

General Specifications English/Metric

Frame	AYR	BYR (& BYRIH)	CYR	DYR (& DYRM)	BYRH (& BYRHH)
Maximum initial gage pressure (psi/bar)	700/48	700/48	700/48	700/48	700/48
Maximum initial temperature (°F/°C)	750/399	750/399	750/399	750/399	750/399
Maximum exhaust pressures (gage, psi/bar)	vac-100/6.9	vac-100/6.9 ⁽¹⁾	vac-90/6.2	vac-100/6.9	250/17 ⁽⁴⁾
Speed Range (rev/min)	1000-7315	1000-6675	1000-6950	1000-5770	1000-7090
Wheel pitch diameter (in./mm)	14/360	18/460	22/560	28/710	18/460
Number of stages (impulse type)	1	1	1	1	1
Number of rows of rotating blades	2	2	2	2	2
Inlet sizes (ANSI, in.)	3"	2", 3", 4"	2", 3", 4", 6"	2", 3", 4", 6"	2", 3", 4", 6"
Inlet location (facing governor)	right	right	right	right	right
Exhaust size (ANSI, in.)	6"	8"	10"	12" ⁽²⁾	8"
Exhaust location (L.H. Standard)	R.H. optional	R.H. optional	R.H. optional	R.H. optional ⁽³⁾	R.H. optional
Approximate range of capacities (hp/kW)	750/560	to 1400/1050	to 2500/1850	to 3500/2600	to 3000/2250
Approximate shipping weight (lb/kg)	870/400	1275/580	2050/930	2600/1180	2300/1050

See Form PP2641 for BYRH and BYRHH details;
Bulletin H-32 for AYR details.

- (1) BYRIH: 160 psi/11 bar
(2) DYRM: 14" max. exhaust pressure 75 psig

- (3) DYRM: Optional R.H. not available
(4) BYRHH: 375 psi/25 bar

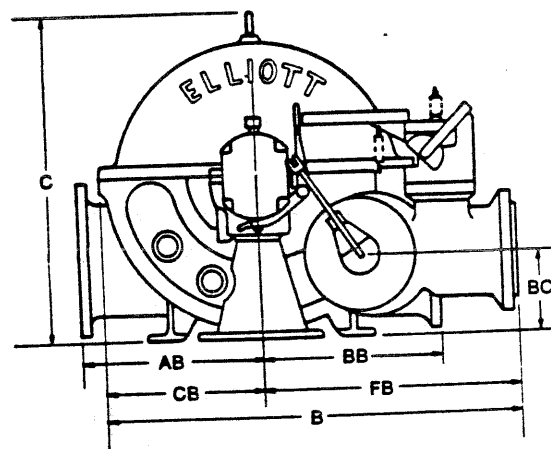
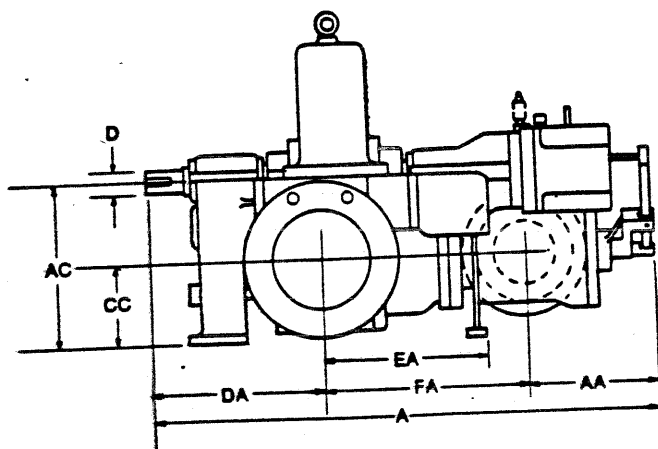
Approximate Dimensions Inches, Millimetres

	A	AA	FA	DA	EA	B ⁽¹⁾	AB ⁽²⁾	BB ⁽²⁾	CB	FB ⁽¹⁾	C	AC	BC	CC	D
YR	40.56	12.00	15.74	12.82	15.18	31.12	13.74	13.74	12.62	18.50	24.00	12.00	5.00	6.50	1.9325
	1030	305	400	325	386	791	349	349	321	470	610	305	127	165	49.09
BYR, BYRIH	45.88	12.26	21.18	12.44	17.30	36.12	16.00	16.00	13.62	22.50	28.00	14.00	7.74	7.00	1.9325
	1165	311	538	316	440	918	406	406	346	572	711	356	197	178	49.09
CYR	51.24	12.26	21.18	17.82	16.06	40.12	19.24	19.24	16.62	23.50	34.74	17.00	8.50	8.50	2.4325
	1301	311	538	452	408	1019	489	489	422	597	883	432	216	216	61.78
DYR	50.88	12.26	21.68	16.94	16.56	43.12	22.24	22.24	19.62	23.50	42.50	22.00	13.50	12.00	2.9325
	1292	311	551	430	421	1096	565	565	499	597	1080	559	343	305	74.48
DYRM	50.88	12.26	22.68	15.94	17.56	43.12	23.00	23.00	19.62	23.50	42.50	22.00	13.50	11.24	2.9325
	1292	311	576	405	446	1096	584	584	499	597	1080	559	343	286	74.48
BYRH, BYRHH	55.38	12.26	27.00	16.12	21.88	39.88	18.74	18.74	16.38	23.50	31.12	17.00	8.50	9.00	2.4325
	1407	311	686	410	556	1013	476	476	416	597	791	432	216	229	61.78

(1) AYR, BYR, BYRIH 3" inlet. 4" inlet for
all other frames.

(2) For Class III

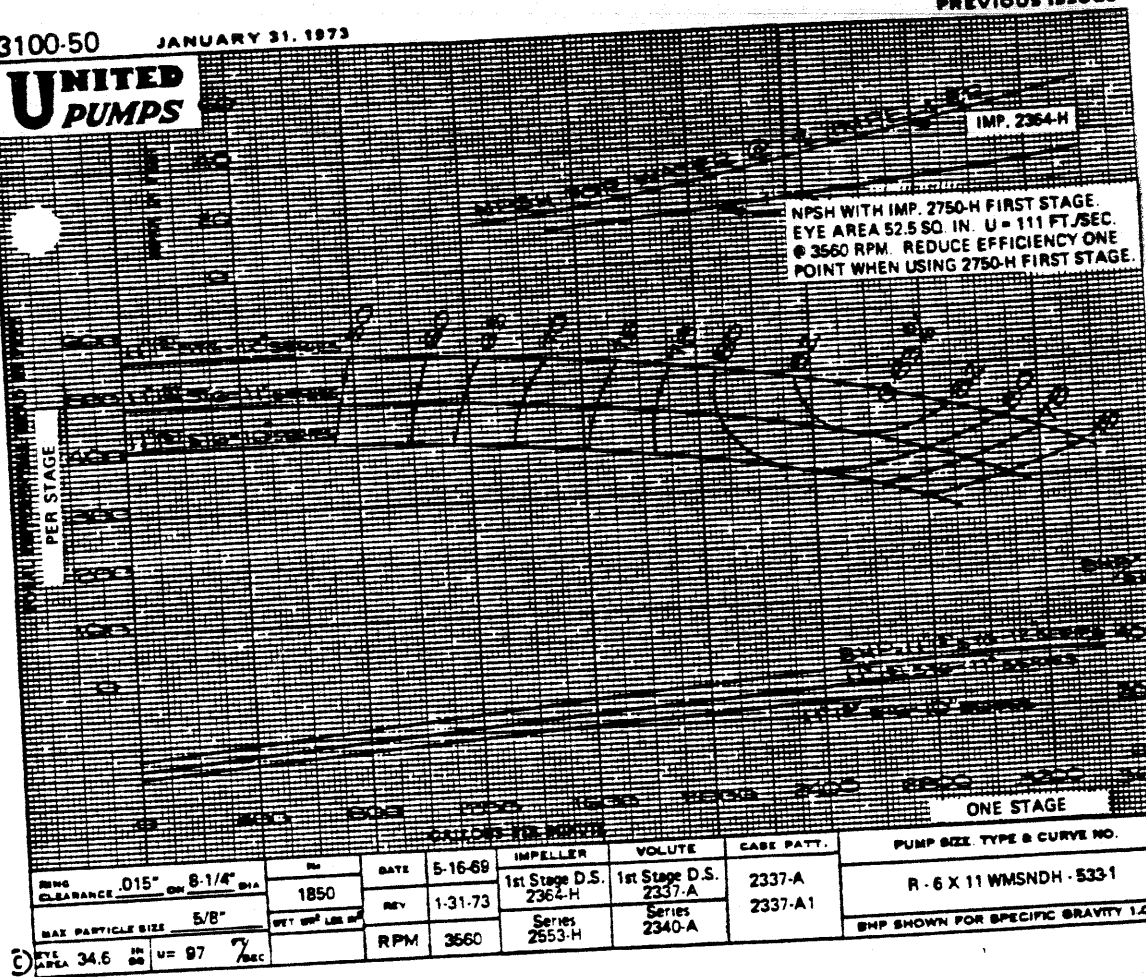
Left hand exhaust (AB dimension) is standard.



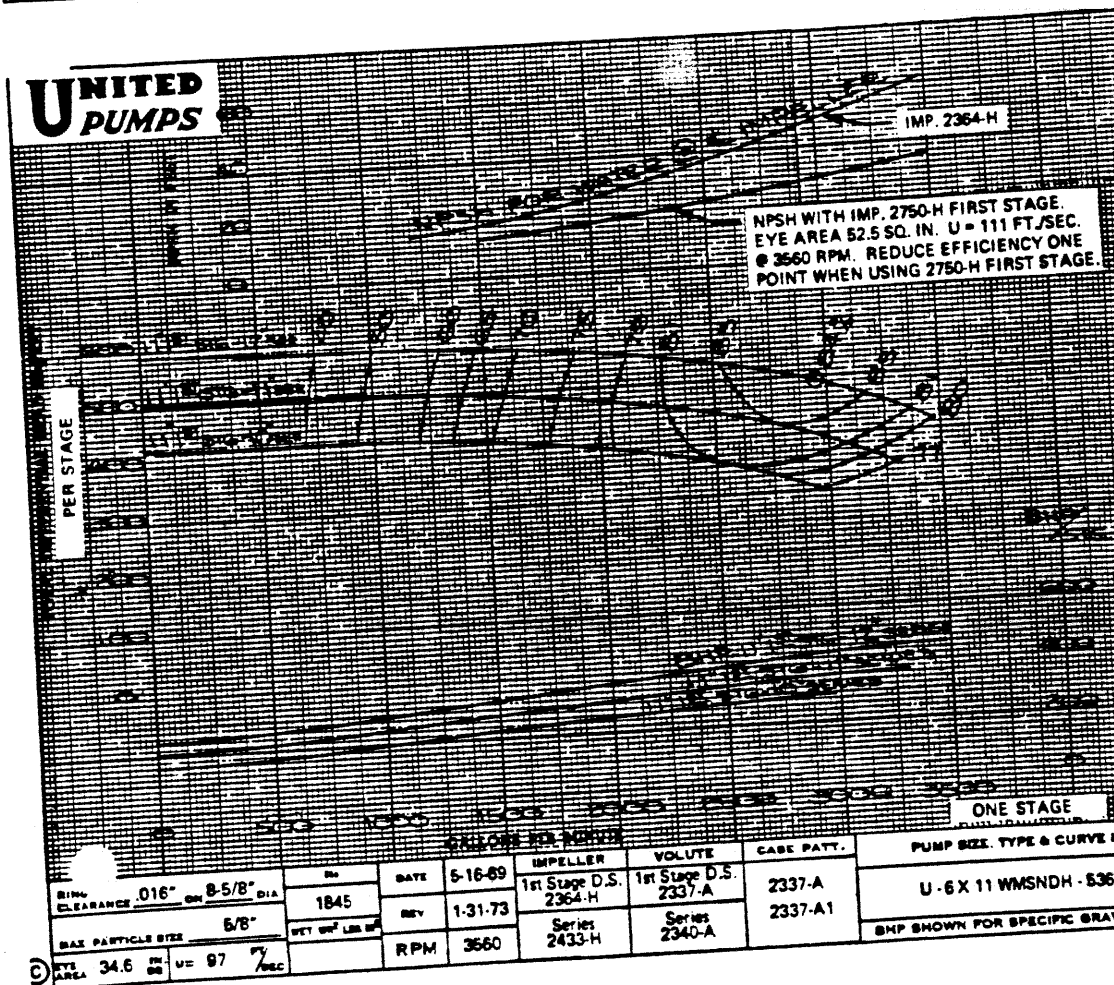
3100-50

JANUARY 31, 1973

**UNITED
PUMPS**



**UNITED
PUMPS**



Inlet Velocity (FT/sec)

$$= \frac{\text{Flow (lb/hr)} \times \text{Sp. Vol (FT}^3/\text{lb)}}{19.65 \times \text{PIPE DIA (in)}^2}$$

$$= \frac{15636 \text{ ~~56.8~~ LB/HR} \times 2.0 \text{ FT}^3/\text{LB}}{19.65 \times \text{ID}^2} < 175$$

$$\therefore \text{ID} = \text{~~2.78~~} 3.012$$

$\therefore 4''$ is OK.

OUTLET Velocity (FT/sec)

$$= \frac{15636 \text{ ~~56.8~~} \times 4.1}{19.65 \times \text{ID}^2 (\text{in})} < 450$$

$$\therefore \text{ID} = 2.69''$$

$\therefore 3''$ pipe is OK.

PROBLEM III - SOLUTION

1) CALCULATE STEAM RATE

TSR = 31.1 LB/KW-HR FROM MOLLIER

BASIC STEAM RATE = 48 LB/HP-HR AT 3000RPM

MECHANICAL LOSS = 9.6 HP

SUPERHEAT = 94°F

SUPERHEAT CORRECTION FACTOR = 1.032

$(48/1.032) \times (275 + 9.6)/275 = 48.1 \text{ LB/HP-HR}$

STEAM FLOW = $48.1 \times 275 = 13200 \text{ LB/HR}$

2) FIND INLET AND EXHAUST VELOCITIES

INLET SIZE = 3 INCH

EXHAUST SIZE = 6 INCH

$$\text{VELOCITY} = \frac{\text{FLOW (LB/HR)} \times \text{SP. VOL (FT}^3\text{/LB)}}{19.65 \times \text{PIPE DIAMETER}}$$

$H_1 = 1263$

$\Delta H = 2545/48.1 = 53$

$H_2 = 1210, P_2 = 115 \text{ PSIA}$

$T_2 = 370^\circ\text{F (30}^\circ\text{ SH)}$

INLET SP. VOL = 2 FT³/LB

EXHAUST SP. VOL = 4.1 FT³/LB

INLET VELOCITY = 149 FT/SEC

EXHAUST VELOCITY = 77 FT/SEC

PROBLEM II - SOLUTION

1) ESTIMATED STEAM RATE

THEORETICAL STEAM RATE (TSR):

USING TABLE - 8.49 LB/KW-HR
MOLLIER - 8.50 LB/KW-HR

BASIC EFFICIENCY = 0.75
SUPERHEAR FACTOR = 1.002 (86°F)
SPEED FACTOR = 0.968)

$$\text{ESTIMATED STEAM RATE} = \frac{(8.49)(.746)}{(.75)(1.002)(.968)} = 8.7 \text{ LB/HP-HR}$$

2) PERCENT MOISTURE IN EXHAUST

$$H_1 = 1272 \text{ BTU/LB}$$

$$\Delta H = 2545/8.7 = 292.5$$

$$H_2 = 979.5 \text{ BTU/LB}$$

3.5 INCHES HG ABS (1.72 PSIA)

FROM MOLLIER, 13% MOISTURE

3) ESTIMATE CONDENSER DUTY AND GPM CONDENSATE

$$(10000 \text{ HP}) \times (8.7 \text{ LB/HP-HR}) = 87000 \text{ LB/HR}$$

$$(87000) \times (.13) = 11310 \text{ LB/HR OF } 120^\circ\text{F CONDENSATE}$$

$$87000 - 11310 = 75690 \text{ LB/HR OF } 120^\circ\text{F VAPOR}$$

$$\text{DUTY} = 75690 \times 1025^* \text{ BTU/LB} = 77.6 \text{ MMBTU/HR}$$

*ASSUMES CONDENSATE RETURN AT 120°F

$$\frac{87000}{(8.33)(.9886)(60)} = 176 \text{ GPM CONDENSATE}$$

1. SAMPLE PROBLEM 1.

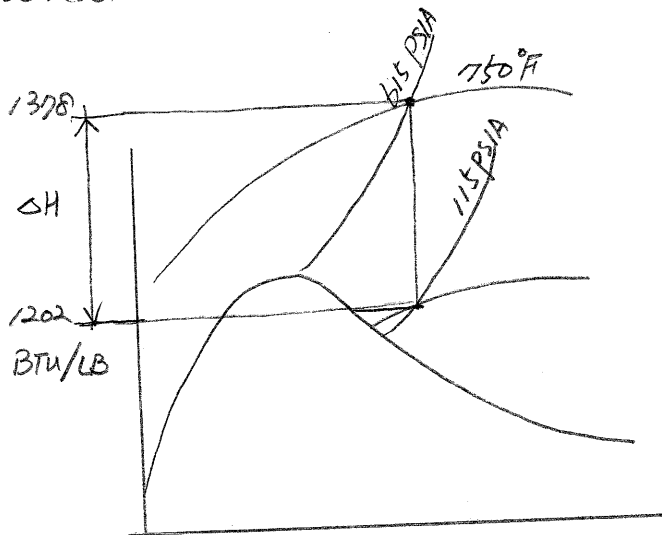
NON-CONDENSING TYPE TURBINE, 6000 RPM, RATED POWER = 4000 HP. (THIS steam turbine rated Horse-power is 110% of the compressor rated power requirement).

Intet steam = 600 PSIG, 750°F.

Exhausted steam = 100 PSIG.

1) Questions

What is the steam rate (Lbs/HR-HP)



$$\Delta H = 176 \text{ BTU/LB} \quad \left(\begin{array}{l} 1 \text{ HP} = 2545 \text{ BTU/HR} \\ 1 \text{ HP} = 0.746 \text{ KW} \end{array} \right)$$

$$\frac{176 \text{ BTU/LB}}{2545 \text{ BTU/HR}} \times \frac{\text{HP}}{1} = 0.069 \text{ (HP-HR)/LB}$$

$$\frac{1}{0.069} = 14.46 \text{ LB/HP-HR} \quad (\text{AT } 100\% \text{ EFFICIENCY})$$

$$\frac{14.46 \text{ LB}}{\text{HP-HR}} \times \frac{\text{HP}}{0.746 \text{ KW}} = 19.38 \text{ LB/KW-HR}$$

THE STEAM TURBINE EFFICIENCY

- ① BASIC EFFICIENCY (FROM FIG 1A) = 0.709.
- ② SUPERHEAT FACTOR (FROM FIG 2) = 1.003 (261°F S.H., ~~SAVED FACTOR 2.1~~)
- ③ PRESSURE RATIO FACTOR (FROM FIG 3)

$$\frac{\text{Exhaust } P \text{ (PSIA)}}{\text{inlet } P \text{ (PSIA)}} = \frac{115}{615} = 0.187$$

$$\therefore \text{PRESSURE RATIO FACTOR} = 1.013$$

- ④ SPEED FACTOR (FROM FIG 4) = 0.981

$$\text{ESTIMATED STEAM RATE} = \frac{14.46}{(0.709)(1.003)(1.013)(0.981)}$$

$$= 20.5 \text{ LB/HR-HP} ; \text{ANSWER}$$

? $\rightarrow 20.5 \text{ LB/HR-HP} \times 4000 \text{ HP} = 82,000 \text{ LB/HR}$

2) WHAT IS THE EXHAUST TEMPERATURE?

$$H_1 = 1378 \text{ BTU/LB}$$

$$H_1 - H_2 = 176 \text{ BTU/LB}$$

$$\text{IDEAL RANKINE CYCLE EFFICIENCY} = (H_1 - H_2)_{IS} / (H_1 - H_W)$$

$$H_1 - H_W = \frac{20.5 \text{ LB}}{\text{HR-HP}} \times \frac{\text{HP}}{2545 \text{ BTU/HR}} = 124.1 \text{ BTU/LB}$$

$$\therefore H_1 - H_W = 124.1 \text{ BTU/LB}$$

THEREFORE,

$$H_W = 1378 - 124 = 1254 \text{ BTU/LB}$$

FROM MOLLIER. $T_2 = 455^\circ \text{F}$. ANSWER.

• IDEAL RANKINE CYCLE EFFICIENCY

$$= (H_1 - H_2) / (H_1 - H_W) \times 100$$

$$= \frac{176.1}{124.1} = 142\%$$

• INTERNAL TURBINE EFFICIENCY

$$= \frac{(H_1 - H_W)}{\Delta H_{is}}$$

$$= \frac{124.1}{176} \times 100 = 70.5\%$$

3) What would the steam rate be if the exhaust pressure were 150 PSIG.

$$H_1 = 1378 \text{ BTU/LB}, \quad 6000 \text{ RPM}, \quad 4000 \text{ HP}, \quad 600 \text{ PSIG}, \quad 750^\circ \text{F}.$$

$$(H_1 - H_2)_{is} = (\text{FROM Keenan \& Keyes}) = 23.83 \text{ LB/HR-KW}$$

STEAM TURBINE EFF

$$\textcircled{1} \text{ BASIC EFF} = 0.709$$

$$\textcircled{2} \text{ SUPERHEAT FACTOR} = 1.003$$

$$\textcircled{3} \text{ PRESSURE RATIO FACTOR} = 1.013$$

$$\textcircled{4} \text{ SPEED FACTOR} = 0.981$$

$$\frac{4000 \text{ HP} \times \frac{2545 \text{ BTU}}{\text{HP-HR}}}{1500 \text{ BTU/lb}}$$

$$\frac{2545 \times 4000 \text{ BTU}}{\text{HR}} \div 1400$$

$$\frac{260 \text{ mthg}}{25.4} = 10.24$$

4

$$\rightarrow (\text{Lb/HP-HR})$$

$$\text{STEAM RATE} = \frac{23.83 \times 0.746}{(0.709)(1.003)(1.013)(0.981)}$$

$$= 25.2 \text{ Lb/HP-HR.}$$

$$\therefore \text{Required steam rate} = 25.2 \times 4000 \text{ HP}$$

$$= 100,623 \text{ lb/HR steam}$$

2. SAMPLE PROBLEM II.

TOTALLY CONDENSING TYPE, 10,000 HP, 7500 RPM.

INLET: 600 PSIG, 575°F.

EXHAUST: 3.5 in HgA. ($= 3.5 \times \frac{14.7}{29.92} = 1.72 \text{ PSIA}$)

1) WHAT IS THE PERCENT MOISTURE IN THE EXHAUST?

FROM THE MOLLIER:

$$H_1 = 1275 \text{ BTU/lb}$$

$$H_2 = ~~910~~ 877.5 \text{ BTU/lb.}$$

$$\therefore (H_1 - H_2)_{15} = 1275 - ~~910~~ 877.5 = 397.5 \text{ BTU/lb.}$$

$$\frac{397.5 \text{ BTU/lb}}{2545 \text{ BTU/HR}} \times \text{HP} = 6.4 \text{ lb/HR-HP}$$

① BASIC EFFICIENCY (FROM FIG 1) = 0.75

② SUPERHEAT FACTOR (FROM FIG 2) = 1.06

③ PRESSURE RATIO FACTOR (~~FROM FIG 3~~) = X

④ SPEED FACTOR (FROM FIG 4) = 0.9655

$$2. \text{ STEAM RATE} = \frac{6.4}{(0.75)(1.06)(0.9655)} = 8.3 \text{ lb/HP-HR}$$

$$\frac{8.3 \text{ lb}}{\text{HP-HR}} \bigg| \frac{\text{HP}}{2545 \text{ BTU/HR}} = 305.2 \text{ BTU/lb}$$

$$H_i - H_w = 305.2 \text{ BTU/lb}$$

$$H_w = 1275 - 305.2 = 970 \text{ BTU/lb}$$

FROM MOLLIER DIAGRAM. (SEE SATURATED STEAM TABLE)

$$\text{Temp} \approx 120$$

$$(970 \text{ BTU/lb}, 1.72 \text{ PSIA})$$

$$\underline{\text{Moisture PERCENT} = 14\%}$$

3) ESTIMATE THE CONDENSATE (GPM)

$$H_i - H_w = 305.2 \text{ BTU/lb}$$

$$10000 \text{ HP} \div 305.2 \text{ BTU/lb}$$

$$= \frac{10000 \text{ HP} \bigg| \frac{2545 \text{ BTU/HR}}{\text{HP}}}{305.2 \text{ BTU/LB}} = 83387.9 \text{ LB/HR (STEAM)}$$

$$\frac{83387.9 \text{ LB}}{\text{HR}} \bigg| \frac{1 \text{ gal}}{8.338 \text{ lb H}_2\text{O @ 60°F}} \bigg| \frac{\text{HR}}{60 \text{ min}} \approx 167 \text{ GPM}$$

3) WHAT IS THE CONDENSER DUTY.

$$(H_i - H_w) \times \text{STEAM RATE} = 305.2 \text{ BTU/lb} \times 83387.9 \text{ lb/HR}$$

$$= \underline{25.5 \times 10^6 \text{ BTU/HR}}$$

3. SAMPLE PROBLEM III.

FOR SINGLE STAGE ST,

$$6000 \text{ RPM, IN. STEAM} = 250 \text{ PSIG, } 500^\circ \text{F.}$$

$$\text{EXHAUST STM} = 60 \text{ PSIG.}$$

$$\text{PUMP RATED POWER} = 250 \text{ HP.}$$

$$\text{T/B. " = 110\% (1.1 \times 250 = 275 \text{ HP})}$$

$$\text{IN. SP. VOL} = 2.0 \text{ FT}^3/\text{LB.}$$

$$\text{EXHAUST SP. VOL} = 4.1 \text{ FT}^3/\text{LB.}$$

$$H_1 = 1264 \text{ BTU/LB}$$

$$H_2 = 1136 \text{ BTU/LB}$$

$$(H_1 - H_2)_{15} = 1264 - 1136 = 128 \text{ BTU/LB.}$$

$$\left\{ \begin{array}{l} \text{① BASIC EFF (FROM FIG 1)} = \\ \text{②} \end{array} \right\}$$

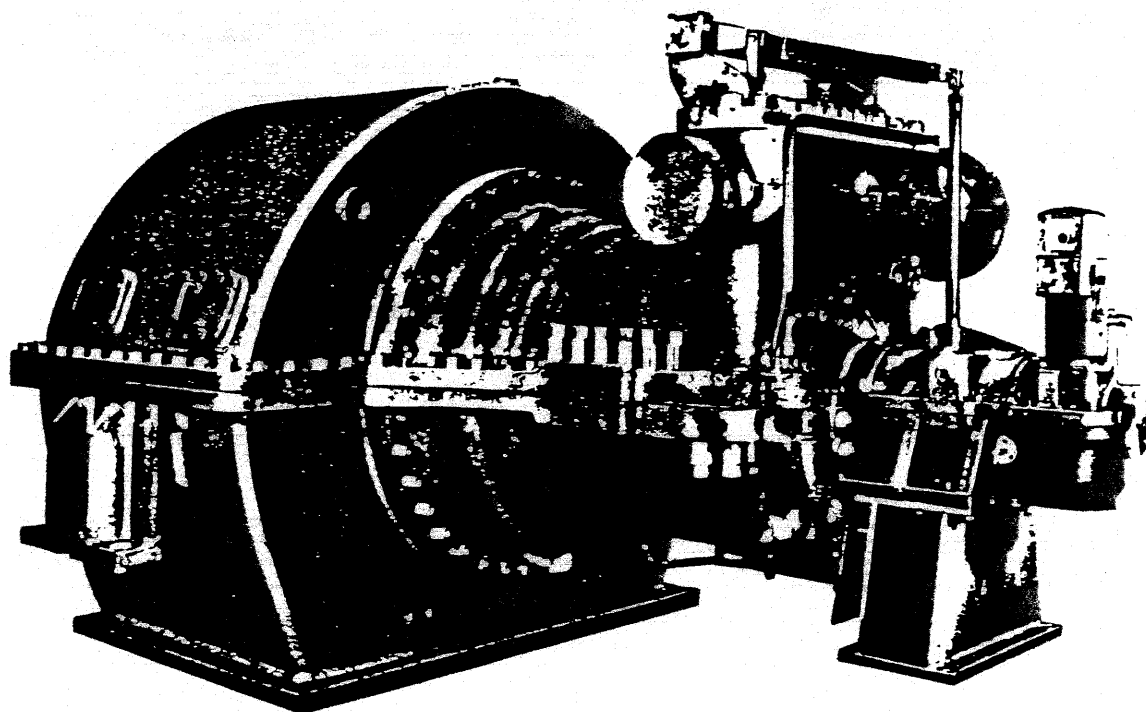
$$\text{FROM SINGLE STAGE EFF} = 35\%$$

TAKEN 35% EFFICIENCY.

$$\frac{128 \text{ BTU/LB} \mid \text{HP}}{2525 \text{ BTU/HR}} = 19.9 \text{ LB/HR-HP STEAM (AT 100\% EFF)}$$

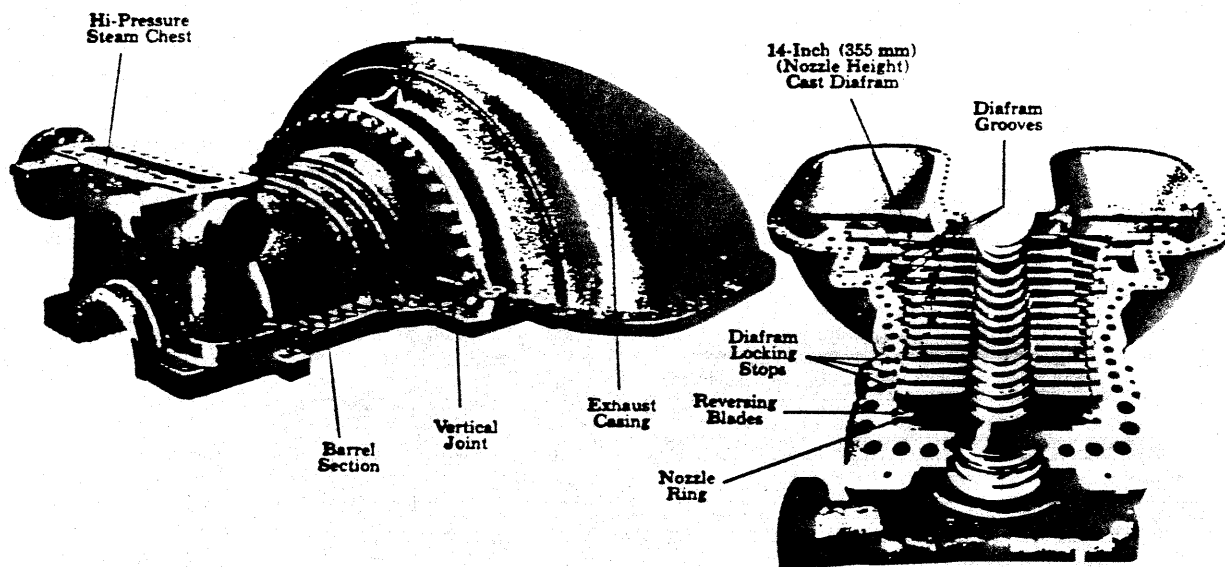
$$275 \times 19.9 \div 0.35 = \underline{15636 \text{ LB/HR STEAM}}$$

MULTIVALVE TURBINE CONSTRUCTION AND ARRANGEMENT

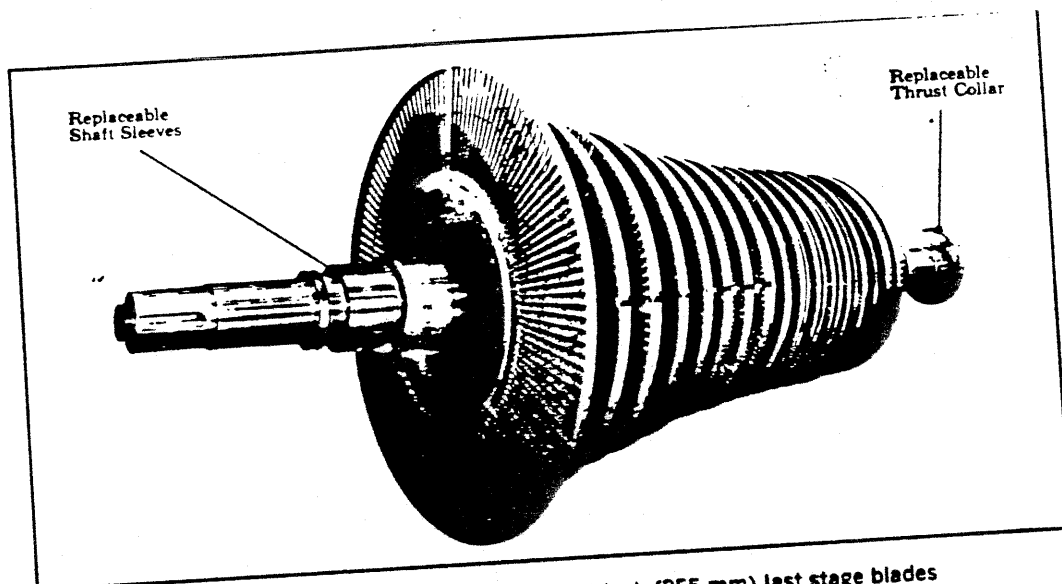


Typical Double-Flow Condensing Steam Turbine

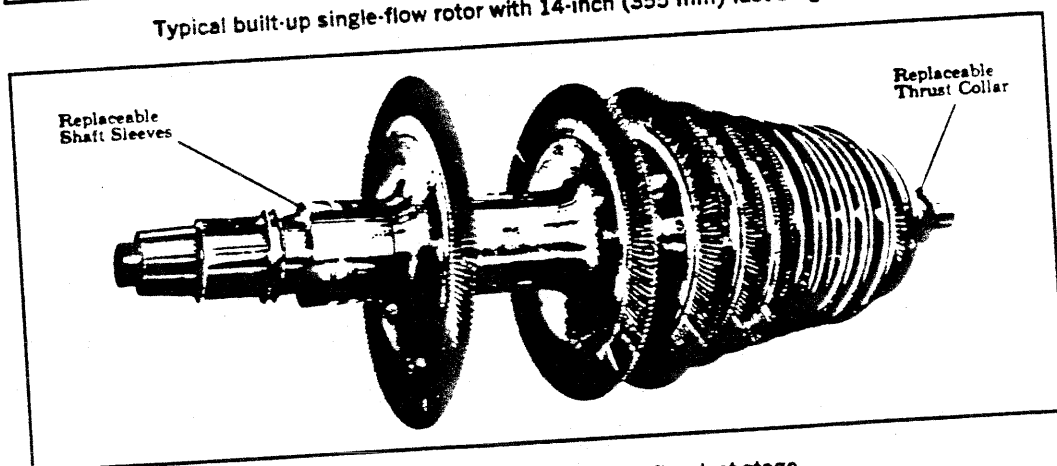
MULTIVALVE TURBINE CASING



MULTIVALVE TURBINE ROTOR ASSEMBLY

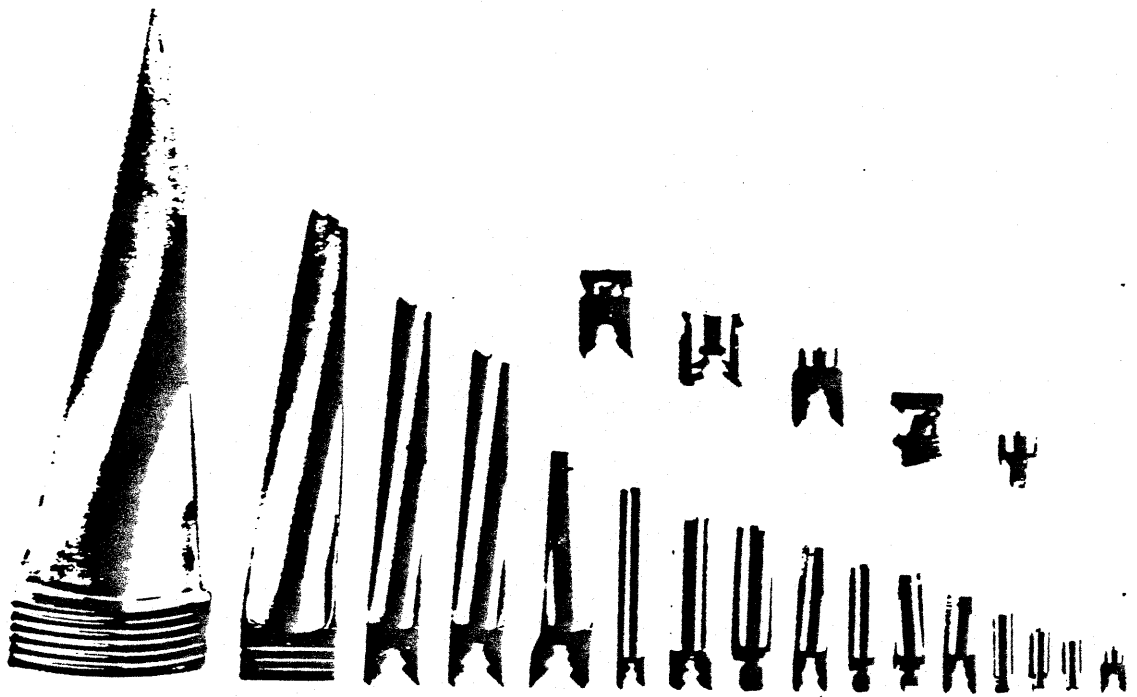


Typical built-up single-flow rotor with 14-inch (355 mm) last stage blades



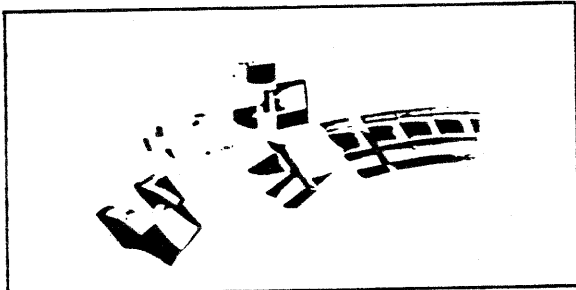
Typical integral rotor with double-flow last stage

TURBINE BLADES

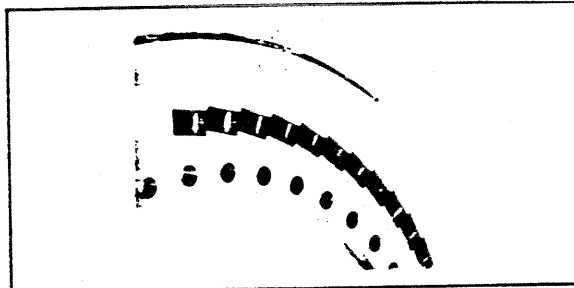


FORM 3446 -374
LITHO IN U.S.A.

