

SECTION 15

Prime Movers For Mechanical Drives

“Prime movers for mechanical drives are used as a common name for machines made for transferring mechanical energy to pumps and compressors, including:

- Steam turbines
- Gas turbines
- Electrical motors
- Internal combustion engines

Special considerations for the use of prime movers as drives for generators are not included in this chapter.

STEAM TURBINE TYPES

Mechanical drive steam turbines are major prime movers for compressor, blower, and pump applications. Steam turbines are available for a wide range of steam conditions, horsepower, and speeds. Typical ranges for each design parameter are:

Inlet Pressure, psig	30 – 2000
Inlet Temperature, °F	saturated – 1000
Exhaust Pressure, psig	saturated – 700
Horsepower	5 – 100,000
Speed, rpm	1800 – 14,000

Steam turbines used as process drivers are usually required to operate over a range of speeds in contrast to a turbine used to drive an electric generator which runs at nearly constant speed. Significant hardware differences exist between these two applications. Only variable speed process drivers will be covered here.

Mechanical drive steam turbines are categorized as:

- Single-stage or multi-stage
- Condensing or non-condensing exhausts
- Extraction or admission
- Impulse or reaction

Single Stage/Multi-Stage

In a single-stage turbine, steam is accelerated through one cascade of stationary nozzles and guided into the rotating blades or buckets on the turbine wheel to produce power. A Rateau design has one row of buckets per stage (Fig. 15-2). A Curtis design has two rows of buckets per stage and requires a set of turning vanes between the first and second row of buckets to redirect the steam flow (Fig. 15-3). A multi-stage turbine utilizes either a Curtis or Rateau first stage followed by one or more Rateau stages.

Single-stage turbines are usually limited to about 2500 horsepower although special designs are available for larger units. Below 2500 horsepower the choice between a single and a multi-stage turbine is usually an economic one. For a given shaft horsepower, a single-stage turbine will have a lower capital cost but will require more steam than a multi-stage turbine because of the lower efficiency of the single-stage turbine.

Condensing/Non-Condensing

The energy available in each pound of steam which flows through the turbine is a function of the overall turbine pressure ratio (inlet pressure/exhaust pressure) and inlet temperature. Condensing turbines are those whose exhaust pressure is below atmospheric. They offer the highest overall turbine pressure ratio for a given set of inlet conditions and therefore require the lowest steam flow to produce a given horsepower. A cooling medium is required to totally condense the steam.

Non-condensing or back-pressure turbines exhaust steam at pressures above atmospheric and are usually applied when the exhaust steam can be utilized elsewhere.

Extraction/Admission

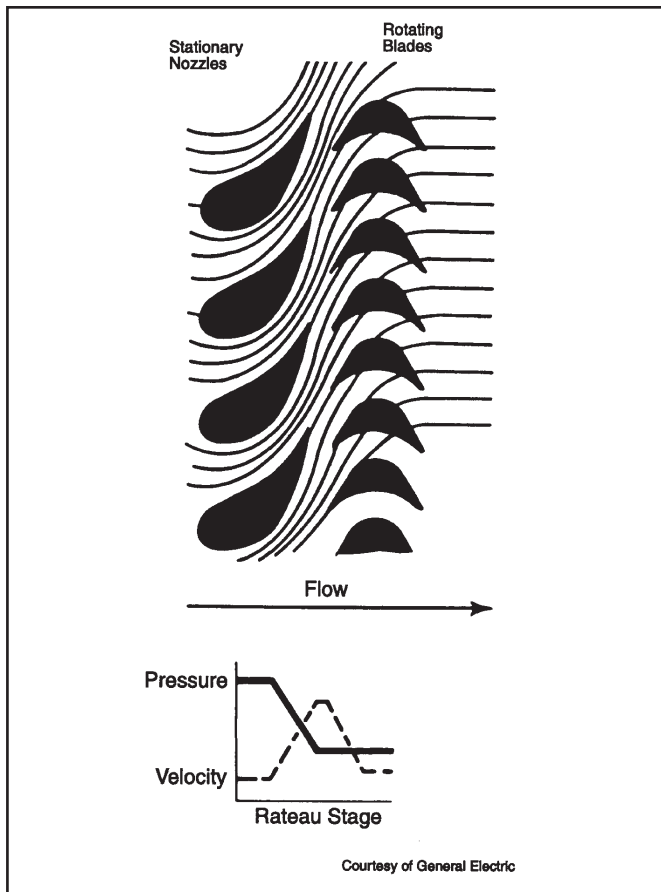
Some mechanical drive steam turbines are either extraction or admission machines. Steam is extracted from, or admitted to, the turbine at some point between the inlet and exhaust (Fig. 15-4). Admission or extraction units may be either controlled or uncontrolled. An uncontrolled turbine accepts or provides steam based only on the characteristics of the steam sys-

FIG. 15-1
Nomenclature

A	=	area, sq in.
ASR	=	actual steam rate, lb/(hp · hr)
BMEP	=	brake mean effective pressure, psi
D	=	diameter, in.
F	=	steam flow, lb/hr
f	=	frequency, Hz
h	=	specific enthalpy of superheated steam, Btu/lb
h _f	=	specific enthalpy of saturated water, Btu/lb
h _g	=	specific enthalpy of saturated steam, Btu/lb

N	=	number of power strokes per min
P	=	number of magnetic poles in motor
s	=	specific entropy of superheated steam, Btu/(lb · °F)
s _f	=	specific entropy of saturated water, Btu/(lb · °F)
s _g	=	specific entropy of saturated steam, Btu/(lb · °F)
S	=	piston stroke, ft
TSR	=	theoretical steam rate, lb/(hp · hr)
v	=	velocity, ft/sec
ρ	=	density, lb/cu ft

FIG. 15-2
Rateau Design



tem to which the extraction or admission line is connected. A controlled turbine will control the flow of extraction or admission steam based on some process measurement such as pressure or flow. In general, if the horsepower associated with the extraction or admission flow is greater than 15% of the total turbine horsepower, a controlled extraction (or admission) turbine is used.

