

Simplified Steam Rate Calculation

English Units

The steam rate determined from these calculations is a quick approximation, accurate to within $\pm 5\%$. For *guaranteed* steam rates, consult your Elliott Company Field Engineer or Representative.

Step 1: Determine THEORETICAL STEAM RATE (TSR) using the Elliott Polar Steam Rate Chart or Keenan & Keyes TSR tables published by ASME.

Step 2: Determine BASE STEAM RATE (BSR) using curves on Pages 10 and 11, based upon TSR, Turbine Speed, and Turbine Frame.

Step 3: Determine the MECHANICAL LOSS using curves on Pages 14 and 15, based upon Turbine Speed, Exhaust Pressure and Turbine Frame.

Step 4: Determine SUPERHEAT (use table at top of Page 18).

Step 5: Determine the SUPERHEAT CORRECTION FACTOR (SCF) using curve on Page 18, knowing Superheat ($^{\circ}\text{F}$) and TSR.

Step 6: Determine CORRECTED STEAM RATE by formula: $\text{Corrected Steam Rate} = \frac{\text{Base Steam Rate}}{\text{Superheat Correction Factor}} \times \frac{(\text{HP} + \text{HP Loss})}{\text{HP}}$

Step 7: Determine Steam Flow (SF) by formula: $\text{Steam Flow} = \text{Corrected Steam Rate} \times \text{HP}$.

Step 8: Determine the required INLET SIZE using the curves on page 20, knowing Steam Flow (SF) and Turbine Inlet Pressure and Temperature.

Step 9: Determine the required EXHAUST SIZE using the curves on page 20, knowing Steam Flow (SF) and Exhaust Pressure.

EXAMPLE

STEAM CONDITIONS
Inlet 250 PSIG
575 $^{\circ}\text{F}$
Exhaust 50 PSIG

DESIGN CONDITIONS
Turbine to Develop
500 Horsepower at
4000 RPM

The Simplified Steam Rate Calculations establish that, for most applications, the DYR turbine offers the lowest corrected Steam Rate (see below), the AYR the highest. The DYR is the most expensive turbine. However, the larger and more efficient turbine will usually have a lower evaluated cost if steam cost is considered.

Step 1: $\text{TSR} = 26.07 \text{ lb/kW-hr}$

Step 2: $\text{BSR} = 36$

Step 3: Mechanical Loss = 55 HP

Step 4: Superheat = 169°F

Step 5: $\text{SCF} = 1.050$

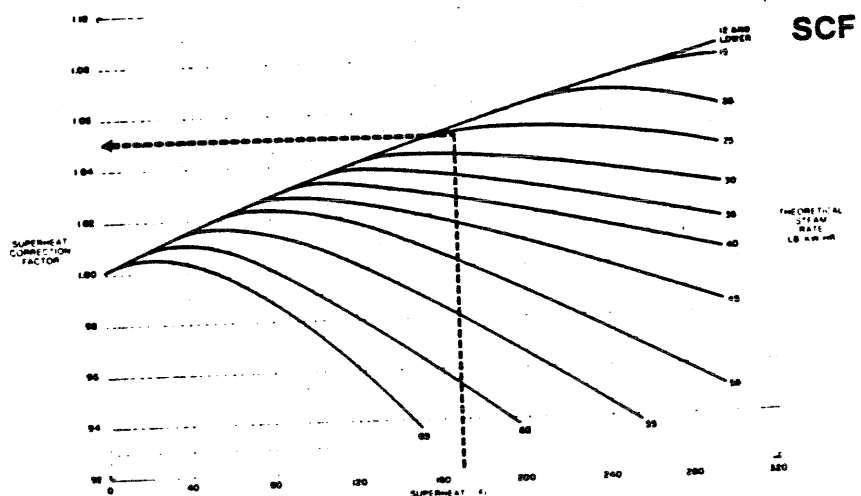
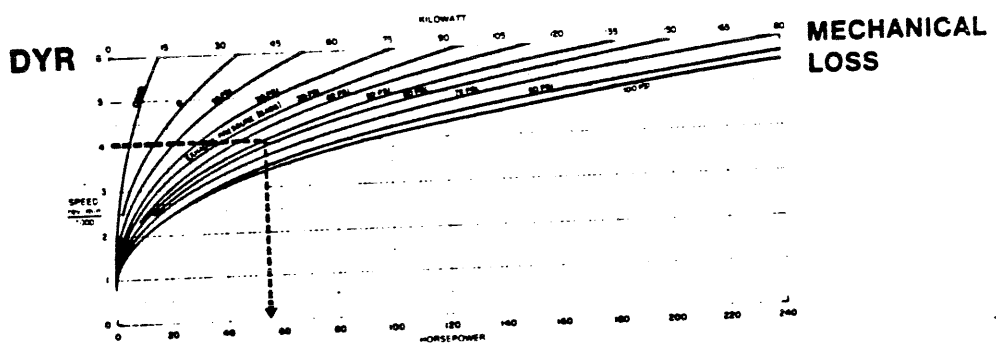
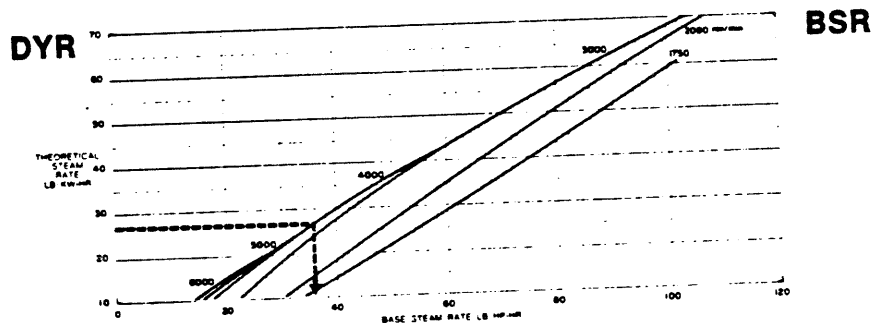
Step 6: $\frac{36}{1.05} \times \frac{(500+55)}{500} = 38.0 \text{ lb/HP-hr}$

Step 7: $\text{SF} = 38.0 \times 500 = 19,000 \text{ lb/hr.}$

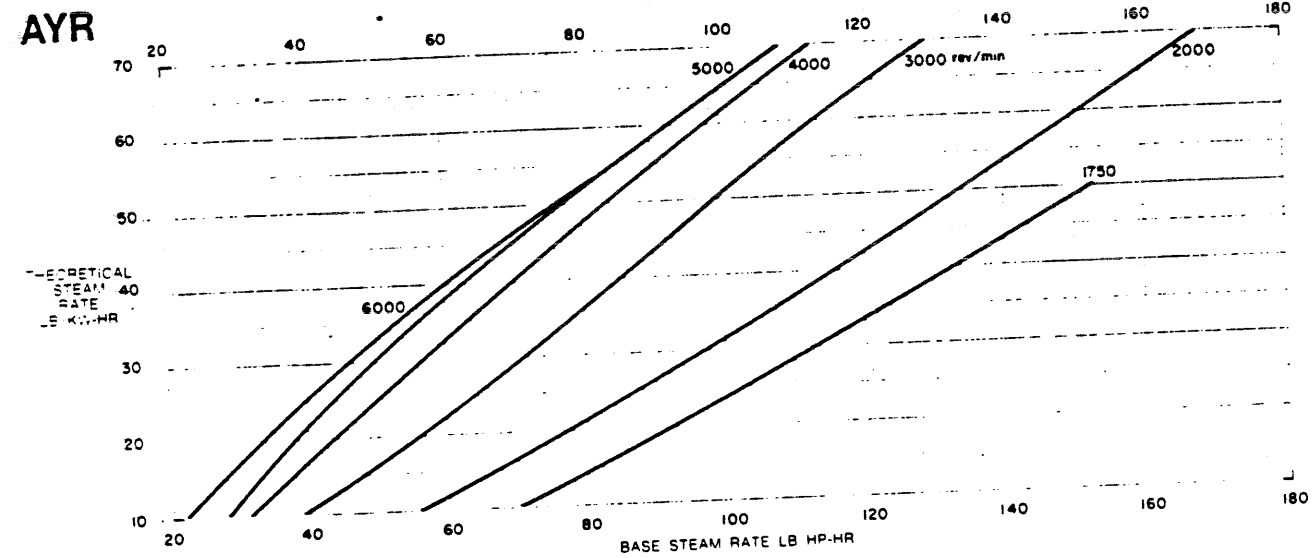
Step 8: Inlet Size = 4" ANSI

Step 9: Exhaust Size = 6" ANSI

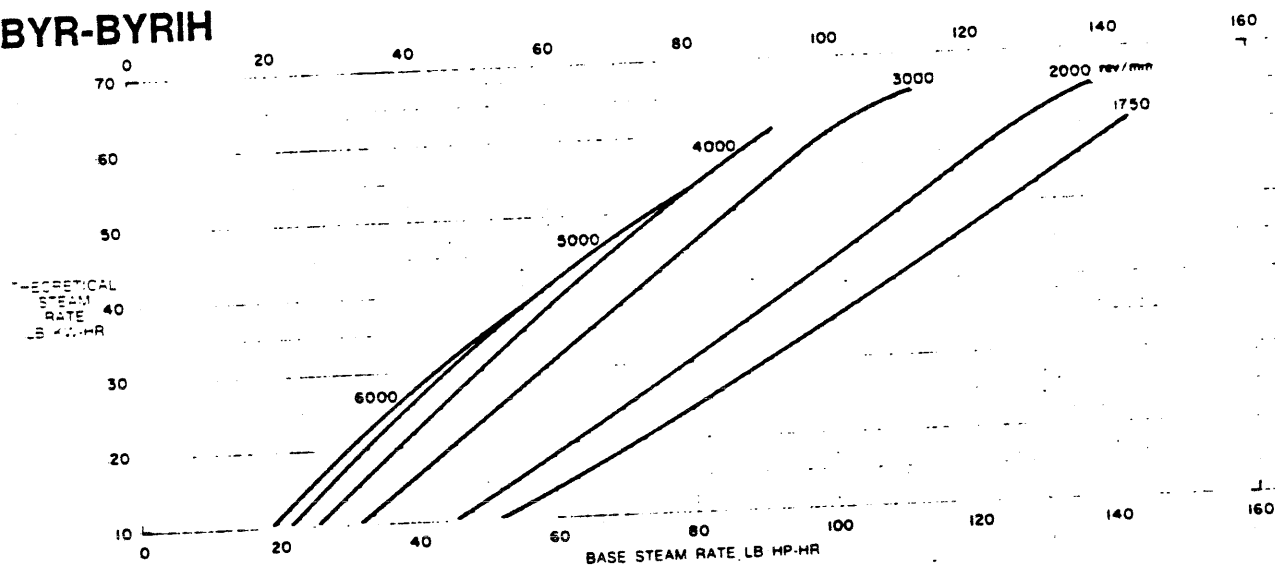
(Similar calculation for AYR turbine results in Corrected Steam Rate of 49.8 lb/hp-hr.)



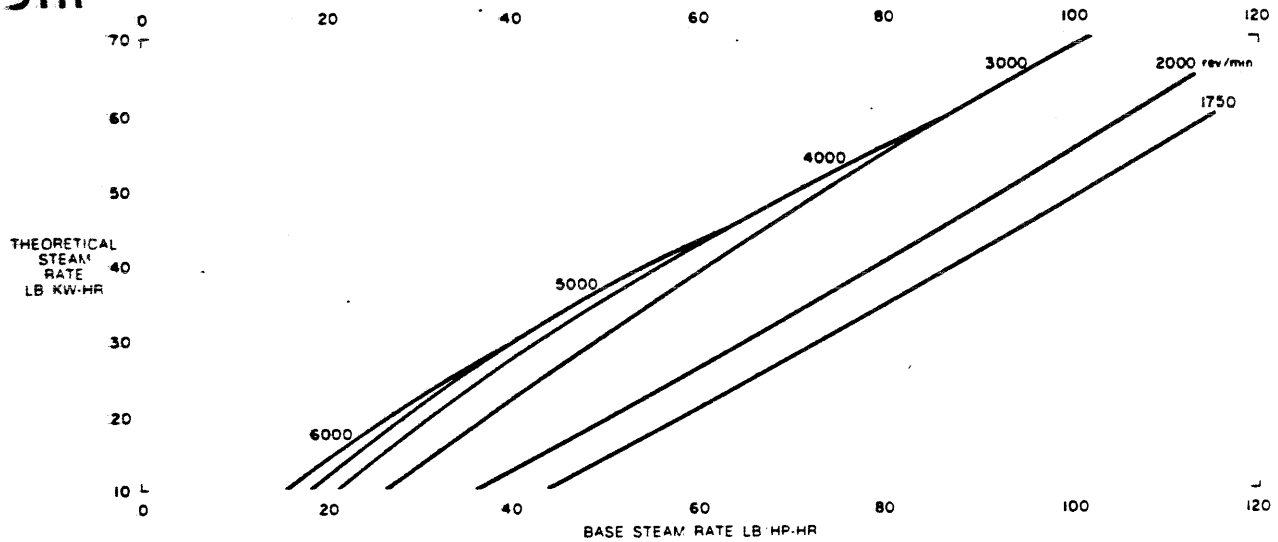
Base Steam Rates English Units



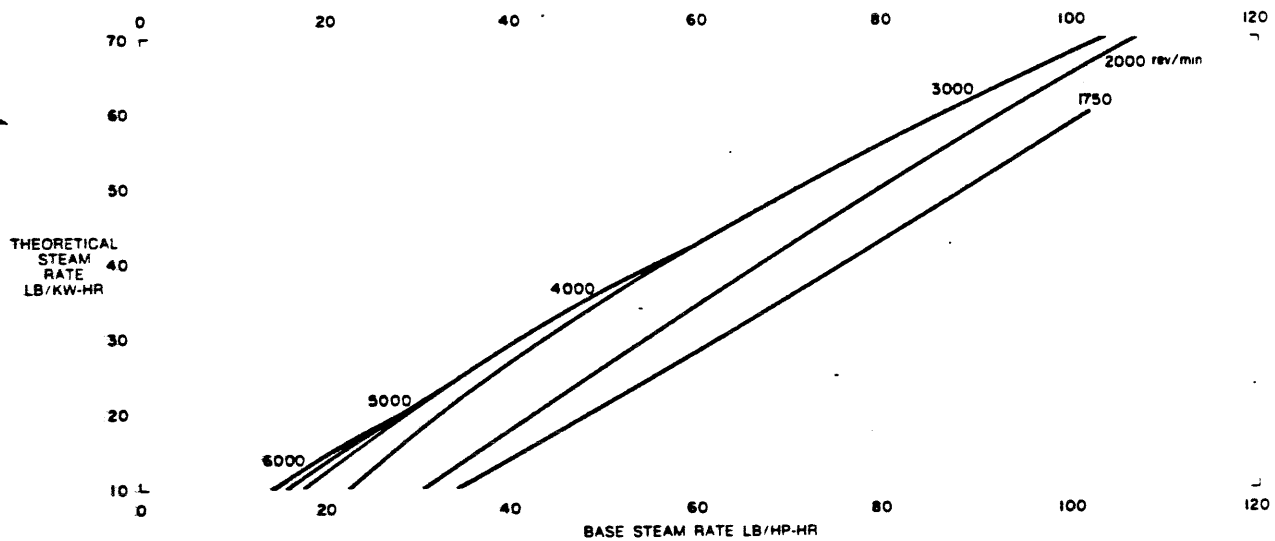
BYR-BYRIH



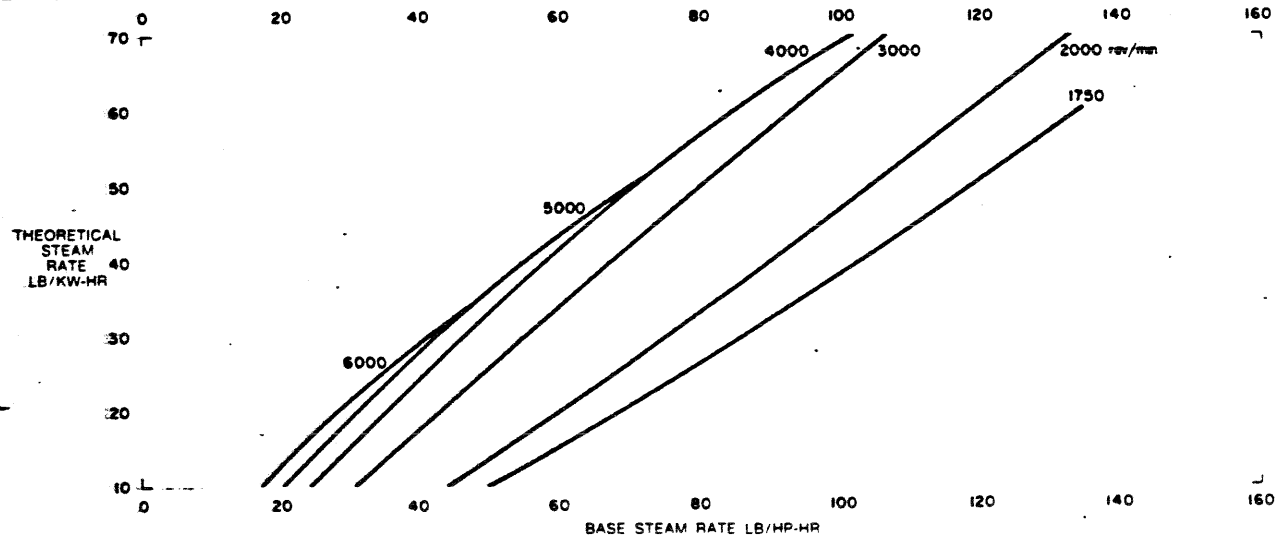
YR



DYR-DYRM

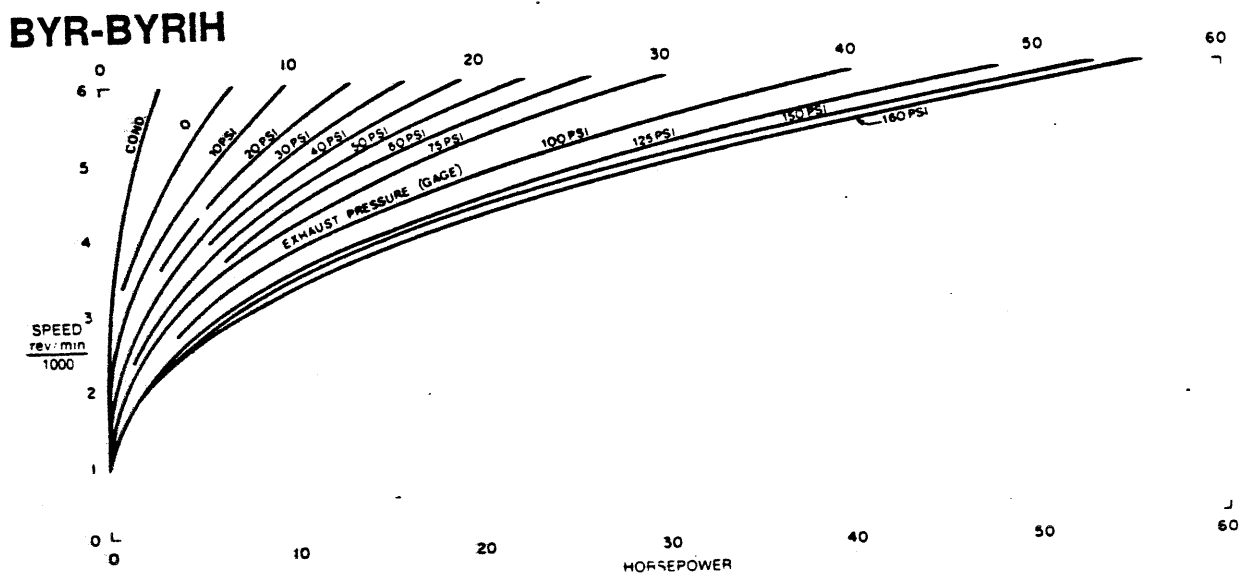
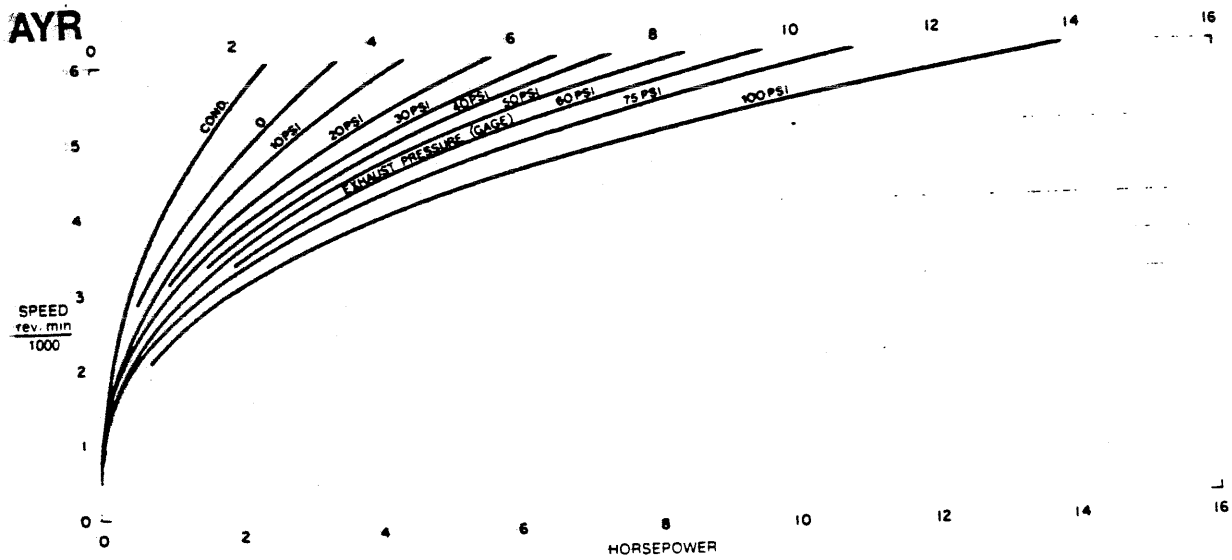


