expander generate its refrigeration at the lowest possible temperature in the process (see Fig. 29-45), and this frequently encounters the condensation temperature of the process stream. Steam-turbine practice advises against operating on condensing streams because efficiency is deteriorated and there also usually is erosion of the rotor blades.

Another advantage of the radial reaction turbine is that it can be designed to accept condensation in any amount without efficiency deterioration or erosion.* This is possible because there are two forces acting on suspended fog particles, the deceleration force and the centrifugal force, and these two forces can be balanced against each other to prevent the droplets from impinging on specially shaped blades. The process is explained as follows:

This expansion of a condensing vapor is highly desirable thermodynamically, but the liquid must not bombard and erode the rotor blades, and, in particular, it must not accumulate in the rotor, since that would cause efficiency loss.

If liquid droplets form as the gas is expanded in the turboexpander, one's first thought may be that a radial inflow design is the last thing to use, but the following explanation will show that this is the only design that can accomplish expansion efficiently.

Figure 29-47 shows the primary nozzles and the rotor. About half of the pressure drop takes place in the primary nozzles, which jet the gas tangentially into the periphery of the rotor. Cooling takes place during this expansion, and the jetted stream entering the rotor may be foggy. This foggy gas flows radially inward within the rotor as the latter rotates, and at the outlet, which is near the center of the rotor, the

* This same principle applies also to the expansion of flashing liquids in which the bubbles are guided away from the blades.

FIG. 29-47 The primary nozzles and the rotor.

fluid is discharged by being jetted backward out of the rotor so as to leave without rotary motion. The second half of the expansion energy is spent by the gas passing radially through the rotor against centrifugal force, and further precipitation of liquid takes place.

The stream from the nozzles enters the rotor with a tangential velocity of about 152.4 or 304.8 m/s (500 or 1000 ft/s; see Fig. 29-48) and follows a path through the rotor of such curvature that the centrifugal force acting on an element of the stream and, therefore, on suspended droplets is of the order of 75,000 G (75,000 times the force of gravity). Also, the stream, because it is moving radially inward in the rotor, is decelerated in the rotor from this tangential velocity of 304.8 m/s (1000 ft/s) down to zero tangential velocity in about a half a revolution of the rotor. This deceleration force amounts to something like 10,000 G. The vector sum of these two forces, therefore, amounts to 75,000 or 100,000 G in a direction 5 to 15° from the radial direction (see Fig. 29-48). This is an acceptable direction for the blades to lie, so the problem of avoiding bombardment of the blades by the droplets is solved by shaping the expander rotor blades parallel to this resultant vector. Then there is no force causing droplets to drift in the direction of any surface. They do drift back upstream slightly but nevertheless are carried on through by the mainstream and discharged. By this method any amount of condensing liquid can pass through, or, in the case of flashing liquids being expanded, the bubbles can pass through without efficiency loss. It would be impossible to construct a turbine blade meeting this requirement without two vector forces.

APPLICATIONS

The important uses of turboexpanders are in:

1. Air separation
2. Recovery of condensables from natural gas
3. Liquefaction of gases, including helium

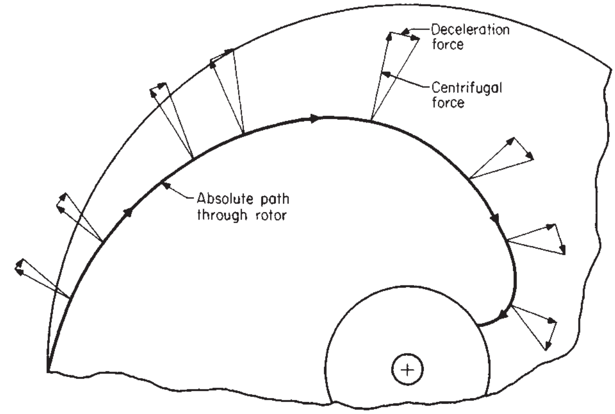


FIG. 29-48 Elemental path radially inward, then axially, through the rotor.

4. Augmenting refrigeration in various cryogenic processes such as the recovery of ethylene
5. Power generation, sometimes referred to as power recovery

A potential application for the turboexpander for power recovery exists whenever a large flow of gas is reduced from a high pressure to some lower pressure or when high-temperature process streams (waste heat) are available to boil a secondary liquid. When such conditions exist, they should be examined to see if use of a turboexpander is justified. In such cases a turboexpander can be used to drive a pump, compressor, or electric generator, thus recovering a large portion of otherwise wasted energy. In applications of this type, careful consideration should be given to the temperature drop which will occur in the expander. It may sometimes be necessary to heat or to dry the inlet gas to avoid low exhaust temperatures that cause the formation of ice or liquids.

Expanders are used because they are in an advanced state of development and reliability, attain high efficiency, and are relatively inexpensive.

LUBRICATION

Expander bearings are usually high-speed, and they should have full film lubrication. This is best assured by using force-feed lubrication at a pressure of the order of 689.5 kPa (100 lbf/in^2) or more. There is no special objection to using pressures as high as 6.895 MPa (1000 lbf/in^2) or higher, if for some reason it is desirable to do so.

Usually, a journal bearing and a thrust bearing are combined in one assembly, and oil is injected so as to feed both of them. The rate of flow usually is adjusted so as to carry the heat away with a temperature rise of the order of 11 to 17° C (20 to 30° F).

The smallest expanders usually use oil with a viscosity at 38° C (100° F) of 60 to 100 SSU, and large machines up to 500 SSU. If the oil is kept in a totally enclosed system in contact with hydrocarbon or another partly soluble gas, which would dissolve and reduce the viscosity of the oil, then a compensating higher viscosity should be used so that the working viscosity after ultimate equilibrium with such gas is suitable for the bearings.

The lubrication system, for reliability reasons, usually has an operating and a standby pump and dual switchable filters. If there is a cooling-water scaling problem, coolers may also be switchable.

BUFFER-GAS SYSTEM

The shaft seal (see subsection "Shaft Seals") generally is a close-clearance labyrinth-type seal. It is desirable that there be available a suitable pressurized buffer gas for injection into the intermediate point in the seal, such gas to be available at an absolute pressure well above the highest shaft pressure to be sealed. Then the seal-gas system may consist of only a filter, a flow-indicating device, and a throttle valve or other flow or pressure control, usually a pressure regulator and a graduated needle valve.

If the available pressure is not far above that of the pressure to be sealed, then with simple throttling the flow may be insufficient when the two pressures come too near together. Then more precise control, such as by differential pressure between the process side and the seal-gas pressure, may be required.

SIZE SELECTION

Size, rotating speed, and efficiency correlate well with the available isentropic head, the volumetric flow at discharge, and the expansion ratio across the turboexpander. The head and the volumetric flow and rotating speed are correlated by the specific speed. Figure 29-49 shows the efficiency at various specific speeds for various sizes of rotor. This figure presumes the expansion ratio to be less than 4:1. Above 4:1, certain supersonic losses come into the picture and there is an additional correction on efficiency, as shown in Fig. 29-50.

The available isentropic head is usually calculated by computer, using any of the various equations of state. In the absence of such facility, a quick and reasonably reliable calculation follows. In fact, this calculation is valuable as a cross-check on other methods because it is likely to be accurate within a few percent.

R = gas constant, 1.986 $\quad T_a$ = average temperature, °R

Z_a = compressibility at T_a $\quad H_i$ = isenthalpic work,

$$H_i = \frac{RT_aZ_a}{\text{mol. wt.}} \ln \frac{P_1}{P_2} \qquad \text{Btu/lb} \tag{29-22}$$

where H_i = reversible incremental enthalpy drop, Btu/lb
R = gas constant, 1.986, Btu/(lb·mol·°R)
T_a = average temperature for the increment, °R
Z_a = average compressibility for the increment
P_1/P_2 = pressure ratio

The use of this equation requires that an average P and T, based on an assumed increment, be used to find Z.

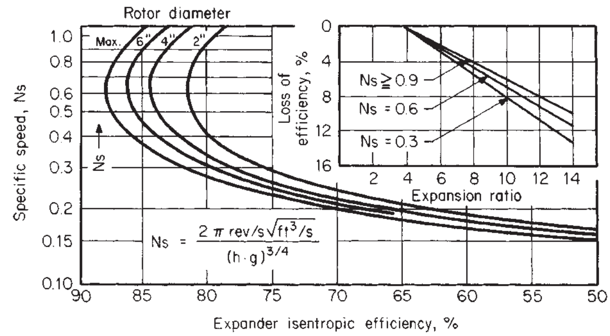


FIG. 29-49 Efficiency at various specific speeds for various sizes of rotor.

FIG. 29-50 Loss of efficiency as a function of the pressure ratio.

INSTRUMENTATION

Process-flow control and buffer-gas control have been discussed under "Variable Nozzles" and "Buffer-Gas System" respectively. Speed is usually self-controlled by a matching speed-sensitive load such as a compressor or a pump. If the load is an induction or synchronous generator feeding into a stable ac system, the system frequency fixes the speed. Otherwise, the speed can be controlled by a conventional governor.

Various protective instruments are used to provide a shutdown signal (to a fast-acting trip valve at the expander inlet) that senses various things, such as overspeed, lubricant pressure, bearing temperature, lubricant temperature, shaft runout, icing, lubricant level, thrust-bearing load, and process variables such as sensitive temperatures, levels, pressures, etc. However, too many safety shutdown devices may lead to excessive nuisance shutdowns.

POWER RECOVERY FROM LIQUID STREAMS

BASIC PRINCIPLES

The potential for power recovery from liquid streams exists whenever a liquid flows from a high-pressure source to one of lower pressure in such a manner that throttling to dissipate pressure occurs. Such throttling represents a system potential for power that is the reverse of a pump—in other words, a potential for power extraction. Just as in a pump, there exists a hydraulic horsepower and a brake horsepower, except that in the recovery they are generated or available horsepowers.

Basically, power recovery from liquids is achieved in industrial installation as shaft horsepower. While this potentially could appear either as reciprocating or as rotating power, most larger applications are rotating. Consideration of power recovery from liquids involves a choice among several possible uses, and usually this choice involves as alternatives (1) driving of a few large-horsepower services versus driving of more smaller-horsepower services; (2) driving of essential versus nonessential services or of spared versus nonspared services; (3) driving as sole driver versus partial driver or full horsepower versus partial horsepower for the selected service; and (4) converting power-recovery energy to some other intermediate energy form, as by driving an electric generator.

In applying power recovery, three basic problems are (1) limitations in designing equipment to recover the power, (2) operating reluctance to consider rotating equipment that is not absolutely necessary, and (3) the way in which the economics of the installed system is evaluated. It is important to recognize that there has always been an operable, acceptable alternative to power recovery from liquid streams in the form of the throttling or letdown valve, whereas no such simple, cheap, foolproof substitute exists for the pump.

Basic to establishing whether power recovery is even feasible, let alone economical, are considerations of the flowing-fluid capacity available, the differential pressure available for the power recovery, and corrosive or erosive properties of the fluid stream. A further important consideration in feasibility and economics is the probable physical location, with respect to each other, of fluid source, power-production point, and final fluid destination. In general, the tendency has been to locate the power-recovery driver and its driven unit where dictated by the driven-unit requirement and pipe the power-recovery fluid to and away from the driver. While early installations were in noncorrosive, nonerosive services such as rich-hydrocarbon absorption oil, the trend has been to put units into mildly severe services such as amine plants, hot-carbonate units, and hydrocracker letdown.

Economics Power-recovery units have no operating costs; in essence, the energy is available free. Furthermore, there is no incremental capital cost for energy supply. Incremental installed energy-system costs for a steam-turbine driver and supply system amount to about \$800 per kilowatt, and the incremental cost of an electric-motor driver plus supply system is about \$80 per kilowatt. By contrast, even the highest-inlet-pressure, largest-flow power-recovery machines will seldom have an equipment cost of more than \$140 per kilowatt, and costs frequently are as low as \$64 per kilowatt. However, at bare driver costs (not including power supply) of \$64 to \$140 per kilowatt for the power-recovery driver versus about \$30 to \$80 per kilowatt for steam turbines or \$50 to \$64 per kilowatt for electric motors, operating costs must be considered to make power-recovery units attractive. Using commonly accepted values for power costs, turbine steam rates, and steam selling prices, operating costs for either motors or steam turbines approximate \$280 per year for 746 W (1 hp).

Thus, barring technical difficulties or operational considerations in application, power-recovery units ought to show payouts of less than 6 months. Actual project payouts run from 1 to 3 years. This difference is due principally to (1) the fact that while the incremental costs just presented are valid in comparing large systems, specific designs encounter frame-size breaks, standardized capacities and horsepowers, code requirements, etc.; and (2) operating requirements, sparing considerations, and a certain lack of confidence in power-recovery units stemming from lack of extensive experience produce equipment-selection schemes that deviate from the straightforward comparison.

Development The following discussion relates specifically to the use of what could be called radial-inflow, centrifugal-pump power-recovery turbines. It does not apply to the type of unit nurtured by the hydroelectric industry for the large-horsepower, large-flow, low- to medium-pressure differential area of hydraulic water turbines of the Pelton or Francis runner type. There seems to have been little direct transfer of design concepts between these two fields; the major manufacturers in the hydroelectric field have thus far made no effort to sell to the process industries, and the physical arrangement of their units, developed from the requirements of the hydroelectric field, is not suitable to most process-plant applications.