Impulse/Reaction

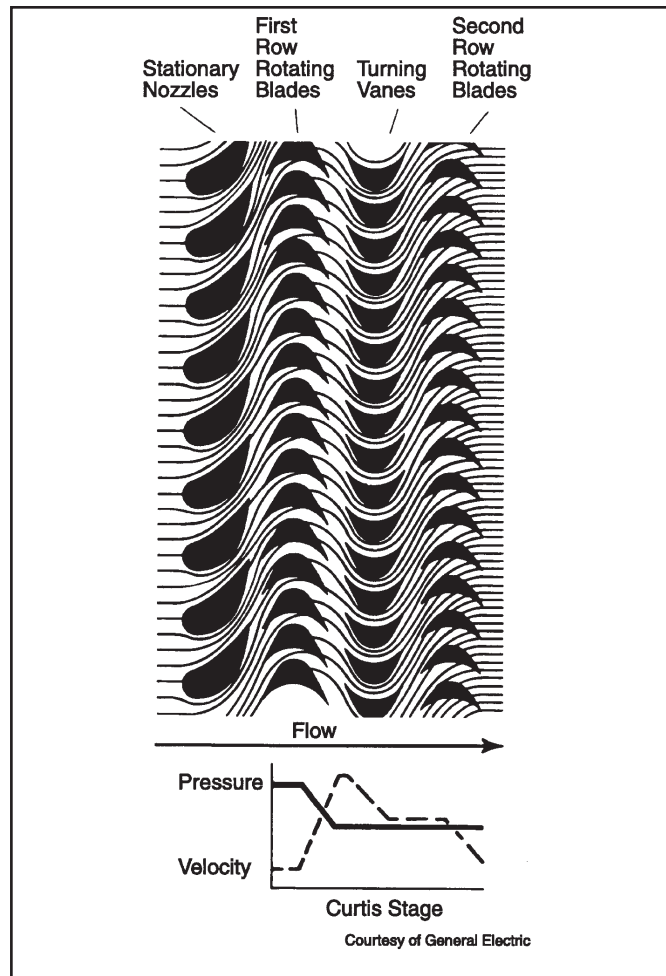
Turbines are further categorized by the philosophy employed in the steam path design and are divided into two major design concepts: impulse and reaction. In an impulse turbine the pressure drop for the entire stage takes place across the stationary nozzle. In reaction designs, the pressure drop per stage is divided equally between the stationary nozzles and the rotating blades (Fig. 15-5). For given horsepower, speed and steam conditions, a reaction turbine will, in general, employ approximately three times more stages than an impulse turbine in the same turbine span. Most U.S. mechanical drive steam turbines are of the impulse type.

STEAM TURBINE COMPONENTS

Trip and Throttle Valve/Stop (Block) Valve

A trip-and-throttle valve or stop valve, or both, may be positioned between the steam supply and the turbine inlet control valve(s) (Fig. 15-6). During normal operation this valve remains

FIG. 15-3
Curtis Design



fully open and its primary function is to shut off the steam supply in response to a trip (shutdown) signal. In addition a trip-and-throttle valve can be used to modulate the steam flow during start-up and can be either manually or hydraulically positioned from zero lift to 100% lift. The stop valve can only be positioned either in the closed or fully open positions. In order to minimize the pressure drop through the trip-and-throttle valve, maximum inlet velocities are usually limited to 150 ft/sec. Velocities above this level will usually result in high pressure drops which will reduce turbine efficiency.

Inlet Control Valves

The primary function of the inlet control valve(s) is regulation of the steam flow to provide the appropriate horsepower and speed. These valves may also close in response to a shutdown signal. Throttling which occurs across the control valve(s) reduces the thermal performance of the turbine. This efficiency loss is a function of the control valve design and overall turbine pressure ratio. For a given amount of throttling, turbines with large pressure ratios suffer smaller efficiency losses than turbines with smaller pressure ratios (Fig. 15-7).

Multi-stage turbines may have a single inlet control valve or several control valves to regulate the inlet steam. Typical multi-valve steam turbines will have from three to eight control valves