BYRH-BYRHH

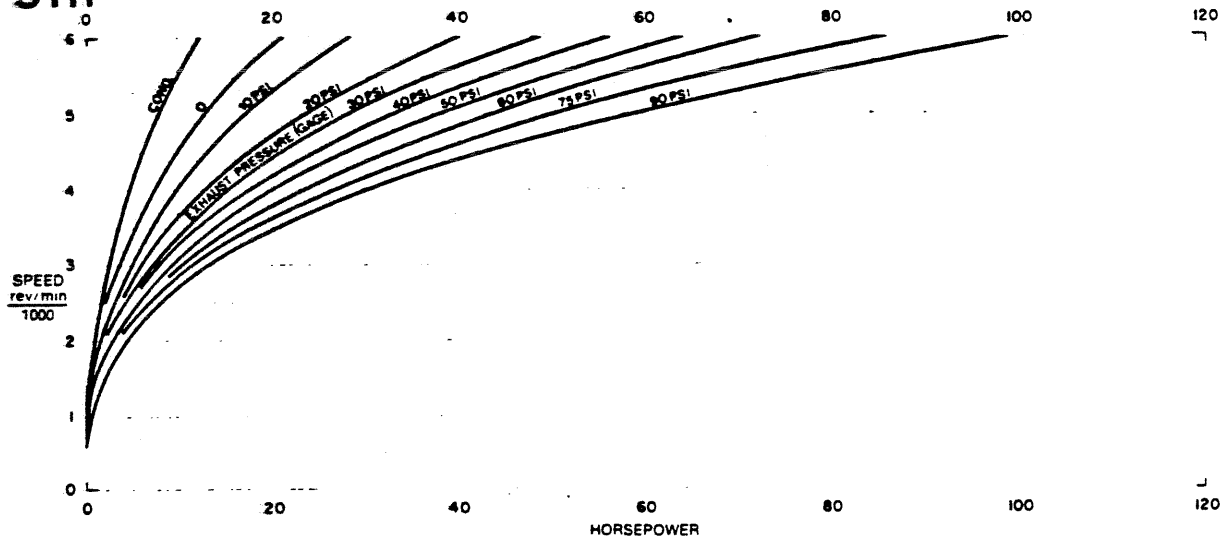


Mechanical Losses

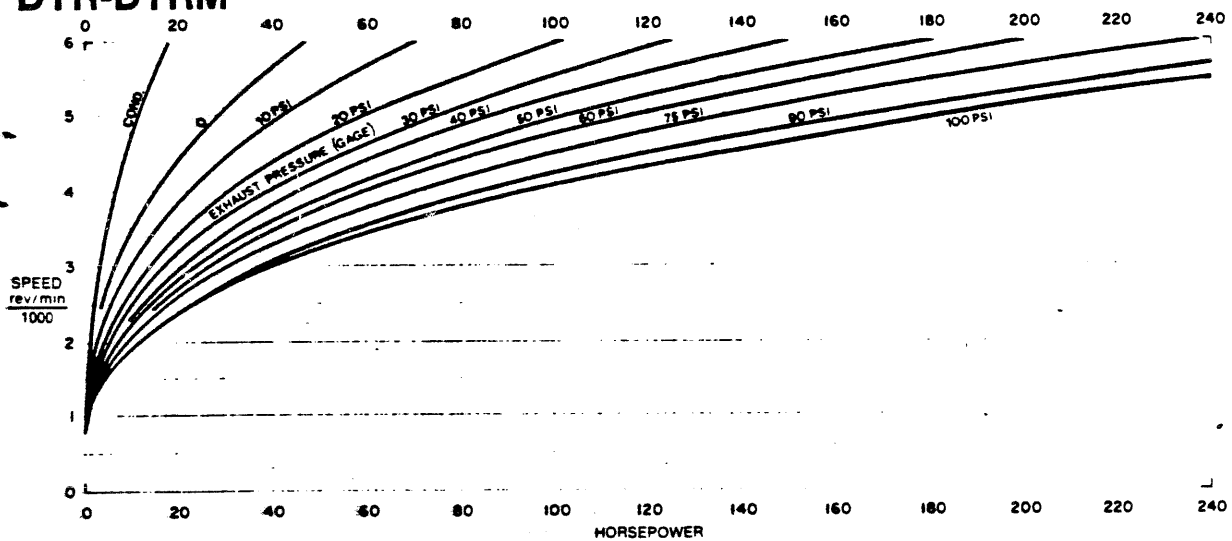
English Units



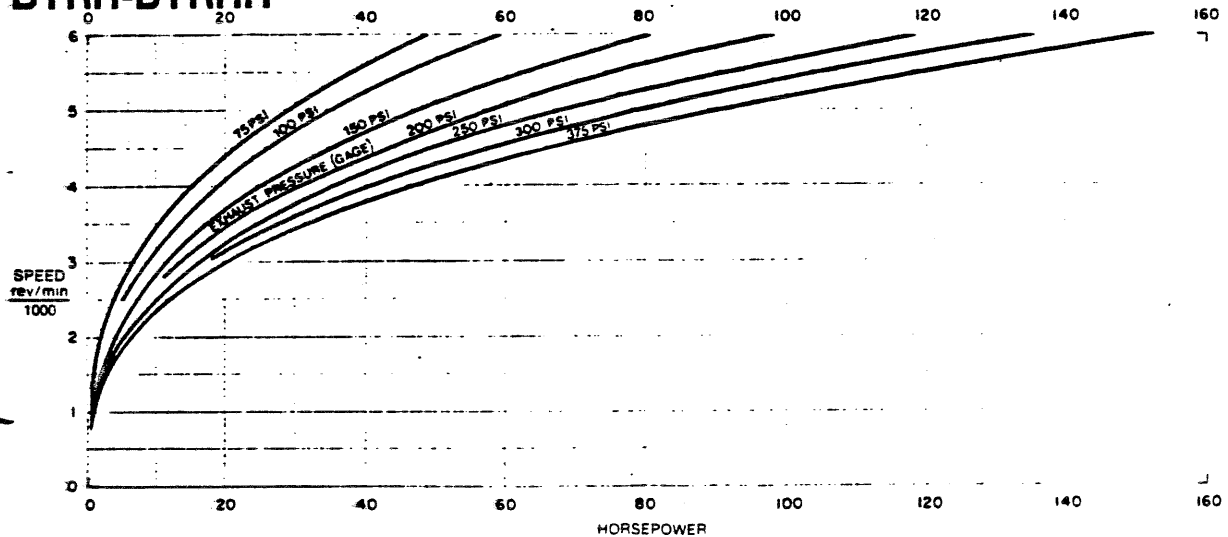
CYR



DYR-DYRM



BYRH-BYRHH



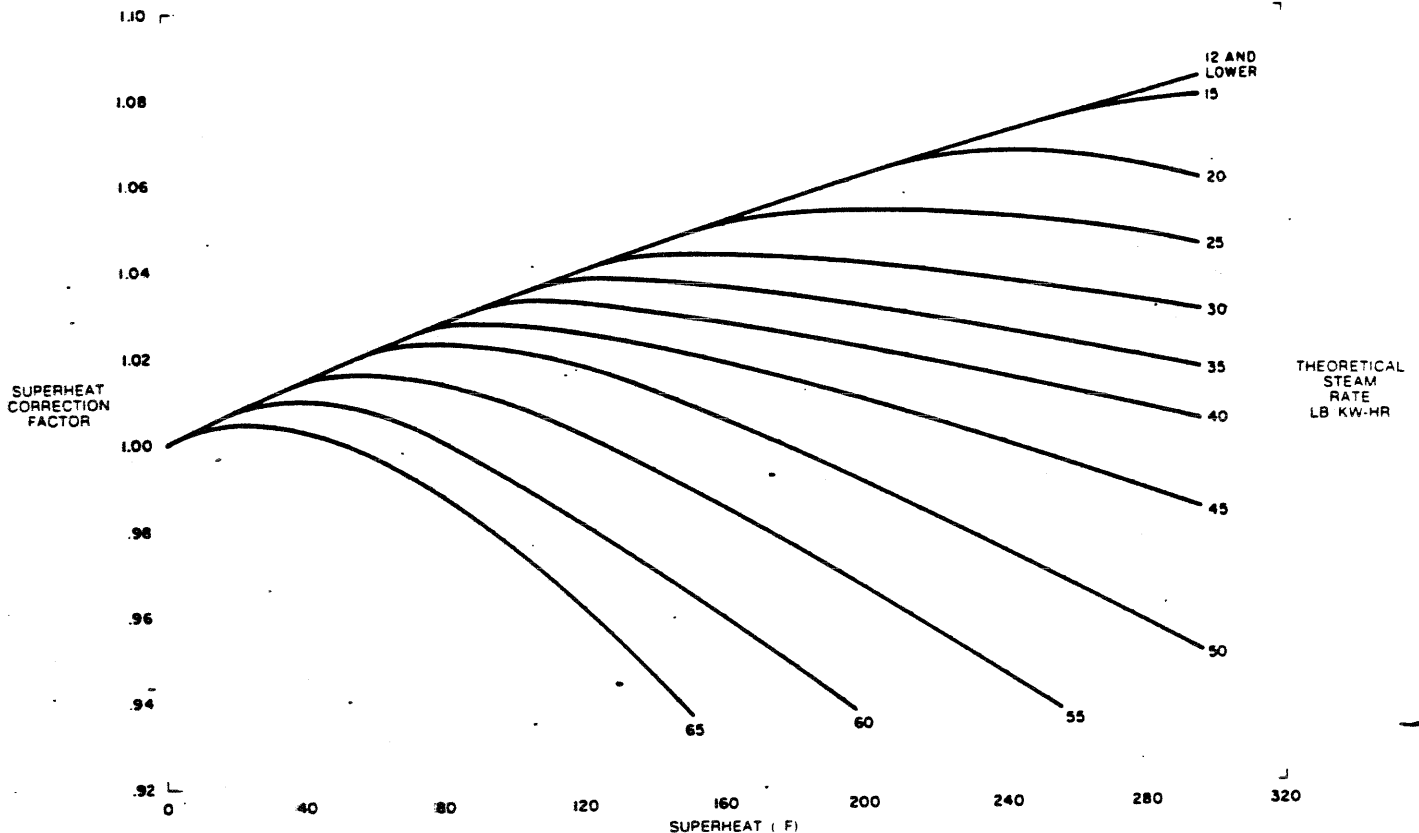
Superheat Correction Factor

English Units

Temperature of Dry and Saturated Steam

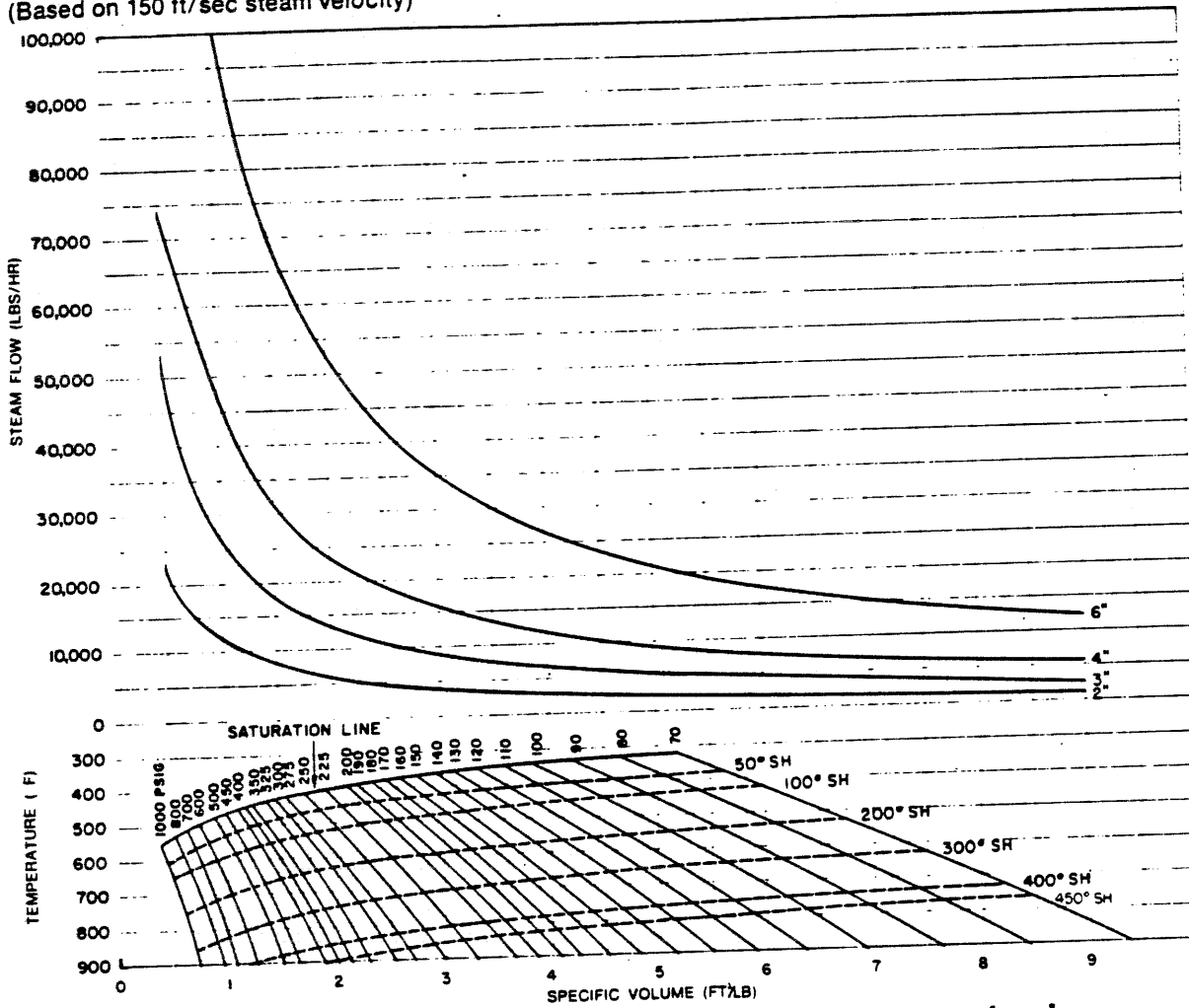
To obtain superheat in degrees F, subtract temperature given in tabulation below from total initial temperature.

PSIG.	Saturation Temp. °F	PSIG.	Saturation Temp. °F	PSIG.	Saturation Temp. °F	PSIG.	Saturation Temp. °F
0	212	150	366	300	422	450	460
5	228	155	368	305	423	455	461
10	240	160	371	310	425	460	462
15	250	165	373	315	426	465	463
20	259	170	375	320	428	470	464
25	267	175	378	325	429	475	465
30	274	180	380	330	431	480	466
35	281	185	382	335	432	485	467
40	287	190	384	340	433	490	468
45	293	195	386	345	434	495	469
50	298	200	388	350	436	500	470
55	303	205	390	355	437	510	472
60	308	210	392	360	438	520	474
65	312	215	394	365	440	530	476
70	316	220	396	370	441	540	478
75	320	225	397	375	442	550	480
80	328	230	399	380	444	560	482
85	328	235	401	385	445	570	483
90	331	240	403	390	446	580	485
95	335	245	404	395	447	590	487
100	338	250	406	400	448	600	489
105	341	255	408	405	449	610	491
110	344	260	410	410	451	620	492
115	347	265	411	415	452	630	494
120	350	270	413	420	453	640	496
125	353	275	414	425	454	650	497
130	356	280	416	430	455	660	499
135	358	285	417	435	456	670	501
140	361	290	419	440	457	680	502
145	364	295	420	445	458	690	504



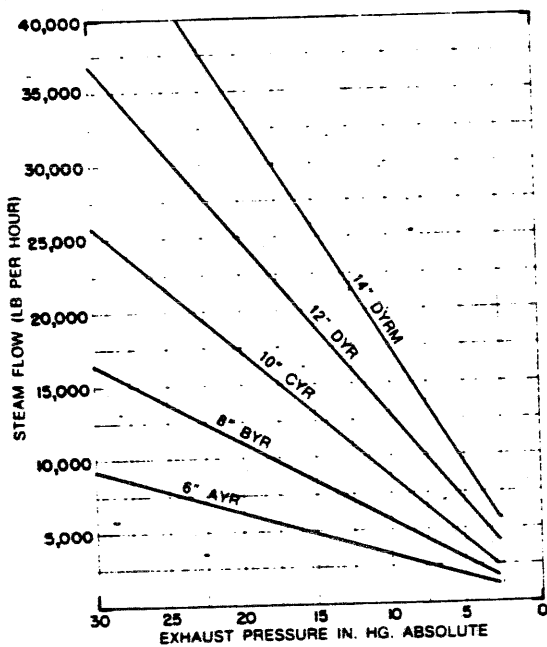
Steam Flow English Units Inlet Size

Read inlet size required to pass maximum flow
(Based on 150 ft/sec steam velocity)



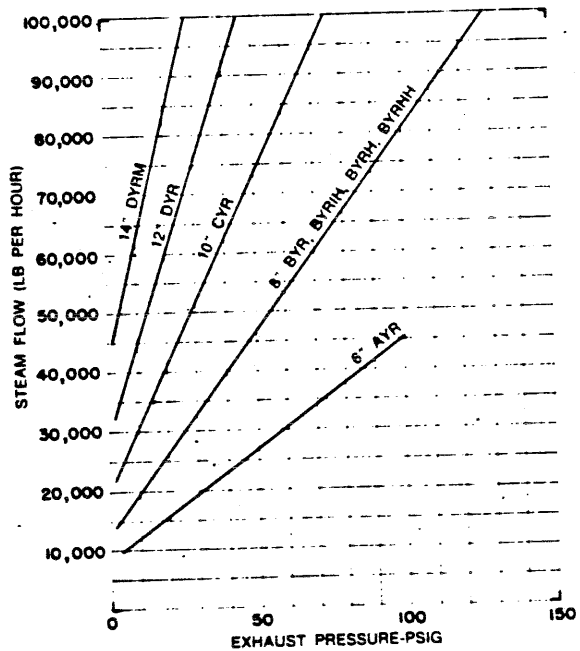
Exhaust Size—Condensing

(Based on 350 ft/sec steam velocity)



Exhaust Size—Non-condensing

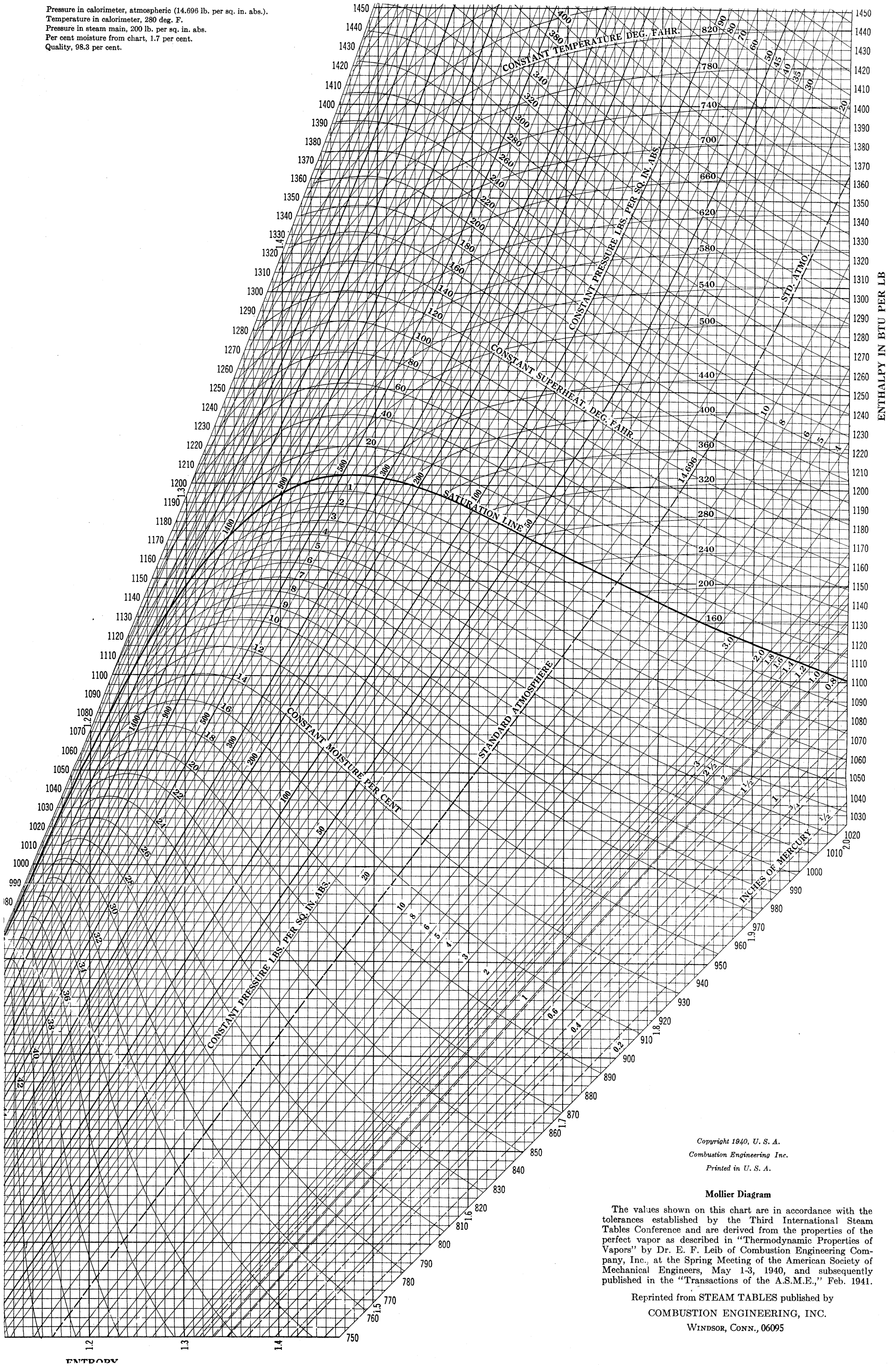
(Based on 250 ft/sec steam velocity)



Source: Fig. 21 of the ASME Steam Tables, Fourth Edition. Reproduced by permission of the publisher, The American Society of Mechanical Engineers, 345 E. 47th St., New York, N.Y.



Pressure in calorimeter, atmospheric (14.696 lb. per sq. in. abs.).
Temperature in calorimeter, 280 deg. F.
Pressure in steam main, 200 lb. per sq. in. abs.
Per cent moisture from chart, 1.7 per cent.
Quality, 98.3 per cent.

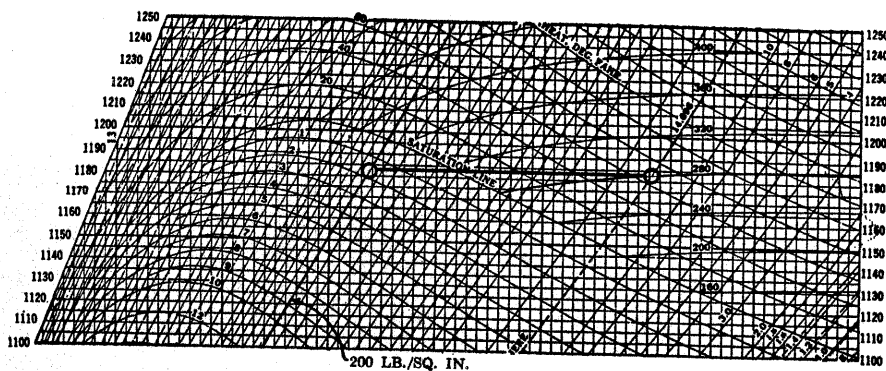


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Mollier Diagram

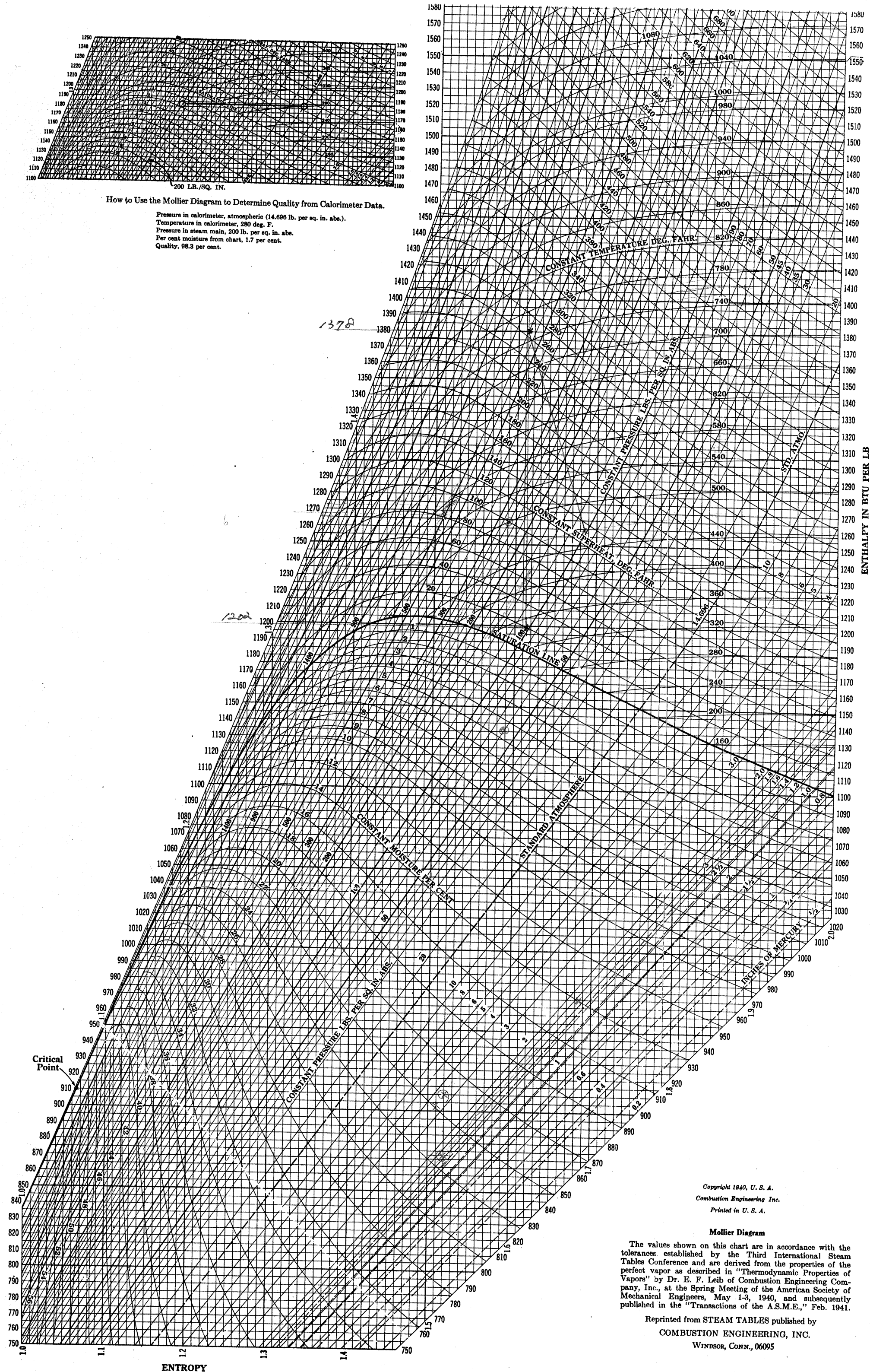
The values shown on this chart are in accordance with the tolerances established by the Third International Steam Tables Conference and are derived from the properties of the perfect vapor as described in "Thermodynamic Properties of Vapors" by Dr. E. F. Leib of Combustion Engineering Company, Inc., at the Spring Meeting of the American Society of Mechanical Engineers, May 1-3, 1940, and subsequently published in the "Transactions of the A.S.M.E.," Feb. 1941.

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How to Use the Mollier Diagram to Determine Quality from Calorimeter Data.

Pressure in calorimeter, atmospheric (14.696 lb. per sq. in. abs.).
 Temperature in calorimeter, 280 deg. F.
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STEAM TURBINES

ENGINEERING DESIGN SEMINAR
TWENTY-EIGHTH SESSION - FALL 1991

I. GENERAL

- A. APPLICATION
- B. TYPES

II. THEORY

- A. THERMODYNAMICS
- B. STAGE DESIGN
- C. EFFICIENCY
- D. VELOCITY LIMITS

III. HARDWARE

- A. CASINGS
- B. BLADES
- C. BEARINGS
- D. SEALS
- E. CONTROL/PROTECTION

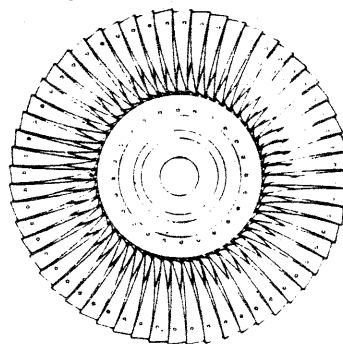
IV. SAMPLE PROBLEMS

GENERAL

APPLICATION
AND
TYPES

Reprint 133A
February 1976

Turbine Selection



*Elliott
Multistage
Steam
Turbines*

Turbine Selection

Steam turbines are designed to meet the varied needs of the process industry for prime movers to drive generators, fans, pumps, centrifugal compressors and various other machines. Turbines to drive generators are constant speed applications and have different mechanical considerations and will be omitted from this discussion.

The reasons for using steam turbines and the relative advantages are described as follows:

1. Thermodynamic

- A. It can translate into mechanical work a large percentage of the heat energy available to the turbine.
- B. It has the capability of producing a high output at a good efficiency level.
- C. It has the potential of good efficiency levels at reduced load ranges.
- D. Automatic extraction and/or induction designs provide thermodynamic flexibility in process heat balances, resulting in low-cost power.
- E. Process temperature control for a given set of conditions can be maintained (acts as reducing valve.)

2. Mechanical

- A. The turbine is designed so that the propulsion force is applied directly to the rotating element, not as in reciprocating engines where the mechanical work must be transmitted through a series of links converting reciprocating motion to rotary motion.
- B. Turbine governors permit adjustable speeds for process flexibility.
- C. The governor automatically responds to process variables (inlet pressure, inlet temperature, load, etc.) to hold a speed setpoint.
- D. Turbines can be installed readily in explosive atmospheres.
- E. High speeds result in elimination of a gear.

3. Operational

- A. Contains few moving parts requiring minimum maintenance.
- B. Operates at relatively low vibration level.
- C. Relatively quiet.
- D. Not affected by local utility's power failures.
- E. Can be operated on overload in many cases.

F. Uprate capability.

There are four basic types of turbines for the process industry:

1. Condensing Type

A minimum amount of high-pressure steam generation required. Steam power requirements greater than process steam demand.

2. Back-Pressure Type

Various steam pressure levels required by the process. Process steam requirements exceed steam power demand. Can be used where condensing water is unavailable and/or insufficient.

3. Extraction Type

Intermediate steam pressure level required with variation in flow.

4. Induction Type

Excess steam exists at intermediate pressure level.

Steam Evaluation

For many steam turbine applications more than one selection is possible. For instance, a turbine with a higher purchase price will usually use less steam than a less expensive unit because the higher-priced turbine will have more and/or higher efficiency staging.

Therefore, it is necessary to compare the total cost (first cost plus operating cost) for each turbine under consideration.

The example on the opposite page makes such a comparison between two turbines. In this case, a single valve and a multivalve turbine will be considered for the same application. You will notice that the single-valve unit has a lower first cost but higher operating costs due to lower efficiency, especially during part-load operations (see Reprint #135, Figure 2).

Only after comparing the *total* cost for each unit can a proper selection be made. In the example given it is assumed these units will be equally reliable for this application. This can only be determined, however, after carefully reviewing such application parameters as nozzle flow limits, blade loading, critical speeds, thrust, etc.

Elliott Reprint #123A discusses turbine application and experience in more detail.

System steam balances and economic evaluations are also important in proper utilization of steam turbines. Figures 1, 2, 3 and 4 show various approaches

to steam turbine application in the process plant steam systems.

Figure 1. One non-condensing turbine used as a topping turbine. Exhaust steam used for driving two other turbines plus process steam.

Figure 2. Two extraction turbines extracting at two different pressure levels (process steam) and one turbine straight condensing.

Figure 3. Two extraction turbines extracting and one pressure level and two different flows. Lower pressure level used for driving condensing turbine and process steam.

Figure 4. One turbine non-condensing used as topping turbine but mechanically connected to another turbine. Exhaust steam feeds four other turbine applications and process.