Despite a rather slow start, centrifugal power-recovery pump-turbines have built a respectable record of process-plant installations extending back to the middle 1950s. Applications have included drives for the following services: cooling-tower fans; reciprocating recycle compressors; gas-treating-solution circulation pumps, as sole drive and as tandem with a steam turbine or motor helper; refinery-unit charge-stock pumps with a helper driver; and floating-online electric generators.

In general, early experiences were in the small-horsepower, nonessential or spared services, using sole drivers at full horsepower. The present trend is more and more toward the few large drivers in essential services, usually supplying only partial horsepower but not spared. If a plant is based on electric drivers and the economics of rate structure, demand charges, etc., permits, the use of a power-recovery unit driving a generator electrically floating on the line has found increasing favor. In general, operating experience in regard to reliability, serviceability, and maintainability has shown the units to be comparable with centrifugal pumps and has resulted in increasing acceptance even as drivers in large-horsepower units on essential service equipment. Presently accepted industrial limits are shown in Fig. 29-51.

Hydraulic Behavior The basic hydraulic behavior of centrifugal pumps operating as power-recovery units (turbines) is not much different from that of centrifugal pumps and follows the same sort of affinity laws over narrow ranges. Typical generalized curves are shown in Figs. 29-52 and 29-53. Note particularly that both torque and horsepower go negative (turn to values indicating power consumption) when head and capacity are within a fairly wide range representing at least startup and shutdown conditions if not also part load. Note also that even if head goes to 125 percent of design, speed (r/min) at zero torque for this unit does not exceed about 130 to 150 percent of design.

Tests conducted to operate centrifugal pumps as hydraulic turbines throughout the head-capacity-speed range show that a good centrifugal pump generally makes an efficient hydraulic turbine. From theoretical considerations it is possible to state that at the same speed

$$H_t = H_p/e_h^2 \tag{29-23}$$

$$Q_t = Q_p/e_h \tag{29-24}$$

$$n_{st} = n_{sp}e_h \tag{29-25}$$

where H = total head at best efficiency point
Q = capacity
n_s = specific speed
e_h = hydraulic efficiency, taken as the same for the turbine and the pump
t, p = subscripts denoting turbine and pump respectively

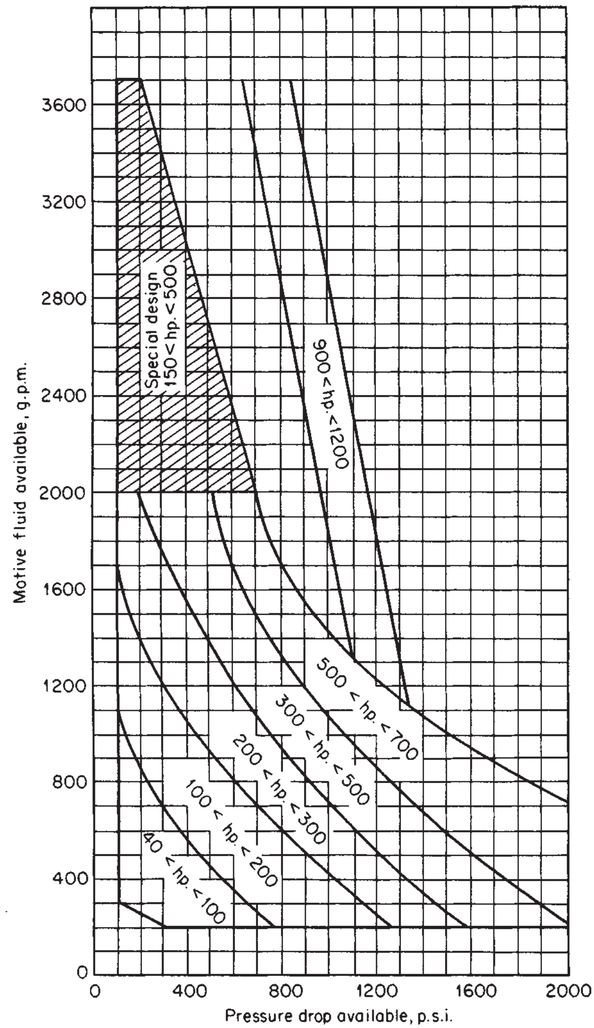


FIG. 29-51 Application areas for centrifugal pump turbines. Curves apply between the following minimum and maximum limits: inlet pressure, 100 to 3000 psig; pressure differential, 100 to 2800 lbf/in^2; flow of motive fluid, 200 to 4000 gal/min; horsepower, 50 to 3000 hp. Curve horsepower is based on a speed of 3600 r/min and a fluid of 1.0 specific gravity; for other fluids, multiply the curve horsepower by specific gravity to get the actual horsepower. To convert pounds-force per square inch to megapascals, multiply by 6.89×10^{-3}; to convert gallons per minute to cubic meters per minute, multiply by 3.79×10^{-3}; and to convert horsepower to kilowatts, multiply by 0.746.

Since the exact value of the **hydraulic efficiency** e_h is never known, $\sqrt{e}$ can be taken as an approximation where e is the gross (hydraulic horsepower/brake horsepower) pump efficiency. Efficiencies of pump designs running as turbines are usually 5 to 10 efficiency points lower than those as pumps at the best efficiency point.

OPERATING BEHAVIOR

By considering the flow as stopped but the turbine casing full of liquid, it is intuitively obvious that to rotate the wheel or impeller in either direction power will have to be put in. As the flow increases

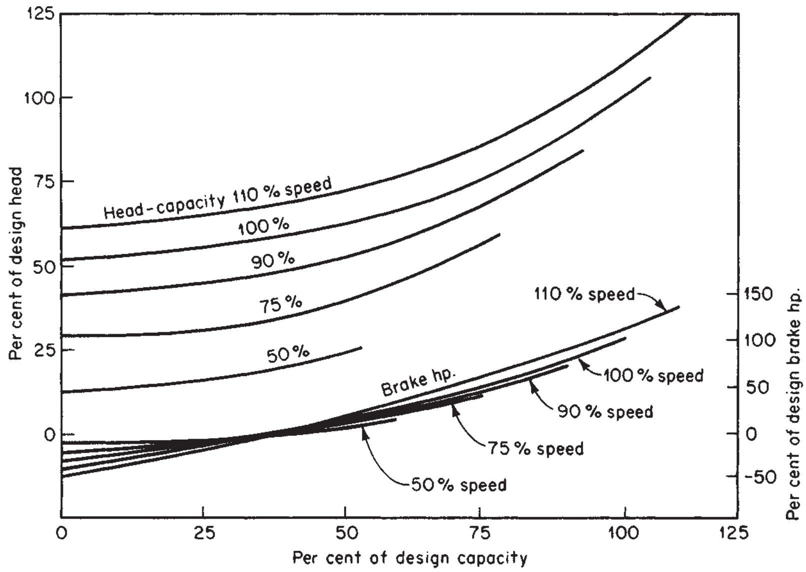


FIG. 29-52 Generalized curves showing hydraulic behavior of centrifugal pumps operating as power-recovery turbines.

from the no-flow conditions, the fluid velocity through the wheel gradually approaches such a rate that it imparts enough energy to the wheel not only to overcome internal friction but also to permit some net power output for consumption; this point usually occurs at about 30 to 40 percent of design flow or capacity. As in any turbine driver, the machine will speed up until the load imposed on the shaft coupling by the driven unit equals the power entering from the power-recovery wheel. Like a pump, a power-recovery unit will ride its characteristic curve and seek a point at which its particular head-capacity-speed-power-output relationship is satisfied. In most applications, the head available to the unit, being largely composed of a static-pressure difference, is nearly constant and varies only to the extent that inlet and exit piping-friction losses vary with flow through the unit. Thus the unit finally acts as an orifice in a relatively fixed differential system, meaning that it has a definite flow limit which also produces a torque and horsepower limit. This can be seen by following the 100 percent head curve of Fig. 29-52.

Performance Characteristics Performance of the power-recovery unit **operating as the sole driver** (Fig. 29-54) is shown in Fig. 29-55. If it is assumed that more liquid at the available head is presented to the power-recovery unit than is needed to generate the horsepower required by the pump, the turbine unit will speed up to handle the liquid, and at the same time the pump speed must go up. In speeding up, the turbine will generate more horsepower, which the pump must absorb while it is at the new speed. Finally, a balance point on horsepower is reached with the driven unit, but the number of revolutions per minute may be off design. If speed control of the driven unit is necessary, throttling some of the available capacity across a valve bypassing the unit permits the unit to satisfy its horsepower-capacity-speed relationship at the desired number of revolutions. A similar problem occurs when the capacity available to the unit is less than that needed at available head and design revolutions. The unit will slow down, try shedding load, and attempt to come to peace with its head-capacity-speed curve sets. Here speed control can be achieved by throttling the available pressure so that the unit sees only that portion of the available head needed to satisfy its head-capacity-speed relationship at the desired number of revolutions.

Performance of the power-recovery unit **operating with a makeup driver** (Fig. 29-56) is shown in Fig. 29-57; specific percentage values are shown, but the general characteristics and curve shapes are typical. It should be noted that the flow scheme, the selection of equipment, and the design of that equipment have produced the relatively inflexible system pattern shown in the curves, in which (1) except at a single point the recovery unit always requires either flow bypassing or inlet-pressure throttling (see bottommost curve); and (2) the horsepower output of the recovery unit is reduced at any point away from design (note the horsepower-difference curve), which, combined with the characteristics of steam turbines, produces the unusual turbine throttle steam-flow curve (second from the top in Fig. 29-57).

DESIGN CONSIDERATIONS

Involved in producing the curves for Figs. 29-53 and 29-55 is a calculation of the so-called **balance point** at which the flow and revolutions per minute required by the recovery unit match those provided by the pump. If the recovery turbine is the sole driver (as for the lean pump of Fig. 29-54), both the speed and the brake horsepower of the recovery turbine and its driven pump must be the same at the so-called balance point. If there is a makeup driver and the recovery unit has available to it just the flow from the pump that it is driving, as for the pump of Fig. 29-56, then the speed and capacity must match at the balance point.

Example 2: Units for a Power Recovery System The scheme of Fig. 29-52 and the actual units supplied (Figs. 29-58 and 29-59) will be used for purposes of illustration. Since this case has the recovery unit as the sole driver, the balance point is set by speed and horsepower.

For purposes of example, assume a flow of 8.71 m^3/min (2300 gal/min) through the tower. The maximum head available to the recovery turbine was calculated to be 604 m (1982 ft); this value will be slightly in error when part of the flow is bypassed since frictional losses into and out of the recovery unit will change. First, assume the lean pump to be at 3.03 m^3/min (800 gal/min) running at 3900 r/min with the semilean pump at 5.68 m^3/min (1500 gal/min) to get the total flow of 8.71 m^3/min (2300 gal/min). At 3.03 m^3/min (800 gal/min) and 3900 r/min the available head of the lean pump is read from the curve. This must be greater than the required head, and the excess is plotted as in Fig. 29-60. The brake horsepower of the lean pump is also read.

Now, at 3900 r/min and a head of 6.04 m (1982 ft), the required flow and generated brake horsepower of the recovery turbine are read. Since the horsepower of the lean pump and the recovery turbine are not identical, this entire process is repeated at another speed with the 3.03 m^3/min (800 gal/min). The difference in brake horsepower between the lean pump and the recovery turbine is then plotted against the speed for these two points, and a line is drawn between

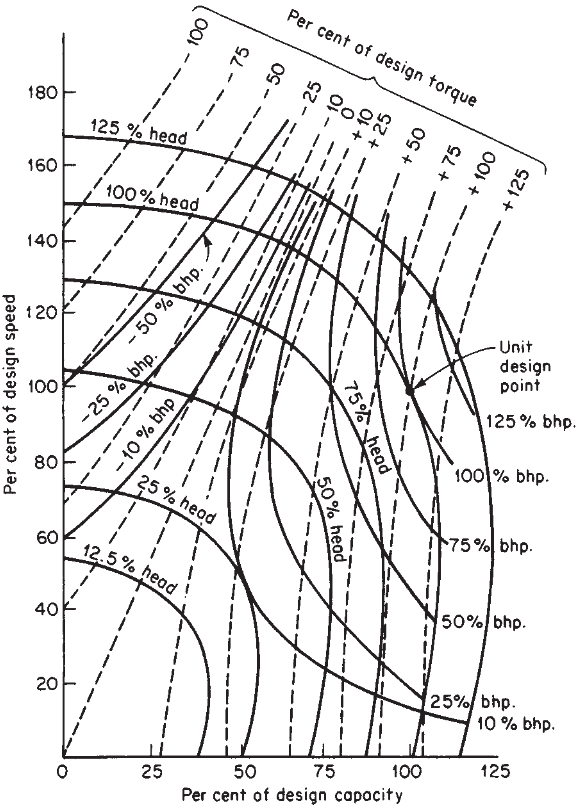


FIG. 29-53 Generalized curves for centrifugal pumps operating as power-recovery turbines.

