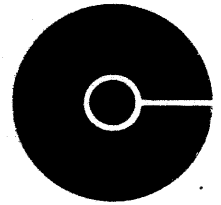
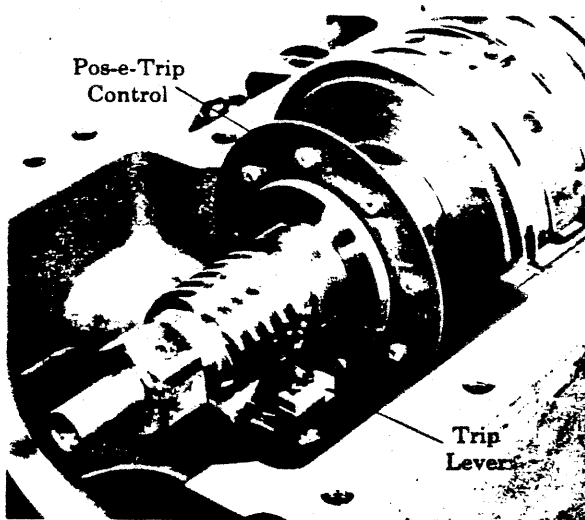


SPECIFICATIONS



POS-E-TRIP* CONTROL

*U. S. Pat. 2973771



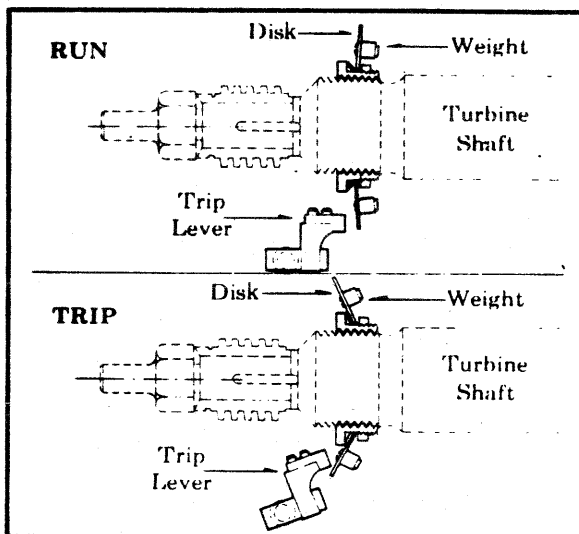
Pos-e-Trip is a positive and precise overspeed trip mechanism. It will trip consistently within 1 10 of 1 percent of set speed. Introduced in 1956, Pos-e-Trip is now used on all Elliott high-speed turbines. Pos-e-Trip is patented by, and is exclusive with, Elliott Company.

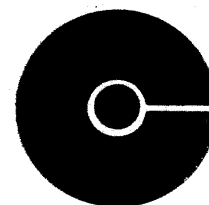
Operating Principle: Heart of the Pos-e-Trip is the speed-sensing device. Instead of the conventional spring-opposed bolt within the turbine shaft, a dished disk is mounted on the shaft. A series of weights are located around the circumference of the disk and the assembly is inherently in dynamic balance. At trip speed, centrifugal forces cause the disk to "snap" to the tripped position. The patented "floating pivot" design permits essentially frictionless snap action.

Each Pos-e-Trip mechanism is laboratory calibrated at the factory for correct trip speed. No further adjustment is required either at field start-up or during the normal life of the turbine. And proper calibration cannot be tampered with by operators, thus assuring that your turbine is being operated within safe speed limits.

When trip speed is reached and the Pos-e-Trip disk snaps to the "tripped" position, a trip lever is struck. Pivoting of the trip lever releases a piston in the control oil circuit, which instantaneously causes the pressure in the control oil circuit to decay, thus closing the trip valve.

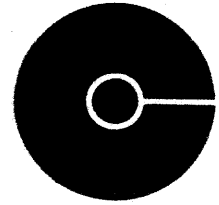
The turbine can be tripped manually at any speed by simply turning the trip handle. After the turbine has slowed down to a speed below running speed, the trip mechanism can be reset by pulling the trip handle outward.





MULTIVALVE TURBINE MATERIALS OF CONSTRUCTION

Part	Material	Commercial Specifications
Steam Chest and Casing		
600 psi—750°F/41 bar—399°C	Cast carbon steel	ASTM A-216 Grade WCB
600 psi—825°F/41 bar—440°C	Carbon-molybdenum steel	ASTM A-217 Grade WC1
900 psi—900°F/62 bar—482°C	Chromium-molybdenum steel	ASTM A-217 Grade WC6
2000 psi—950°F/138 bar—510°C	Chromium-molybdenum steel	ASTM A-217 Grade WC9
Exhaust Casing		
Condensing and		
Non-Condensing (Cast)	High strength cast iron	ASTM A-278 Class 40
Non-Condensing (Cast)	Cast steel	ASTM A-216 Grade WCB
Fabricated	Steel	ASTM SA 285 Grade C
Nozzles	12% Chromium-stainless steel	AISI-405
Diafram Centers		
Fabricated	Steel	ASTM SA 285 Grade C
Cast	High strength cast iron	ASTM A-278 Class 40
Disks		
Forged	Chromium-nickel-molybdenum steel	AISI 4340
Cross-Rolled Plate	Constructional alloy steel	USS T-1
Integral with Shaft	Chromium-nickel-molybdenum-vanadium steel	ASTM A-470 Class 4, 7 or 8
Blades	12% Chromium-stainless steel	AISI Type 403
Shroud Bands	12% Chromium-stainless steel	AISI Type 410
Damping Wire	15% Chromium steel	Inconel X750
Shaft		
Built-up	Chromium-molybdenum steel	AISI 4140
Integral	Chromium-nickel-molybdenum-vanadium steel	ASTM A-470 Class 4, 7 or 8
Bearing Shells	Steel	ASTM SA 285 Grade C
Bearing Liners	Bonded tin-base babbitt	ASTM B-23 Alloy #7
Labyrinth Seals		
Shaft Sleeves up to 750°F/399°C	Carbon steel	ASTM A-179
751° to 875°F/400° to 468°C	Nickel-chromium-molybdenum-steel	AISI 4340
Stationary Baffles	Chromium-molybdenum-steel forgings	AISI 4140
Governor Valves	12% Chromium-stainless steel	AISI Type 410
Governor Valve Stems and Seals	12% Chromium-stainless steel, nitrided	AISI Type 416
Governor Valve Seats	12% Chromium-stainless steel	AISI Type 416
Bar Lift Rods and Bushings	12% Chromium-stainless steel, nitrided	AISI Type 416
Steam Strainer Screen	Stainless steel	AISI Type 321
Bearing Housings	Ductile iron	ASTM A-536 Grade 60-45-12



BAR-OPERATED GOVERNOR VALVES

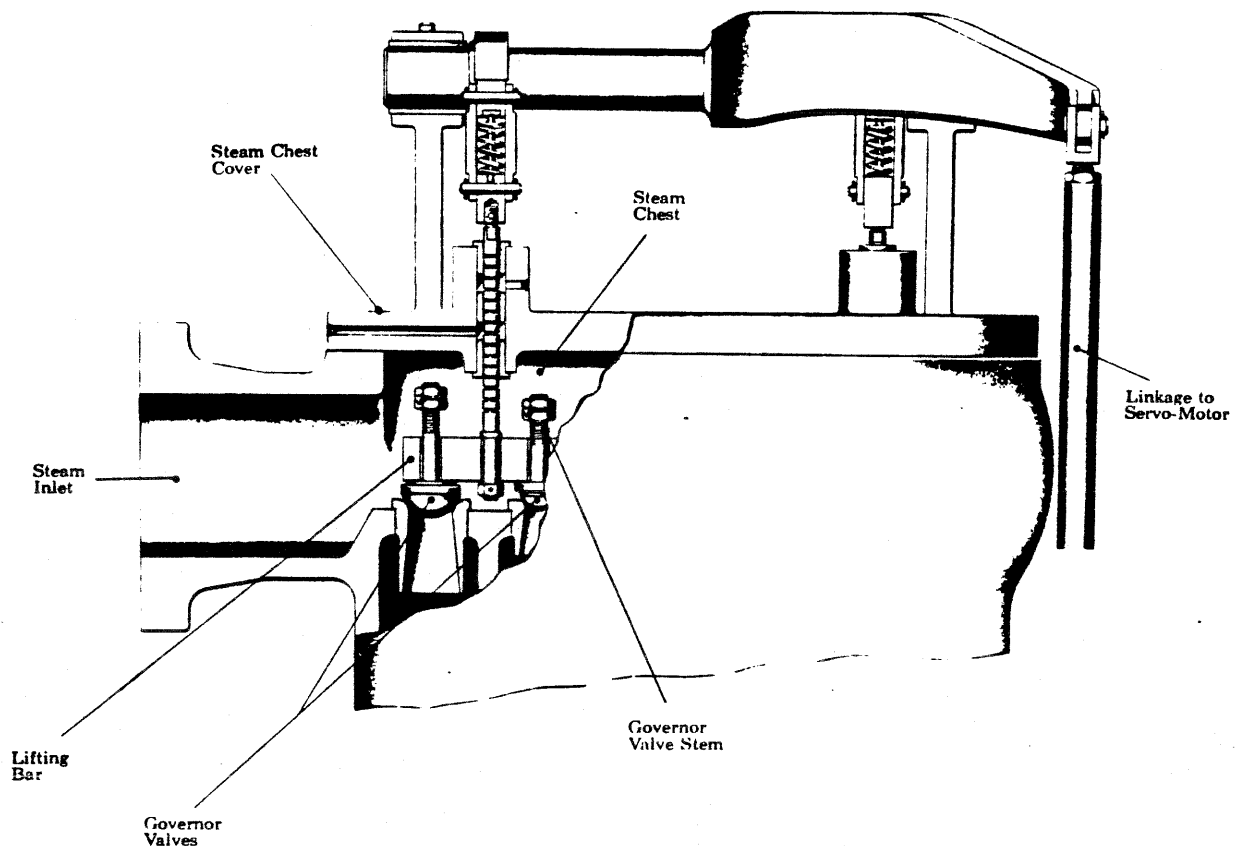
A multivalve governing arrangement is provided in which the governor-controlled valves open and close sequentially with increasing and decreasing load. All valves are located in the top half of the casing with valve stems set at different lengths in the lifting bar. Thus, as load increases, the valves open in succession, admitting live steam from the steam chest to separate groups of first stage nozzles. Each succeeding valve is not opened until ring pressure on the preceding nozzle group has increased to approximately throttle pressure.

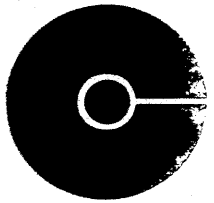
Each valve is of the single-seated venturi-type with carefully ground seats shrunk and tack-welded to bored openings in the steam chest. Valves and/or

valve seats are stellited when required.

Valve motion is positively controlled by a large hydraulic servo-motor acting through the rigid bar lift arrangement. The servo motor is mounted on the lower half turbine casing. The substantial valve-operating forces are, therefore, not transmitted to the foundation through the bearing housing-steam end attachment. Instead, these forces are transmitted directly to the bar lift assembly and contained entirely within the turbine casing.

The two nitrided bar-lifting stems are sealed against leakage by labyrinth grooves and leak-off connections to low-pressure stages of the turbine.





SPEED CONTROL FOR STRAIGHT CONDENSING AND STRAIGHT NONCONDENSING TURBINES

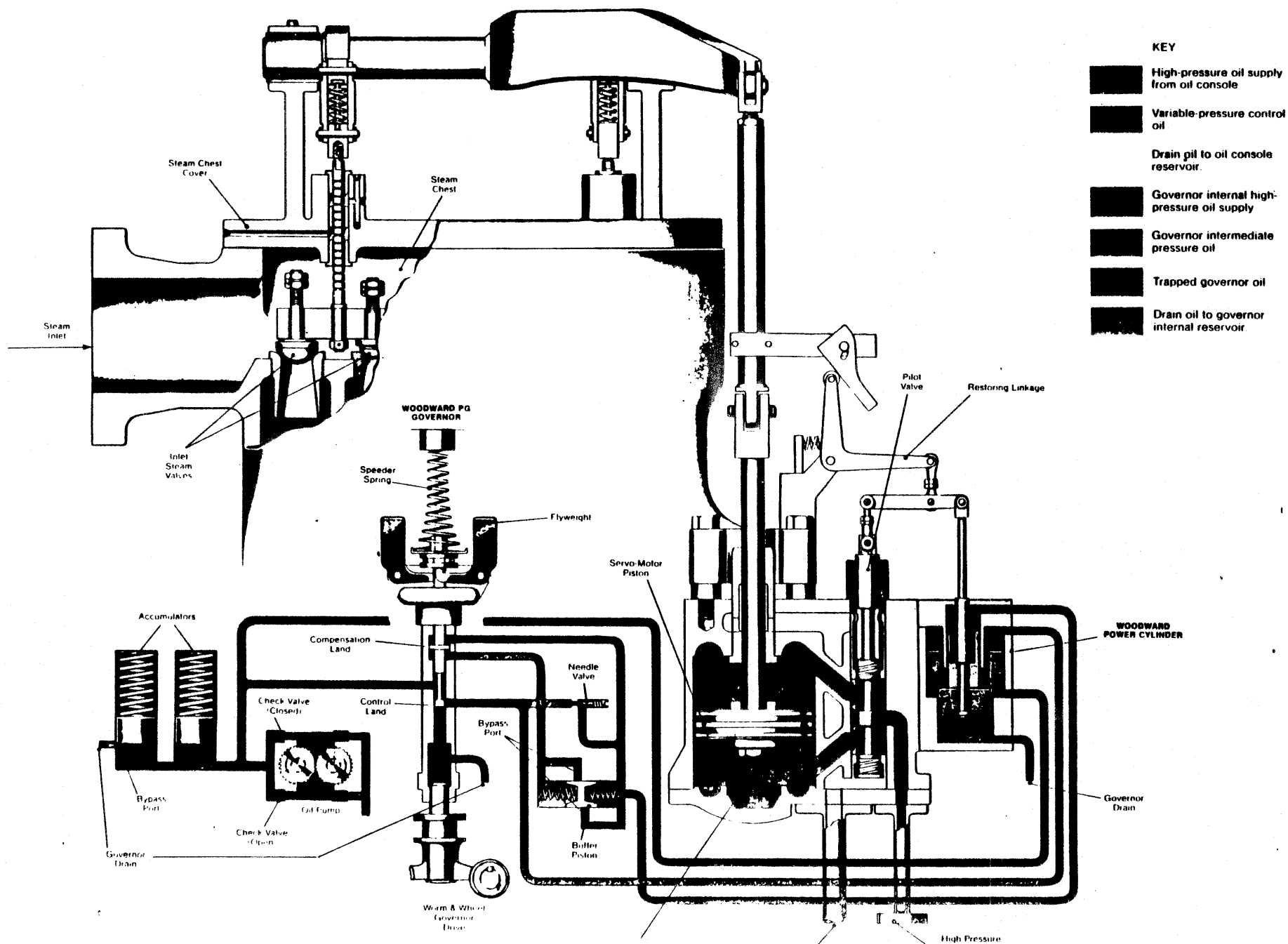
The speed of Elliott multivalve mechanical drive steam turbines is controlled by a Woodward PG Series oil relay governor. This governor is well suited for the variable speed applications found in the process industries. It conforms to NEMA D standards and has won worldwide acceptance because of its unequalled performance and reliability.

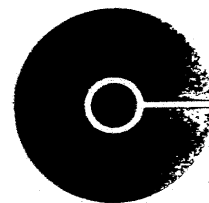
The governor is completely self-contained and features built-in flyweights, oil pump, accumulators, and a fully compensated oil relay system. Accessories such as remote pneumatic or electric speed setting control, weatherproof governor construction and shutdown features are readily available.

To generate the forces required to position the

multiple steam inlet valves, an Elliott servo-motor is required. The Woodward power cylinder is mounted to this servo and acts as a relay. The Woodward power cylinder positions the pilot valve which directs the flow of high pressure oil from the lube system to the servo motor piston. The restoring linkage re-centers the pilot valve after the servo motor piston reaches its new position and the set speed is satisfied.

Alternate control systems are available to meet other process applications. For example, a safety feature is available that closes the steam inlet valves when the turbine is shut down. Elliott has also supplied the Woodward electric governor (Type EG) and other specialized systems engineered to the customer's requirements.





AUTOMATIC EXTRACTION PRESSURE CONTROL

Steam at constant pressure for process use can be supplied from a steam turbine by combining a speed and a pressure control system. The combined speed and pressure control system, called automatic extraction control, is illustrated inside the fold.

The speed control system consists of a Woodward PG Series oil relay governor. This governor is well suited for the variable speed applications found in the process industries. It conforms to NEMA D standards and has won worldwide acceptance because of its unequalled performance and reliability. The governor is completely self-contained and features built-in flyweights, oil pump, accumulators and a fully compensated oil relay system. Accessories such as remote pneumatic or electric speed setting control, weatherproof construction, and shutdown features are readily available.

The extraction pressure regulator is the heart of the pressure control system. It maintains the desired extraction pressure and turbine speed by combining inputs from these variables and sending corrective signals to the servo-motors controlling the inlet and extraction steam valves.

The sketches below show, in simplified form, how the Elliott automatic extraction control system responds to changes in turbine speed and to variations in extraction steam demand. Assume for a moment that the extraction steam demand is constant but the load on the turbine is reduced slightly so that it momentarily begins to speed up. The

Woodward governor will respond and call for reduced steam flow to the turbine (Figure 1). The three-arm linkage is designed to reduce the steam flow equally through both the steam inlet valves and the extraction valves. Thus, the extraction flow remains unchanged while both the inlet and exhaust flows are reduced. This reduced steam flow through the turbine reduces the steam torque on the rotor and the turbine slows to its set speed.

Now assume that the load on the turbine is constant, but there is a change in the process and the demand for extraction steam decreases. This will cause an increase in the extraction pressure and compress the bellows in the extraction regulator which will rotate the three-arm linkage as shown in Figure 2. This action simultaneously closes the inlet valves and opens the extraction valves. The inlet and extraction flows are thus reduced while the exhaust flow is increased. The speed is maintained at its set point because the three-arm linkage is proportioned to maintain a constant steam torque on the rotor during these changes in extraction flow.

In actual operation both speed and extraction steam demands are constantly changing and both actions described are occurring simultaneously.

Elliott turbines are also available for automatic induction. Turbines of this type are the opposite of automatic extraction designs in that excess process steam is admitted into the turbine and expanded to the exhaust pressure. Units designed to extract and induct are also available.