FIG. 15-4
Extraction/Admission Flow Turbines

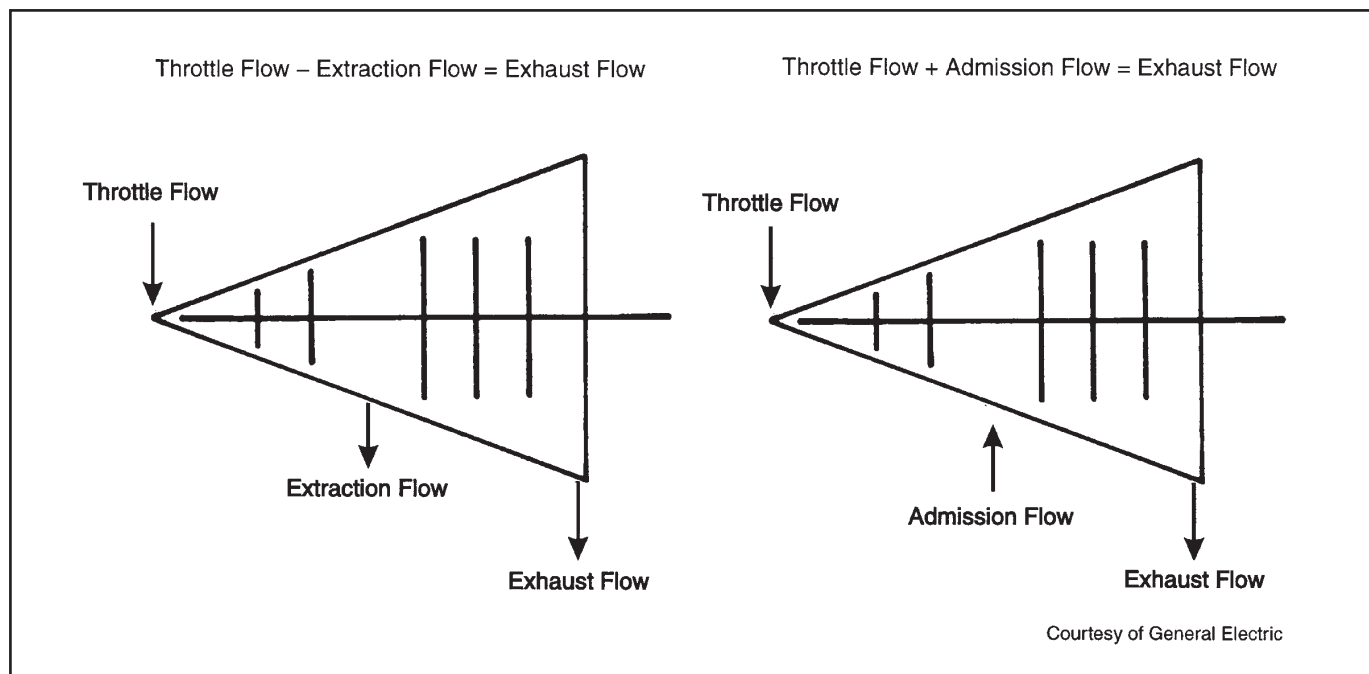
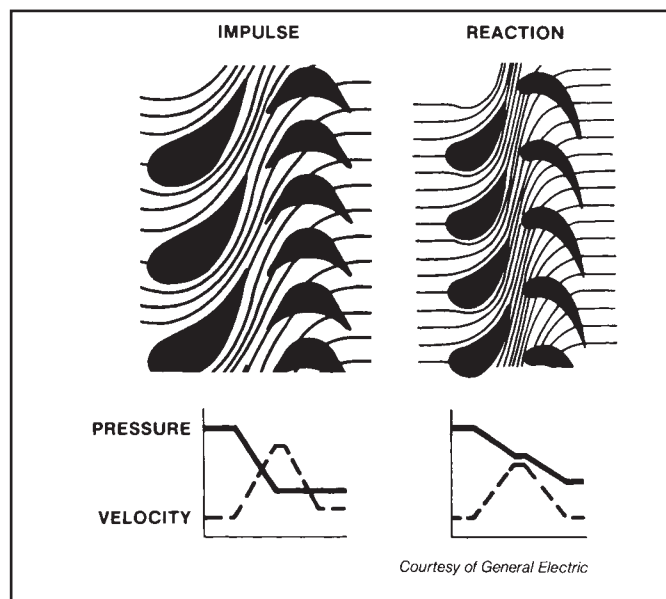


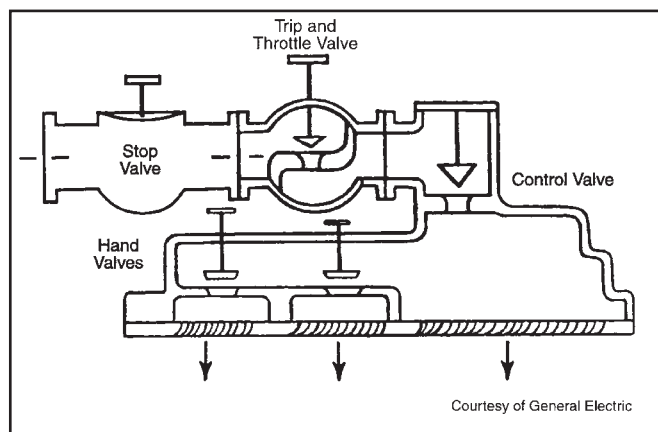
FIG. 15-5
Turbine Types



(Fig. 15-8). Multi-valve turbines have higher efficiencies at reduced loads because only the flow through one of the control valves is incurring a throttling loss (Fig. 15-9).

Turbines with a single control valve will often employ hand valves to improve efficiency at reduced loads. For the turbine shown in Fig. 15-6 both hand valves would be open at or near full load. As the load on the unit is reduced one or both of these hand valves can be closed to reduce throttling loss. Fig. 15-10 shows the efficiency advantage at reduced loads.

FIG. 15-6
Single Valve with Hand Valves



Nozzles/Blades (Buckets)

On constant speed turbines a design objective is to avoid all bucket resonances at the operating speed. On variable speed turbines, although the design objectives remain the same, it is seldom possible to avoid all blade resonance because of the wide operating speed range. In these cases it is important to identify all blade resonance and to verify that all stresses are well below the material strength.

Exhaust Casings

Turbine exhaust casings are categorized by pressure service (condensing or non-condensing) and number of rows of the last stage buckets (single flow, double flow, triple flow). Non-condensing exhausts are usually cast steel with most of the applications between 50 and 700 psig exhaust pressure. Most con-

FIG. 15-7

Loss in Available Energy of Steam due to 10% Throttling

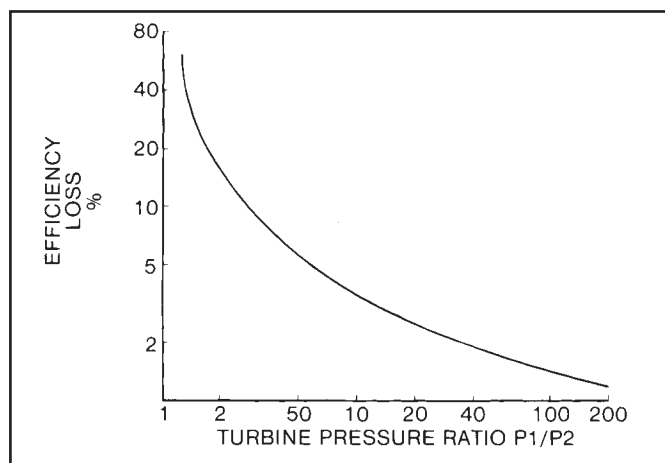
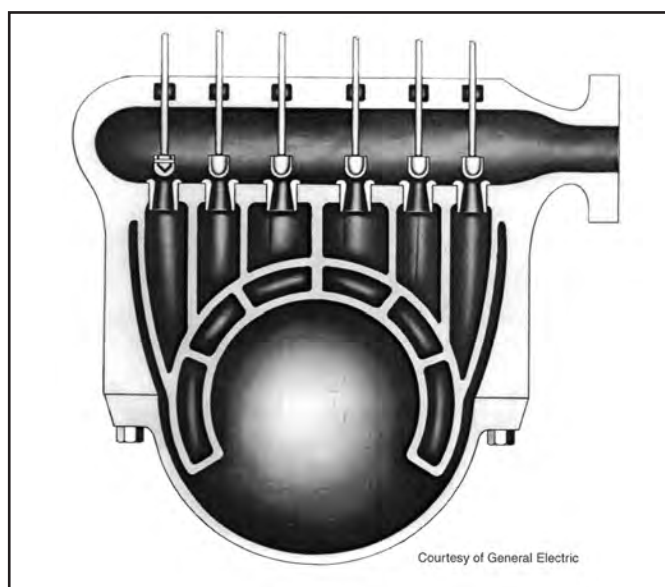