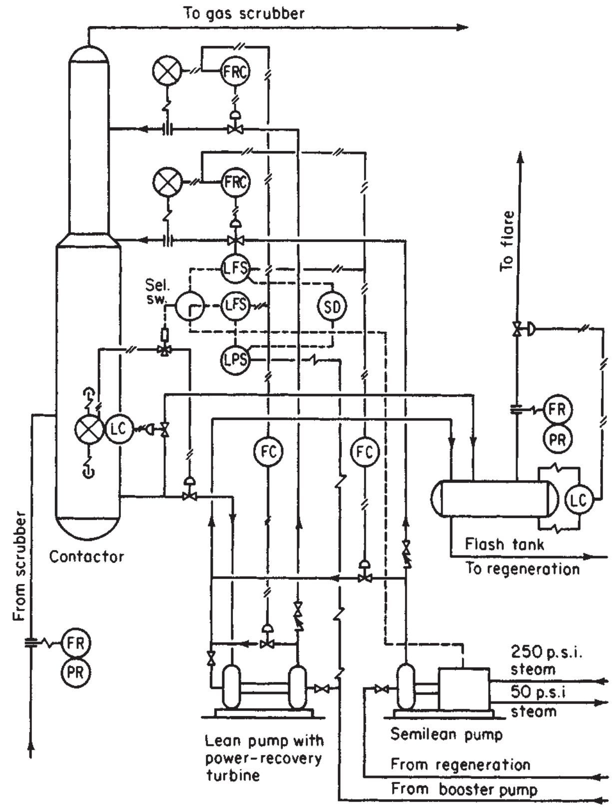


FIG. 29-54 Flow diagram of a power-recovery unit operating as the sole driver.

them. Where this line crosses the zero-difference brake-horsepower line is the balance point at 8.71 m^3/min (2300 gal/min) through the tower and 3.03 m^3/min (800 gal/min) through the lean pump.

The same procedure may be used at other pump flows to permit plotting the series of balance-point curves as has been done in Fig. 29-61. From such curves, one can establish the maximum lean pump at any total tower outflow, and combining this with the semilean-pump performance curve results in Fig. 29-55. Bypass flow plotted in Fig. 29-55 is obtained by adding simultaneous lean- and semilean-pump flows and subtracting the recovery pump-turbine flow required to make the balance point at that lean-pump flow.

Design Bases It is apparent that the balance point is always determined by the power r/min characteristics of the driven unit as sensed at the coupling by the shaft of the power-recovery pump-turbine. If the driven unit can simply soak up any (all) of the generated horsepower, for instance, a floating electric generator, then capacity control and pressure throttling may not be needed. When a speed-controlling variable-horsepower unit such as an electric motor or a steam turbine provides a tandem helper or a makeup driver, these units will hold revolutions per minute constant and make up just enough horsepower to permit the power-recovery pump-turbine to satisfy its head-capacity curve at virtually any flow rate.

It is the **combined unit characteristics** which must be considered, and these characteristics must be evaluated over the full operating range as well as for the startup condition. The consumption of power, up to almost 40 percent of design output, on starting up and coming up to speed and the fact that under a relatively fixed head condition the maximum speed at zero torque is about 140 percent of design have both already been noted. These, of course, bear particular significance for starting the unit up or shutting it down and must be considered. Failure to perform a complete system analysis can frequently lead to a process design that proves, upon installation, to be an operating trap.

In realizing the advantages of competitive designs for hydraulic power-recovery systems, there is usually an investment premium which must be paid for the **operating flexibility** illustrated in Fig. 29-55. This takes the form of additional reduced-capacity pumps and steam-turbine drivers, as well as some sacrifice in power recovery that results from bypassing or throttling even at the design point. If investment is to be minimized, a tightly designed system for full power recovery with a single pump-turbine helper, as in Fig. 29-54, may be worth considering. In such systems, the only fluid available to the pump-turbine is that provided by the pump it drives, and no separate pump with auxiliary driver is available for startup. Operating personnel will immediately see that such systems are relatively inflexible and difficult to operate (Fig. 29-57). Thus, while the tightly designed system has a minimum investment and more power recovery, it may not be the most desirable if operation away from the design point is anticipated, since only at that one point is neither throttling nor bypassing needed. Furthermore, it becomes apparent that the maximum steam requirements may be set by a partial-load condition rather than by design conditions or overload.

While the foregoing examples have dealt with applications on centrifugal pumps, the same sort of analysis can be made for reciprocating pumps, reciprocating compressors, or other rotary users like cooling-tower fans.

INSTALLATION FEATURES

In addition to performing the system analysis, a number of details or peculiarities of the units must be considered with respect to:

1. Vaporization, flashing, or cavitation
2. Fluid volumes
3. Process-stream controls
4. Speed control

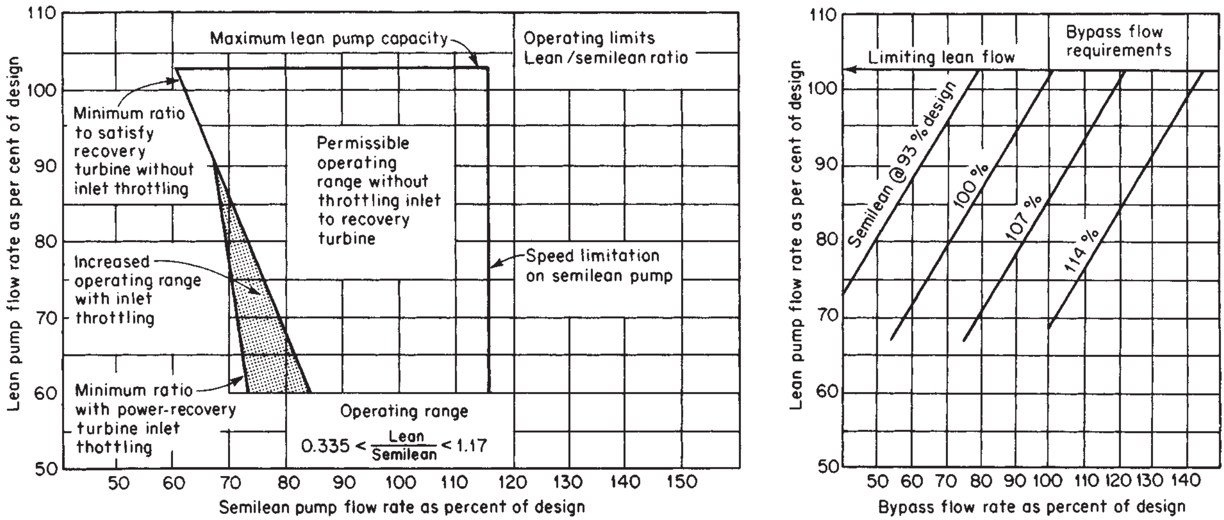


FIG. 29-55 Performance of a power-recovery unit operating as the sole driver.

5. Startup and overcapacity
6. Electrical-system characteristics if the recovered power is used to generate electricity

Vaporizing Fluids Many pump-turbines are installed on gas-saturated liquid streams, and loss in pressure can cause problems whenever this occurs across balancing drums or pressure-reducing labyrinth seals. Piping that carries bleed streams from the drums and seals to the low-pressure (outlet) side of the pump-turbine must be sized generously to allow for some gas evolution.

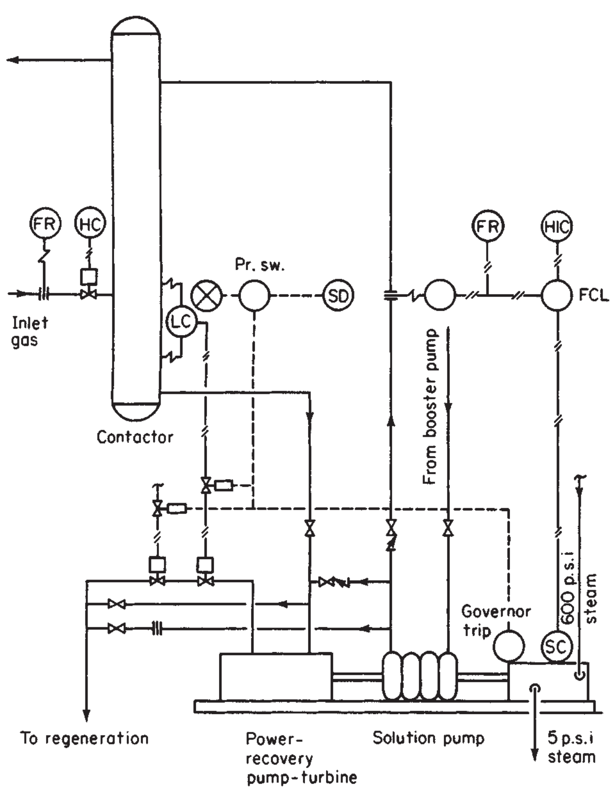


FIG. 29-56 Flow diagram of a system with a power-recovery turbine operating with a makeup driver.

In general, gas evolution is not evident in the pump-turbine's casing proper, for the corresponding added horsepower (as would be anticipated from the gas expansion) has never been evidenced in reports on field testing. It appears that the liquid passage through the casing is too fast for vapor-liquid equilibrium to be attained. However, slower shearing passage through a balancing drum and return line does permit gas evolution, and this line may become vapor-locked if it is undersized. Similarly, the high points of the pump-turbine casing may be vapor-locked with released gas if the unit stands idle; this can cause damage to seal chamber, balancing-drum chambers, etc., when the unit is started up again.

There is another potential hazard due to vaporization which does not generally occur in process-plant installations. The hazard results from the fact that a pump sees only the head of fluid, and if significant flashing occurs in the inlet piping, the head of fluid represented by the pounds-force-per-square-inch-gauge inlet pressure can be many times greater than design head, resulting in an attempt by the unit to increase its speed greatly.

There also appears to be a minimum outlet or impeller eye pressure below which cavitation and its attendant physical damage can occur (similar to net positive suction head, or NPSH, this could be called net positive discharge head, or NPDH). Thus it is often advisable to design by using only a part of the full pressure differential available in the process for the pump-turbine. If throttling is to be provided, outlet throttling is probably better than inlet throttling, and if used, any mechanical seals, as well as the unit casing outlet flange, must be adequate to withstand the full inlet pressure when the throttle valve is closed.

Fluid Volumes Many process-plant installations of these units are made by handling *rich liquid* out of an absorber. In most cases both liquid volume and liquid density will change from input to output. While such changes are not normally significant, the sensitivity of the balance point to the volume of flow makes it mandatory to consider volumetric swell by absorption in checking the suitability and adequacy of the pump-turbine unit and the controls and bypassing arrangements as part of a system analysis.

Process Controls If the inlet or outlet liquid to a pump-turbine is regulated by a level controller on the liquid-supply vessel, a falling liquid level inside the vessel will cause this controller to throttle a valve, reducing the differential pressure available to the pump-

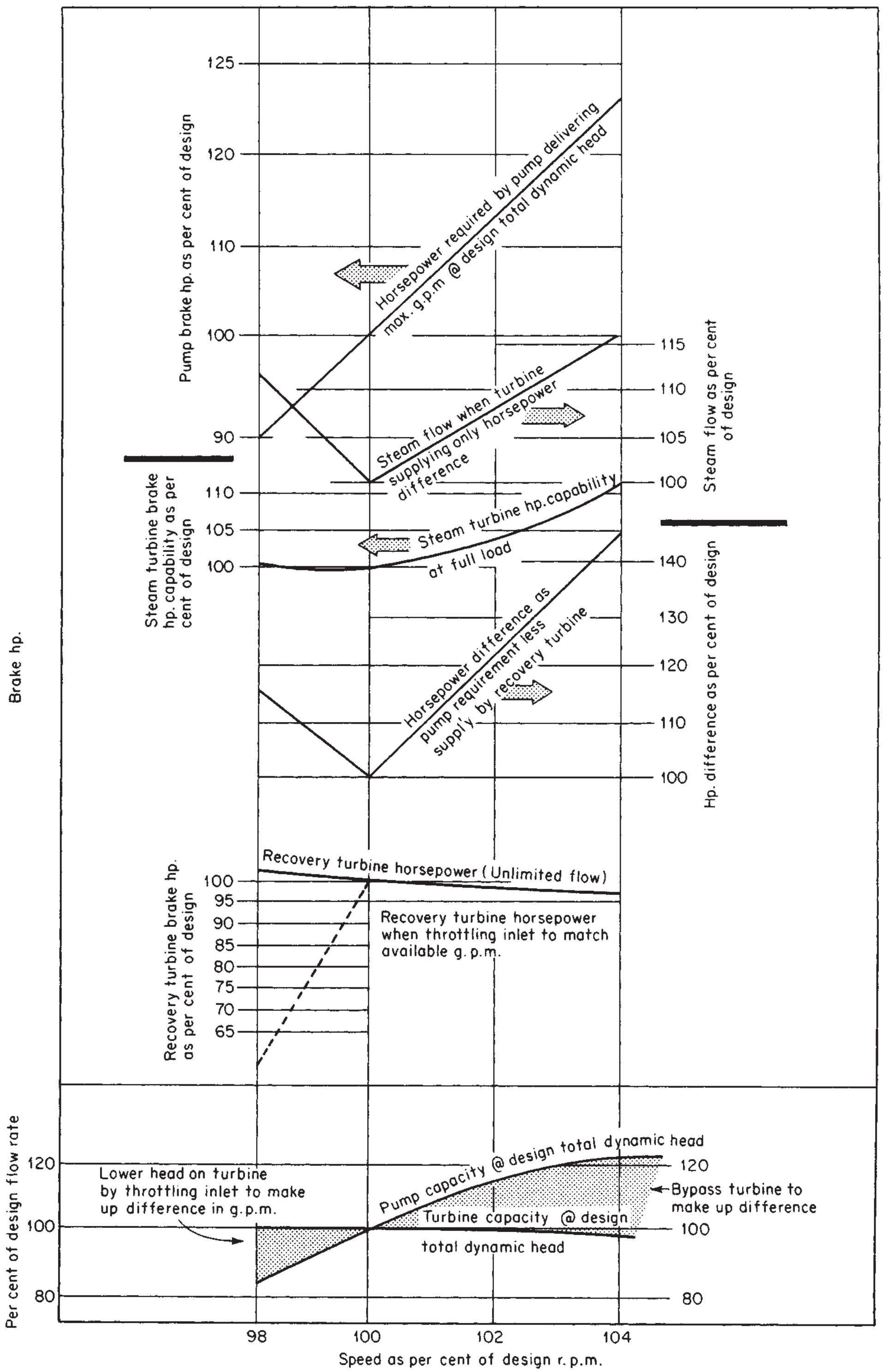


FIG. 29-57 Performance of a power-recovery turbine operating with a makeup driver.