	English units	Metric units
An Example:		
Speed 9000 rev/min		
Power	3000 HP	2237 kW
Steam Conditions	600 PSIG - 750°F - 4" Hga	41.4 bar (gage) - 399°C - 135 mbar (abs)
Steam Cost	\$1.50 per 1000 pounds	\$3.31 per 1000 kg
Payout Period 3 years		

Selection 1—Single-Valve Multistage Turbine

Model SBPG 5

SR 8.40 #/HP HR 5.11 kg/kWh

Price is \$154,500*

Selection 2—Multivalve, Multistage Turbine

Model SRV 6

SR 7.93 #/HP HR 4.82 kg/kWh

Price is \$270,000*

*Same Bill of Material, except one versus five governor valves

The operating cost per year can be calculated as follows:

Annual operating cost = (SR) x (Power) x (Operating Time) x (Steam Cost)

Where SR—steam rate,

Steam Cost = \$1.50/1000# \$3.31/1000 kg

Operating Time = 8000 hours/year

Selection 1

Operating Cost/Year	$\frac{(8.40 \times 3000) (8000) (1.50)}{1000}$	$\frac{5.11 \times 2237 \times 8000 \times 3.31}{1000}$
	= \$302,000/year	= \$302,000/year

Selection 2

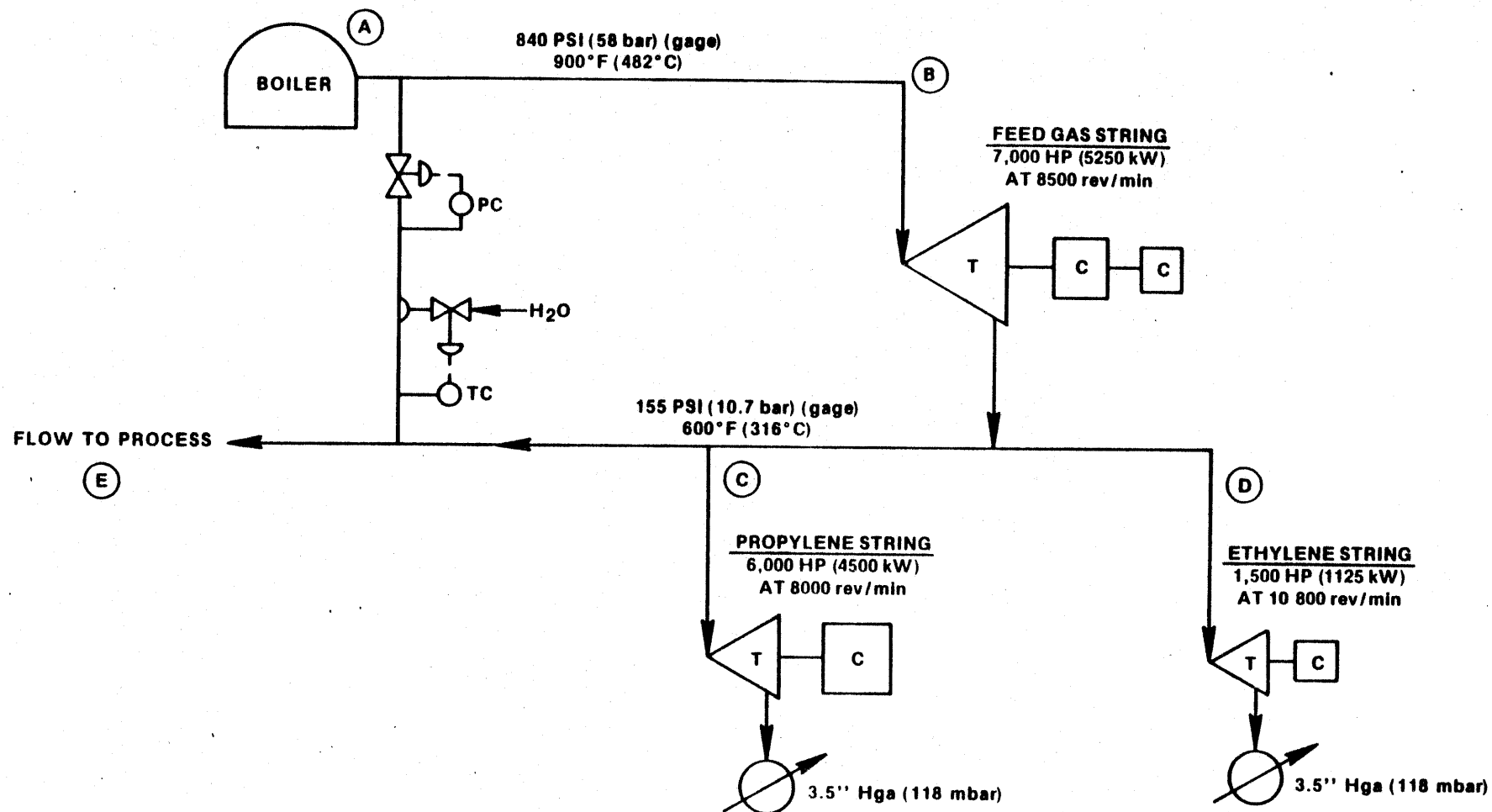
Operating Cost/Year	$\frac{(7.93 \times 3000) (8000) (1.50)}{1000}$	$\frac{4.82 \times 2237 \times 8000 \times 3.31}{1000}$
	= \$285,000/year	= \$285,000/year

Total Investment - Initial Cost + Operating Cost

Selection 1—\$154,500 + \$906,000 = \$1,060,500

Selection 2—\$270,000 + \$855,000 = \$1,125,000

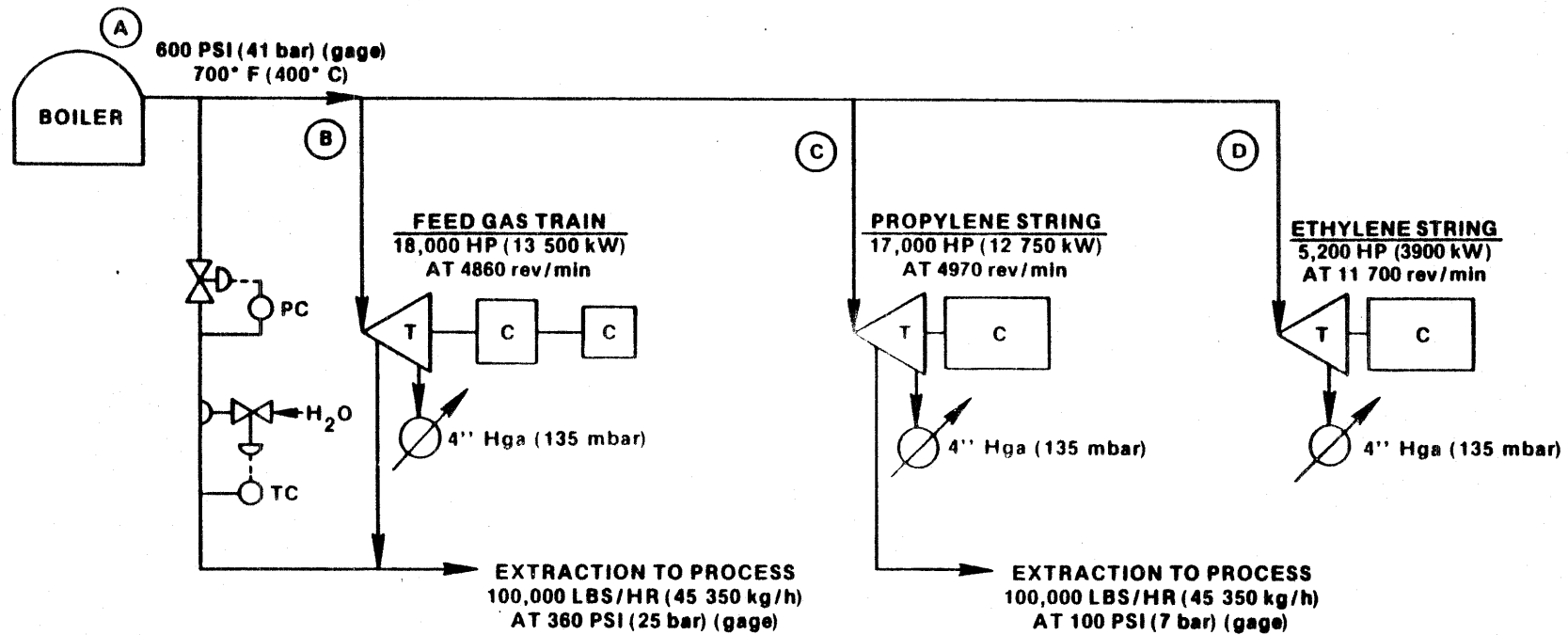
**FIGURE 1 ETHYLENE PLANT TYPICAL STEAM BALANCE
200 MILLION POUNDS (90 700 tonnes) PER YEAR**



FLOW	C	65,000 LBS/HR (29 500 kg/h)
FLOW	D	16,000 LBS/HR (7250 kg/h)
FLOW	E	64,000 LBS/HR (29 000 kg/h)

TOTAL BOILER FLOW	A	145,000 LBS/HR (65 750 kg/h)
FLOW	F	145,000 LBS/HR (65 750 kg/h)

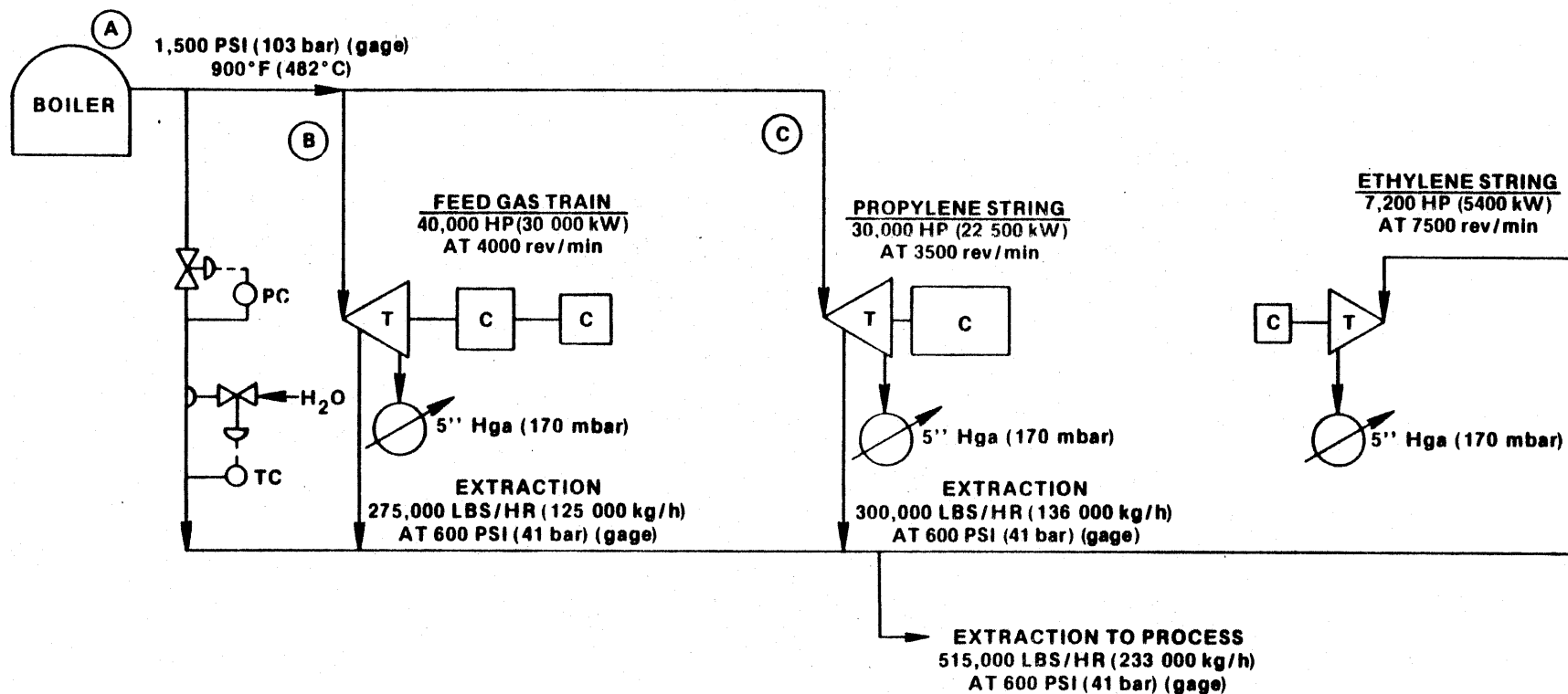
FIGURE 2 ETHYLENE PLANT TYPICAL STEAM BALANCE
500 MILLION POUNDS (225 000 tonnes) PER YEAR



FLOW B 240,000 LBS/HR (108 800 kg/h)
 FLOW C 198,000 LBS/HR (89 800 kg/h)
 FLOW D 41,500 LBS/HR (18 800 kg/h)

TOTAL BOILER FLOW A 479,500 LBS/HR (217 400 kg/h)

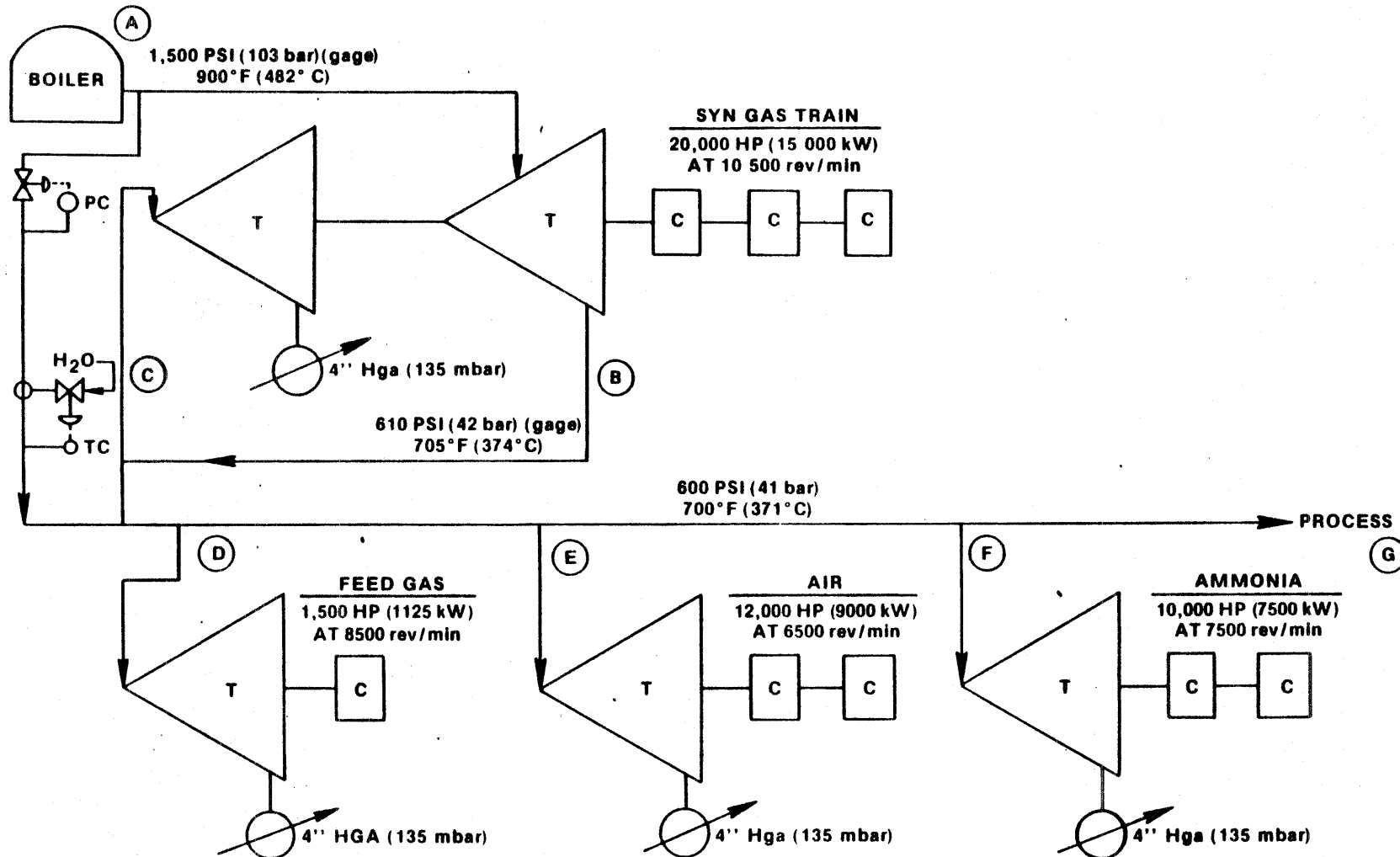
**FIGURE 3 ETHYLENE PLANT TYPICAL STEAM BALANCE
1.2 BILLION POUNDS (550 000 tonnes) PER YEAR**



FLOW B 440,000 LBS/HR (200 000 kg/h)
 FLOW C 410,000 LBS/HR (185 000 kg/h)

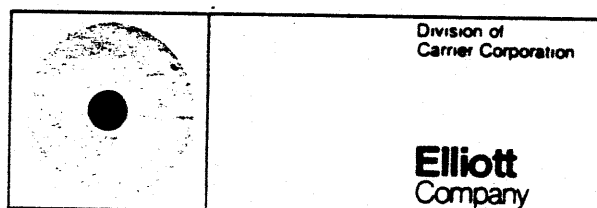
TOTAL BOILER FLOW A 850,000 LBS/HR (385 000 kg/h)

FIGURE 4 AMMONIA PLANT TYPICAL STEAM BALANCE
1,200 TONS (1 090 tonnes) PER DAY



FLOW	C	17,000 LBS/HR (7650 kg/h)
FLOW	D	14,000 LBS/HR (6350 kg/h)
FLOW	E	94,000 LBS/HR (42 600 kg/h)
FLOW	F	77,000 LBS/HR (35 000 kg/h)
FLOW	G	379,000 LBS/HR (171 900 kg/h)

TOTAL BOILER FLOW	A	581,000 LBS/HR (263 500 kg/h)
FLOW	B	581,000 LBS/HR (263 500 kg/h)



- | | |
|-----------------------------|---------------------|
| ■ Mechanical Drive Turbines | ■ Tube Cleaners |
| ■ Air and Gas Compressors | ■ Tube Expanders |
| ■ Turbochargers | ■ Roller Burnishers |

Elliott Company General Offices:

Jeannette, Pennsylvania 15644
Other plants at Dayton and Springfield, Ohio;
Isle of Wight, England

Canada: Elliott Industrial Equipment Company
24 Queen St. East, Brampton, Ontario

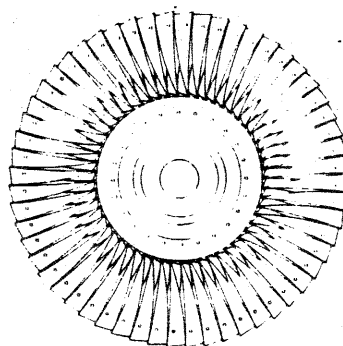
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Latin America: Elliott Overseas Company
1570 Madruga Avenue
Coral Gables, Florida 33146

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P.O. Box 1112, City of Industry, California 91749

Reprint 163
March 1976

Steam Turbine Nomenclature



*Elliott
Multistage
Steam
Turbines*

Elliott Steam Turbine Nomenclature

The following tables describe the frame nomenclature for Elliott multistage turbines. Please note that the data is general in nature and is to be used for general information only.

A. Basic Frame Designation

Nominal Wheel Diameters	Frame Size & Nominal Power/Speed Range	
	SVMS	MVMS
15" (380mm)		J-Line 10,000-30,000 HP (7500-22 500kW) 10,000-13,000 rev/min
18" (460mm)	B-Line 1000-8000 HP (750-6000kW) 8000-12,000 rev/min	R-Line 2500-12,000 HP (1850-9000kW) 8000-12,000 rev/min
25" (635mm)	E-Line 1000-8500 HP (750-6350kW) 3000-8500 rev/min	Q-Line 5000-15,000 HP (3700-14 000kW) 4000-8500 rev/min
25" & 34" (635 & 860mm)		N-Line 10,000-60,000 HP (7500-45 000kW) 3000-6000 rev/min

B. Steam Conditions

Frame Size	Pressure (Gage)		Temperature	
	psig	bar	°F	°C
E-Line	900	62	900	482
B-Line	700	48	825	440
R-Line	2000	138	950	510
Q-Line	650	45	825	440
N-Line	2000	138	950	510
J-Line	1500	103	900	482

C. Construction Characteristics

Nomenclature	Construction Item	Frame Size				
		SVMS		MVMS		
		B	E	R	Q	N
S	Solid Rotor	X	X	X	X	X
DF	Double-Flow Exhaust	X	—	X	X	X
V	Bar-Operated Governor Valves	N/A	N/A	X	X	X
C	Cam-Operated Governor Valves	N/A	N/A	—	—	X

X Available

N/A Not Applicable

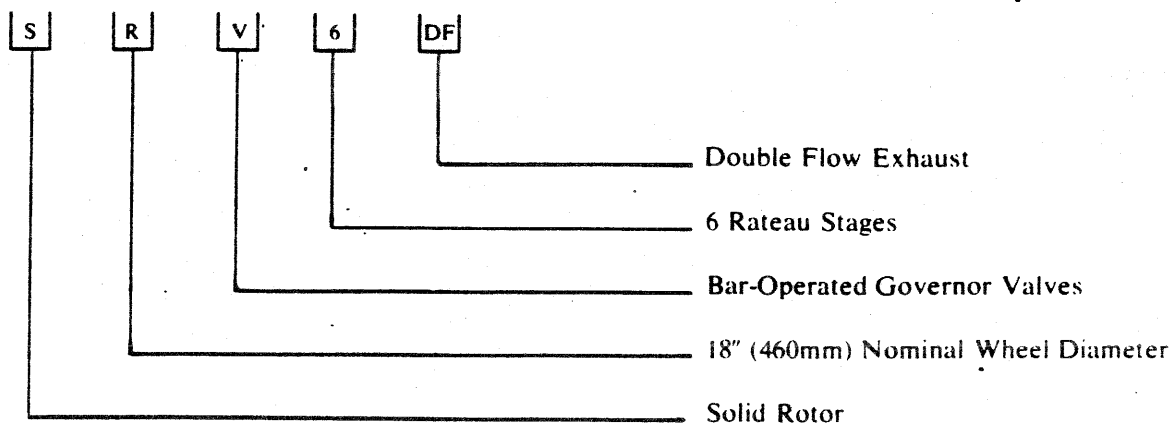
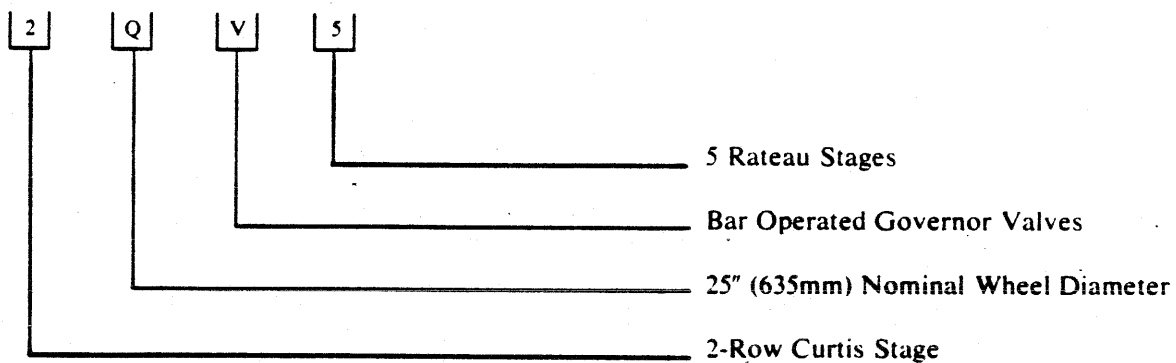
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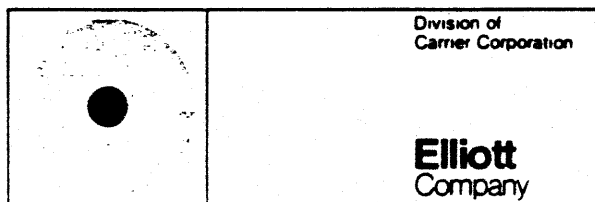
D. Staging

Curtis — The numbers preceding the frame designation indicate the number of rotating wheels. A single Curtis stage (2 rows of rotating blades) is designated by a 2 preceding the frame designation.

Rateau— The number following the frame designation indicates the number of Rateau stages.

EXAMPLES





- Mechanical Drive Turbines
- Air and Gas Compressors
- Turbochargers
- Tube Cleaners
- Tube Expanders
- Roller Burnishers

Elliott Company General Offices:

Jeannette, Pennsylvania 15644

Other plants at Dayton and Springfield, Ohio;

Isle of Wight, England

Canada: Elliott Industrial Equipment Company

24 Queen St. East, Brampton, Ontario

Europe, Middle East, Africa:

Elliott Turbomachinery Ltd.

197 Knightsbridge, London, England

Latin America: Elliott Overseas Company

1570 Madruga Avenue

Coral Gables, Florida 33146

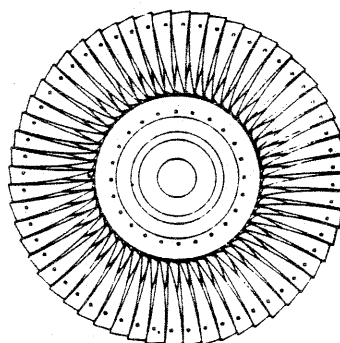
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Reprint 134A
March, 1976

Thermodynamics



***Elliott
Multistage
Steam
Turbines***

THERMODYNAMICS

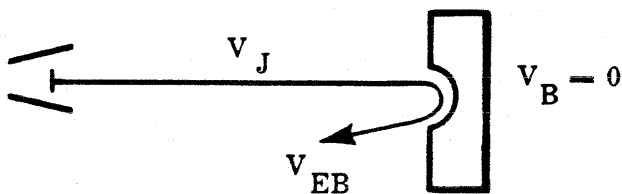
A steam turbine operates by utilizing heat energy which it abstracts from the steam and which it converts into mechanical work. Heat and mechanical work are both forms of energy, and, therefore, can be converted from one to the other.

First, heat energy is converted into velocity (kinetic) energy. In this first step, steam expands in a nozzle discharging at high velocity. The total heat (enthalpy) of the steam is converted to kinetic energy (velocity). This can readily be shown by the general energy equation. (See Addendum A for derivation and details.)

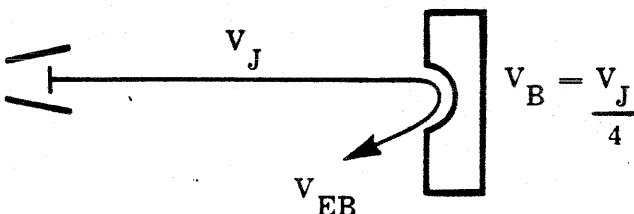
$$V_J = 224 \sqrt{\Delta h_{is}}$$

Second, a steam turbine does mechanical work by virtue of the steam velocity which strikes moving blades. There are two types of stages to convert this velocity energy into usable work, namely, impulse and reaction. With an impulse design, the pressure drop is taken across the nozzles only, whereas with reaction design the drop is evenly distributed across the nozzle and blades. If the nozzle is fixed and the force of the jet is directed against a crescent shape (the blade), the jet's IMPULSE force pushes the blade in the direction of the jet. Further, the leaving velocity is less than the initial jet velocity. Such an example would be a water hose jetting water against a wall. If the nozzle is free to move, the reaction pushes against the nozzle and it will travel in a direction opposite to the jet. The leaving velocity is greater than the jet velocity. An example would be a revolving lawn sprinkler.

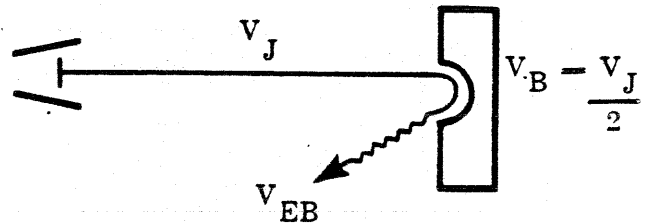
Focusing on the impulse principle, it can be shown how work and force are related. See Addendum B for derivation and details.



(a) Steam jet exerts maximum force on locked impulse bucket, but no work is done.



(b) When bucket moves at $\frac{1}{4}$ jet speed, the steam jet force diminishes. Steam leaves the bucket at low speed but does some work.



(c) When bucket moves at $\frac{1}{2}$ jet speed, steam jet force drops to $\frac{1}{2}$ the locked condition. Steam exits the bucket at zero speed but does maximum work.

Basic characteristics of impulse type turbines and reaction type turbines are summarized below:

Impulse (Standard Domestic Design)

1. Large bucket clearances permit quicker loading with minimal danger from thermal stress.
2. Since most of the pressure drop is in stationary nozzles, efficiency does not depend on blade clearances.
3. Low thrust bearing load.

Reaction

1. Expansion takes place in stationary nozzles and moving blades.
2. More bucket sealing (sealing strips) is required at end of blades.
3. Efficiency is theoretically greater but depends on close clearances.
4. The so-called greater efficiency of a reaction machine over an impulse machine in practice becomes a function of the staging (number of stages, efficiency of stages available, etc.) A reaction machine will have more stages because to keep losses to a minimum, the pressure drop per stage must be kept low.
5. Reaction turbines require larger thrust bearings and/or a balancing piston.

Turbine Staging

Staging in a steam turbine is the division of the process of converting heat energy into mechanical work.

There are these basic classes of staging which are defined as follows:

1. **Velocity-Compounded (Curtis) Staging —**

This type of impulse staging converts the heat energy of the steam into mechanical

work by using stationary impulse jet nozzles to direct the steam against one row of moving blades, then redirect the steam through reversing buckets (not nozzles) to an additional row of rotating blades.

The steam velocity is absorbed in a series of constant pressure steps. The large pressure drop through the nozzle results in a high-speed steam jet and the dual rotating rows of blades permit effective use of the high steam jet velocity while keeping wheel diameter small and the blade tip speeds to a minimum. At the expense of overall efficiency, Curtis staging permits fewer overall stages and a shorter, more rugged turbine for a given rating. Curtis stages are generally used for inlet stages and occasionally built with two Curtis stages. Figure #1A shows a typical two-row Curtis Stage.

2. Pressure Compounded (Rateau) Staging—

Here, the heat energy of the steam is converted into work by employing stationary impulse jet nozzles to direct the steam against the following row of moving blades.

Similar stationary nozzles (or diaframs) follow each row of moving blades except the last. Pressure drops occur almost entirely across the stationary nozzles, and are smaller per stage than those of a velocity-compounded impulse stage. However, Rateau staging is relatively more efficient and can be used as an inlet stage and/or any stage thereafter. Figure #1B shows Rateau Staging.

3. Reaction Staging —

This type is similar in concept to Rateau staging except that pressure drop occurs through both the stationary nozzle and matching blades. Figure #1C shows reaction staging and can readily be compared with Rateau staging.