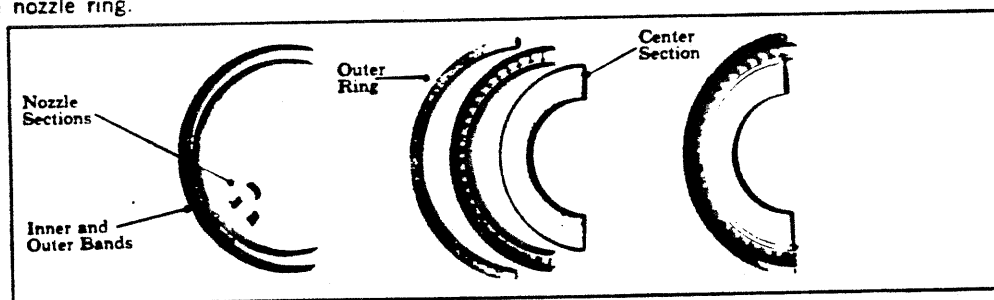
MULTIVALVE TURBINE NOZZLES AND DIAFRAMS



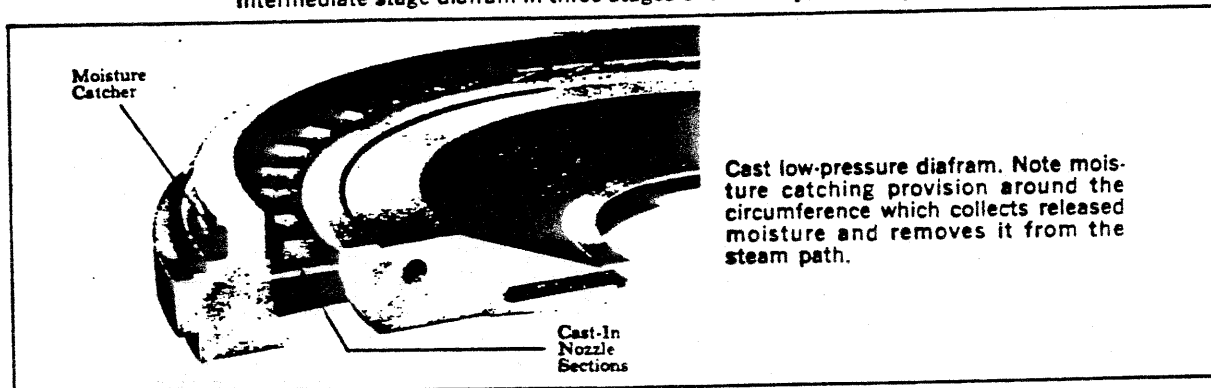
Milled nozzle blocks which form the first stage nozzle ring.



Section of completed first stage nozzle ring.

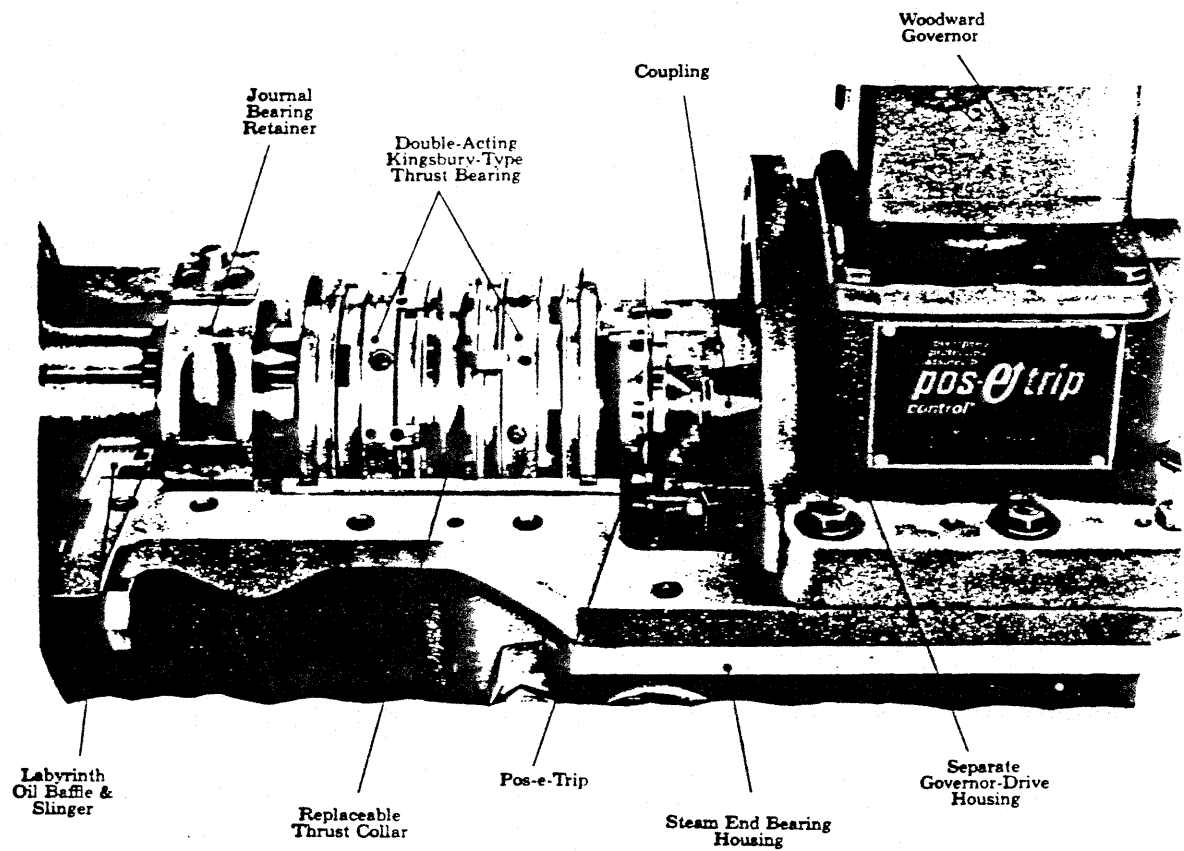


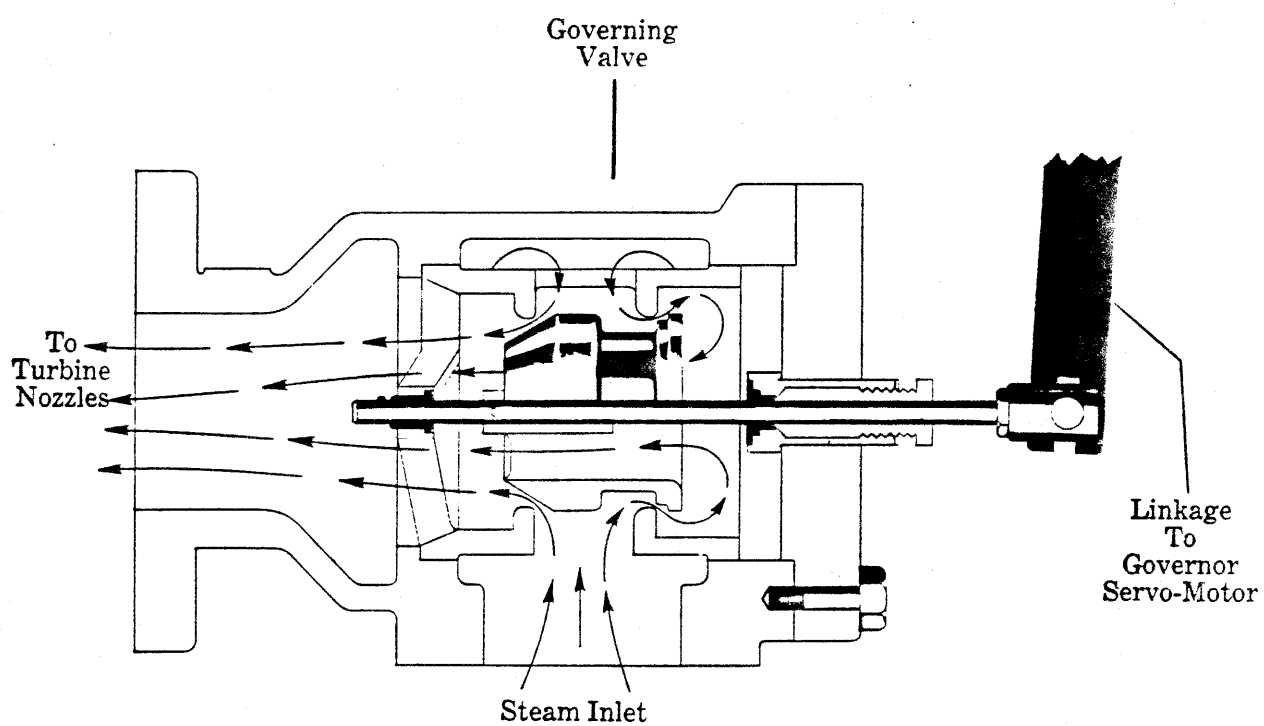
Intermediate stage diafram in three stages of assembly, left to right.

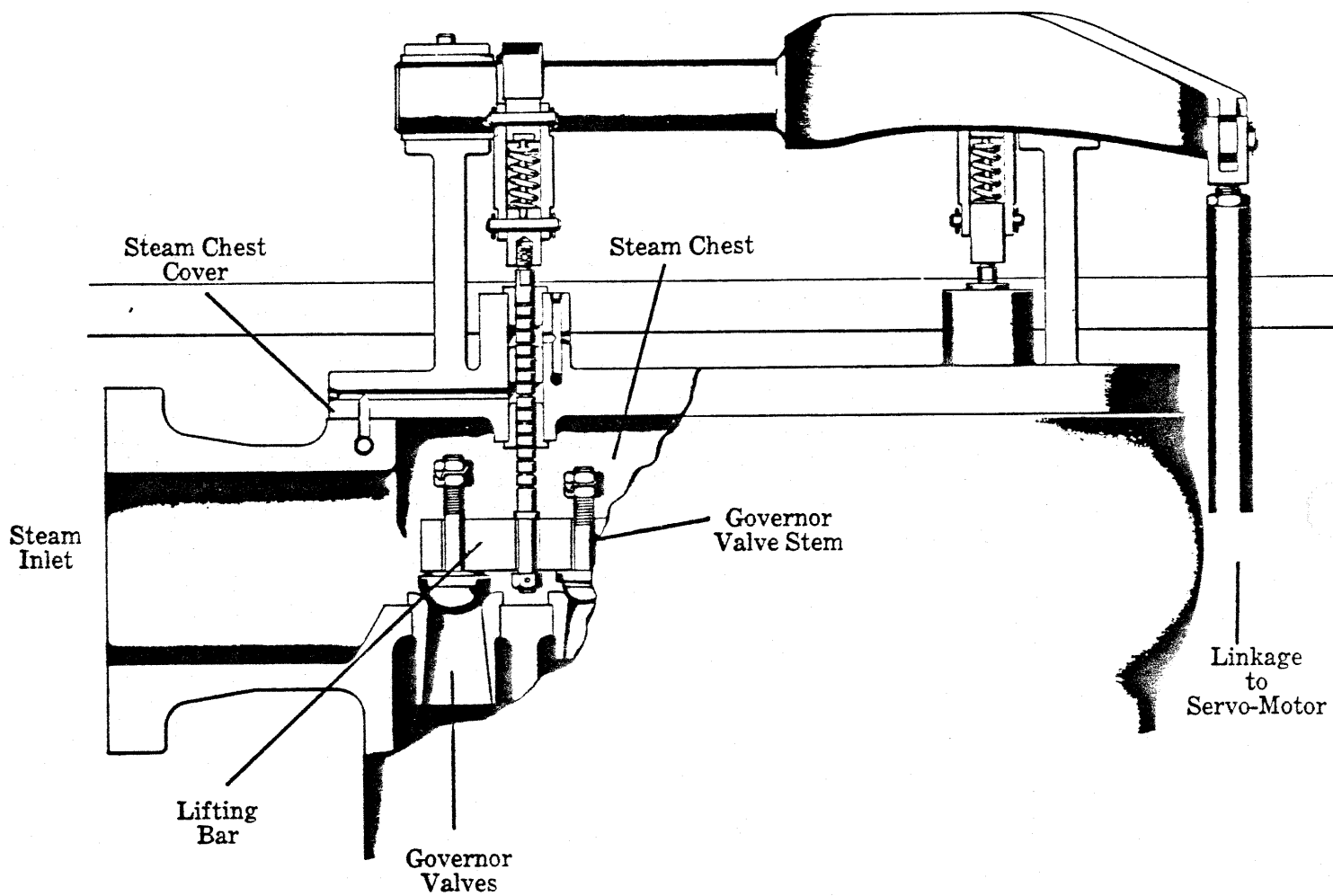


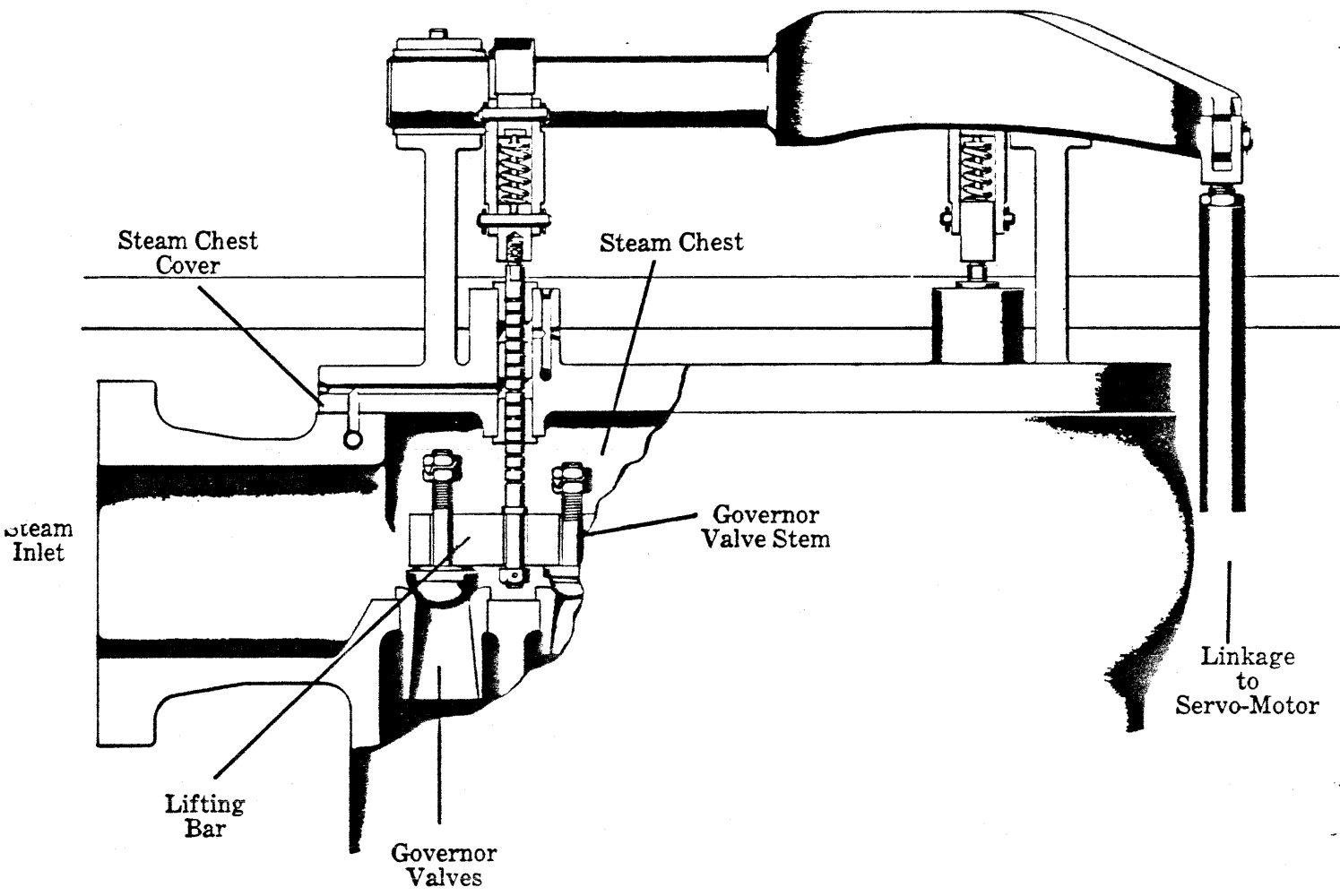
Cast low-pressure diafram. Note moisture catching provision around the circumference which collects released moisture and removes it from the steam path.

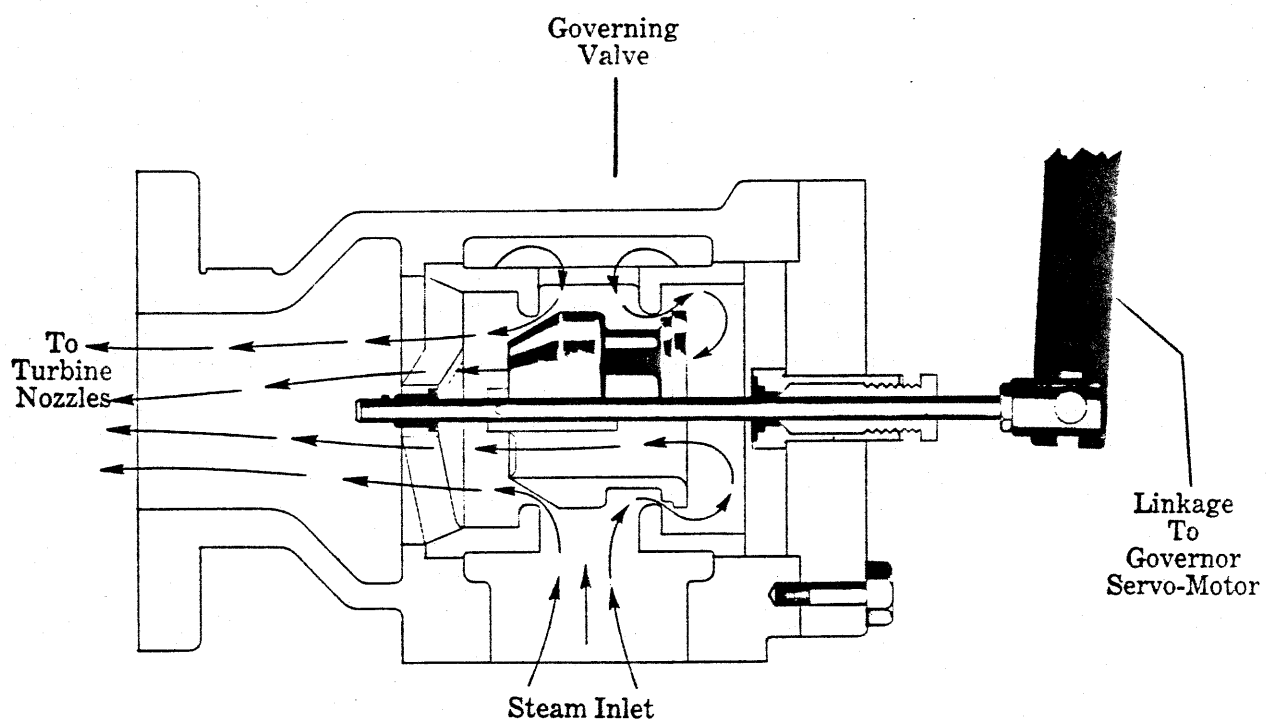
MULTIVALVE TURBINE BEARINGS











GOVERNING SYSTEM TERMINOLOGY

SPEED REGULATION

CHANGE IN SPEED WHEN THE POWER OUTPUT IS GRADUALLY CHANGED FROM RATED OUTPUT TO ZERO OUTPUT.

SPEED VARIATION

FLUCTUATIONS FROM THE SET POINT INCLUDING DEAD-BAND AND SUSTAINED OSCILLATION.

SPEED RISE

SPEED INCREASE WHEN THE LOAD IS DROPPED INSTANTANEOUSLY

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

SAMPLE PROBLEMS

DETERMINING THE STEAM RATE.....

- 1) DEFINE STEAM CONDITIONS AT INLET AND OUTLET
- 2) DEFINE ΔH_{IS} USING MOLLIER OR KEENAN AND KEYES
- 3) FIND TURBINE EFFICIENCY USING VENDOR'S DATA

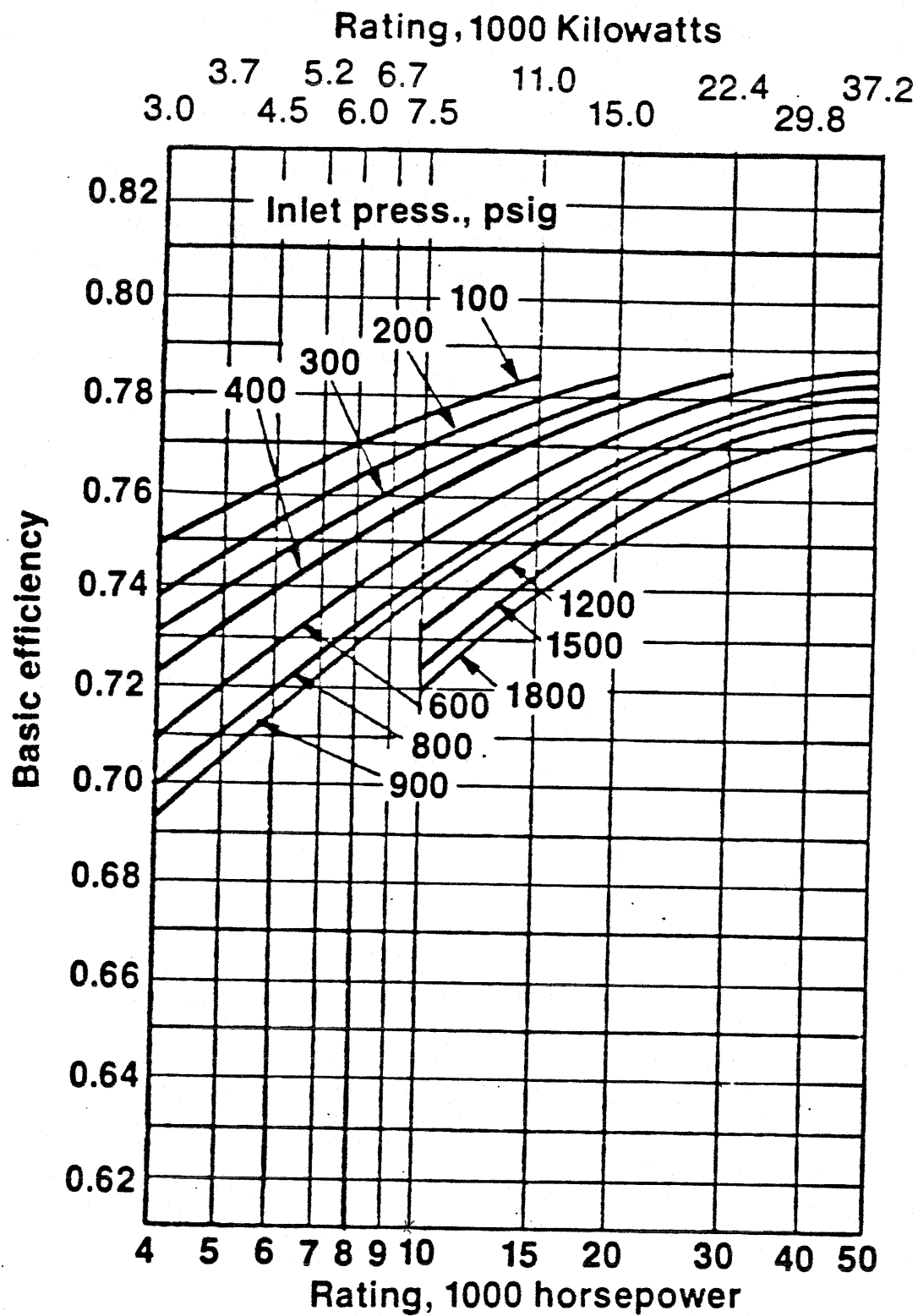


Fig. 1. Basic efficiency of multi-valve, multi-stage condensing turbines.

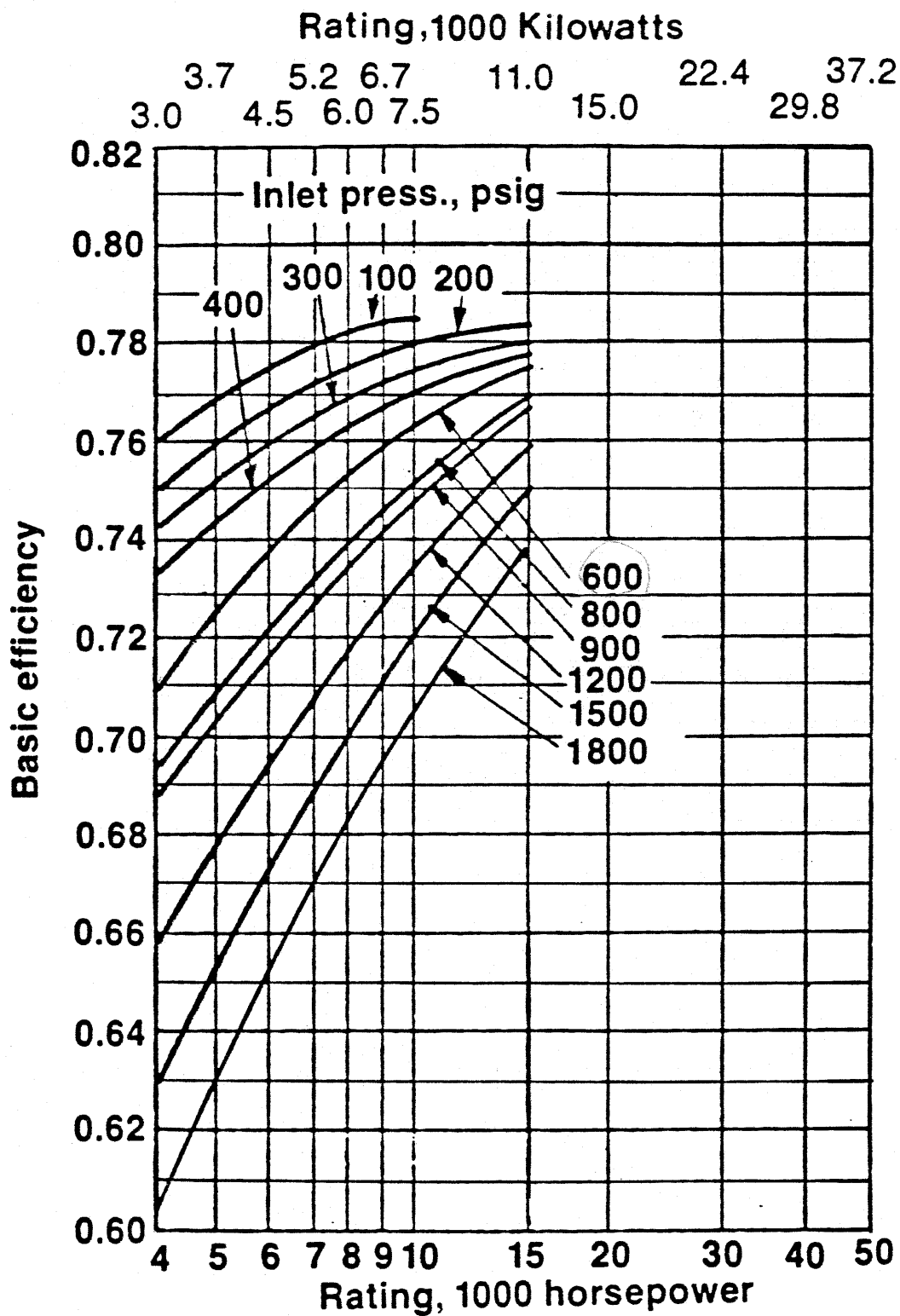


Fig. 1A. Basic efficiency of multi-valve, multi-stage non-condensing turbines.