FIG. 15-8

Multi-Valve Inlet



condensing exhausts are steel fabrications although some utilize cast iron construction. Maximum exhaust flange velocities are typically 450 ft/sec. Velocities above this level will usually result in substantial increases in exhaust hood losses and will decrease turbine efficiency.

Moisture Protection

As steam expands through the turbine both the pressure and temperature are reduced. On most condensing and some non-condensing exhaust applications, the steam crosses the saturation line thereby introducing moisture into the steam path. The water droplets which are formed strike the buckets and can cause erosion of the blades. In addition, as the water is centrifuged from the blades, the water droplets strike the stationary components, also causing erosion. Where the moisture content is greater than 4%, moisture separators, which are in-

FIG. 15-9

Multi-Valve vs Single-Valve Performance Characteristic (Typical Non-Condensing Turbine)
Turbine Pressure Ratio = 8.0

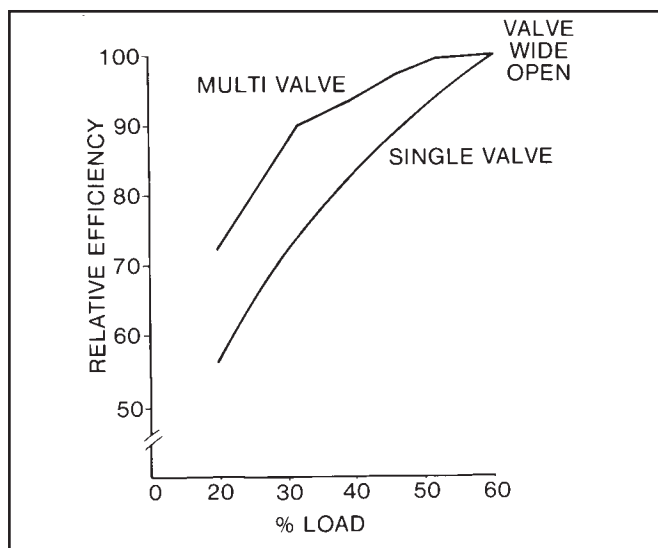
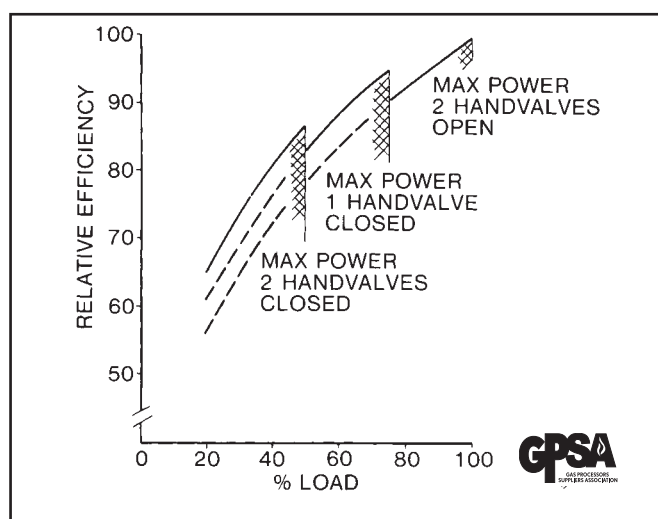


FIG. 15-10

Single-Valve with Hand Valves Performance Characteristic (Typical Non-Condensing Turbine)
Turbine Pressure Ratio = 8.0



ternal to the turbine, can be used to remove a large percentage of the moisture, improving the turbine efficiency and reducing the impact erosion on the buckets. Stainless steel moisture shields can also be used to minimize the impact erosion of the stationary components.

Control Systems

Mechanical governors were the first generation control systems employed on mechanical drive turbines. Shaft speed is sensed by a fly-ball governor with hydraulic relays providing the input to the control valve. A second generation control system was developed and utilized analog control circuitry with

the fly-ball governor replaced by speed pick-ups and the hydraulic relays with electronic circuit boards. A third generation control system was developed and replaced the electronic circuitry with digital logic. A microprocessor is used and the control logic is programmed into the governor. The major advantage of this system is the ability to utilize two governors simultaneously, each capable of governing the turbine alone. If the primary governor incurs a fault, the back-up governor assumes control of the turbine and provides diagnostic information to the operator.

STEAM TURBINE EFFICIENCY

Factors Affecting Efficiency

The objective of the steam turbine is to maximize the use of the available steam energy where the available steam energy is defined as the difference between the inlet and exhaust energies (enthalpies) for a 100% efficient constant entropy (i.e., isentropic) process. There are numerous loss mechanisms which reduce the efficiency from the isentropic such as throttling losses, steam leakage, friction between the steam and the nozzles/buckets, bearing losses, etc. Efficiency can range from a low of 40% for a low horsepower single-stage turbine to a high approaching 90% for a large multistage, multi-valve turbine.