Turbine Calculations

A perfect steam turbine would theoretically convert all heat energy to mechanical work. Thus,

Mechanical Work = Heat Liberation

Work = Function Heat Drop

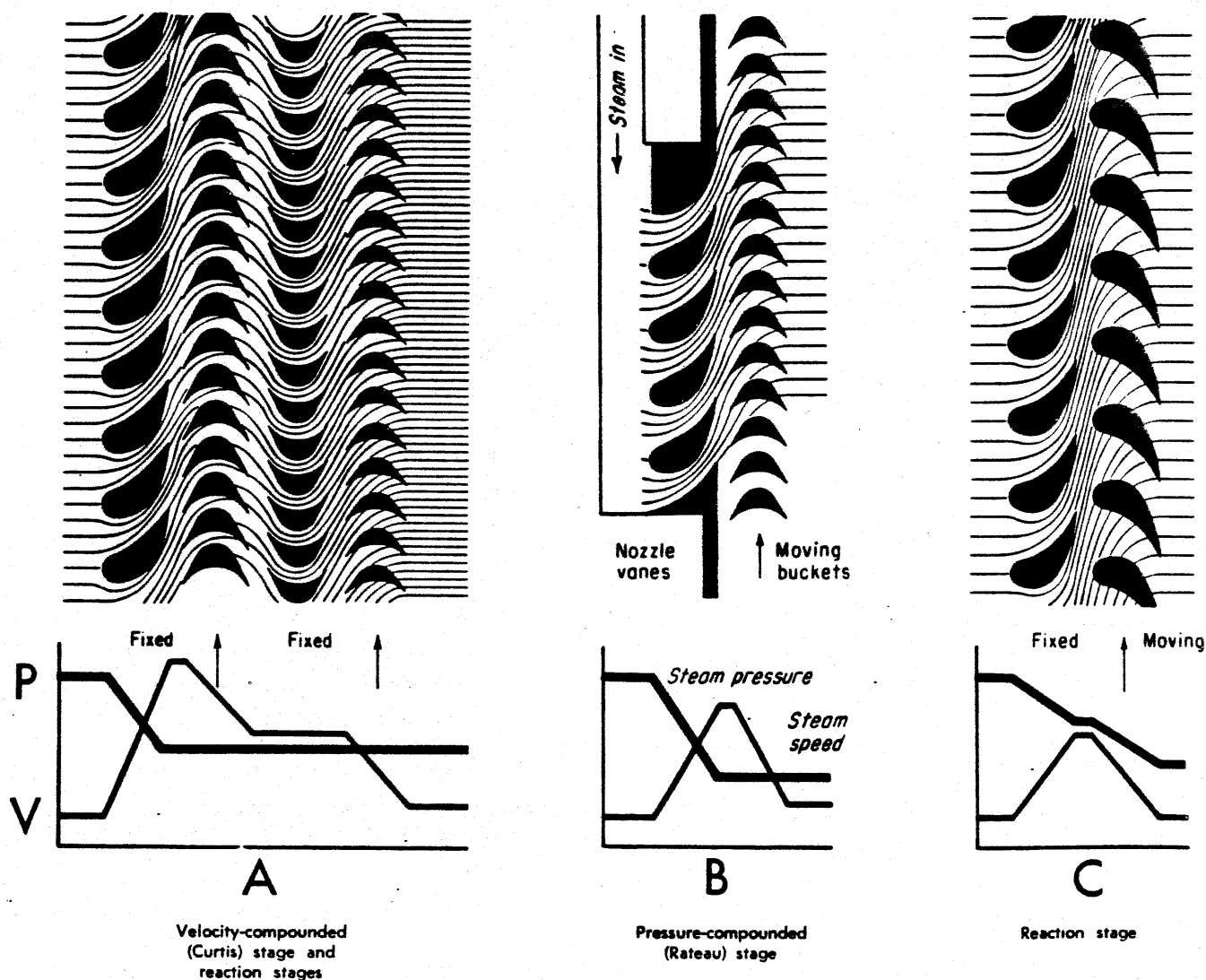


Figure 1

From a Mollier Chart, the thermodynamic characteristics of a steam turbine can be determined. Digressing for a moment, this chart enables one to obtain the following data:

- Enthalpy - BTU/# - total heat
- Entropy - BTU/# °F - arbitrary abscissa
- Temperature - °F - measure of sensible heat
- Superheat - °F - temperature above saturated
- Pressure - PSIA - absolute
- Moisture - % - # vapor/# mixture

Thus a graphical "picture" of steam is obtained to apply to a steam turbine.

Theoretically, for a perfect steam turbine an isentropic expansion occurs (i.e., $\Delta = 0$). Thus, by graphing on a Mollier Chart, the theoretical energy drop can be obtained.

By knowing this drop, the theoretical steam rate can be obtained.

$$\frac{2545}{\Delta h_{is}} = \frac{2545}{h_1 - h_2} = \text{TSR} (\#/\text{HP HR})$$

$$\text{where } \frac{550 \text{ ft. \#}/\text{Sec. HP}}{778 \text{ ft. \#}/\text{BTU}} \times$$

$$3600 \text{ Sec./HR} = 2545 \text{ BTU/HP HR}$$

However, steam turbines are not perfect machines and losses do occur.

Thus,

$$\eta_s = \frac{\Delta h_{act}}{\Delta h_{ideal}} = \frac{h_1 - h_{2e}}{h_1 - h_2} = \frac{\text{TSR}}{\text{ASR}}$$

(See Figure 2)

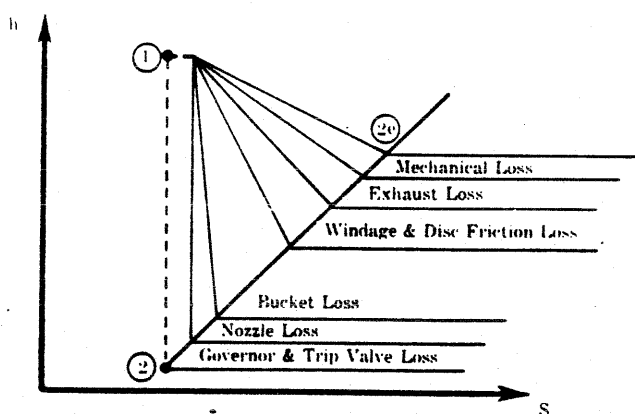


Figure 2

Factors which can and do affect turbine efficiencies are as follows:

- Leakage loss — Such losses can be attributed to:
 - High pressure shaft gland packing.
 - Low pressure shaft gland packing.
 - Governor valve stem packing.
 - Packing at shaft under diaphragms.

2. Rotational losses

- Disc friction due to fluid friction between steam and disc and also the centrifugal pumping action of disc.
 - Windage or fanning action caused by blades fanning the internal fluid.
 - Bearing loss caused by journals and thrust friction.
- Steam pressure losses through trip valves, governor valves, and passages to the nozzle ring.
 - Radiation and conduction losses in general. These losses can be kept minimal by the use of insulation.
 - Exhaust losses occur in two forms and can be classified as leaving losses and hood losses.
 - Leaving loss is the loss of kinetic energy represented by the velocity of the steam leaving the last stage.
 - Hood loss results from a pressure drop of the steam passing through the exhaust casing.
 - Moisture losses caused by the inability of accelerating water droplets as readily as the gaseous steam.
 - Carry over losses due to large axial stage spacing and a discontinuity of blade heights.
 - Displacement losses caused by partial admission stages.

Obviously, "high" losses for a low horsepower machine affect overall turbine efficiency more adversely than identical losses for a large horsepower machine.

Returning to the discussion on blade velocities in relation to heat liberation in the nozzle block, stage efficiency can be substituted for work and the basic stage efficiency determined. In more specific terms, the relationship between blade velocity and jet velocity is referred to as velocity ratio.

Mathematically,

$$V_B = \frac{D\omega}{2} = \frac{\pi DN}{720}$$

Where

$$\omega = \frac{2\pi \text{ Rad.}}{\text{Rev.}} \times \frac{\text{Rev.}}{\text{Min.}} \times \frac{\text{Min.}}{60 \text{ Sec.}} \times \frac{\text{Dia. (in.)}}{2} \times \frac{\text{FT}}{12 \text{ in.}}$$

Previously, it has been shown that,

$$V_J = 224 \sqrt{\Delta h_{is}}$$

Thus,

$$\eta_s \propto \frac{V_B}{V_J} \propto \frac{DN}{229 \times 224 \sqrt{\Delta h_{is}}} \propto \frac{DN}{51400 \sqrt{\Delta h_{is}}}$$

Note: See Addendum Sections C, D, E and F.

Figure #3 depicts a typical efficiency curve for a Rateau and Curtis Stage.

Stage Performance

The following examples illustrate the performance of various combinations of impulse staging. It should be understood, of course, that stage selection and overall turbine efficiency are affected by many important considerations other than stage efficiency. Speed limitations, mechanical stresses, leakage and throttling losses, windage, bearing friction and reheat, all must be factored into the ultimate turbine design.

The approximate efficiencies of a Curtis stage and a Rateau stage are shown in the curves at right.

Basic Formulae

$$V_b = \frac{\pi DN}{720} \quad V_j = 224 \sqrt{h_1 - h_2} = 224 \sqrt{\Delta H}$$

V_b = Pitch line (blade) velocity, ft/sec.

D = Pitch diameter of wheel, inches (base diameter plus height of blade)

N = Rotative speed, rpm

V_j = Steam jet velocity, ft/sec.

h_1 = Inlet steam enthalpy, BTU/lb.

h_2 = Isentropic exhaust steam enthalpy, BTU/lb.

h_{2e} = Stage exit steam enthalpy, BTU/lb.

ΔH = Isentropic heat drop, BTU/lb. ($h_1 - h_2$)

V_b/V_j = Velocity ratio, dimensionless

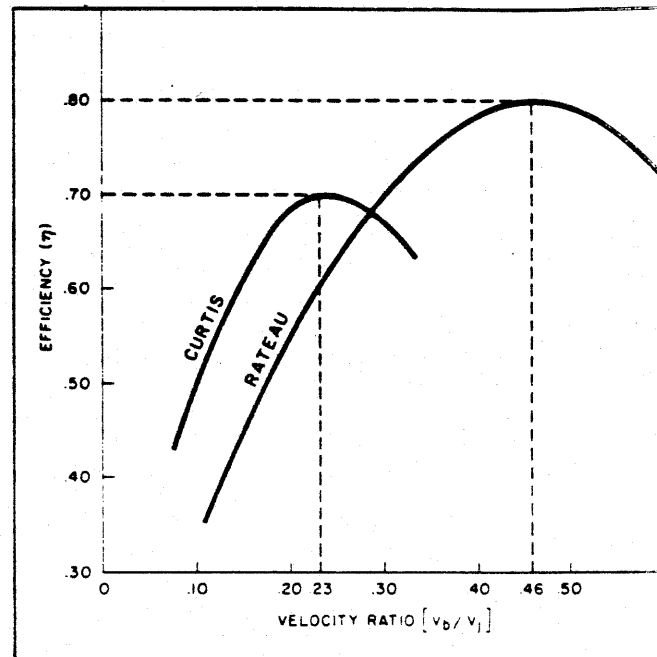


Fig. 3 Efficiency of Curtis and Rateau Stages

EXAMPLE 1. Rateau Stage Performance

Conditions: 400 psig (415 psia); 600F.
5000 rpm E-line (25-inch wheel diameter)
Assume 1-inch blade height

Find isentropic heat drop and end point.

(See Figure 4)

$$V_b = \frac{\pi DN}{720} = \frac{(3.14)(25 + 1)(5000)}{720} = 568 \text{ ft/sec.}$$

From Figure 3 Velocity Ratio
for optimum Rateau stage efficiency = .46

$$V_b/V_j = .46 \quad V_j = V_b/.46 = \frac{568}{.46} = 1235 \text{ ft./sec.}$$

$$V_j = 224 \sqrt{\Delta H} = 1235 \text{ ft./sec.}$$

$$\Delta H = 30.4 \text{ BTU/LB}$$

$$h_1 \text{ (from Mollier Chart or Steam Tables) } = 1305.7 \text{ BTU/LB}$$

$$h_2 = 1305.7 - 30.4 = 1275.3 \text{ BTU/LB}$$

Exhaust Pressure (from Mollier Chart) = 300 PSIA

Assuming a stage efficiency of 80%, the stage exit conditions are:

exhaust pressure 300 PSIA

$$h_{2e} = 1305.7 - (.8)(30.4) = 1281.4 \text{ BTU/LB}$$

EXAMPLE 2. Curtis Stage Performance

Conditions: 400 psig (415 psia); 600F
5000 rpm E-line (25-inch wheel diameter)
Assume 1-inch blade height

Find isentropic heat drop and end point.

(See Figure 4)

$$V_b = \frac{\pi DN}{720} = \frac{(3.14)(25 + 1)(5000)}{720} = 568 \text{ ft/sec.}$$

From Figure 3 Velocity Ratio
for optimum Curtis stage efficiency = .23

$$V_b/V_j = .23 \quad V_j = \frac{V_b}{.23} = \frac{568}{.23} = 2470 \text{ ft./sec.}$$

$$V_j = 224 \sqrt{\Delta H} = 2470 \text{ ft./sec.}$$

$$\Delta H = 121.5 \text{ BTU/LB}$$

$$h_1 \text{ (from Mollier Chart or Steam Table) } = 1305.7 \text{ BTU/LB}$$

$$h_2 = 1305.7 - 121.5 = 1184.2 \text{ BTU/LB}$$

Exhaust pressure (from Mollier Chart) = 115 PSIA

Assuming a stage efficiency of 70%, the stage exit conditions are:

exhaust pressure 115 PSIA

$$h_{2e} = 1305.7 - (.7)(121.5) = 1220.6 \text{ BTU/LB}$$

EXAMPLE 3. Straight Rateau Staging

Conditions: 400 psig (415 psia); 600F.

Exhaust pressure 100 psig (115 psia)

Find the number of Rateau stages required assuming optimum stage efficiency.

(See Figure 5)

From Mollier Chart, the isentropic heat available is:

$$1305.7 - 1184.2 = 121.5 \text{ BTU/LB}$$

Alternately, using a TSR table, or the Elliott Polar Mollier:

$$\text{TSR} = 28.08 \text{ LB/KW-HR}$$

Using the basic definition of TSR:

$$\Delta H = h_1 - h_2 = \frac{3413}{\text{TSR}} = \frac{3413}{28.08} = 121.5 \text{ BTU/LB}$$

From Example 1, the optimum isentropic heat drop per Rateau stage = 30.4 BTU/LB

$$\text{Approximate stages required} = \frac{121.5}{30.4} = 4.0 \text{ (thermodynamically)}$$

The turbine will require 4 Rateau stages.

Assuming 80% stage efficiency.

$$121.5 \times .80 = 97.2 \text{ BTU/LB}$$

$$1305.7 - 97.2 = 1208.5 \text{ BTU/LB}$$

EXAMPLE 4. Curtis & Rateau Staging

Conditions: 400 psig (415 psia); 600F.

Exhaust pressure 35 psig (50 psia)

Find the number of Rateau stages required when using one Curtis stage, assuming optimum stage efficiencies.

(See Figure 6)

From Example 2, we found that a Curtis stage removes 121.5 BTU/LB

$$1305.7 - 121.5 = 1184.2 \text{ BTU/LB pressure} = 115 \text{ psia}$$

Assuming 70% stage efficiency:

$$121.5 \times .70 = 85.05 \text{ BTU/LB}$$

$$1305.7 - 85.05 = 1220.6 \text{ BTU/LB}$$

From this end point to 50 psia, the isentropic heat drop for Rateau stages is:

$$1220.6 - 1151.3 = 69.3 \text{ BTU/LB}$$

From Example 1, the optimum isentropic heat drop per Rateau stage = 30.4 BTU/LB

$$\text{Approximate stages required} = \frac{69.3}{30.4} = 2.28 \text{ (thermodynamically)}$$

The turbine will require one Curtis stage and 2 Rateau stages.

It is interesting to note that with Example 4 conditions, a straight Rateau machine would optimally require 6 stages. The proof is left to the reader.

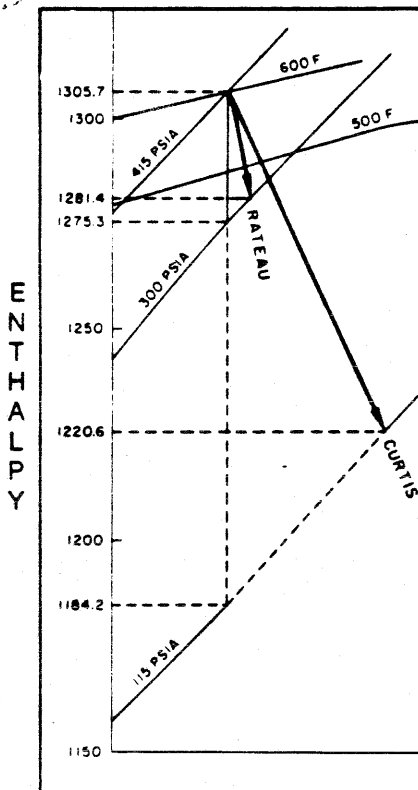


Fig. 4 Stage Performance

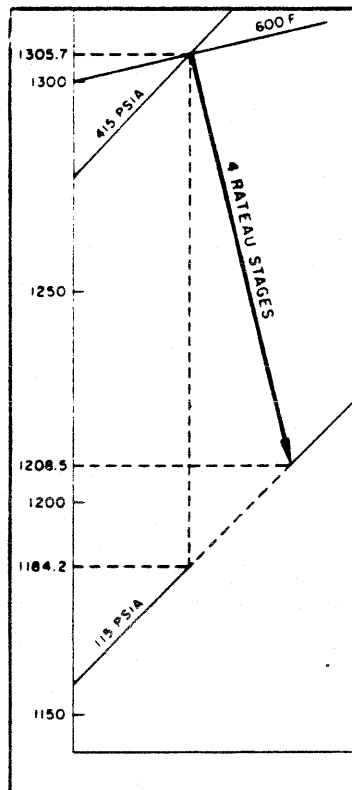


Fig. 5 Straight Rateau Staging

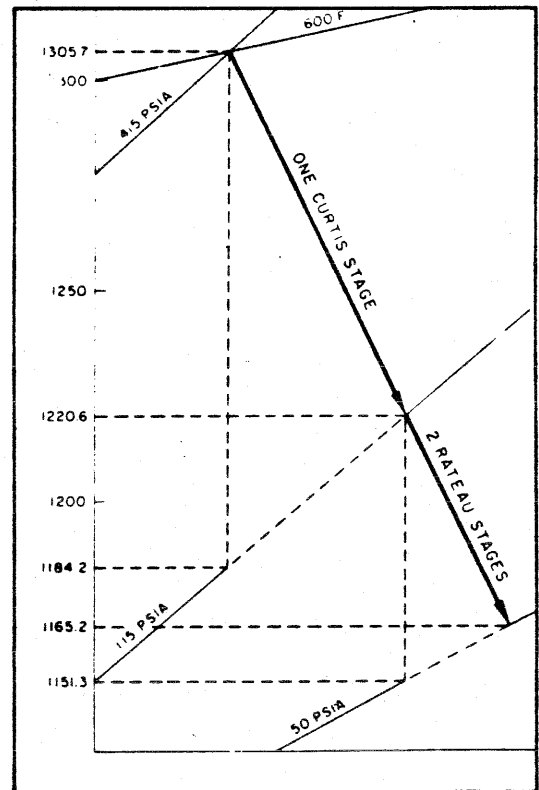


Fig. 6 Curtis and Rateau Staging

(The above schematics are not to actual scale)

ADDENDUM

A. Derivation of Jet Velocity

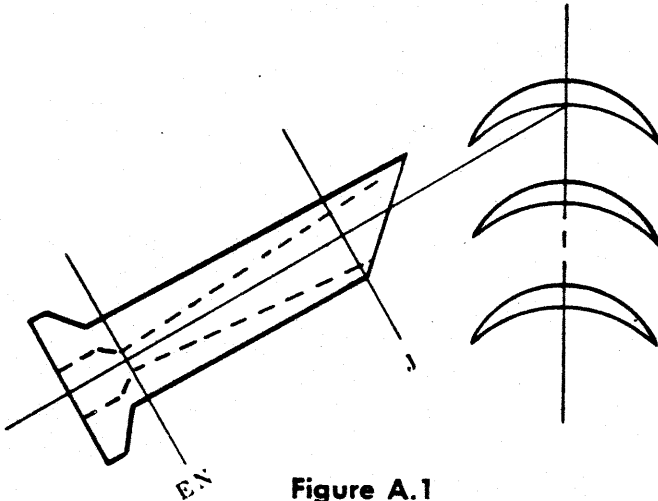


Figure A.1

For a nozzle:

$$(\text{Internal Energy})_{EN} + (\text{External Work})_{EN} + (\text{Kinetic Energy})_{EN} =$$

$$(\text{Internal Energy})_J + (\text{External Work})_J + \text{Kinetic Energy}_J$$

$$\left[\left(\frac{u}{J} + \frac{144 p \mathcal{V}}{J} \right) + \left(\frac{U}{J} \right) + \left(\frac{Z}{J} + \frac{V^2}{2gJ} \right) \right]_{EN} = \left[\left(\frac{u}{J} + \frac{144 p \mathcal{V}}{J} \right) + \left(\frac{W}{J} \right) + \left(\frac{Z}{J} + \frac{V^2}{2gJ} \right) \right]_J \quad (\text{A.1})$$

For an adiabatic expansion:

$$(a) \quad h = \frac{u}{J} + \frac{144 p \mathcal{V}}{J}$$

For flow through a nozzle:

$$(a) \quad \text{Work supplied} = W/J = 0$$

$$(b) \quad \text{Heat supplied} = u/J = 0$$

$$(c) \quad Z_{EN} = Z_J$$

$$(d) \quad V_{EN} = 0$$

∴ Equation (1)

$$h_{EN} = \left(h + \frac{V^2}{2gJ} \right)_J \quad (\text{A.2})$$

or

$$V_J = \sqrt{2gJ(h_{EN} - h_J)} = 244 \sqrt{\Delta h_{is}} \quad (\text{A.3})$$

B. Derivation of Force and Work for a Simple System

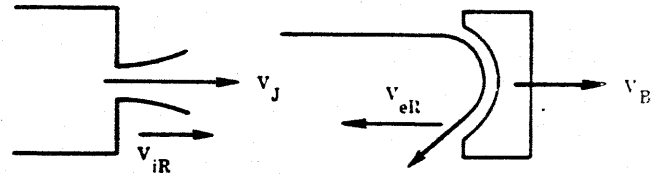


Figure A.2

V_J — Jet Velocity from Nozzle

V_B — Blade Velocity

V_{iR} — Jet Inlet Relative Velocity

V_{eR} — Blade Exit Relative Velocity

V_{eB} — Blade Absolute Exit Relative Velocity

Work = Force x Distance Moved (B.1)

Force = Mass x Acceleration = $m \, dV/dt$ (B.2)

$$\text{Where } m = \frac{1 \#}{\text{ft./sec.}^2} = 1 \text{ slug}$$

$$= 1/g \text{ (for 1# steam)}$$

For Above System:

$$\text{Force} = \frac{1}{g} (V_J - V_{eB}) / dt \quad (\text{B.3})$$

Substitution Eq. (B.3) into (B.1)

$$W = \frac{(V_J - V_{eB})}{g} \frac{dx}{dt} \quad (\text{B.4})$$

$$\text{Where } dx/dt = V_B$$

$$\therefore W = \frac{(V_J - V_{eB})}{g} V_B \quad (\text{B.5})$$

C. Solution of Maximum Force and Maximum Work for Simple System

Assume No Friction or Flow Change

$$V_{iR} = V_J \Rightarrow V_B = V_J - V_{eR} \quad (\text{C.1})$$

$$V_{eB} = V_{eR} \Rightarrow V_B = (-V_J + V_B + V_B) = 2V_B - V_J \quad (\text{C.2})$$

Substituting Eq. (C.2) into Eq. (B.5)

$$W = \frac{[V_J - (2V_B - V_J)] V_B}{g} = \frac{(2V_J - 2V_B) V_B}{g} = \frac{2(V_J V_B - V_B^2)}{g} \quad (\text{C.3})$$

Differentiation of Eq. (C.3) with respect to V_B

$$\frac{dw}{dV_B} = \frac{2(V_J - 2V_B)}{g} \quad (C.4)$$

Solving (C.4) for maximum:

$$V_J - 2V_B \text{ or } V_B/V_J = +1/2 \quad (C.5)$$

(See Figure A.3)

Solving for Work:

Substitution of Eq. (C.5) into Eq. (C.3) for Work

$$\begin{aligned} W_{\text{max.}} &= 2/g [V_J (V_J/2) - (V_J/2)^2] \\ &= 2/g (V_J^2/4) = V_J^2/2g \end{aligned} \quad (C.6)$$

(See Figure A.3)

Solving for Force:

From Eq. (B.3) Force is a Linear Equation, therefore, the slope can be determined by differentiation:

$$F = (V_J - V_{EB})/gdt \quad (B.3)$$

$$V_{EB} = 2V_B - V_J \quad (C.2)$$

Substitution of Eq. (C.2) into Eq. (B.3)

$$F = \frac{2(V_J - V_B)}{g dt} \quad (C.7)$$

Differentiation of Eq. (C.7)

$$dF/dV_B = 2/g dt (-1) \text{ or negative slope}$$

(See Figure A.3)

Again using Eq. (C.7) and rearranging

$$F = 1/g dt (1 - V_B/V_J) \quad (C.8)$$

F maximum occurs when $V_B/V_J = 0$ or when blade is fixed

F minimum occurs when $V_B/V_J = +1$ or when jet velocity equals blade velocity

The above is shown on Figure A.3.

E. From the above sections a theoretical velocity curve can be drawn:

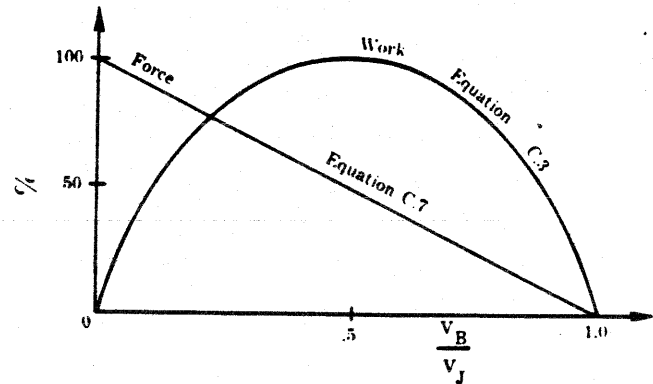


Figure A.3

NOTE: All of the above equations, etc. have assumed an ideal nozzle-bucket combination. The manufacture of this type of system is impossible for the following reasons:

- (1) It is impossible for the entrance angle of the bucket to be in the plane of the wheel. This could cause mechanical interference.
- (2) It is impossible for the steam to leave the bucket at a 0° angle. There would be no opportunity for the steam to pass axially through the turbine.

Section F depicts an actual impulse nozzle-bucket system which closely represents the conditions, velocities, angles, etc. in actual design.

F. Actual Velocity Diagram for Single Stage Impulse Turbine

Using the preceeding analysis, it can be shown that the following relationships can be used to calculate the work produced by the velocities of a single-stage impulse turbine.

D. Calculation of Optimum Efficiency:

$$\eta_s = \frac{\text{Energy produced by blade}}{\text{Energy supplied}}$$

$$= \frac{\text{Eq. (C.3)}}{\text{Kinetic Energy of Jet}}$$

$$= \frac{2/g (V_J V_B - V_B^2)}{V_J^2/2g}$$

$$= \frac{2 (V_J V_B - V_B^2)}{V_J^2/2}$$

$$= \frac{4 (V_J V_B - V_B^2)}{V_J^2} \quad (D.1)$$

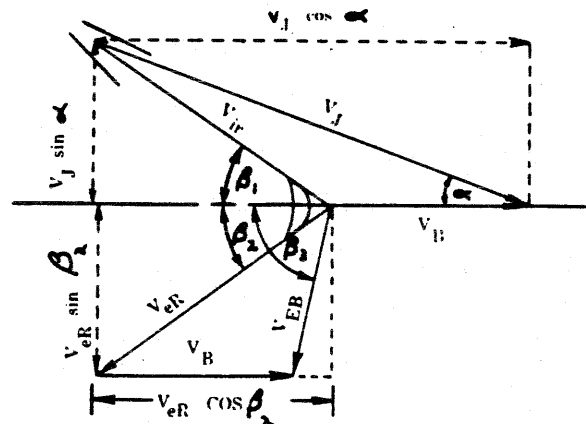


Figure A.4

$$\text{Where } \tan \beta_1 = \frac{V_J \sin \alpha}{V_J \cos \alpha - V_B}$$

$$\text{Then } V_{iR} = \frac{V_J \sin \alpha}{\sin \beta_1}$$

Due to the loss caused by friction and eddies, the relative leaving velocity (V_{eR}) is less than the relative entering velocity (V_{iR}). This loss is dependent upon the magnitude of the relative entering velocity and the blade geometry. Such a loss can be as low as 5% at low (250 ft./sec.) relative entering velocities and can reach 30% at high (3500 ft./sec.) relative entering velocities.