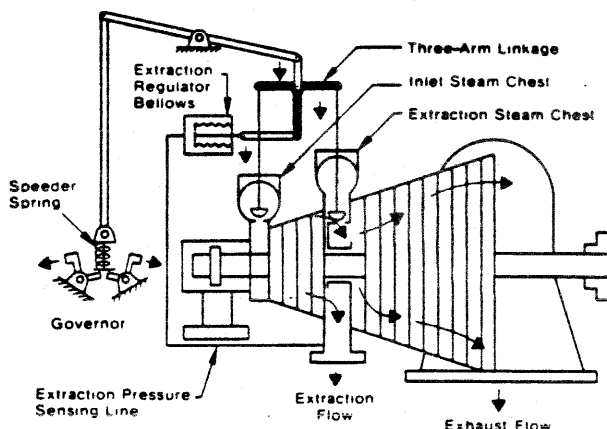


Figure 1

Turbine speed changes; extraction steam demand is constant

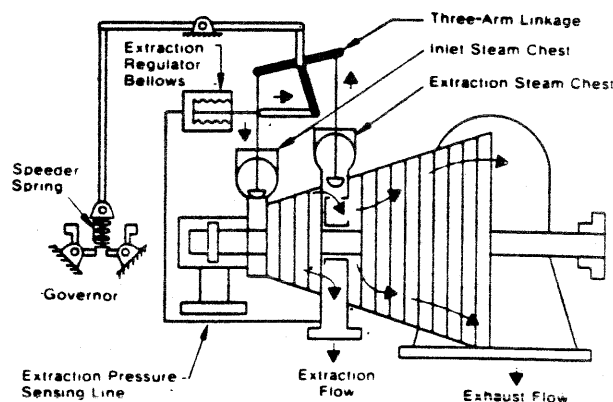
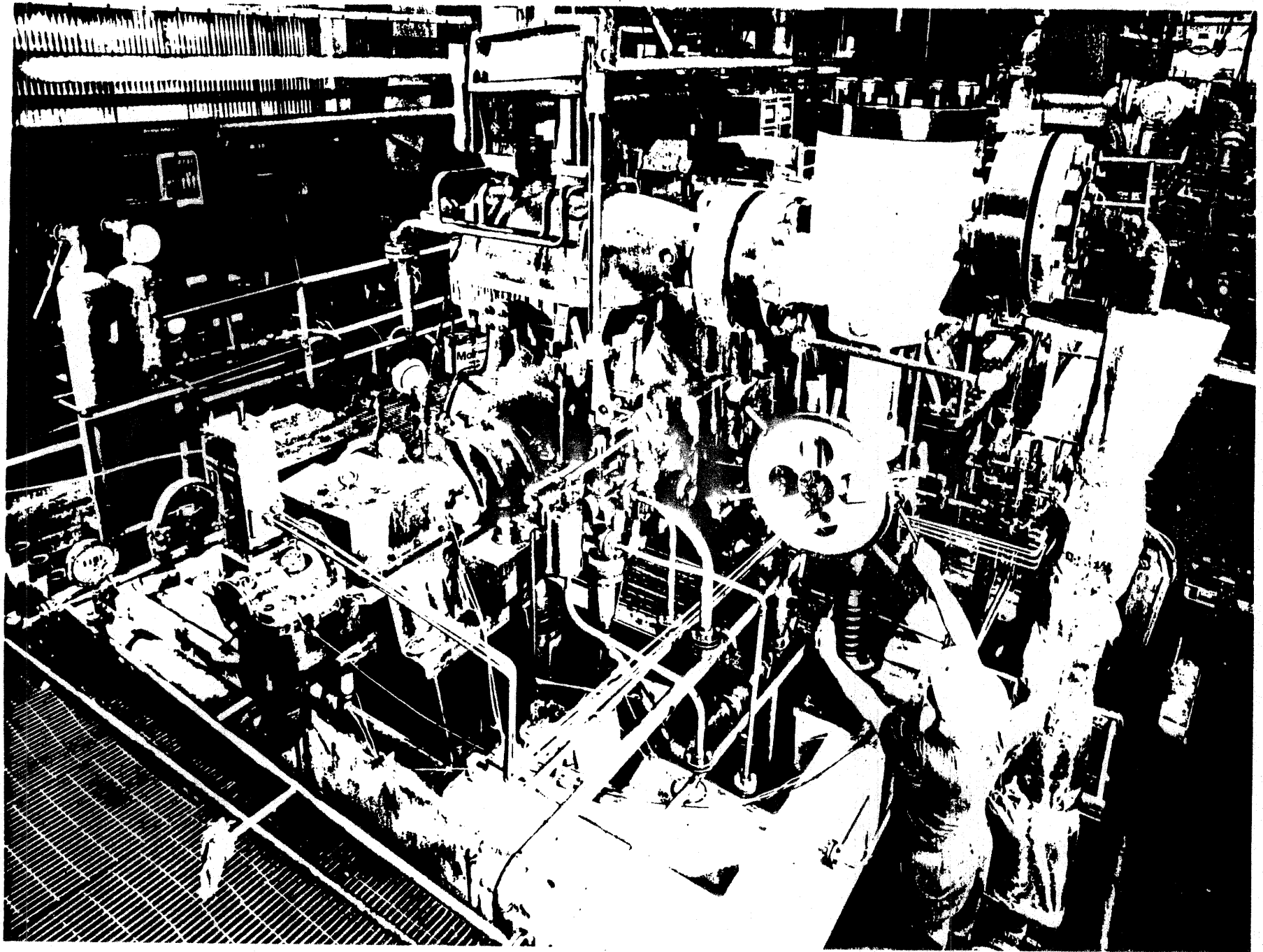
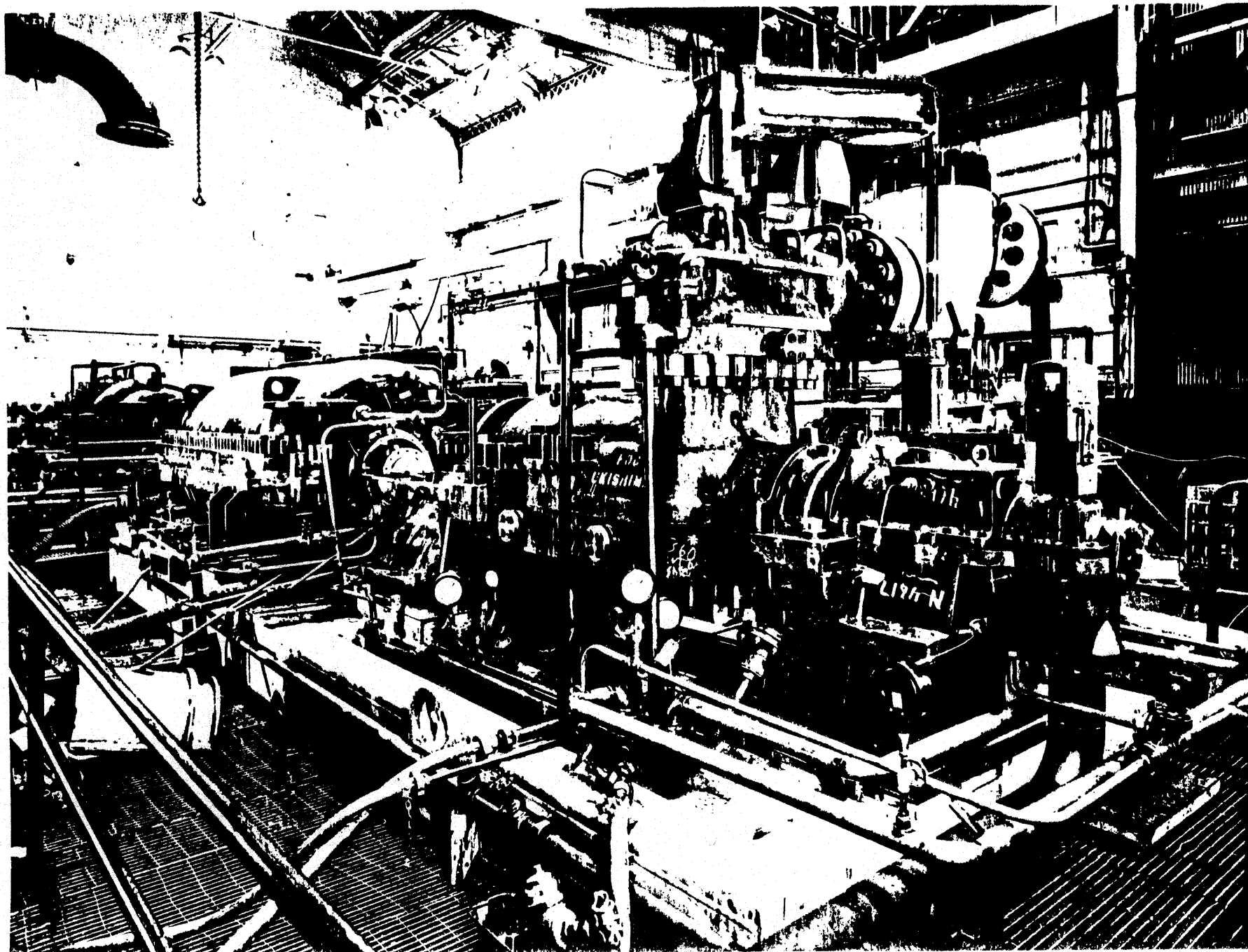


Figure 2

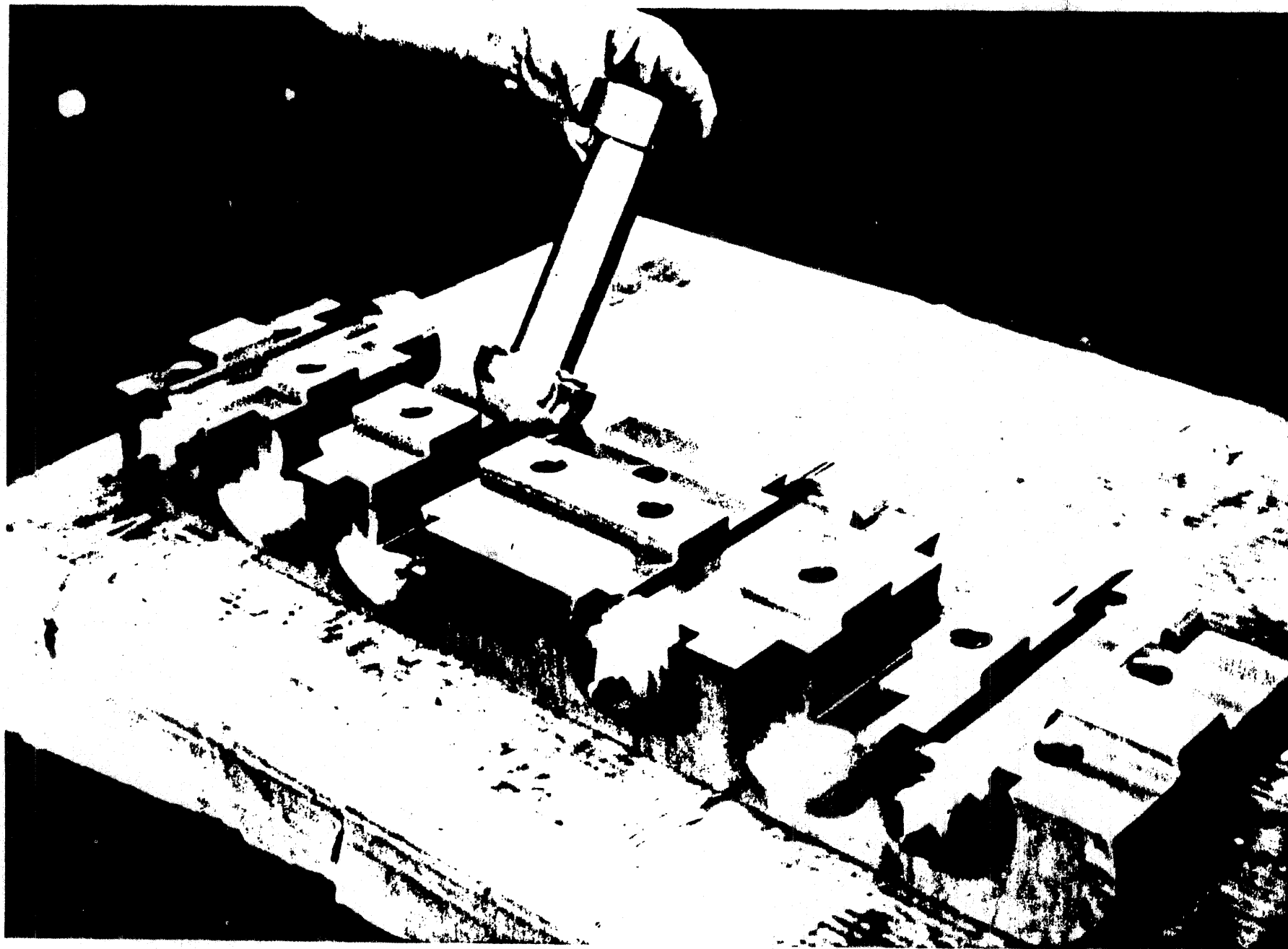
Extraction steam demand reduces; turbine speed is constant



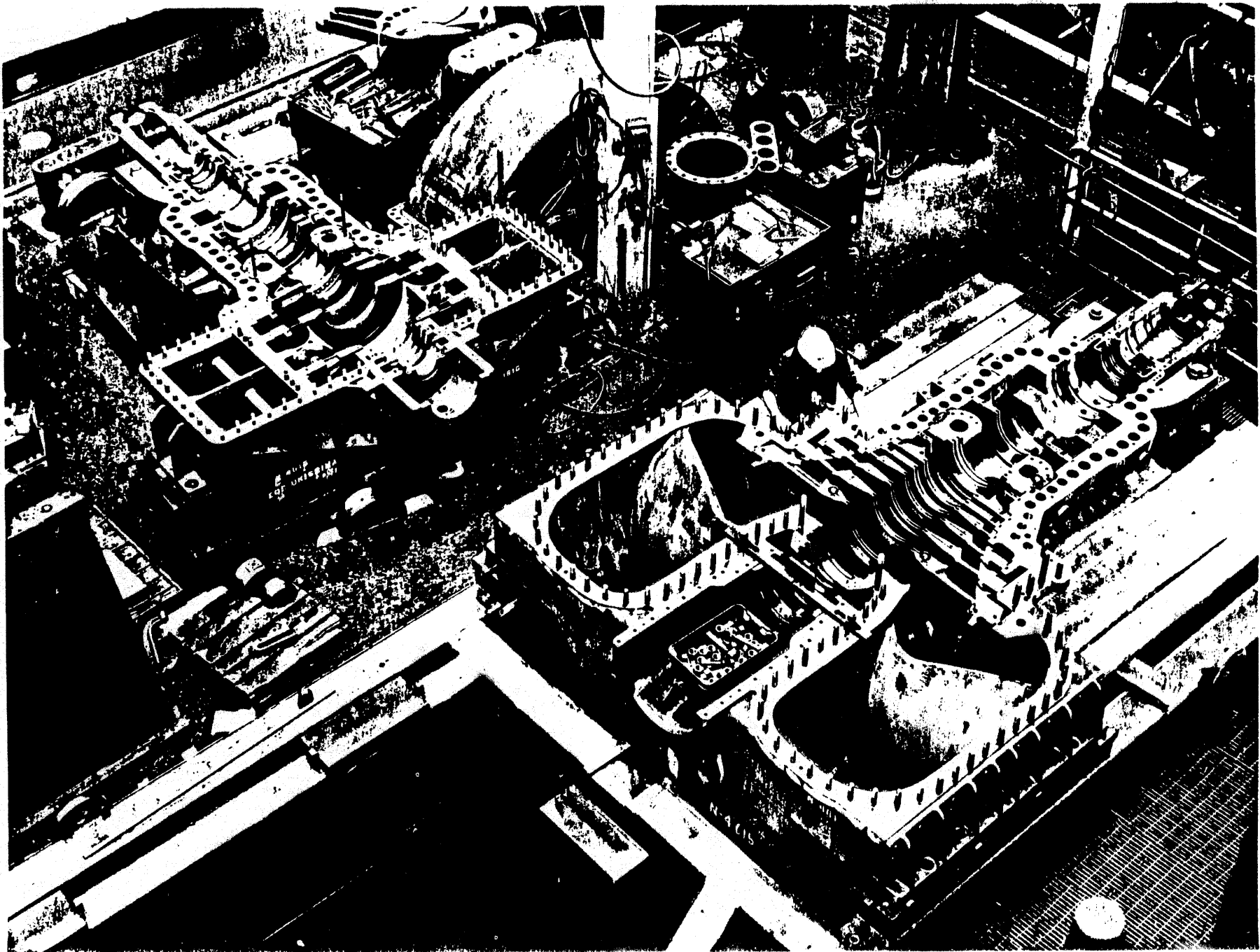
Mechanical test of 24,800 HP (18 500 kW) double-flow, extraction-condensing unit.



2SNV-5 back structure on test

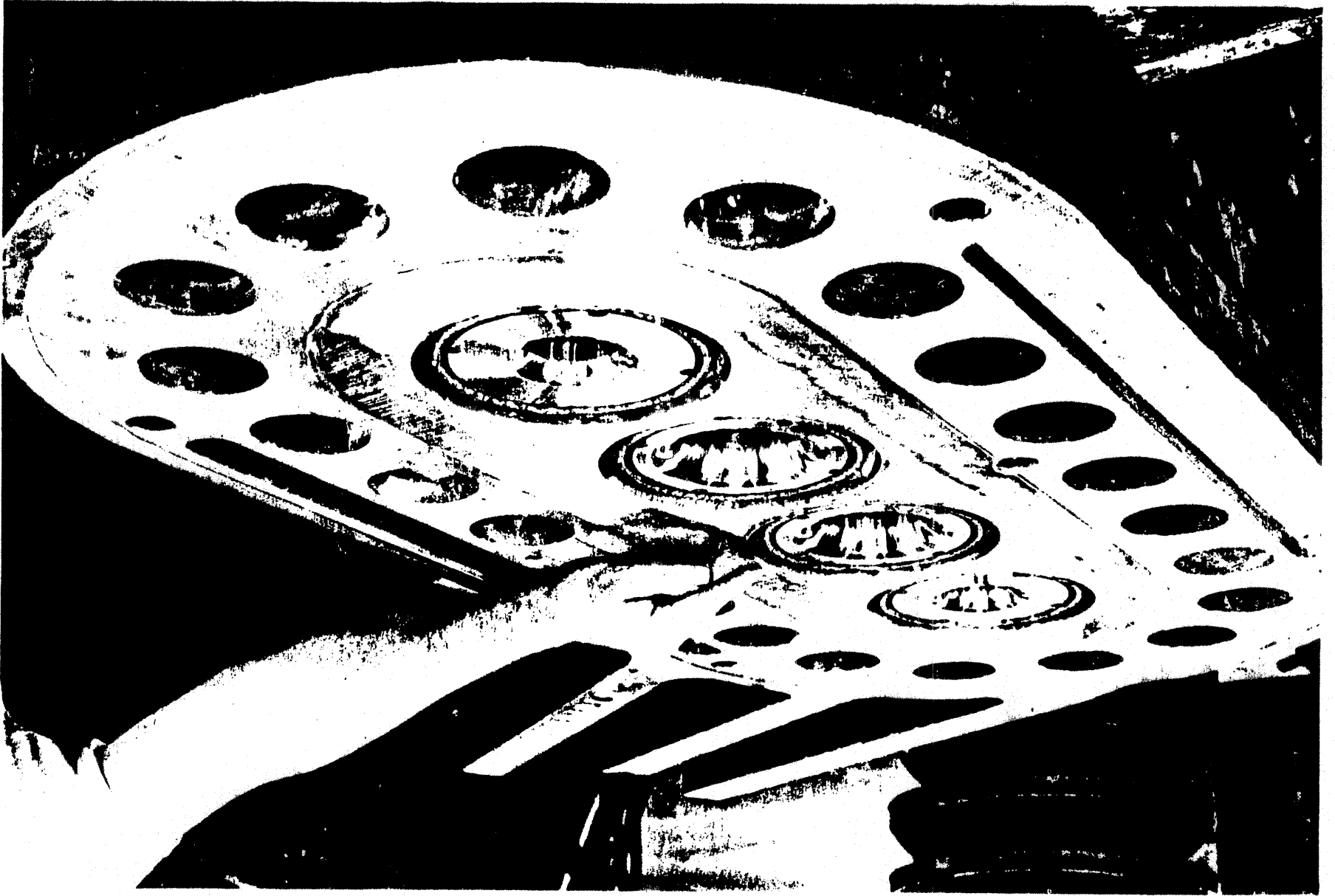


Split valve lifting bar showing square valve stems.