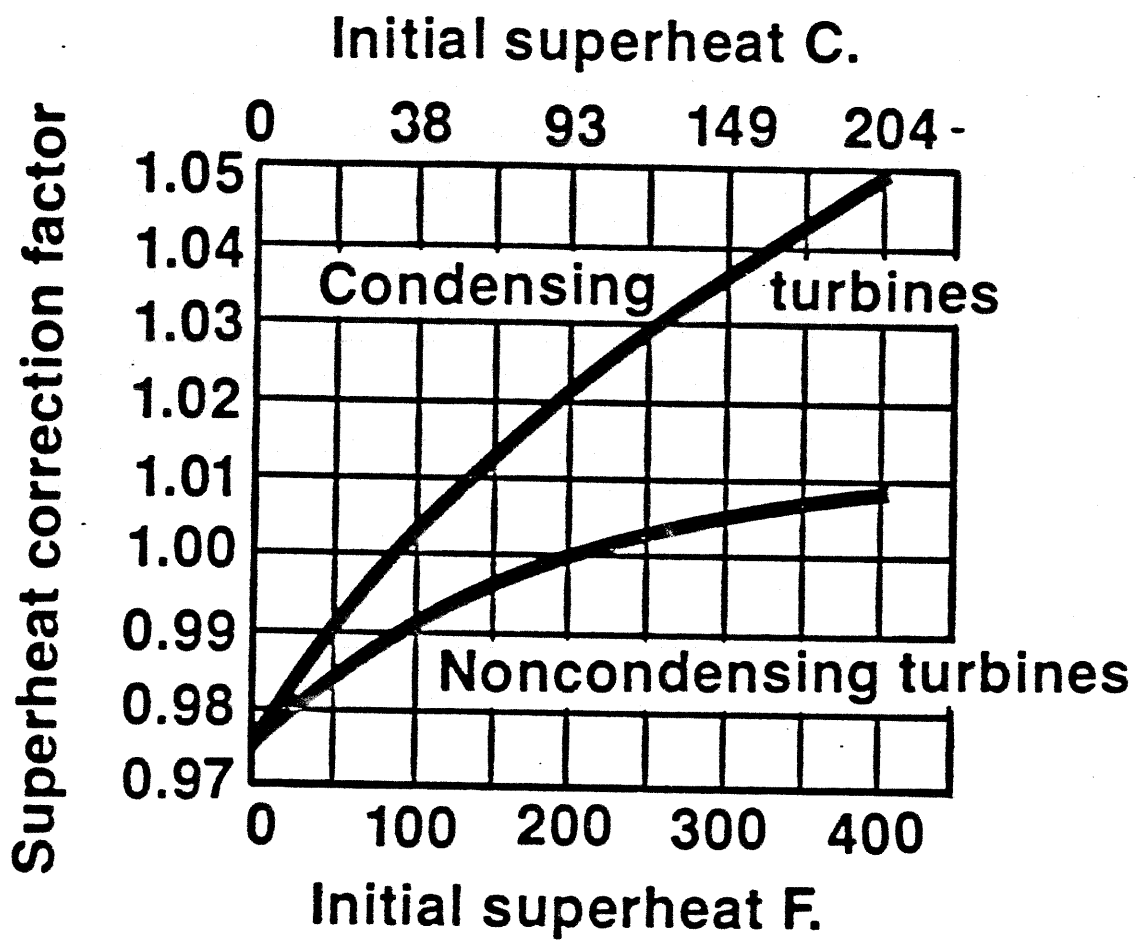


Fig. 2. Superheat correction factor.

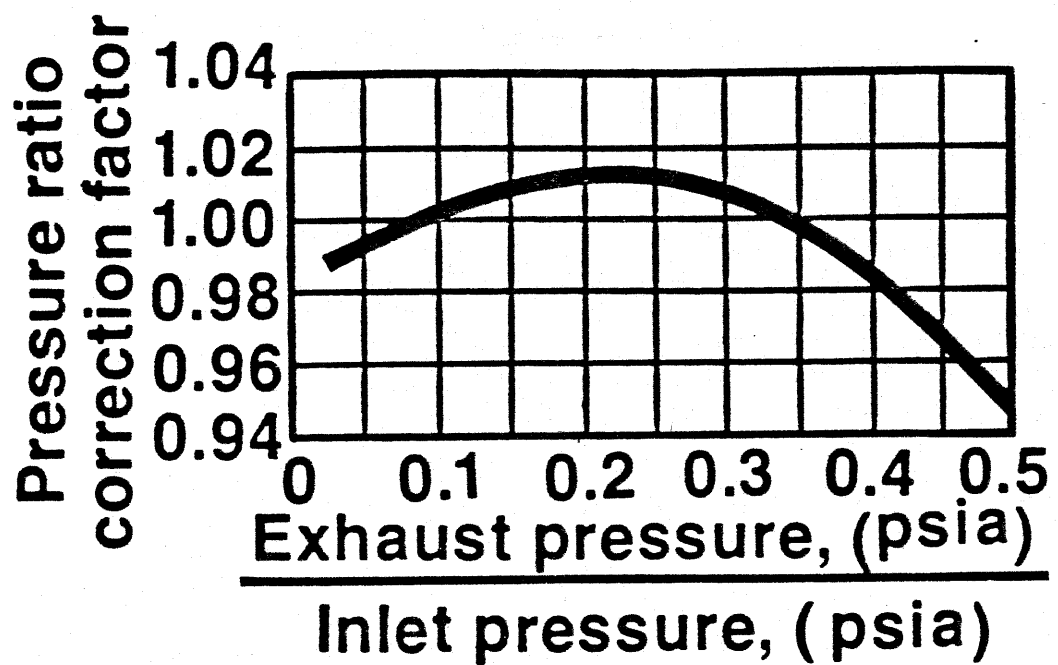


Fig. 3. Pressure ratio correction non-condensing turbines.

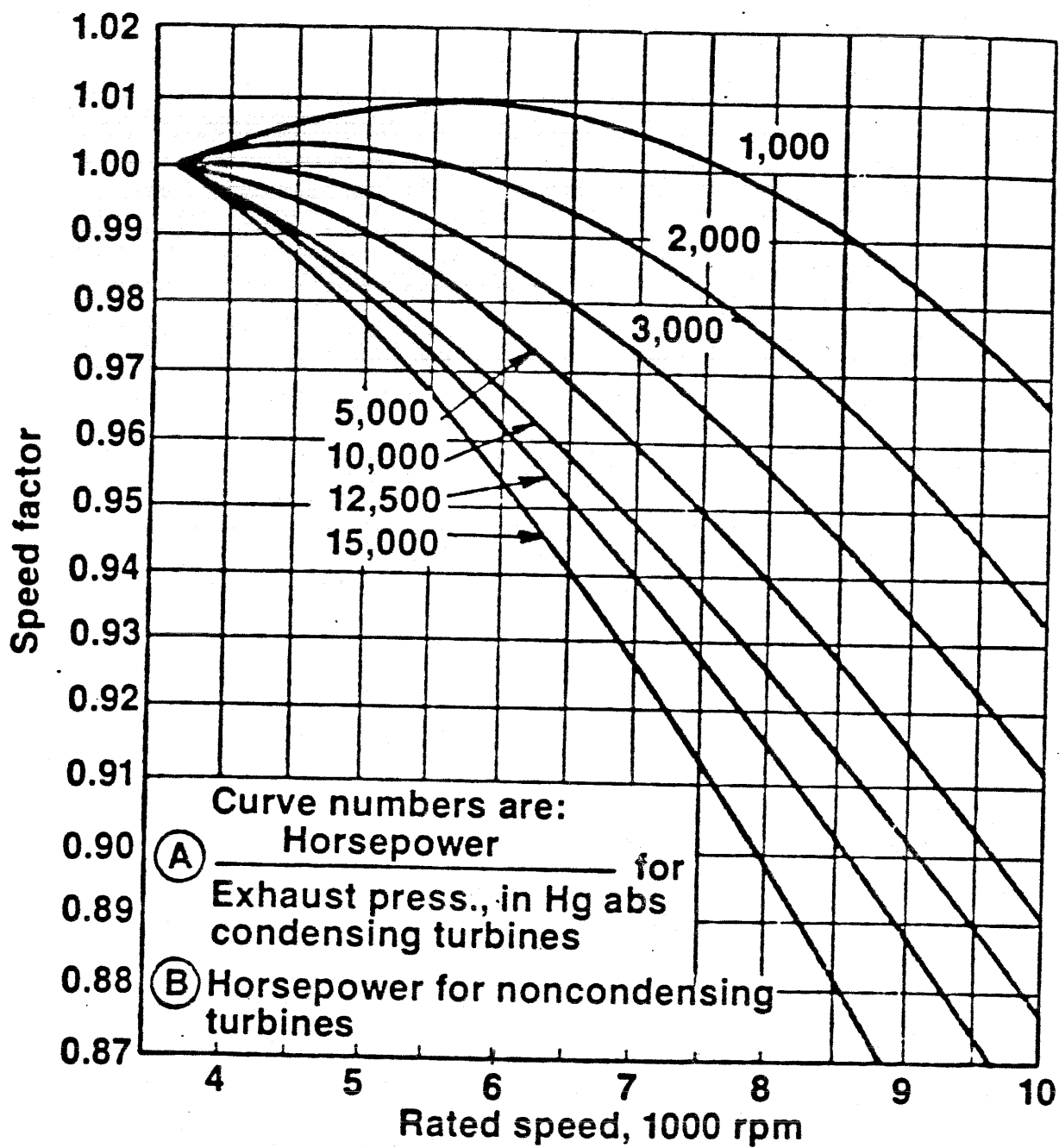


Fig. 4. Speed factor.

Plot II

Problem I

16

Exhaust Pressure	125 #G 580' FTT 147' FS	150 #G 580' FTT 134' FS	200 #G 348' FTT 8' FS	200 #G 500' FTT 112' FS	250 #G 406' FTT 8' FS	250 #G 500' FTT 94' FS	400 #G 448' FTT 8' FS	400 #G 750' FTT 302' FS	600 #G 575' FTT 84' FS	600 #G 750' FTT 281' FS	600 #G 825' FTT 334' FS	850 #G 750' FTT 223' FS	850 #G 825' FTT 298' FS	850 #G 900' FTT 373' FS	1250 #G 825' FTT 251' FS	1250 #G 900' FTT 326' FS	1500 #G 825' FTT 228' FS	1500 #G 900' FTT 302' FS	1800 #G 900' FTT 278' FS
3 in HgA	10.57	10.28	10.62	9.87	10.22	9.59	9.48	7.70	8.34	7.40	7.05	7.20	6.85	6.53	6.69	6.36	6.64	6.30	5.86
4 in HgA	11.10	10.78	11.12	10.31	10.66	10.00	9.85	7.99	8.64	7.65	7.28	7.42	7.06	6.73	6.88	6.54	6.83	6.48	6.01
5 in HgA	11.57	11.21	11.54	10.70	11.05	10.35	10.17	8.22	8.89	7.86	7.48	7.61	7.24	6.89	7.04	6.69	6.98	6.62	6.14
6 #G	18.55	17.50	17.51	16.08	16.30	15.16	14.26	11.19	11.96	10.40	9.82	9.87	9.32	8.81	8.90	8.40	8.76	8.25	7.54
10 #G	23.18	21.48	21.10	19.28	19.31	17.90	16.43	12.71	13.49	11.64	10.96	10.94	10.30	9.71	9.75	9.18	9.56	8.95	8.16
20 #G	27.96	25.47	24.56	22.34	22.12	20.44	18.34	14.02	14.79	12.68	11.90	11.82	11.10	10.43	10.43	9.80	10.20	9.56	8.64
30 #G	33.3	29.73	28.11	25.46	24.91	22.85	20.15	15.23	15.98	13.62	12.76	12.60	11.80	11.07	11.03	10.34	10.75	10.07	9.06
50 #G	47.2	39.9	35.99	32.3	30.8	28.21	23.67	17.56	18.19	15.36	14.31	14.01	13.07	12.21	12.07	11.28	11.72	10.94	9.77
60 #G	57.2	46.4	40.55	36.2	34.8	31.07	25.45	18.74	19.26	16.18	15.06	14.66	13.66	12.74	12.55	11.71	12.16	11.34	10.08
75 #G	79.1	59.1	48.5	42.9	39.4	35.8	28.21	20.57	20.85	17.40	16.17	15.61	14.51	13.51	13.22	12.32	12.77	12.85	10.53
100 #G	—	96.6	66.6	58.1	50.1	45.21	33.14	23.86	23.53	19.43	18.07	17.14	15.88	14.77	14.28	13.26	13.74	12.75	11.21
150 #G	—	—	—	—	—	—	45.21	31.93	29.24	23.83	22.15	20.15	18.62	17.33	16.28	15.05	15.52	14.33	12.44
250 #G	—	—	—	—	—	—	—	—	43.97	35.40	32.91	26.84	24.80	23.08	20.22	18.67	18.89	17.37	14.78
400 #G	—	—	—	—	—	—	—	—	—	—	—	—	—	—	27.05	24.99	24.30	22.32	18.39

Keenan & Keyes

16/HR KW

SAMPLE PROBLEM I

A NON-CONDENSING STEAM TURBINE IS DRIVING A 6000 RPM CENTRIFUGAL COMPRESSOR REQUIRING A RATED POWER OF 4000 HP. THE STEAM TURBINE RATED HORSEPOWER IS 110% OF THE COMPRESSOR RATED POWER REQUIREMENT.

INLET STEAM IS 600 PSIG, 750°F AND EXHAUST STEAM IS AT 100 PSIG.

QUESTIONS:

- $H_1 - H_w$
115
LBS/HR-HP
- 1) WHAT IS THE STEAM RATE (LBS/HR/HP)?
 - 2) WHAT IS THE EXHAUST TEMPERATURE?
 - 3) WHAT WOULD THE STEAM RATE BE IF THE EXHAUST PRESSURE WERE 150 PSIG?

PROBLEM I SOLUTION

1) CALCULATE THE STEAM RATE.

THEORETICAL STEAM RATE: TABLE 1: 19.43 LB/KW-HR

$$\begin{aligned}\text{MOLLIER: } H_1 &= 1378 \text{ BTU/LB} \\ H_2 &= 1202 \text{ BTU/LB} \\ \Delta H &= 176 \text{ BTU/LB}\end{aligned}$$

$$33000 \text{ FT-LBS/MIN} = 1 \text{ HP}$$

$$778 \text{ FT-LBS} = 1 \text{ BTU}$$

$$33000/778 = 42.4 \text{ BTU/MIN} = 1 \text{ HP}$$

$$42.4 \times 60 = 2545 \text{ BTU/HR} = 1 \text{ HP}$$

$$1 \text{ HP} = 0.746 \text{ KILOWATTS}$$

$$2545/0.746 = 3412 \text{ BTU/HR} = 1 \text{ KW}$$

$$\frac{3412 \text{ BTU/KW-HR}}{176 \text{ BTU/LB}} = 19.4 \text{ LB/KW-HR}$$

$$\text{BASIC EFFICIENCY (FIGURE 1A)} = 0.709$$

$$\text{SUPERHEAT FACTOR (FIGURE 2)} = 1.003 \text{ (261°F S. H., 0.187 FACTOR)}$$

$$\text{PRESSURE RATIO FACTOR (FIGURE 3)} = 1.013$$

$$\text{SPEED FACTOR (FIGURE 4)} = 0.981$$

$$\text{LB/(KW-HR)} : \Delta H$$

$$\text{ESTIMATED STEAM RATE} = \frac{(19.43) \times (0.746)}{(.709)(1.003)(1.013)(.981)} = 20.5 \text{ LB/HP-HR}$$

ACTUAL STEAM REQUIRED

$$\text{LB/(HP-HR)} : \Delta H$$

$$20.5 \times 4000 = 82000$$

$$\frac{16}{100}$$

SAMPLE PROBLEM I SHEET 2

2) ESTIMATE EXHAUST TEMPERATURE

$$H_1 = 1378 \text{ BTU/LB}$$

$$\Delta H = \frac{2545 \text{ BTU/HR-HP}}{20.5 \text{ LB/HP=HR}} = 124.1 \text{ BTU/LB}$$

$$H_2 = 1378 - 124 = 1254 \text{ BTU/LB} \quad P_2 = 100 \text{ PSIG}$$

FROM MOLLIER, $T_2 = 455^\circ\text{F}$

AT 100 PSIG, $T_{\text{SAT}} = 338^\circ\text{F}$

117° F SUPERHEAT

WASTON

3) STEAM RATE FOR 150 PSIG EXHAUST

THEORETICAL STEAM RATE = 23.83 LB/KW-HR

BASIC EFFICIENCY = 0.709

SUPERHEAR FACTOR = 1.003

PRESSURE RATIO FACTOR = 1.013 (0.268)

SPEED FACTOR = 0.981

$$\text{ESTIMATED STEAM RATE} = \frac{(23.83)(.746)}{(.709)(1.003)(1.013)(.981)} = 25.2 \text{ LB/HP-HR}$$

SAMPLE PROBLEM II

CALCULATE THE ESTIMATED STEAM RATE FOR A TOTALLY CONDENSING STEAM TURBINE RATED FOR 10000 HP AND 7500 RPM AND DRIVING A CENTRIFUGAL COMPRESSOR.

THE INLET STEAM IS 600 PSIG, 575°F AND THE EXHAUST STEAM IS AT 3.5 INCHES HG ABS AND IS ROUTED TO A WATER COOLED CONDENSER.

QUESTIONS:

- 1) WHAT IS THE PERCENT MOISTURE IN THE EXHAUST?
- 2) ESTIMATE THE CONDENSATE GPM.
- 3) WHAT IS THE CONDENSER DUTY?

SAMPLE PROBLEM III

FOR A SINGLE STAGE STEAM TURBINE
DIRECTLY DRIVING A PUMP AT 6000 RPM,
DETERMINE THE RATED STEAM
CONSUMPTION AND INLET AND
EXHAUST VELOCITIES GIVEN THE
FOLLOWING:

TURBINE MODEL - AYR

INLET STEAM - 250 PSIG, 500°F
EXHAUST STEAM - 60 PSIG

PUMP RATED POWER = 250 HP

TURBINE RATED POWER = 110% OF PUMP HP

INLET SP. VOL = 2.0 FT³/LB
EXHAUST SP. VOL = 4.1 FT³/LB

FAN LAWS

INTRODUCTION

The affinity laws or fan laws express the mathematical relationship between the several variables involved in pump performance. They apply to all types of centrifugal and axial flow pumps. They are as follows:

1. With impeller diameter, D, held constant:

$$A. \quad \frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

Where Q = Capacity, GPM
 H = Total Head, Feet
 BHP = Brake Horsepower
 N = Pump Speed, RPM

$$B. \quad \frac{H_1}{H_2} = \left(\frac{N_1}{N_2} \right)^2$$

$$C. \quad \frac{BHP_1}{BHP_2} = \left(\frac{N_1}{N_2} \right)^3$$

2. With speed, N, held constant:

$$A. \quad \frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

$$B. \quad \frac{H_1}{H_2} = \left(\frac{D_1}{D_2} \right)^2$$

$$C. \quad \frac{BHP_1}{BHP_2} = \left(\frac{D_1}{D_2} \right)^3$$

When the performance (Q_1 , H_1 & BHP_1) is known at some particular speed (N_1) or diameter (D_1), the formulas can be used to estimate the performance (Q_2 , H_2 & BHP_2) at some other speed (N_2) or diameter (D_2). The efficiency remains nearly constant for speed changes and for small changes in impeller diameter.

HORSEPOWER

$$\text{BHP} = \frac{\text{GPM} \times \text{HEAD} \times \text{SP.} \times \text{GR.}}{3960 \times \text{EFF.}} = \frac{\text{LB/MIN} \times \text{HEAD}}{33,000 \times \text{EFF.}} = \frac{\text{GPM} \times \text{PSI}}{1714 \times \text{EFF.}}$$

60 - 50 CYCLE

$$Q_{60} = 1.2 Q_{50}$$

$$H_{60} = 1.44 H_{50}$$

$$\text{NPSH}_{60} = \frac{\text{NPSH}_{50}}{0.76}$$

VII. PUMP CONTROL

INTRODUCTION

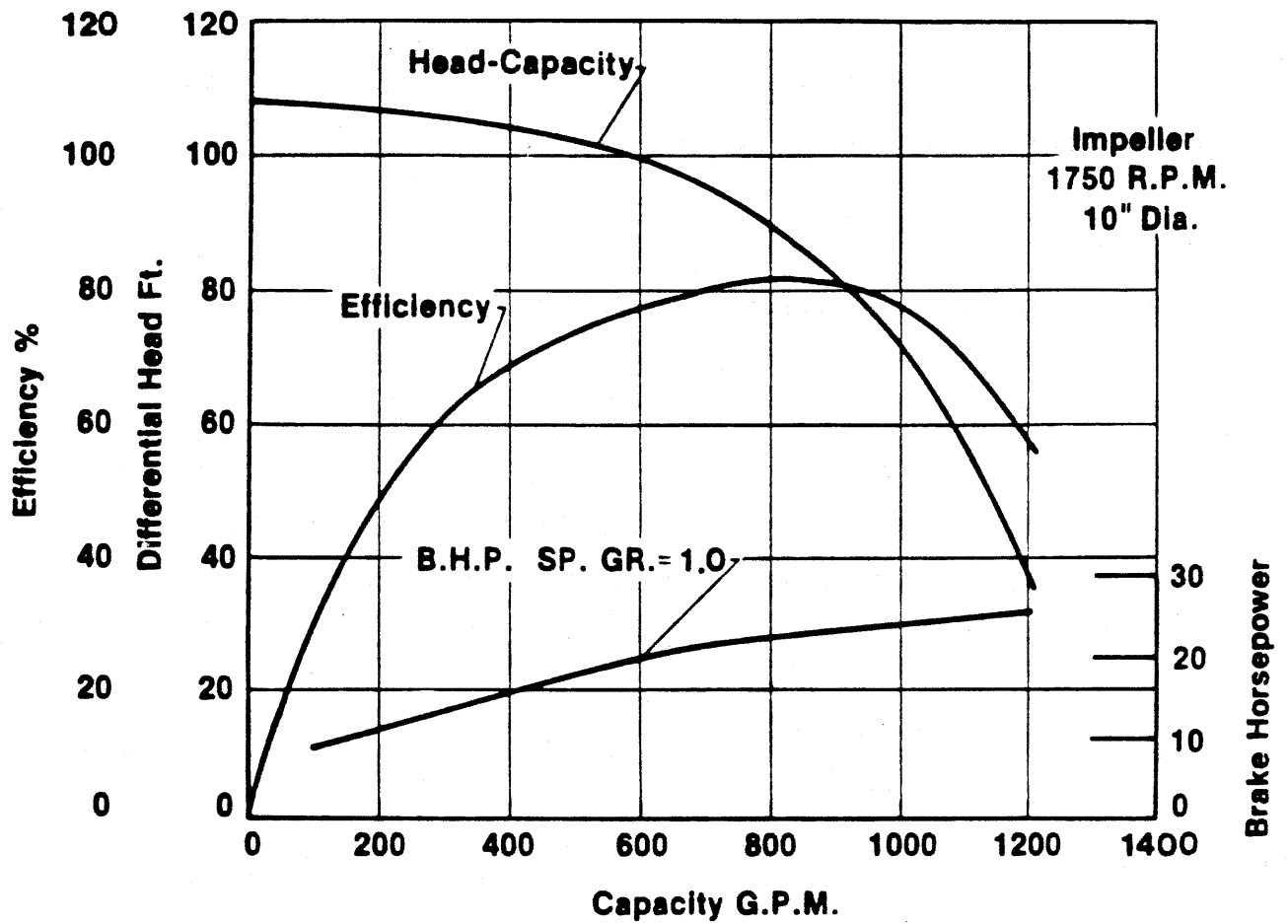
With a single-speed pump, should a turndown case require less head than the head capacity curve, the excess must be taken over the control valve.

With a variable speed driver (variable speed motor or turbine), the pump speed can be reduced to create a lower head capacity curve to meet the required head at turndown. This uses less power, as nothing is wasted across the control valve.

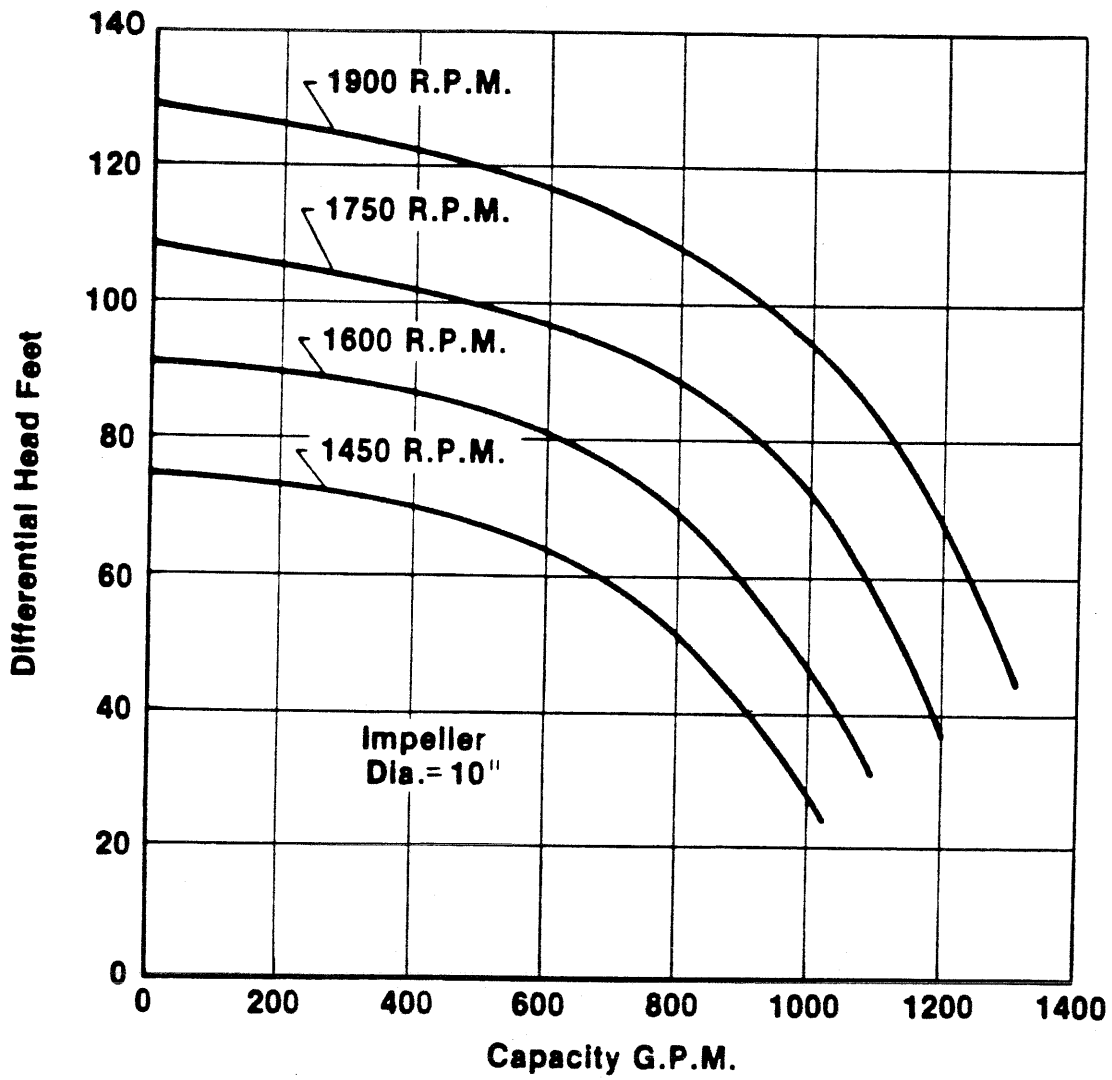
When pumps are operated in parallel, the combined capacities of both pumps for any head are equal to the sum of the individual capacities. For pumps operating in series, the combined heads for any flow are equal to the sum of the individual heads.

The typical field arrangement is one operating motor-driven pump and one spare motor-driven pump. For some critical services, a spare turbine-driven pump may be provided. Auto starts for both motor and turbine are typically specified.