Thus,

$$\tan \beta_3 = \frac{V_{eR} \sin \beta_2}{V_{eR} \cos \beta_2 - V_B}$$

then

$$V_{EB} = \frac{V_{eR} \sin \beta_2}{\sin \beta_3}$$

$$W = \frac{1}{2g} [(V_J^2 - V_{EB}^2) - (V_{iR}^2 - V_{eR}^2)] \quad (E.1)$$

Substitution of a simple nozzle/wheel per Sec. C.

$$W = \frac{1}{2g} \left[V_J^2 - (2V_B - V_J)^2 - \left[(-V_{eR})^2 - (V_{eR})^2 \right] \right]$$

$$= \frac{2}{g} (V_B V_J - V_B^2)$$

agrees with Eq. (C.3) (E.2)

If Eq. (C.3) or (E.2) is the work per one pound of steam and since Power is the rate of doing work, the power can be calculated by the following:

$$\text{Power} = W \cdot \frac{\text{ft. \#}}{\text{\# steam}} \times \frac{\text{BTU}}{778 \text{ ft. \#}} \times$$

$$g' \cdot \frac{\text{\# steam}}{\text{HP}} \times \frac{\text{HP HR}}{2545 \text{ BTU}}$$

$$= \frac{Wg'}{1.98 \times 10^6}$$

Expressing again as efficiency:

$$\eta_s = \frac{\text{Energy produced by blade}}{\text{Energy available}}$$

$$= \frac{\text{Eq. (E.2)}}{\text{Heat available}}$$

$$= \frac{W/778}{\Delta h_{is}}$$

SYMBOLS

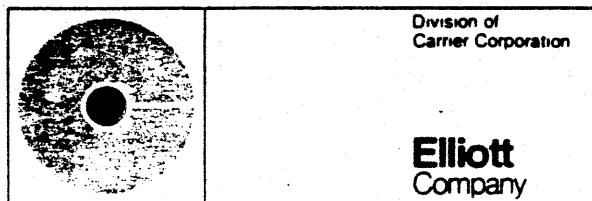
D — Diameter (in.)	h — Enthalpy (BTU/#)
F — Force	m — Mass
HP — Horsepower	p — Pressure (PSIA)
J — Mechanical equivalent heat (778 ft. #/BTU)	s — Entropy (BTU/# °F)
N — RPM - Revolutions per minute	t — Time (sec.)
U — Heat (ft. #/#)	u — Internal energy (ft. #/#)
V — Velocity (ft./sec.)	
W — Work (ft. #/#)	v — Specific Volume (ft. ³ /#)
Z — Height above datum (ft.)	ω — Angular velocity (Radian/sec.)
g — Gravity (ft./sec. ²)	η — Efficiency
g' — Flow (#/HR.)	

SUBSCRIPTS

B — Blade	eR — Exit Relative
EB — Exit Absolute	iR — Inlet Relative
EN — Enter	is — Isentropic
J — Jet	s — Stage
	1 — Inlet Conditions
	2 — Isentropic Exhaust Conditions
	2e — Actual Exhaust Condition

ANGLES

α — Nozzle Inlet
β_1 — Bucket Inlet
β_2 — Bucket Exit



- | | |
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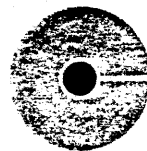
Isle of Wight, England

Canada: Elliott Industrial Equipment Company
24 Queen St. East, Brampton, Ontario

Europe, Middle East, Africa:
Elliott Turbomachinery Ltd.
197 Knightsbridge, London, England

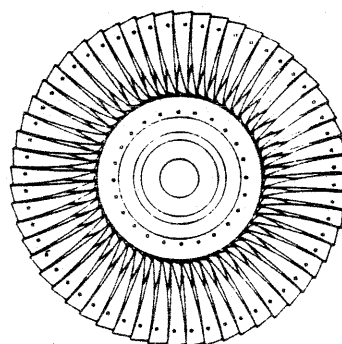
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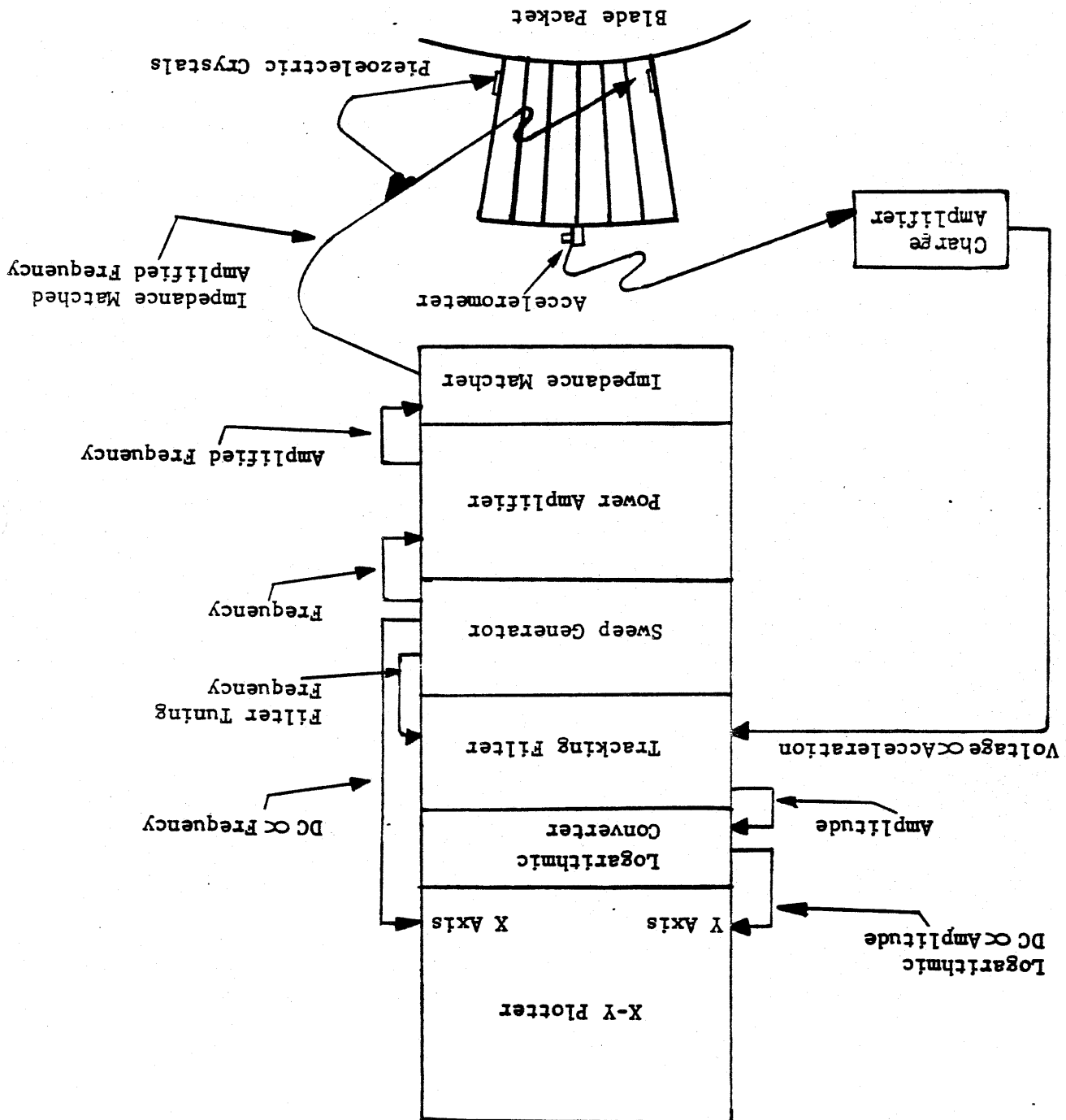
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March 1976

Blade Design



*Elliott
Multistage
Steam
Turbines*

Fig. 1. Typical static test hardware arrangement.



Blade Design

The steam turbine evolution has seen steam pressures and temperatures constantly being increased along with rotor speeds and horsepower. During this period of development, the turbine concept has been adapted to a countless number of different driver applications. Because of these changing design parameters, more sophisticated design methods are constantly being required to insure reliable turbine operation.

The turbine blade would undoubtedly be at the top of the list of the turbine components which require a high degree of engineering expertise. This unique subject of blade design has always been intriguing to the engineer. The greatest portion of the blade analysis techniques follow precise mathematical models, but the complete blade evaluation can best be described as an "art." Even when the basic design limits are established, we must always look at what has been accomplished in the past.

The blade mechanical design is analyzed for each stage of every turbine application. This analysis includes a review of the blade mechanical speed limits, and generating Goodman and Campbell Diagrams.

Blade mechanical speed limits are dictated by the stresses generated in the blade/disk geometry (i.e., blade airfoil, blade root, blade shroud, lashing wire, disk rim, locking piece, and locking pin).

The blade mechanical reliability is evaluated by means of various analytical techniques. This reliability analysis can be subdivided into three areas:

1. Determination of the blade packet natural frequencies and mode identifications.
2. Graphically describing the blade natural frequencies and sources of significant excitation as a function of turbine speed, (i.e., Campbell Diagrams).
3. Evaluating the combined effects of alternating stress and steady state stress in the turbine blade, (i.e., Goodman Diagrams).

Excellent blade reliability records can be achieved and sustained over a long period of time by combining all of the current mathematical models such as Goodman and Campbell Diagrams along with the manufacturers past operating experience.

Frequencies and Mode Identifications

Optimum mechanical integrity is one of the most important considerations in the application of a particular blade geometry. The key to success in this type of analysis is to carefully identify the various natural frequencies and accurately assign the corresponding frequency values.

Frequency analysis techniques have improved considerably in the past few years. The classical method for blade frequency analysis was to use the individual blade natural frequencies. These values were obtained from bench tests conducted on individual blades.

This information is misleading because, when the blade is assembled with the turbine disk, the blade packets and disk form a system which does not necessarily vibrate at the same frequencies as those obtained from the individual blade tests.

The current blade testing techniques can be classified into two basic types, static tests and dynamic tests. The static tests are generally used to determine blade natural frequencies. The blade frequencies that are obtained from these static test procedures are later used to determine the resonant conditions for any particular application.

The static testing technique consists of testing actual rotor assemblies as manufactured in the normal production cycle. Fig. 1 shows a typical static test set-up arrangement. The following paragraphs describe the actual static test procedure.

Piezoelectric crystals are attached at the base of the first and last blade in the test packet. An accelerometer is attached to the shroud band and located approximately in the middle of the test packet. This accelerometer is oriented in the direction that blade response is to be measured (i.e., tangential or axial).

It is important to note that the piezoelectric crystals excite all resonant frequencies in all directions, however, the accelerometer (unidirectional) senses that response for the orientation in which it is attached.

A constant input voltage is supplied to the crystals by means of a sine wave generator. The frequency of the input voltage is varied between 100 and 20,000 cps at a constant sweep rate.

As the frequency of this input voltage (excitation frequency) coincides with a natural frequency of the packet, a resonant condition develops. The result is a build-up in packet vibration amplitude. This response or displacement is sensed by the accelerometer.

The accelerometer will measure the acceleration associated with any particular resonant condition at a specific frequency. Integrating this acceleration twice results in a peak to peak displacement which represents the amplitude of response of the packet relative to the input voltage. This automatic integration and resultant plotting of the raw test data is a unique feature of this test procedure.

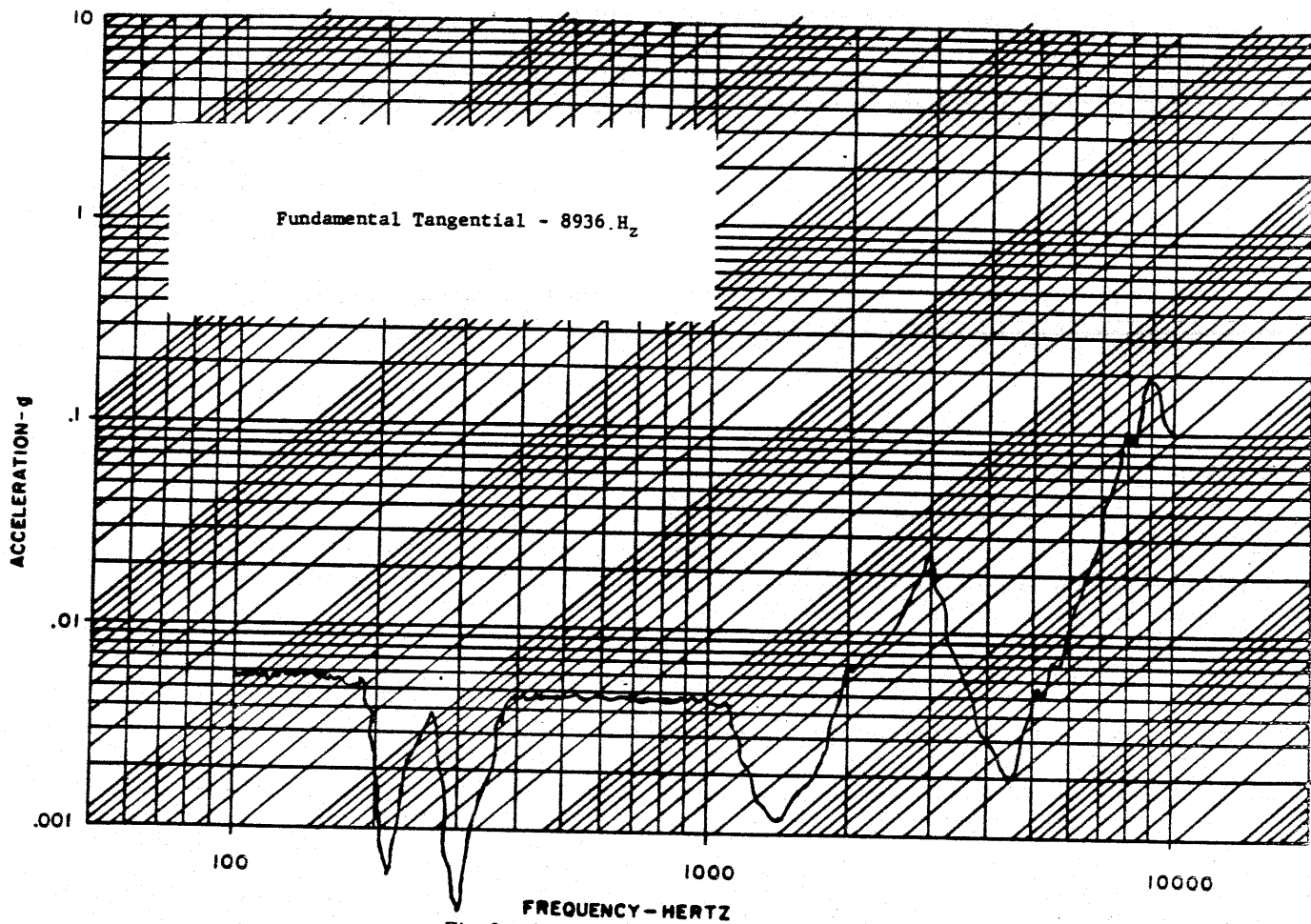


Fig. 2. Fundamental static packet test diagram.

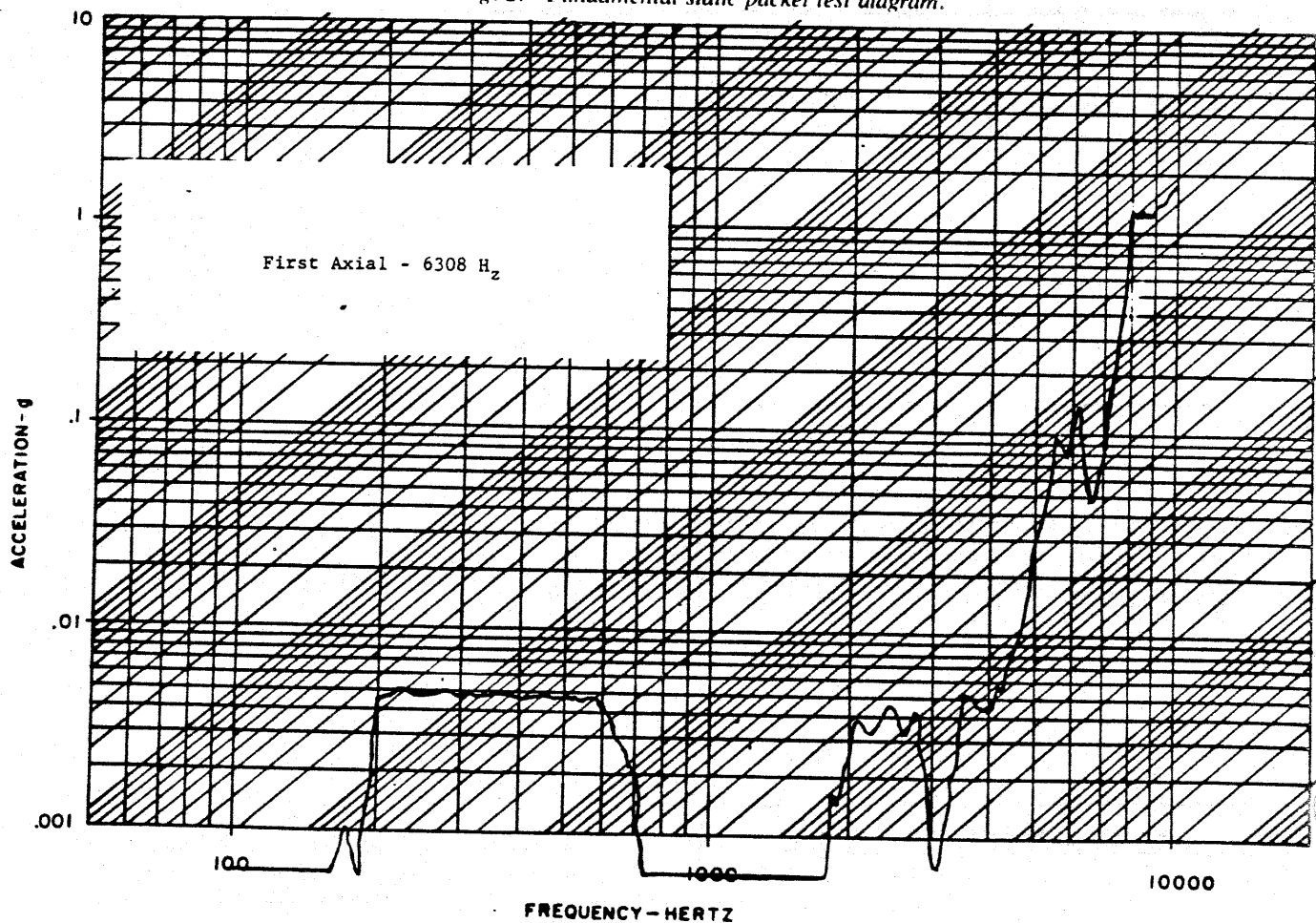


Fig. 3. First axial static packet test diagram.

The response of any packet for both the tangential and axial directions is thus achieved and permanently recorded for future reference. (Figs. 2 and 3 are plots presenting the results of a static packet test.)

After this plotting phase of the static test has been completed, all of the significant resonant peaks are identified according to their mode shape i.e., fundamental tangential, first axial, torsional, etc.

Mode shapes are identified by the characteristic phase relationships between adjacent blades in packet at a resonant condition. The basic mode shapes are categorized by the direction of motion and order of natural frequency i.e., first, second, third, etc. (Fig. 4 contains an illustrative example of mode shape orientation.)

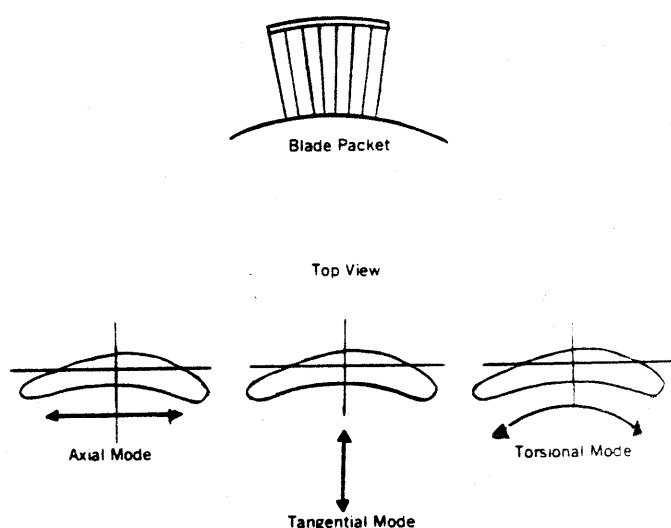


Fig. 4. Blade packet mode directions.

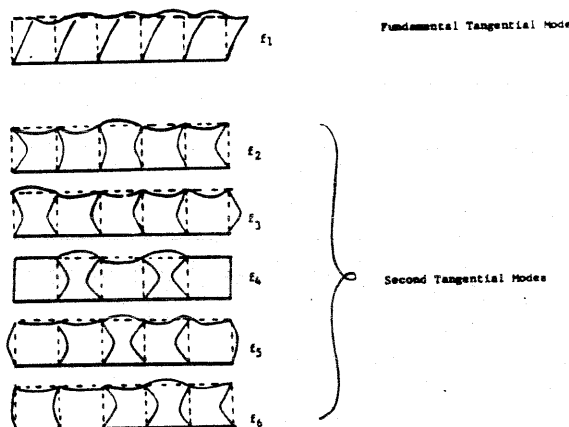


Fig. 5. Blade-to-blade phase relationship for tangential direction modes.

The minimum and maximum axis of inertia for this blade section are perpendicular and parallel to the plane of the disk containing the blades. The three basic modes of motion that can exist in this example are tangential, axial, and torsional. (See Fig. 4.)

The following discussion expands upon this subject of mode type. The more common tangential modes and their unique characteristics are used as an illustrative example. Similar discussion also applies to the other basic modes. (See Fig. 5.)

First Tangential Mode:

This mode is more commonly termed the fundamental tangential mode. This mode is characterized by having all of the blades moving in phase with the maximum motion occurring at the blade tip and having no nodes along the blade length.

The blade packet response in resonance can be quite large when excited by a force whose frequency corresponds with a low harmonic of running speed or nozzle passing frequency.

Second Tangential Mode:

There is one less natural frequency in this group than the number of blades contained in a given packet. This mode is characterized by large amplitudes near the midpoint of the blades while the blade tips remain essentially stationary. The natural frequencies for this mode group are normally close together in a relatively narrow frequency band.

The individual blade phase relationships between adjacent blades can vary from in-phase to out-of-phase symmetry. The possibility of a strong amplification in resonance for any of these modes is unlikely, but must be given consideration in the design.

The mode identification procedure is performed with a hand-held stylus-type accelerometer and an oscilloscope. The first step is to excite the assembly at the exact frequency corresponding to the resonant peak being classified. Then each blade in the test packet is probed with the stylus accelerometer.

The individual blade phase relationship or relative direction of motion with respect to a known reference signal (input voltage to crystals) is determined with the oscilloscope. The oscilloscope screen will display a Lissajous Pattern exhibiting the phase relationship of each blade in the test packet. The mode shape at the specific frequency can then be identified by knowing the blade phase relationships. (Fig. 6 illustrates the mode identification techniques.)

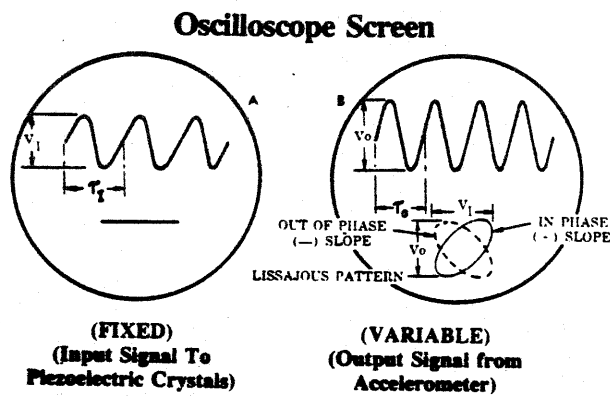


Figure 6A—Input signal to Piezoelectric Crystals on horizontal axis of oscilloscope.

Figure 6B—Output signal from accelerometer on vertical axis of oscilloscope.

Lissajous Pattern—Comparison of the input signal with respect to the output signal or phase relationship.

V_1 —Input voltage

τ_1 —Period of input signal.

V_0 —Output voltage from accelerometer proportional to acceleration.

τ_0 —Period of output signal.

Accelerometer Probe in Tangential Direction

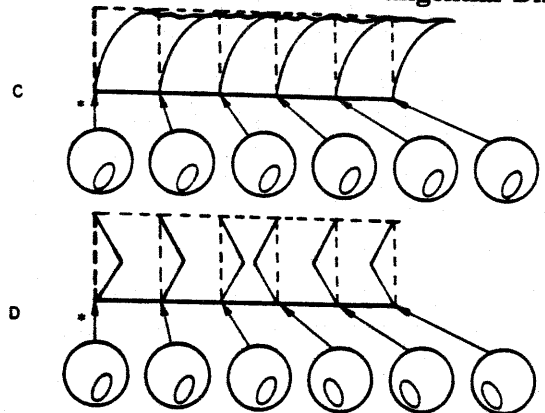


Figure 6C—All buckets in phase and no nodal lines along vane other than the one at the base of the vane. This is characteristic of the fundamental tangential mode.

Figure 6D—Half the buckets were in phase and the other half out of phase. No nodal lines along vane other than the one at the base of the vane. This is characteristic of a second tangential mode.

*Lissajous Pattern (Comparison of input signal with respect to output signal).

Fig. 6. Illustrative example of mode identification from Lissajous patterns.

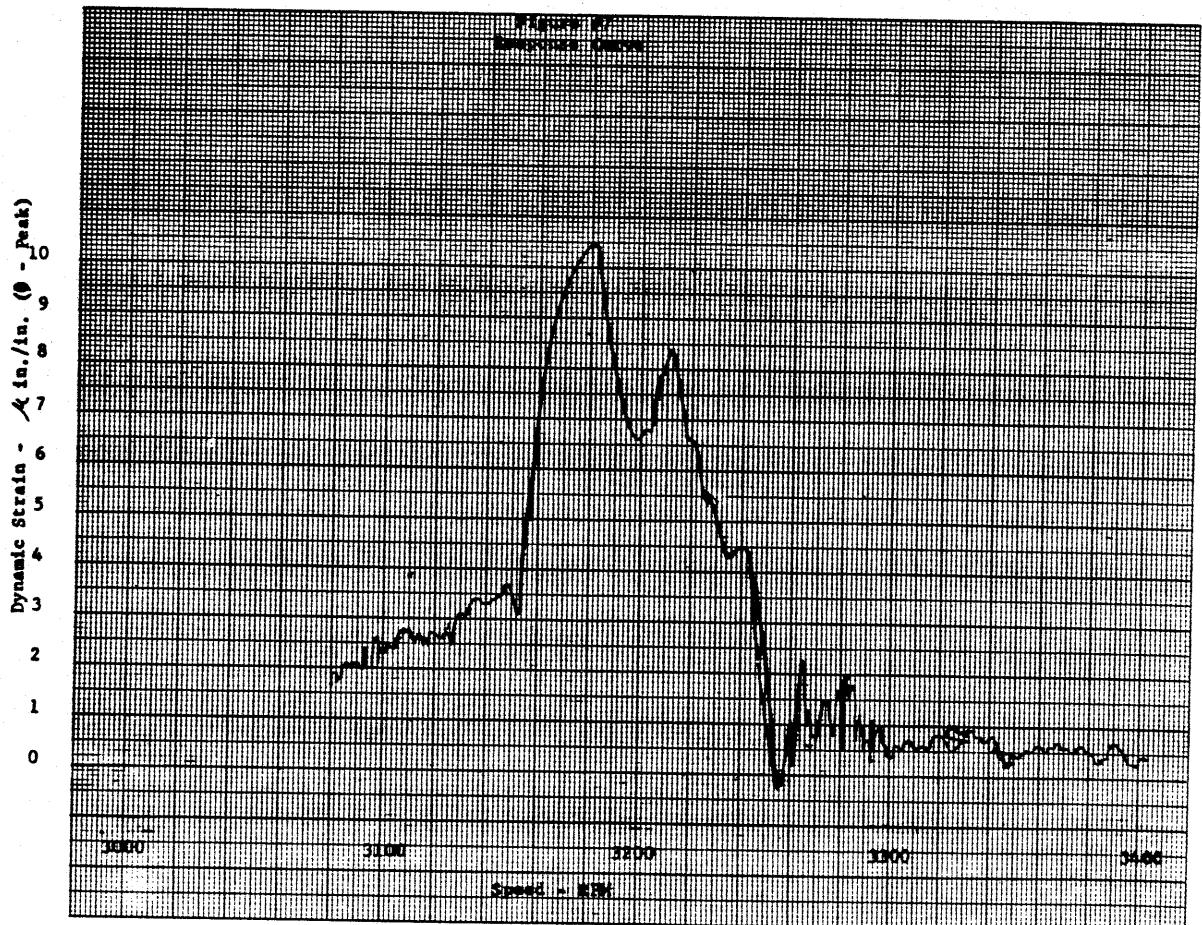


Fig. 7. Response curve.