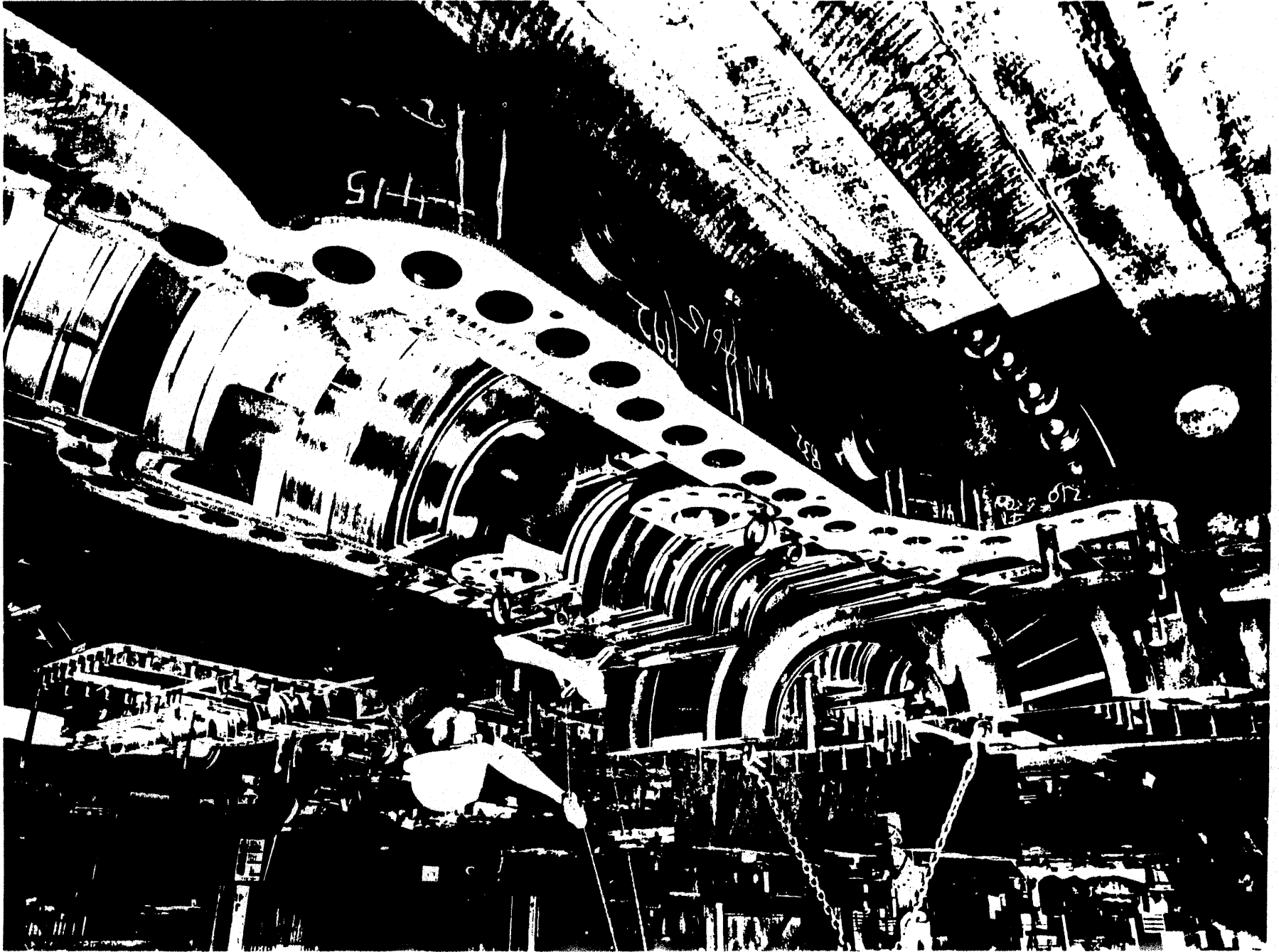


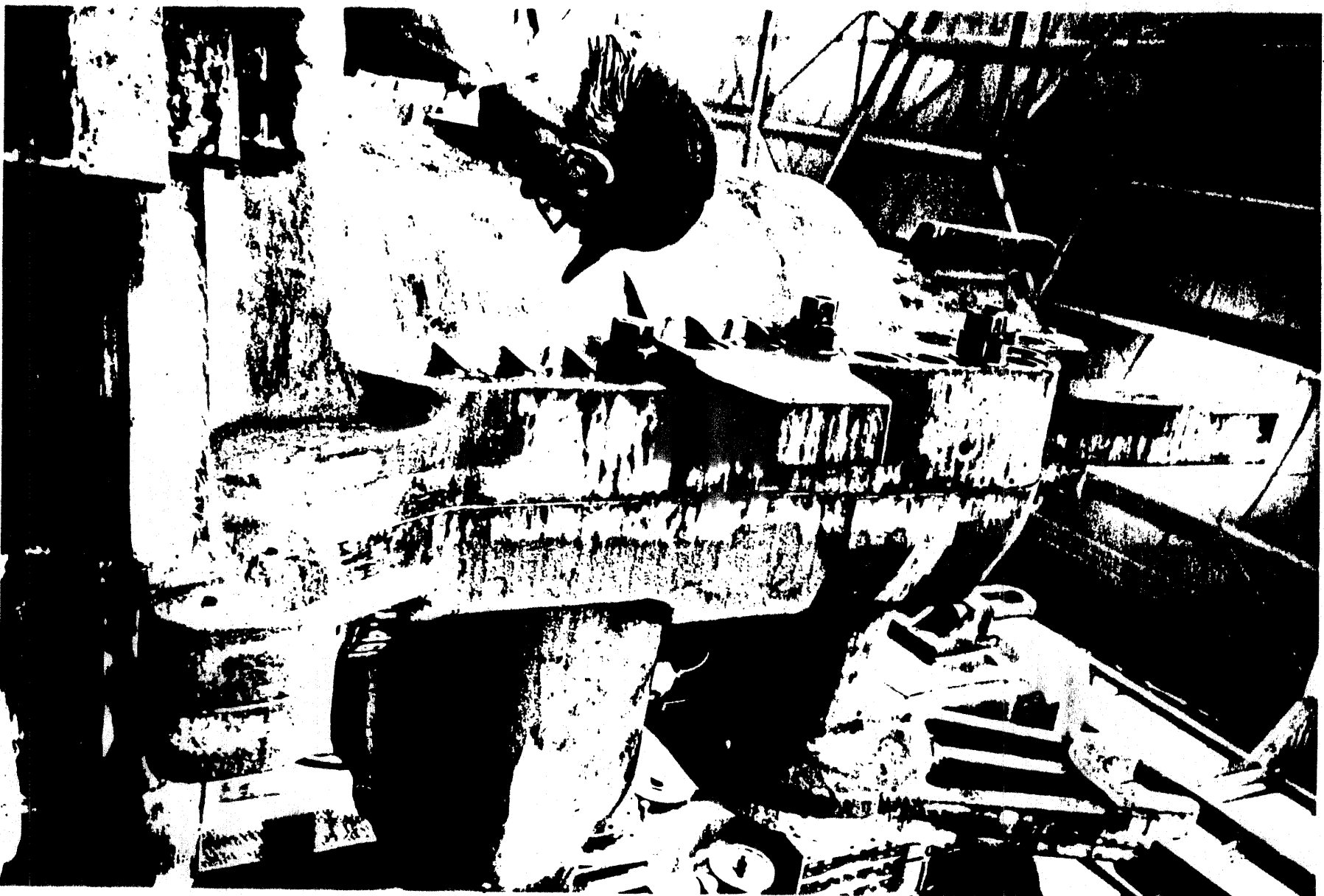
Two extraction-condensing turbines totaling almost 8 IP (59 700 kW) being assembled on Elliott's test floor.

Inlet valves installed in steam chest.

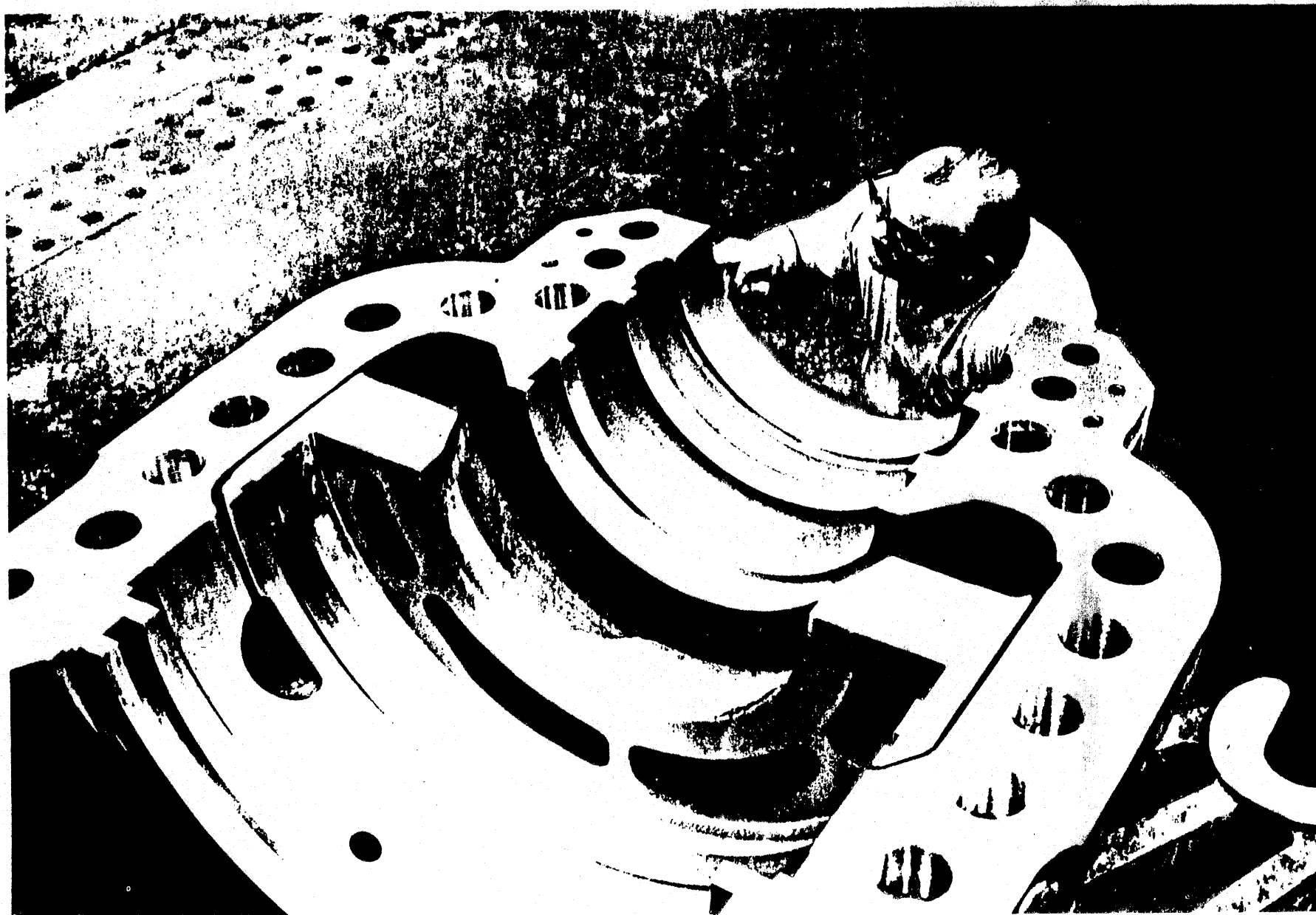


Lowering last-stage diagram into lower-half casing

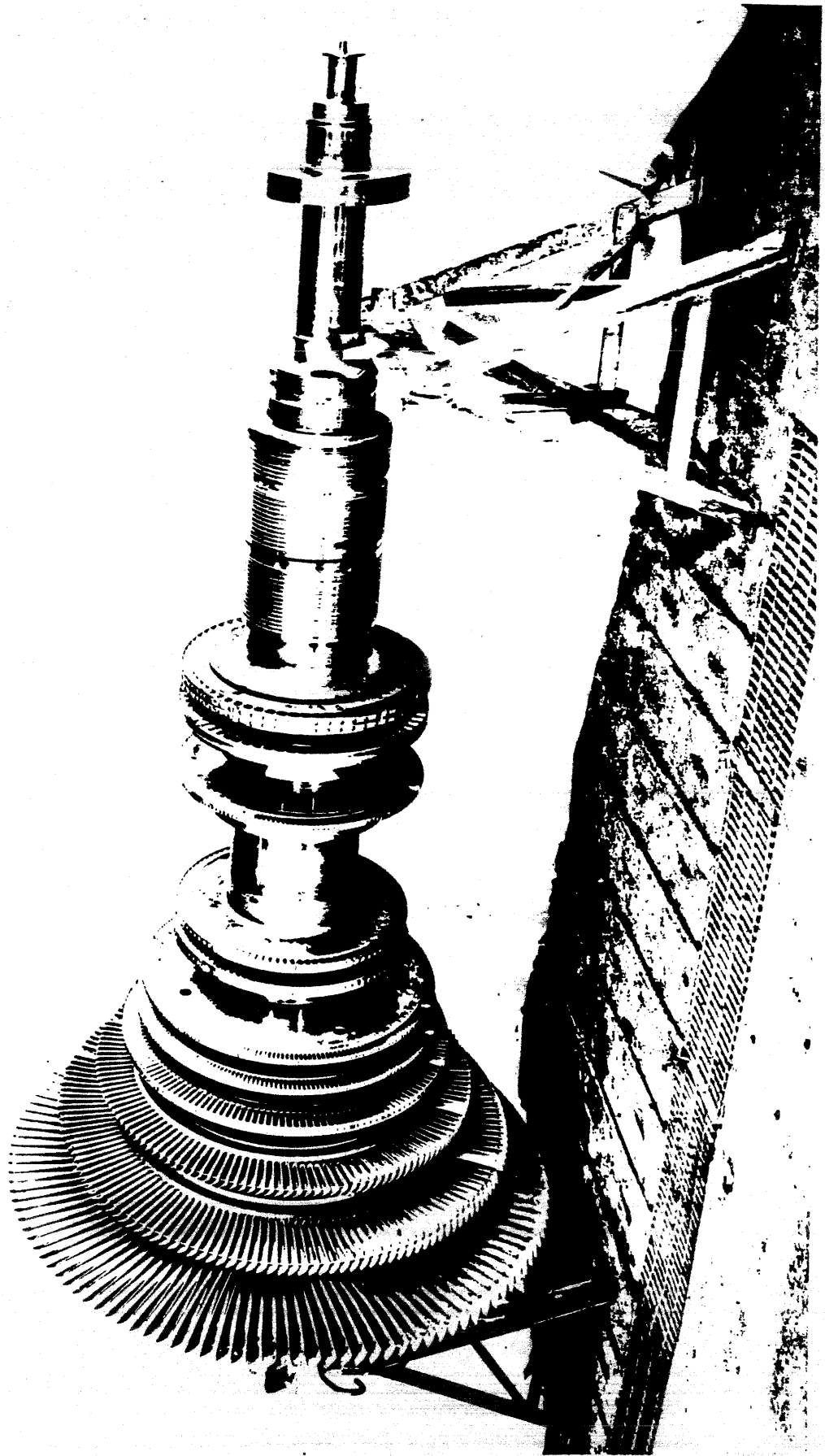




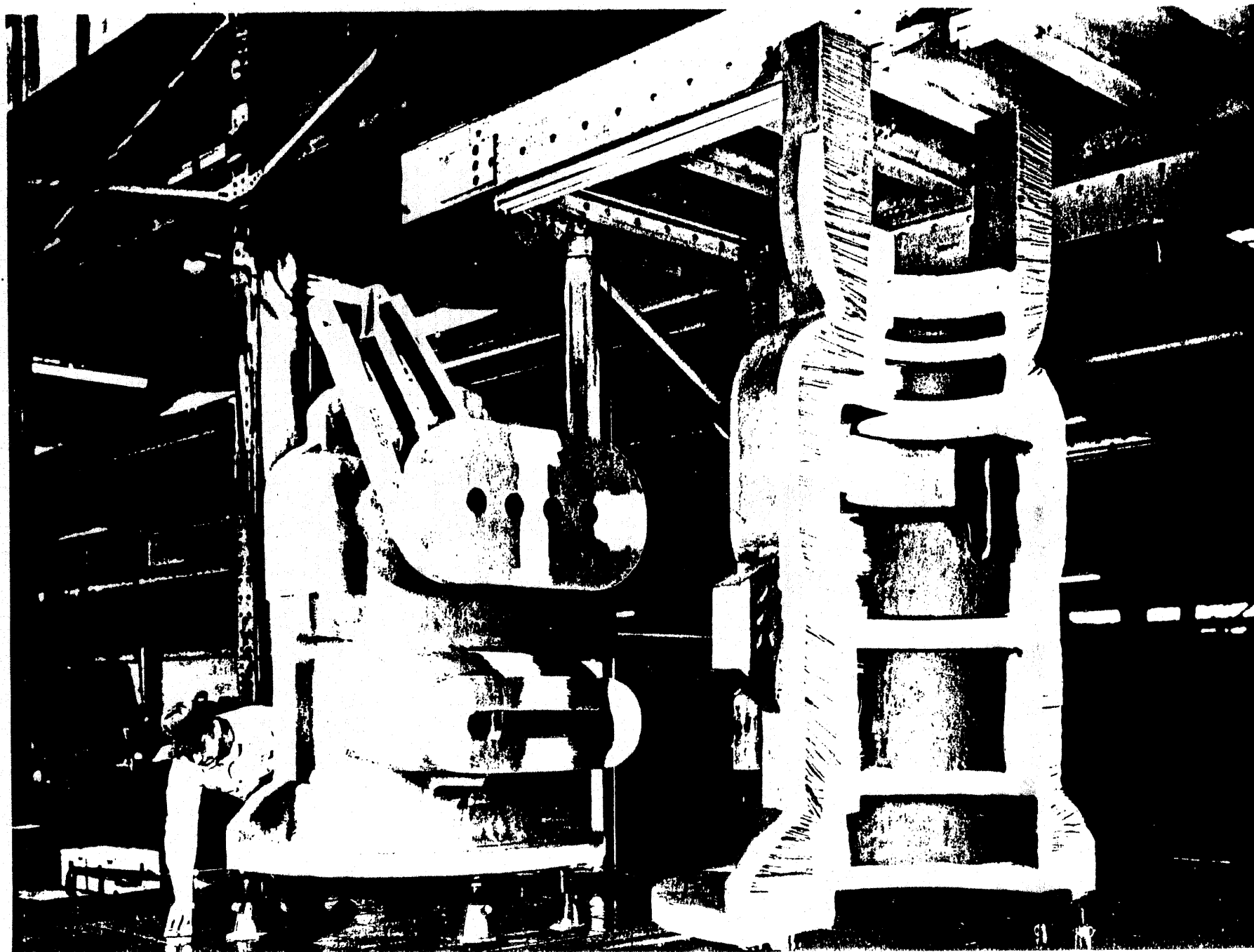
Inlet casing after rough machining.



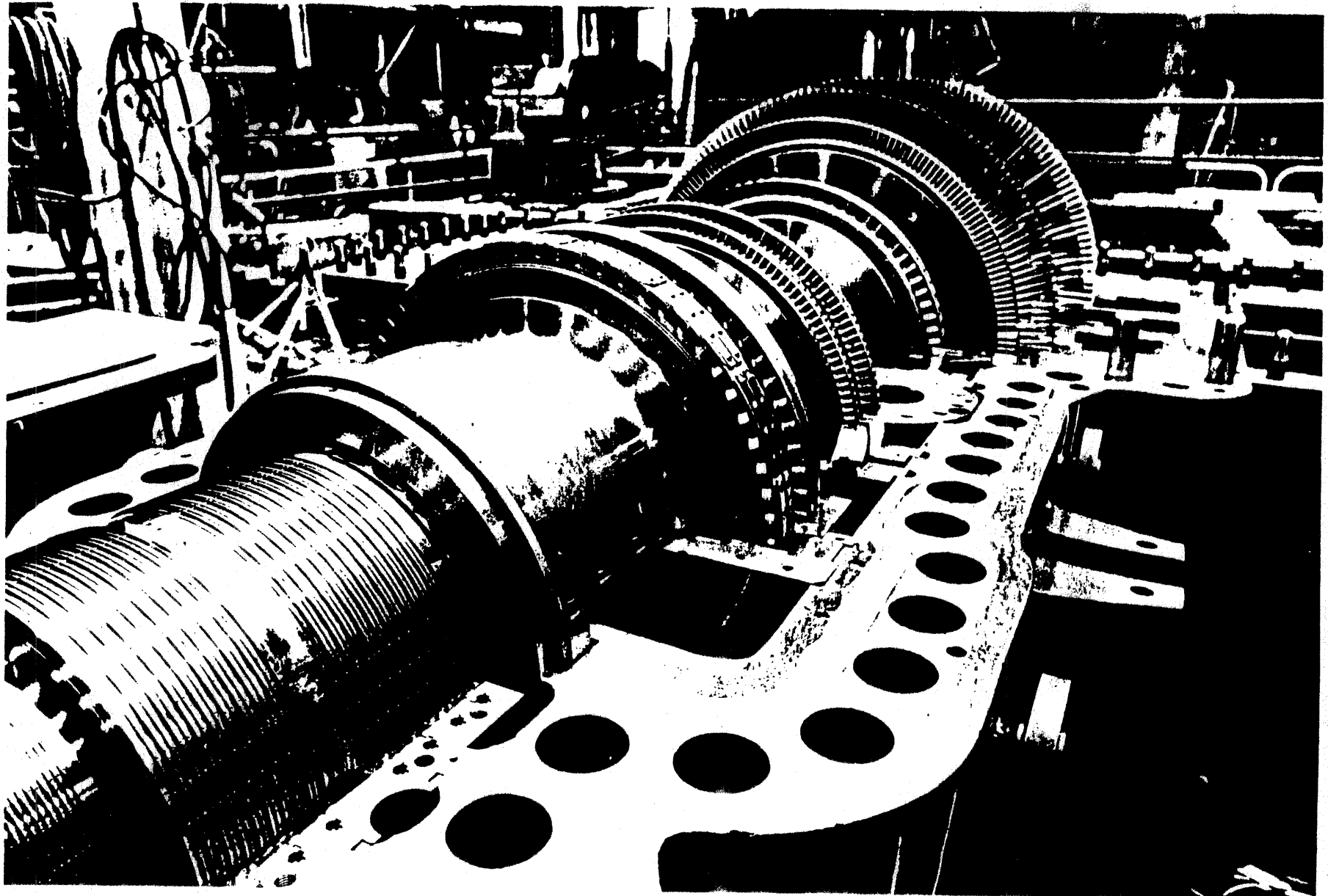
Inverted upper-half casing showing how the banks are separated from the horizontal flange.



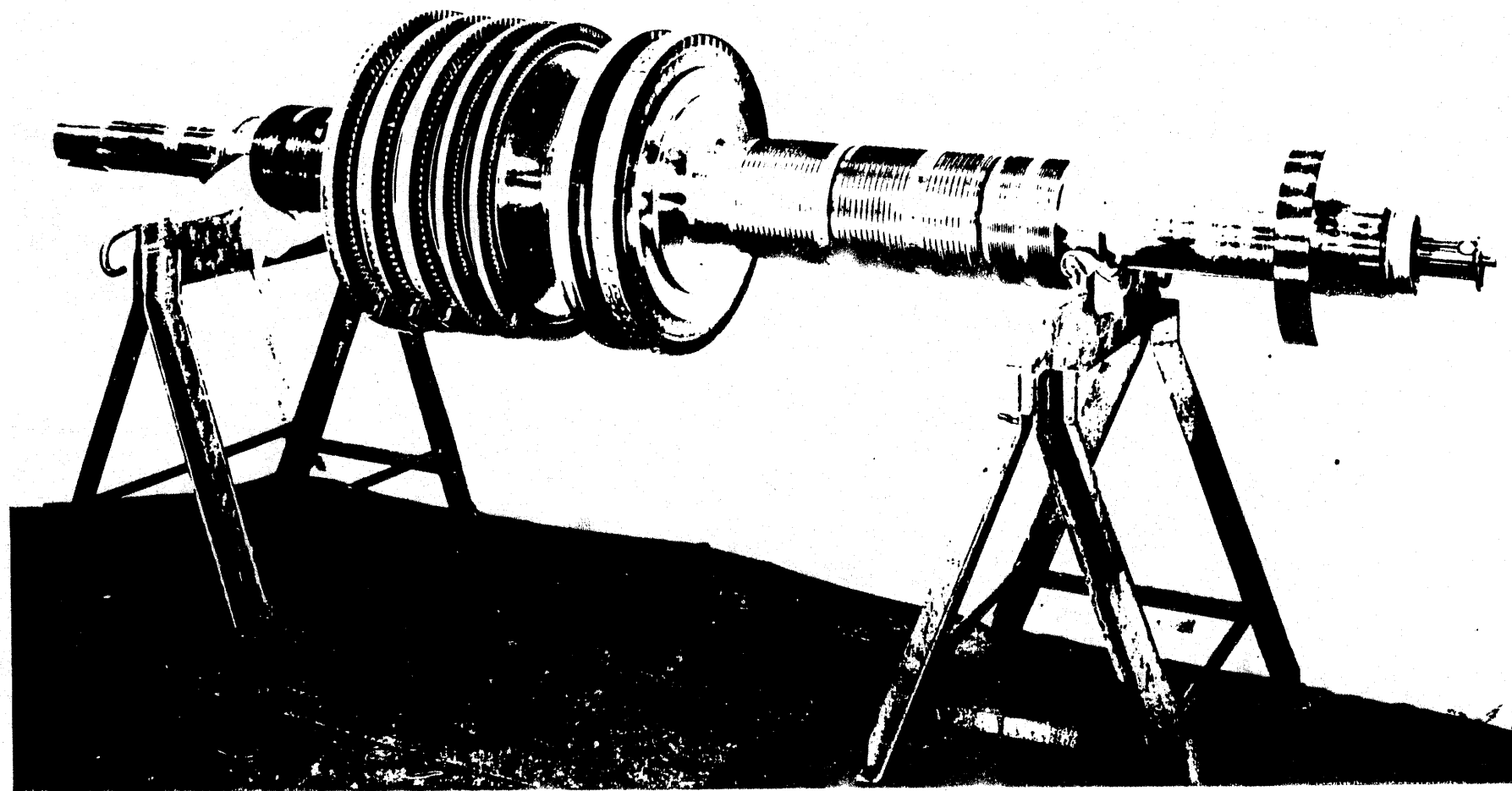
Completed rotor for 54300 H.P. (40500 kW) condensing-extraction turbine.