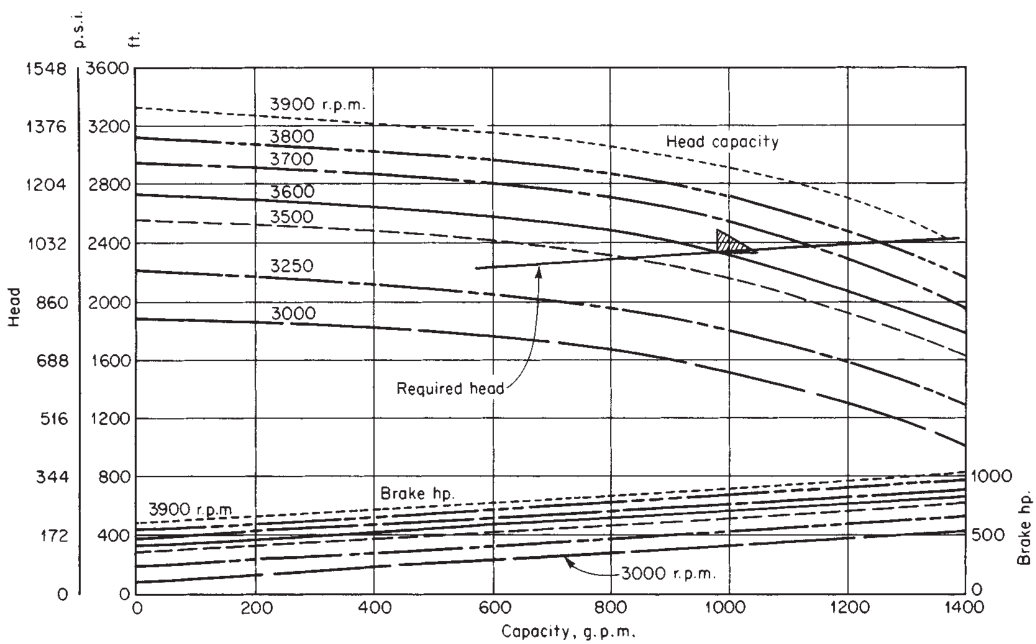


FIG. 29-58 Head-horsepower-capacity characteristics of a lean pump tandem-connected with a power-recovery turbine operating as the sole driver. To convert gallons per minute to cubic meters per minute, multiply by 3.79×10^{-3}; to convert horsepower to kilowatts, multiply by 0.746; and to convert pounds-force per square inch to megapascals, multiply by 6.89×10^{-3}.

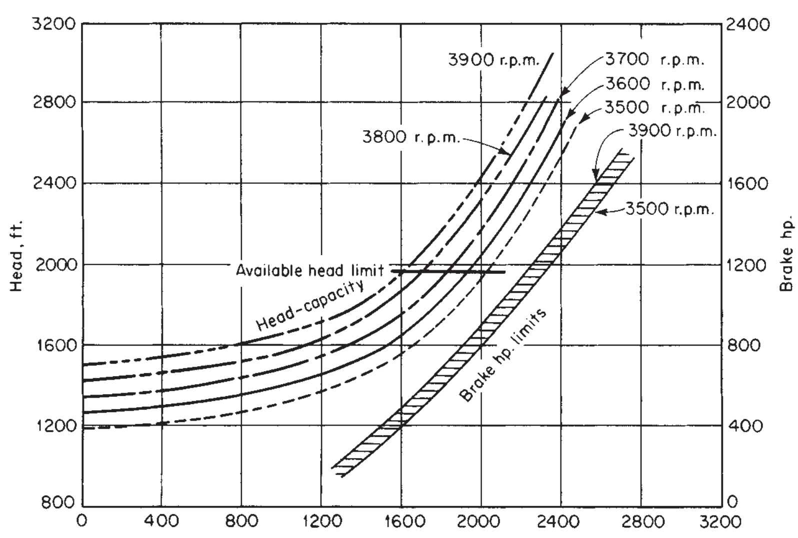


FIG. 29-59 Head-horsepower-capacity characteristics of a power-recovery turbine operating as the sole driver of a lean pump. If the total capacity of lean and semilean pumps exceeds the values indicated by "available head limit," bypass must be used. Net recovery-pump head at 8.71 m^3/min (2300 gal/min) is figured as follows:

Tower	957 lbf/in^2	6.598 MPa
Flash tank	–75	–0.517
Suction piping (62.4 lb/ft^3)		
Friction loss	–11.8	–0.081
Elevation	+2.8	+0.019
Discharge piping (57.8 lb/ft^3)		
Friction loss	–8.8	–0.061
Elevation	–5.6	–0.038

To convert gallons per minute to cubic meters per minute, multiply by 3.79×10^{-3}; to convert horsepower to kilowatts, multiply by 0.746; and to convert pounds-force per square inch to megapascals, multiply by 6.89×10^{-3}.

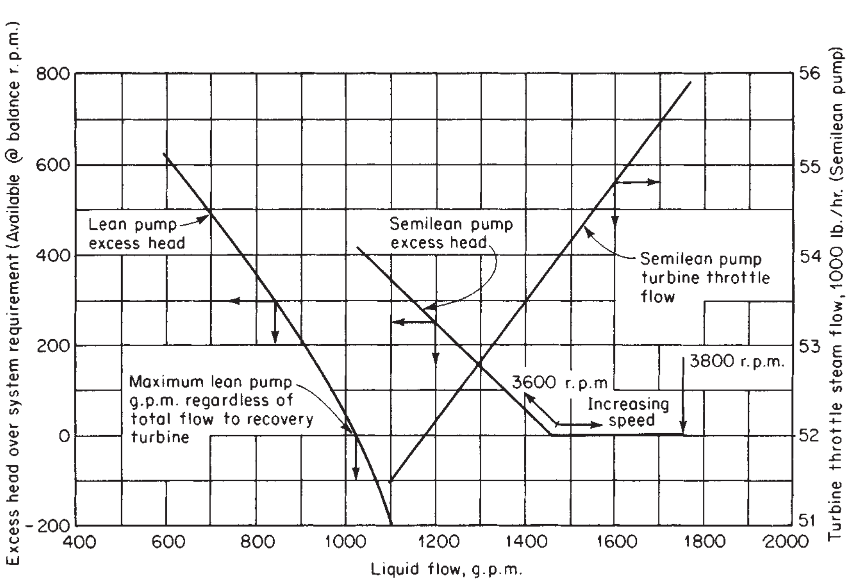


FIG. 29-60 Excess head developed by lean and semilean pumps and the steam-throttle flow for a semilean-pump turbine. To convert gallons per minute to cubic meters per minute, multiply by 0.00379; to convert pounds per hour to kilograms per second, multiply by 1.260×10^{-4}.

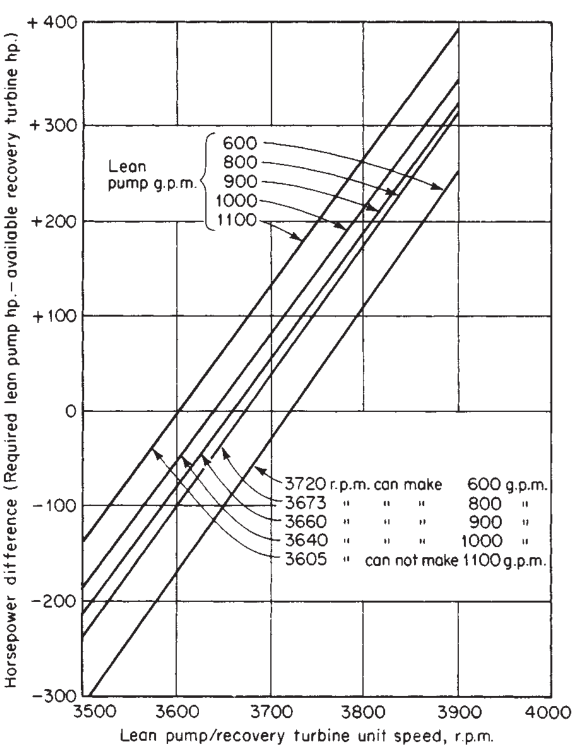


FIG. 29-61 Horsepower–r/min balance for a lean pump tandem-connected with a power-recovery turbine operating as the sole driver. Horsepower differences are calculated from excess head requirements as typically shown in Fig. 29-60. To convert gallons per minute to cubic meters per hour, multiply by 0.2271; to convert horsepower to kilowatts, multiply by 0.746.

turbine; or if the liquid level is rising, the control valve tends to open wide, so that the pump-turbine sees its full available head. Since under this latter condition the head is at its maximum, no more liquid will flow through the pump, and if the level continues to rise, the system goes off level control. One performance like this inevitably leads to a request from operating personnel for a bypass around the pump-turbine. On a startup it is frequently necessary to have a bypass from a point upstream of the driven pump's discharge check and block valves to the associated pump-turbine.

Speed Speed control can be critical with pump-turbines. Considering the characteristics shown in Fig. 29-57 (and the curves on which it is based), a 4 percent change in speed, from 98 to 102 percent, produces a 33 percent change in pump flow, from 82 to 115 percent, and about a 22 percent horsepower change, from 118 to 100 to 122 percent. A NEMA Class A steam-turbine mechanical governor has about 10 percent steady-state regulation, so that for the 22 percent horsepower speed change one should not expect better than a 2 percent speed regulation. Since 4 percent is the total speed change being considered, it becomes apparent that a problem exists in governing the speed of the steam turbine. Since this unit is supposed to serve to hold pump speed by supplying horsepower as needed to get a match of pump and system curves at the desired flow rate, this is a serious concern. In this case the problem can be solved by eliminating the steam-turbine speed governor, leaving only the overspeed trip, and placing a control valve actuated directly by pump-flow measurement in the steam-supply line to the turbine.

In contrast to steam turbines, in which runaway overspeeding is always a problem, pump-turbines operating at design head go to zero torque at about 130 to 140 percent of design speed. Thus, overspeed protection may not be necessary if the pump-turbine can withstand 140 to 150 percent of design speed and it is the sole driver. When a steam-turbine helper is used, it should be provided with the usual overspeed trip-out mechanism.

Startup and Overcapacity From a design standpoint and also operationally, it is important to remember that pump-turbines not only do not generate power before they attain about 40 percent of design flow but actually consume power in decreasing amounts as the flow is increased from zero to 40 percent. This means that they should be brought up to operating speed as rapidly as possible. Another solution, and a more desirable one from an operating and maintenance standpoint, is to install a free-wheeling or overriding clutch (such as that made by the Marland Division of Zurn Industries, Inc., of La Grange, Illinois) between the turbine and the pump. With such an installation, the pump does not have to turn until fluid is available to it. Also, it is not connected to the pump until it tries running faster than the pump, at which time it is putting out power. Under this arrangement the startup sequence can be selected so that the turbine unit goes from zero speed to operating speed along the zero-torque curve.

At design head, on the other hand, capacity does not change markedly with speed, so that once the design point has been passed the pump-turbine acts as a restriction in the line. Since most of these units operate on a relatively fixed pressure differential, they then tend to act like an orifice to limit flow, and little or no benefit can be realized from any overcapacity in terms of fluid flow available to the unit in the actual installation.