The dynamic-type blade testing is the second basic type of analysis. Dynamic tests are conducted to determine blade natural frequencies as well as blade response to forced excitation at operating speeds.

The dynamic testing methods are recognized as the most complete analysis, however, this method involves a considerable amount of time and expense. Therefore, dynamic-type testing is only used in critical areas where static testing is not completely suitable.

The dynamic-type test hardware basically consists of a test casing assembly, driver, and dynamometer. The casing is usually a cylindrical chamber equipped with a series of nozzles. These nozzles are spaced in such a manner to provide excitation at various orders of rotor speed and nozzle passing frequency.

The test rotor is positioned inside the casing and consists of a full size disk assembly fitted to the test rig shaft. This shaft is drilled in both radial and axial directions to permit lead wires from the test instrumentation to be brought out from the test casing to a slip ring assembly fitted into the end of the shaft. (Photographs 1 and 2 illustrate typical dynamic test hardware.)

The dynamic testing performed on a 16" blade is used as an illustrative example to demonstrate the techniques and results which can be obtained from this type of testing.

The testing sequence is to first define the various frequencies, mode shapes, and relative responsiveness for the assembly by static-type testing methods. This enables both the blade and disk frequencies to be determined throughout the frequency range of interest prior to the actual rotating test.

The next step is to perform the predetermined series of dynamic tests. The data obtained from the dynamic tests is recorded and preserved on magnetic tapes for further reference and analysis. (Fig. 7 is a sample of a typical response plot obtained from a 16" blade testing.) These response curves are plots of the strain gage output (blade stress) versus shaft speed.

The strain gage output or stress will increase as a resonant condition is approached. (Fig. 7 demonstrates the blade response at the resonant condition of the fundamental tangential frequency with a three per revolution exciting force.)

These response plots are used to develop the desired dynamic test information. For example, dynamic testing indicates the effects of rotation on the blade natural frequency.

Obviously, this effect cannot be detected by the static test approach. The centrifugal forces acting on the blades exhibit a stiffening effect which raises the blade natural frequency during rotation. This effect is small and insignificant for short blades, but must be considered on longer blades.

The blade natural frequency versus shaft speed characteristic can be derived from the information contained on the response curves. This centrifugal stiffening effect is shown on the Campbell Diagram for a 16" blade in Fig. 8. As can be seen from this Campbell Diagram, both of the plotted fundamental frequencies increased by more than 30%.

Dynamic-type testing also enables the engineer to evaluate the blade system damping. Damping is an ex-

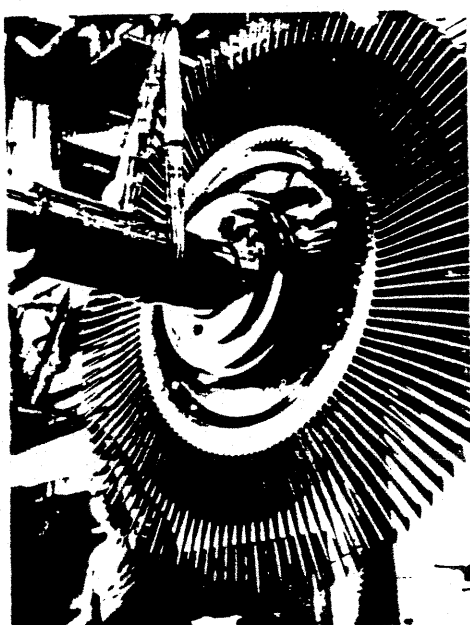


PHOTO 1 Bladed disk is lowered into rotating test assembly to determine the effect of rotation on blade and disk frequencies.

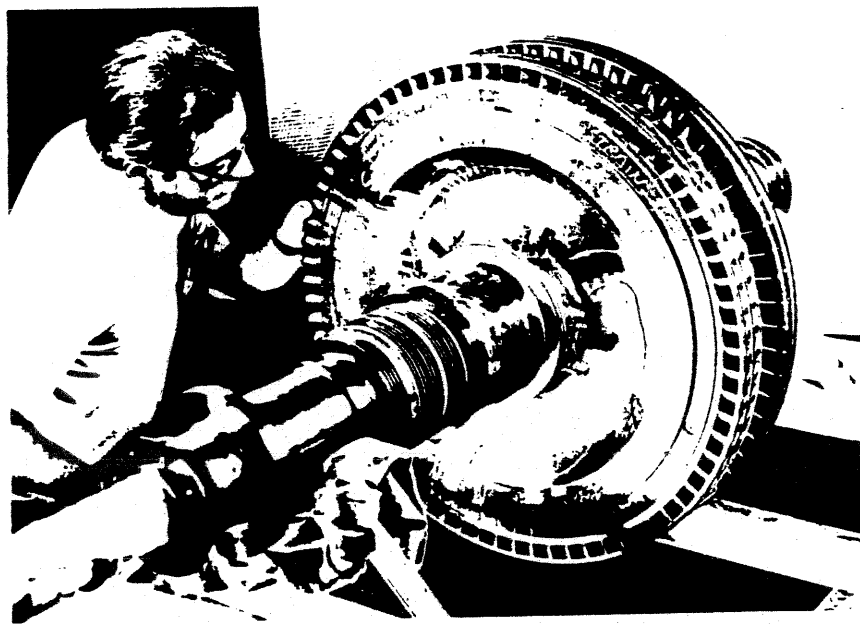


PHOTO 2 First-stage Curtis disk assembly being prepared for dynamic testing to determine mechanical and thermodynamic characteristics.

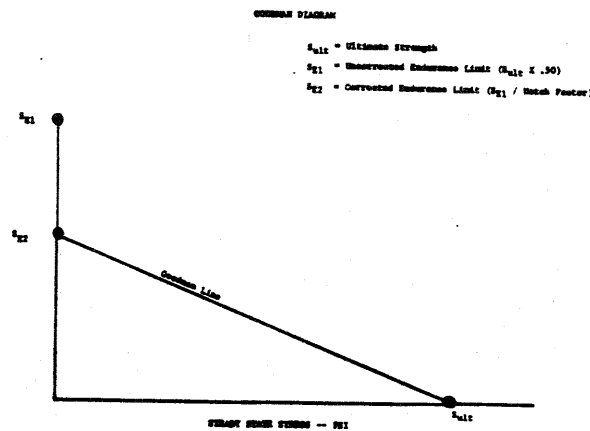


Fig. 13. Goodman Line.

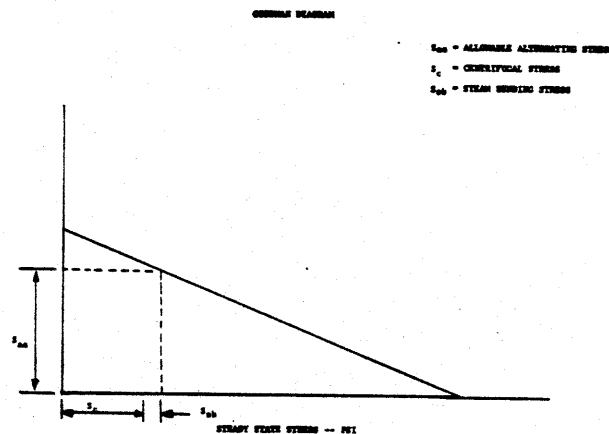


Fig. 14. Goodman Diagram stresses.

the allowable limits. A blade path selection for a particular application should always attempt to avoid an interference condition.

The likelihood of selecting all of the blades in a particular multistage turbine where their natural frequencies are not close to a resonant condition is generally impractical, if not impossible. When an interference condition cannot be avoided, the machine must be designed so operating in resonance can be achieved reliably.

Therefore, if a resonant condition (interference on the Campbell Diagram) does exist, the allowable margin for stress amplification must be evaluated. This evaluation is in the form of a Goodman Factor calculation.

Goodman Diagrams

The Goodman Factor calculation is the design tool used to evaluate the reliability of a blade operating in resonance. This factor is nothing more than the ratio of allowable cyclic stress to the maximum steam bending stress. This stress ratio must have a minimum value, above which reliable operation in resonance can be assumed for a particular geometry.

The manufacturer's blade testing programs and past successful operating experience has established these safe

minimum Goodman Factor guidelines. Factor calculation procedure is derived from the construction of a Goodman Diagram.

The Goodman Diagram is used as a means of evaluating the combined effects of alternating stresses in turbine blading. (Figs. 13 and 14 show the basic Goodman Diagram.) The diagram shows the steady state stresses, and the ordinates of the alternating stresses in the component being evaluated. The theoretical failure line is constructed by a line between the Ultimate Strength on the steady stress axis to the Corrected Endurance Limit on the alternating stress axis.

This Goodman Line, in theory, represents the maximum combinations of steady state stress and alternating stress which would produce a blade fatigue failure. Any point below the Goodman Line is safe; anything above the line is unsafe.

The analysis utilizes three stresses in the Goodman Diagram. These stresses are:

1. Centrifugal Stress
2. Steam Bending Stress
3. Vibratory Stress

The centrifugal stresses are caused by the rotation about the shaft centerline. The steam bending stresses are caused by the steam momentum and changes in a particular turbine stage. The vibratory stresses are usually caused by variations in the steam flow stress.

The centrifugal and steam bending stresses are considered steady state stresses, and the vibratory stress is considered a cyclic stress. (See Fig. 14.)

The steady state stress component of the Goodman procedure is relatively straight forward. The vibratory stress calculation. Considerable research is underway to find methods to calculate

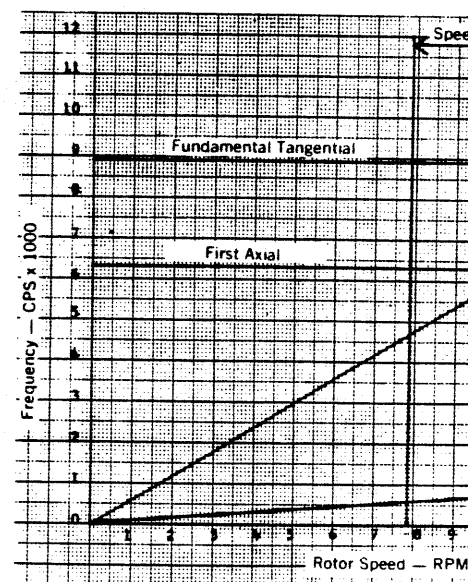


Fig. 15. Example calculation, Campbell Diagram.

Modified Goodman Diagram

CALCULATION DATA

1. S_{ult} (Corr) 98,200 PSI
2. Notch Factor (KF) 1.667
3. Steam Bending Stress (S_{sb}) Use Higher Value of (SLE) or (STE)
1199 PSI
4. Cent. Stress Factor (CSF) 90
5. S_{AA} - Allowable Alternating - FROM DIAGRAM

1. ENDURANCE LIMIT

$$S_e = S_{ult}/2 = (98,200)/2 = 49,100 \text{ PSI}$$

2. CORRECTED ENDURANCE LIMIT

$$S_e' = S_e/KF = (49,100)/(1.667) = 29,454 \text{ PSI}$$

3. STEAM BENDING STRESS

$$S_{sb} = (SLE) \text{ or } (STE) = 1199 \text{ PSI}$$

4. CENTRIFUGAL STRESS

$$S_c = (CSF) \times (\text{Speed}/1000)^2 = (90) \times (11235/1000)^2 = 11,360 \text{ PSI}$$

5. STEADY STATE STRESS

$$S_{ss} = S_{sb} + S_c = (1199) + (11,360) = 12,559 \text{ PSI}$$

6. GOODMAN FACTOR

$$AF = S_{AA}/S_{sb} = (25,500)/(1199) = 21.2$$

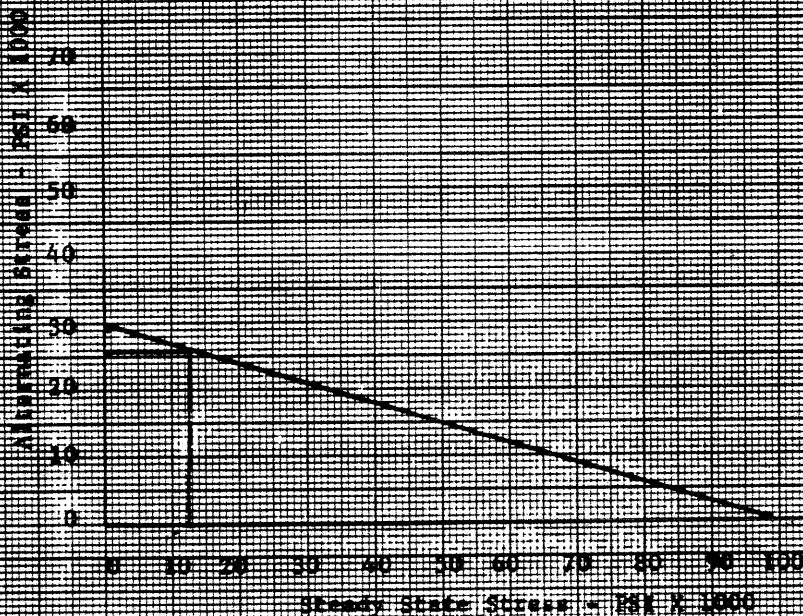


Fig. 16. Example calculation, Goodman Diagram.

stresses in blades. These efforts, primarily theoretical, serve to provide insight into the nature of blade response to excitation and damping.

Since these analytical methods are still being developed, and for the most part are unproven over long periods of operating time, an intuitive approach to vibratory stress is required. Essentially, it is assumed that the vibratory stress is directly related to the steam bending stress. The Goodman Diagram is the analytical tool used to analyze this vibratory stress situation.

The Goodman approach can be described as follows: It is intuitive that as a resonant condition is approached, there will be an increase in the turbine blade stress levels. The magnitude of this stress build-up will be dependent upon the type of excitation and damping in the blade group. Because the magnitude and nature of the exciting force is quite intangible, the absolute value of the expected vibratory stress cannot be accurately calculated.

Therefore, turbine blades must have an adequate margin of safety above the steady state stress, in which vibration may increase the stress without exceeding the endurance limit of the blade material. The Modified Goodman Diagram is used to evaluate this margin of safety which exists in a turbine blade, with respect to the allowable vibratory stress.

This evaluation is in the form of a Goodman Factor. The Goodman Factor is defined as the allowable alternating stress divided by the steam bending stress.

The allowable alternating stress is obtained from the Goodman Diagram by entering the abscissa axis at the steady state stress value and proceeding vertically until the Goodman Line is intersected and reading the corresponding allowable alternating stress on the ordinate axis. The calculated Goodman Factor can then be compared to the design guideline limits that have been established from past experience at similar operating conditions.

In order to demonstrate the use of the available turbine blade design tools, let us review a sample application. The example is an intermediate pressure stage, .8475" blade height on a 18.75" base diameter. (Figs. 2 and 3 contain the actual "Elliott Diagrams" for the referenced blade geometry.)

The Campbell Diagram can now be drawn to determine if a resonant condition exists. (Fig. 15 contains the Campbell Diagram for this application.)

From the diagram, it can be seen that the fundamental axial mode frequency has an interference condition with nozzle passing frequency in the operating speed range. The

prognosis might be that the blade will fail. Many engineers feel that if any interference exists in the operating speed range, the blade will fail; but is this true?

It is important to point out that the Campbell Diagram alone cannot predict reliable or unreliable blade operation. This diagram is nothing more than a graphical representation of frequencies, with no reference to stress levels. A Modified Goodman Diagram must be drawn to evaluate the stresses acting on this blade. (Fig. 16 contains the Goodman Diagram for this application.)

Past operating history has established safe Goodman Factor guidelines. These guidelines are a function of steam inlet geometry, and whether the blade does or does not have a significant resonant condition in the operating speed range. The example blade has a Goodman Factor of 21.2; more than twice the minimum requirement for this stage. This would, therefore, be a reliable blade and would be used for this application even though a Campbell Diagram shows frequency interference.

How can we be so sure? It can be proven because this example stage has been successfully running for seven years, as of this printing, with the first axial mode in resonance with nozzle passing frequency within the operating speed range. This example emphasizes the fact that many parameters effect the turbine blade mechanical design evaluation, and *all* factors must be considered: frequency, blade stress and operating experience.

Summary

Turbine manufacturers' past experience indicates that a multistage mechanical drive rotor, without a blade resonance within the operating speed range in at least one stage, is the exception rather than the rule. These resonant conditions are avoided in the initial design stages if possible, but it is generally impractical, if not impossible, to avoid such occurrences. The turbine must, therefore, be designed such that operating in resonance can be achieved reliably.

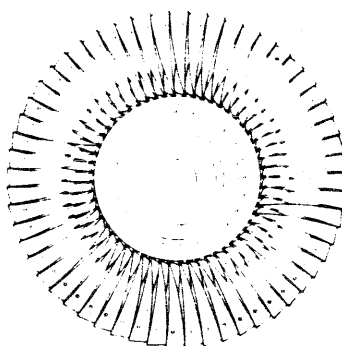
This reliability analysis consists of evaluating the blade stress levels, and comparing the results with past successful operating experience. In order to be consistently successful, the engineer must use all of the current mathematical techniques, along with a great wealth of successful blade operating experience.

The turbine manufacturer must be committed to a continuing turbine blade development program to keep current with the state of the art. These projects must include extensive testing in the Engineering Laboratory, as well as, computer development programs. These continuous advancements and data correlation enable the manufacturer to confidently offer, and more importantly successfully operate, reliable turbine blading.



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April 1976

Governors and Control Systems



*Elliott
Multistage
Steam
Turbines*

GOVERNORS AND CONTROL SYSTEMS

General

Steam turbine control systems can, at first, appear complex and mysterious, but in reality can be described as a combination of simple elements. For any system, each element must be considered and analyzed independently. Basically these elements are sensing, transmitting and correcting.

Consider first the **sensing** element. Generally, this is speed. The primary function of a speed governor is to maintain the speed of a turbine within set limits for the total load range of the unit. Speed sensing devices use (1) the "flyball" weight to sense changes to centrifugal force as a function of speed change, (2) an electric generator or pickup whose signal can be modified and corrected for control purposes, or (3) the pressure from a positive displacement pump where a flow (and pressure) change reflects speed.

The **transmitting** elements can be mechanical linkage, hydraulic or pneumatic pressures, electric signals, or a combination of above. Some systems require amplifying elements to put "muscle" into a "weak" signal. For example, an ounce in force at the speed governor may be felt as several hundred pounds at a steam admission valve. Such amplifiers are called pre-pilots and servo-motors. The "tail-end" of the control system (**correcting** element) is the steam admission valve. These are found in all manner of sizes and shapes. Most single valve turbines utilize a double-seated valve. This type of valve has characteristically a high flow at relatively low lifts with small unbalance forces over the normal lift range. A properly designed double-seated valve will exhibit the same magnitude of unbalance at any valve lift point. For high flows and pressures a venturi type valve is used. This valve has superior flow lift characteristics over the double-seated type valve. Valves per se are discussed later.

If controls are combinations of simple elements, why is there trouble with them and about them? Like any other mechanical or hydraulic device, governors are prone to certain deterrents. Principally, these are:

1. Friction
2. Lost Motion
3. Misunderstanding

Friction cannot be eliminated entirely, as we well realize, but it can be reduced to a low acceptable value (i.e. proper alignment, lubrication, proper valve stem design and adjustment). Most steam chest covers are designed with a valve stem leakoff arrangement and connection to reduce valve stem leakage without excessively increasing the pressure of packing on the valve stem. Frictional binding sensed by the packing can upset an otherwise reliable control system.

Similarly, **lost motion** can be controlled. For example, excessive tolerance in connector pins will cause lost motion which is nothing more than the governor or sensing element responding to a small change in speed which is not great enough to effect a change in the steam admission valve due to the linkage pin or connector tolerance. This problem can usually be traced and corrected.

Principally the **understanding** of a governor and or system is most necessary for its proper application and installation. Even an excellently designed control system with proper linkage and adjustment can offer poor or unacceptable performance if the selection of the system was not properly made. An attempt will be made to detail our various types of governors and controls along with a classification, specification, variation, performance and recommended application of each type.

GOVERNOR SYSTEM TERMINOLOGY

As a means of rating the performance of a governor or control system, the requirements are usually specified in terms of percent speed regulation.

Speed Regulation

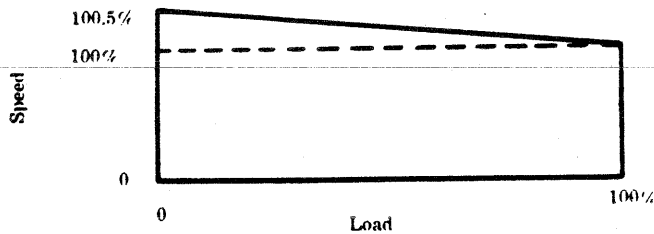
Speed-regulation, expressed as a percentage of rated speed, is defined as the change in speed when the power output of the turbine is **gradually** changed from rated output to zero output, under the following conditions:

1. Steam conditions constant
2. Speed-changer set at rated conditions
3. Turbine under no external control device

$$\% \text{ Speed-Regulation} = 100 \text{ times}$$

$$\frac{(\text{RPM @ zero output}) - (\text{RPM @ rated power output})}{\text{RPM at rated output}}$$

Most steam turbines have some "speed-droop"—that is, there occurs a drop in speed as the load is applied to the turbine. A governor that has no droop—i.e., constant speed, with load changes—is termed an "isochronous" governor. A NEMA, Class "D" governor is basically an isochronous governor.



Speed Variation

Speed variation, expressed as a percentage, is the total magnitude of speed-change, or fluctuations from the set speed. This characteristic includes dead-band and sustained oscillation.

$$\% \text{ Speed Variation} = 100 \text{ times}$$

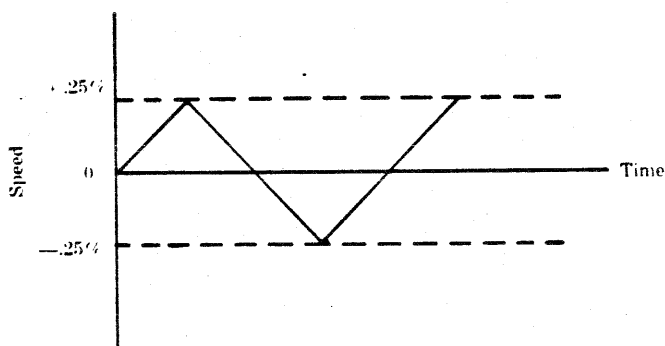
$$\frac{(\text{Change in RPM above set speed}) + (\text{Change in RPM below set speed})}{\text{Rated Speed}}$$

Dead Band

Dead band is a characteristic of the speed-governing system which is commonly known as wander. It is the insensitivity of the speed governing system that is defined as the total speed change during which there is no resultant change in the position of the governing valves to compensate for the speed change.

Stability

Stability is the measure of the ability of the speed-governing system to position the governor-controlled valves so that sustained oscillations of speed are not produced during a sustained load demand, or following a change to a new load demand.



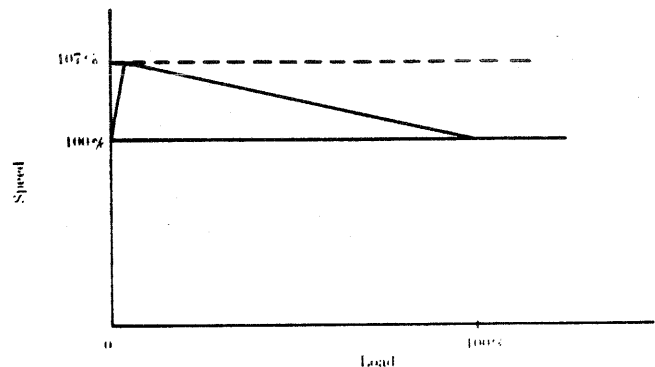
Extreme oscillation about the set point is commonly called "hunting."

Speed Rise

The governor must be capable of catching the speed-increase when the load is dropped **instantaneously** and return the system to the set point. Speed-rise is defined as the maximum momentary increase in speed obtained when the turbine is developing rated power output at rated speed, and the load is suddenly dropped to zero.

$$\% \text{ Speed Rise} = 100 \text{ times}$$

$$\frac{(\text{Max. speed at zero power output}) - \text{Rated Speed}}{\text{Rated Speed}}$$



Miscellaneous

Set point is the value of the controlled variable which the governor or the regulator is set to maintain under fixed conditions.

Speed compensation is a device that responds to movement in the governing system and loads the governor a predetermined amount. This loading compensates for the inherent regulation of the governor and results in a settled regulation of lower value than the inherent. It is often referred to as reset device, droop correction, proportional speed or floating action.

Isochronous governor is a governor that controls for constant speed, or theoretically has no regulation.

Direct-acting governor system allows only the speed sensitive portion of the governor to supply the force to position the governor valve.

Relay governor system provides for a servo or a power supply element to amplify the force positioning the governor valve.

NEMA Classifications

Governors in general are rated by NEMA classifications as shown in the following table and are used by industry in specifying governors. However, there are some "modifications" to these NEMA standards. For example, a governor may meet the requirements of NEMA "A" but not capable of a 65% speed range. Further, a NEMA D recommends 10% trip speed, but for a compressor drive, 5% could be sufficient because a maximum speed rise (instantaneous loss of load) would never occur unless a coupling failed. In some cases, a 5% trip has been used on compressor drives.

NEMA CLASSIFICATIONS Speed Governing Systems

NEMA Class	Adjustable Speed Range (%)	Maximum Steady State Speed Regulation (%) No Load to Full Load	Maximum Speed Variation Plus or Minus (%) At Rated Cond.	Maximum Speed Rise (%)
A	10 thru 65	10	0.75	13
B	10 thru 80	6	0.50	10
C	10 thru 80	4	0.25	7
D	10 thru 90	0.50	0.25	7

Overspeed Trip System Settings

NEMA CLASS	Trip Speed (% Above Rated Speed)
A	15
B	10
C	10
D	10

VALVES

Single Valve Turbines

The correcting element of a control system is the steam admission valve. Most $\frac{3}{4}$ " and 1" valves are single seated with a multiple selection of tapered plugs to permit proper selection of desired valve lift or travel at full load flow. These small

valves are used for relatively low flow rated turbines. Valve sizes 2", 3", 4", 5½" and 7" are double seated and are available in steam chests of the same size or larger than the valve size. Figure #1 shows such a valve.