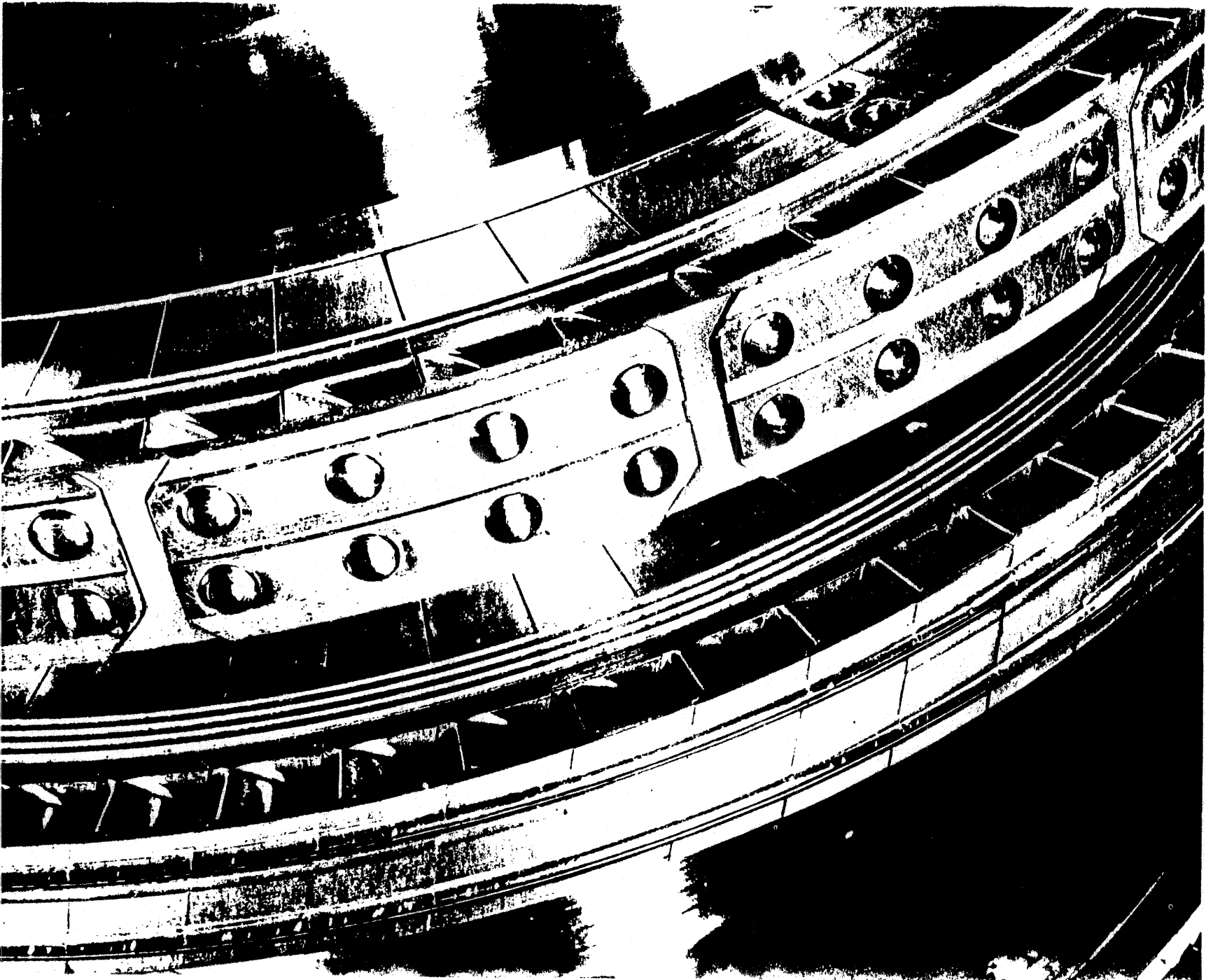
Turbine inlet component on layout table.



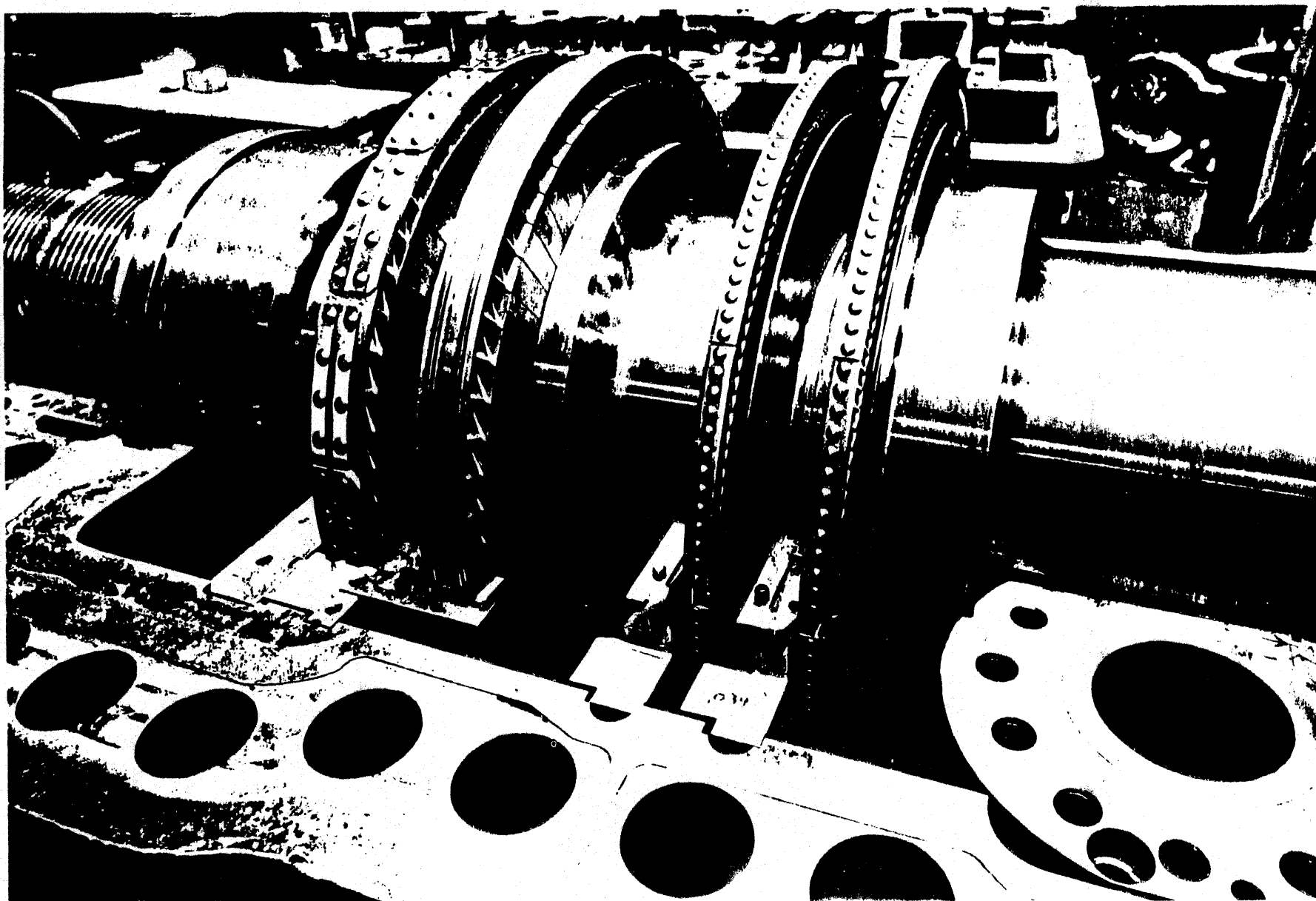
Rotor in lower-half casing showing labyrinth seals.



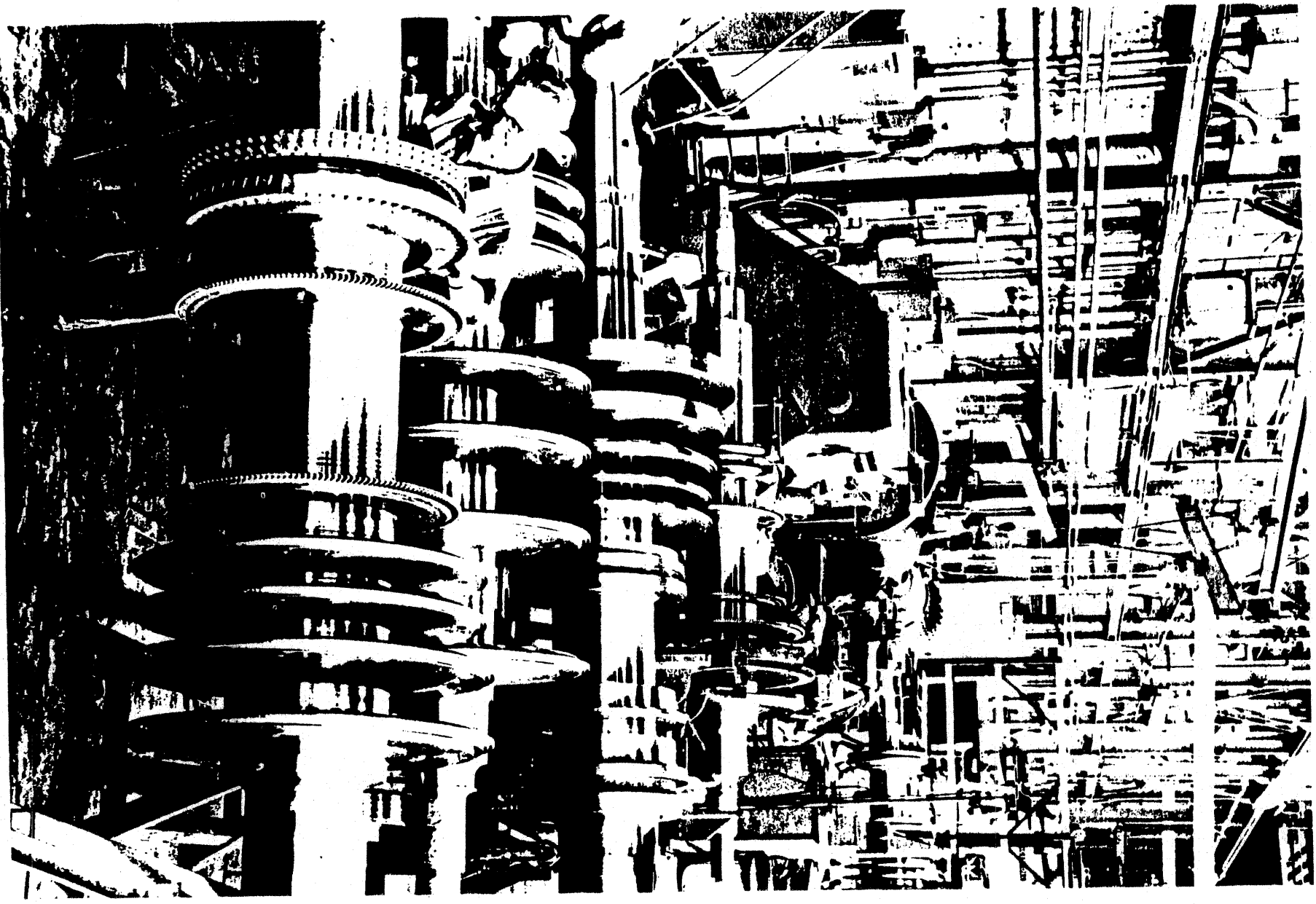
Completed rotor for back-pressure turbine.



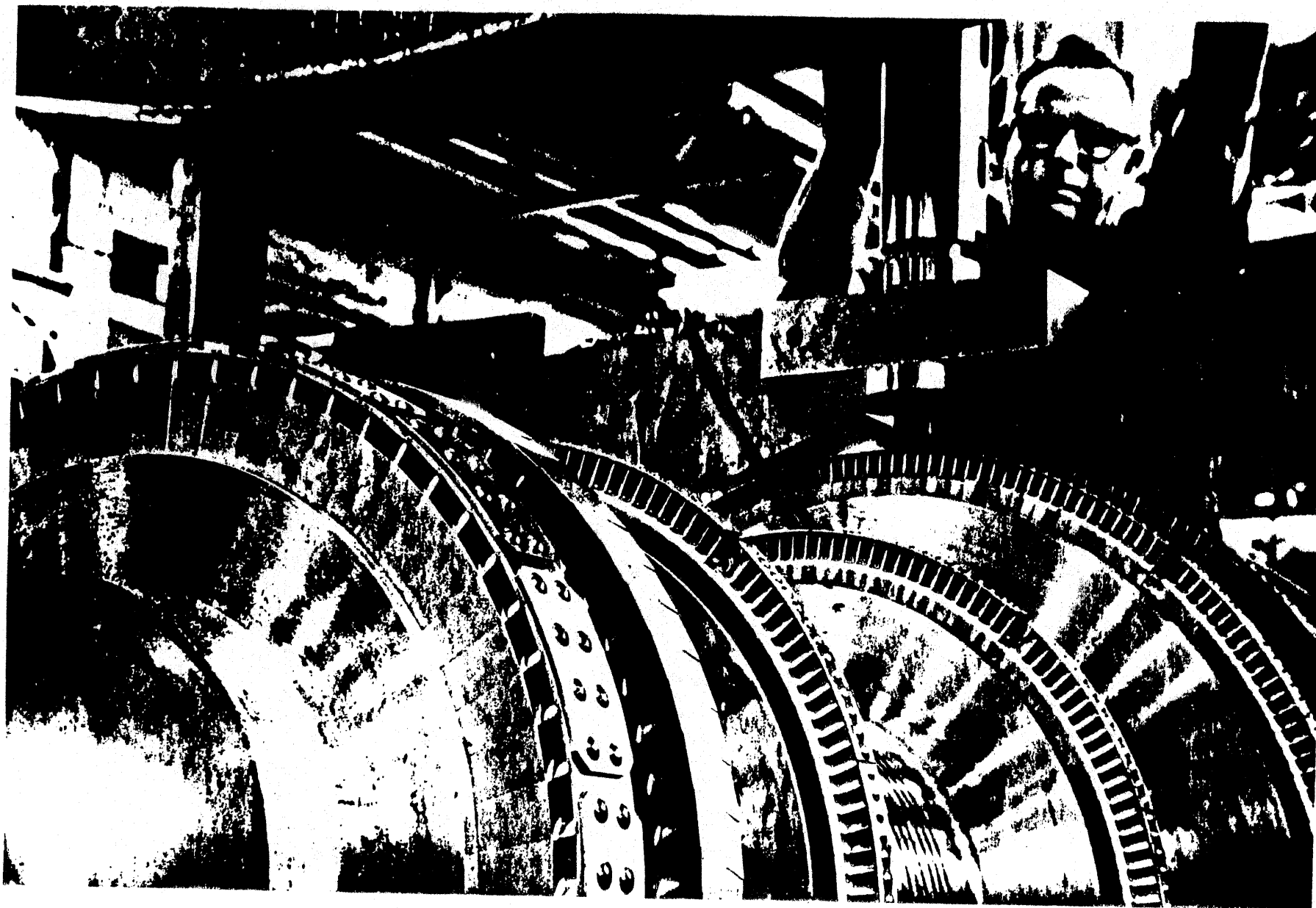
Detail view of 2000 psi (138 bar) Curtis stage.



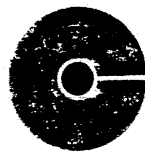
Rotor in lower-half casing showing Curtiss and Rateau staging and extraction diafram.



Installing blades on finish-machined rotors.

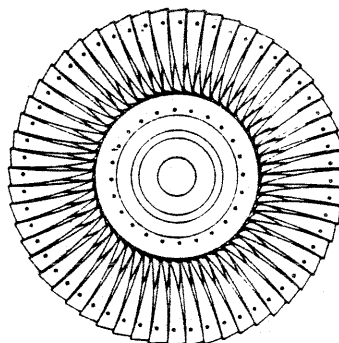


Rolling blade tangs on partially completed rotor



Reprint 138A
March 1976

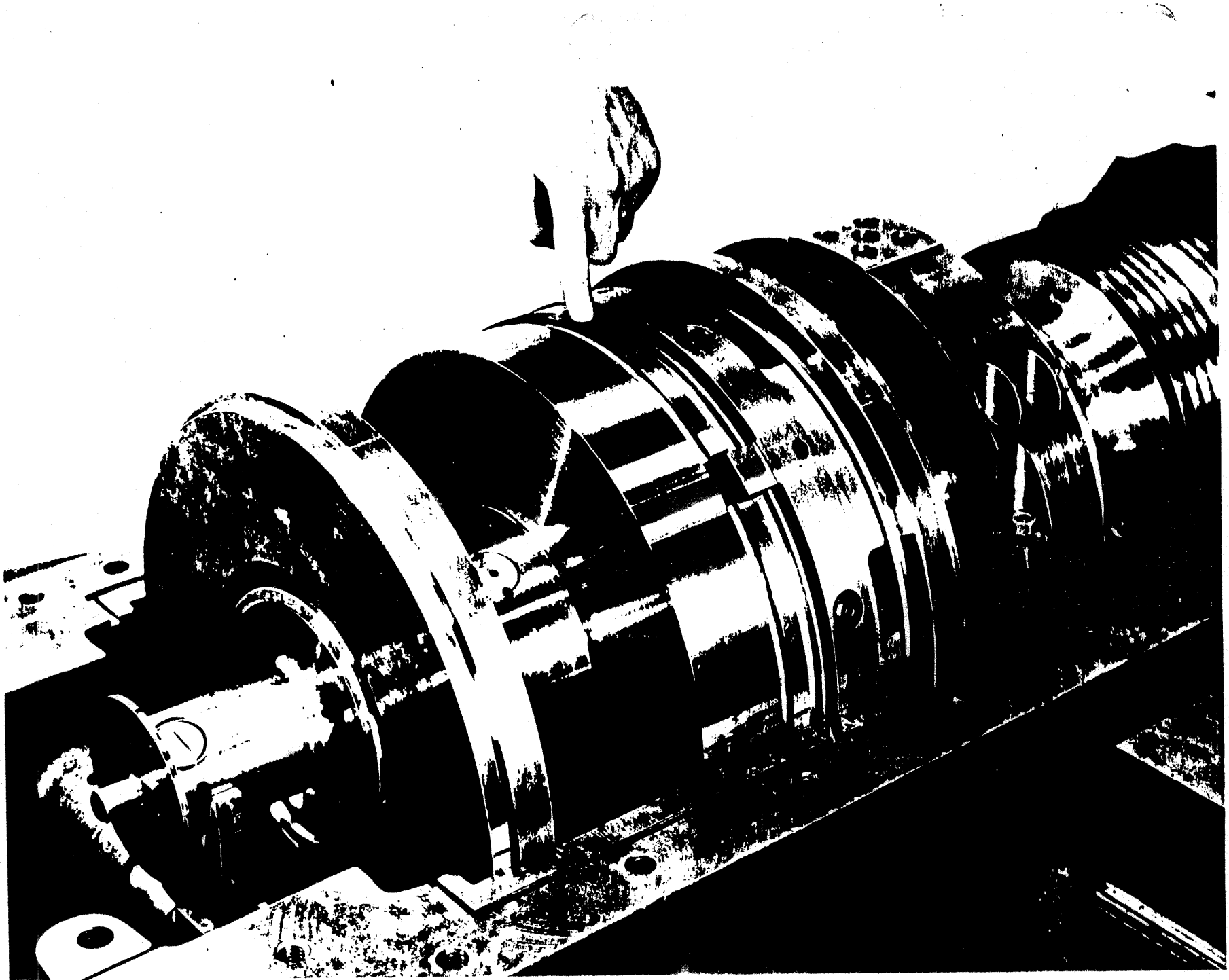
2000 psig (138 bar) Steam Turbines



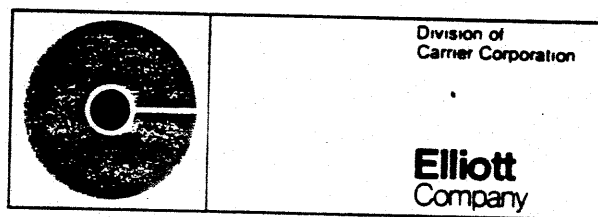
***Elliott
Multistage
Steam
Turbines***



Shaft forgings as received from forge shop.