Electrical Generation When pump-turbines are used to generate electricity, the units should be tied into electrically strong networks of such a size that the pump-turbines cannot swing the system frequency but are governed by that frequency and become constant-speed machines. Inlet or outlet throttling, as well as bypassing, must be used for such installations. Consequently, with speed fixed and available head essentially constant because of the process, the maximum amount of recoverable power is established by the design. Once installed, a pump-turbine cannot be pushed into generating more power if more fluid is available except by a redesign of its internals. The electrical portion of such an installation is straightforward, controls are simply inlet throttling and a bypass (if needed) for the pump-turbine, and these controls can be operated on a split range from the same level controller on the liquid-source vessel.

Integral Units A relatively recent development is the integral or packaged pump-turbine unit assembled in a single casing, having a common discharge port and designed for mounting directly in the piping. A cutaway view of one such design is shown in Fig. 29-62. It is obvious that the use of such a unit presupposes the compatibility and desirability of having the pump discharge material contaminated by all the powering fluid which passed through the driving turbine impeller. So far, this unit has seen little service in process plants, but its simplicity, its independence of normal power sources, and its ease of installation make it an attractive candidate for consideration when the process-flow-scheme can be adapted to its characteristics. It has been proposed for use in place of liquid eductors or ejectors, for it has a much higher efficiency than such units, and furthermore throttling motive fluid flow will cause the pumped fluid flow to follow according to balance-point requirements.

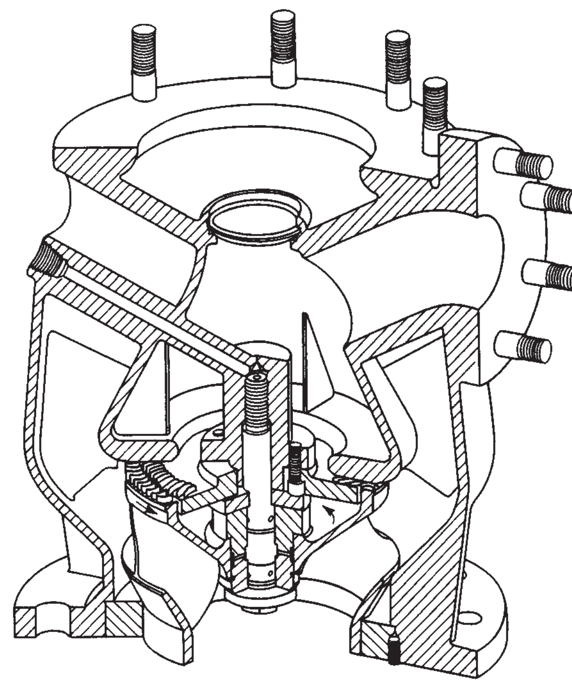


FIG. 29-62 Integral or packaged pump-turbine unit. (*Worthington Division, Dresser-Rand Co.*)

MECHANICAL POWER TRANSMISSION

Process machinery usually involves a driver and a driven machine, and these machines are connected by power-transmission equipment. Such equipment can either be mechanical or hydraulic.

For instance, power can be transmitted from one machine to the other through shafts, flexible couplings, and gear reducers (mechanical equipment). Power can be transmitted through a torque converter (hydraulic equipment) or by a combination of mechanical and hydraulic equipment.

Although power-transmission equipment is generally simpler than the machines it connects, it nevertheless fails in service as often as—if not more often than—motors, turbines, pumps, and the like. Care must be given to the proper selection and maintenance of power-transmission equipment; otherwise, the best process machines cannot perform as expected.

This segment will discuss the more commonly used power-transmission equipment; also discussed will be the machinery bearings, which not only support the rotors of machinery, but the power-transmission equipment as well.

The segment is divided into the following parts:

- Bearings
- Power transmission without speed change
- Power transmission with change in speed
- Lubrication of power transmission equipment

BEARINGS

Rotating shafts must be supported by the machine housing, not only against gravity, but also against a variety of forces that are imposed on rotors inside machines (including axial thrust). Bearings have a significant influence on the performance of process machines, both because they limit their continuous operation and also since they influence the level of vibration and critical speeds of the machines.

It was generally accepted that "machinery users do not select bearings" (see the sixth edition of this handbook); this is no longer valid: users have realized that they can have a voice in the selection of bearings of new machines and that they can select better bearings for their old machines than the ones supplied by the original equipment manufacturer (OEM). A number of independent bearing manufacturers can be found in most industrial areas.

Until quite recently, all types of bearings required lubrication; the advent of magnetic bearings has eliminated the need for lubrication. Lubricated bearings have a significant power loss, as oils or greases are sheared by the relative motions between shafts and housings; magnetic bearings eliminate the losses in lubricants, but they require electrical power to maintain the shaft journals in the desired position.

Three types of bearings will be discussed: oil-film bearings, rolling-element bearings (also known as antifriction bearings), and magnetic bearings.

Oil-Film Bearings The name *oil-film bearing* derives from the fact that, in such bearings, shafts are supported by a film of oil under significant pressure. This pressure is generated by the rotation of the shaft and by the fact that the clearance between the shaft and its bearing has a wedge shape. A typical radial oil-film bearing is illustrated in Fig. 29-63. The bearing has a diametrically split shell, having a layer of low-friction material (babbitt) on the inside surface. The outside diameter fits tightly in the machine housing, and an antirotation pin (or protrusion) is provided. This pin also serves the purpose of axially locating the bearing. Between the inside surface of the bearing and the shaft journal, there is a clearance filled with oil under a pressure of about 20 psi (1.4 bar, 138 kilopascals).

This clearance is very important for bearing and machine performance. A too-tight clearance will allow very little oil to flow through the bearing, which will operate hot; a too-loose clearance will prevent the formation of a high-pressure film of oil, and the bearing will fail through mechanical contact. It is customary to use the following clearances:

- For low to moderate speeds, $0.001 \times d + 0.002$ (inches)
  $0.001 \times d + 0.05$ (mm)
- For high speeds, $0.002 \times d$

where d is the journal diameter.

The clearance in oil-film bearings is necessary for the formation of a film of oil; however, it is detrimental because it allows shafts to vibrate within. Machinery vibrations tend to appear with wear and with the unbalance created by blade fouling. In many cases, machines can be made less sensitive to vibrations by changes in bearing design; there are types of oil-film bearings that resist shaft motions without decreasing the internal clearances. To understand how such bearings work, an explanation of *preload* is in order.

The normal operating position of a shaft inside a bearing is shown in Fig. 29-64. It can be seen that, due to radial forces, the geometric center of the shaft does not coincide with the one of the bearing. This displacement creates a "wedge," which combined with the shaft motion, forces the oil into a continuously decreasing area, and a

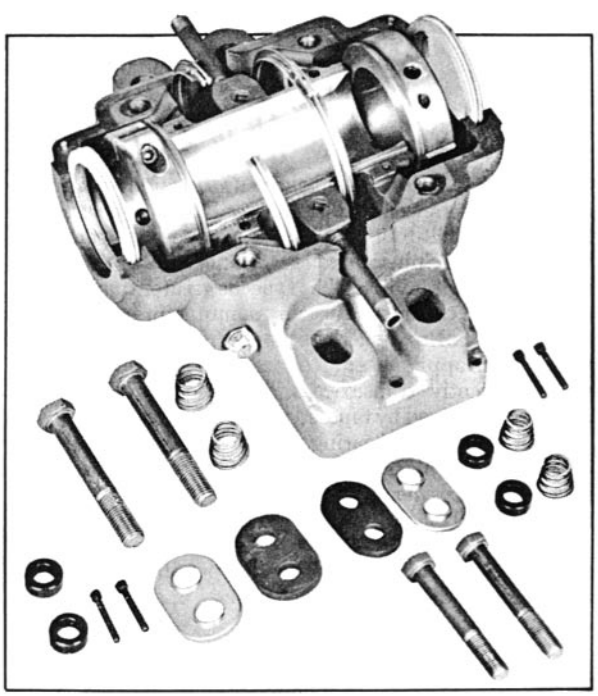


FIG. 29-63 Typical oil-film pillow-block bearing. Lubrication is provided both by an oil flow and by oil rings.

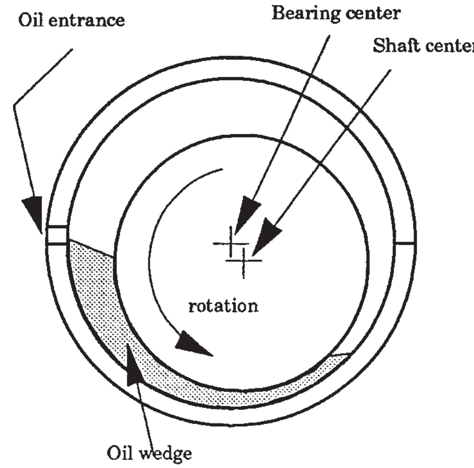


FIG. 29-64 Operation of oil-film radial bearings.

hydraulic pressure is generated. This pressure acts on the journal and supports it against gravity or other internal forces. To stabilize a vibrating shaft, bearings with larger oil pressures are used. A larger pressure is created whenever the shaft moves closer to the bearing (steeper wedge), which is not desirable. Therefore, an "apparent" proximity is created by altering the geometry of the bearing.

A lemon-bore bearing configuration is shown in Figure 29-65. It can be seen that a steep wedge was generated by *apparently* making the bearing diameter larger; this is similar to a normal bearing with a larger load; therefore, these types of bearings are known as preloaded bearings. *Preload* in this case does not refer to forces but rather to the resulting geometry.

Another way to stabilize a shaft inside a bearing is to use "pressure-dam" designs, as shown in Fig. 29-66. The oil supply pressure is directed to the top of the shaft; it forces the shaft downwards, and it stabilizes its motions.

The most modern oil-film bearings use movable pads, which tilt in order to create, automatically, the optimum wedge shape. They are known as the "tilting pad" bearings and are illustrated in Fig. 29-67.

Oil-film bearings are also used for positioning shafts axially, against thrusts created by the flow of the processed fluid through machines.

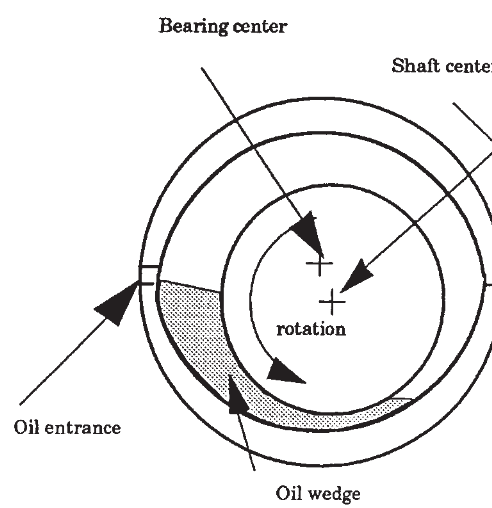


FIG. 29-65 Oil-film bearing with geometrically-created preload.

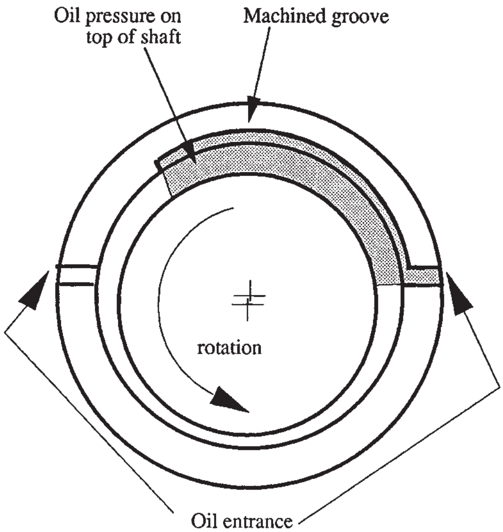


FIG. 29-66 Oil-film bearing with oil pressure created preload (pressure dam design).

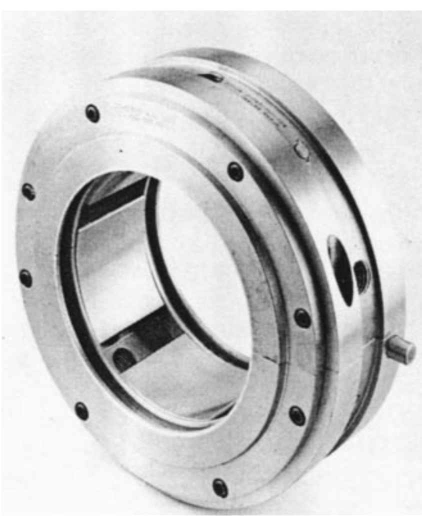


FIG. 29-67 Radial oil-film bearing with tilting pads.

Thrust bearings have an active side, against which the bearing collar or disc is forced; they also have an inactive side, which has the purpose of limiting the total axial float of a rotor, should the thrust reverse for any reason. Just as radial bearings require a clearance between the journal and the bearing, so do thrust bearings require a float of the thrust disc between the two sides of the bearing. A typical oil-film thrust bearing is illustrated in Fig. 29-68. Thrust bearings, just as radial bearings, can have a fixed geometry or can be equipped with tilting pads.

Rolling-Element Bearings With this type of bearing, shafts are supported by small parts that roll with no friction, at least theoretically. The rolling elements can be spherical (ball bearings), cylindrical (roller and needle bearings), or conical (tapered roller bearings). Although more complicated than fluid-film bearings, rolling-element bearings are less costly, mainly because they are manufactured in very large quantities. Their main disadvantage is a limitation in maximum operating speed. The main types of rolling-element bearings are shown in Fig. 29-69.