CHARACTERISTIC CURVES AT CONSTANT SPEED

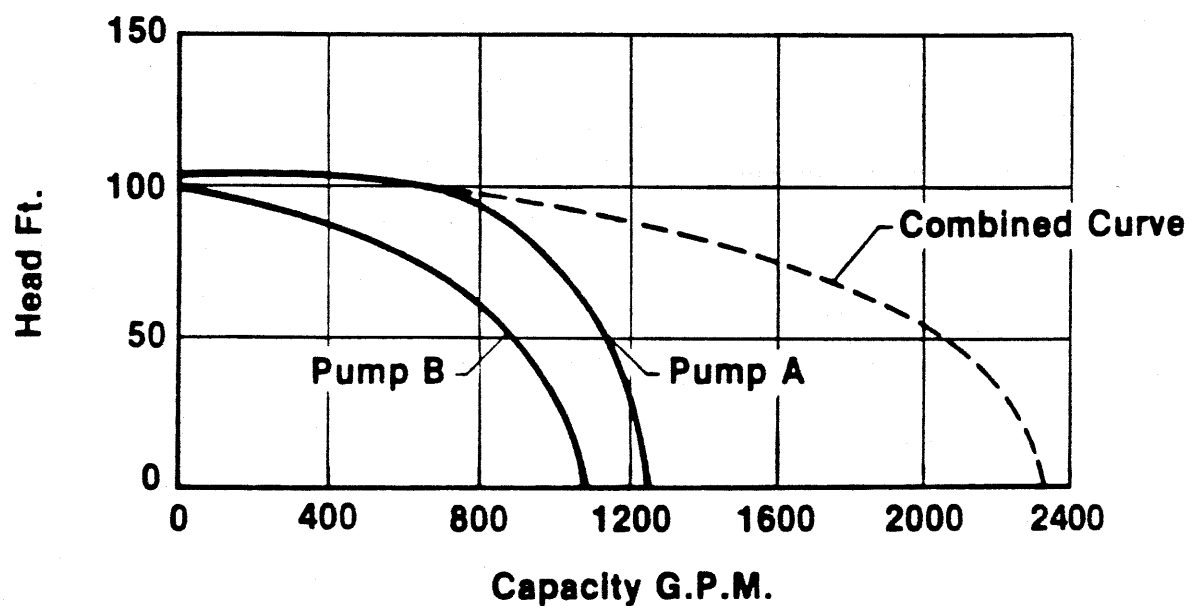


SPEED VARIATION

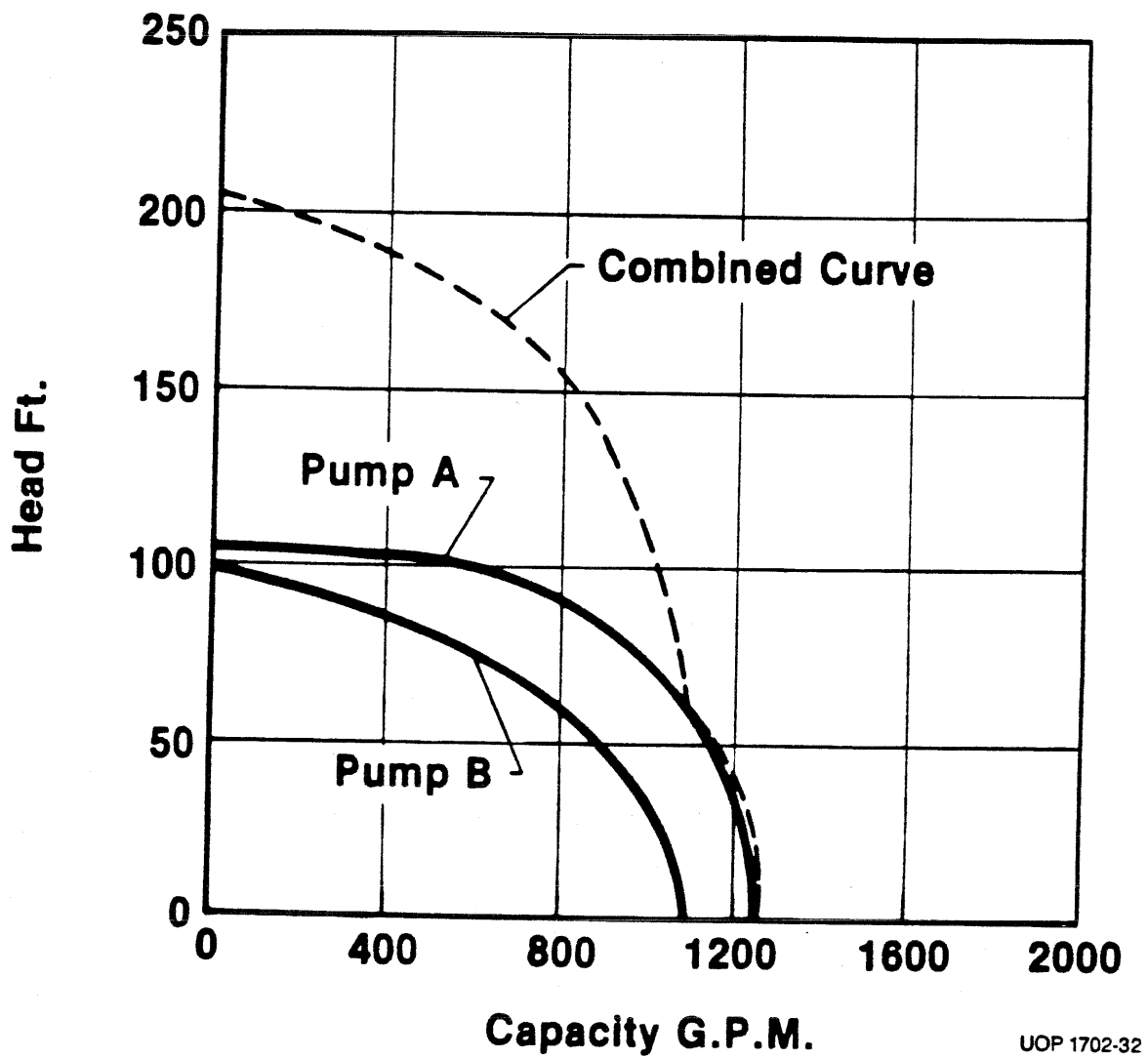


UOP 1702-31

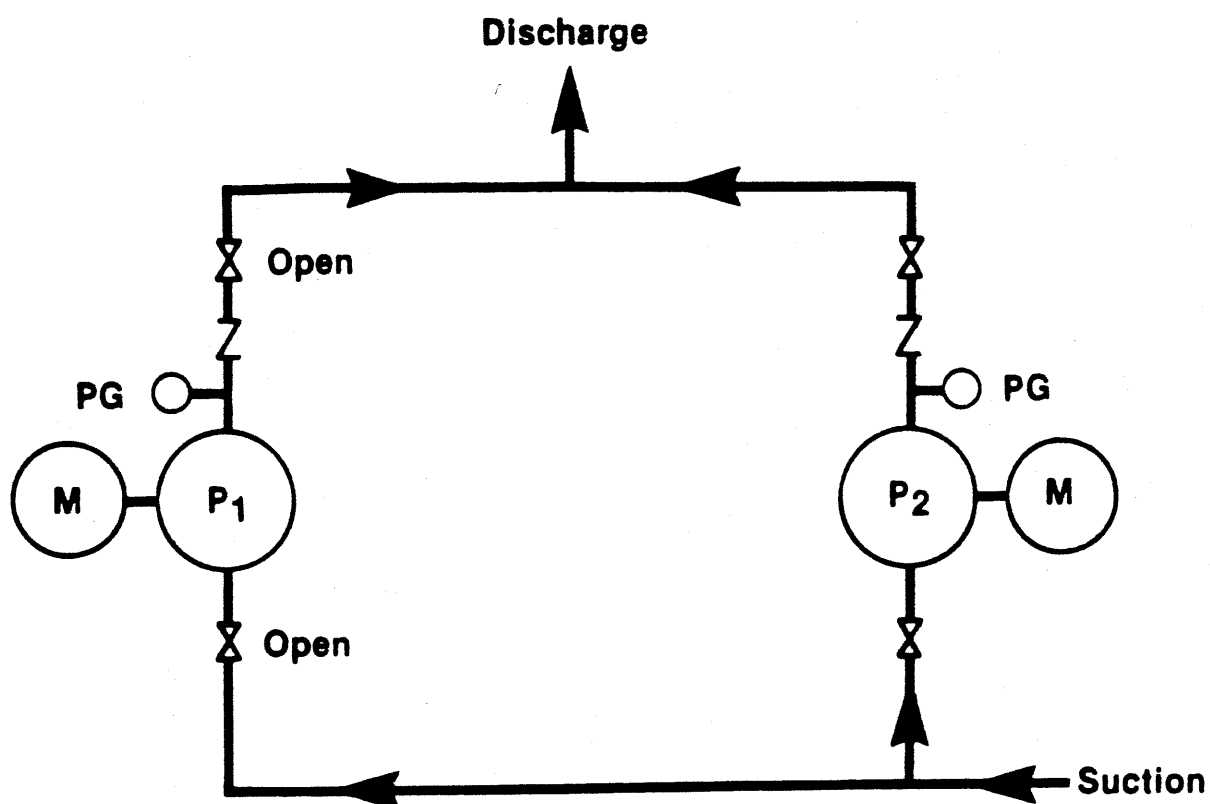
TWO CENTRIFUGAL PUMPS IN PARALLEL



TWO CENTRIFUGAL PUMPS IN SERIES

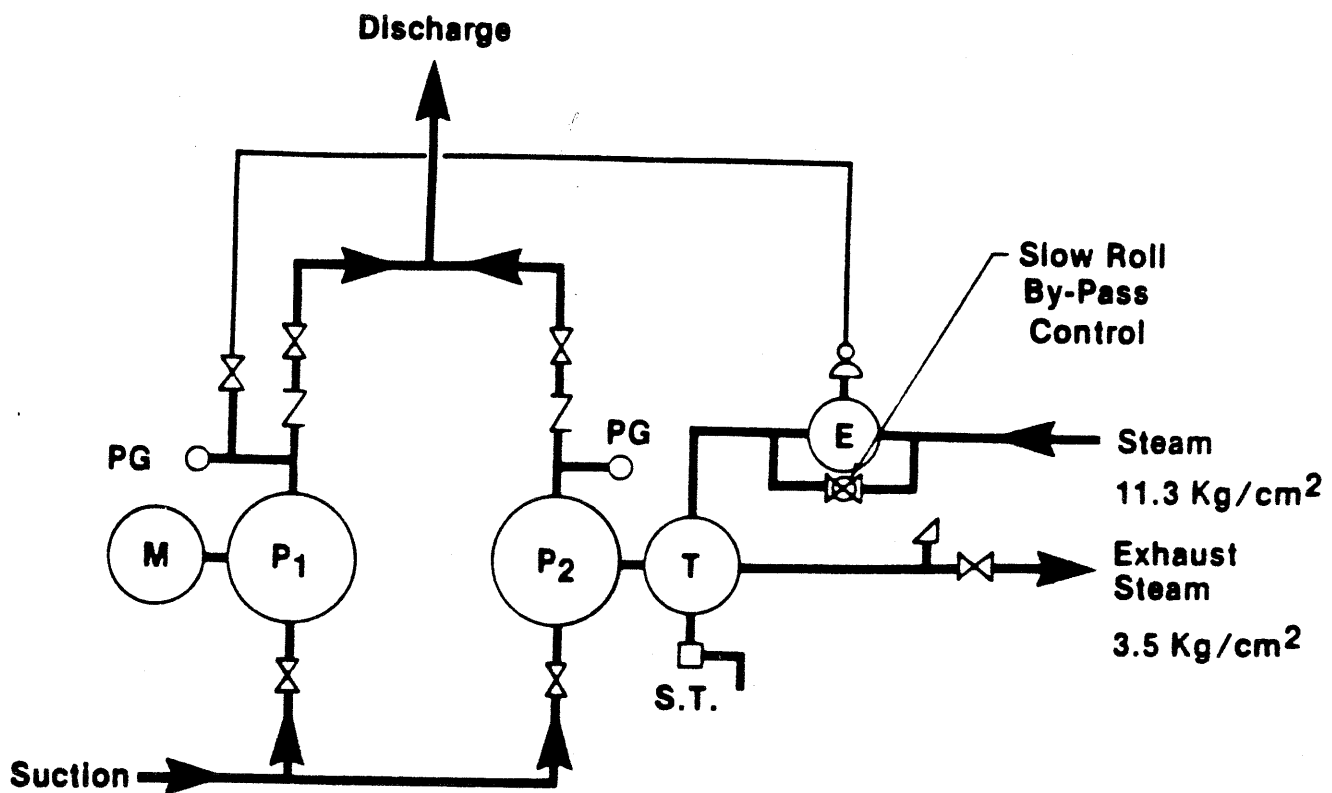


TYPICAL MOTOR/MOTOR SPARE PUMP ARRANGEMENT



UOP 1702-35

TYPICAL MOTOR -AUTO CUT -IN TURBINE SPARE PUMP ARRANGEMENT



UOP 1702-36

VIII. STANDARDS

INTRODUCTION

Most refinery pumps are designed to API (American Petroleum Institute) Standard 610. Lately, many refineries are accepting the less expensive ANSI (American National Standards Institute) designed pumps for non-hydrocarbon services. 5-11-2 is UOP's supplemental standard for API pumps and 5-21-0 is UOP's supplemental standard for ANSI pumps.

ANSI VS. API

	<u>ANSI</u>	<u>API</u>
Pump Type	End Suction Single-Stage Only	End or Top Suction, Single or Multi-Stage
Dimensions	Standardized	Non-standardized
Impeller	Open or Closed	Closed
Std. Materials	Cast Iron or 316SS	Steel or 11-13% Cr
Stuffing Box	Packed or Mechanical Seal	Mechanical Seal
Support	Foot	Centerline
Minimum Bearing Life	17,500 HR	25,000 HR
Wearing Rings	Optional	Required
Flanges	CL 125 or CL 150 Min.	CL 300 Min.
Running Clearances	None Specified	Specified

	<u>ANSI</u>	<u>API</u>
Shaft Runout	0.05 MM	0.025 MM
Temperature Limits	500°F 300°F for Cast Iron	800°F
Throat Bushing	Optional	Required
Cost	30 to 50% of API	

1. 5-11-2

2. API STD 610

3. ANSI (AVS)

4. 5-21-0

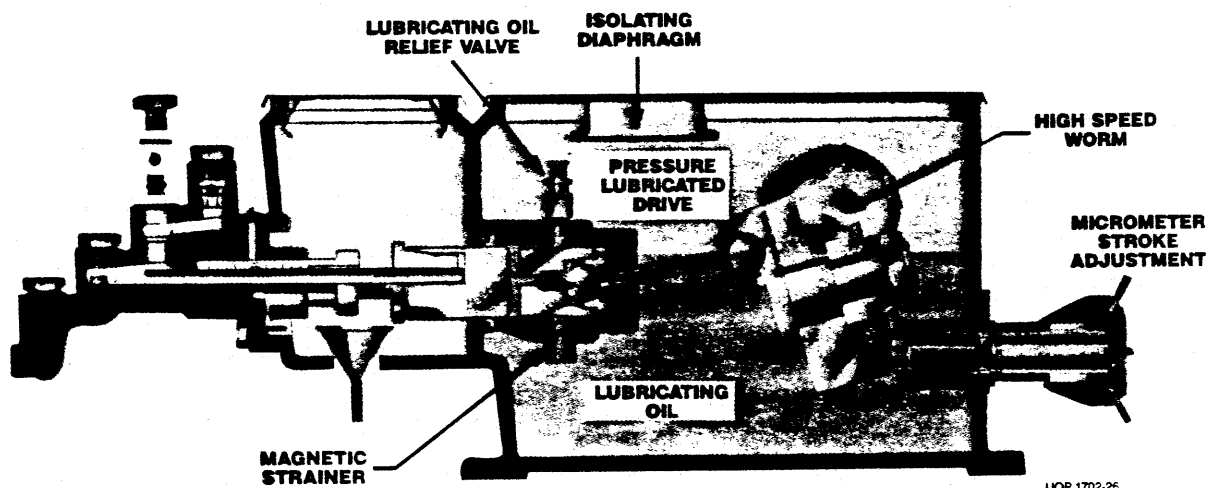
IX. OTHER PUMPS

INTRODUCTION

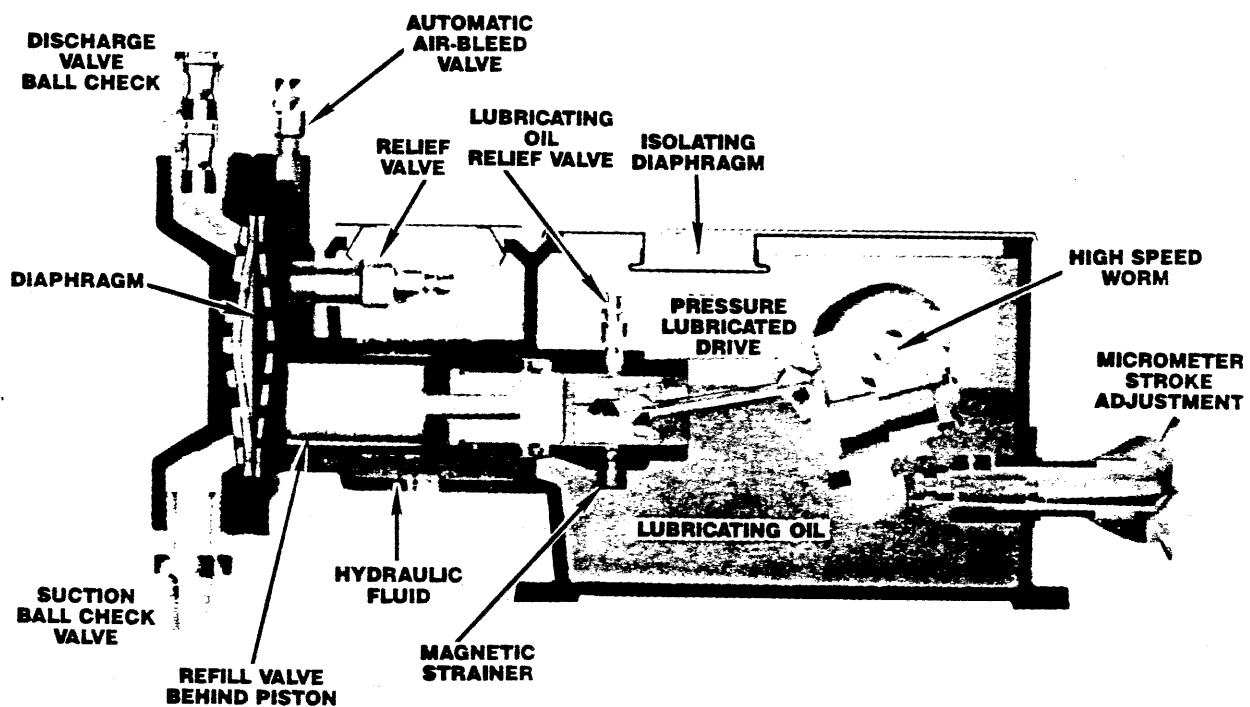
Other types of pumps used in the industry include piston and plunger, rotary and special pumps such as on-line, sump pumps and seal-less pumps.

PISTON AND PLUNGER-TYPE PUMPS

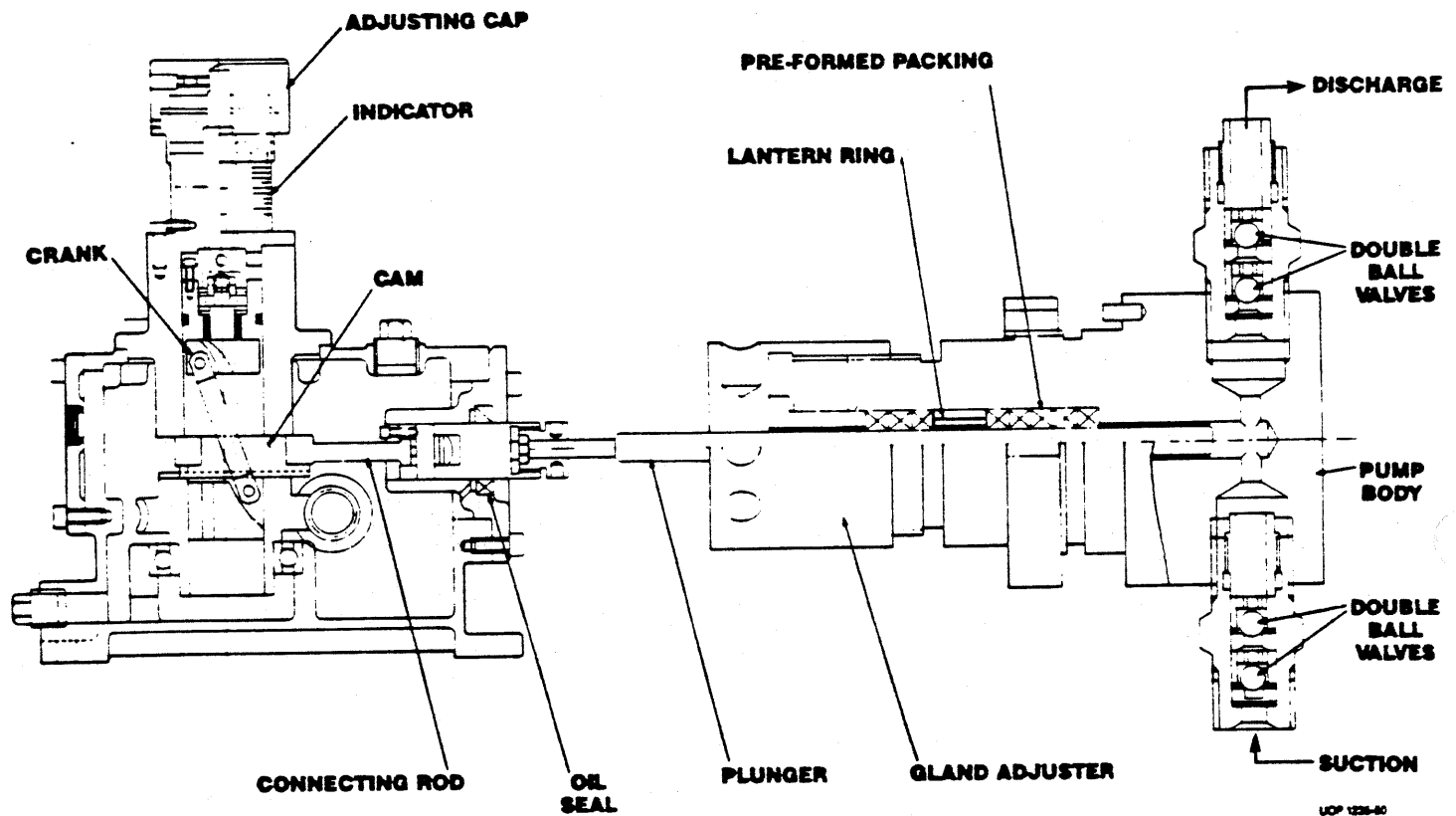
The family of positive displacement type pumps is used in the refinery for various injection services and for other duties where the low flow, high pressure requirements make a centrifugal selection poor. These pumps have efficiencies in the 80% range and greater and are very dependable. They do require more maintenance in general, but this can be minimized by insuring proper lubrication and conservative piston and plunger speeds. Various drive arrangements are available which afford wide variations in capacity. Steam driven, variable speed transmission, and variable stroke give an effective tool for applying these pumps to a variable capacity situation. The pulsations produced by these pumps are reduced by using pulsation bottles and by using multiple stroking. However, many times the centrifugal type pump is used where the flow and pressure requirements would strongly point towards a piston or plunger design due to the need for a very smooth and steady flow. Another problem with injection pumps is that the liquid being pumped tends to gum up the valves and they no longer seal properly, so the valves have to be removed and cleaned.

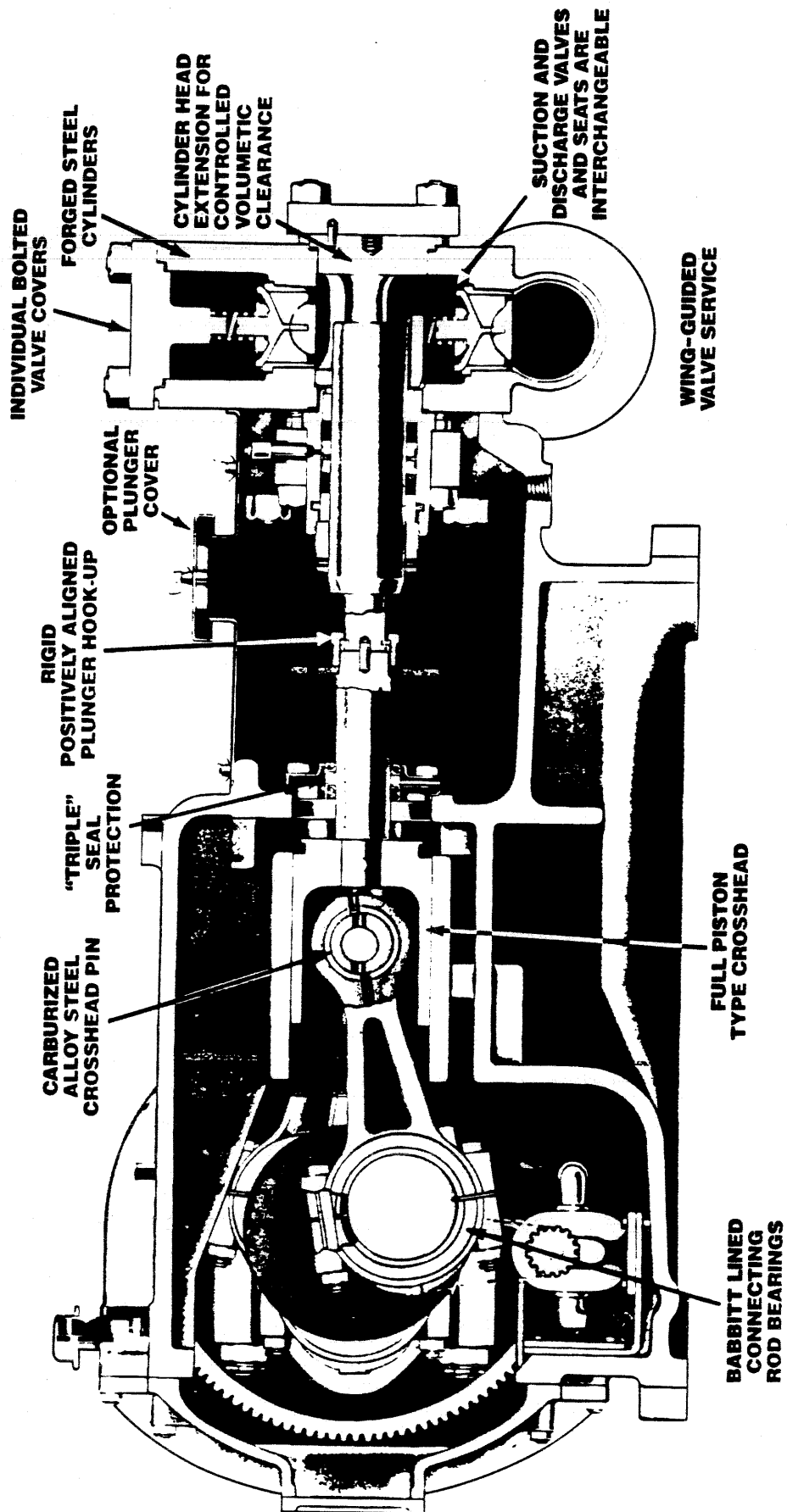


UOP 1702-26



INJECTION PUMP





ROTARY PUMPS

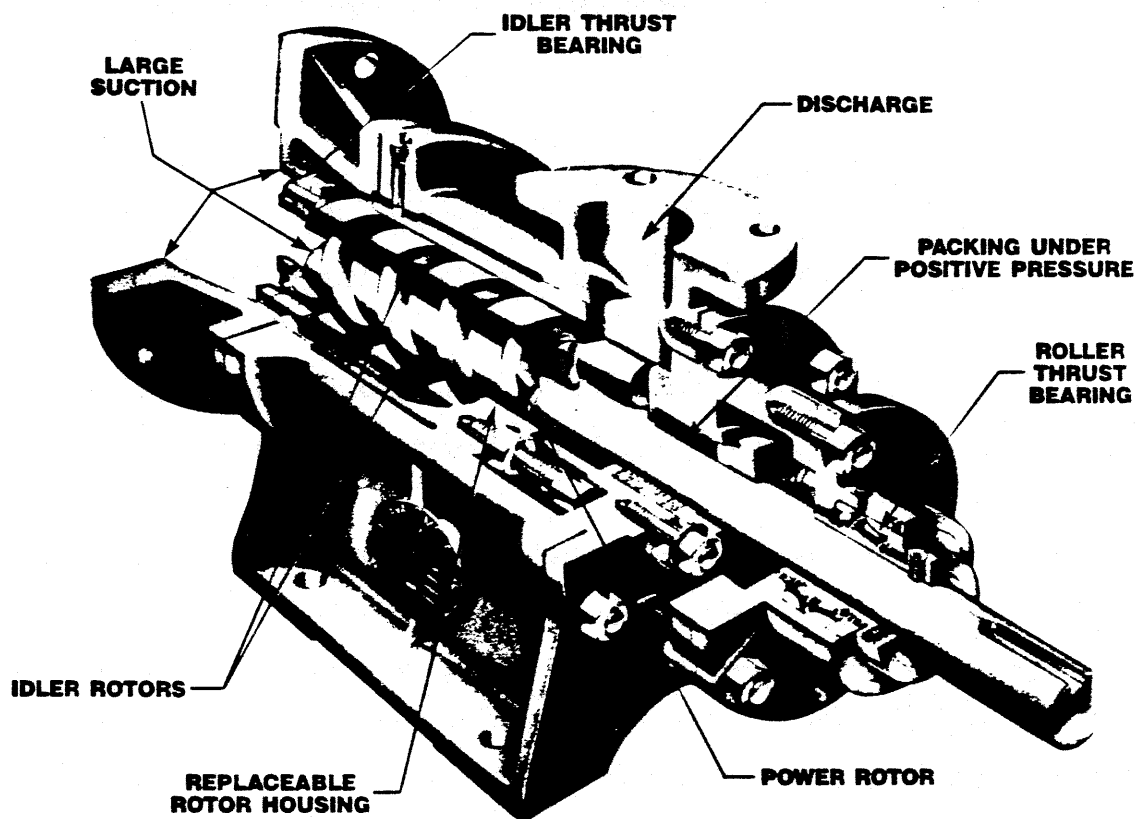
The rotary pump is a versatile and widely used type of pump. Two very prominent uses are in fluid power systems and seal and lubricating oil consoles. The rotary pump is built in capacities from a fraction of a gallon to more than 5000 gpm with pressures ranging up to 10,000 psi and handling viscosities up to 1,000,000 SSU. A lower limit of something like 200 SSU is needed to give a reasonable life when differential pressures approach 1000 psi or more. The rotary pump is defined as a positive displacement type pump because of the characteristic close-running clearances generally employed which substantially limit internal leakage. In the rotary pumps, mechanical displacement of the fluid from inlet to outlet is produced by trapping a slug of fluid between one or more moving elements such as gears, cams, screws, vanes, lobes, plungers, or other similar devices within a stationary housing or casing. The rotary motion of the centrifugal pump is combined with the positive-pressure characteristic of the reciprocating pump resulting in a displacement device which delivers a given quantity of fluid with each shaft revolution.

A number of types are listed and described below.

1. **Vane:** vanes, blades, or rollers located in the periphery of a rotor surrounded by a stator to form cavities between two successive vanes which carry fluid from inlet to outlet.
2. **Piston:** a number of pistons reciprocating within cylinders arranged axially around the periphery of a rotor moving past inlet and outlet posts.
3. **Gear (external):** fluid is carried between teeth of two external gears and displaced as they mesh.

4. **Gear (internal):** fluid is carried between teeth of one internal and one external gear and displaced as they mesh.
5. **Lobe:** fluid is carried between one or more lobes on each of two rotors which are timed by separate means.
6. **Screw:** fluid is carried between screw threads on two or more engaged rotors and is displaced axially as they mesh.
 - a. **Timed:** separate timing gears, located either internally or externally, are required to maintain proper meshed relationship of the screw threads, and rotors are also generally supported by separate sets of bearings.
 - b. **Untimed:** rotors incorporate the use of generated thread forms which provide a synchronized gearing action making separate timing unnecessary.

In the gear-type pump, the flowing liquid is trapped in the gap between the teeth on each gear wheel and is carried around from the suction side to the discharge side of the pump in a clockwise direction on the top gear and on the lower gear counter-clockwise to a point where the teeth of the gears mesh and the liquid is forced out into the discharge. Normally, there is a built-in relief system so if the pressure exceeds the set pressure, it relieves back from the discharge to the suction. This particular pump has a jacket so it may be steam heated for very viscous liquids if required. This family of pumps is very susceptible to solids and is very dependent upon sufficient lubricity for the sliding contact of the gear teeth. A minimum viscosity of 400-600 SUS is required for a reasonable life in a moderate pressure level application. For pressure levels above 500 psig or so, the viscosity levels must increase as the pressure requirement goes up. Small injection type gear pumps are also available and offer a high degree of accuracy and low flow capabilities.

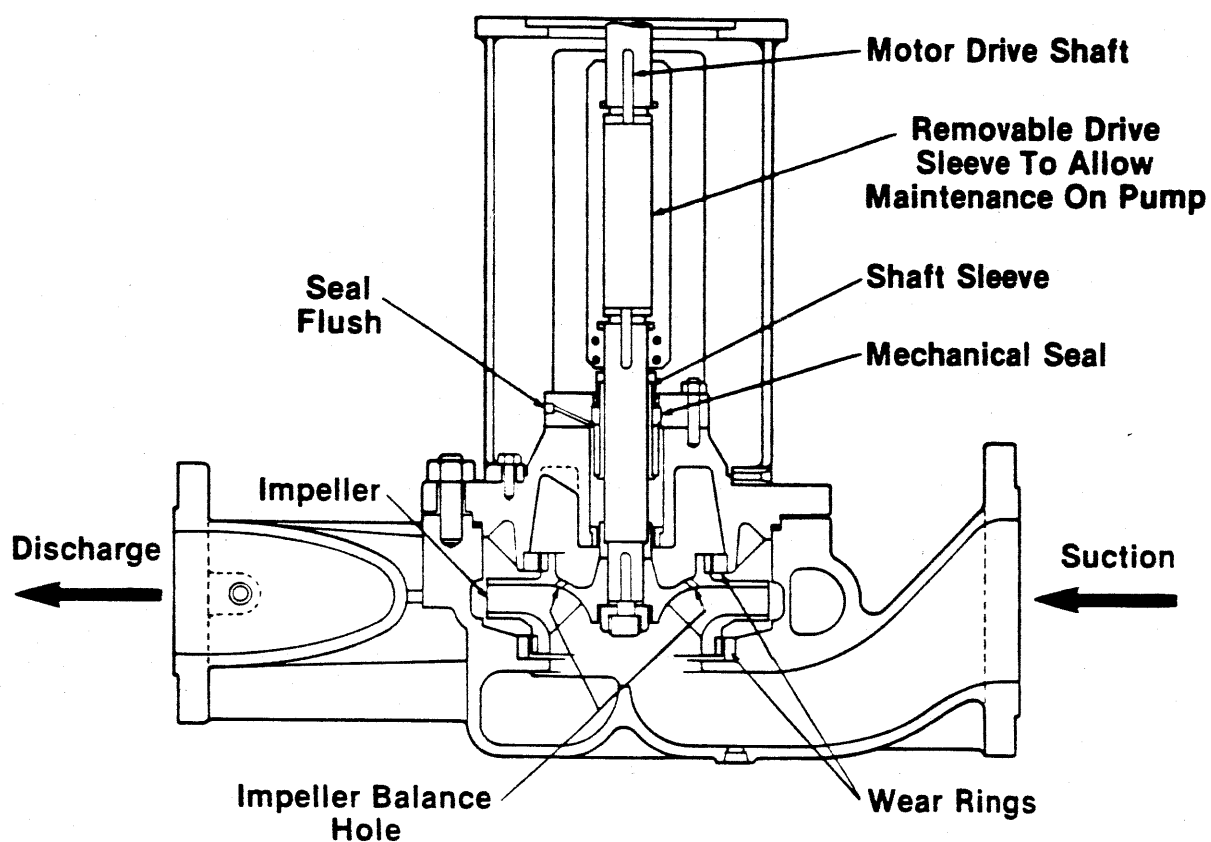


SPECIAL PUMPS

The following figures show in-line pumps, a sump pump, and a seal-less pump to illustrate some of the unusual designs which are possible when confronted with extraordinary process requirements. These pumps, in general, are not as rugged as conventional process pumps and will not offer interchangeability with conventional process pumps. However, each has its own particular application where it has proven to be a reliable and dependable alternative.

1. **In-line pump:** light duty, simplified piping and foundation requirements, no alignment distortion in high temperature, low npsh requirement; however, relies on driver bearings usually and should not be used for medium and heavy duty applications.
2. **Sump pumps:** convenient "packaged" method of controlling a level in a sump - low head and temperature capability.
3. **Seal-less pump:** used where mechanical seal capabilities are extended - has limited coverage.

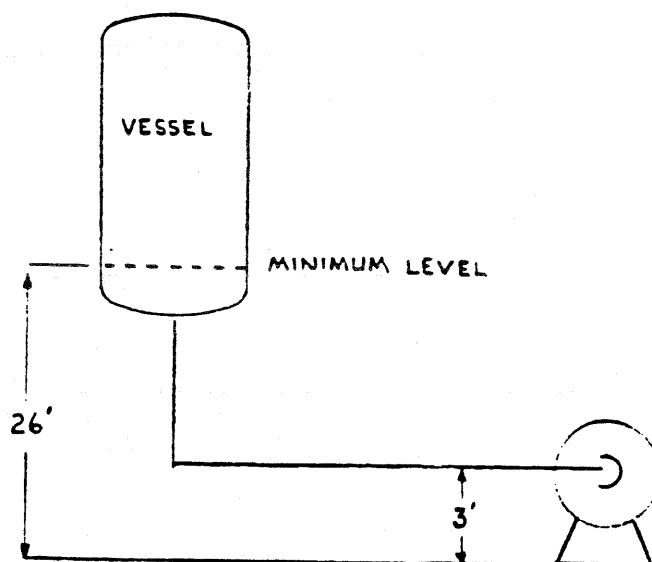
VERTICAL INLINE PUMP



UOP 1702-34

X. PROBLEMS

PROBLEM I



Liquid is at bubble point. Friction loss is 2 PSIG.

$$\text{NPSHA} = 23 - 6 = 17 \text{ Ft.}$$

Two existing pumps (operating and spare) Byron Jackson 4 x 6 x 13-1/4L (curve attached)

Present pump duty	Flow	600 GPM
	Head	500 Ft.
	NPSHR	14 Ft.
	SG	0.80
	Temp.	150°F
		60 Cycle

New conditions require flow to be increased to 780 GPM.

Now need 18 Ft. NPSH. What can be done?