Bearing housing for large units showing downstream half of 112 in² (720 cm²) thrust bearing. Upstream half removed.



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

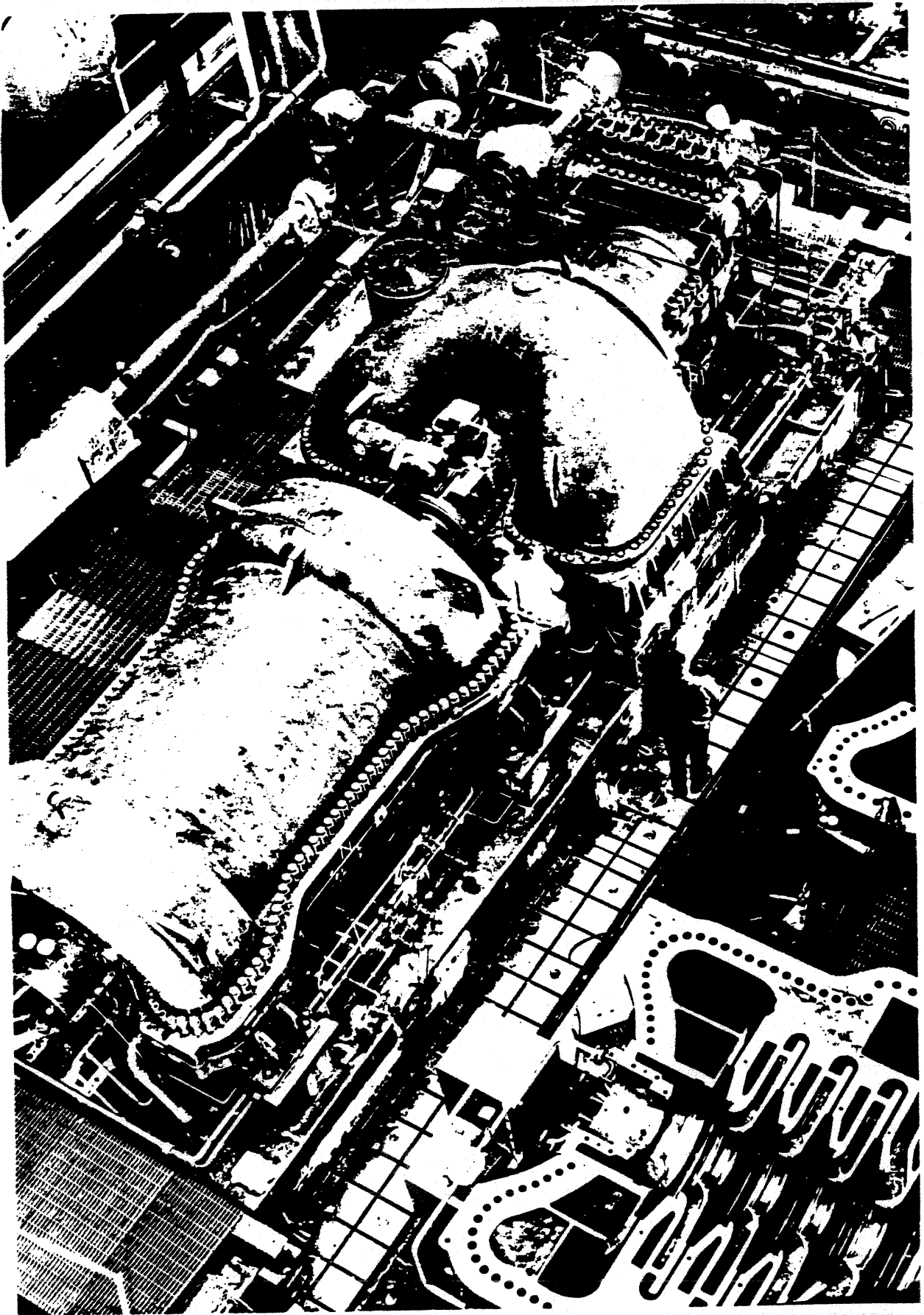
Elliott Company General Offices:
Jeannette, Pennsylvania 15644
Other plants at Dayton and Springfield, Ohio;
Isle of Wight, England

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

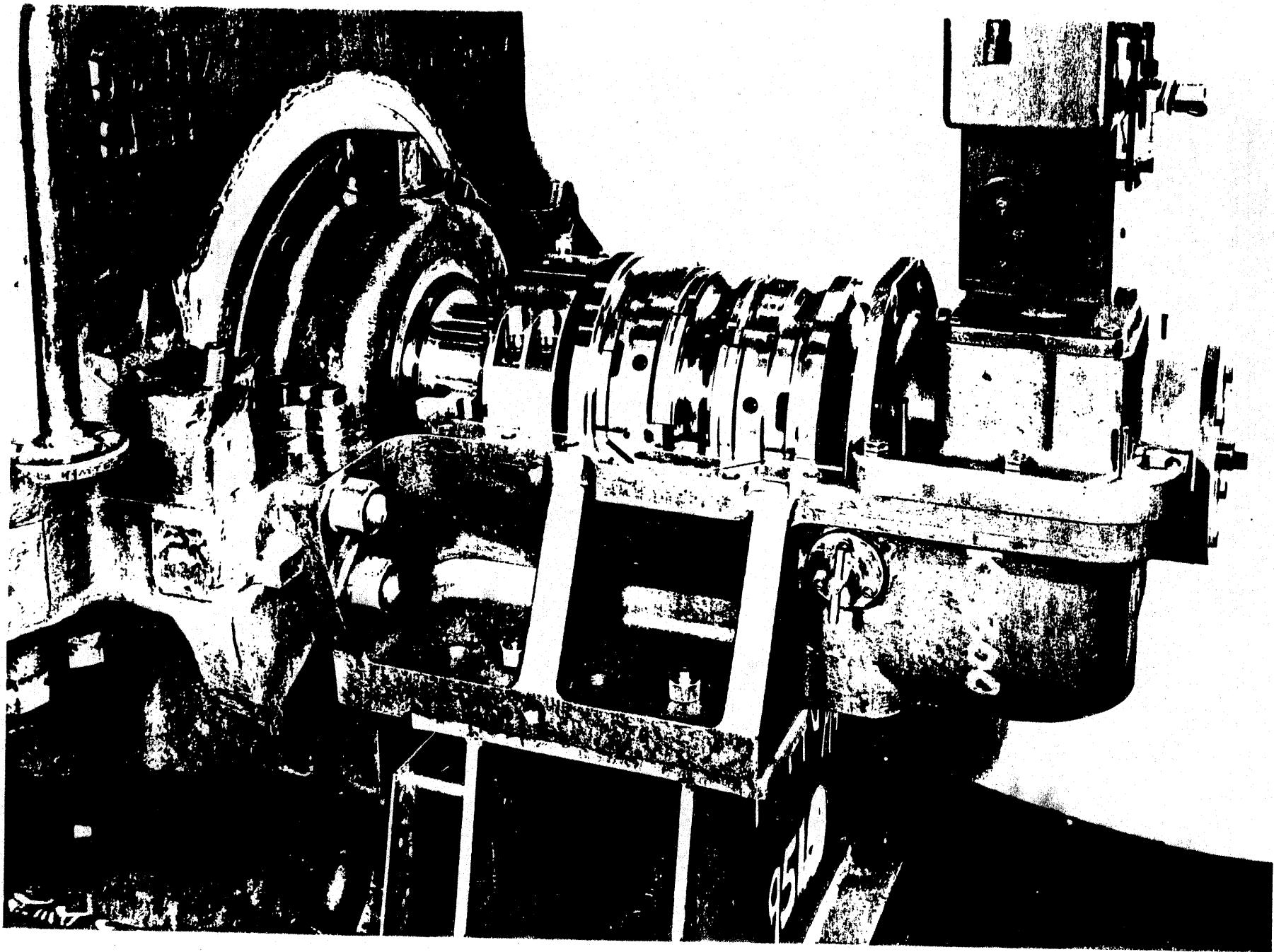
Europe, Middle East, Africa:
Elliott Turbomachinery Ltd.
197 Knightsbridge, London, England

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

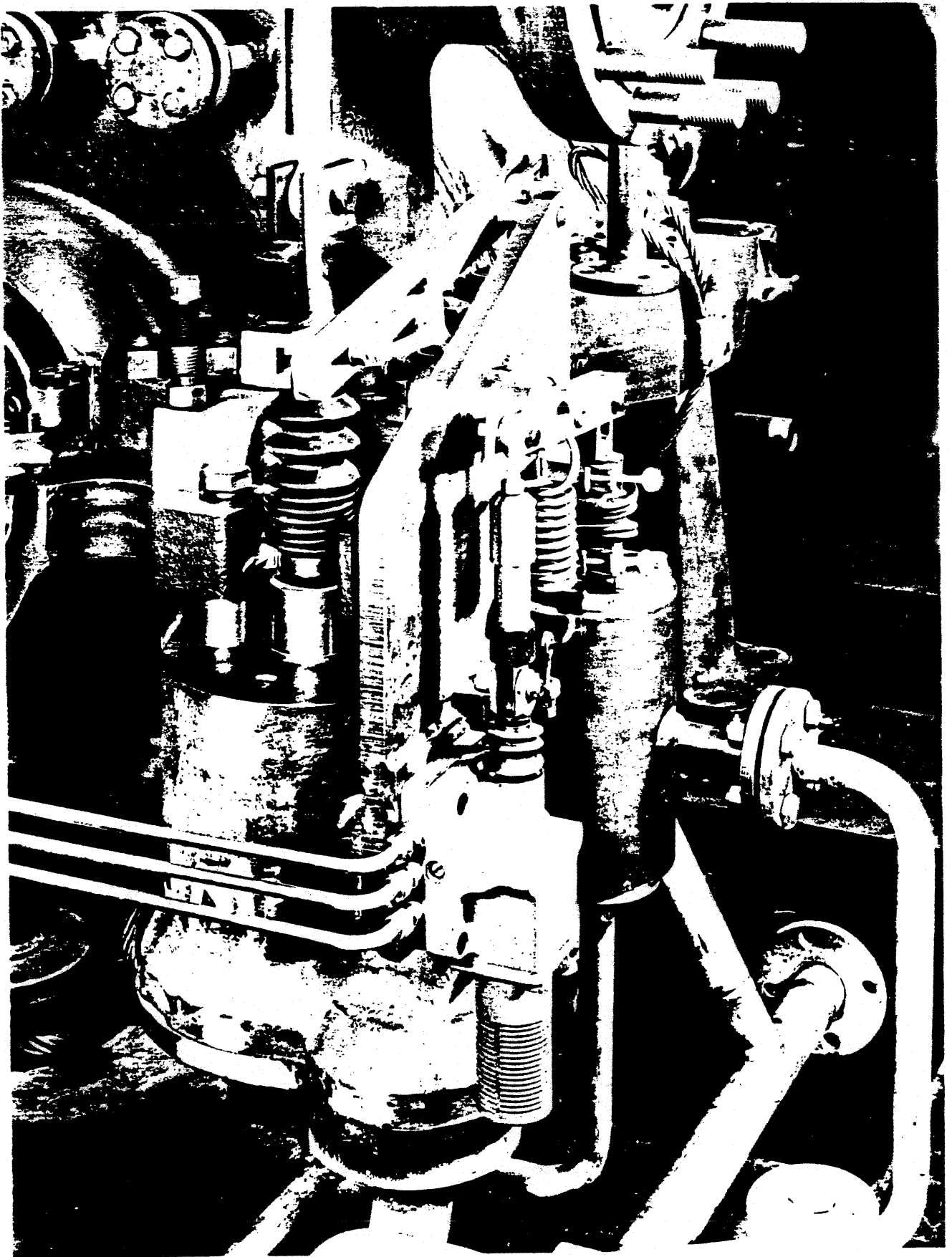
Far East: Elliott Overseas Company
P.O. Box 1112, City of Industry, California 91749



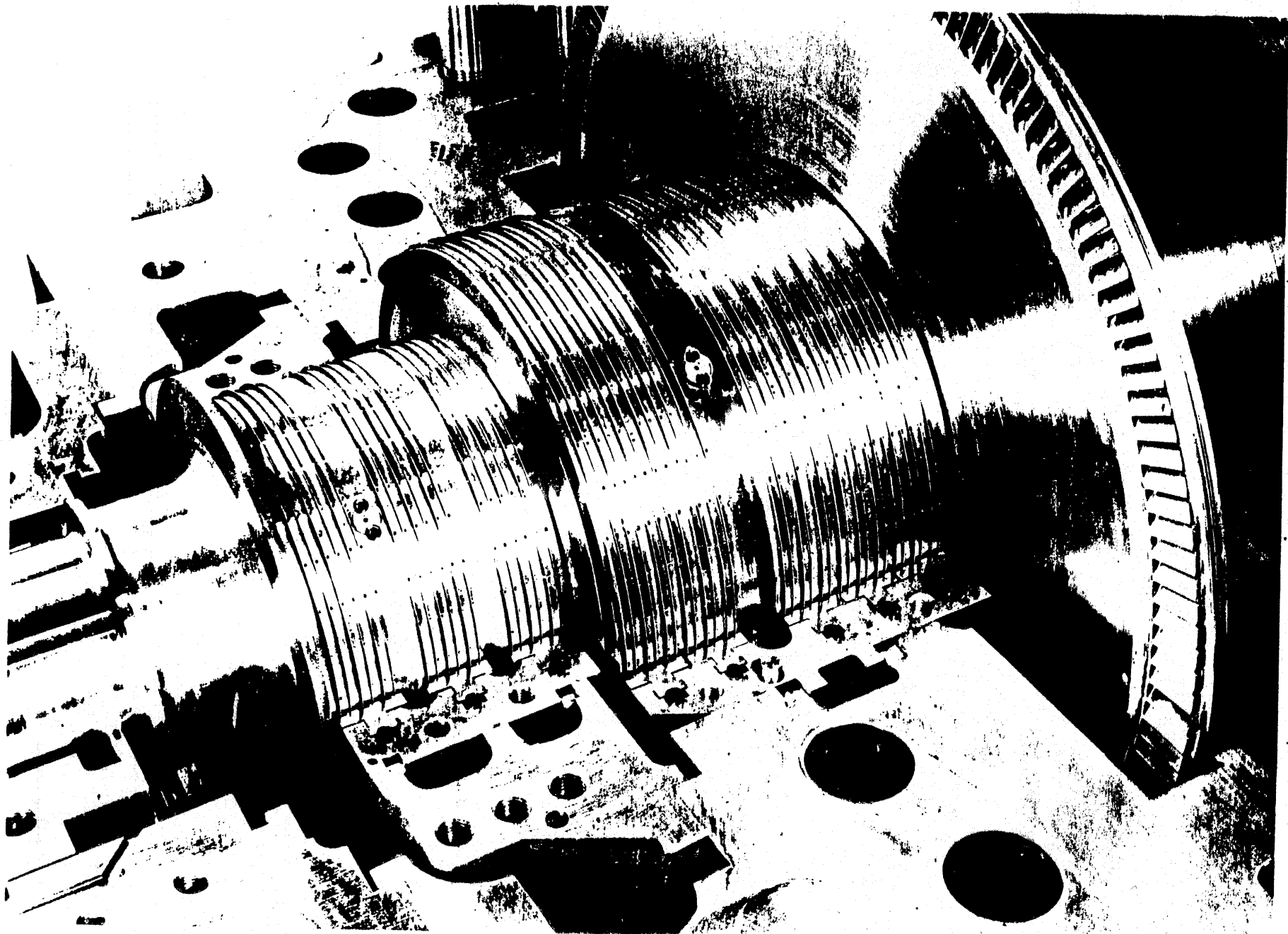
Turning gear being adjusted on a large ethylene plant string.



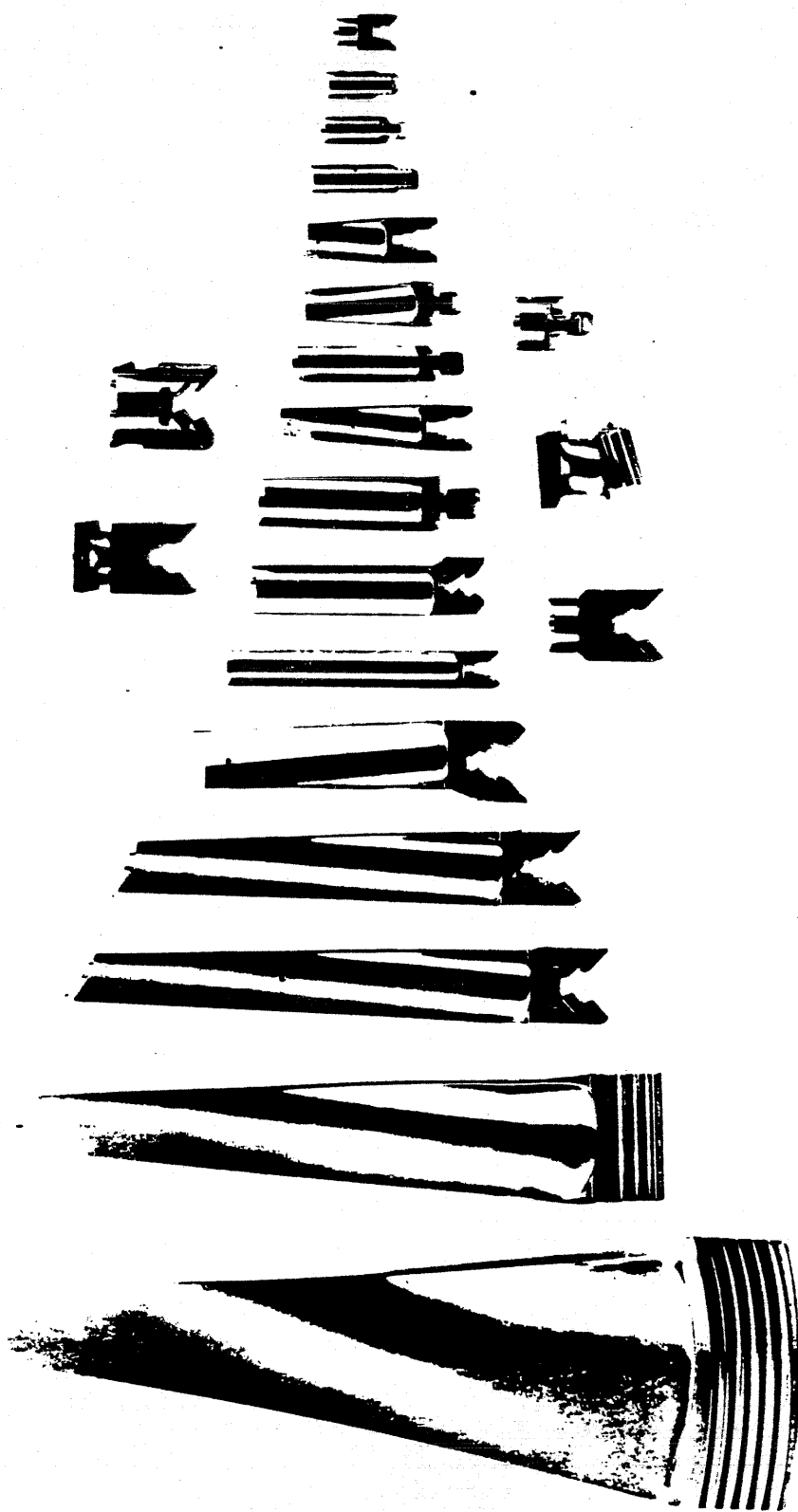
Bearing housing for moderate size turbines.