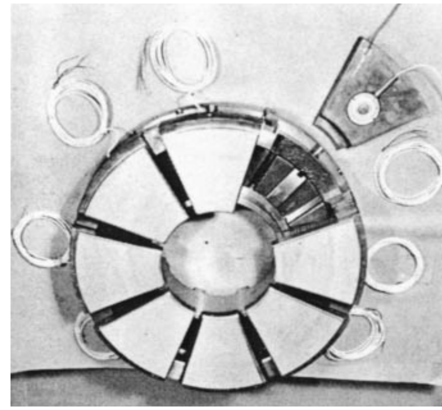


FIG. 29-68 Axial (thrust) oil-film bearing with tilting pads and embedded temperature sensors.

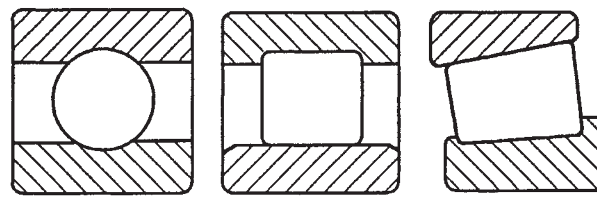


FIG. 29-69 Three types of rolling-element bearings: ball, cylindrical roller, and tapered roller.

Generally, the rolling elements ride on an internal race, installed on the shaft, and an external race, installed in the machine housing. The races and the rolling elements are made of hardened alloy steels, because the contact pressure caused by the radial (or thrust) forces can be very high.

The rolling elements are installed in a cage that performs the very important role of reducing the friction inside bearings. The conditions without a cage are shown in Fig. 29-70. It can be demonstrated that the rotational speed of a rolling element around its geometric center is:

$$\text{rpm}_1 = \frac{\text{rpm}_2}{2} \times \frac{R}{r}$$

For example, if the inner race diameter is 4 in and the balls have a diameter of ⅜ in, the balls will rotate 5.3 times faster than the shaft. If balls touch, then the relative velocity occurs at twice their revolution, and the friction so generated will overheat the bearing. Depending on the rated speed of a bearing, cages are made (in increasing order of speed) of steel, phenolics, or bronze. Brass and polymers are also used.

Rolling-element bearings require lubrication for minimizing the friction between the rolling elements and their cage and for dissipat-

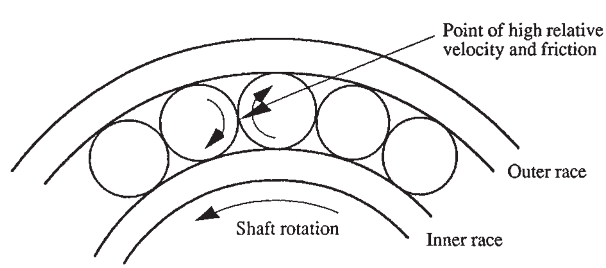


FIG. 29-70 Cageless roller bearings would operate hot and wear rapidly.

ing the heat generated by friction. Depending on the operating speed, lubrication can be provided by greases, liquid oil, or oil mist.

The selection of rolling-element bearings is based on the radial and axial loads they must support, on the operating speed, and on the expected life. All bearing catalogs provide comprehensive selection data.

Rolling-element bearings are provided with a small radial clearance that almost disappears when they are pressed on the shaft or in the housing. Basically, a rolling element bearing is pressed on its shaft and has a small clearance in the housing when the shaft rotates and the housing is stationary; the reverse is true when the housing rotates, as is the case for idler pulleys.

Magnetic Bearings Magnetic bearings replace the hydrodynamic shaft support by a magnetic field. Their advantages and disadvantages are summarized in Fig. 29-71. While oil-film bearings create the necessary power to support the shaft from the movement of the shaft, magnetic bearings need an outside power supply. A magnetic bearing system consists of four major components: magnetic actuators, power supply, shaft-positioning sensors, and electronic controls. Magnetic forces act all around the shaft; sensors detect the relative position between the shaft and the housing and send the signals to the controller, which supplies more or less electrical power to the electromagnets so that the shaft remains centered independent of the forces acting on it. The principle of operation of a magnetic bearing is shown in Fig. 29-72.

Because magnetic bearings do not require lubrication, they are particularly suited to applications such as canned pumps, vacuum pumps, turbo-expanders, and some centrifuges. As of 1994, some centrifugal compressors in the 5000-kW power range had been equipped with magnetic bearings.

The rotordynamic study of a machine with magnetic bearings is quite different from the one of either oil-film or rolling-element bearings. The stiffness and damping properties of magnetic bearings can be adjusted (within limits) by altering the feedback and response time between the sensors and the electromagnets.

Advantages	Disadvantages
Low power usage	Small load capacity
Very long life	Large size
No Oil required	Higher investment
Low *system* weight	Requires mechanical backup for power interruptions
Reduced fire hazard	New technology
Vibration sensors built-in	Possible rotor-dynamic problems

FIG. 29-71 Advantages and disadvantages of magnetic bearings.

POWER TRANSMISSION WITHOUT SPEED CHANGE

Whenever the process machine operates at the same speed as its driver, the two can be directly coupled. This direct coupling still allows for a variable speed, through adjustments of the speed of the driver. Steam turbine speed can be easily adjusted, and electric motor speed can also be varied by the use of special drives that vary the *frequency* of the power applied to the motor. Whether the speed is fixed or variable, direct coupling of two machine shafts presents the problem of accommodation of misalignment. To this purpose, machines are coupled through a *flexible coupling*.

Variable-Speed Electric Motor Drives Whether of the induction or synchronous type, ac motor speed is a direct function of the supply voltage frequency. The advent of high-power solid-state controllers made possible the manufacture of frequency converters. These converters can generate high power (over 14,000 kW) at a variable frequency, which can be lower or higher than the standard 60 Hz (cycles/second). Frequency converters made possible not only variable-speed motors but also high-speed (up to 10,000 rpm) electric motors. The cost of these converters is often higher than the cost of a gear unit, but the elimination of gear units makes machines simpler and more reliable.

Although motors and controllers can be bought separately, the trend is to purchase a system from a given manufacturer. There are six types of variable-speed drives, as shown in Fig. 29-73. Dynamic response indicates the ability of the drive to respond to a change in command; it is measured in radians/second; the higher the number, the faster the drive response.

Variable-Speed Mechanical Drives Although rather common until the late 1960s, mechanical variable-speed drives, Figs. 29-74 and 29-75, are seldom used today. Their relative complexity makes these drives more maintenance-prone than the newer variable-frequency electric motor drive systems. However, variable-speed fluid drives (Fig. 29-76) are finding widespread use in major machinery drives demanding precise speed control in applications ranging to 50 MW and even higher. These variable-speed turbo couplings can be combined with one or more gear stages in a common housing. The bottom part of this compact unit forms an oil sump. From the basic concept consisting of a speed-increasing gear followed by a variable-speed turbo coupling, other models have been derived to provide stepless speed control for both high-power, high-speed machines such as boiler feed pumps and compressors; and low-speed machines with a speed-reducing gear such as coal mills, ID fans, and crude oil pumps.

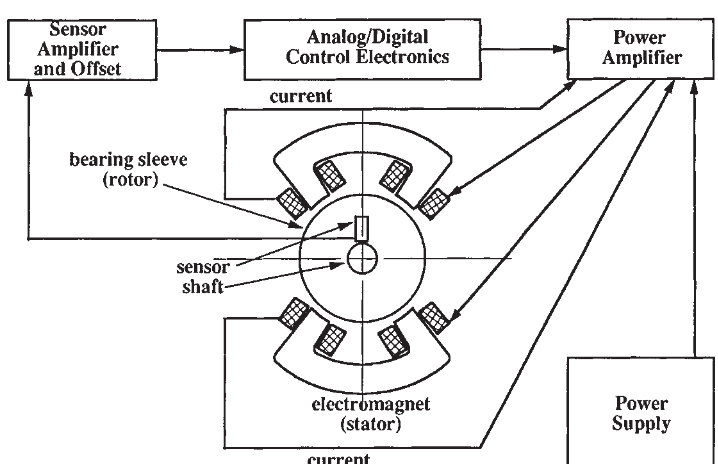


FIG. 29-72 Principle of operation of magnetic bearings.

Type of Drive	Speed Range	Starting Torque	Maximum Speed	Dynamic Response
Thyristor dc	100:1	150%	3000	15
Brushless dc	100:1	150%	3000	15
Std. Inverter	10:1	100%	6000	5
Vector Inverter	100:1	150%	6000	50
Servo	2000:1	200%	6000	500
Switched Reluctance	100:1	150%	10,000	50

FIG. 29-73 Main characteristics of typical electric adjustable-speed drives. (*Source: Reliance Electric Co.*)

The power developed by the prime mover is converted into kinetic energy in the impeller (primary wheel) of the turbo coupling and converted back into mechanical energy in the turbine wheel (secondary wheel), which is connected to the driven machine. As there is no metal-to-metal contact between primary and secondary wheels, there is no wear. Hydraulic oils with additives are used for power transmission. The amount of oil in the coupling can be varied during operation using the scoop tube. This in turn regulates the power-transmitting capability of the coupling and provides stepless speed control dependent on the load of the driven machine.

The coupling has a regulating range of 4:1 to 5:1 for driven machines with increasing parabolic torque load characteristics such as centrifugal pumps and fans. For machines with approximately constant torque load characteristics, the regulating range is 3:1. With centrifugal machines, this method of speed regulation is much more efficient than throttling the machine output, giving considerable power savings. The motor is started under no-load conditions with the coupling drained. When running the load, the motor can be controlled by the coupling. Moreover, by draining the fluid coupling, the prime mover can be disconnected from the driven machine while the prime mover is still running.

A mechanically driven oil pump on the primary side of the coupling pumps oil from the reservoir underneath the coupling through a control valve into the working chamber of the turbo coupling. The level of the oil in the working chamber and therefore the power that the turbo coupling can transmit depends on the radial position of the adjustable sliding scoop tube. The scoop tube can pick up more oil than the pump can deliver. The oil picked up by the scoop tube passes through an oil cooler/heat exchanger and control valve back to the working chamber and/or the oil reservoir. The heat exchanger dissipates the heat originating from the slip of the turbo coupling. The scoop tube actuator can be operated either electrically, hydraulically, or pneumatically.

Synchro-Self-Shifting ("SSS") Clutches The automatic free-wheel action of an SSS clutch simplifies plant startup and shutdown sequences. It is, of course, receiving its power input from a gas turbine or similar driver, while its output is connected to a driven machine. The drive disengages automatically when the driven machine speed exceeds that of the driver. Since large gas turbines are usually brought up to speed with either a startup gear motor or a helper turbine, thousands of SSS clutches (Fig. 29-77) are finding application here.

Upon unit startup, the turning gear motor can continue to rotate

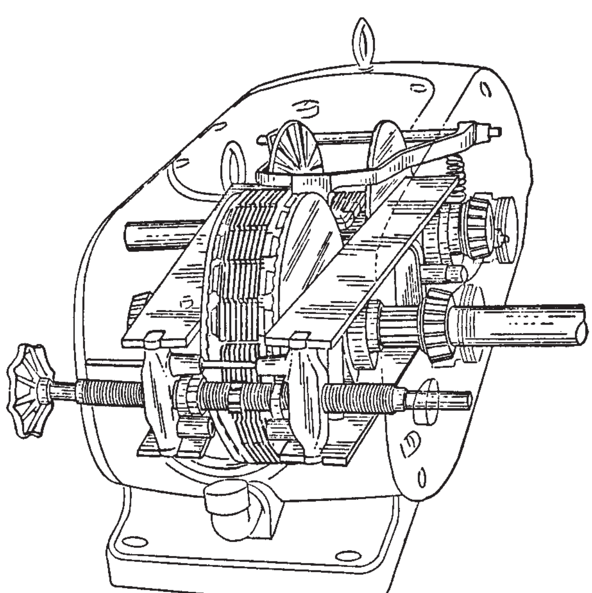


FIG. 29-74 PIV speed changer. (*From Kent,* Mechanical Engineers' Handbook, *12th ed., Wiley, New York, 1961.*)

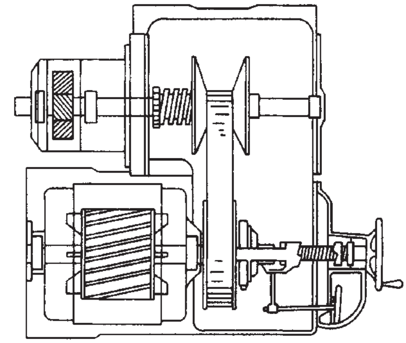


FIG. 29-75 Combined variable-speed and motor drive. (*Reeves Pulley Co.; from Kent,* Mechanical Engineers' Handbook, *12th ed., Wiley, New York, 1961.*)