Techniques to Improve Efficiency

Various techniques are employed to maximize turbine efficiency, each designed to attack a specific loss mechanism. For example, the number of stages utilized can range from the fewest possible to develop the load reliably to the thermodynamically optimum selection. Spill bands can be utilized to minimize throttling losses. High efficiency nozzle/bucket profiles are available to reduce friction losses. Exhaust flow guides are available to reduce the pressure within the exhaust casing.

The specific features employed on a given application are usually based on the trade-off between capital investment and the cost to produce steam over the life of the turbine.

Operation at Part Load

Most equipment driven by steam turbines are centrifugal machines where horsepower varies as the cube of speed. Part load efficiency varies as a function of speed, flow, and the number of stages. By assuming horsepower to vary as the cube of speed the turbine part load efficiency can be approximated as a percentage of the design efficiency (Fig. 15-11).

EXAMPLES

Figs. 15-11 through 15-19 and 24-30 and 24-31 allow estimates to be made of steam rate, turbine efficiency, number of stages, and the inlet and exhaust nozzle diameters. The following examples illustrate the use of these figures:

Example 15-1 — Given a steam turbine application with the following characteristics:

Inlet Pressure	600 psia
Inlet Temperature	750°F
Exhaust Pressure	2 psia
Required Horsepower	6000 hp
Speed	7000 rpm

Determine

- The actual steam rate (ASR)
- The inlet and exhaust nozzle diameters
- The approximate number of stages
- The steam rate at a partial load of 4000 hp and 6100 rpm

Solution Steps

Using Figs. 24-30 and 31, the theoretical steam rate (TSR) may be determined from the difference in the inlet enthalpy and the theoretical exhaust enthalpy (i.e. isentropic exhaust

FIG. 15-11

Part Load Efficiency Correction Factor vs Percent Power Multi-Valve Steam Turbines

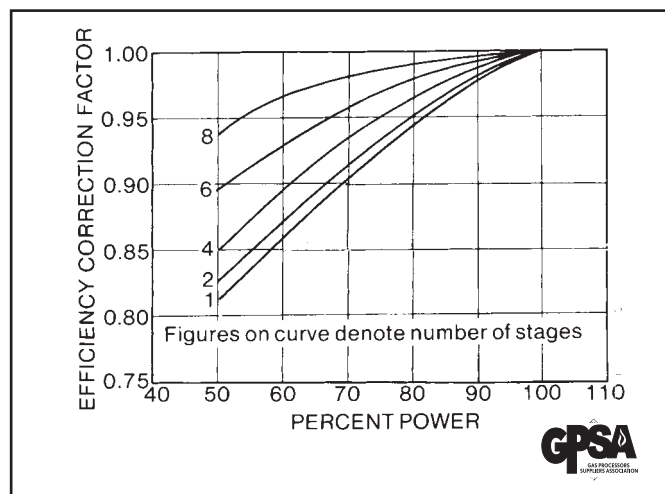
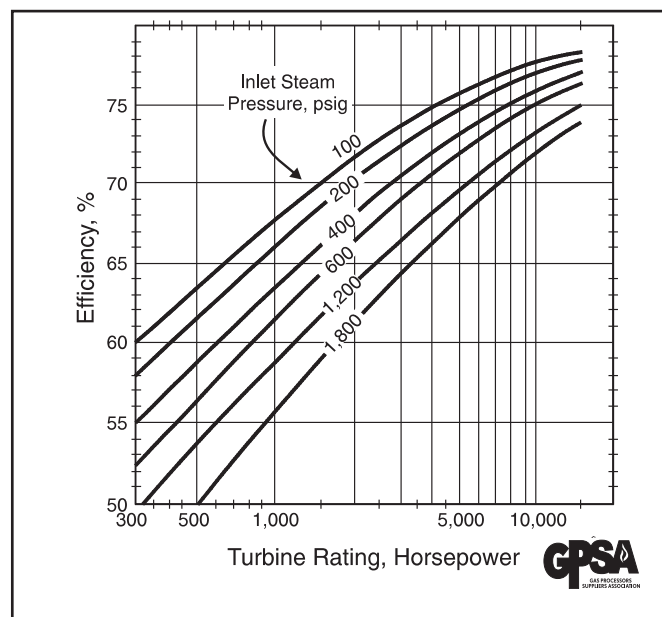


FIG. 15-12

Basic Efficiency of Multi-Valve, Multi-Stage Condensing Turbines



enthalpy), but first the inlet and exhaust states should be confirmed. Fig. 24-31 for superheated steam indicates that the inlet is superheated (i.e., 750°F is above the saturation temperature of 486.2°F), and gives an inlet entropy of 1.6109 Btu/(lb · °F). From Fig. 24-30, for saturated steam at the turbine exhaust pressure of 2 psia absolute, the liquid and vapor entropies are 0.1750 and 1.9200 Btu/(lb · °F). Since the inlet entropy is within this range, the theoretical exhaust must be two-phase. Had the exhaust-vapor entropy been equal to the inlet entropy, the exhaust would be single-phase vapor (i.e. at its dewpoint). Had the exhaust-vapor entropy been below the inlet entropy, the assumed two-phase exhaust would have been incorrect and Fig.