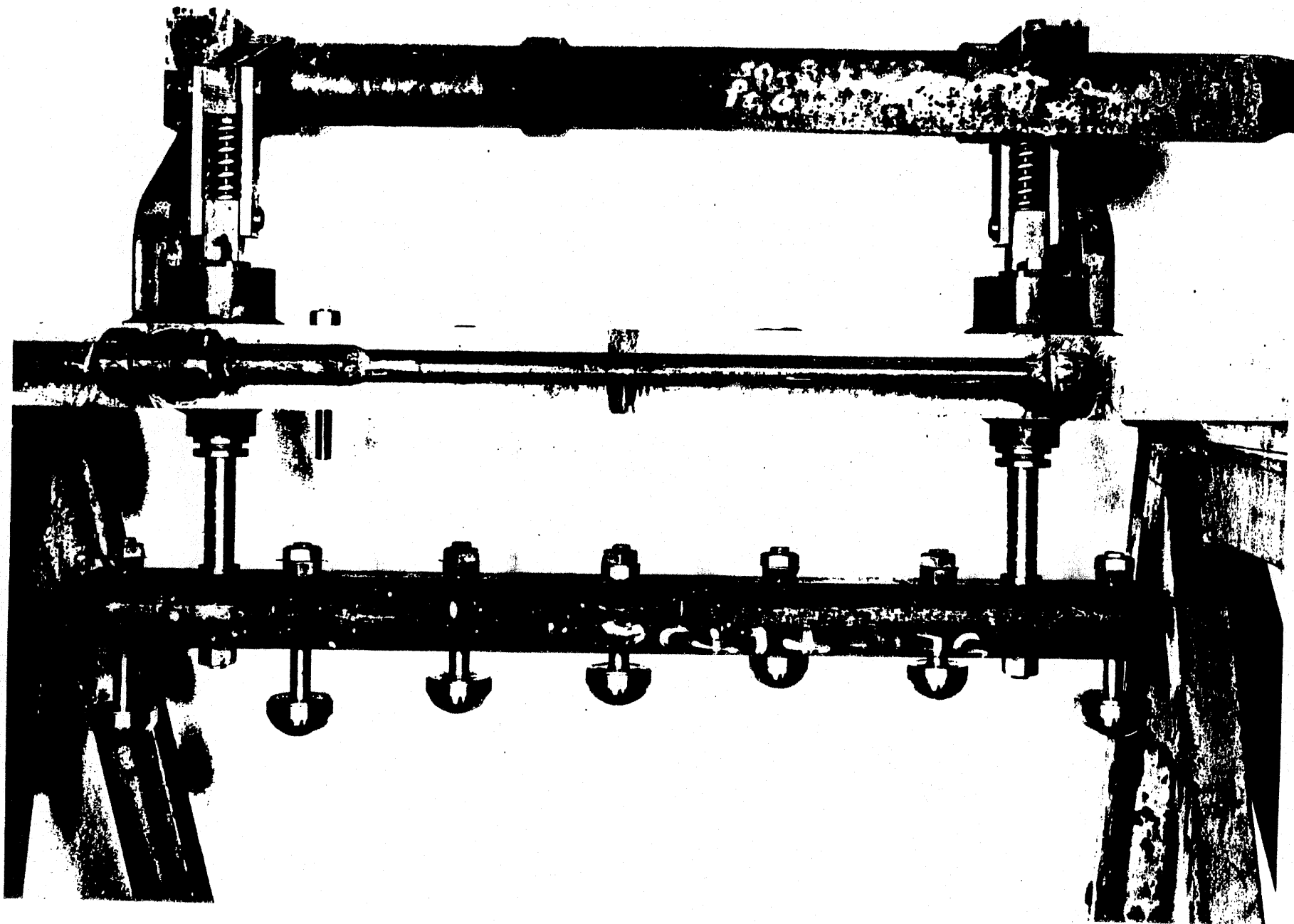
Servo-motor for operating governor valves.



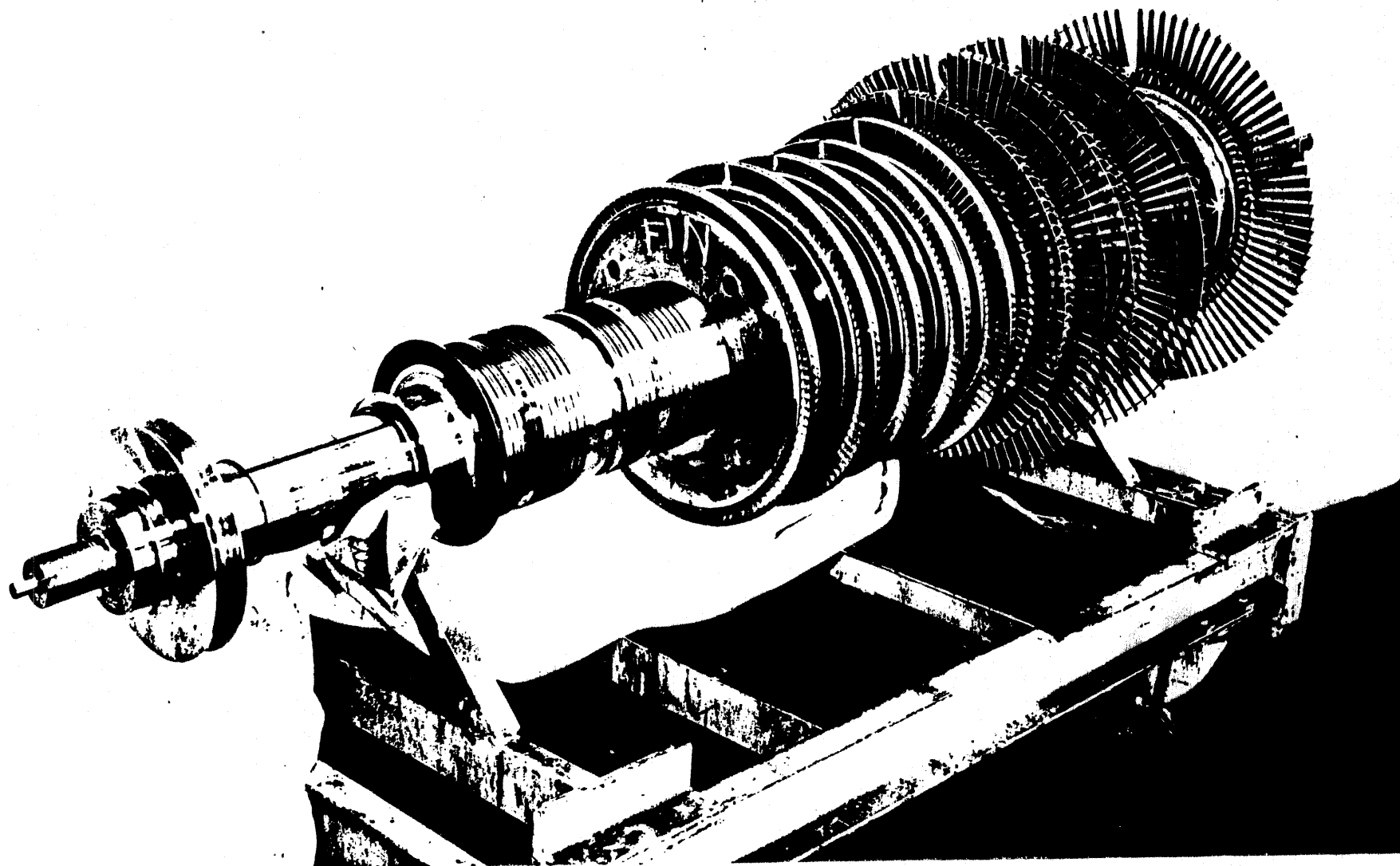
Labyrinth seal arrangement.



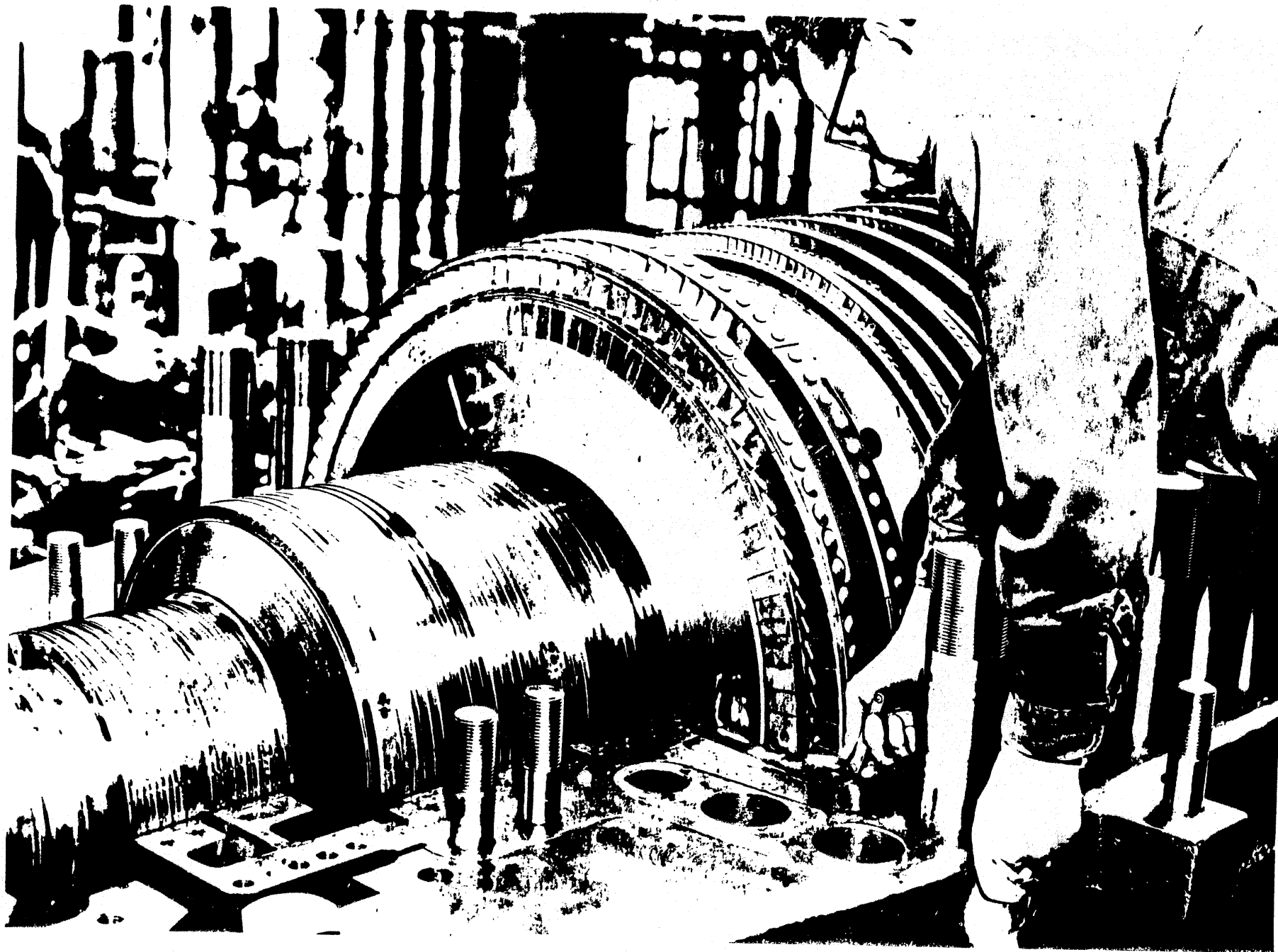
Some of the many blade designs available to meet any combination of power, speed and steam conditions.



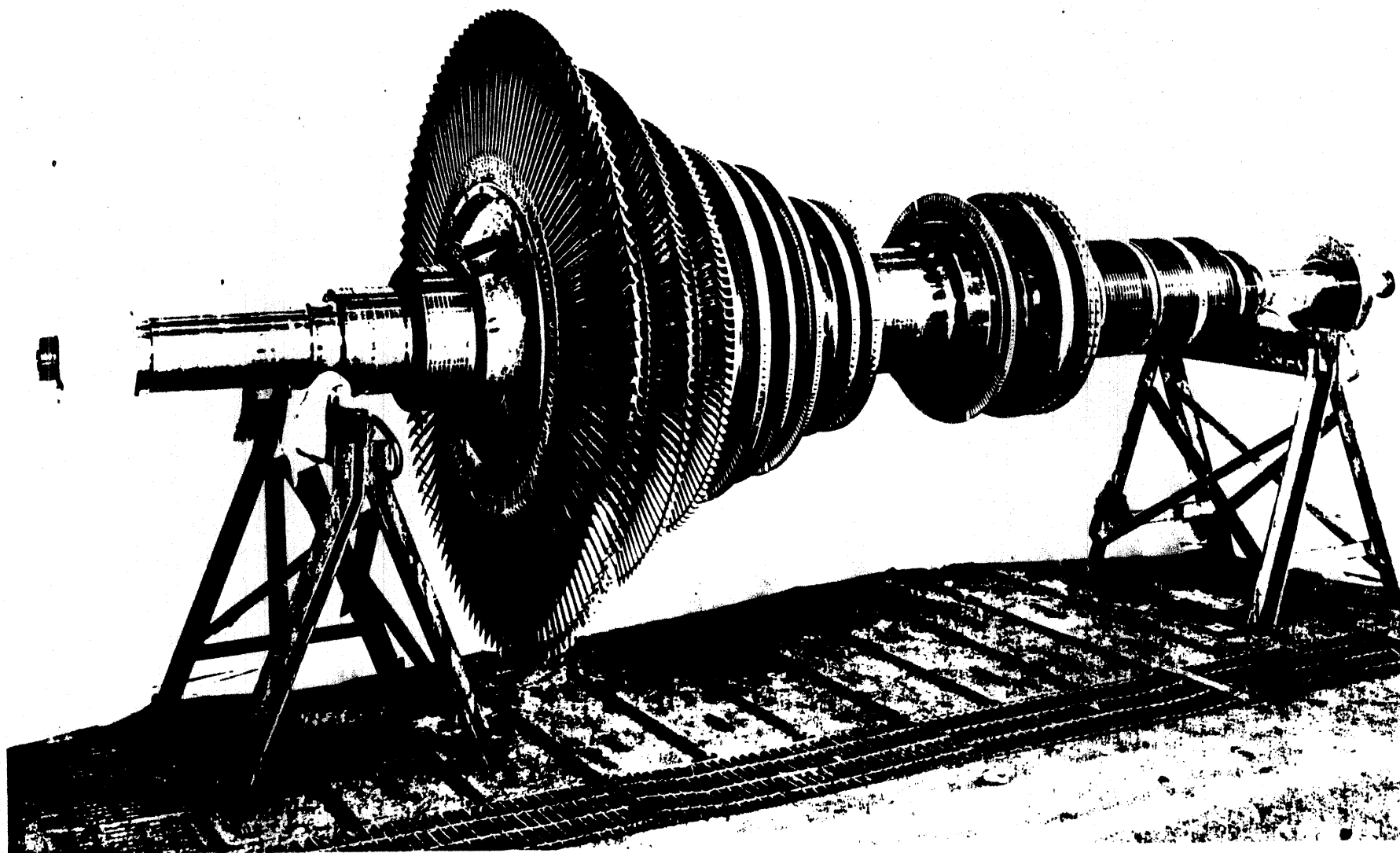
Bar-lift governor valve arrangement for sequential valve opening.



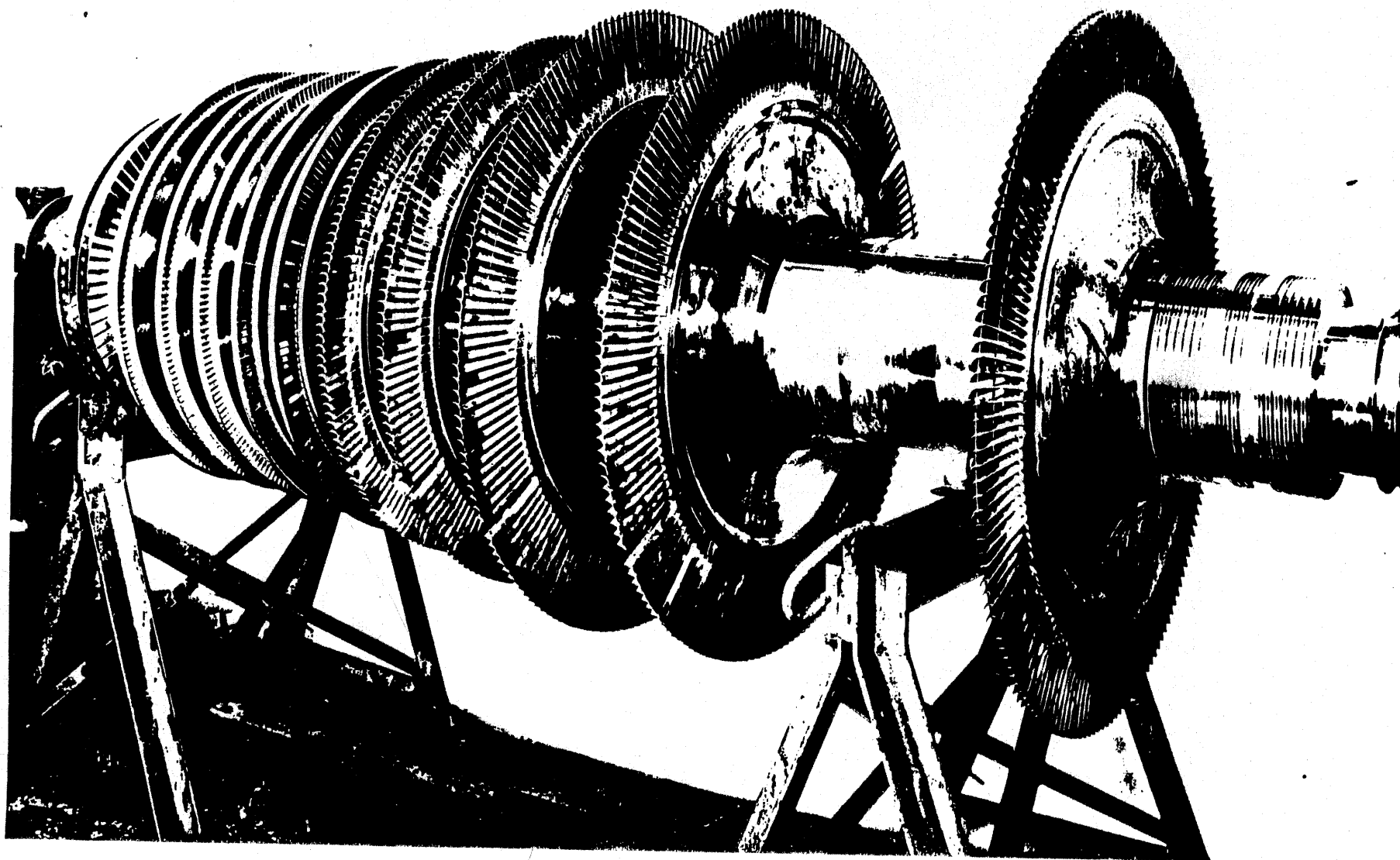
High speed double flow rotor for 900 psig (62 bar) service.



Large rotor installed in lower half casing



Nine stage, single-flow 54,300 HP (40 520 kW) extraction-condensing rotor with 16" (406mm) last stage blades.



Ten stage, double-flow 27,260 HP (20 340 kW) rotor.

INTRODUCTION TO PROCESS DESIGN

SECTION TWELVE

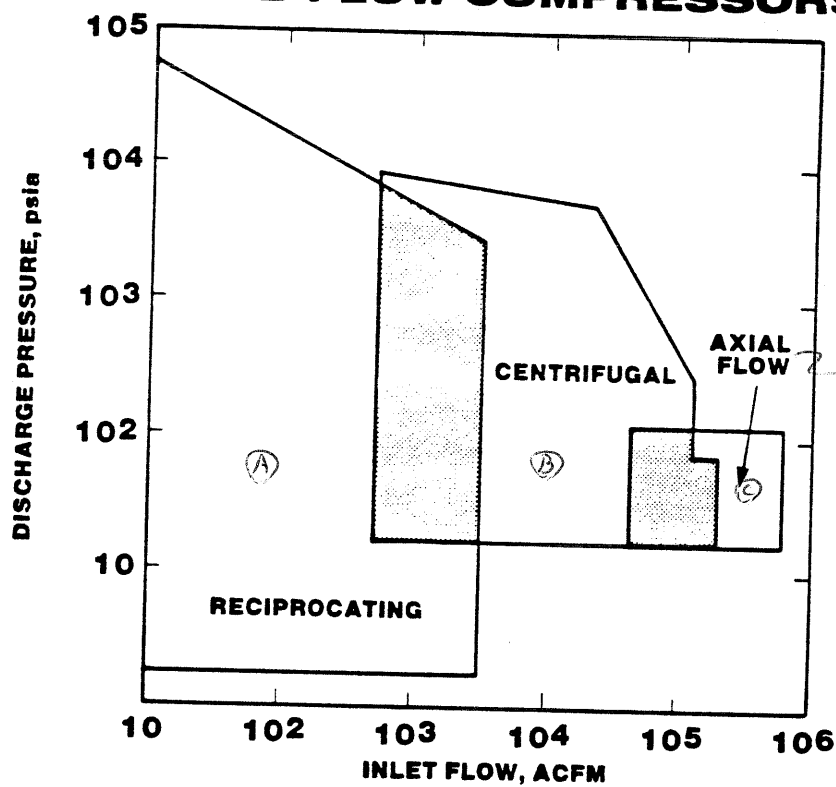
COMPRESSORS

**CHOOSING THE RIGHT COMPRESSOR -
RECIPROCATING, CENTRIFUGAL, OR AXIAL**

Engineering Seminar

M. J. Wilcox

FIGURE 1
**APPROXIMATE RANGES OF APPLICATION
 FOR RECIPROCATING, CENTRIFUGAL
 AND AXIAL-FLOW COMPRESSORS**



UOP 537A-2

A:

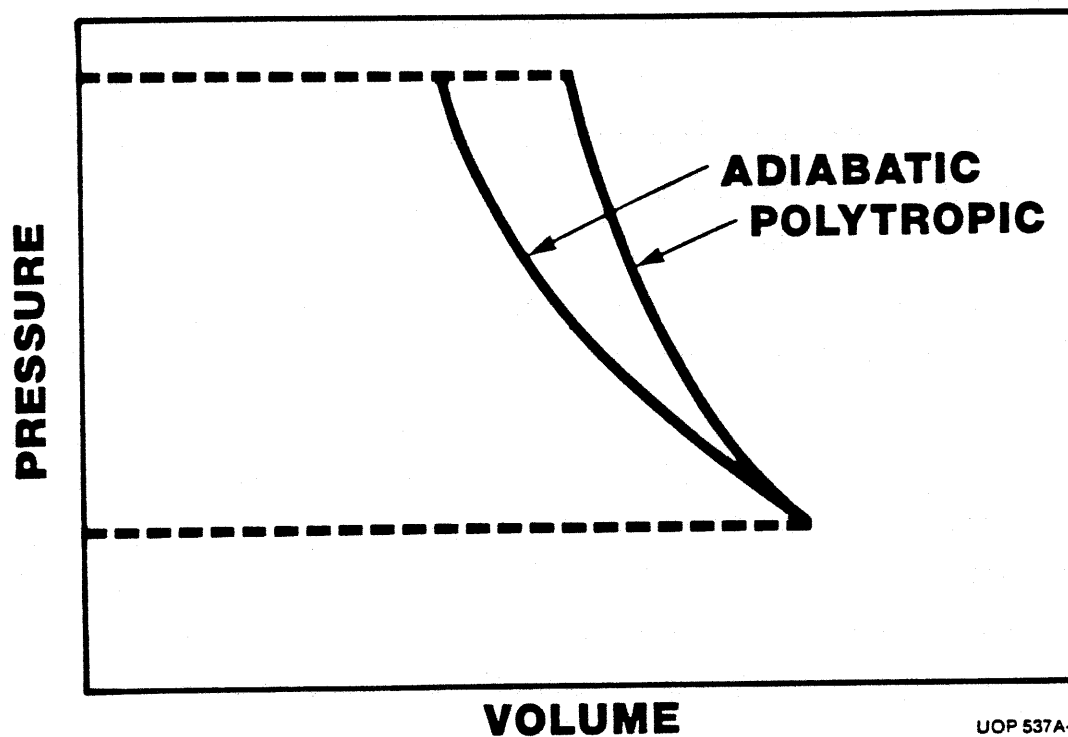
C: H₂ + MIXER

CHOOSING THE RIGHT COMPRESSOR - RECIPROCATING, CENTRIFUGAL, OR AXIAL

Reciprocating compressors have been compressing gases for the past one hundred years or so with the same basic design as is used in refineries today. Naturally the slow speed horizontal steam engines would have been inappropriate in handling the 25,000 BHP/50,000 psig polyethylene service in today's plant, but the basic drive train and thermodynamic principles still apply. In the mid 1940's, technology met industries need for a smooth flow large capacity device with the development of the centrifugal machine, which has been the dominant force in compression applications since that time. We owe to the jet engine/gas turbine age a third type of compressor, an axial flow machine. Axial compressors were used in wind tunnels and for other research ventures in the mid to late fifties, but it took the space program, during the 1960's, to fine tune the blade aerodynamics to allow the 85% and higher efficiencies which are achievable today.