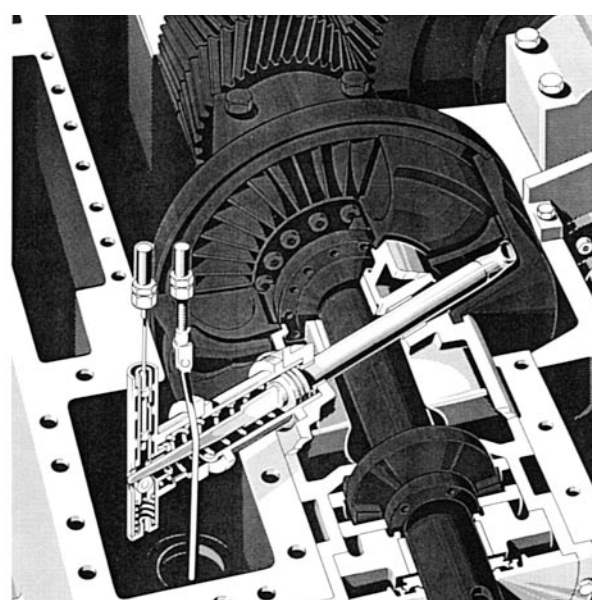


FIG. 29-76 Variable-speed turbo coupling (hydrodynamic fluid coupling). (*Courtesy of Voith Transmissions, Inc., York, Pennsylvania and Heidenheim, Germany.*)

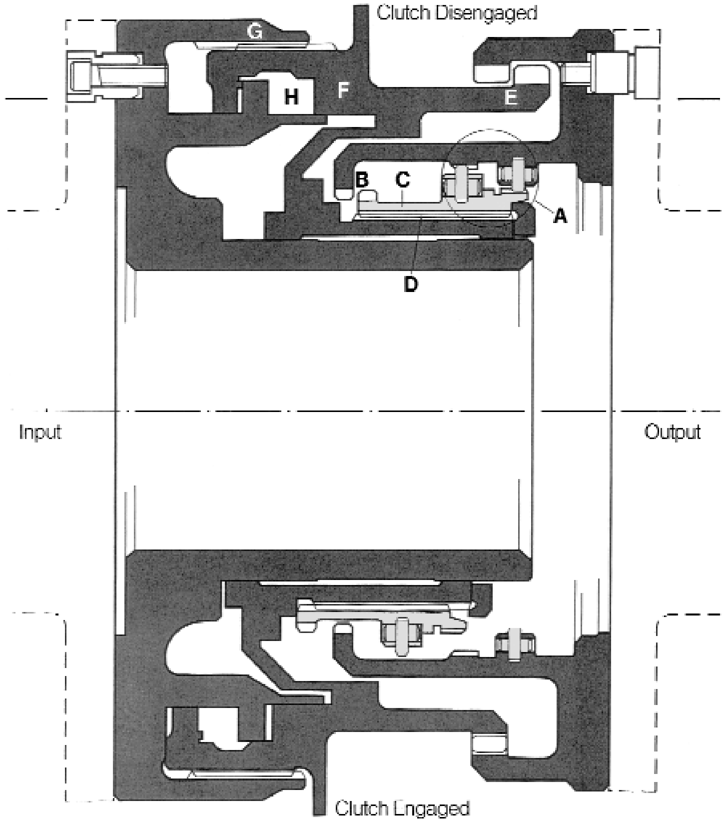


FIG. 29-77 Synchro-self-shifting clutch for large turbogenerator applications. (*Courtesy SSS Clutch Company, Inc., New Castle, Delaware.*)

or be stopped at any time. Similarly, it can be started at any time during the shutdown sequence, but the drive will only engage at turning gear speed.

Considerably larger units, with power ratings in the 300-MW range, are often fitted between a gas turbine driver and a utility power generator. This mode of application allows the generator to be used for voltage control/synchronous condensing with the gas turbine at rest but readily "on call" for peak load generation duty.

The sequence of clutch-engaging action is depicted in Fig. 29-77. Primary or secondary pawls (A) initiate engagement at clutch synchronism by aligning and engaging the relay clutch teeth (B). As the lightweight relay clutch (C) slides along its helical splines (D), the pawls are unloaded.

Primary pawls operate at low generator speeds. Secondary pawls operate at high turbine speeds. Therefore, all pawls are inert during synchronous condensing and when the clutch is engaged during power generation.

The relay clutch teeth (B) align and initiate engagement of the main clutch teeth (E). As the main clutch (F) slides along its helical splines (G), the relay clutch is unloaded.

When the main clutch teeth (E) are fully engaged, power is transmitted from the turbine to the generator. Engagement and disengagement of the main clutch is cushioned by the oil dashpot (H).

Flexible Couplings Whenever two machine shafts in substantial alignment are directly coupled, a flexible coupling is used to transmit the torque and accommodate the inevitable misalignment. A number of reasons make misalignment inevitable: thermal growth of machine housings and shafts, piping strain, foundation settlement, and so on. Without the ability to accommodate misalignment, machine shafts would fatigue and fail. Couplings accommodate misalignment either through sliding of one component over another or through flexing one or more of their components. As sliding requires lubrication, it is customary to categorize couplings as either lubricated or dry.

The most popular lubricated couplings in process machinery are the gear type (Fig. 29-78) and the grid coupling (Fig. 29-79). Lubricated couplings have the advantage of small size and weight but the disadvantage that machines must be stopped for the couplings to be relubricated (most machinery can be relubricated while running). Gear-type couplings also have the unique advantage that they can accommodate *any* axial shaft motions that machines require.

Dry couplings are divided into metallic elements and elastomer elements. For a given torque, metallic elements (disks and diaphragms) are more compact and lighter than elastomer element couplings but are less flexible; therefore, they impose larger forces on the bearings. On the other hand, elastomer elements become quickly distorted by centrifugal forces; therefore, they cannot usually operate safely at speeds larger than standard motor speed. These couplings are shown in Figs. 29-80 and 29-81.

It is important for coupling users to understand that *all flexible couplings resist being misaligned.* Hence, good alignment reduces the forces on machine bearings and increases machine reliability. Couplings designed to accommodate *large misalignments* usually can do this to the detriment of other features, such as reduced torque transmission or increased bearing forces.

It can be said that the trend in the 1980s/1990s was the gradual

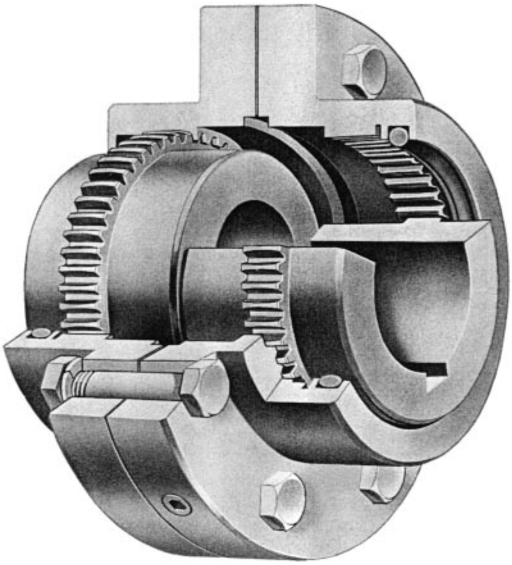


FIG. 29-78 Typical general-purpose gear-type coupling. (*Source: Kop-Flex Inc.*)

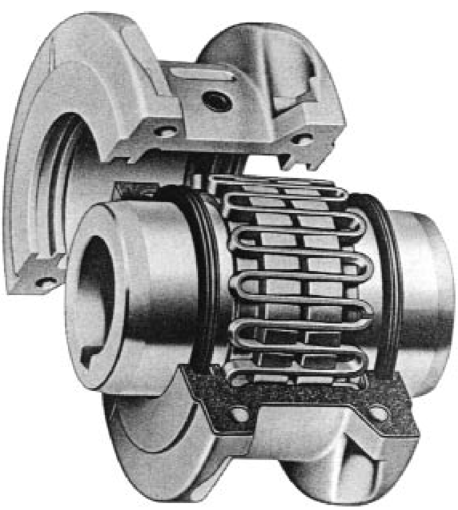


FIG. 29-79 Typical general-purpose steel-grid coupling. (*Source: The Falk Corp.*)

replacement of lubricated couplings with dry ones, particularly in machines designed for high power and high speeds. While there still are many machines equipped with lubricated couplings, new equipment is ordered with nonlubricated couplings.

POWER TRANSMISSION WITH SPEED CHANGE

Many process machines operate at speeds different from the one of their drivers. Typical of cases where the machine rotates *slower* than the driver are reciprocating compressors; typical examples of machines rotating *faster* than the drivers are centrifugal compressors driven by electric motors. In either case, *gears* are used to match the two speeds. Gears can also be designed to accommodate shafts that

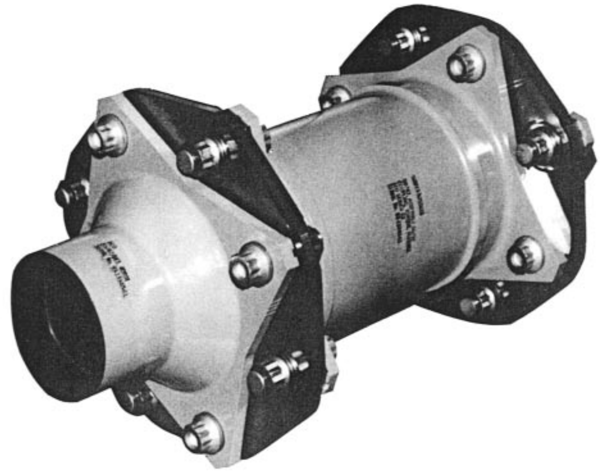


FIG. 29-80 Typical special-purpose disk-pack coupling. (*Source: Rexnord/Thomas Coupling Div.*)

FIG. 29-81 Typical general-purpose elastomer coupling. (*Source: Rexnord Corp.*)

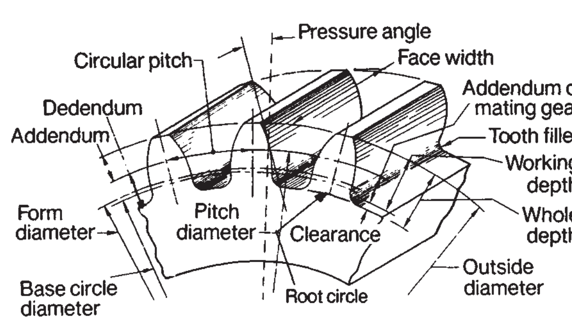


FIG. 29-82 Basic nomenclature of a gear.

are not parallel; gears can accommodate any angles between two shafts, including 90°. The following discussion is limited to gears for parallel shafts, which is the most common configuration in process machinery.

Basically, two engaging gears are two cylinders of different diameter with teeth machined on their periphery (Fig. 29-82). A gear is defined by its basic diameter (pitch diameter), its width, the number of teeth, and the angle that the teeth make in respect to the axis.

The ratio between the input-to-output shaft speeds is determined either by the ratio of the two pitch diameters or the ratio between the number of teeth:

$$i = \frac{PD_1}{PD_2}, \quad \text{or} \quad i = \frac{n_1}{n_2}, \quad \text{therefore} \quad \frac{PD_1}{PD_2} = \frac{n_1}{n_2}, \quad \text{or} \quad \frac{PD_1}{n_1} = \frac{PD_2}{n_2}$$

The ratio between the pitch diameter and the number of teeth is called the diametral pitch, and for two gears to mesh, they must have the same diametral pitch. The diametral pitch is standardized and is always an integer. For gears used in process machinery, the diametral pitch varies between 6 (coarse) and 20 (fine).

Teeth have an *involute* profile, a curve that is generated by the rolling of a straight line over a circle, as shown in Fig. 29-83. Although many curves could be used for gear teeth, the involute is the preferred one because it allows the correct engagement, even when the distance between the centers of the gears is not accurately held.

A tooth rolls over the meshing tooth only for a very small distance at the pitch diameter; above and below the pitch diameter, there is tangential sliding between the teeth, and lubrication is necessary to prevent premature wear. Because of this sliding, gears also have a certain power loss, which is about 2–3 percent per gear mesh. The heat generated by this power loss is dissipated by the lubricating oil. In high-speed gears, the oil jet is directed to where the teeth leave the engagement, as there is the point of maximum temperature.

The three most common types of cylindrical gears are shown in Fig. 29-84. *Spur* gears are the simplest, but they have the disadvantage of rough operation because as few as one or two teeth are in contact at any time. *Helical* gears have a smoother operation, but the angle of the helix generates an axial thrust on the shafts, which requires special bearings. The *double-helical* (also known as *herringbone*) gears have all the advantages of the helical gears without the axial thrust. They are also the most complicated to manufacture and are very sensitive to axial thrusts imposed from the outside on the gear shafts.