FIG. 15-13

Basic Efficiency of Multi-Valve, Multi-Stage Non-Condensing Turbines

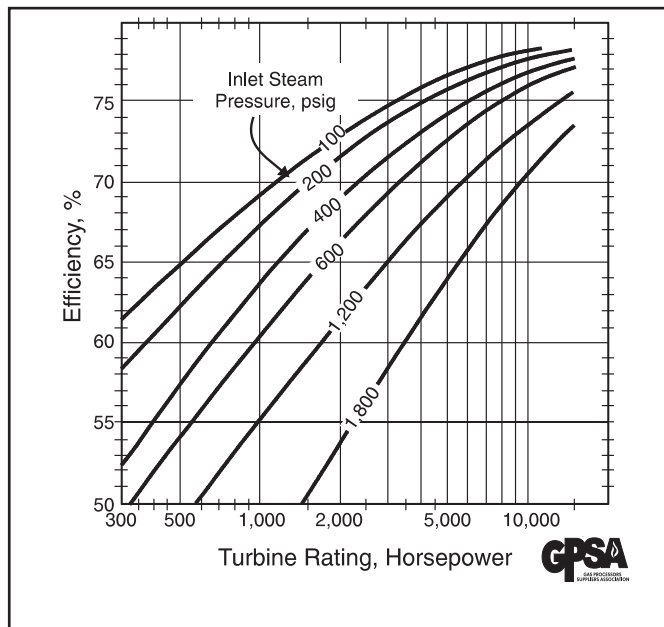


FIG. 15-14

Superheat Efficiency Correction Factor for Condensing Turbines

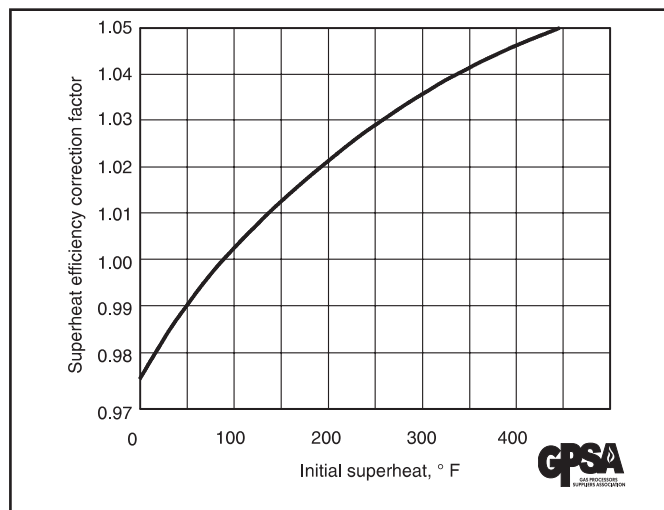
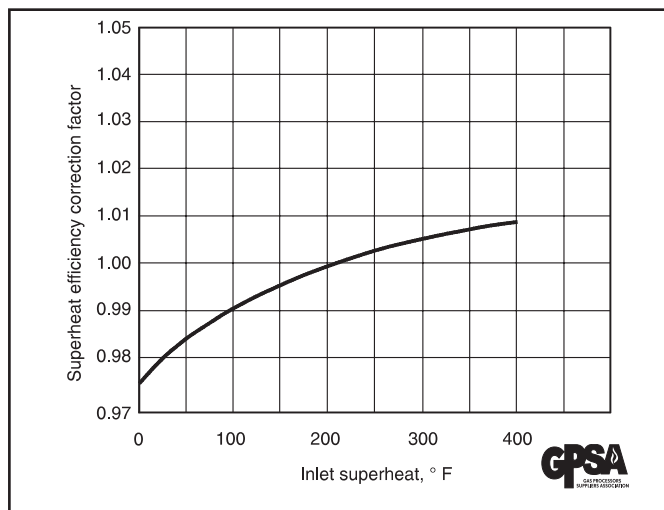


FIG. 15-15

Superheat Efficiency Correction Factor for Non-Condensing Turbines



24-31 instead of 24-30 would be applicable.

Inlet conditions at 600 psia and 750°F (the average of the values at 700°F and 800°F on Fig. 24-31):

$$s = 1.6109 \text{ Btu/(lb} \cdot \text{°F)}$$

$$h = 1379.4 \text{ Btu/lb}$$

Exhaust conditions at 2.0 psia:

$$s_f = 0.1750 \text{ Btu/(lb} \cdot \text{°F)}$$

$$s_g = 1.9200 \text{ Btu/lb} \cdot \text{°F}$$

$$h_f = 94.03 \text{ Btu/lb}$$

$$h_g = 1116.2 \text{ Btu/lb}$$

Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

$$1.6109 = x(0.1750) + (1 - x)(1.9200)$$

$$x = 0.1771$$

$$1 - x = 0.8229 \text{ (vapor fraction in the exhaust)}$$

$$\begin{aligned} \text{Exhaust enthalpy} &= (0.1771)(94.03) + (0.8229)(1116.2) \\ &= 935.2 \text{ Btu/lb} \end{aligned}$$

$$\text{Enthalpy change} = 935.2 - 1379.4$$

$$= -444.2 \text{ Btu/lb}$$

Substituting $\text{Btu} = (\text{hp} \cdot \text{hr})/2544$:

$$\text{Enthalpy change} = (-444.2/2544) = (-1/5.727)(\text{hp} \cdot \text{hr})/\text{lb}$$

$$\text{TSR} = \text{the absolute value of the inverse of the enthalpy change} = 5.727 \text{ lb/}(\text{hp} \cdot \text{hr})$$

$$\text{Basic efficiency} = 0.729 \text{ (Fig. 15-12)}$$

$$\text{Inlet saturation temperature} = 486.2^\circ\text{F} \text{ (first column Fig. 24-31)}$$

$$\text{Inlet superheat} = 750 - 486 = 264^\circ\text{F}$$

$$\text{Superheat efficiency-correction factor} = 1.03 \text{ (Fig. 15-14)}$$

Speed efficiency-correction factor = 0.957 (Fig. 15-16)

Corrected efficiency = $(0.729)(1.03)(0.957) = 0.719$

ASR = $5.727/0.719 = 7.97 \text{ lb/(hp} \cdot \text{hr)}$

$F = (6000 \text{ hp}) 7.97 \text{ lb/(hp} \cdot \text{hr)}$

$= 47,800 \text{ lb/hr}$

The inlet and exhaust diameters may be estimated from the equation:

$$D = \sqrt{\frac{(0.051)(F)}{(\rho v)}} \quad \text{Eq 15-1}$$

A reasonable rule of thumb for maximum velocity of the inlet steam is 150 (ft/sec).

$\rho = 0.88 \text{ lb/ft}^3 \text{ @ } 600 \text{ psia and } 750^\circ\text{F}$

$$D = \sqrt{\frac{(0.051)(47,800)}{(0.88)(150)}}$$

$D = 4.3 \text{ in.}$

A 4 in. NPS (minimum) inlet nozzle would be selected.