At the same time that centrifugal and axial compressors were coming of age, UOP was at the forefront of petroleum processing technology, employing these machines in new and larger processes with increasing pressures, capacities, and potential gas corrosion and toxicity. At various steps in the development of these compressors, UOP technology was a force giving direction to the maturation process. For example, the first barrel type compressor applied to a petroleum process was made by Dresser Clark for a UOP Platformer. Also, the largest barrel type compressor made up until 1970 was manufactured by Elliott Company for a UOP Pacol Unit. In addition, UOP has been involved with the manufacturers in solving many of the problems which the ever increasing severities of our processes demand, such as stress corrosion cracking and high pressure seal technology. UOP continues to participate in the race of compressor development versus process technology advances as a member of the American Petroleum Institute subcommittee on mechanical equipment.

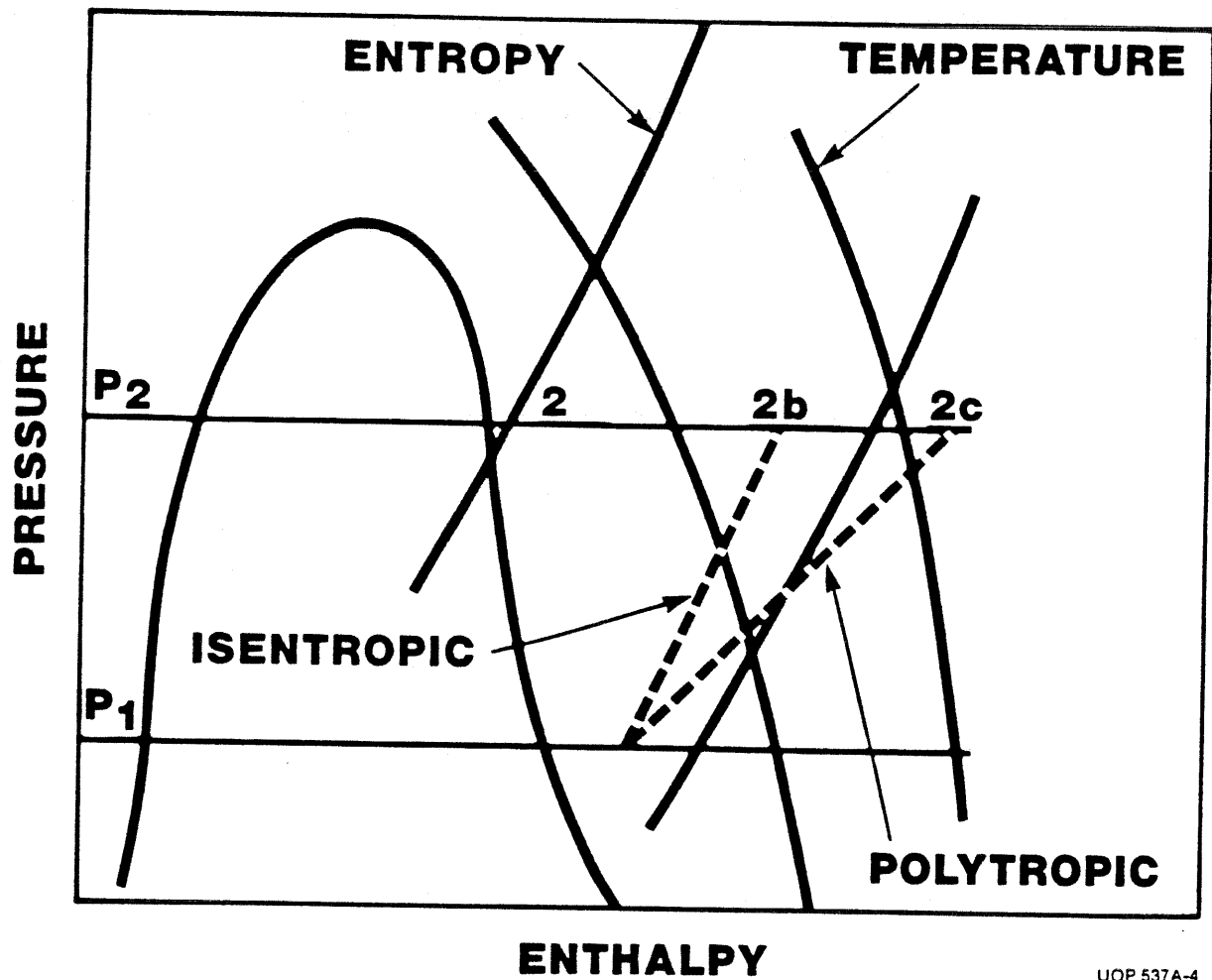
FIGURE 2
PROCESS OF COMPRESSION
DEFINED ON PV DIAGRAM



UOP 537A-3

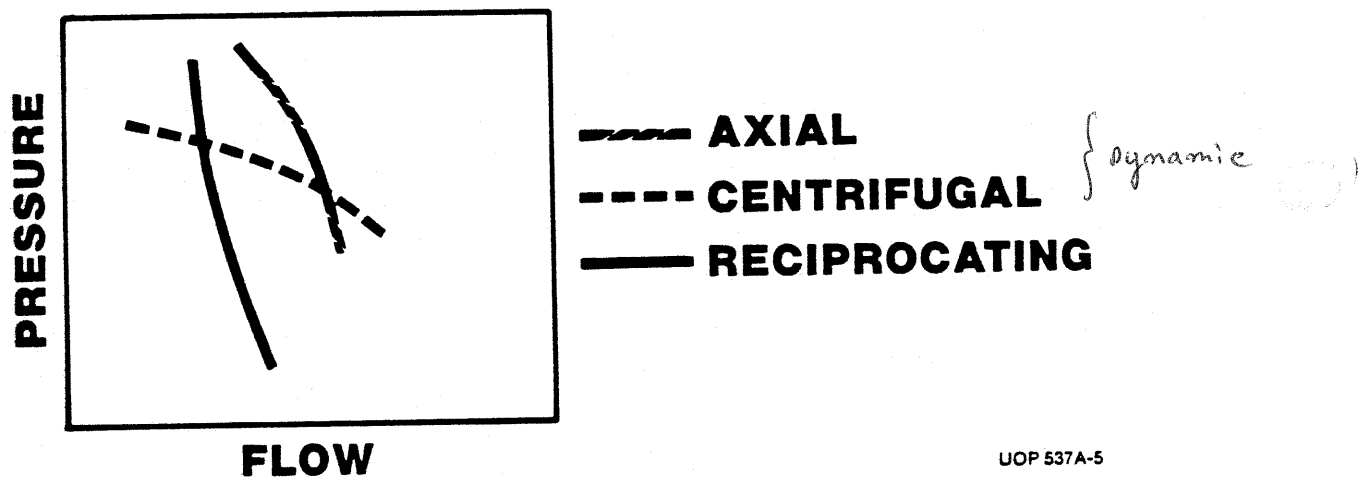
GAS always high T/P .
 so, ADIABATIC is HP. PV is T constant.

FIGURE 3
TYPICAL MOLLIER DIAGRAM



UOP 537A-4

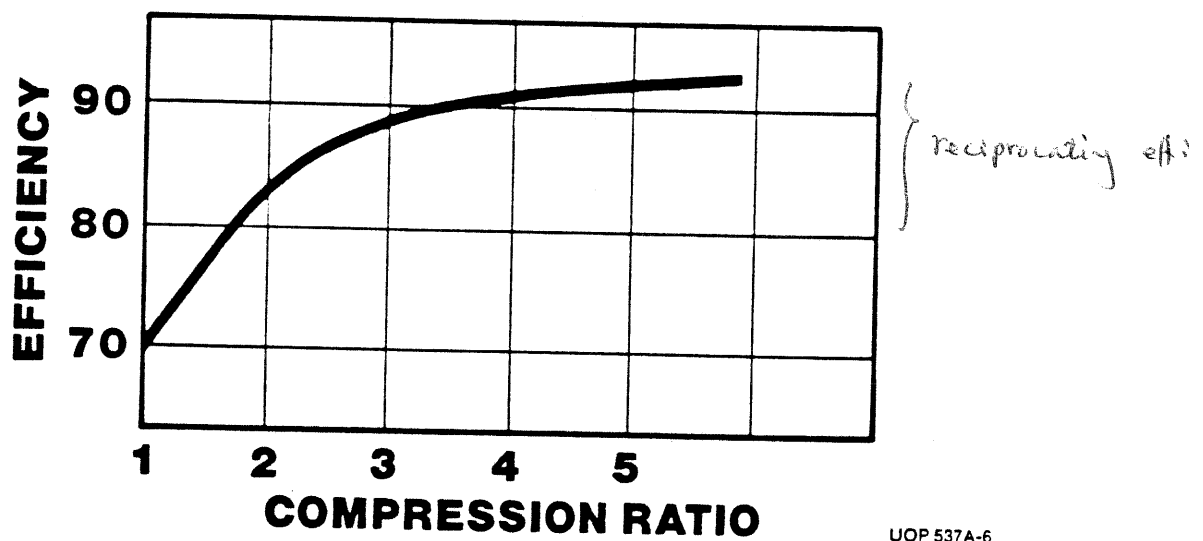
FIGURE 4 COMPRESSOR PERFORMANCE CHARACTERISTICS



UOP 537A-5

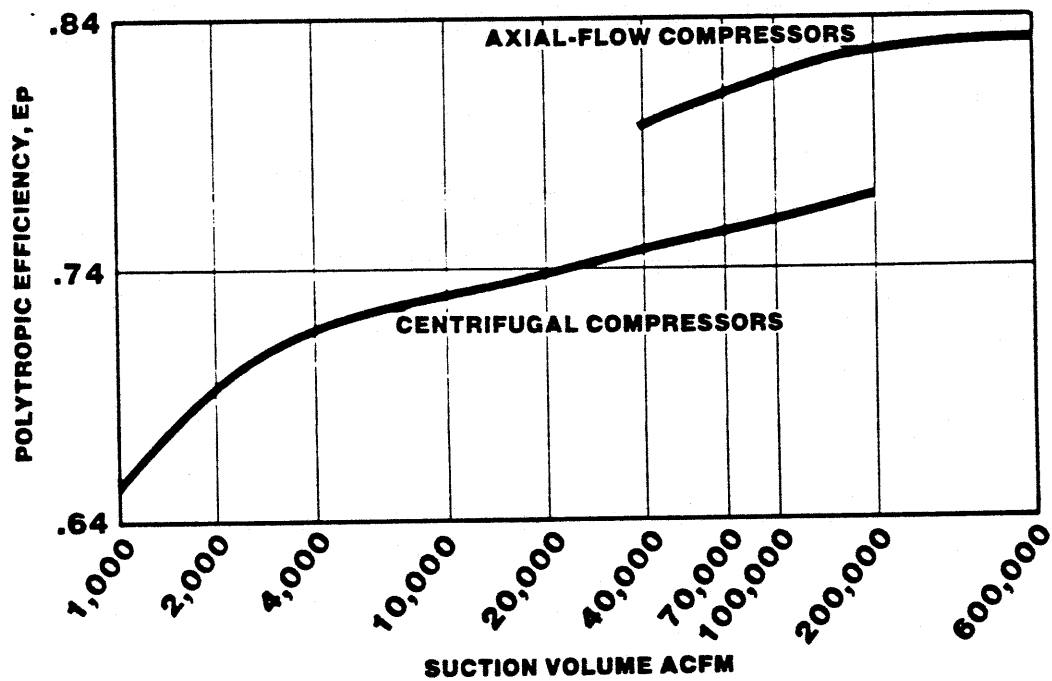
Reciprocating : Fix Volume .
 Centrifugal : Different Volume, Fix pressure .

FIGURE 5
RECIPROCATING COMPRESSOR
EFFICIENCY



UOP 537A-6

FIGURE 6
APPROXIMATE POLYTROPIC EFFICIENCIES FOR
CENTRIFUGAL AND AXIAL-FLOW COMPRESSORS



UOP 537A-7

This chart is to help you know which is more efficiency one.
 ex) 100,000 ACFM is Axial / Cent의 효율이 높음.

The development of the three basic types of compressors mentioned satisfied the needs of industry. However, another result of having different types of machines available is that, for a given application, many times a thorough study is required to insure that the optimum compressor type is chosen. As can be seen in Figure 1, two of the primary variables in making the selection are flow and pressure. The overlapping areas, however, are large. Depending upon utility costs, particulars of the application, operator philosophy and overall economics, the best choice for a given application can be determined.

THERMODYNAMICS

A reciprocating compressor moves a volume of gas from one pressure level to another and is best modeled as an adiabatic compression device. Centrifugal and axial compressors convert velocity head into pressure head, thereby utilizing a dynamic characteristic rather than the positive displacement mode which reciprocating machines employ. See Figures 2 and 3 for a pressure - volume diagrams and a Mollier diagram for the reciprocating compressor's isentropic (adiabatic and reversible) path and the polytropic path of the centrifugal and axial compressors. Note that the polytropic compression path consumes more enthalpy, owing to the higher hydraulic losses incurred during velocity head to pressure head conversion.

CHARACTERISTIC CURVES

Figure 4 illustrates the characteristic curves for the three subject compressors. Note that the centrifugal machine would provides a fairly constant pressure for a range of flows. Conversely, the reciprocating machine curve compresses a fairly constant volume of gas for a range of pressures. The axial compressor curve is a compromise between the reciprocating and centrifugal characteristics.

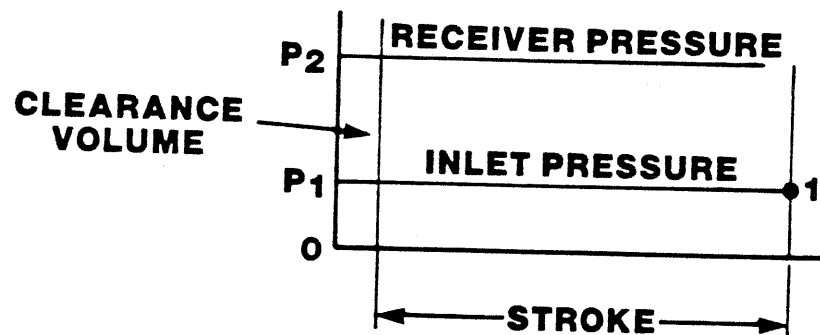
EFFICIENCY

Figures 5 and 6 show average efficiencies for the compressor types. Note that reciprocating compressor efficiency is dependent upon materials, cylinder size, valve design, compression ratio, and other variables to a lesser degree. As Figure 6 illustrates, an axial compressor is typically 6 to 8 points more efficient than a centrifugal compressor at the same flow. Since a large portion of the inefficiency in centrifugal and axial compressors is due to hydraulic losses, the gas flow path surface area to volume ratio is the best parameter to compare to relative efficiency. Therefore, a low volume machine with narrow passages and small impellers would have a high surface area to volume ratio and high hydraulic losses. Predictably, small centrifugals are not very efficient.

RECIPROCATING COMPRESSOR BASIC THEORY

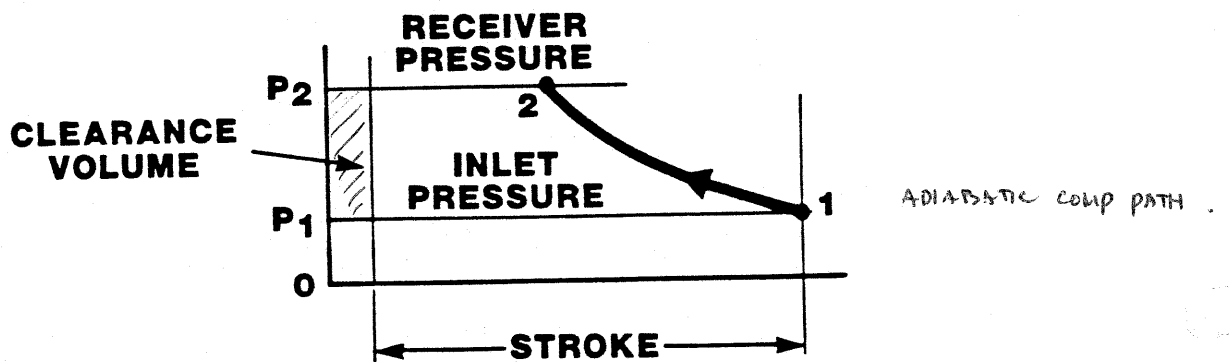
See Figures 7-11 for step by step diagrams of the pressure-volume to piston location relationships. Remember that, while an internal combustion engine is in many ways similar to a reciprocating compressor, there is a basic difference - your automobile engine has a fixed compression ratio due to mechanically timed valves while a reciprocating compressor uses automatic spring loaded valves which open only when the proper differential pressure exists across the valve. Inlet valves open when the pressure in the cylinder is slightly below the intake pressure. Discharge valves open when the pressure in the cylinder is slightly above the discharge pressure.

FIGURE 7
RECIPROCATING COMPRESSOR
BASIC THEORY



UOP 537A-24

FIGURE 8 **COMPRESSION**



UOP 537A-25

CLEARANCE volume : HOT TEMP .