Effective OCT. 76

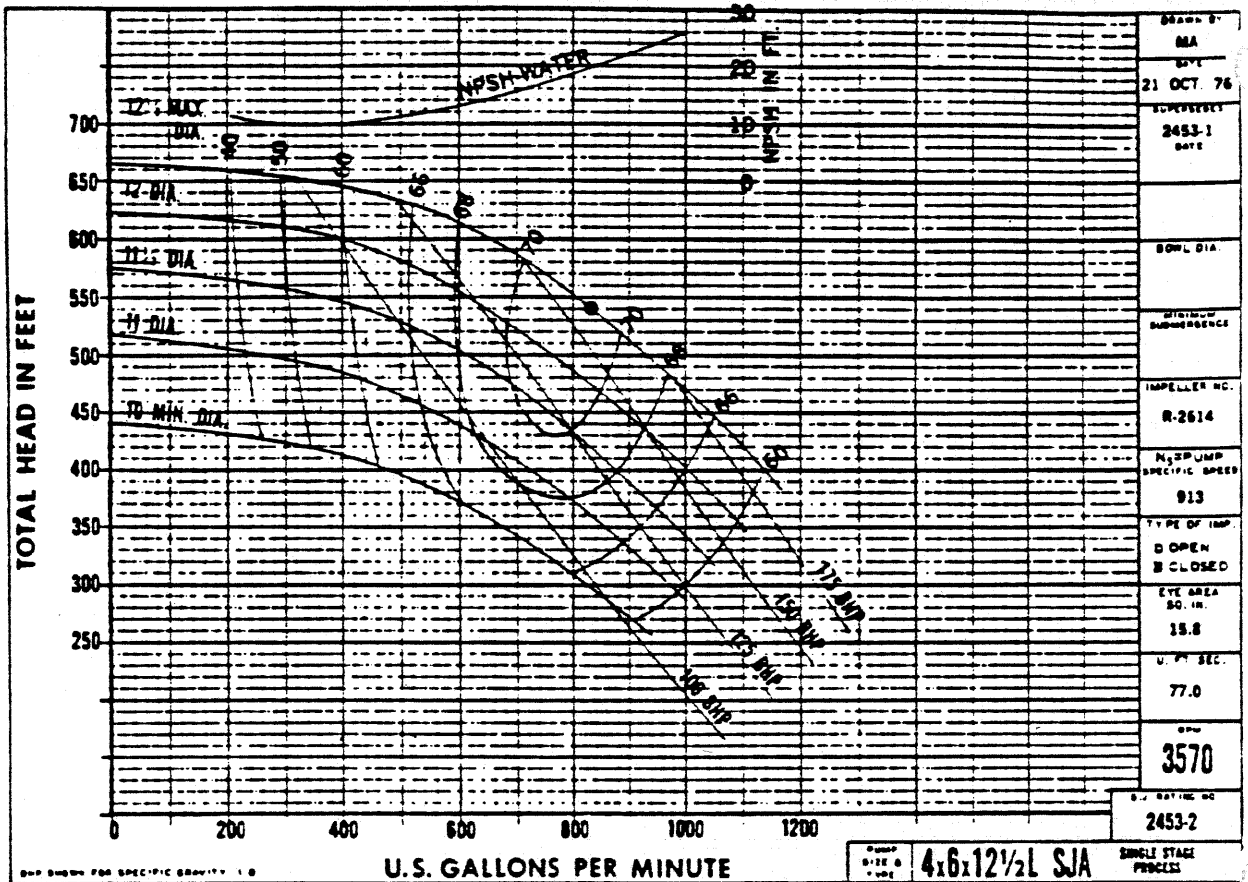
Byron Jackson Pump Division

BORG-WARNER CORPORATION



Section 1-250

Page 1-250-39



PROBLEM II

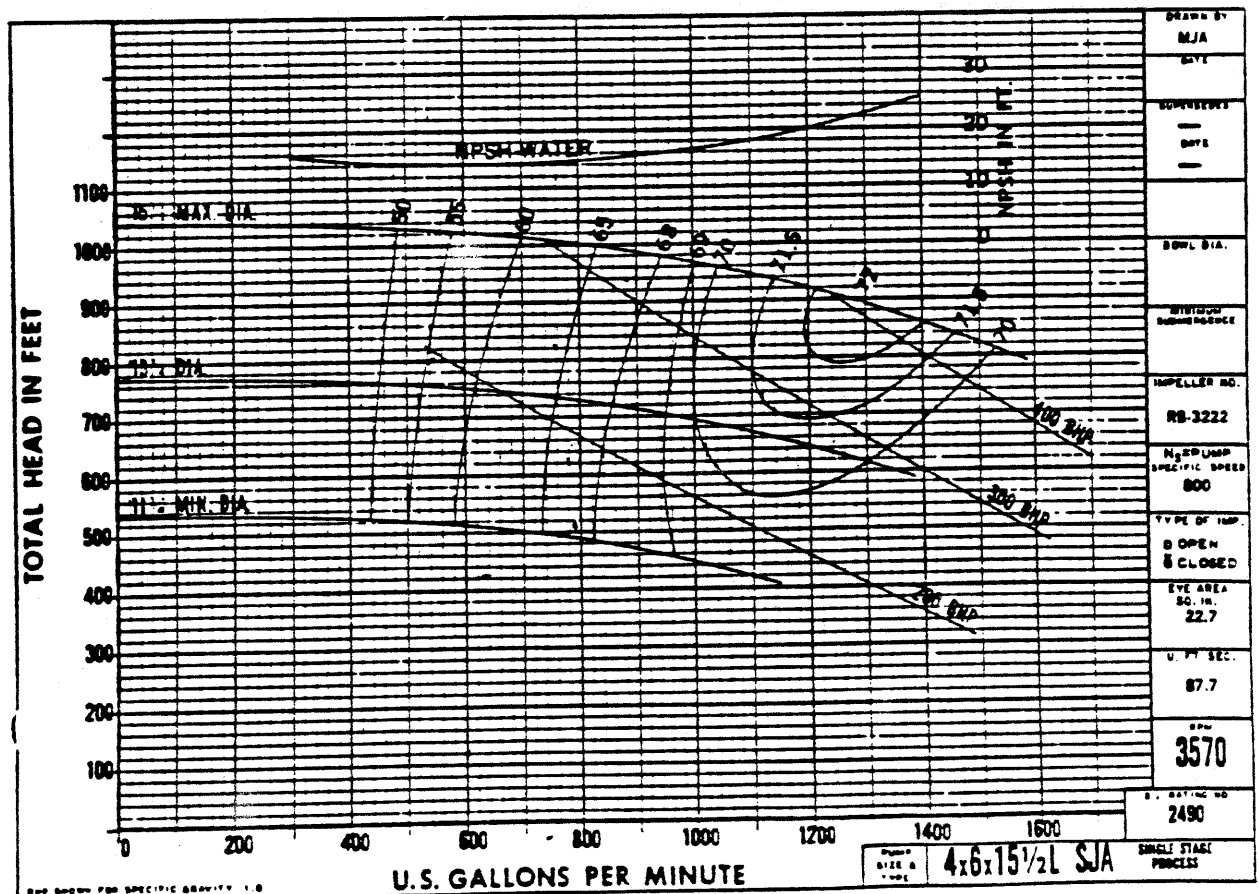
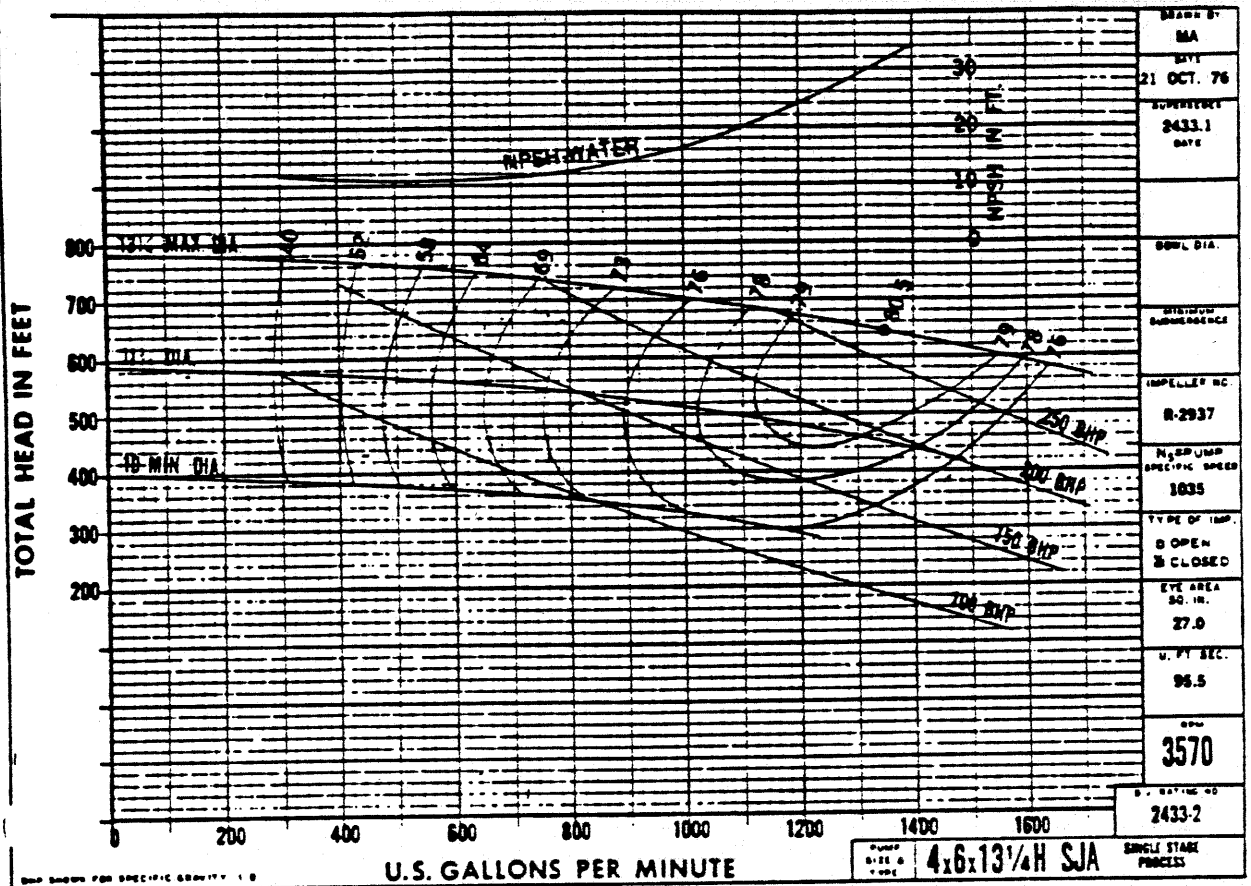
Pump is specified for 1200 GPM, 450 feet of head, 60 cycle.

Quotations are received from three vendors as follows:

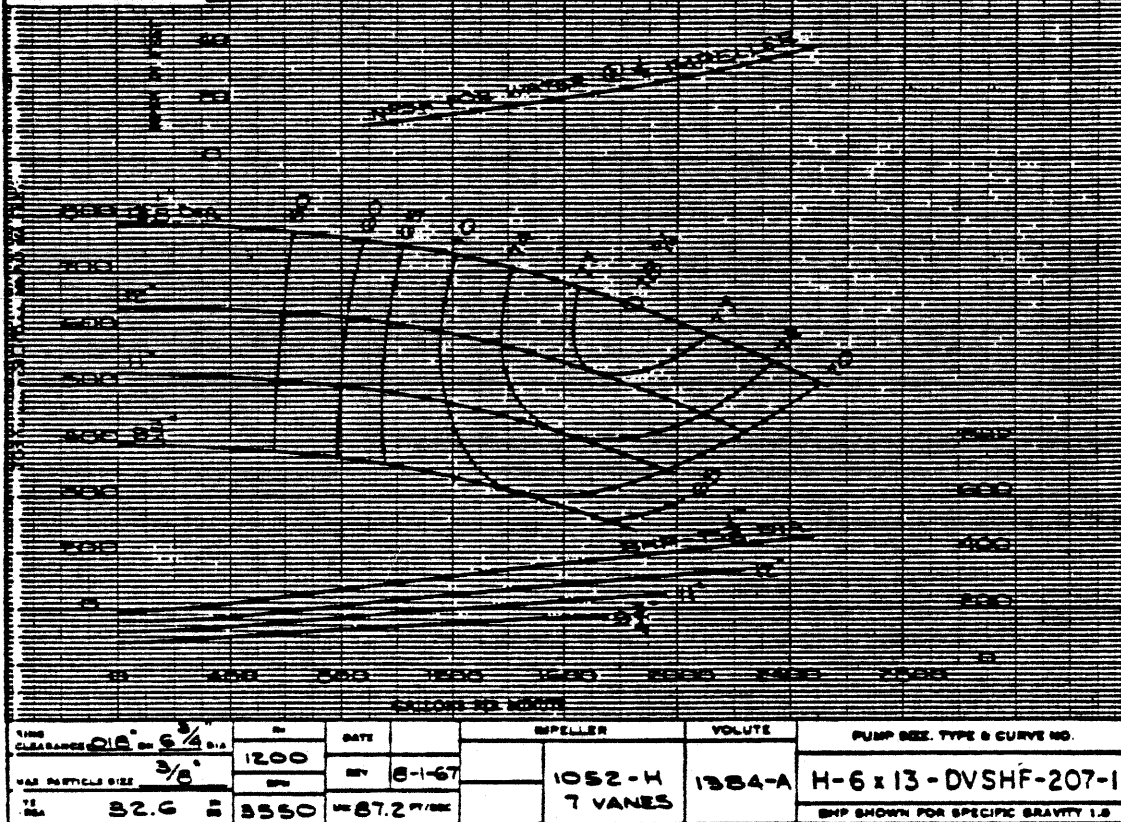
Byron Jackson	4 x 6 x 13-1/4H	Single Suction, 3550 RPM
United	H x 6 x 13	Double Suction, 3550 RPM
Pacific	8 x 23HVC	Double Suction, 1750 RPM

Pump curves are attached.

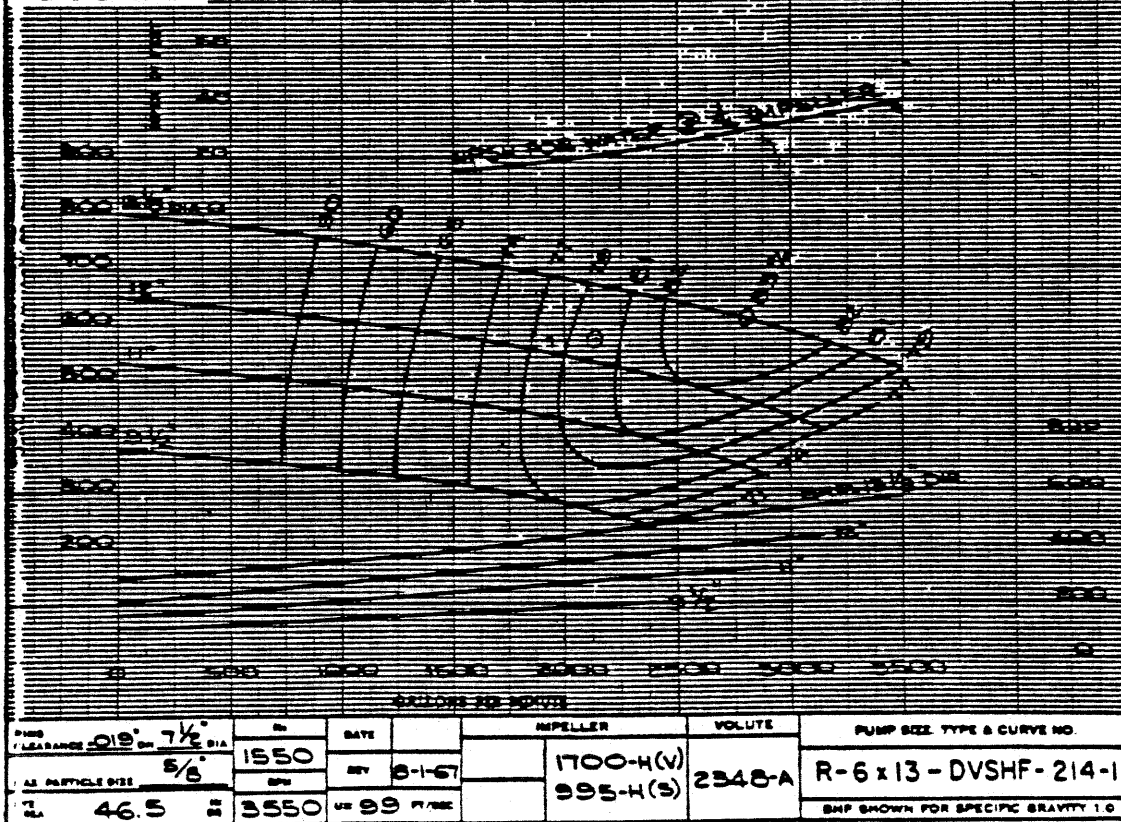
What are positive and negative qualities of each pump?



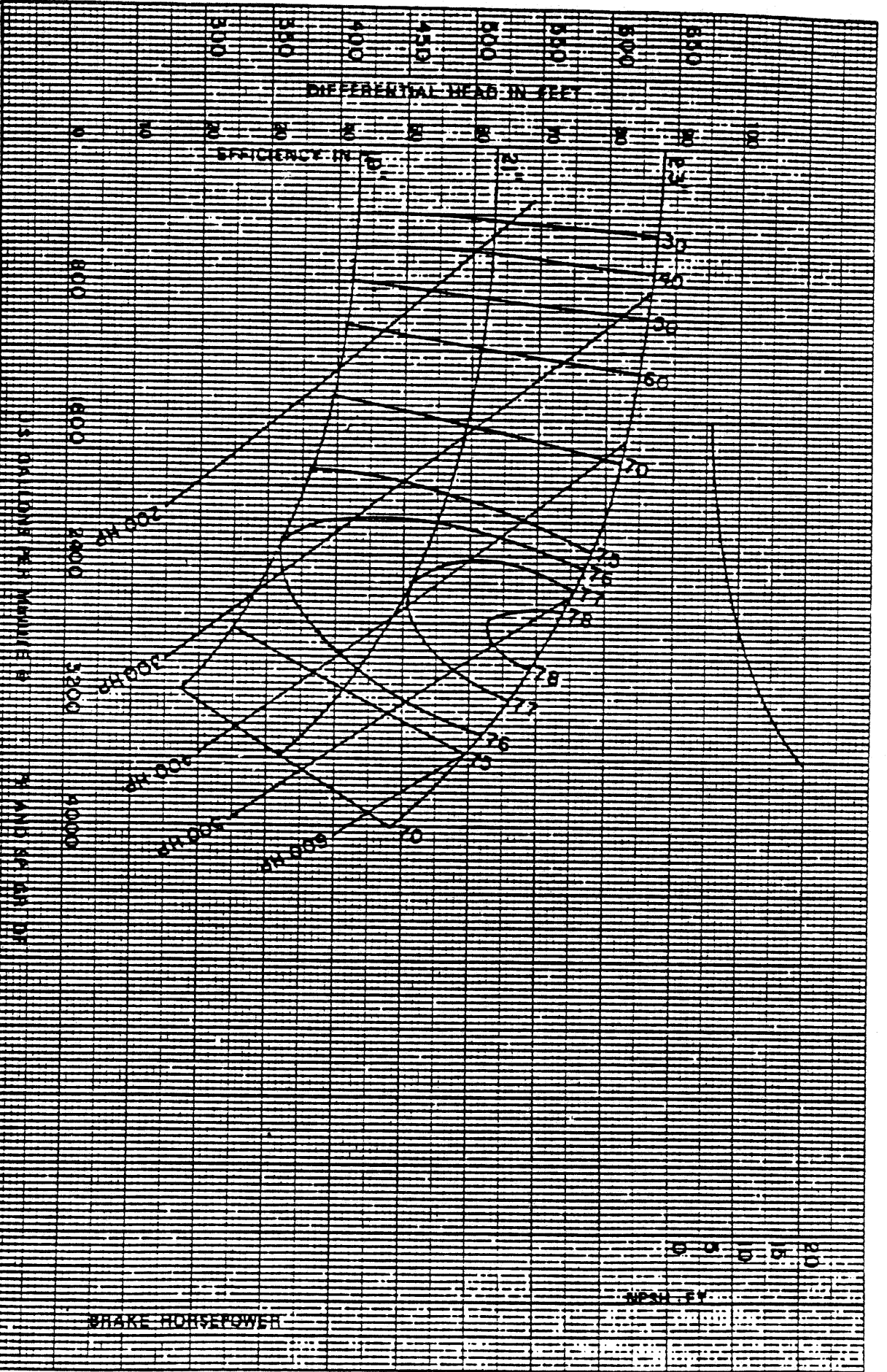
**UNITED
PUMPS**



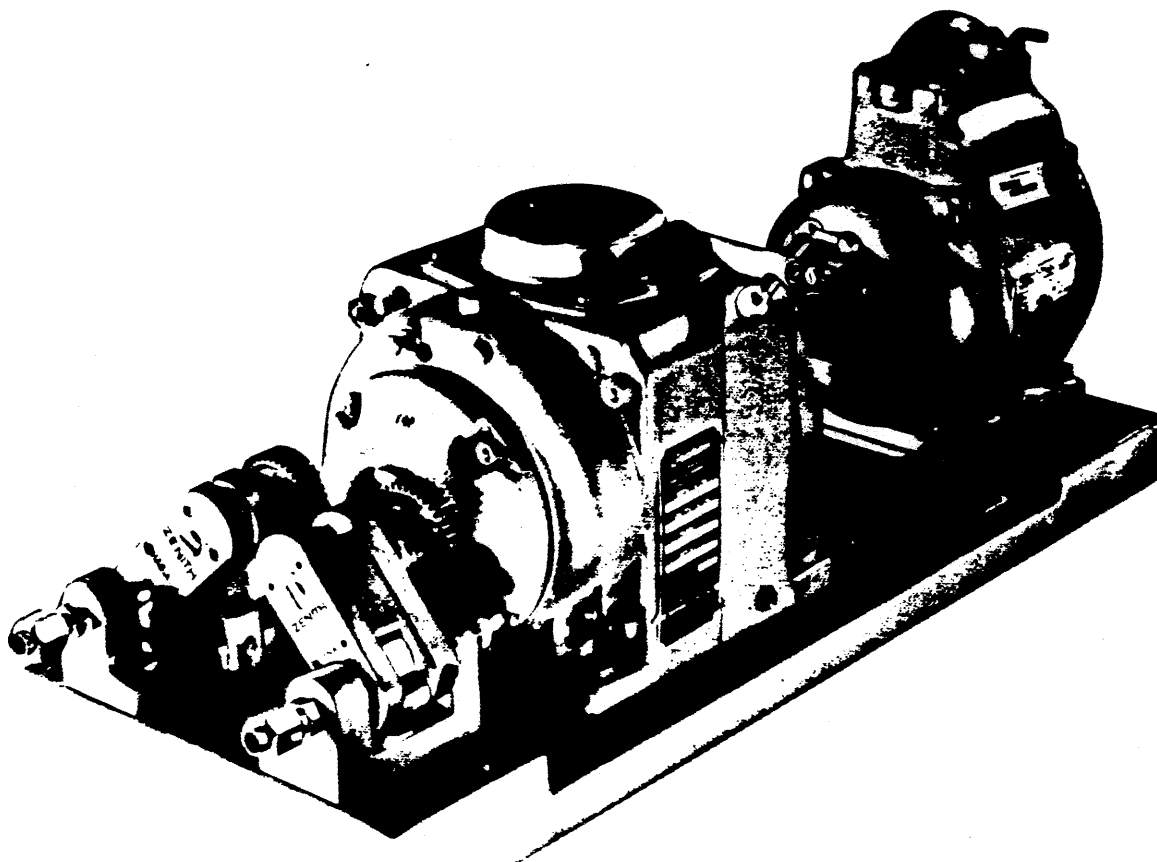
**UNITED
PUMPS**



SUPSEDES CURVE DATED 2-18-66

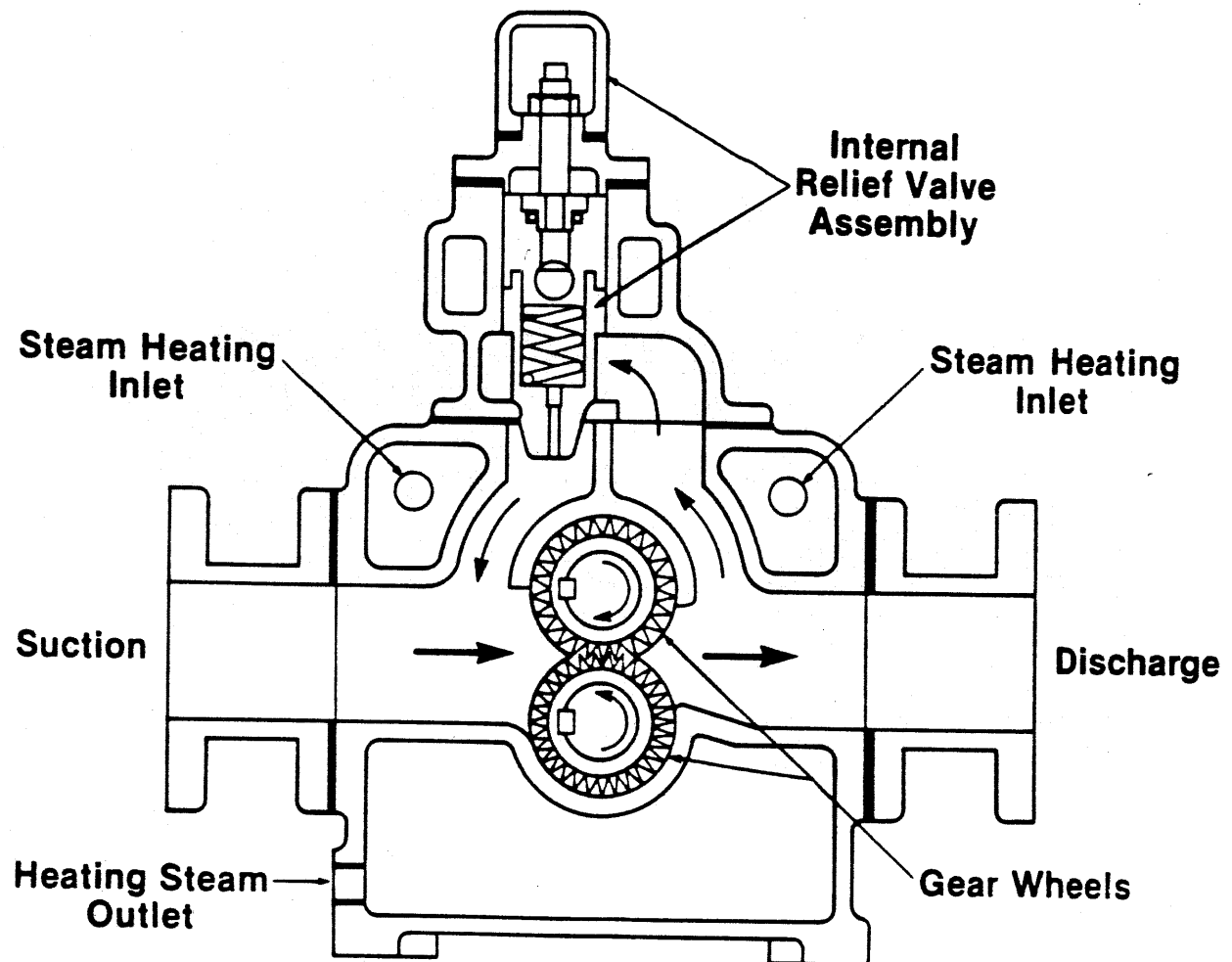


PACIFIC PUMPS PACIFIC PUMPS DIVISION DRESSER INDUSTRIES, INC.	R.P.M. 1770	IMPELLER PATTERN D-2765	IMP. EYE AREA
PUMP SIZE AND TYPE 23 HVC	NO. STGS. 1	WEARING RING CLEARANCE .018 IN	MIN. INT. DIM. 15/16
DATE 3-1-78	PAGE NO.	E. NO. 177	177

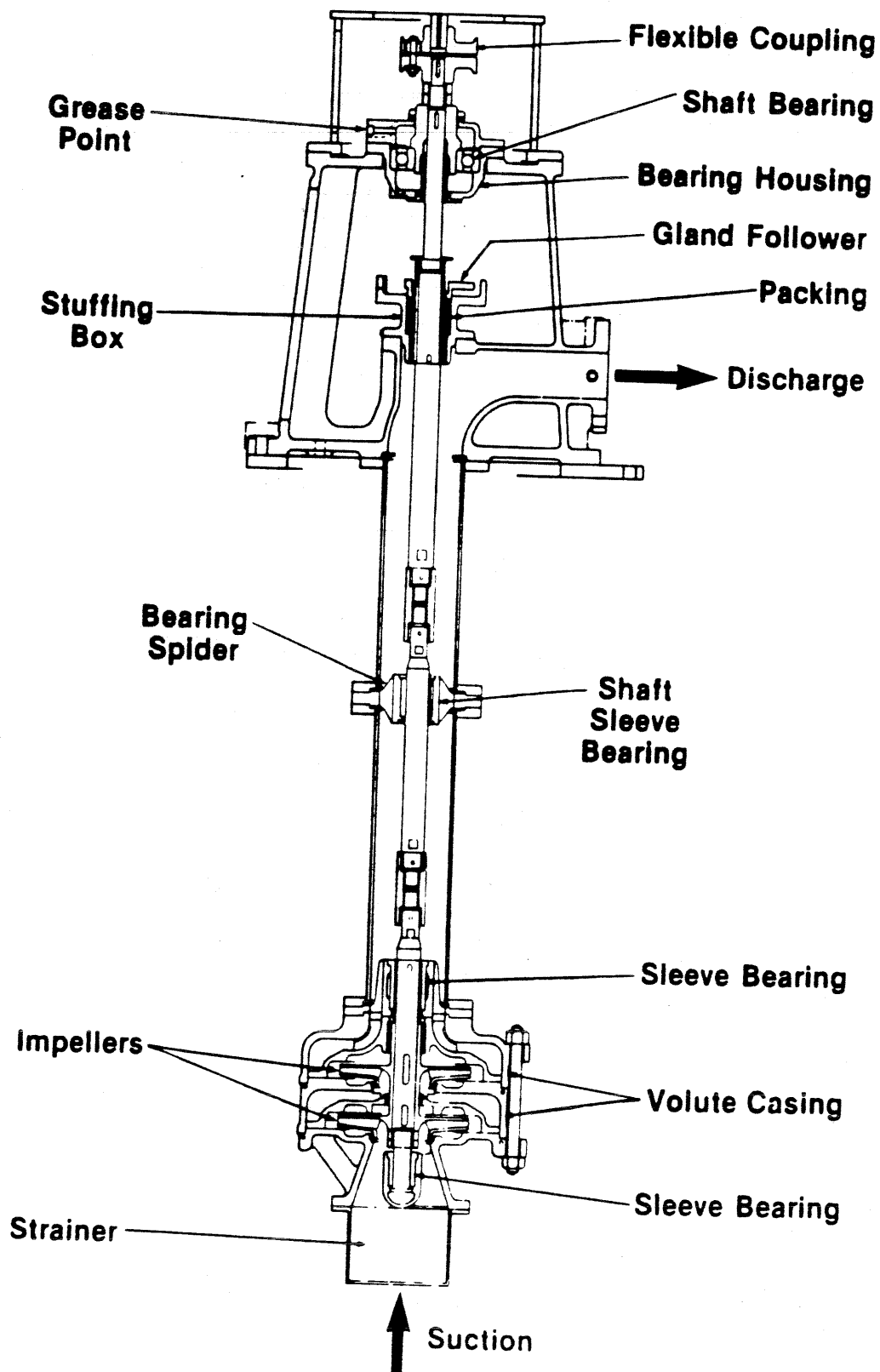


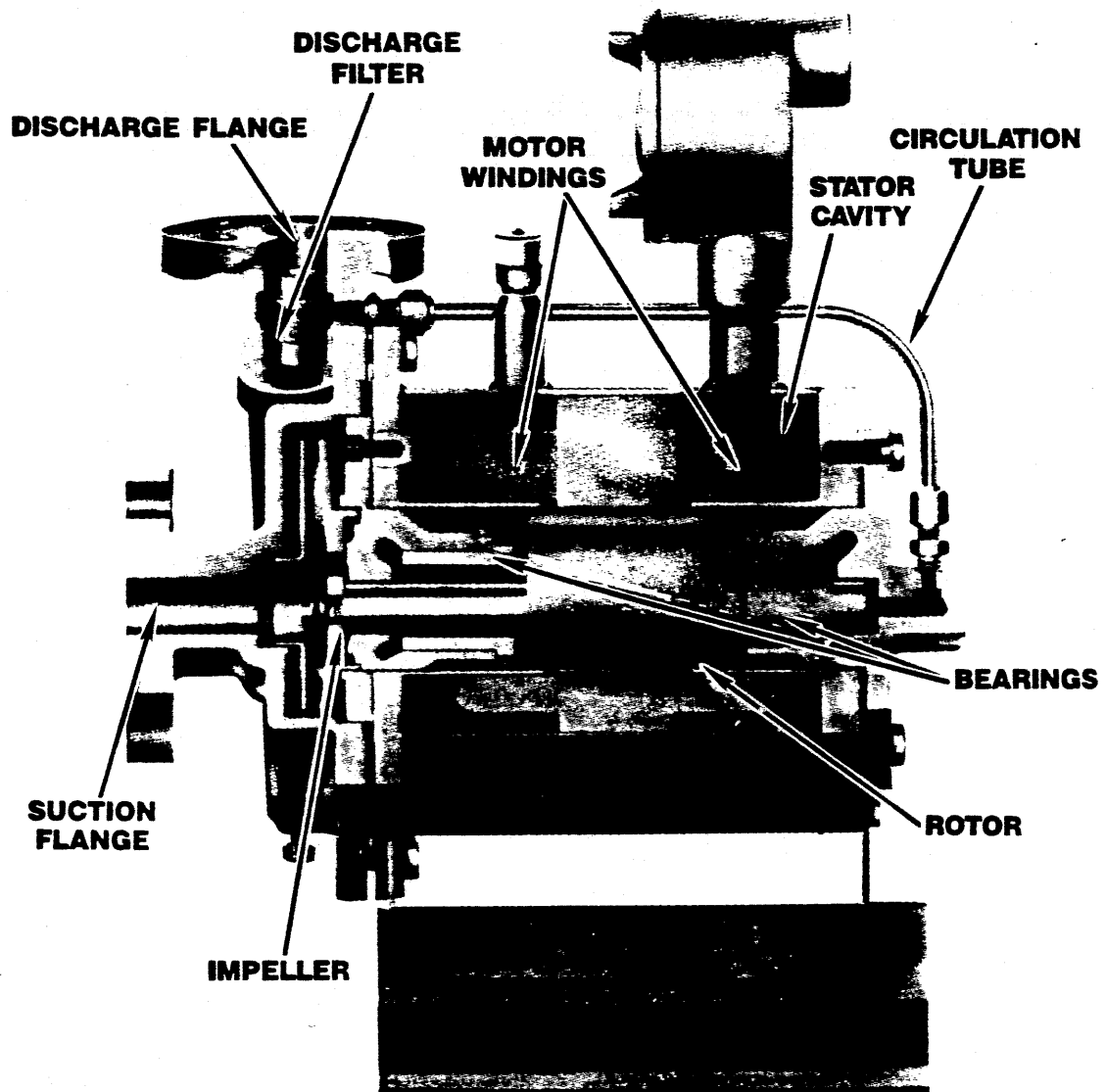
UOP 1702-21

GEAR PUMP



2-STAGE SUBMERSIBLE LIFT PUMP





EFFICIENCY

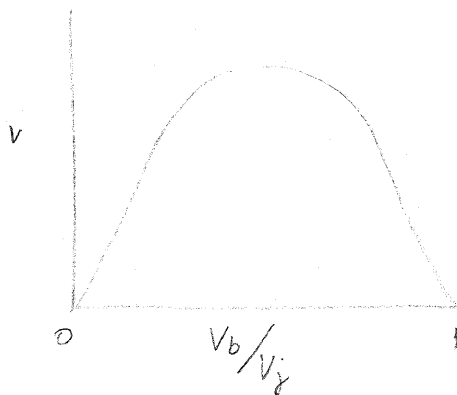
$$\text{BLADE SPEED} = DW/2 = \pi DN/720$$

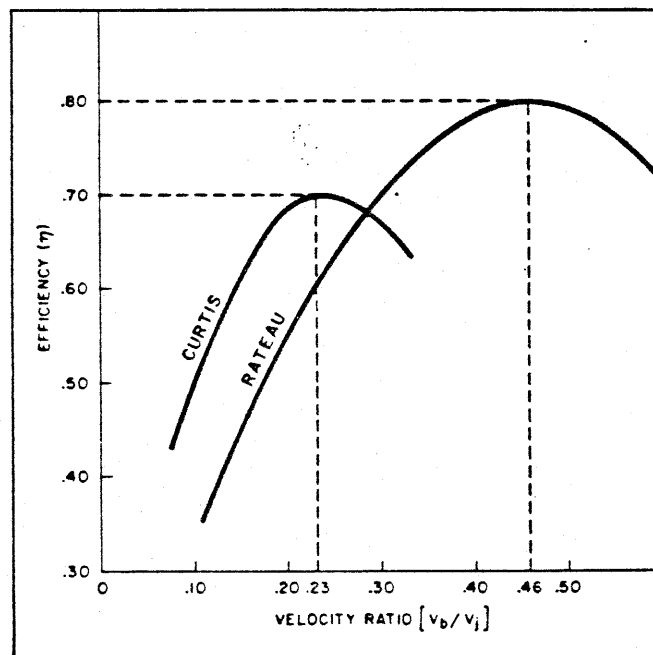
WHERE $W = 2\pi \text{ radians/rev} \times \text{rev/min} \times \text{min/60 sec} \times D/2 \times \text{ft/12 inch}$

$$\text{JET SPEED} = 223.7 (\Delta H)^{0.5}$$

EFFICIENCY IS PROPORTIONAL TO THE RATIO
OF BLADE SPEED TO JET SPEED.....

$$\text{EFFICIENCY} \approx \boxed{V_b/V_j}$$





Efficiency of Curtis and Rateau Stages

EFFICIENCY

ESTIMATED EFFICIENCY.....