For exhaust sizing a maximum steam velocity of 450 ft/sec is a reasonable rule of thumb.

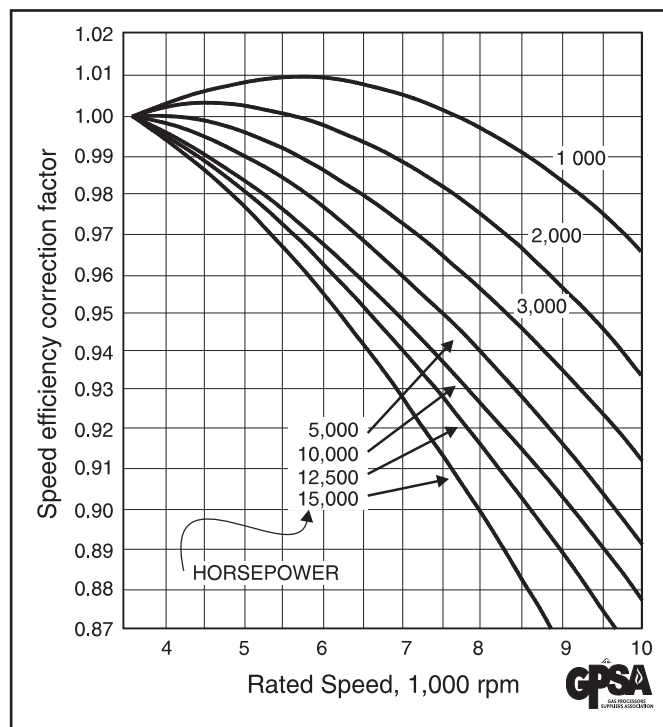
$\rho = 0.0057 \text{ lb/ft}^3 \text{ @ } 2 \text{ psia}$

$$D = \sqrt{\frac{(0.051)(47,800)}{(0.0057)(450)}}$$

$D = 30.8 \text{ in.}$

FIG. 15-16

Speed Efficiency Correction Factor for Condensing and Non-Condensing Turbines



A 30 in. exhaust nozzle would be selected.

The number of stages may be estimated using Fig. 15-18. Drawing a horizontal line from the 7000 RPM indicates that between 1.5 and 2 stages per 100 Btu/lb of available energy would be acceptable.

Available Energy (theoretical)(i.e., the isentropic enthalpy change calculated above)

$= 444.2 \text{ Btu/lb}$

Number of Stages

$$= \frac{(1.5)(444)}{(100)} = 7 \text{ (approximately)}$$

or, Number of Stages

$$= \frac{(2)(444)}{(100)} = 9 \text{ (approximately)}$$

Nine stages would provide increased efficiency but at additional cost.

At partial load of 4000 hp and 6,100 RPM and assuming seven stages from Fig. 15-11, a part load efficiency factor of approximately 0.96 is obtained. From Fig. 15-12, the basic efficiency at 4000 hp and 6,100 RPM is estimated to be 0.71.

Efficiency = $(0.96)(0.71) = 0.68$

Actual Steam Rate = $5.73/0.68 = 8.43 \text{ lb/(hp} \cdot \text{hr)}$

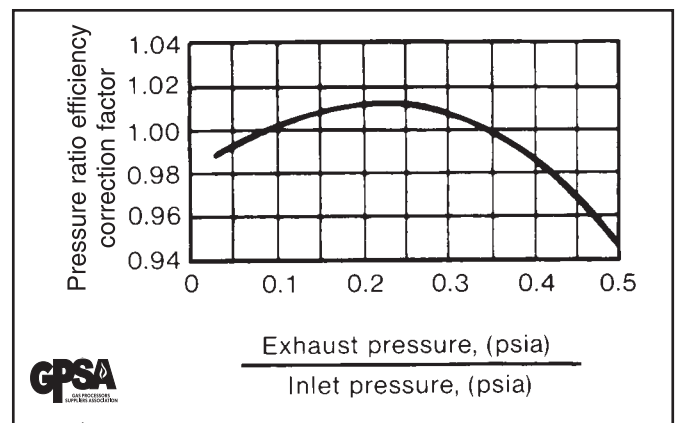
$F = (4000)(8.43) = 33,700 \text{ lb/hr}$

Example 15-2 — Determine the ASR and total steam requirements for a multi-stage turbine and a single-stage turbine at the following conditions:

Inlet Pressure	250 psig
Outlet Pressure	100 psig
Inlet Temperature	500°F
Horsepower	900 hp
Speed	5000 rpm

FIG. 15-17

Pressure Ratio Efficiency Correction Factor, Non-Condensing Turbines



Solution Steps

For a multi-stage turbine:

Examining Figs. 24-30 and 31 in the same way as in Example 15-1, the turbine inlet is superheated, and the exhaust is two-phase.

Inlet conditions at 250 psig (264.7 psia) and 500°F (interpolating linearly between 240 and 260 psia on Fig. 24-31):

$$s = 1.5873 \text{ Btu/(lb} \cdot ^\circ\text{F)}$$

$$h = 1261.8 \text{ Btu/lb}$$

Exhaust conditions at 100 psig (i.e. 114.7 psia). From Fig. 24-30 interpolating linearly between 89.64 psia at 320°F and 117.99 psia at 340°F, get the following for 114.7 psia:

$$s_f = 0.4872 \text{ Btu/(lb} \cdot ^\circ\text{F)}$$

$$s_g = 1.5918 \text{ Btu/(lb} \cdot ^\circ\text{F)}$$

$$h_f = 308.9 \text{ Btu/lb}$$

$$h_g = 1189.5 \text{ Btu/lb}$$

Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

$$1.5873 = x(0.4872) + (1 - x)(1.5918)$$

$$x = 0.0041$$

$$1 - x = 0.9959 \text{ (fraction vapor in exhaust)}$$

$$\text{Exhaust enthalpy} = (0.0041)(308.9) + (0.9959)(1189.5) = 1185.9$$

$$\text{Enthalpy change} = 1185.9 - 1261.8 = -75.9 \text{ Btu/lb}$$

Substituting 1 Btu = (hp · hr)/2544:

$$\text{Enthalpy change} = (-75.9/2544) = (-1/33.5)(\text{hp} \cdot \text{hr})/\text{lb}$$