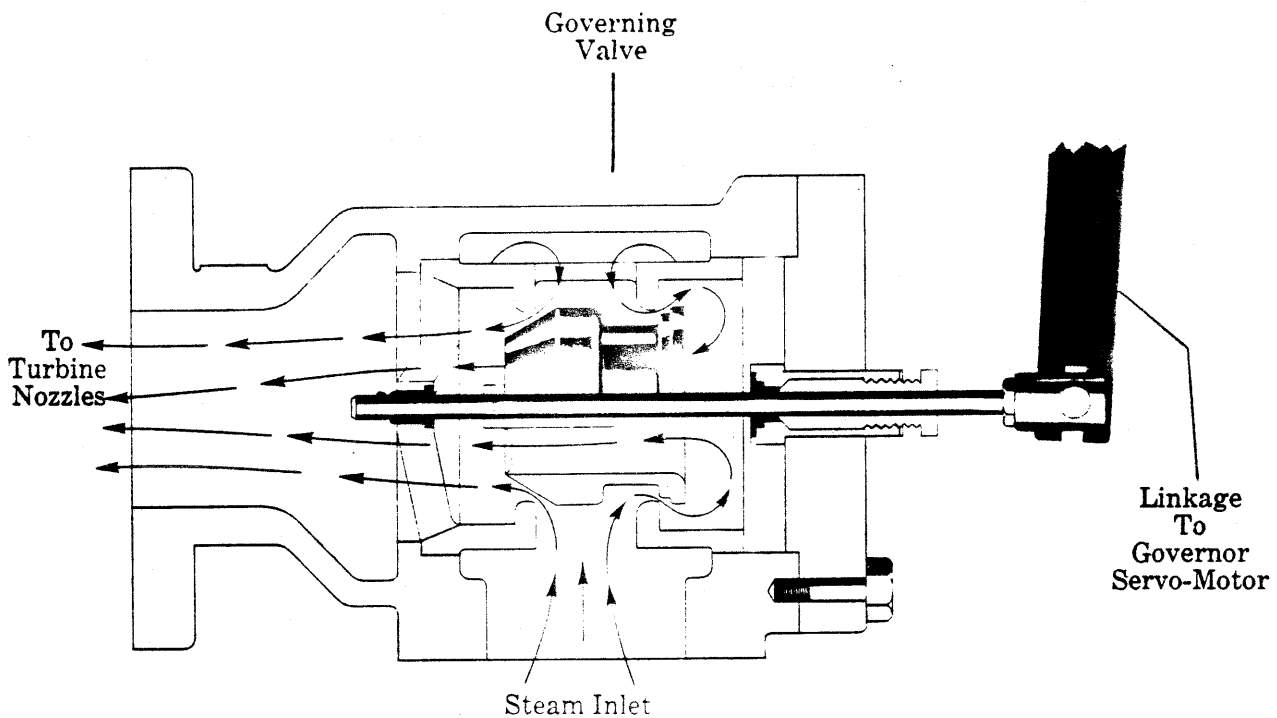


Figure 1

In a single-valve turbine, all the steam flows through a single governing or throttling valve to the turbine nozzles. Since the quantity of steam passing thru the turbine nozzles depends on the pressure of the steam (density), the quantity of steam flowing thru these nozzles will depend directly on the pressure in the turbine steam-chest. Hence, variations in steam flow are accomplished by changing (throttling) the steam pressure.

At part-loads, wherein much throttling may be required, the turbine will be inefficient, due to the energy dissipated by throttling. To reduce the part-load steam consumption, one or more hand-valves are usually provided, which can reduce some of the nozzles, thereby reducing the nozzle-area, and consequently the steam flow. A typical Willans Line is shown on Figure 2.

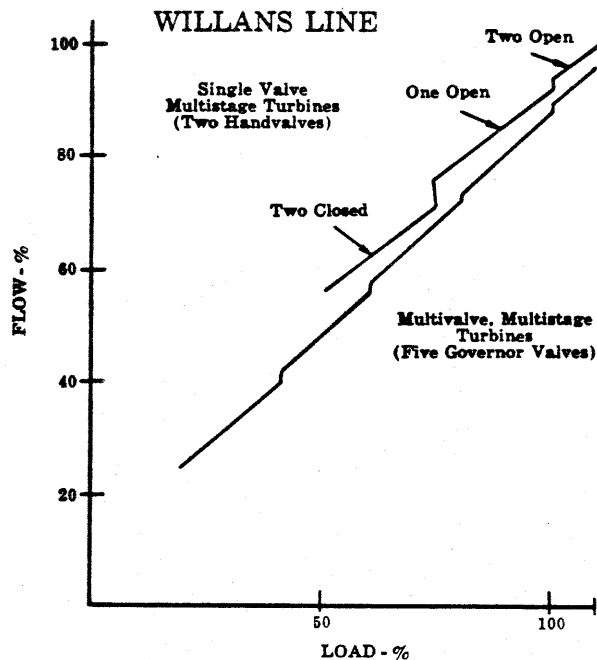


Figure 2

Multivalve Turbines

Multivalve turbines usually use a different type of valve entirely than single valve machines. Such a valve is commonly called a venturi valve. It is used for higher pressures and flows because of its superior flow-lift characteristics. A series of such valves are then used (i.e. 4, 5, 6, or 7) for finer control and better part load economy. Such valve size varies up to a nominal 5" diameter.

Grid type valves are also used, usually for extraction service. This valve offers very high flow areas required for extraction pressures of less than 50 psig. Occasionally, grid valves have been used for inlet service. Venturi valves are used for extraction pressures greater than 50 psig.

On multivalve machines, venturi valves are located in the top half of the steam chest casing and are lifted by a bar mechanism. The valves and stems are set in the bar with different stem heights to allow them to be lifted from their seats in the proper sequence and move positively in the opening direction. Motive force to move the valves in the closing direction comes from the weight of the valve itself and the unbalanced steam pressures across it. See Figure #3.

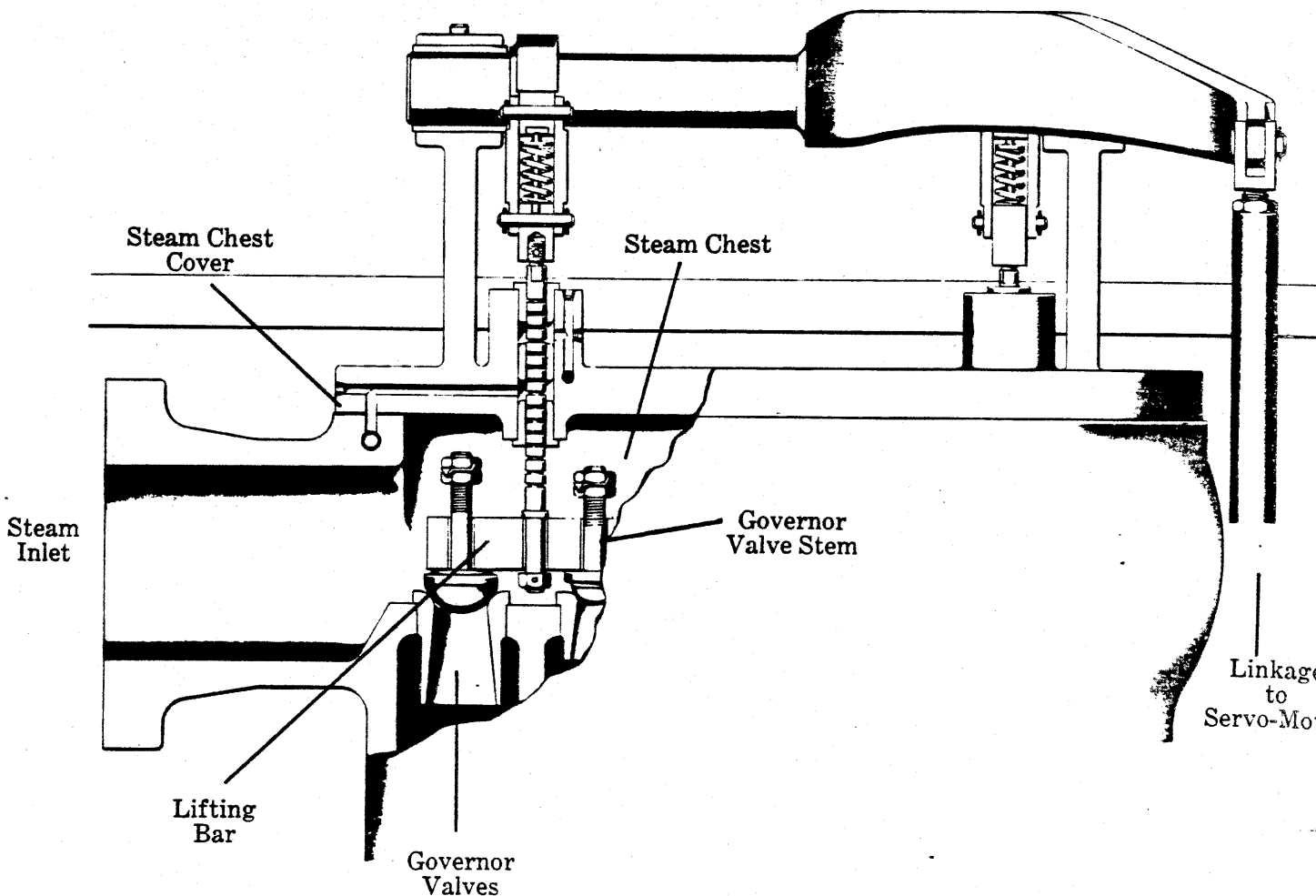


Figure 3

In a multivalve turbine, the steam flow is divided to several nozzle-groups. Each nozzle-group is controlled by an individual valve. An increase in steam flow thru the turbine is obtained by opening the valve on successive nozzle-groups, thereby, increasing the nozzle flow area. The lifting-beam has the valve-stem length so graduated that the minimum number of nozzle-groups are passing steam in order to satisfy the load requirements. When these individual valves are open, there is no throttling.

In essence, the multivalve turbine changes steam flow by cutting-in or cutting-out small increments of nozzle flow area.

Since little, if any, throttling is done, the multivalve turbine is more efficient, particularly at part-loads, or under varying-load conditions, See Figure #2.

PG GOVERNORS

Automation and process requirements are expanding and demanding more and finer control. Thus, we have more need for a governor system with muscle and precision control with specific optional modifications. The Woodward PG governor with its power cylinder (sometimes referred to as a servo motor) and a compensating system meets these additional governor system requirements and has become the industry standard. The governor is a NEMA Class C or D governor recommended for centrifugal process compressor, generator drive, reciprocating process compressor, decoking and debarking pumps, sugar mills and paper mill applications.

All PG governors have similar basic elements regardless of how simple or complex the control may be. These basic elements include:

1. Oil pump with complete built-in governor oil system including the relief valve to limit maximum oil pressure.
2. Centrifugal fly weight head pilot valve assembly to control flow of oil to and from the governor power cylinder assembly.
3. Power cylinder assembly to operate turbine control valve.
4. Compensating system to give stability to the governor.
5. Means of adjusting the governor speed setting.

This governor including the governor oil pump is driven by a worm wheel from the turbine shaft. The standard drive gear ratios are selected to provide governor speed of from 150 rpm to 1500 rpm. It is desirable to limit the maximum governor speed to 1200 rpm to eliminate the need for a governor oil cooler. The governor drive arrangement is available with a pressure turbine lube system supplying approximately 0.5 gpm of oil for drive gear lubrication. Proper worm and worm wheel backlash of 0.006" is obtained by moving governor assembly transversely and doweling the governor assembly to the housing.

The Woodward PG governor is a mechanical oil relay governor. The mechanical force developed by the governor weights is transmitted by oil pressure to a power piston moving the governor valve. The advantage of this system is that only

a small amount of force is required to move the pilot valve, as opposed to the large amount of mechanical force necessary in a direct-acting system. Consequently, smaller, more accurate elements may be used. Friction losses are thus reduced to a minimum. The result is a better governor control. The governor system is not truly isochronous due to governor valve(s) unbalance and system friction. The speed is adjusted manually by means of an external knob which is connected to the governor spring. The speed is, therefore, varied by changing the spring tension which changes the force on the governor pilot. This governor is also available with an electrical speed changer which uses an electric motor connected to the governor spring.

The PG-PL governor includes an integral air speed changer, otherwise, it is similar in operation to the standard PG governor. The air signal positions a secondary pilot valve. This pilot valve directs oil to the speed setting piston mounted on the speeder spring to change the tension of the speeder spring, similar to action on the hand speed changer in the PG governor. With a given air signal applied, the governor speed may be changed, but governor speed should remain within the following limits:

Normal speed between 250 rpm and 1050 rpm, but a low speed of 150 rpm and a high speed of 1500 rpm can be obtained if required. Governor speed changes of from 8 to 120 rpm per psi control air pressure change can be effected for PG-PL governors.

The PG-PH Governor is a modified PG Governor used for turbine-generator units where the regulation is required to permit the generator to parallel a utility system or another generator.

The output of the power piston is fed back into the speeder spring. The governor valve movement is, therefore, proportional to spring force. This provides the governor with regulation. The governor also has a built-in regulation adjustment. The regulation can be adjusted from 0 to 6 per cent. The regulation adjustment control is mounted outside the governor permitting the regulation to be readily set for each application. This governor also has a load limit feature.

ELECTRONIC GOVERNORS

The Woodward electric type governor is designated by "EG" and is intended for speed control of steam turbines driving generators, pumps, fans, compressors and paper mill machinery. The EG governor consists basically of three separate assemblies: a control box, a speed adjusting potentiometer and a hydraulic actuator.

There are two basic types of EG control boxes. The EGA model is used primarily for generator drive and receives its power supply and speed signal from the generator system. The EGM model is used primarily for mechanical drive applications and requires a separate power source. The speed signal originates from the electrical impulses generated by a magnetic pickup.

The operation of the EGM governor system is as follows (See figure 4)

1. A magnetic proximity speed pickup is mounted on the turbine to sense speed from a gear mounted on the turbine shaft. The pickup is mounted as close to the bearing as possible to reduce speed errors caused by shaft vibration and runout.
2. The speed reference signal from the pick-up is fed into the EGM control box. These speed pulses are then converted into a D.C. voltage proportional to speed. This voltage signal is then fed from the control box to the electro-hydraulic transducer portion of the gover-

nor system. The EGM consists of a chassis cover, and two encapsulated printed circuit boards. The chassis internally carries terminal strips for the input-output connections, transformer, diodes, transistors, potentiometers, etc.

The speed setting potentiometer establishes the reference voltage at the input to the EGM box and thus establishes the set point. This speed setting potentiometer can be motor operated if remote control manual speed setting is required. A pneumatic speed changer is also available.

3. The EGM signal feeds the EGR hydraulic actuator of the system. The hydraulic actuator adjusts turbine speed as dictated by the signal produced by the EGM control box. The essential element of the EGR actuator is the electro-hydraulic transducer which directs oil to and from the power piston or servo motor. This servo is remotely mounted on the turbine similar to present PG governors of multistage turbines.

Since the rotating pilot of the EGR is similar to the standard PG series, the EGR requires a motor driver and a lube system. This system is usually offered as a complete package with the only customer connection being an electrical power supply connection to the motor.

GOVERNOR SYSTEMS

General

The sensing element for the multivalve turbine control is the Woodward governor. This governor contains its own servo with given power outputs. For single valve applications this power output is generally sufficient to move a double-seated valve throughout its design range. However, for multivalve applications the forces required to operate the valves can approach 5000#. The Woodward servo cannot generate forces of this magnitude. To achieve these force levels, the Woodward servo is remotely mounted to serve as a prepilot (or slave) to a master pilot which controls the flow of high pressure oil to a large piston. This assembly of Woodward servo (prepilot), pilot valve and piston is called a servo motor. Such a control system is shown in Figure #5. Such servo motors are available in various diameters. For example, using 75 psig oil pressure and an 8-inch diameter servo motor piston, the force that can be generated is 3760# and with a 10-inch diameter servo motor piston utilizing 125 psig oil pressure the force generated is in the neighborhood of 9800#. Thus, it is readily seen how a few ounces in governor force can be multiplied through a hydraulic mechanical advantage to generate forces required to operate multivalve turbine governor valves.

Extraction Control

Steam at constant pressure for process use can be supplied from a steam turbine by adding a pressure regulating system to the speed control system. This combined speed-pressure control system is called extraction control.

The simplest manner in which to explain extraction principles and the action of the turbine is to think of the turbine as having two steam flows, exhaust and extraction, as indicated by Figure 6. The power generated by the steam depends upon the available energy and the quantity, in the conventional manner. Thus, if exhaust flows and extraction flows are equal, the exhaust flow will generate more horsepower than extraction flow, because it is subjected to a greater pressure drop.

These simple rationalizations are shown in the extraction diagram, Figure 7, in terms of tangible pounds of steam and effective horsepower. The vertical coordinate represents the total steam flow through the turbine inlet, the sum of exhaust and extraction flows, and the horizontal coordinate shows the total horsepower developed. The zero extraction line represents the steam flows

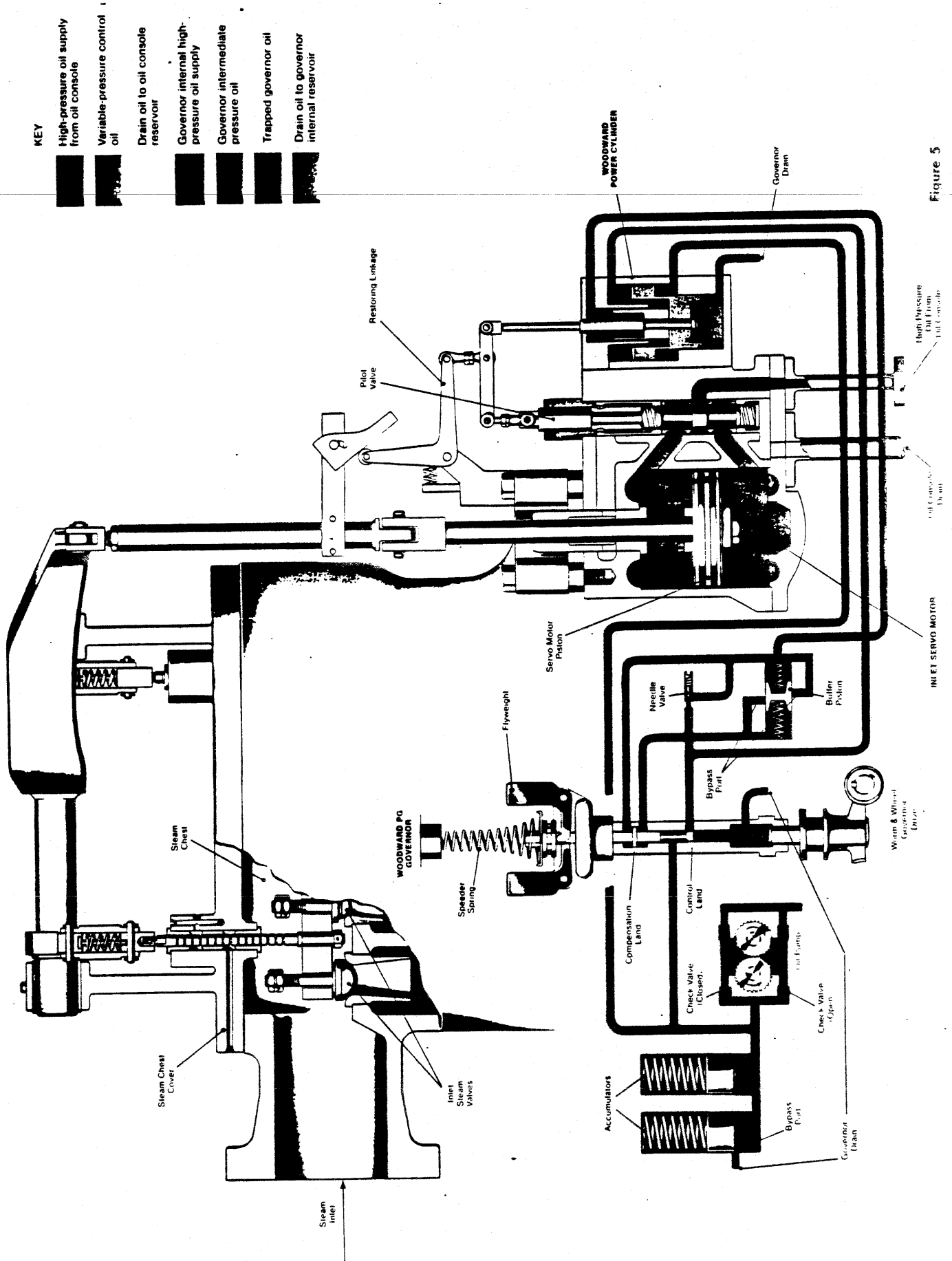


Figure 5

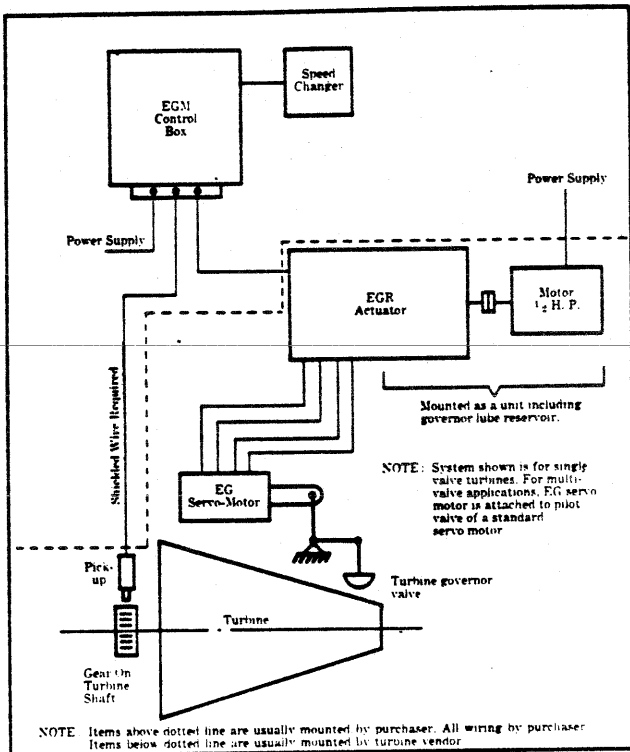


Figure 4

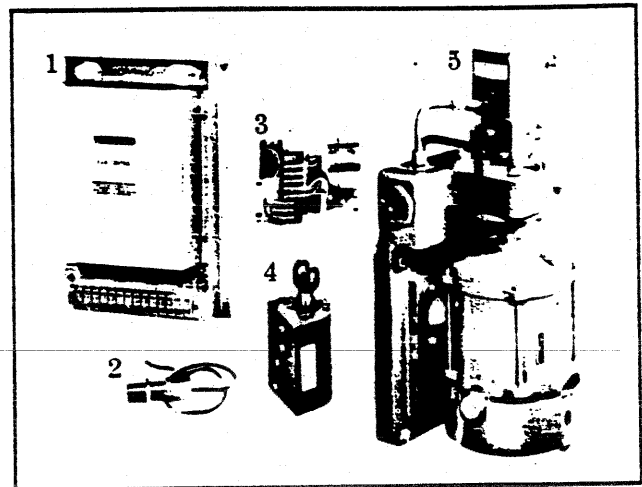


Figure 4A

1. EGM Control Box
2. Magnetic Proximity Speed Pickup
3. Speed Setting Potentiometer (Speed Changer)
4. Servo Motor
5. EGR Actuator (Also showing motor driver and lube system)

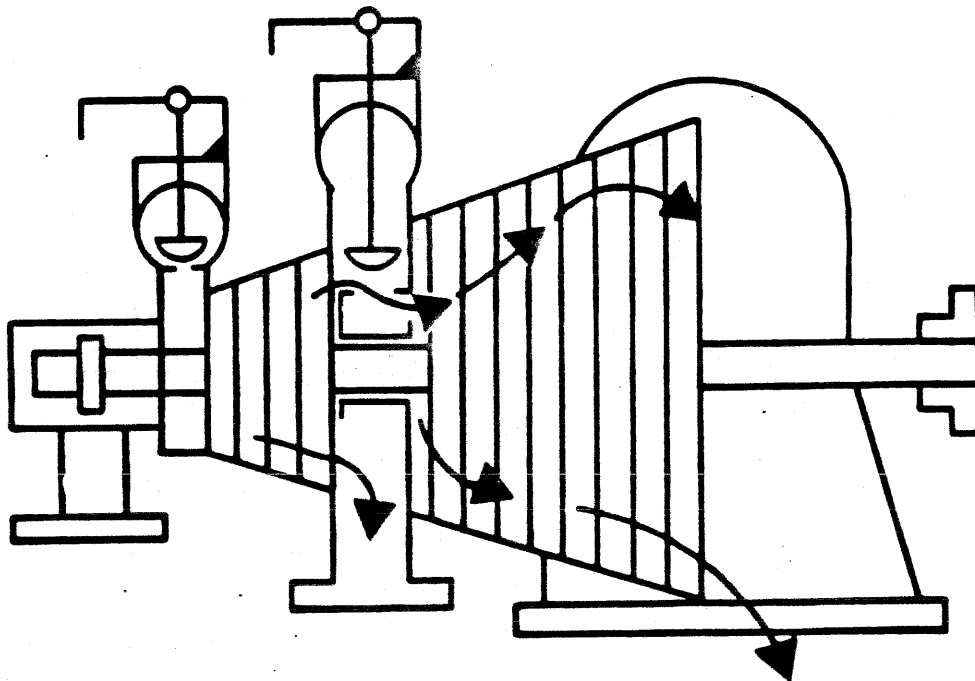


Figure 6

required for various loads when extraction flow is zero. The total extraction line represents the steam flow required when exhaust flow is zero (except for cooling steam). The extraction lines show the total inlet steam flow for various horsepower loads when the flow indicated is extracted. Thus, 32500 horsepower with an extraction flow

to the extraction line of 250,000 pounds per hour would require a total flow of 350,000 pounds per hour. By indulging in some "kitchen arithmetic," exhaust flow is obtained as the difference between total flow and extraction flow, that is 100,000 pounds per hour.

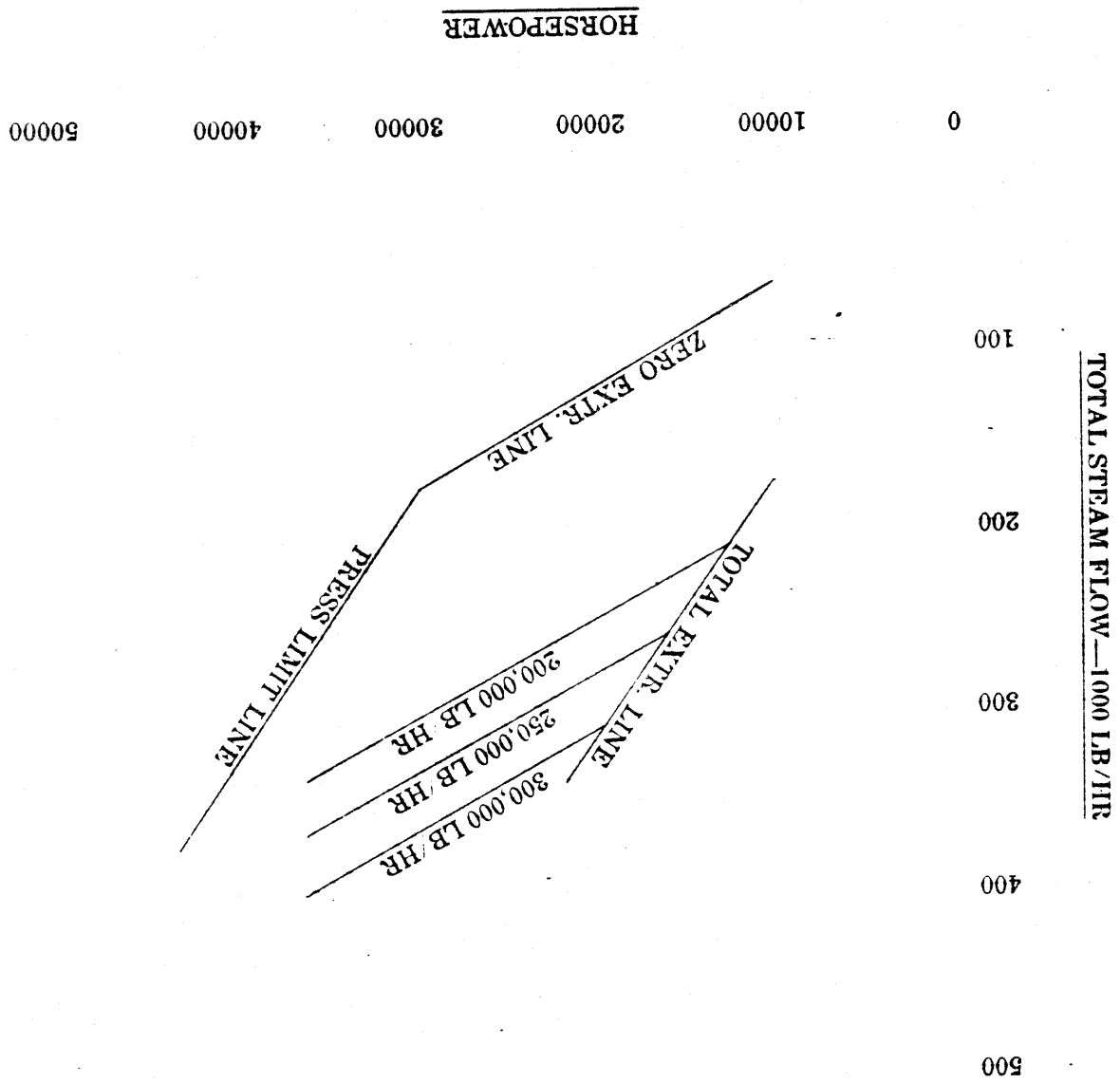


Figure 7

At 16000 horsepower the 250,000-pound per hour line terminates in the total extraction line, and at 12500 horsepower there is no 250,000-pound per hour extraction line. The story as told by the chart is that if the process needs 250,000 pound per hour of steam, the turbine must be loaded to at least 16000 horsepower. At 12500 horsepower the turbine will give 200,000 pound per hour for extraction flow and, if 250,000 # hr. is required an additional 50000 # hr. must be supplied by a reducing valve in the steam header system or by another source.