SINGLE STAGE.....~ 35%

MULTI-STAGE:

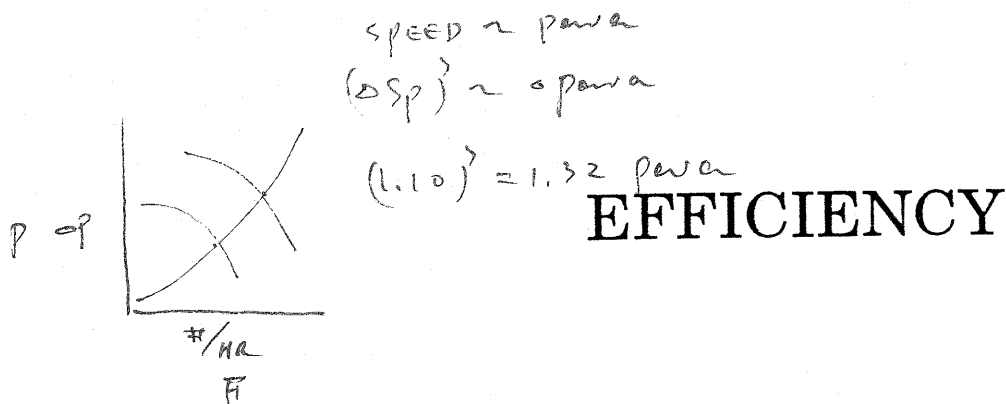
1000 HP, 3000 RPM.....~50%

INCREASE PER 1000 RPM - 2%

INCREASE PER 1000 HP - 2%

MAXIMUM EFF. FOR COND. - 75-80%

MAXIMUM EFF. FOR NON-COND. - 65-75%



EFFICIENCY DROPS AS SPEED AND HORSEPOWER ARE REDUCED. AS AN ESTIMATE, INCREASE THE STEAM RATE (LBS/HR) FOR PART LOAD IN THE FOLLOWING MANNER AT CONSTANT SPEED:

3/4 LOAD.....+4%
 1/2 LOAD.....+10%
 1/4 LOAD.....+35%

STEAM RATE INCREASE.

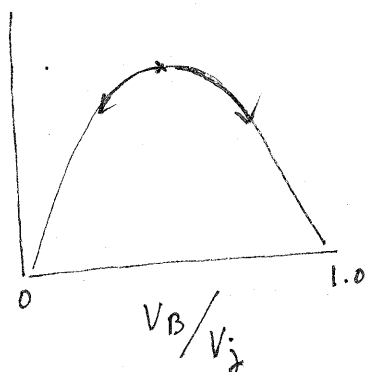
$$\frac{\#}{\text{HR-HP}} = \frac{10}{500 \text{ HP}} = 0.02$$

$$\frac{1000 \text{ HP}}{1000 \text{ #/HR}} = 1.0$$

$$500 \times 10 \times 1.10 = 5500$$

ON DECREASE OF SPEED USE THE FOLLOWING TO ESTIMATE CHANGE IN STEAM RATE:

$$\Delta \text{ Steam Rate} \sim (\text{RPM}_{\text{ACTUAL}} / \text{RPM}_{\text{DESIGN}})^{1/2}$$



Δ Speed

IF 90% η_D

$$\sim (90\%)^2 = 81\%$$

$$\frac{1.0}{.81} =$$

VELOCITY LIMITS

NEMA SM 23

APZ STDS 611/612

IMPORTANT DUE TO CONTROL, PERFORMANCE,
AND NOISE EFFECTS.....

$$\text{VELOCITY (Ft/sec)} = \frac{\text{Flow (lb/hr)} \times \text{Sp. Vol. (ft}^3\text{/lb)}}{19.65 \times \text{pipe diameter (In)}^2}$$

Maximum Velocities (Ft/sec): PER NEMA SM 23

	<u>INLET</u>	<u>EXHAUST</u>
NON-CONDENSING	175	250
CONDENSING	175	<u>450</u> →
INDUCTION	175	175
EXTRACTION	250	250

HARDWARE

A. CASINGS

B. BLADES

C. BEARINGS

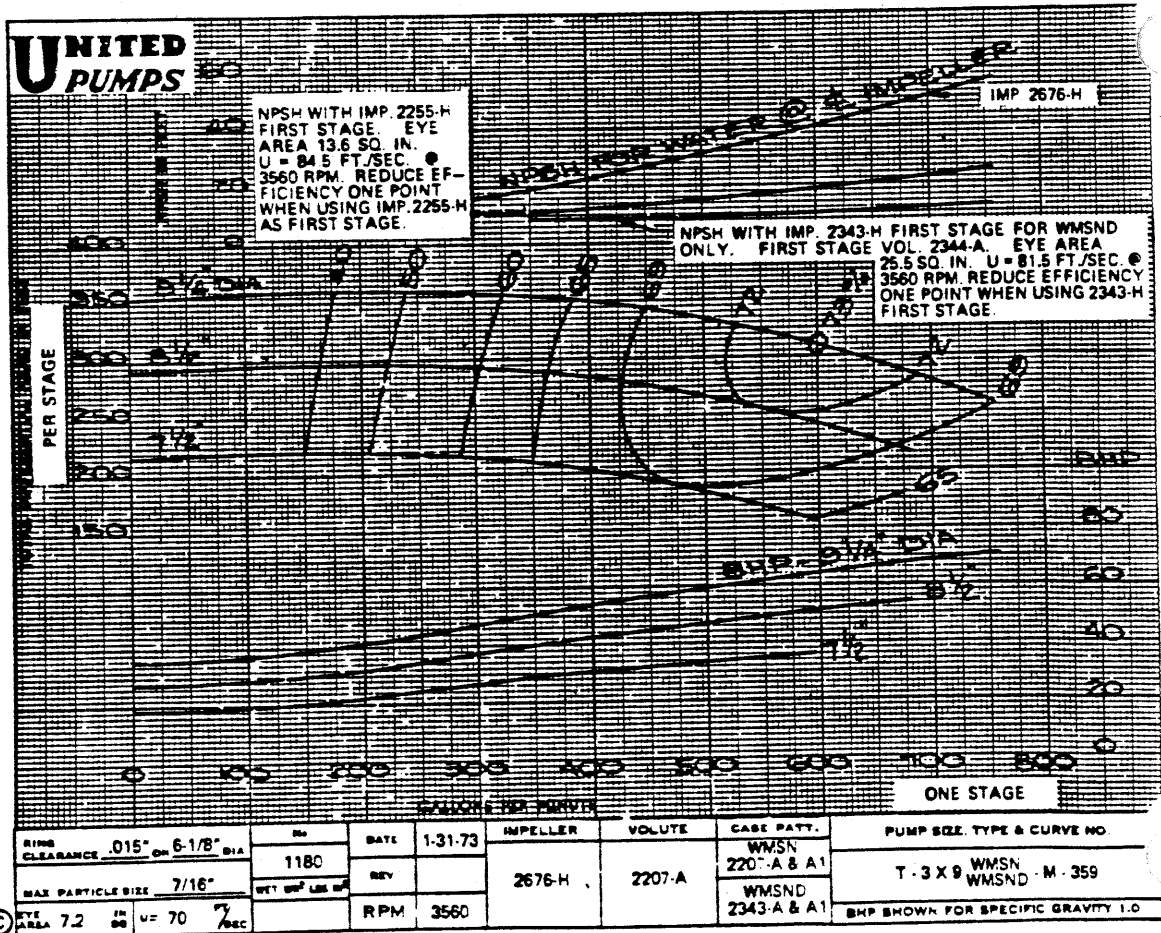
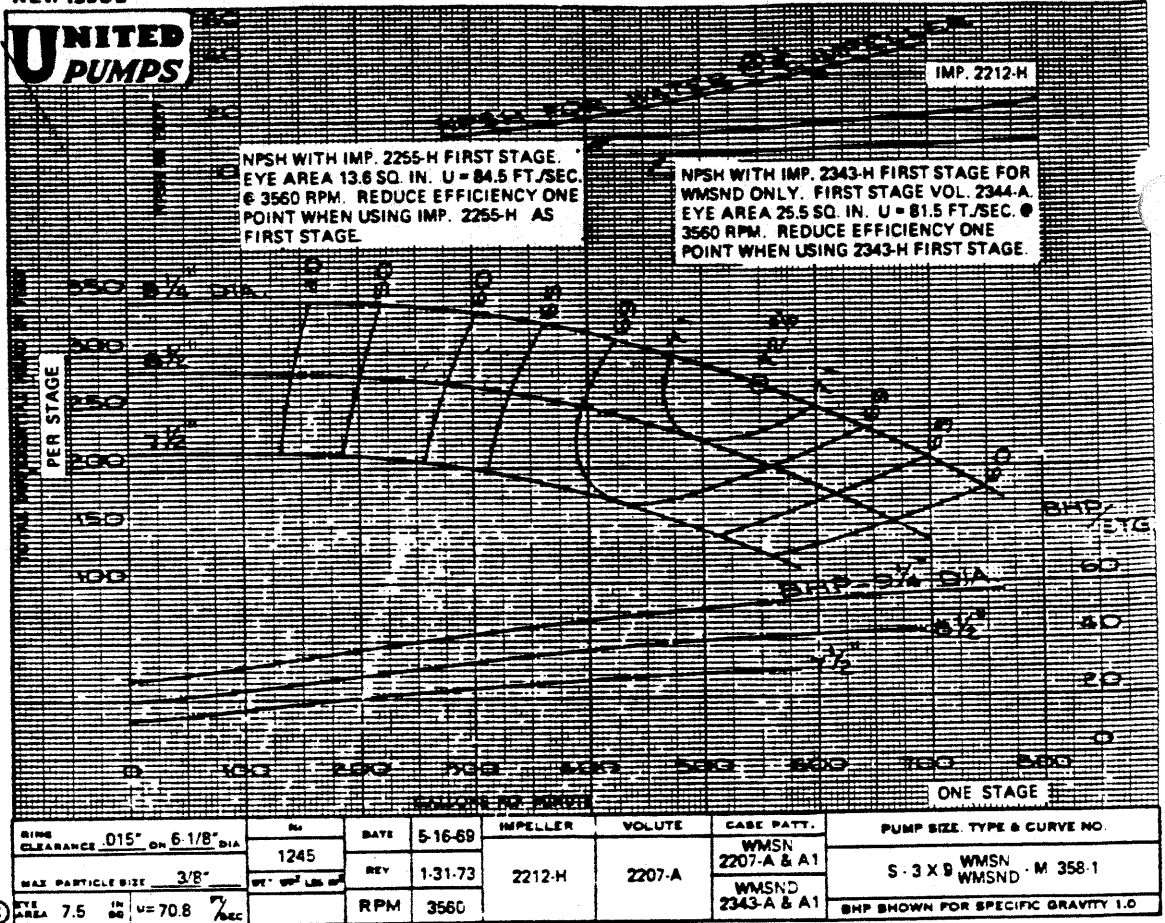
D. SEALS

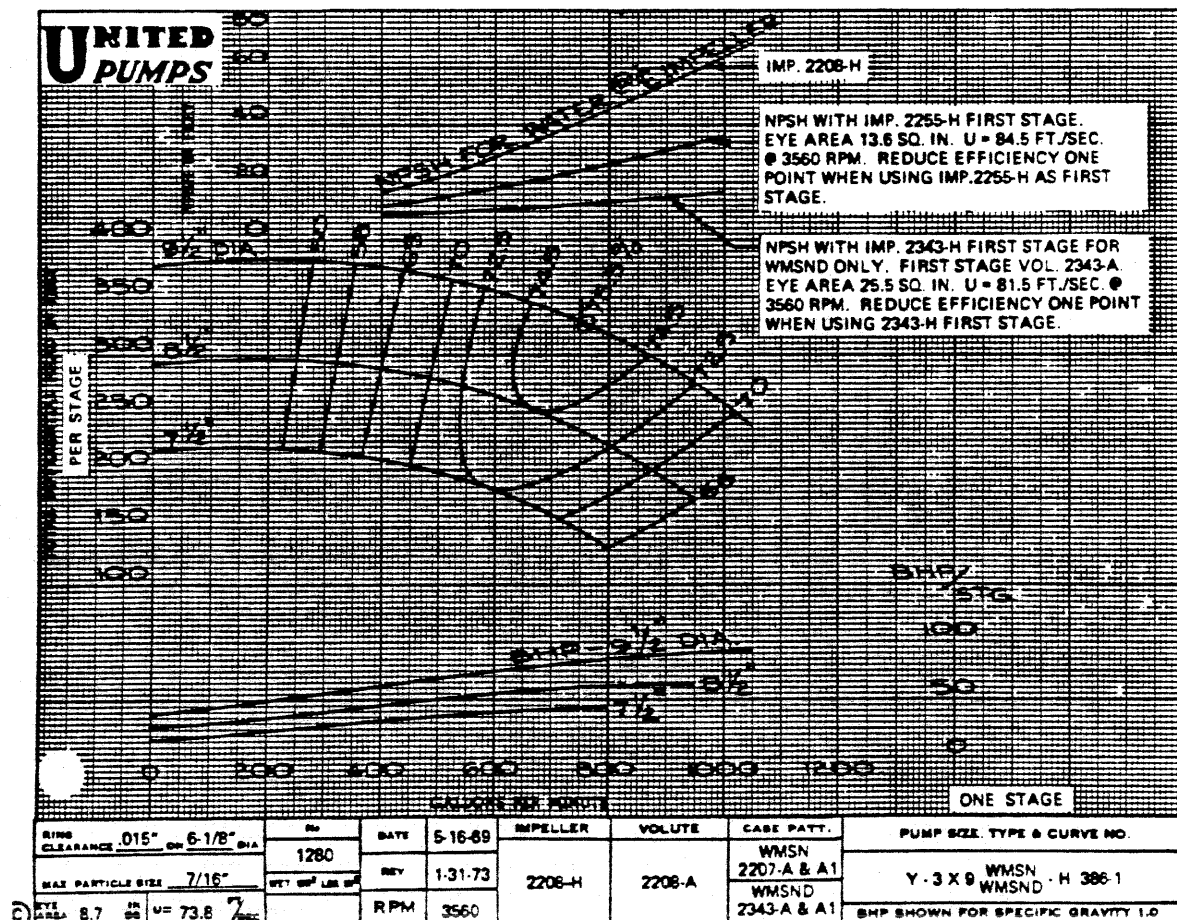
E. CONTROL/PROTECTION

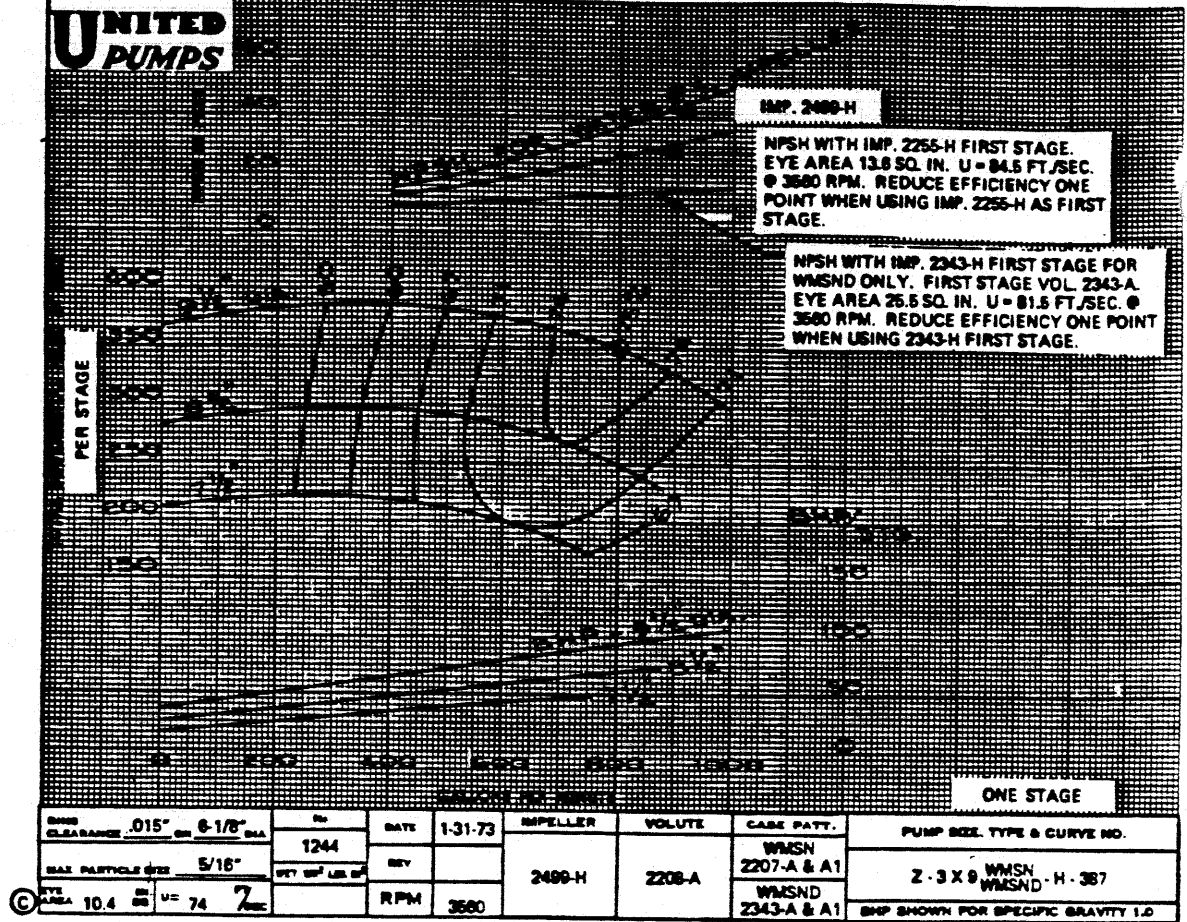
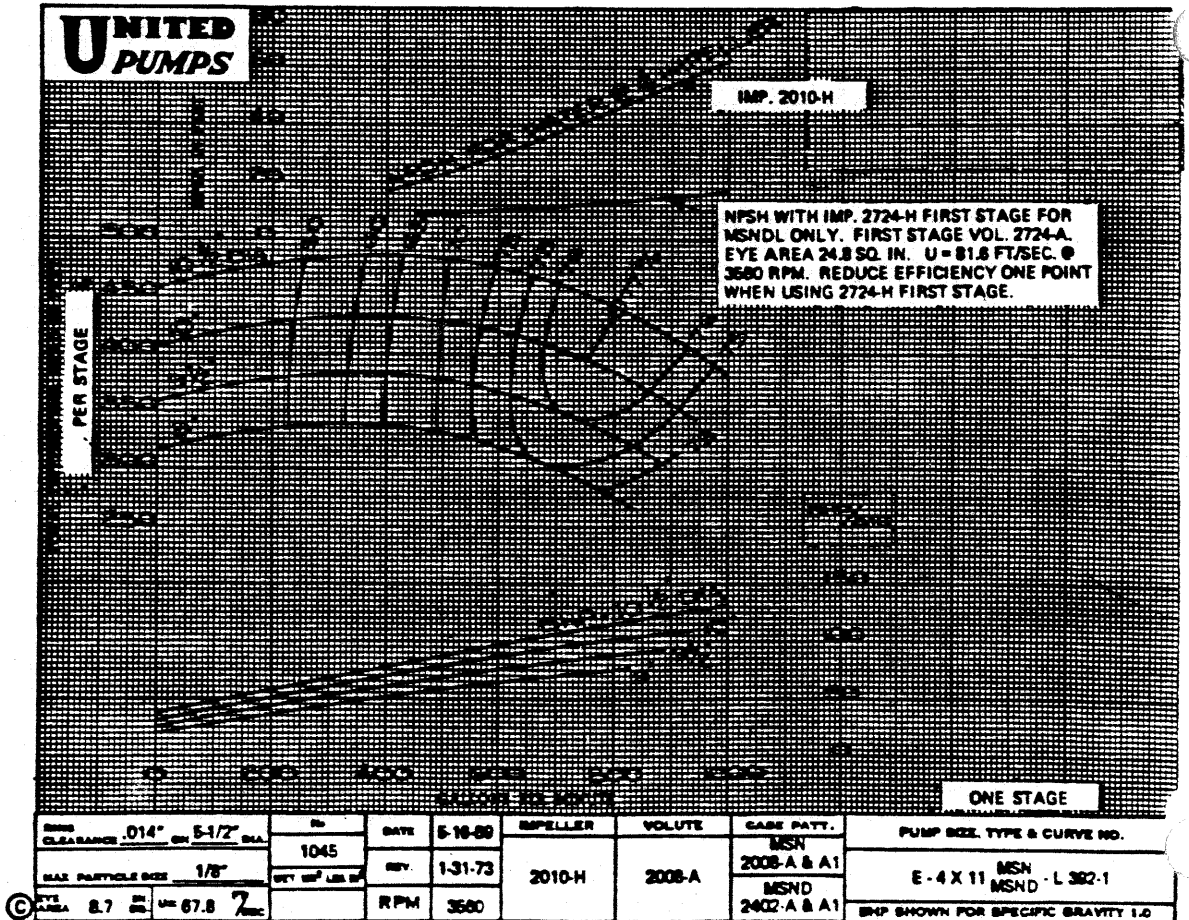
PROBLEM III

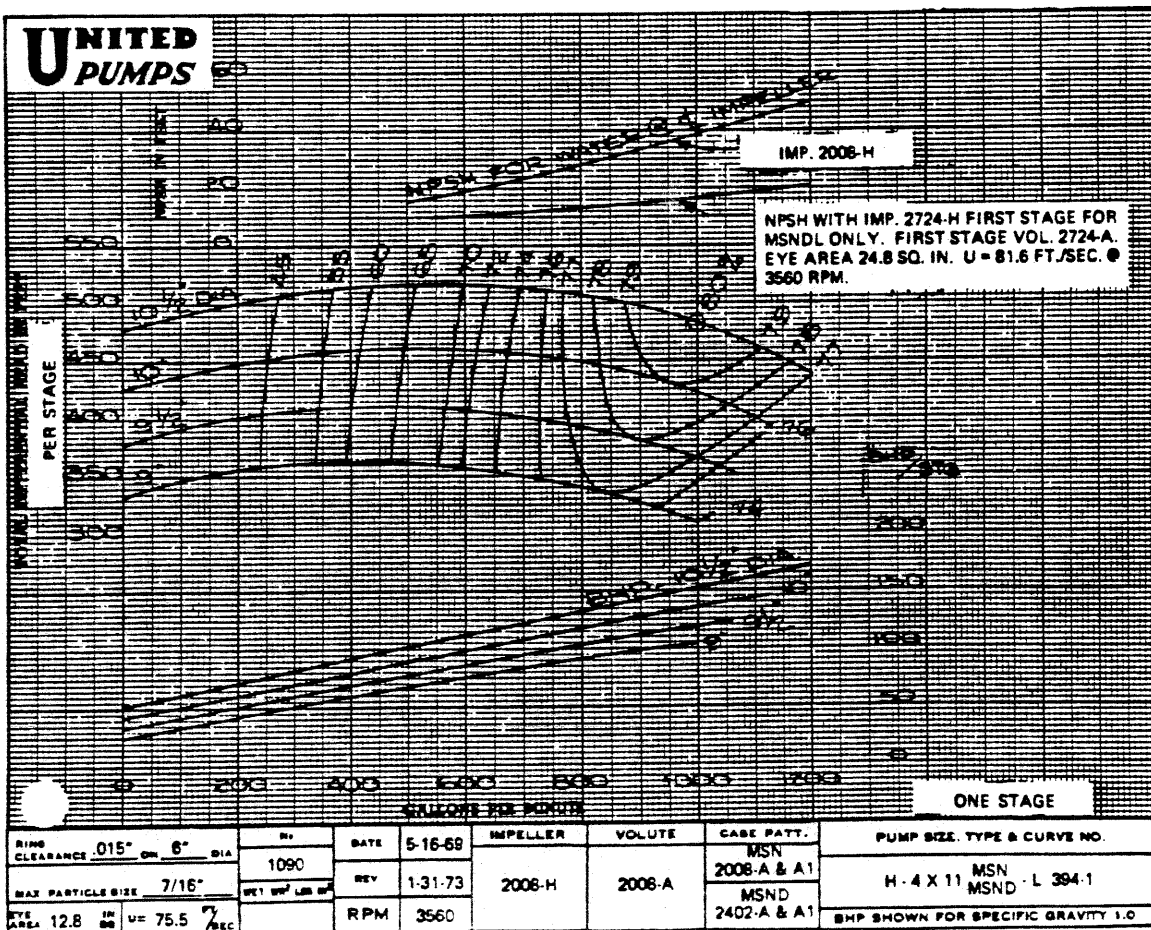
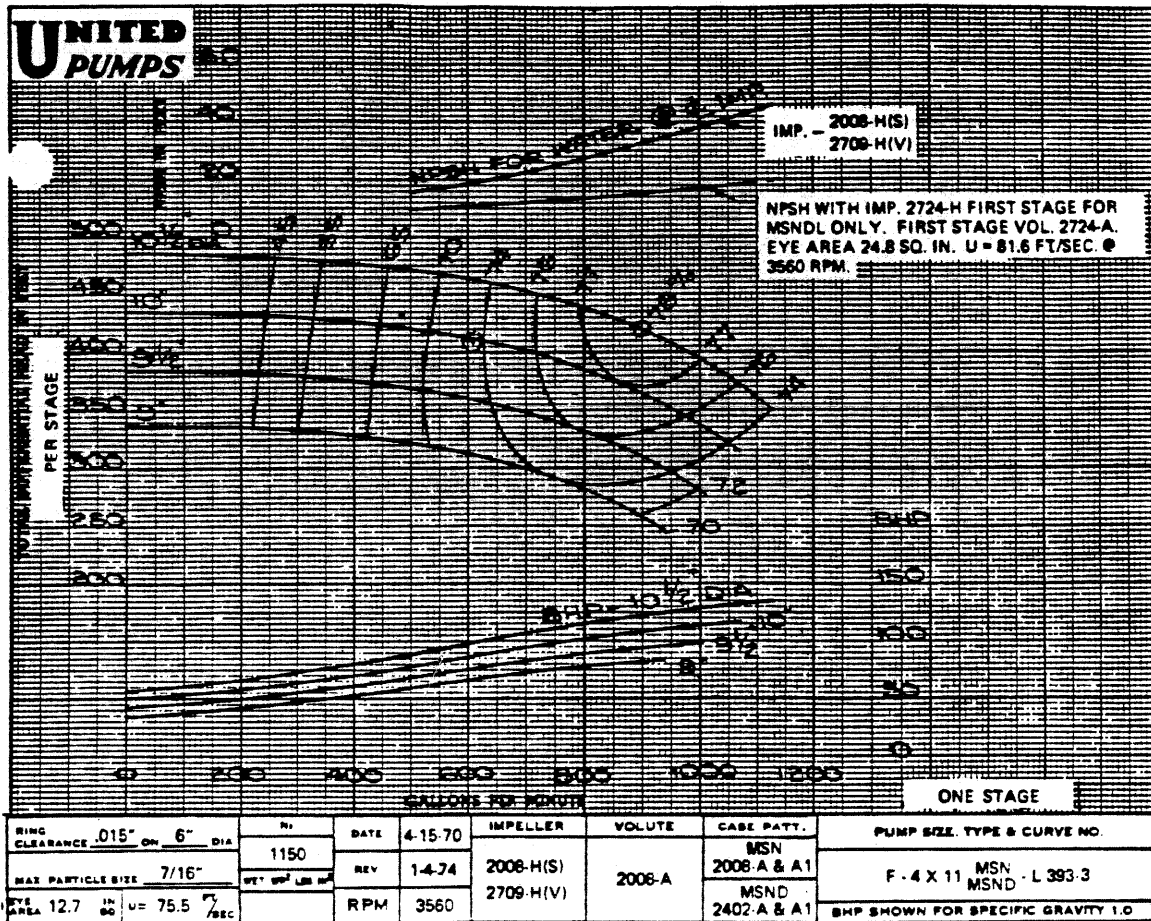
Required Pump Duty:	159 M ³ /hr	Rated Capacity
	145 M ³ /hr	Normal Capacity
	7.03 Kg/cm ² (g)	Suction Pressure
	49.22 Kg/cm ² (g)	Discharge Pressure
	0.95	Specific Gravity
	149°C	Suction Temperature
	Water	Fluid Handled
	4.0 Meters	NPSH Available
	50 Cycle	

1. Make the best pump selection from the attached curves.
2. How many stages?
3. What is the efficiency?
4. What is the horsepower (KW)?
5. What is the required NPSH?