TSR = the absolute value of the inverse of the enthalpy change = 33.5 lb/(hp · hr)

Basic efficiency = 66% (Fig. 15-13)

Inlet saturation temperature = 406.0 °F (interpolating between 260 and 280 psia on Fig. 24-31)

Inlet superheat = 500 – 406 (Fig. 24-31) = 94°F

Efficiency-correction factor for superheat = 0.99 (Fig. 15-15)

Efficiency-correction factor for speed = 1.01 (Fig. 15-16)

Pressure ratio = (114.7 psia)/(264.7 psia) = 0.433

Efficiency-correction factor for pressure ratio = 0.97 (Fig. 15-17)

$$\text{ASR} = [33.5 \text{ lb/(hp} \cdot \text{hr)}] / [(0.66)(0.99)(1.01)(0.97)] = 52.3 \text{ lb/(hp} \cdot \text{hr)}$$

$$F = [52.3 \text{ lb/(hp} \cdot \text{hr)}] (900 \text{ hp}) = 47,100 \text{ lb/hr}$$

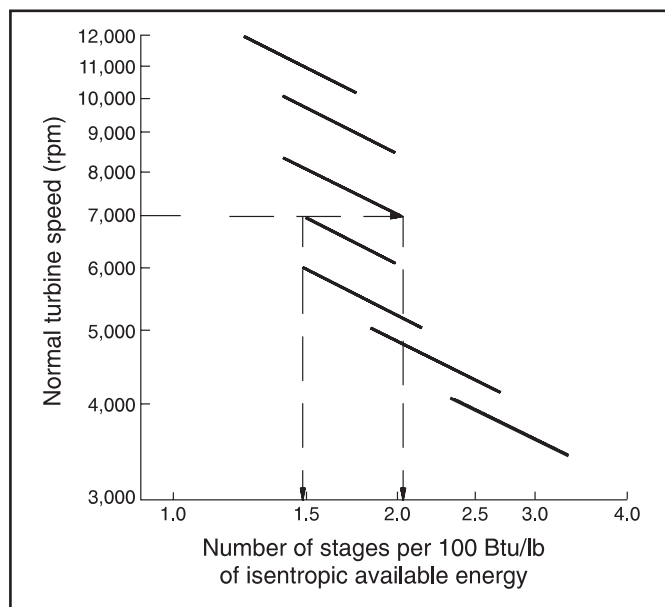
For a single-stage turbine

$$\text{ASR} = [75 \text{ lb/(hp} \cdot \text{hr)}] (0.93) \text{ (Fig. 15-19)} = 70 \text{ lb/(hp} \cdot \text{hr)}$$

$$F = [70 \text{ lb/(hp} \cdot \text{hr)}] (900 \text{ hp}) = 63,000 \text{ lb/hr}$$

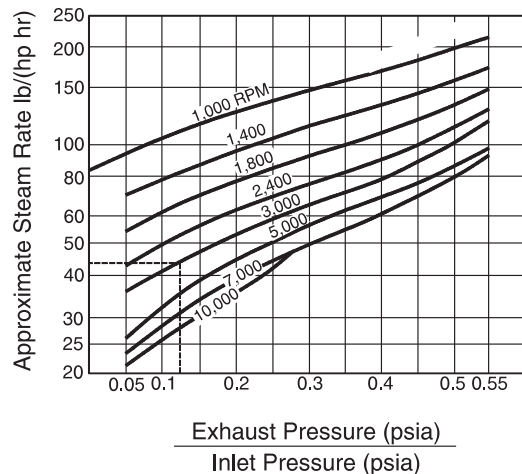
FIG. 15-19
Single-Stage Application

FIG. 15-18
Stages Required per 100 Btu/lb of Available Energy as a Factor of Normal Turbine Speed



Steam Rate Calculation

- 1 - Read approximate steam rate (below) at turbine exhaust pressure/inlet pressure and rated speed
- 2 - Read superheat correction (right) at turbine inlet superheat
- 3 - Multiply approximate steam rate from Step 1 by correction factor from Step 2 to obtain single stage steam rate.



Courtesy of General Electric

GAS TURBINES

General

Gas turbines are extensively used in all phases of the gas industry as a source of shaft power. They are used to drive compressors, generators, and other equipment required to produce, process, and transport natural gas. The main advantages of gas turbines are:

- Compact, light weight design
- Minimal maintenance
- Short installation time

Compact, Lightweight Design

The compact, lightweight design of gas turbines makes them ideally suited for offshore platform installations, portable generating sets, remote sites, or any application where size and weight are important considerations.

Maintenance

Once installed, the gas turbine requires a minimum of routine maintenance. It is important to monitor the operating parameters of the turbine (pressures, temperatures, speed, vibra-

tion levels, etc.). This can often be done by an operator at a location remote from the actual turbine installation.

Installation

The relatively light weight, compact size, and simple design of gas turbines make them an attractive choice where power must be quickly installed in the field. The gas turbine is often delivered on an integral one-piece baseplate with all auxiliary equipment installed and tested by the manufacturer. Thus, construction and start-up time are minimized.

GAS TURBINE TYPES

The gas turbine was first widely used as an aircraft power plant. However, as they became more efficient and durable, they were adapted to the industrial marketplace. Over the years the gas turbine has evolved into two basic types for high-power stationary applications: the industrial or heavy-duty design and the aircraft derivative design.

Heavy Duty

The industrial type gas turbine is designed exclusively for stationary use. Where high power output is required, 35,000 hp and above, the heavy duty industrial gas turbine is normally specified. The industrial gas turbine has certain advantages

FIG. 15-20
Typical Gas Turbine Skid Layout