The pressure limit line is the limit line for exhaust flow and is the upper terminus for the extraction lines. To the right of this line, exhaust flow exceeds the capacity of the turbine exhaust and no more steam can make its exit by that route without an extraction stage pressure increase.

The operating limits of the extraction turbine are between the total extraction line and the pressure limit line, with the size of the turbine inlet as the upper limit, and the zero extraction line as the lower limit (exhaust size). Inside of these limits the turbine is free to perform and satisfy demands for extracted steam. And if these limiting conditions are fully understood and considered in the original design, a most satisfactory system, capable of the necessary flexibility, will result.

The extraction control system consists of a speed governor, a pressure regulator, and two sets of prepilots and servo motors (for steam admission valves and extraction valves). Refer to Figure 8. In essence, the pressure regulator receives two different simultaneous signals. One from the governor and the other from the process. Each function must be analyzed separately although one must realize that each action occurs simultaneously.

The regulator receives the signal from the governor through the Woodward hydraulic lines, the Woodward servo mechanically positions the regulator rod. Note that the Woodward servo motor is mounted on the regulator and not the servo motors. The regulator rod is mechanically connected to the outer springs of the two-cup valves in the pressure regulator, which in turn transmit a hydraulic signal to the prepilots on each servo motor. When on governor control, each set of valves moves proportional to the load on the turbine for a given (constant) extraction flow demand.

The regulator receives a pressure signal from the process (shown on Figure 8), and through the regulator hydraulic and mechanical linkage works on the innersprings to regulate the cup valve pressure which in turn is transmitted to the prepilots on each servo motor. In this case the two valve gear mechanisms move the opposite direction for a given load and changing extraction demand. For higher extraction flows, the inlet valves open, the extraction valves close. For lower extraction flows, the inlet valves close and the extraction valves open. Note that the extraction valves dictate the turbine exhaust flow and the resulting extraction flow is a secondary effect.

To reiterate, each of the above changes occur simultaneously and the regulator acts accordingly within a given extraction map.

In conclusion, one must keep in mind two important and often misunderstood aspects of extraction control. First, the "heart" of the system is the pressure regulator which receives signals from two sources (governor and process) and transmits two intelligence signals to the valves. Second, the extraction valves are an integral part of the turbine and are not valves in the process line. These valves limit turbine exhaust end flow. When closed, maximum extraction flow occurs and when open, maximum turbine exhaust end flow results.

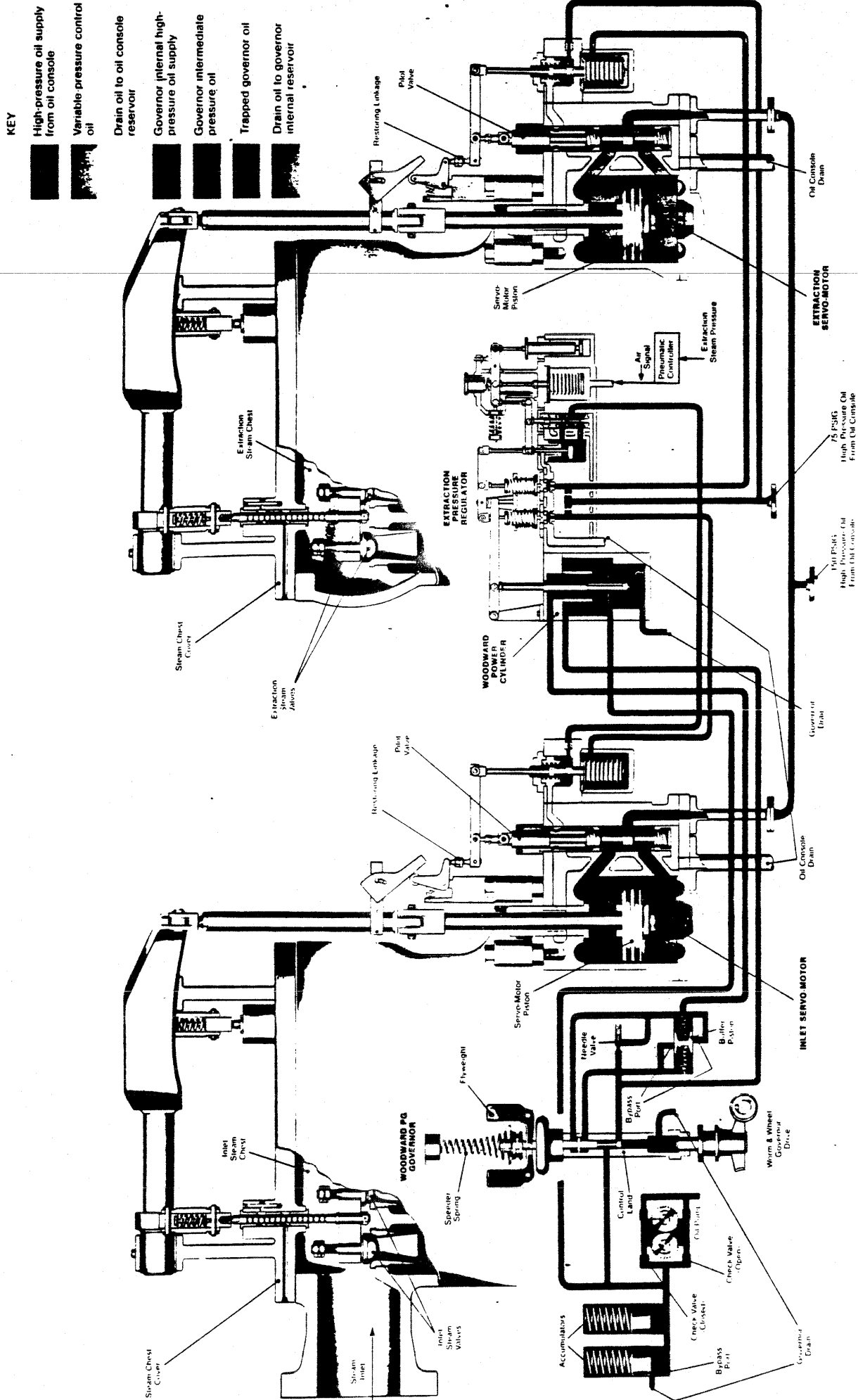
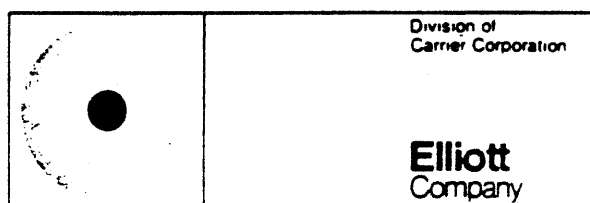


Figure 8



- Mechanical Drive Turbines
- Air and Gas Compressors
- Turbochargers
- Tube Cleaners
- Tube Expanders
- Roller Burnishers

Elliott Company General Offices:

Jeannette, Pennsylvania 15644

Other plants at Dayton and Springfield, Ohio;
Isle of Wight, England

Canada: Elliott Industrial Equipment Company
24 Queen St. East, Brampton, Ontario

Europe, Middle East, Africa:

Elliott Turbomachinery Ltd.
197 Knightsbridge, London, England

Latin America: Elliott Overseas Company

1570 Madruga Avenue
Coral Gables, Florida 33146

Far East: Elliott Overseas Company

P.O. Box 1112, City of Industry, California 91749

The dynamic-type blade testing is the second basic type of analysis. Dynamic tests are conducted to determine blade natural frequencies as well as blade response to forced excitation at operating speeds.

The dynamic testing methods are recognized as the most complete analysis, however, this method involves a considerable amount of time and expense. Therefore, dynamic-type testing is only used in critical areas where static testing is not completely suitable.

The dynamic-type test hardware basically consists of a test casing assembly, driver, and dynamometer. The casing is usually a cylindrical chamber equipped with a series of nozzles. These nozzles are spaced in such a manner to provide excitation at various orders of rotor speed and nozzle passing frequency.

The test rotor is positioned inside the casing and consists of a full size disk assembly fitted to the test rig shaft. This shaft is drilled in both radial and axial directions to permit lead wires from the test instrumentation to be brought out from the test casing to a slip ring assembly fitted into the end of the shaft. (Photographs 1 and 2 illustrate typical dynamic test hardware.)

The dynamic testing performed on a 16" blade is used as an illustrative example to demonstrate the techniques and results which can be obtained from this type of testing.

The testing sequence is to first define the various frequencies, mode shapes, and relative responsiveness for the assembly by static-type testing methods. This enables both the blade and disk frequencies to be determined throughout the frequency range of interest prior to the actual rotating test.

The next step is to perform the predetermined series of dynamic tests. The data obtained from the dynamic tests is recorded and preserved on magnetic tapes for further reference and analysis. (Fig. 7 is a sample of a typical response plot obtained from a 16" blade testing.) These response curves are plots of the strain gage output (blade stress) versus shaft speed.

The strain gage output or stress will increase as a resonant condition is approached. (Fig. 7 demonstrates the blade response at the resonant condition of the fundamental tangential frequency with a three per revolution exciting force.)

These response plots are used to develop the desired dynamic test information. For example, dynamic testing indicates the effects of rotation on the blade natural frequency.

Obviously, this effect cannot be detected by the static test approach. The centrifugal forces acting on the blades exhibit a stiffening effect which raises the blade natural frequency during rotation. This effect is small and insignificant for short blades, but must be considered on longer blades.

The blade natural frequency versus shaft speed characteristic can be derived from the information contained on the response curves. This centrifugal stiffening effect is shown on the Campbell Diagram for a 16" blade in Fig. 8. As can be seen from this Campbell Diagram, both of the plotted fundamental frequencies increased by more than 30%.

Dynamic-type testing also enables the engineer to evaluate the blade system damping. Damping is an ex-

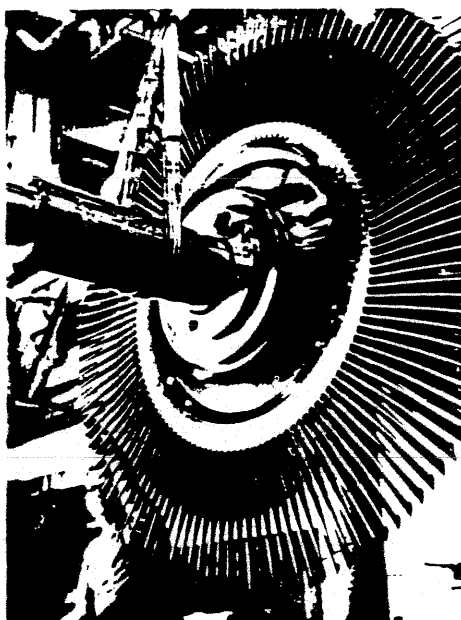


PHOTO 1 Bladed disk is lowered into rotating test assembly to determine the effect of rotation on blade and disk frequencies.

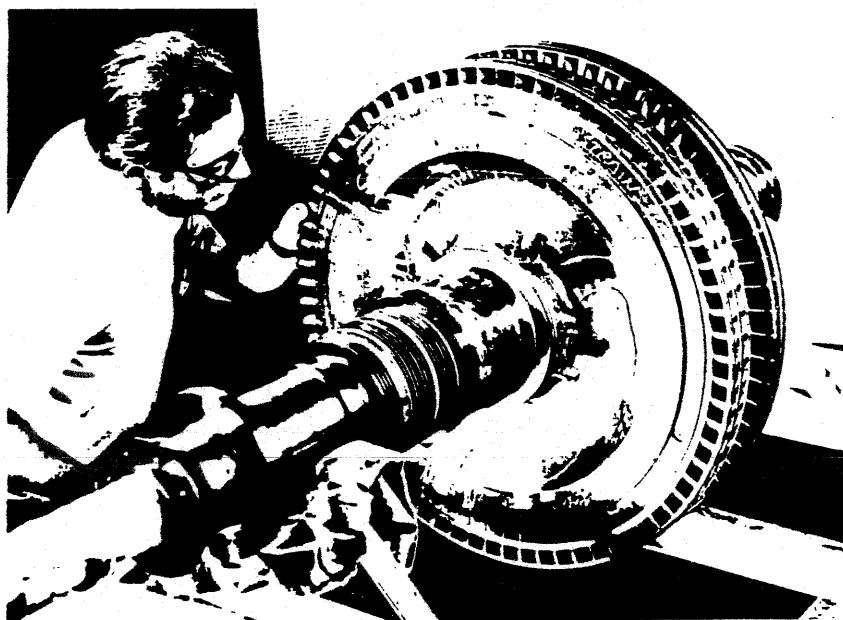


PHOTO 2 First-stage Curtis disk assembly being prepared for dynamic testing to determine mechanical and thermodynamic characteristics.

remely important factor for blades which operate at a resonant condition within the operating speed range of the equipment. The system damping can be derived from the dynamic test response plots. This aspect is a very important result of the testing.

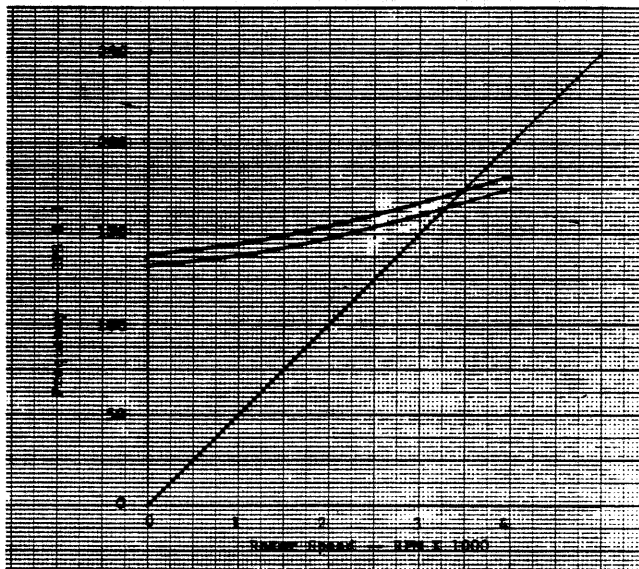


Fig. 8. 16" blade Campbell Diagram.

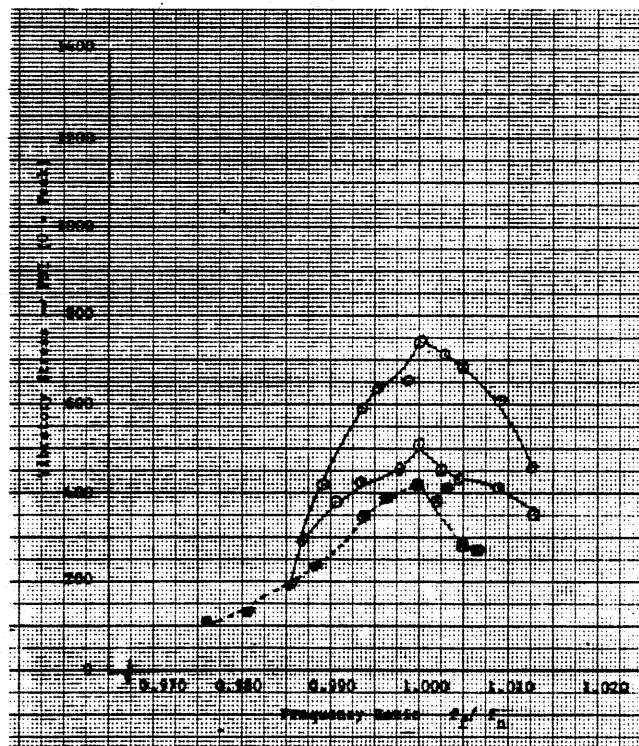


Fig. 9. Sample of a typical resonant response curve.

Once a resonant condition is found during the test, the rotor speed can be controlled in such a manner as to slowly sweep above and below this resonant location until the condition is fully recorded on magnetic tape. The response curves obtained from these tests can be transposed and replotted as the ratio of forcing frequency to natural frequency as shown in Fig. 9.

The sharpness of the resonant response curve (Fig. 9) is related to the amount of energy dissipation by damping in the system. The actual value for the system damping can then be calculated from the resonant response curve. Having determined the actual value for the system damping, the engineer is now able to accurately calculate the vibratory stress in the blade.

The previous paragraphs have illustrated the important role that static and dynamic testing play in the turbine blade analysis. Mathematical models such as Goodman and Campbell Diagrams also contribute to the total evaluation.

Campbell Diagrams

The Campbell Diagram is one blade design tool used by turbine blade designers. This type of diagram is the most common and generally accepted format used when presenting blade natural frequency information as a function of turbine speed. This diagram will graphically display those operating areas where a blade system resonant condition will exist.

The Campbell Diagram contains frequency data pertaining to the potential sources of significant blade excitation forces. The frequency of these significant sources are at multiples of running speed and nozzle passing frequency. These exciting forces are described in the following few paragraphs.

The turbine blades are subjected to periodic steady forces from the nozzles they pass. This relatively high frequency excitation is caused by the variations in the steam force as the blade passes the wakes of successive nozzle vanes which are uniformly pitched around the annulus. (See Figs. 10 and 11.)

The frequency of this exciting force is defined as the "nozzle passing frequency." This stimulus has a period equal to the time required for the blade to travel a distance of one nozzle pitch or, in other words, the stimulus has a fundamental frequency equal to the number of nozzles which the blades pass per unit of time.

$$f = \frac{nN}{60}$$

where: f = Nozzle Passing Frequency—Hertz
 n = Number of Nozzles per Circle
 N = Shaft Rotational Speed—r/min

The turbine blades are also subjected to periodic exciting forces which are multiples of the shaft rotational speed. The sources of these forces may be classified as internal or external to the turbine. However, in general, the main sources for these exciting forces are internal to the turbine.

The source of blade excitation is mainly due to discontinuities in the steam flow path in the area around the nozzle annulus upstream of the turbine blading.

A single discontinuity in the steam flow path upstream of the blade is an example of a one times running speed excitation.

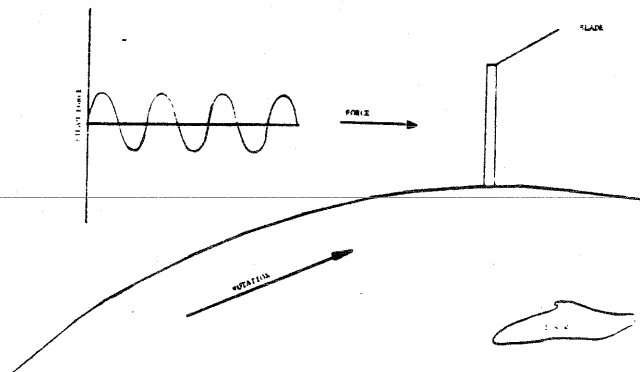
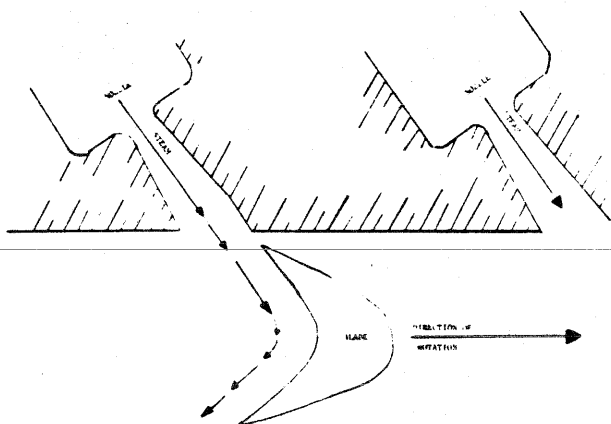


Fig. 10 and 11. Nozzle passing frequency sketches.

This discontinuity could be caused by an enlarged or reduced pitch in the stator blades, or a single obstruction upstream of the rotating blades. An example of an exciting force equivalent to a two times running speed would be the nozzle pitch variation in the area of the horizontal split of a steam turbine diaphragm.

These examples occur on any number of running speed multiples. The reason is because many of the sources for these exciting forces are discontinuities or interruptions in the steam flow path which are random, undefined, and to a certain extent, cannot be avoided.

$$f = \frac{N}{60} \times \text{Order of Running Speed Harmonic}$$

where f = the exciting force frequency in Hertz imposed on the rotating blade operating behind any number of equally spaced interruptions.

N = blade rotation speed in r/min

The Campbell Diagram must include all of significant sources of blade excitation (orders of running speed and nozzle passing frequency). The obvious questions then arise, which multiples of running speed and nozzle passing frequency should be considered in the Campbell analysis.

This item remains somewhat arbitrary, for one could indicate as many multipliers as required to obtain a resonant condition within the operating speed range for a particular machine. This is a ridiculous statement but points out the need for some additional study in this area.

The resonant response of a blade diminishes as the harmonics of running speed increase (i.e., greatest resonant response at one times running speed and decreasing at two and less at three, etc.). This fact is also well documented in published literature. Based upon the blade operating environment, experience, and the manufacturer's blade testing programs, all of the significant sources of blade excitation are identified on the Campbell Diagram. (Fig. 12 is a sketch of a typical Campbell Diagram.)

The following paragraphs will summarize uses and interpretations of the Campbell Diagram and its relationship with the Modified Goodman Diagram.

In reality, the Campbell Diagram can best be described as an "Interference Diagram." An interference occurs when one of the blade natural frequencies intersect various operating modes such as harmonics of running speed and one times nozzle passing frequency.

A blade operating at an interference condition is likely to have a fatigue failure if the vibratory stress levels exceed

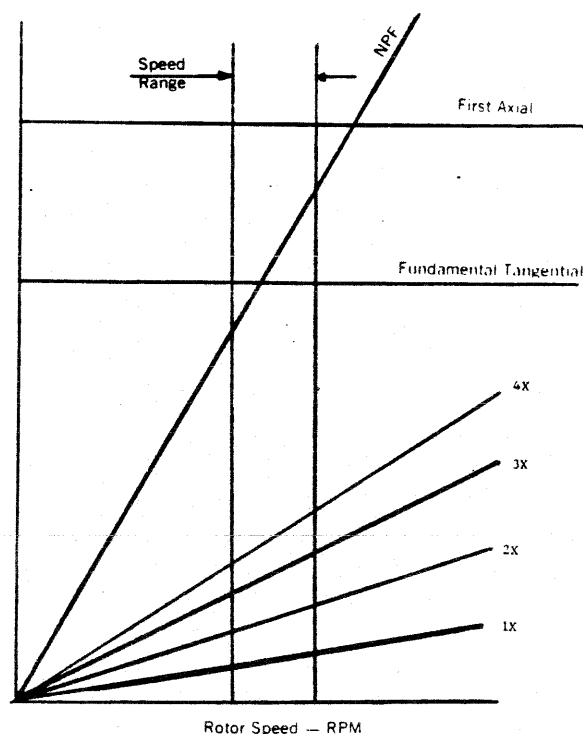


Fig. 12. Typical Campbell Diagram.

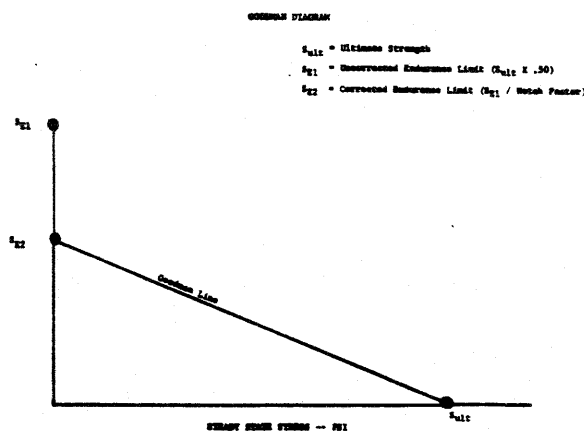


Fig. 13. Goodman Line.

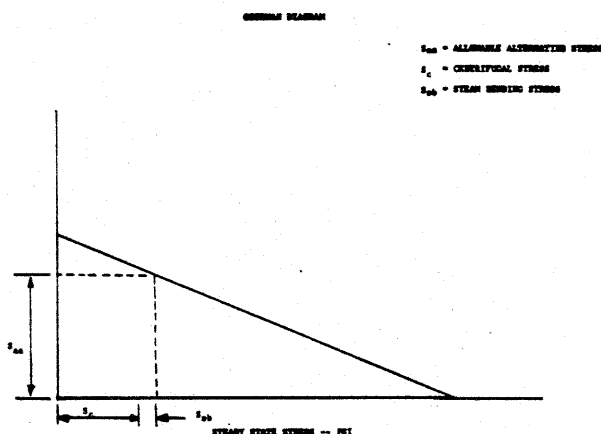


Fig. 14. Goodman Diagram stresses.

the allowable limits. A blade path selection for a particular application should always attempt to avoid an interference condition.

The likelihood of selecting all of the blades in a particular multistage turbine where their natural frequencies are not close to a resonant condition is generally impractical, if not impossible. When an interference condition cannot be avoided, the machine must be designed so operating in resonance can be achieved reliably.

Therefore, if a resonant condition (interference on the Campbell Diagram) does exist, the allowable margin for stress amplification must be evaluated. This evaluation is in the form of a Goodman Factor calculation.

Goodman Diagrams

The Goodman Factor calculation is the design tool used to evaluate the reliability of a blade operating in resonance. This factor is nothing more than the ratio of allowable cyclic stress to the maximum steam bending stress. This stress ratio must have a minimum value, above which reliable operation in resonance can be assumed for a particular geometry.

The manufacturer's blade testing programs and past successful operating experience has established these safe

minimum Goodman Factor guidelines. This Goodman Factor calculation procedure is derived from the construction of a Goodman Diagram.

The Goodman Diagram is used as a method for evaluating the combined effects of alternating and steady state stresses in turbine blading. (Figs. 13 and 14 are illustrations of the basic Goodman Diagram.) The abscissa represents the steady state stresses, and the ordinate represents the alternating stresses in the component being evaluated. The theoretical failure line is constructed by drawing a straight line between the Ultimate Strength on the steady state stress axis to the Corrected Endurance Strength on the alternating stress axis.

This Goodman Line, in theory, represents the minimum combinations of steady state stress and alternating stress which would produce a blade fatigue failure. Theoretically, any point below the Goodman Line is safe, and anything above the line is unsafe.

The analysis utilizes three stresses contained on the Goodman Diagram. These stresses are tabulated below:

1. Centrifugal Stress
2. Steam Bending Stress
3. Vibratory Stress

The centrifugal stresses are caused by the blade rotating about the shaft centerline. The steam bending stresses are caused by the steam momentum and static pressure changes in a particular turbine stage. The vibratory stresses are usually caused by variations in the steam bending stress.

The centrifugal and steam bending stresses are considered steady state stresses, and the vibratory stress is considered a cyclic stress. (See Fig. 14.)

The steady state stress component calculation procedure is relatively straight forward, compared to the vibratory stress calculation. Considerable analytical work is underway to find methods to calculate the vibratory

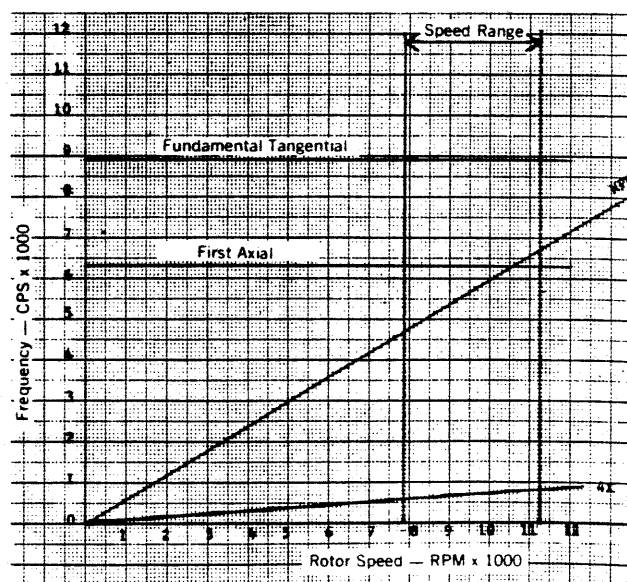


Fig. 15. Example calculation, Campbell Diagram.