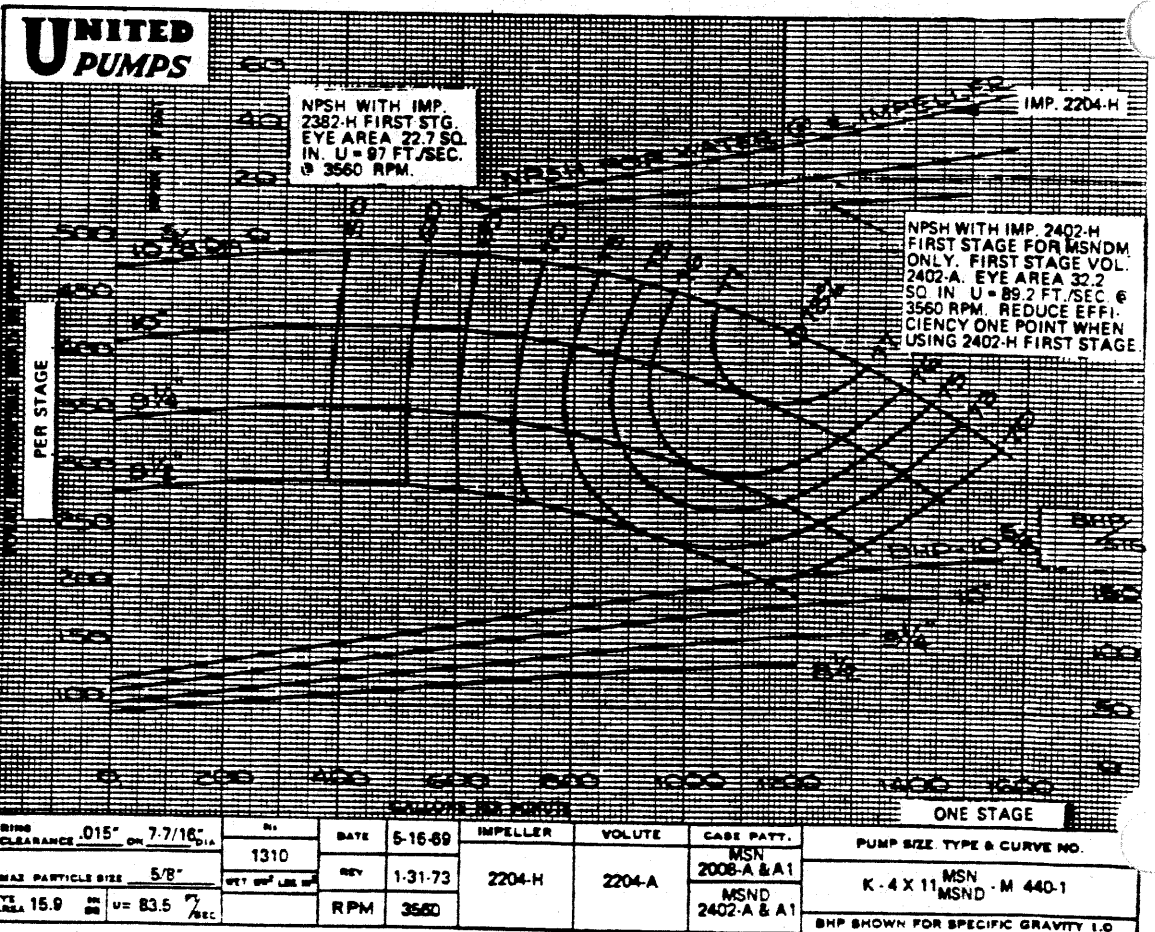
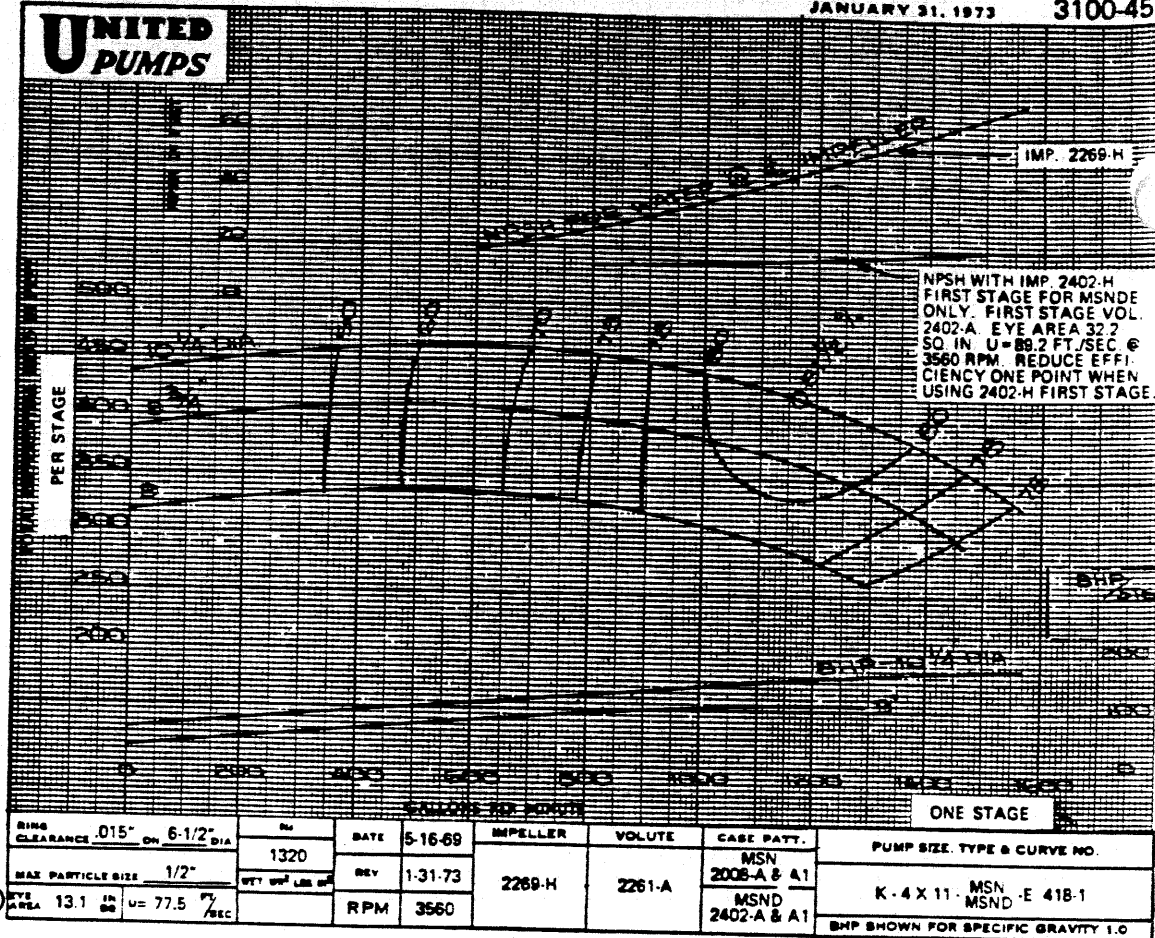
**UNITED
PUMPS**


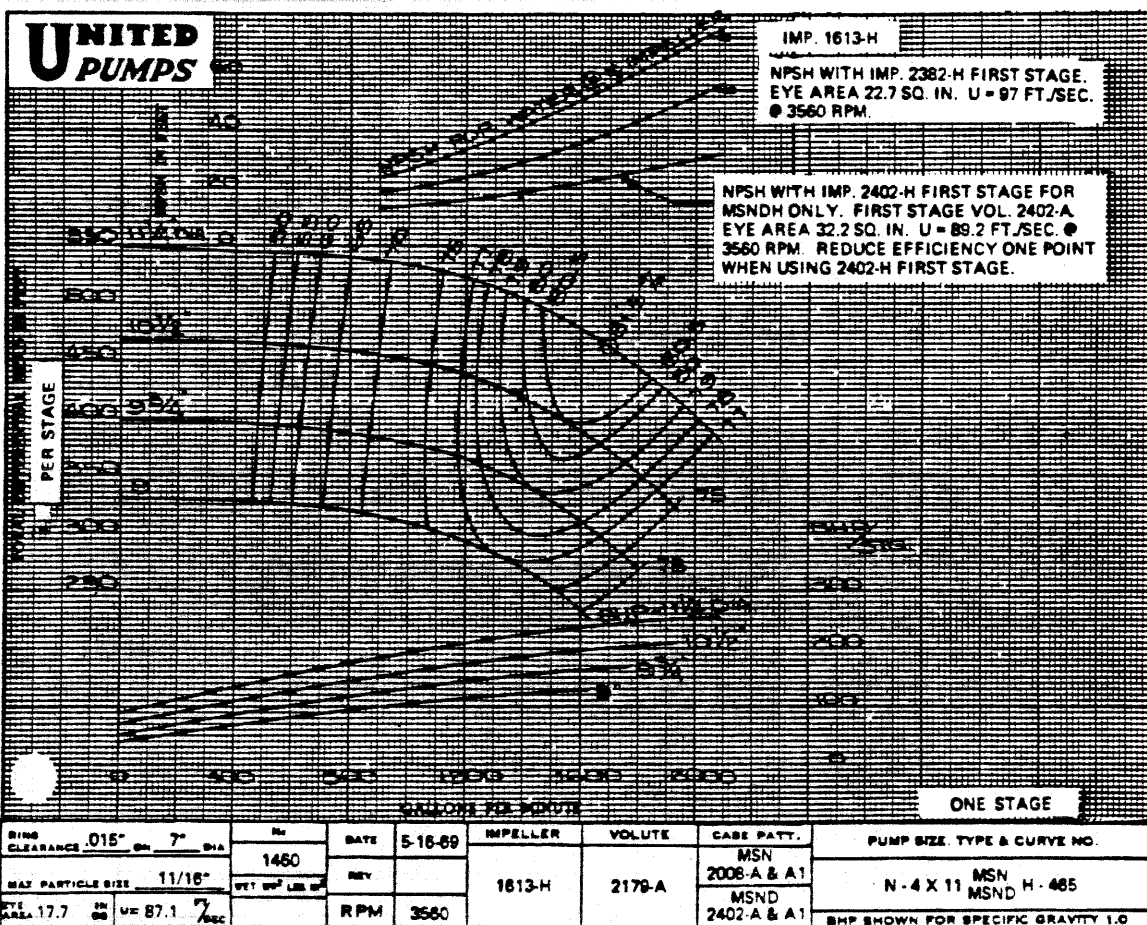
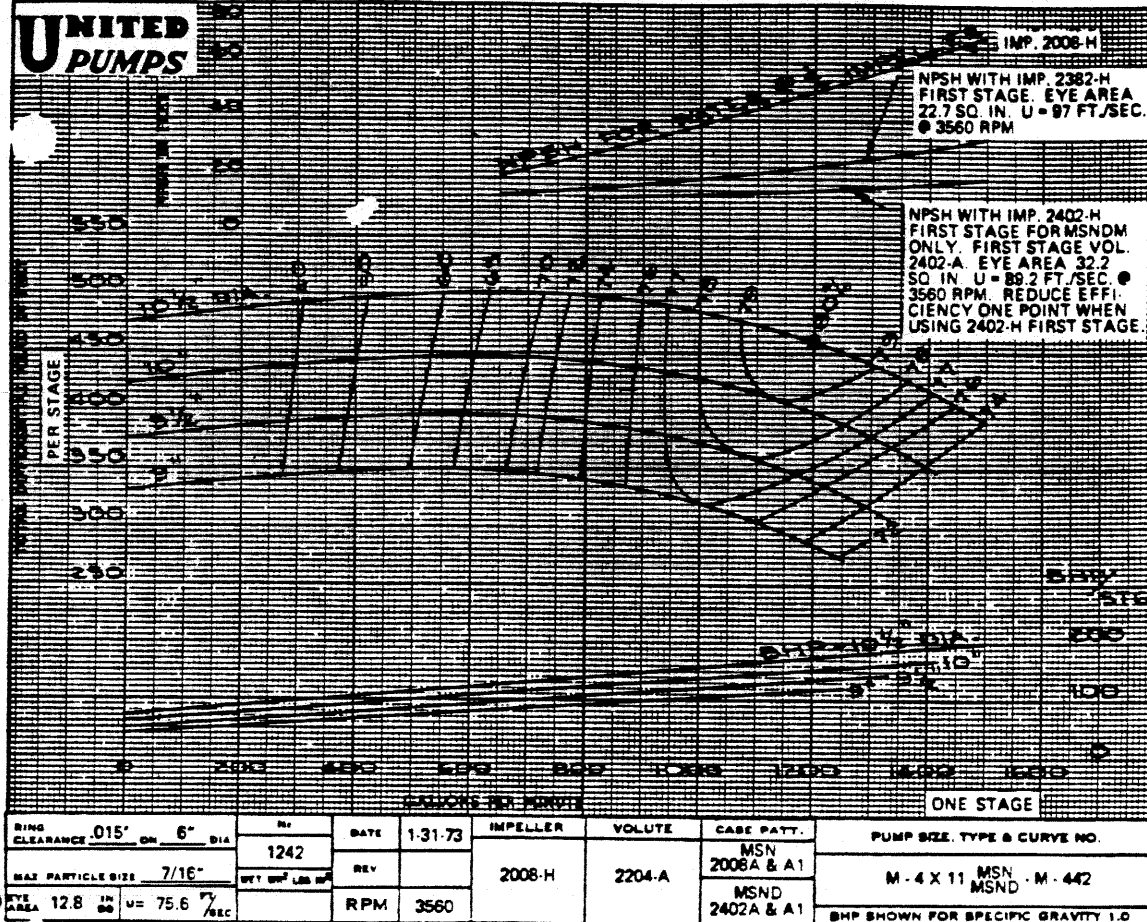


**UNITED
PUMPS****UNITED
PUMPS**

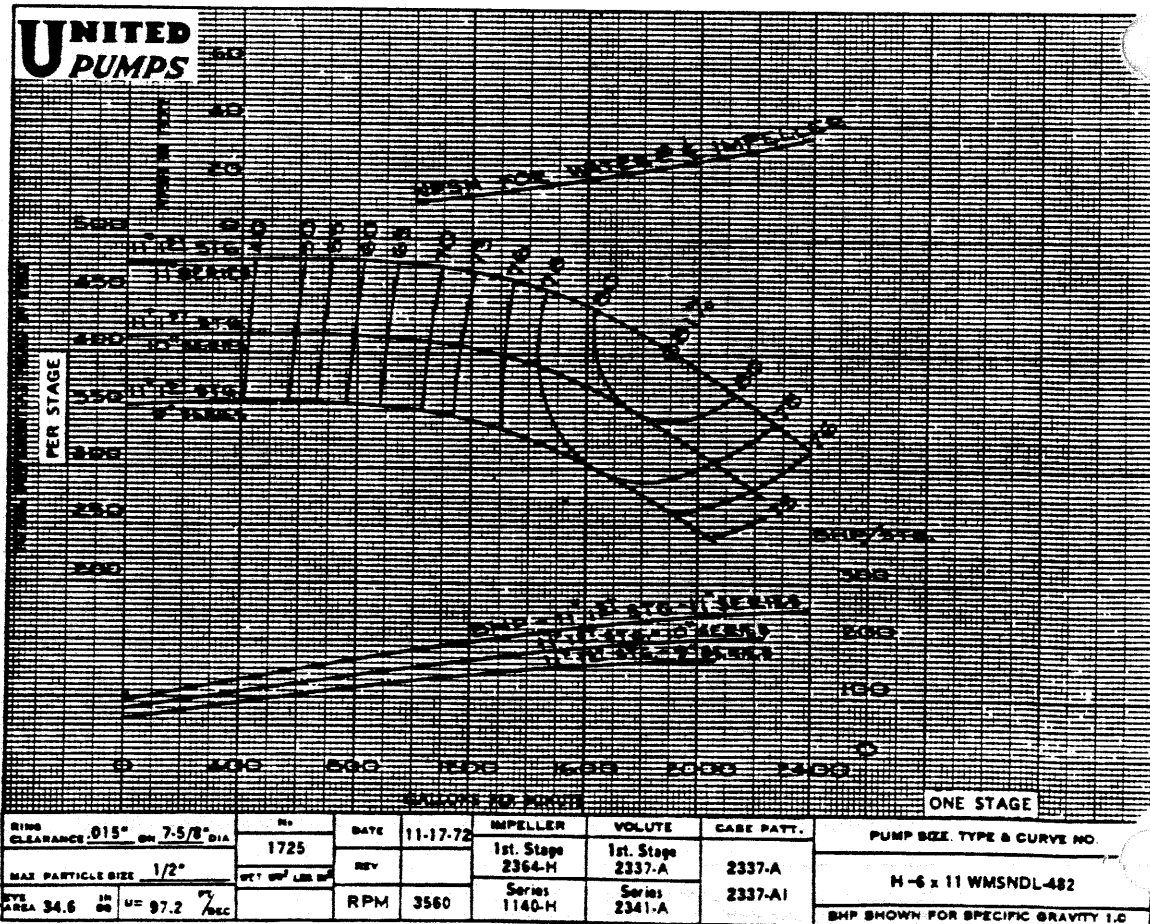
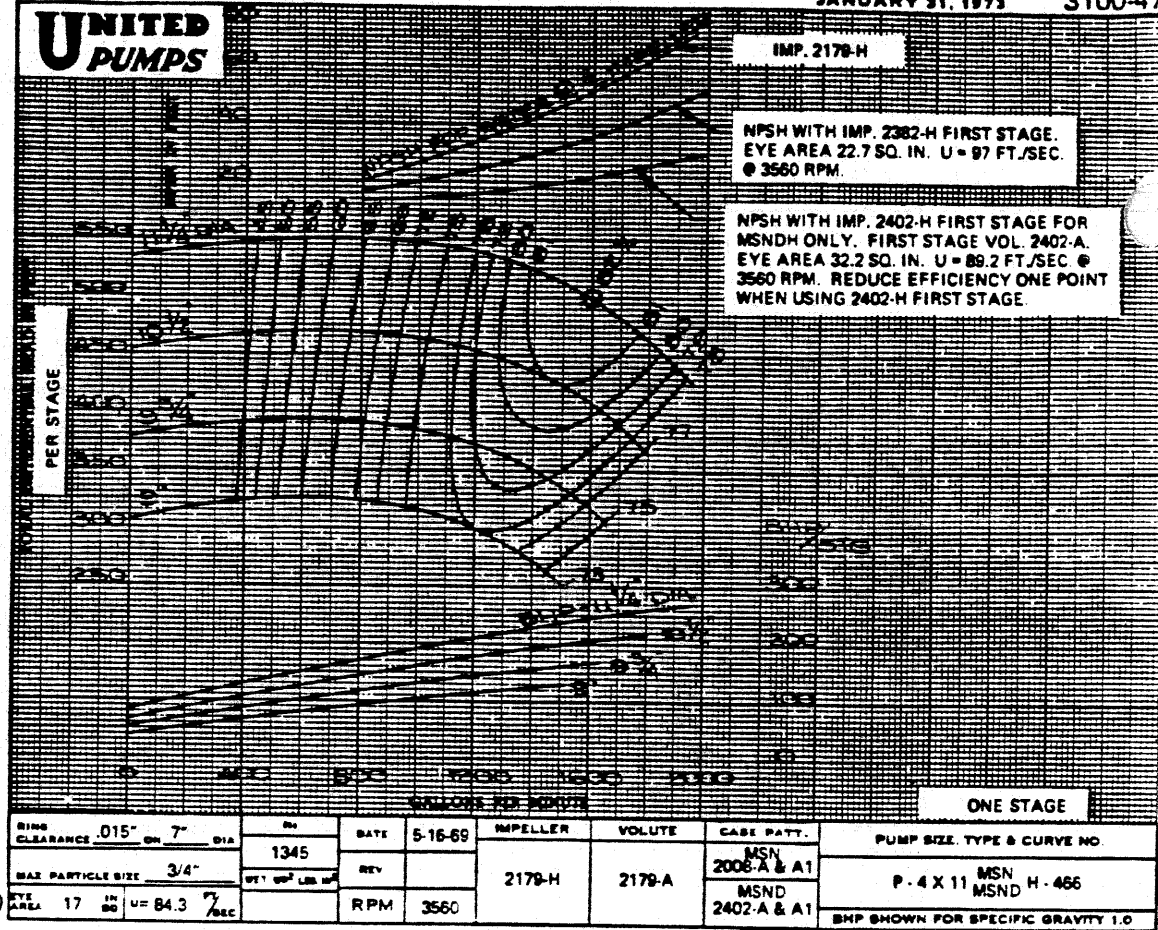


**UNITED
PUMPS**

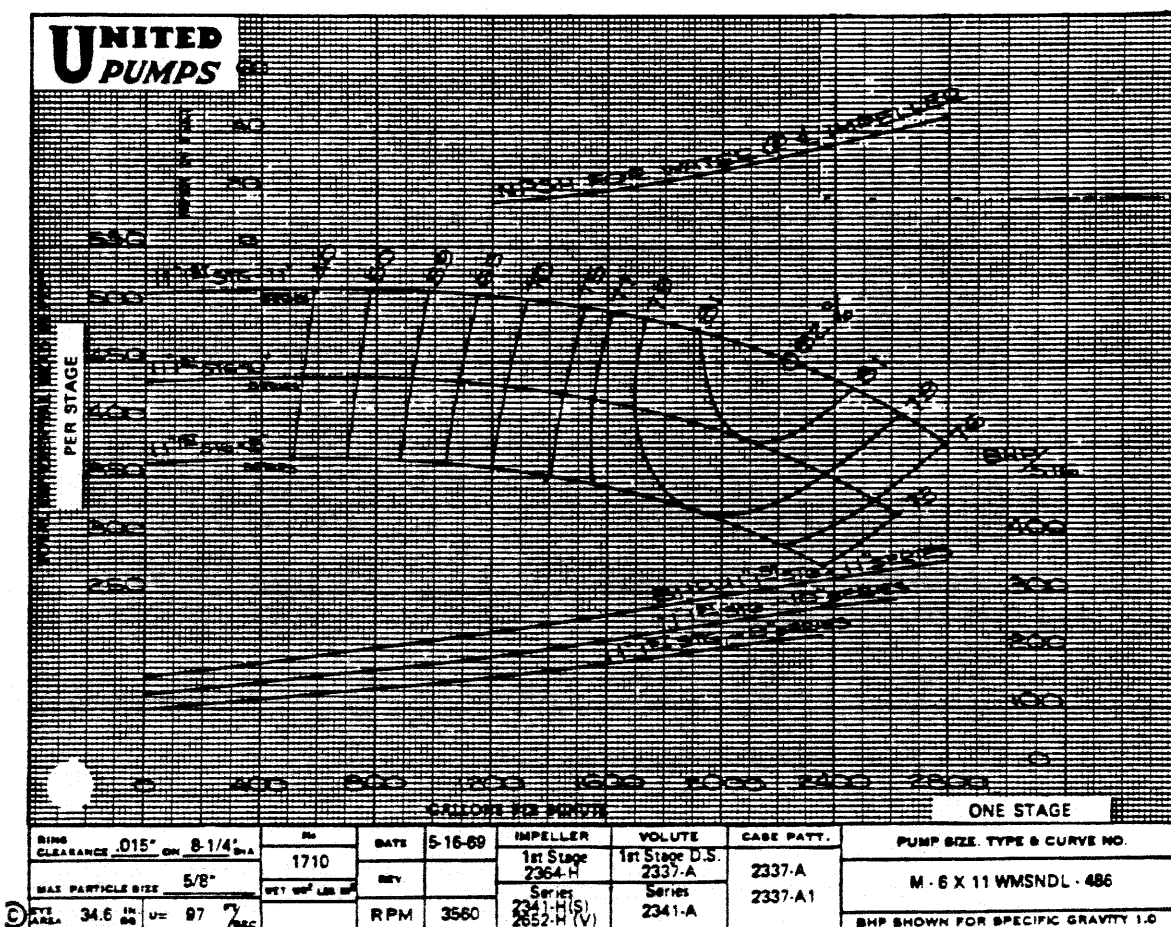
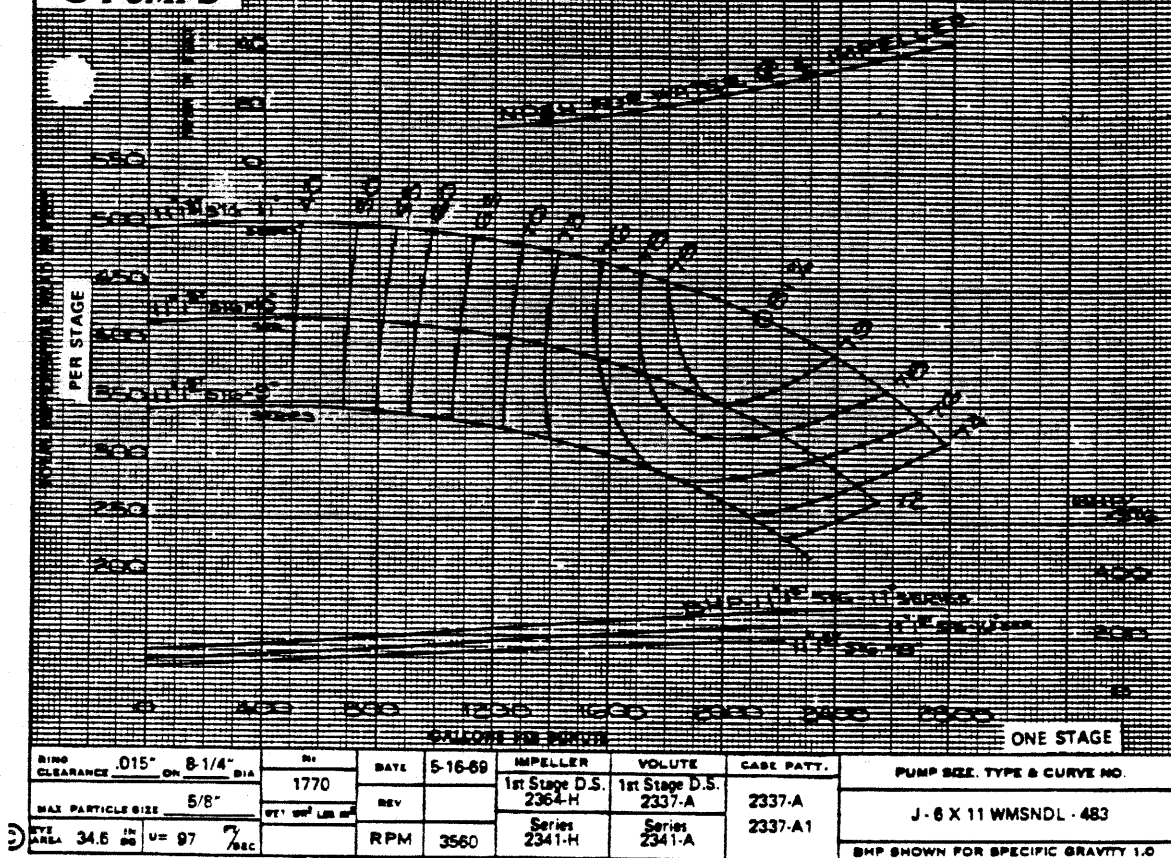




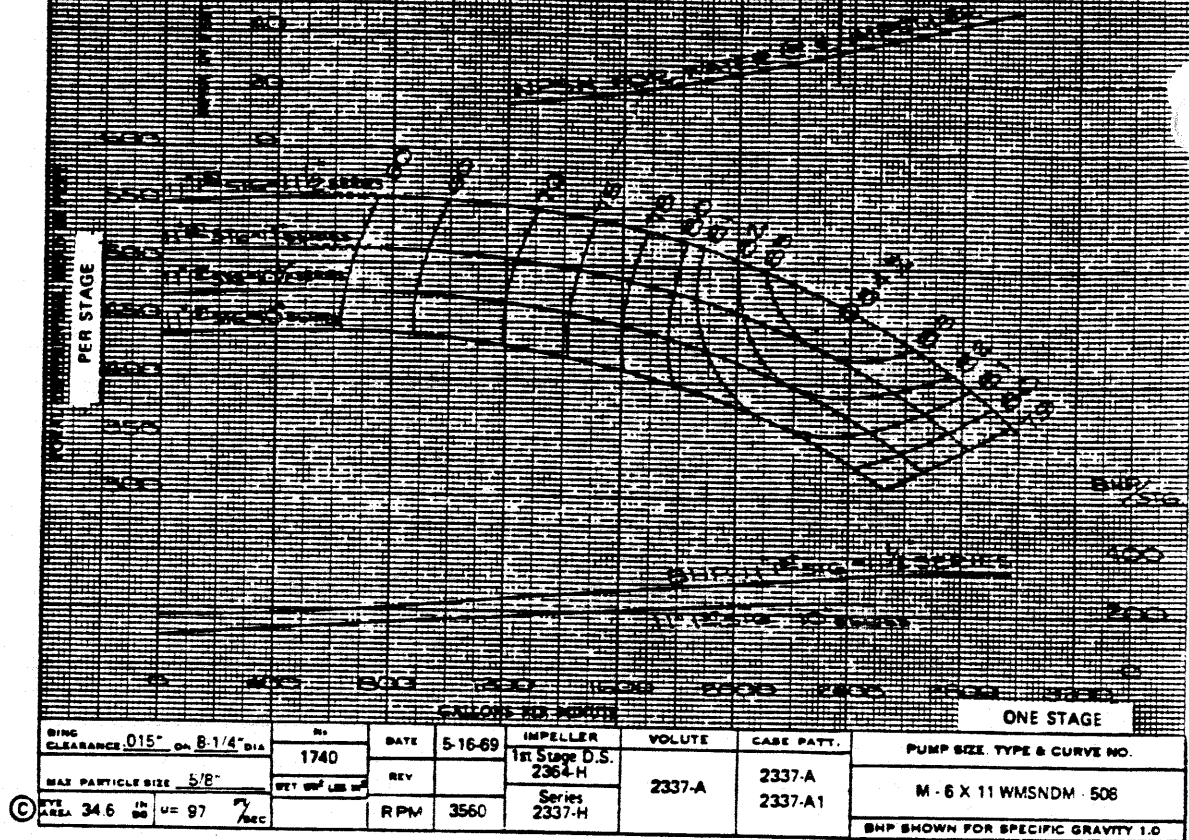
UNITED PUMPS



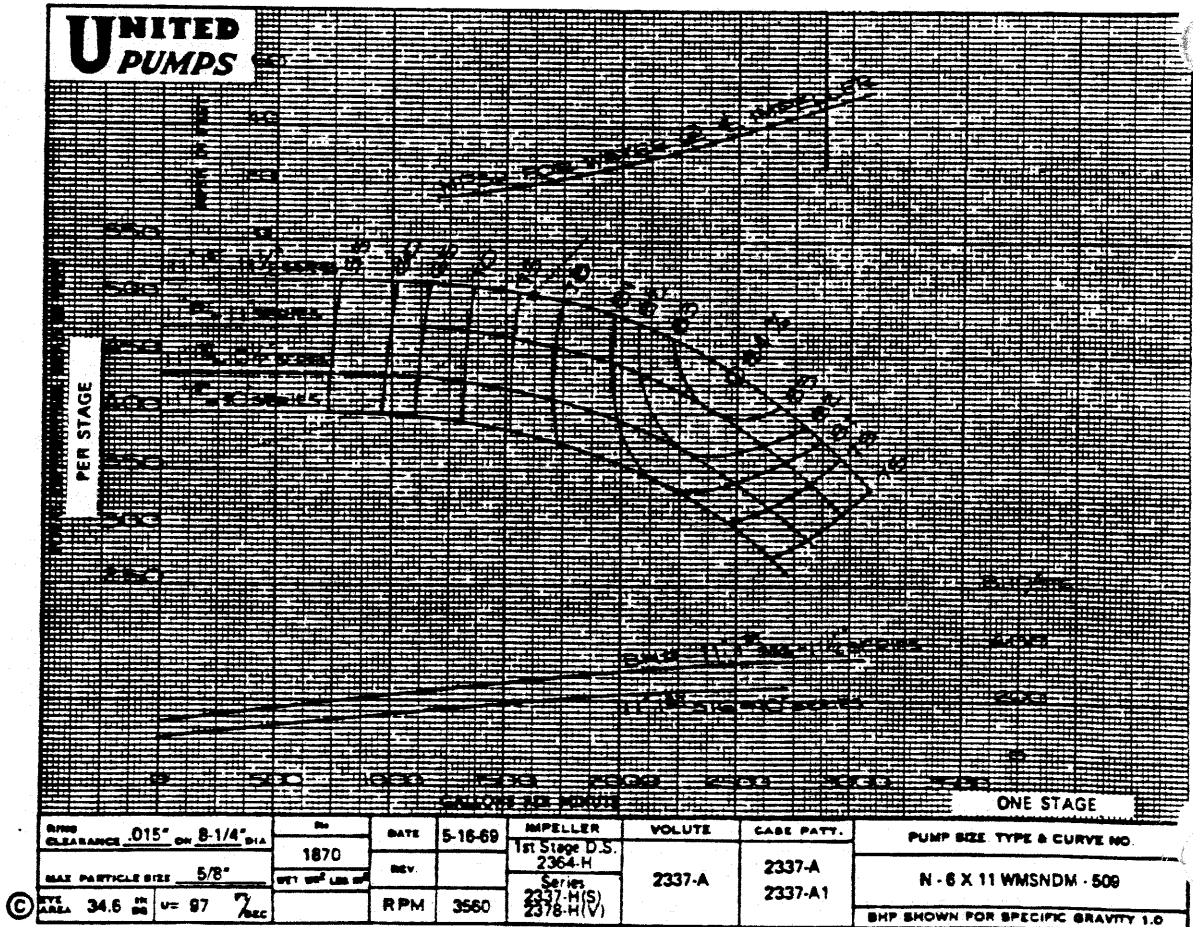
UNITED PUMPS



**UNITED
PUMPS**



**UNITED
PUMPS**



THERMODYNAMIC

- EFFICIENT CONVERSION OF HEAT ENERGY INTO MECHANICAL ENERGY
- HIGH POWER LEVELS *< 0.4-0.5*
- GOOD EFFICIENCY AT OFF-DESIGN POINTS
- STEAM BALANCE SOLUTION
- GOOD PROCESS TEMPERATURE CONTROL VARIABLE (SUBSTITUTE FOR VALVE)

Back pressure steam Turbine

low HP

MECHANICAL

- RELIABLE CENTRIFUGAL ACTION
no shut down
- SPEED/POWER FLEXIBILITY WITH GOVERNOR CONTROL
- GOVERNOR RESPONDS TO PROCESS VARIABLE, HOLDING SIGNAL
- AREA CLASSIFICATION NO PROBLEM
Electrical problems
- USUALLY NO GEARS

(Can not get the idea)

OPERATIONAL

- FEW MOVING PARTS, LOW MAINTENANCE
- LOW VIBRATION LEVELS
- LOW/MODERATE NOISE LEVELS
- RUN THROUGH ELECTRICAL POWER LOSS
- CAN HANDLE OVERLOAD OPERATION
- UPRATE CAPABILITY

Boiler s/b noise ^{11/11} ~~26/04~~ 27 1/2 m.

• Revamp in 2 m.

WHY DO WE USE STEAM
TURBINES IN REFINERIES?