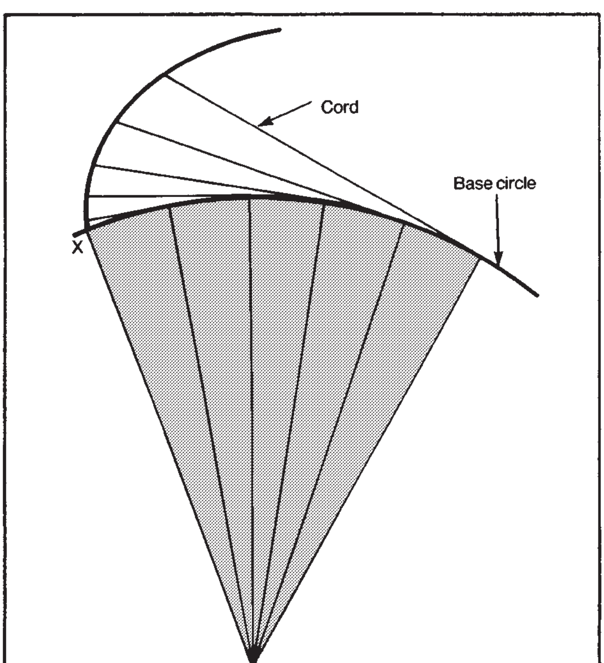


FIG. 29-83 The involute curve and its generation.

LUBRICATION OF POWER TRANSMISSION EQUIPMENT

Most machine bearings, all gears, and some couplings require lubrication. Even though a system can be designed with a direct drive, dry couplings, and magnetic bearings, such systems are still rare, and lubrication systems are ubiquitous. Lubrication systems can be as simple as a Zerk grease fitting at a motor bearing or as complicated as a console that includes a large oil tank, four or more electric-driven oil pumps, multiple filters, and oil coolers.

Grease as a Lubricant Greases are mechanical mixtures of lubricating oils and thickeners. Traditionally, metallic soaps (lithium, sodium, aluminum) were used for thickeners; although still in wide use, they are gradually being replaced with synthetic materials such as polyethylene. Greases blended with synthetic thickeners tend to be more stable but have lower resistance to high temperatures.

Greases are used as lubricants strictly because they are easily confined in a housing; greases have no additional *lubricating* properties over the oils used to blend them. Good greases are blended with special additives (just as good oils are) that can enhance their wear protection and rust protection characteristics. Greases have a paste consistency and can be very soft or very hard. The National Lubricating Grease Institute (NLGI) has established a scale for the consistency of greases that ranges from #00 (the softest) to #6. Technically, the consistency of a grease is determined by measuring the penetration of a weighted cone into the surface of a grease; a larger penetration indicates a softer grease. Unfortunately, the penetration value gives no indication about the lubricating properties of a grease; two greases can have the same consistency, but one may be blended with a high-viscosity, highly refined oil and have only 6 percent thickener, while the other may be blended with a low-viscosity, low-quality oil but have as much as 20 percent thickeners. Manufacturers may only label a grease container as "#2 lithium grease," which is not sufficient information. A complete specification sheet, however, is provided upon request. Larger manufacturers also publish a *digest,* in booklet form, that contains technical data on all their products.

There are hundreds of types of greases available, and machinery manufacturers usually provide a guideline for the greases to be used on their equipment. There are also highly specialized greases, such as the ones that can be used in food processing machinery, or the ones used in nuclear power stations.

To explain why greases cannot be used indiscriminately, the grease requirements for ball bearings and flexible couplings will be compared. Ball bearings require greases that channel, bleed, and contain a low-viscosity oil. Couplings require greases that do not channel, do not bleed, and contain a high-viscosity oil.

Channeling is required for ball bearings so that the balls can roll freely inside the bearing. Without a channel, there will be a high resistance to ball movement, and bearings would operate hot. Bleeding (slow release) of oil is required so that the balls are continuously supplied with the needed lubricant. A high-viscosity oil in a ball bearing would cause the balls to skid (hydroplane) and suffer severe scoring.

Couplings need greases that are very soft (NLGI #0 or #1) so that the spaces around the teeth are always filled with grease. Channeling would allow for metal-to-metal contact and rapid wear would occur. Because coupling rotation subjects greases to high centrifugal acceleration (that can exceed 10,000 Gs), bleeding would rapidly separate the soaps from the grease, and the oils would quickly escape, causing couplings to operate dry. Coupling teeth do not roll on each other; the

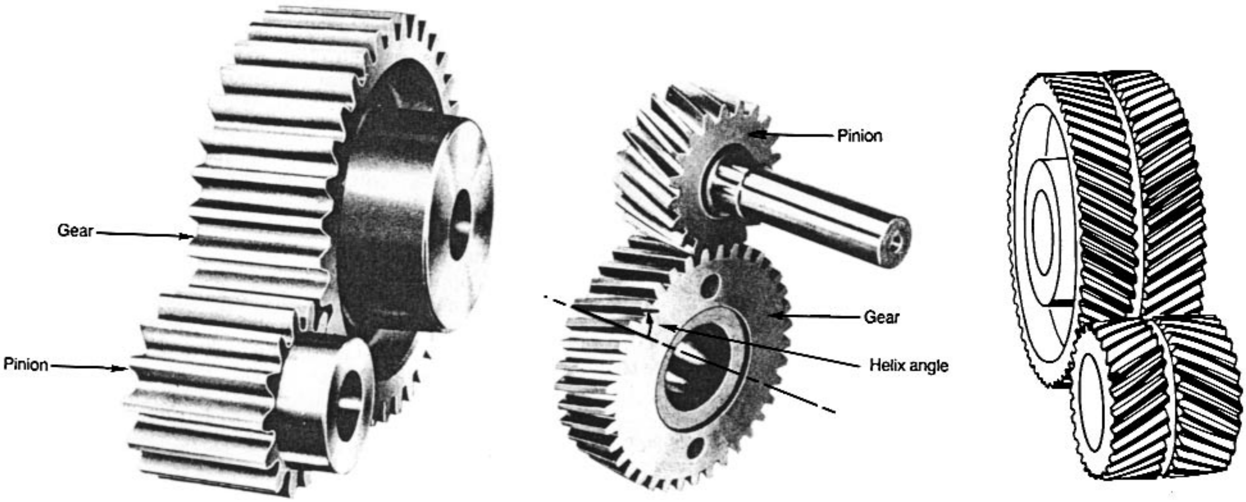


FIG. 29-84 The spur, helical, and double-helical gears.

motions in couplings are reciprocating and sliding. These motions are best lubricated by a highly viscous oil.

As was shown, using a coupling grease in a bearing could cause its failure; using a bearing grease in a coupling could cause its rapid wear.

Oil as a Lubricant Oils are used for lubrication and heat dissipation; therefore, oils are circulated through a machine (through bearings, seals, or gears), returned to a tank where dirt and water are allowed to settle out, and then pumped through coolers and filters back into the machine. Without a continuous flow of oil, machines could rapidly fail; this is why backup pumps are always provided. In systems where the main oil pump is directly driven by the machine (such as in gear boxes), two backup pumps are provided: one driven by an ac motor, which is the same size as the main one, and one driven by a dc motor (battery operated), which is small, since machines are always shut down if the main and the spare pumps fail. The dc-driven pump must only supply oil during the coast-down operation.

Filters separate and retain the dirt from the oil. They are rated based on the size of the largest particle that can pass through. Most machines in petrochemical plants use 10-micron filters. Alarms are provided to signal when a filter becomes clogged up with dirt. A valving system allows the oil to be bypassed to an alternate filter, while the element of the first one is replaced. Filter elements can be made of a wire mesh, a wire loop (closely spaced), or pleated fiber mats. The wire-loop filters are self-cleaning, as the cylindrical element slowly rotates against a wire comb; however, they cannot filter fine particles.

Oil coolers either dissipate the heat into a water stream or the air. Water coolers (Fig. 29-85) are significantly more compact, but a supply of water is required. Air coolers (Fig. 29-86) are large and require a fan that increases the air flow over the cooling fins.

A complete lubrication console schematic is shown in Fig. 29-87. Instrumentation for the console includes thermometers, pressure switches, and flow meters.

Oils used in process machinery lubrication are generally of low viscosity and are known as turbine oils or compressor oils. Such oils would have a viscosity of 78 centiStokes at 40° C (40 Saybolt Seconds

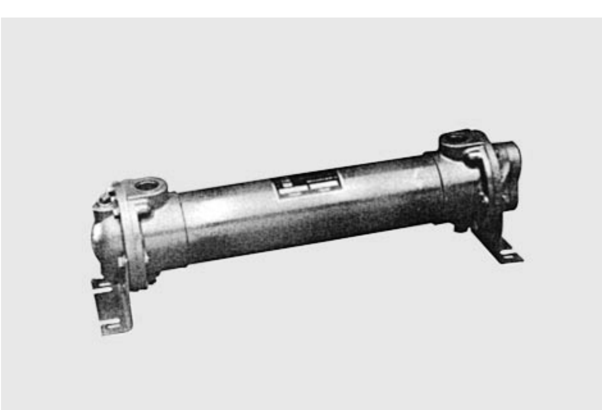


FIG. 29-85 Typical oil cooler, where the heat is transferred to water.

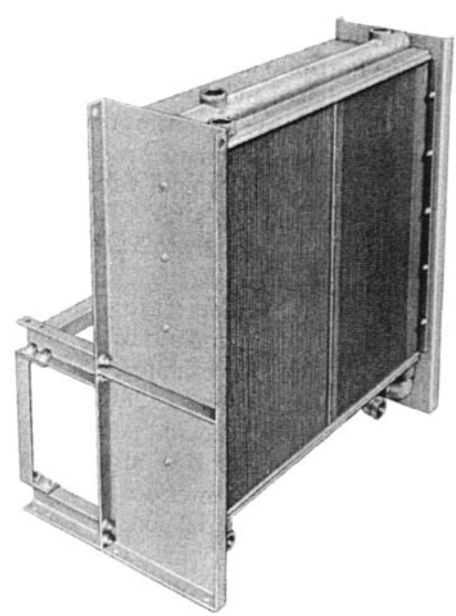


FIG. 29-86 Typical oil cooler, where the heat is transferred to air.

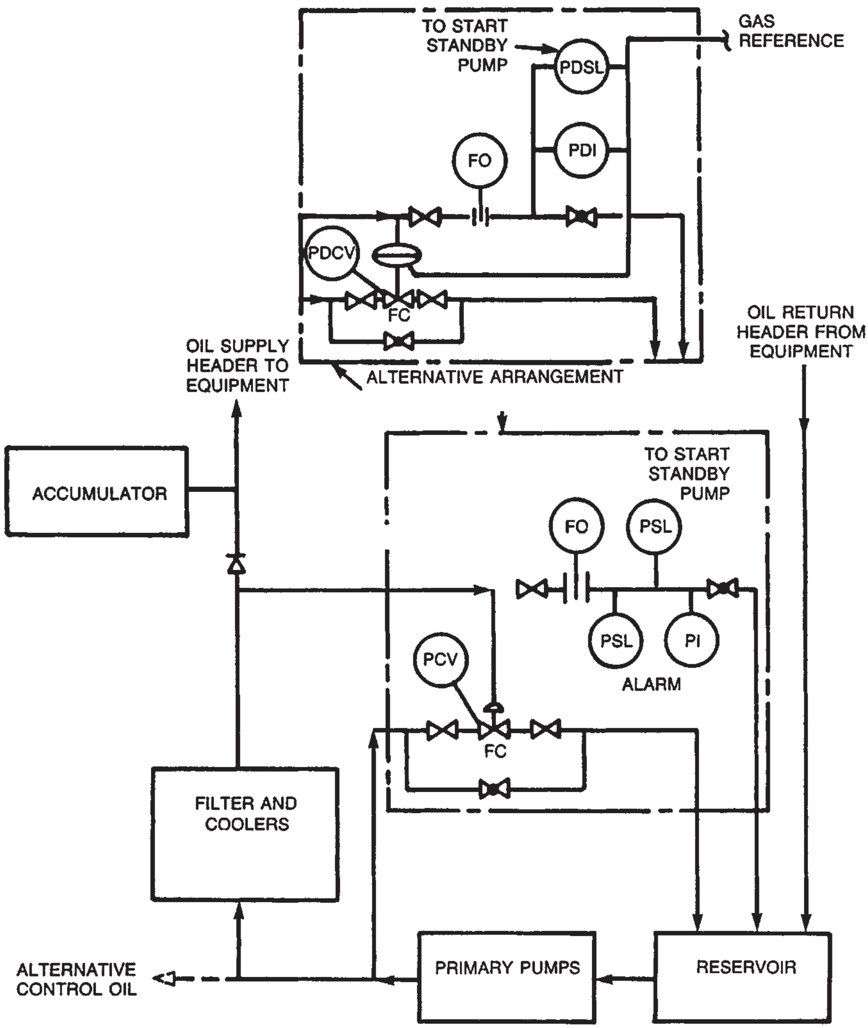


FIG. 29-87 Typical lubricating oil system for medium to large process machinery.

Universal at 100° F) and a viscosity index of 95. They contain necessary oxidation inhibitors and antiwear agents, as the same oil is used for bearings and gear lubrication. Oils tend to retain water, either from steam condensation or from condensation that occurs in the tanks. The presence of water is detrimental as it encourages the formation of acids and sludge, which can severely limit the useful life of machine components. It is customary to analyze periodically samples taken from the oil tanks and replace oils when the contamination is severe. A number of catastrophic coupling failures have been attributed to corrosion by contaminated oils.