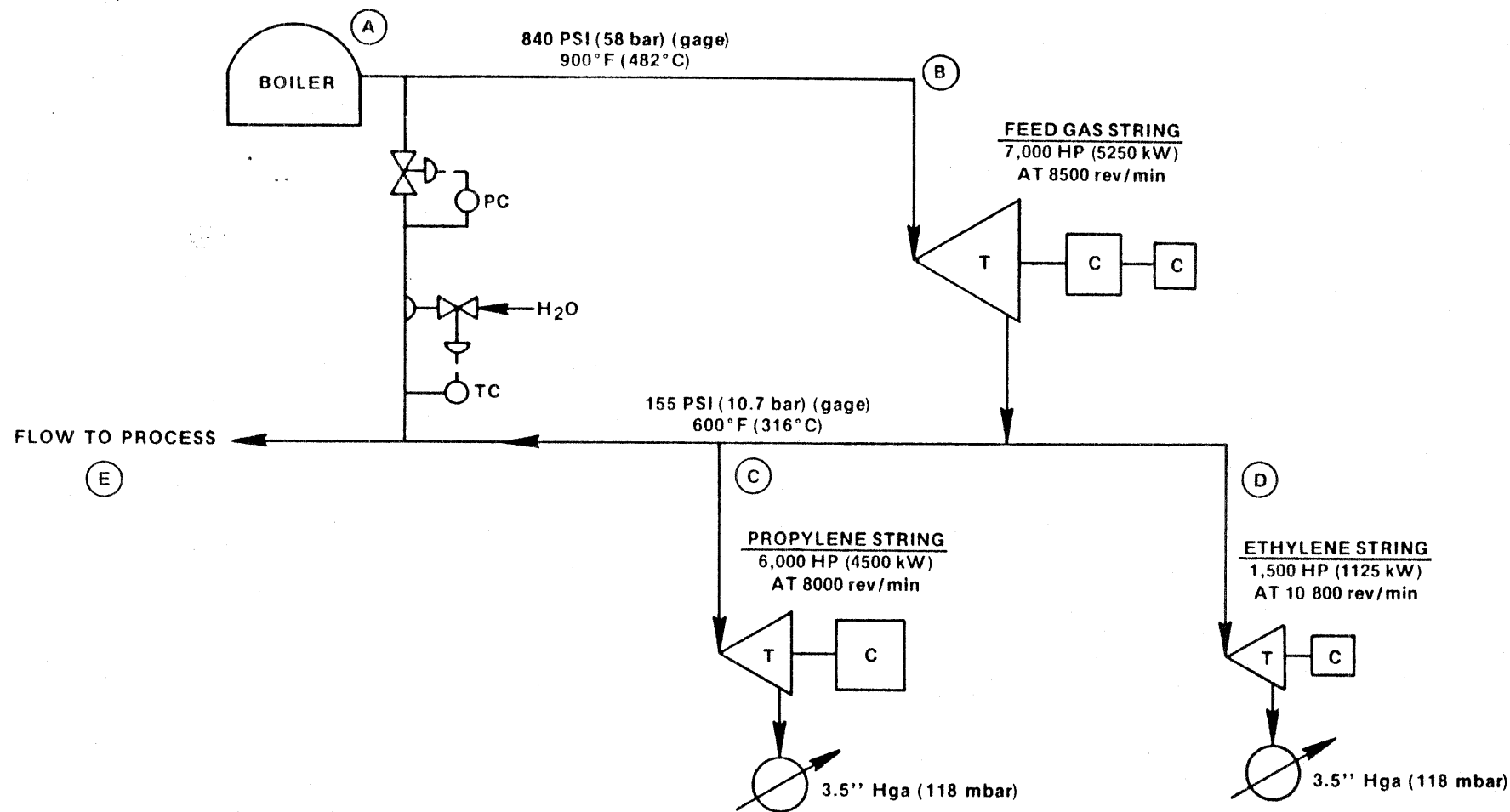
ANSWER:

THERMODYNAMIC ;

MECHANICAL ;

OPERATIONAL ;

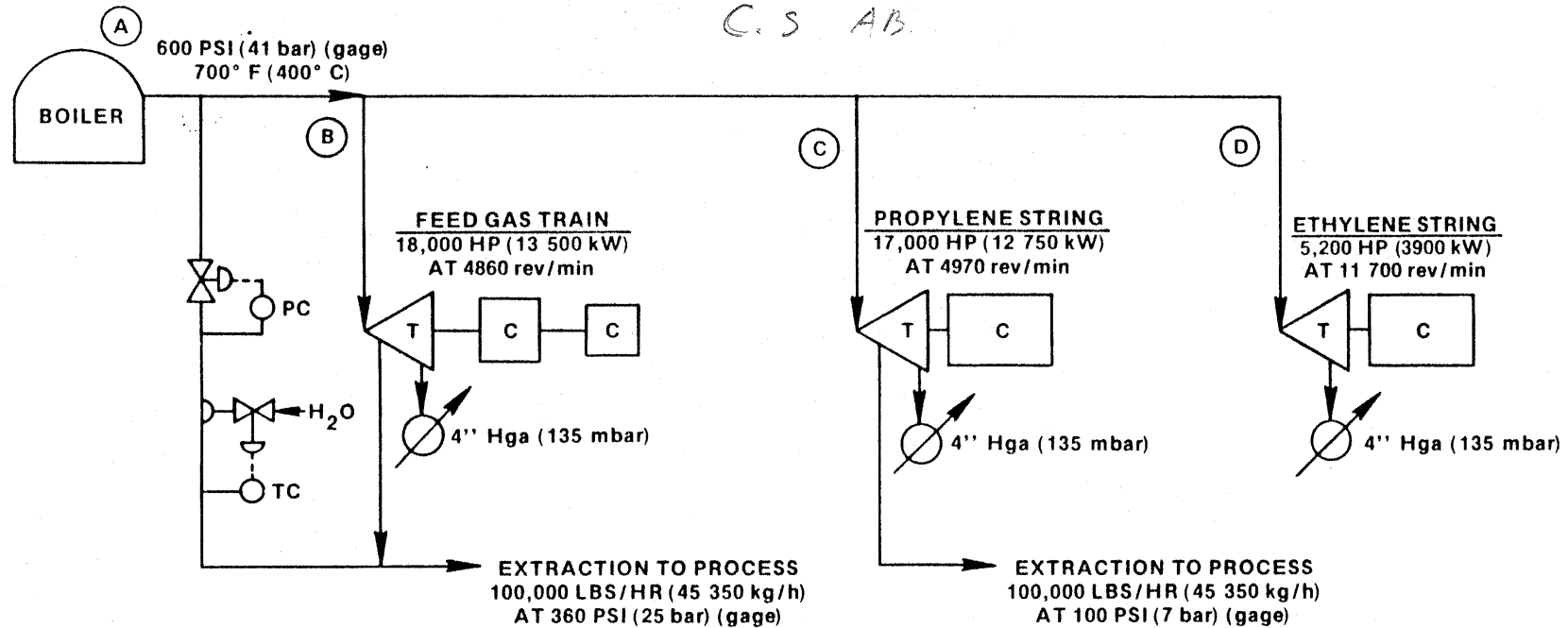
FIGURE 1 ETHYLENE PLANT TYPICAL STEAM BALANCE
200 MILLION POUNDS (90 700 tonnes) PER YEAR



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	B	145,000 LBS/HR (65 750 kg/h)

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

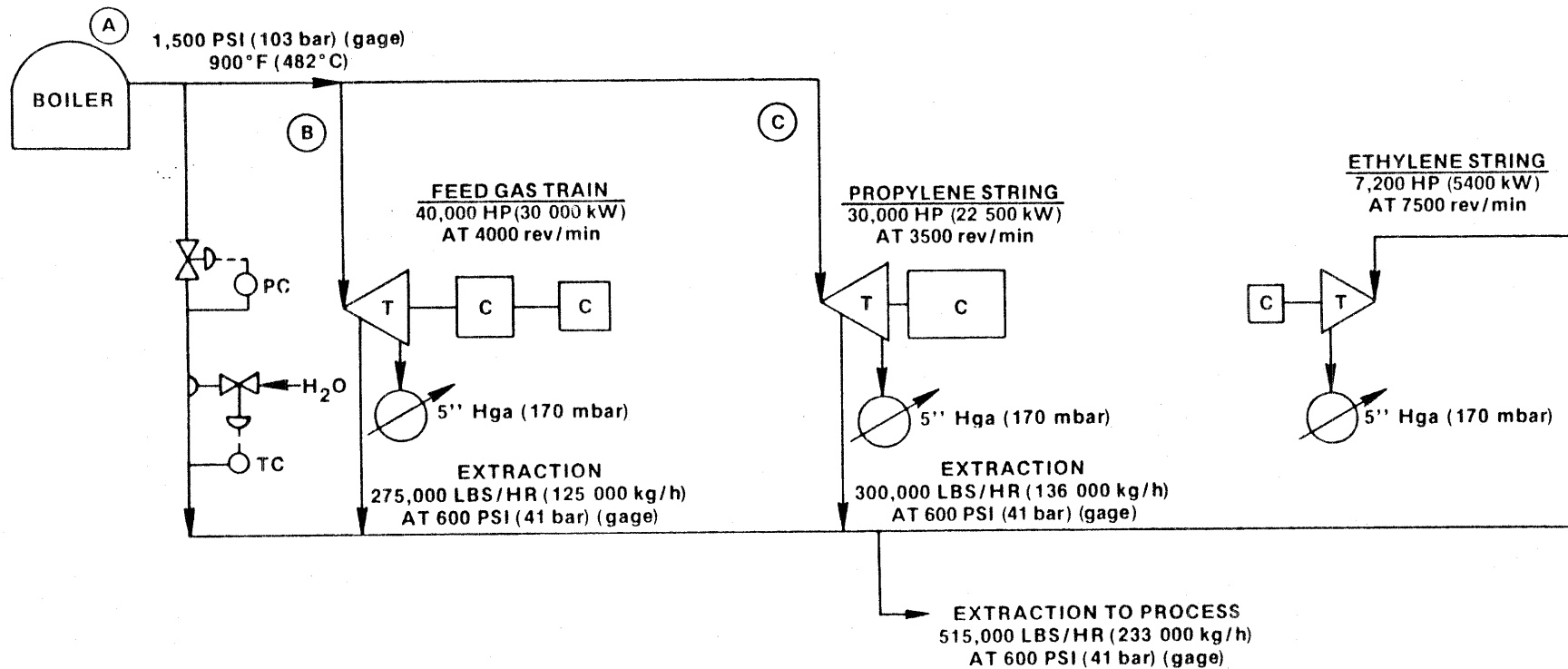


FLOW B 240,000 LBS/HR (108 800 kg/h)
 FLOW C 198,000 LBS/HR (89 800 kg/h)
 FLOW D 41,500 LBS/HR (18 800 kg/h)

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

CS AVAILABLE



TYPES

DEFINED BY CONDITIONS AT INLET AND
EXHAUST, AND INTERMEDIATE PORTS.....

supran

CONDENSING

NON-CONDENSING
(BACK PRESSURE)

EXTRACTION

प्रसृत्योत्पन्न - INDUCTION

CONDENSING

- REQUIRES LEAST AMOUNT OF STEAM
- AIR OR WATER COOLED CONDENSERS
- MORE EXPENSIVE THAN BACK PRESSURE
- MORE CONCERNS THAN BACK PRESSURE
(HEAT UP, COOL DOWN, AIR LEAKS, MOISTURE)
- USUALLY USED FOR COMPRESSOR DRIVES
AND SIZES ABOVE 1000 HP (750 KW)

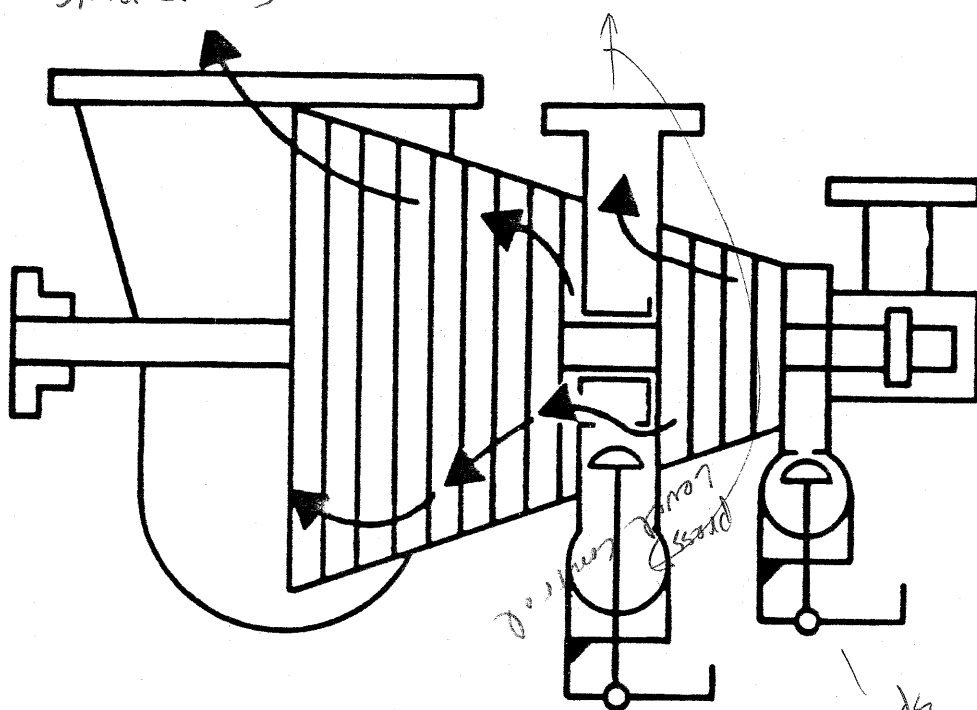
NON-CONDENSING

- STEAM CONSUMPTION IS TWO TIMES
- SMALLER CASING, SHORTER ROTOR
||
52220400
- LESS EXPENSIVE
- COMMON FOR PUMP DRIVERS AND LOWER POWER LEVELS

Automatic start over 1000 rpm.

EXTRACTION/INDUCTION

WHEN AN INTERMEDIATE STEAM PRESSURE
LEVEL IS REQUIRED TO SOLVE A PLANT
STEAM BALANCE - EITHER SHORT TERM OR
CONTINUOUSLY.....



1016 - DESIGN
 2516 - HYDRO
 Relief valves
 Problem
 Pressure high 244

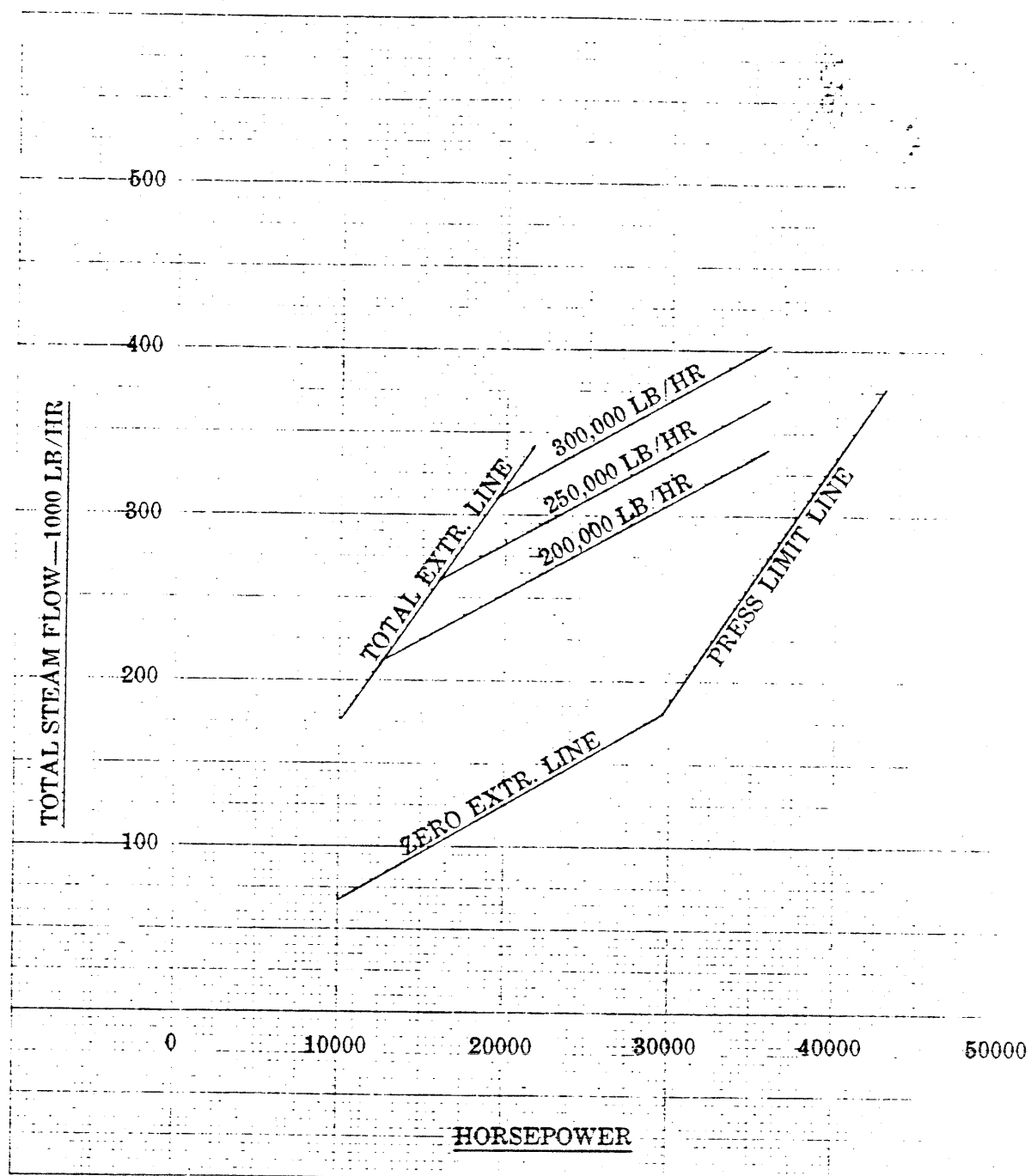
2/1

1/6

Speed Control

Pressure Control

CONDENSING



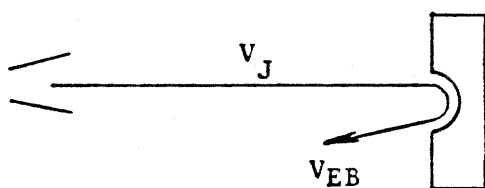
THEORY

A. THERMODYNAMICS

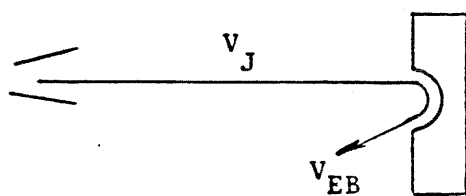
B. STAGE DESIGN

C. EFFICIENCY

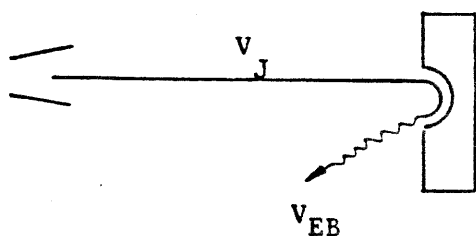
D. VELOCITY LIMITS



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

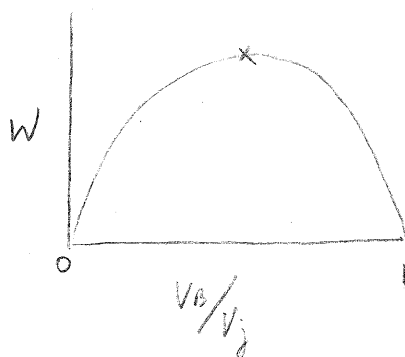


- (b) When bucket moves at $1/4$ jet speed, the steam jet force diminishes. Steam leaves the bucket at low speed but does some work.



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

$$W = F \times D$$



HOW DOES A TURBINE WORK?

HEAT ENERGY IS COLLECTED
FROM THE STEAM AND
CONVERTED INTO SHAFT WORK.

THE ENTROPY INCREASES DUE
TO STEAM FRICTION AND
THROTTLING LOSSES.

STEP 1: HEAT ENERGY CONVERTED INTO
VELOCITY ENERGY

STEP 2: VELOCITY IMPACTS MOVING
BLADES, IMPARTING ENERGY

CONVERSION OF HEAT ENERGY INTO VELOCITY ENERGY, V_j

FOR ADIABATIC EXPANSION ACROSS A
NOZZLE:

$$H_{in} = (H_{out} + V_j^2/2g(778))$$

steam velocity.

$$V_j = (2g(778)(H_{in} - H_{out}))^{1/2} \quad (778 \text{ ft lbs/Btu})$$

$$= 223.7(\Delta H_{is})^{1/2}$$

LESSON:

JET VELOCITY IS DIRECTLY PROPORTIONAL
TO THE ENERGY AVAILABLE.

CONVERSION OF VELOCITY ENERGY INTO WORK.....

TWO TYPES OF STAGES

- IMPULSE
- REACTION

IMPULSE TYPE: *USA type*

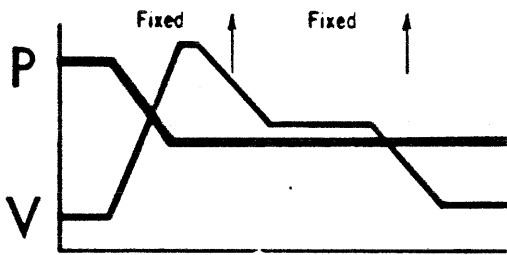
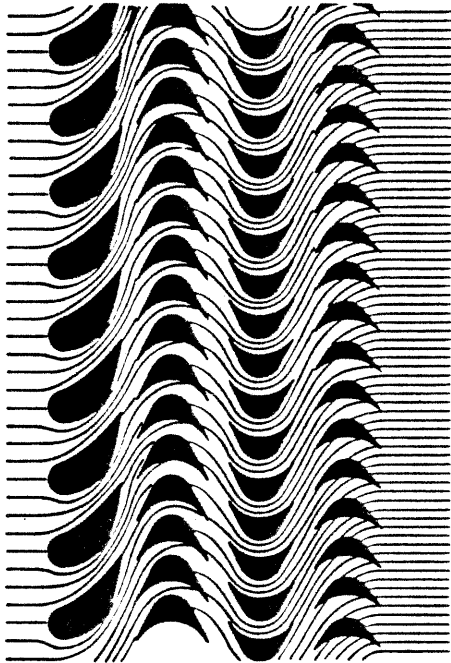
PRESSURE DROP TAKEN ACROSS NOZZLES ONLY

REACTION TYPE: *Europ type*

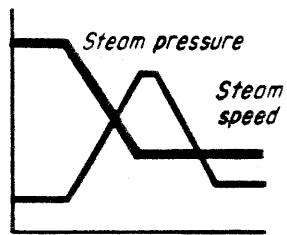
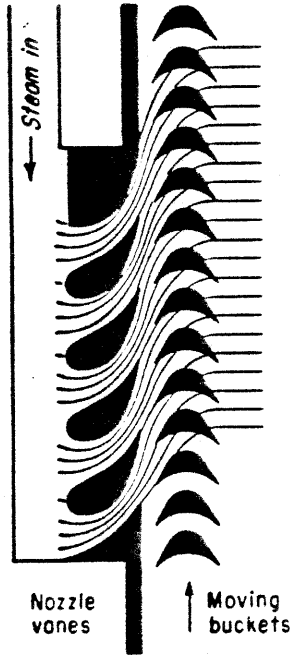
PRESSURE DROP DISTRIBUTED ACROSS NOZZLE
AND BLADES

THREE CLASSES OF STAGING USING THE TWO TYPES OF BLADES:

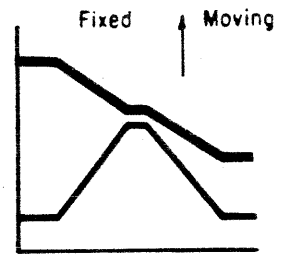
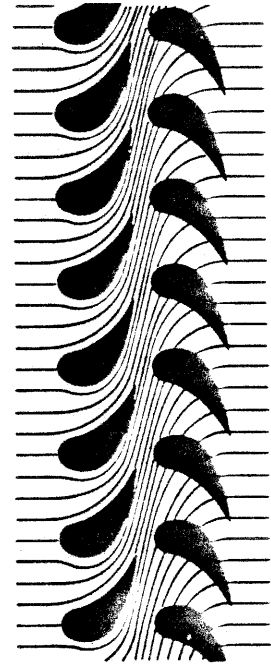
- 1. VELOCITY COMPOUNDED (CURTIS) STAGING**
- 2. PRESSURE COMPOUNDED (RATEAU) STAGING**
- 3. REACTION STAGING**



A
Velocity-compounded
(Curtis) stage and
reaction stages



B
Pressure-compounded
(Rateau) stage



C
Reaction stage

Typical

CHARACTERISTICS OF IMPULSE DESIGN.....

1. LARGE BLADE CLEARANCES

Thrust transfer is good

2. SMALL ΔP ACROSS BLADES

3. LOW THRUST BEARING LOAD

CHARACTERISTICS OF REACTION DESIGN.....

1. ΔP ACROSS NOZZLES AND BLADES, HIGHER PRESSURE IN FRONT PART OF CASING
2. SEALING AT END OF BLADES
3. HIGHER EFFICIENCY-ABOUT 5%
4. TWO TIMES NO. OF STAGES
5. LARGER THRUST BEARING AND/OR BALANCING PISTON

ADVANTAGES OF IMPULSE TYPE

- LARGER CLEARANCES
- FEWER STAGES
- MORE TOLERANT TO UPSET
- MORE TOLERANT TO
MOISTURE/DIRT
- SIMPLER MAINTENANCE
- LESS SOPHISTICATED, MORE
RUGGED DESIGN

ADVANTAGES OF REACTION TYPE

- HIGHER EFFICIENCY

LESSON:

USUAL CHOICE FOR REFINERY IS
IMPULSE TYPE

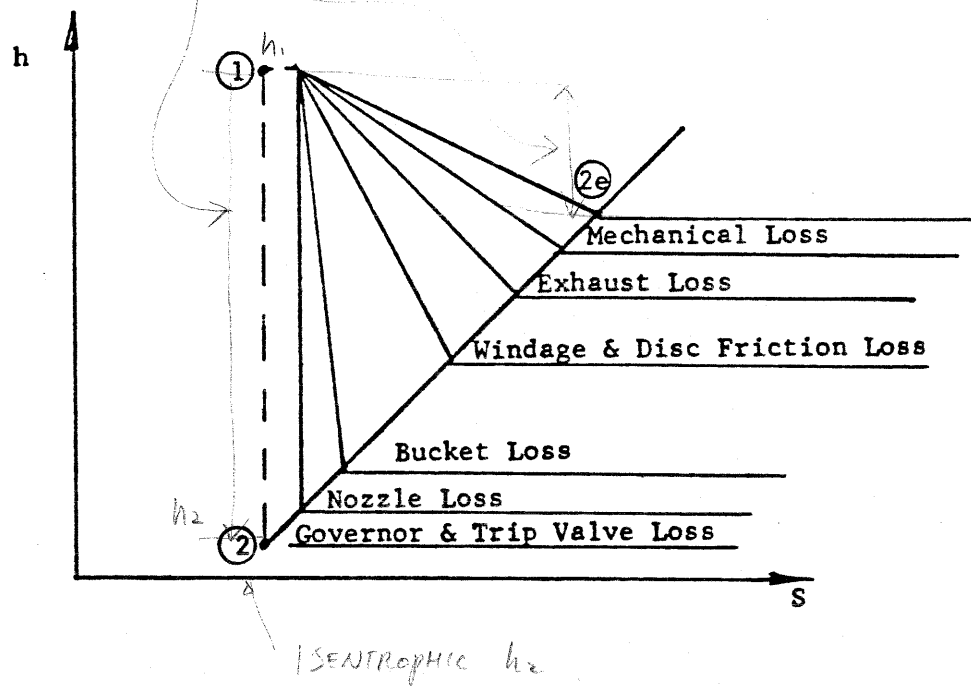
Problem 이걸다.

EFFICIENCY

LOSSES:

- STEAM FRICTION
- STEAM THROTTLING
- MECHANICAL LOSSES
- LEAKAGE *Condensing Type loss*

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



EFFICIENCY

AVAILABLE ENERGY: ISENTROPIC ΔH

IDEAL RANKINE CYCLE EFFICIENCY

$$= (H_1 - H_2)_{IS} / (H_1 - H_W)$$

is for work

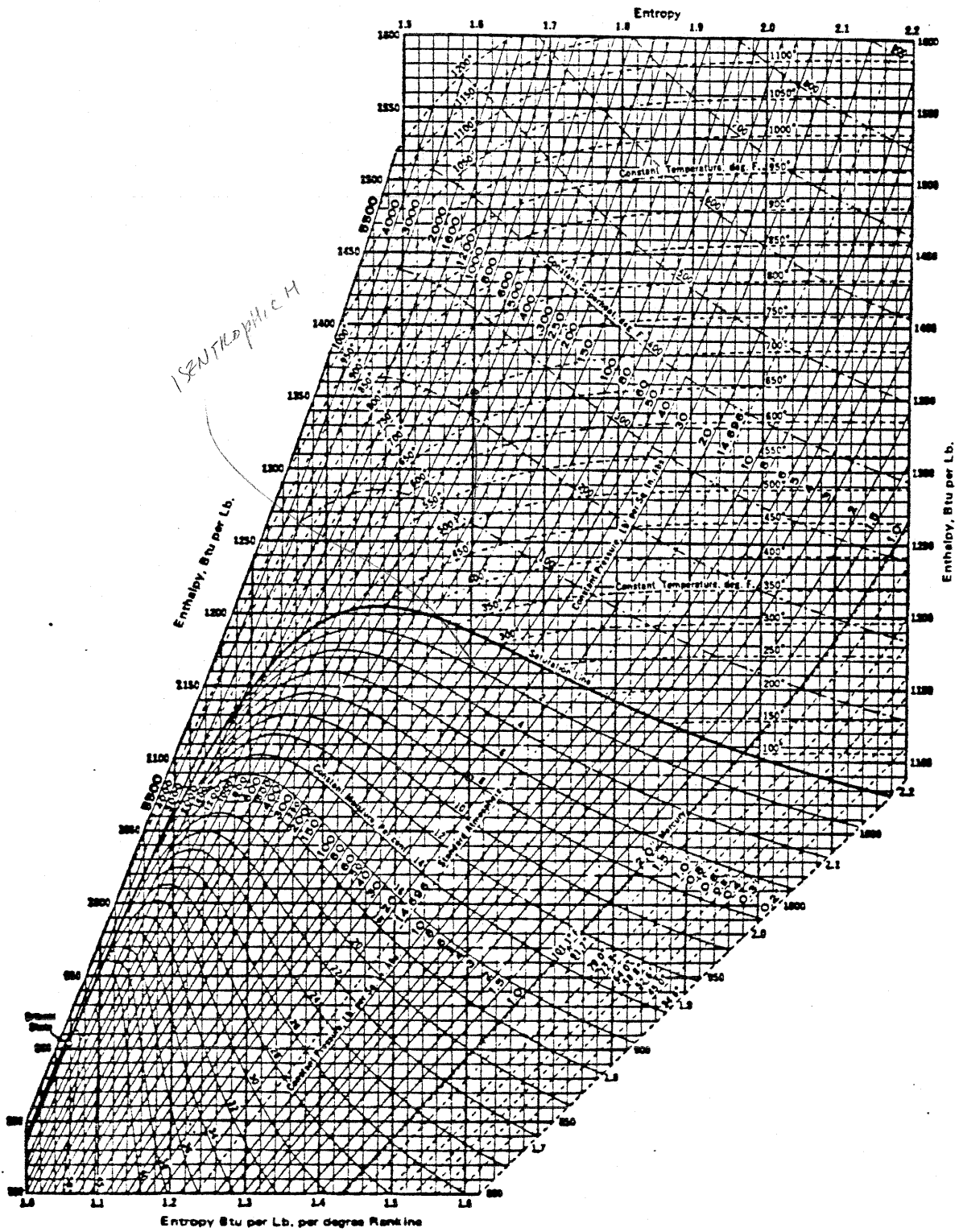
INTERNAL TURBINE EFFICIENCY

$$= (H_1 - H_2) / \Delta H_{IS}$$

is for actual

LESSON:

EFFICIENCY IS BASED ON A RELATIONSHIP BETWEEN BLADE SPEED AND JET VELOCITY.



Exhaust Pressure	125 #G 500' FTT 147' FS	150 #G 500' FTT 134' FS	200 #G 388' FTT 0' FS	200 #G 500' FTT 112' FS	250 #G 406' FTT 0' FS	250 #G 500' FTT 94' FS	400 #G 448' FTT 0' FS	400 #G 750' FTT 302' FS	600 #G 575' FTT 86' FS	600 #G 750' FTT 261' FS	800 #G 825' FTT 336' FS	850 #G 750' FTT 223' FS	850 #G 825' FTT 298' FS	850 #G 900' FTT 373' FS	1250 #G 825' FTT 251' FS	1250 #G 900' FTT 329' FS	1500 #G 825' FTT 228' FS	1500 #G 900' FTT 302' FS	1800 #G 1800' FTT 378' FS
3 in HgA	10.57	10.28	10.62	9.87	10.22	9.59	9.48	7.70	8.34	7.40	7.05	7.20	6.85	6.53	6.69	6.36	6.64	6.30	5.86
4 in HgA	11.10	10.78	11.12	10.31	10.66	10.00	9.85	7.99	8.64	7.65	7.28	7.42	7.06	6.73	6.88	6.54	6.83	6.48	6.01
5 in HgA	11.57	11.21	11.54	10.70	11.05	10.35	10.17	8.22	8.89	7.86	7.48	7.61	7.24	6.89	7.04	6.69	6.98	6.62	6.14
0 #G	18.55	17.50	17.51	16.08	16.30	15.16	14.26	11.19	11.96	10.40	9.82	9.87	9.32	8.81	8.90	8.40	8.76	8.25	7.54
10 #G	23.18	21.48	21.10	19.28	19.31	17.90	16.43	12.71	13.49	11.64	10.96	10.94	10.30	9.71	9.75	9.18	9.56	8.99	8.16
20 #G	27.96	25.47	24.56	22.34	22.12	20.44	18.34	14.02	14.79	12.68	11.90	11.82	11.10	10.43	10.43	9.80	10.20	9.56	8.64
30 #G	33.3	29.73	28.11	25.46	24.91	22.95	20.15	15.23	15.98	13.62	12.76	12.60	11.80	11.07	11.03	10.34	10.75	10.07	9.06
50 #G	47.2	39.9	35.99	32.3	30.8	28.21	23.67	17.56	18.19	15.36	14.31	14.01	13.07	12.21	12.07	11.28	11.72	10.94	9.77
60 #G	57.2	46.4	40.55	36.2	34.0	31.07	25.45	18.74	19.26	16.18	15.06	14.66	13.66	12.74	12.55	11.71	12.16	11.34	10.08
75 #G	79.1	59.1	48.5	42.9	39.4	35.8	28.21	20.57	20.85	17.40	16.17	15.61	14.51	13.51	13.22	12.32	12.77	12.85	10.53
100 #G	—	96.6	66.6	58.1	50.1	45.21	33.14	23.86	23.53	19.43	18.07	17.14	15.88	14.77	14.28	13.26	13.74	12.75	11.21
150 #G	—	—	—	—	—	—	45.21	31.93	29.24	23.83	22.15	20.15	18.62	17.33	16.28	15.05	15.52	14.33	12.44
250 #G	—	—	—	—	—	—	—	—	43.97	35.40	32.91	26.84	24.80	23.08	20.22	18.67	18.89	17.37	14.78
400 #G	—	—	—	—	—	—	—	—	—	—	—	—	—	—	27.05	24.99	24.30	22.32	18.39

Isentropic OH