LARGE vol : CAPACITY ↓
COST ↑

FIGURE 9
DISCHARGE

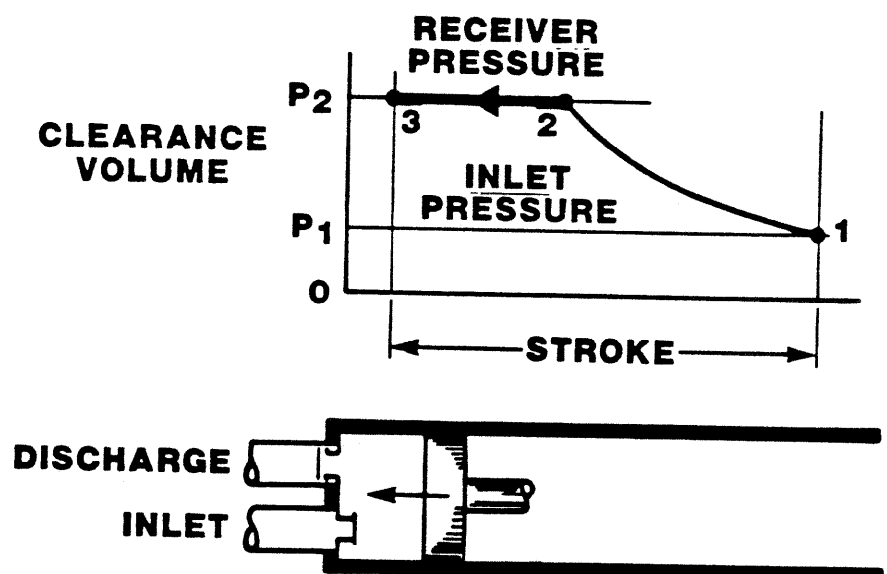
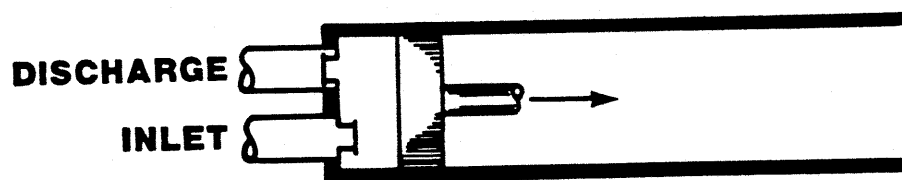
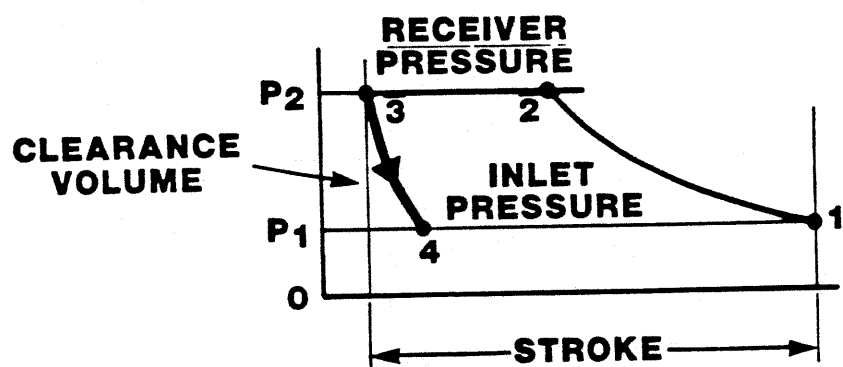
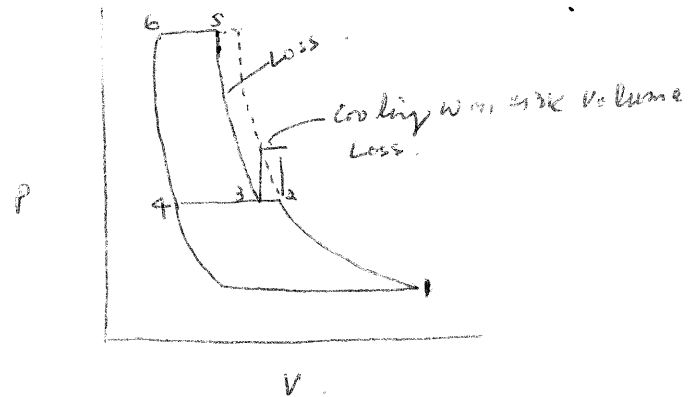


FIGURE 10
EXPANSION



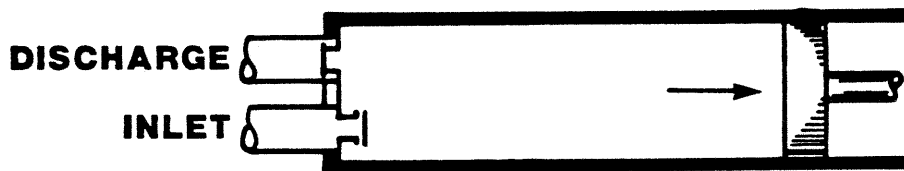
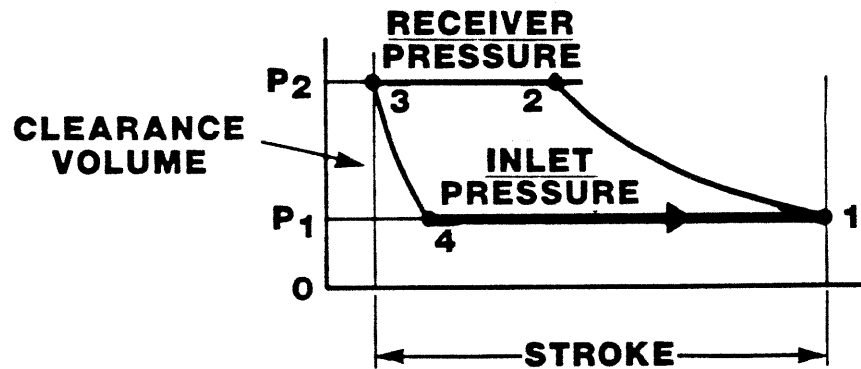
UOP 537A-23

Whe



**FIGURE 11
INTAKE**

TWO STAGE COMPRESSOR



UOP 537A-22

Inside material : peek ← plastic material
 { corrosion m. 75 m. (corrosion resistant)
 High Temp m. 75 m.
 peek
 in m. change in 2.5 m. 50 m.

CENTRIFUGAL COMPRESSOR BASIC THEORY

A centrifugal compressor is a dynamic machine - it does not move gas to a higher potential, but gives the gas a certain amount of potential or head. A centrifugal compressor will only develop a certain amount of pressure and, therefore, does not usually need a relief valve to avoid self-destruction in the event it is isolated. From Figure 12, it can be seen that at each impeller, gas enters with a given velocity C_1 . The impeller vane has a velocity at the inlet of U_1 and the resultant velocity of the gas relative to the impeller is V_1 . The same three velocities can be detailed at the discharge. Using vectoral analysis, the characteristic performance curve shape, surge, and stonewall can be explained.

A centrifugal compressor polytropically compresses a gas, with the resultant horsepower and discharge temperature somewhat higher than an adiabatic compression would develop. Figure 13 shows where the losses occur and gives a typical range for each.

Centrifugal horsepowers can be calculated from the following equation:

$$\text{BHP} = \left[\frac{(wZRT)(r^{n-1}/n-1)(n/n-1)}{(33,000N_p)} \right]$$

$$\text{where } \frac{K-1}{k N_p} = \frac{n-1}{n}$$

N_p = polytropic efficiency

R = 1545/mw

Z = average compressibility

T = inlet temperature

r = compression ratio

w = lbs per minute

FIGURE 13
CENTRIFUGAL COMPRESSOR LOSSES

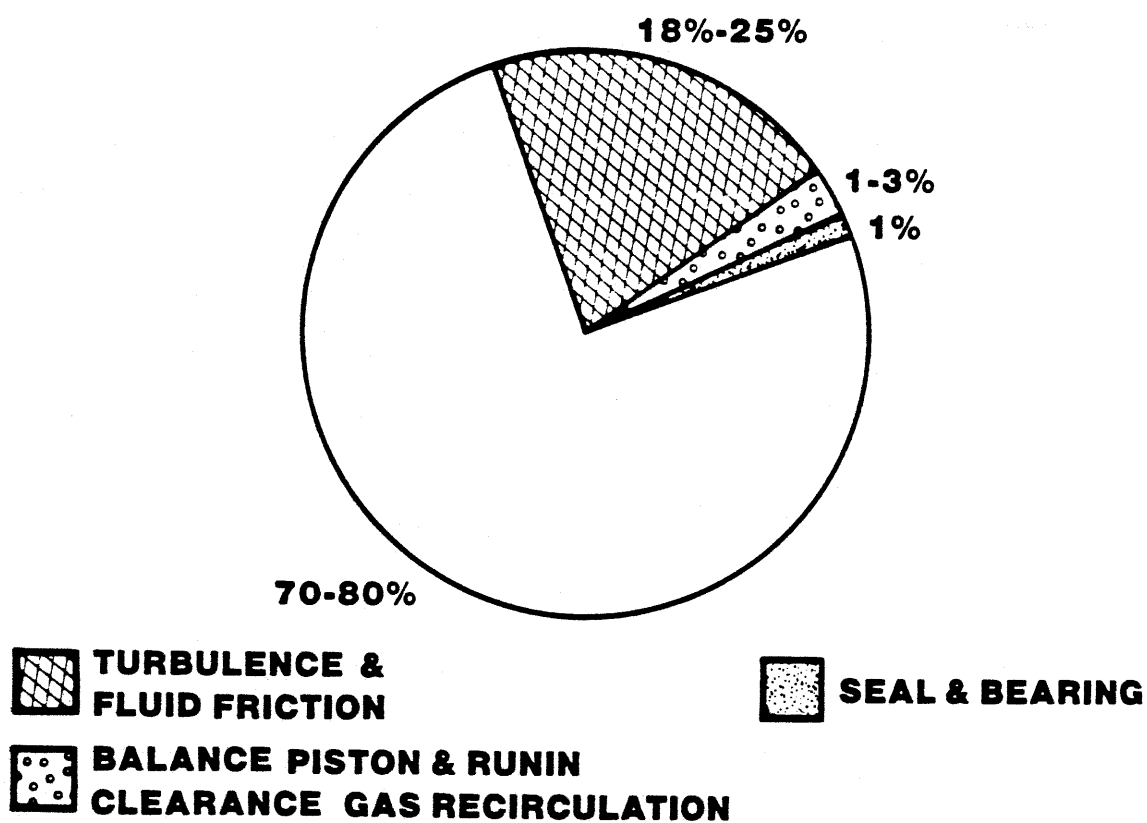
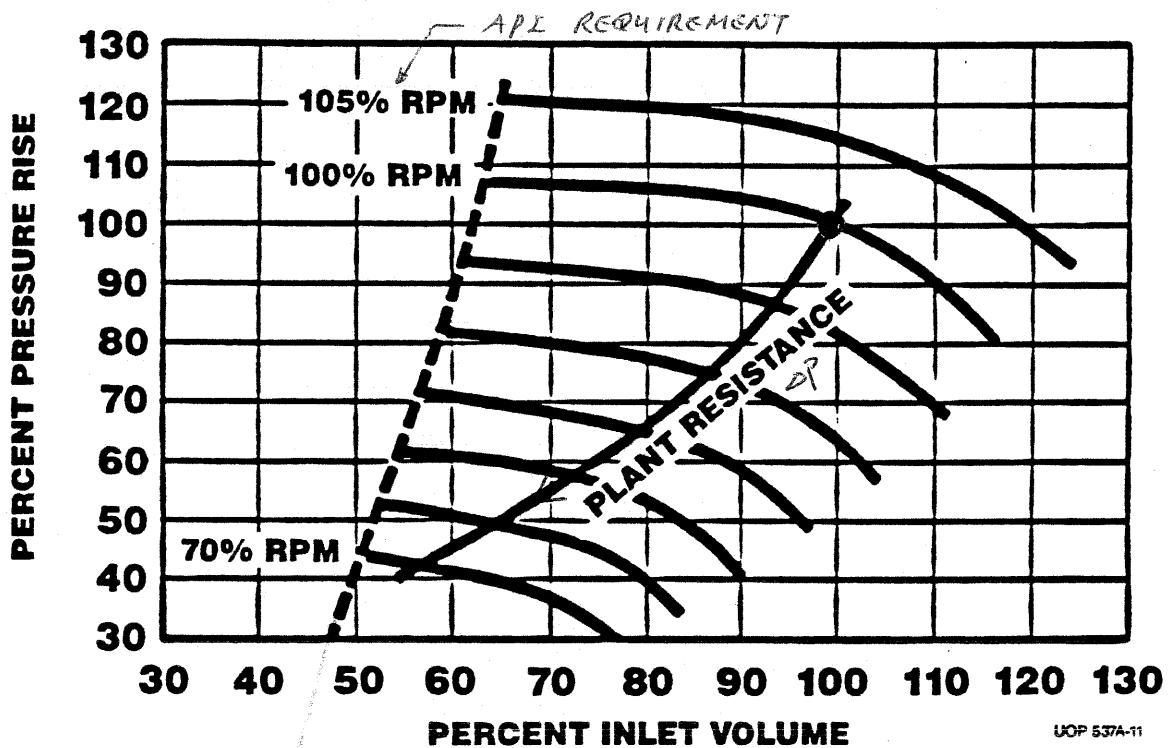


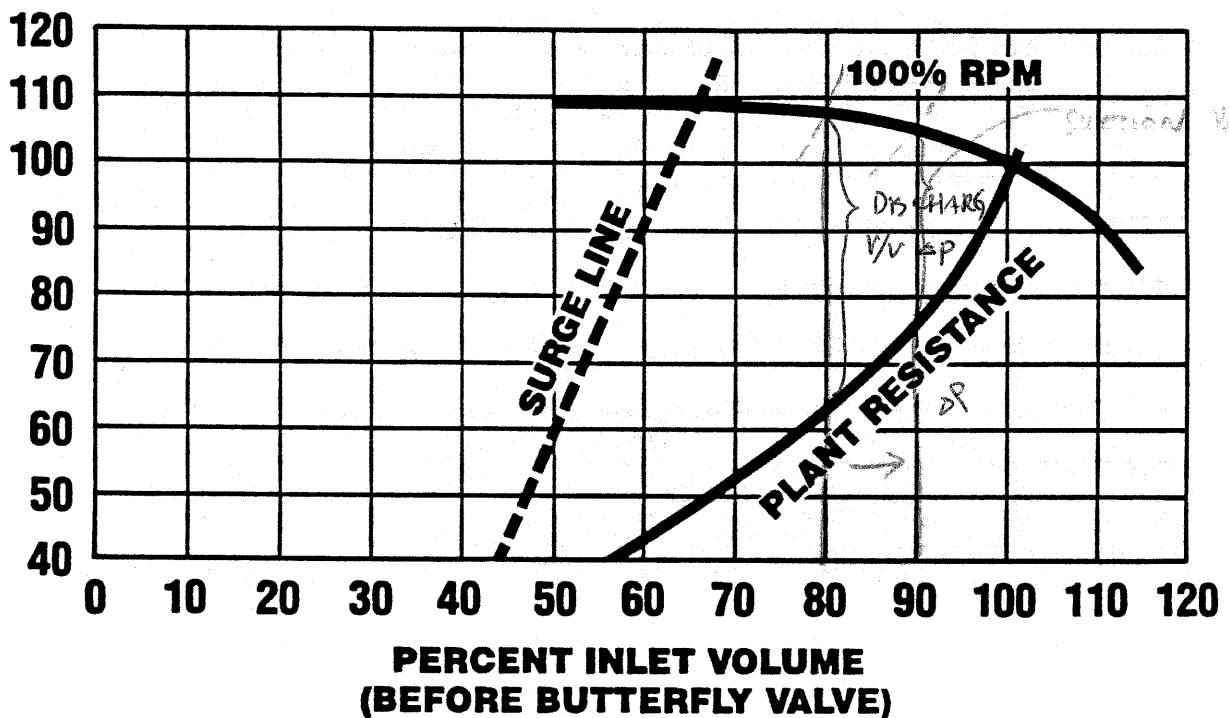
FIGURE 14

CENTRIFUGAL COMPRESSOR TYPICAL VARIABLE SPEED PERFORMANCE CURVE



①. SPEED CONTROL of 7/28 Compressor Control on 7/28

FIGURE 15
CENTRIFUGAL COMPRESSOR
TYPICAL CONSTANT SPEED
PERFORMANCE CURVES



UOP 537A-10

AIR COMPRESSOR

CENTRIFUGAL COMPRESSOR CONTROL

Due to its smooth flow characteristics, fairly regular curve over a wide capacity range, and well defined critical speed regions, a centrifugal machine is easy to control. The following three modes are used:

1. Variable Speed - As can be seen on Figure 14, a variable speed driver or transmission will allow the compressor to satisfy plant operation at a wide range of flows and pressures.
2. Throttling - A throttle valve can be used to alter the plant resistance curve such that the point of operation can be varied to suit process needs. Suction throttling is normally preferred because less pressure and, therefore, less horsepower is required to be throttled unlike a valve on the discharge. In addition, the throttled suction results in an increased acfm at the machine inlet which normally results in a higher efficiency. See Figure 15 for a fixed speed centrifugal compressor curve.
3. Bypass - After the above two options have been considered, a cooled bypass will allow infinite flexibility.

Depending upon the characteristics of the process, the control scheme may be required to protect the machine against operation at too low (surge) or too high (stonewall or choke) a flow.

CENTRIFUGAL COMPRESSOR ADVANTAGES

1. High Reliability, Low Maintenance - Centrifugal compressors are currently running five years and longer without being opened for inspection. Vibration monitoring techniques now allow planned shutdowns and a fairly predictable maintenance plan.

Most of this success story is due to improved quality control by the manufacturer, more rigid and thorough standards, higher degree of competence by users, and mandatory mechanical and performance testing.

2. **Large Volume Capability** - A four to six foot diameter barrel can handle the same capacity that an eight to ten cylinder reciprocating compressor, consuming about 1000 square feet of floor space, could handle.
3. **Liquid and Dirt** - A centrifugal does not have the valves and sliding seals of a reciprocating compressor and can handle small amounts of dirt and liquid without being damaged. In fact, many refineries have used on-stream washing to clean up rotors which have become unbalanced due to a non-uniform deposit, such as salts or tars. However, a large slug of liquid or solid objects, such as a bolt, will most likely cause extensive damage.
4. **Smooth Flow** - The characteristically smooth flow allows for good flow measurement and control, minimal foundation requirements, and simplified piping support.
5. **Capacity Range** - For a process which requires a wide capacity range at a fairly constant pressure, the centrifugal is well suited.
6. **Process Gas Contamination** - Seals are designed such that seal oil which comes into contact with process gas is diverted from the process. This small quantity is typically routed to the flare from waste oil traps.
7. **Compatibility With Driver** - A centrifugal compressor does not have the torsional problems of a reciprocating machine. The torque - speed relationships are compatible with motors and turbines.

CENTRIFUGAL COMPRESSOR DISADVANTAGES

1. **Molecular Weight Range** - A centrifugal compressor must be designed in consideration of the full molecular weight range. Many times the polytropic head requirement for the lowest molecular weight is much larger than for the heaviest. The result is a less than efficient operation at the high molecular weight operation. A variable speed driver can accommodate molecular weight variance more efficiently than a fixed speed machine which must throttle excess head. Sometimes the level of confidence in the predicted molecular weight is low and, in that event, a centrifugal's limitations could be costly.
2. **Low Flow Inefficiency** - Great strides have been made in the past five years to improve the efficiency of the centrifugal machines under 1000 acfm. Even so, a reciprocating machine will probably consume 20% to 30% less power for these applications.
3. **Low Flow Limits** - Approximately 400-600 acfm at the last stage is required in order for a centrifugal machine to be viable. At the low flows, the centrifugal cannot develop the head per stage that the larger impellers can; therefore, the result is lighter and longer rotors which have had bearing stability problems.
4. **Surge Potential** - While some centrifugal machines can repeatedly surge for short durations without damage, many suffer extensive damage after surging for only a short time. Any time process requirements demand compressor operation at a point which is anticipated to be a surge area, the control scheme must protect the machine to avoid surge.

5. **Low Molecular Weight Limitations** - While a reciprocating compressor can attain a compression ratio of about 3:1 on a 2 to 4 mw gas, a centrifugal would need two casings and 15-20 impellers to accomplish the same ratio.
6. **Sophistication** - Centrifugal compressors require a much stronger understanding and more complex capabilities than do reciprocating machines. In the last five to ten years, however, most areas of the world have gained the exposure and experience necessary to operate and maintain high speed rotating equipment.

COMPRESSOR CASE STUDY I

SMALL CENTRIFUGAL VS. RECIPROCATING

SERVICE - Crude Unit Off Gas

DRIVER(S) - Electric Motor

Comparison is between one 100% motor and gear driven centrifugal and two 100% motor and gear driven reciprocating compressors.

GAS COMPOSITION (MOL PER CENT)

Water Vapor	4.82
H ₂ S	1.09
C ₁	3.77
C ₂	19.99
C ₃	32.77
C ₄	<u>37.56</u>
	100.00

MOLECULAR WEIGHT 44.0

PROCESS REQUIREMENTS

MMSCFD	1.77
ACFM	1239
T ₁ , °F	100
P ₁ , PSIA	15.7
P ₂ , PSIA	69.7
P	54
COMP. RATIO	4.44
C _p /C _v	1.115
ZAVE	1.0
M, #/MIN	143
Polytropic Head, Ft.	32,540
Centrifugal BHP	200
Reciprocating BHP	175

COST ANALYSIS FOR ONE 100% CENTRIFUGAL COMPRESSOR

Item	Material	Man Hours	Lab. Cost	Subcontract	Total
Vessels and Internals	\$ 8,400	48			
Compressors and Drivers	510,000	270			
Instruments	20,600	330			
Electrical	-----	---		43,500	
Piping	36,600	3,670			
Insulation	-----	---		3,900	
Fireproofing	100	30			
Steelwork	1,900	70			
Concrete Work	700	200			
Painting	100	100			
Buildings	-----	---		20,000	
Project Productivity		909			
DIRECT FIELD COST	578,400	5,627	42,200	67,400	688,000
Indirect Field Costs	8,300		14,000	6,700	29,000
Labor Benefits			18,000		18,000
TOTAL FIELD COST	\$586,700		\$74,200	\$74,100	\$735,000

COST ANALYSIS FOR TWO 100% CENTRIFUGAL COMPRESSOR

Item	Material	Man Hours	Lab. Cost	Subcontract	Total
Vessels and Internals	\$ 8,400	48			
Compressors and Drivers	240,000	620			
Instruments	18,400	230			
Electrical	-----	---		66,300	
Piping	22,800	1,840			
Insulation	-----	---		2,400	
Fireproofing	100	30			
Steelwork	1,900	70			
Concrete Work	1,100	250			
Painting	100	100			
Buildings	-----	---		25,000	
Project Productivity		612			
DIRECT FIELD COST	292,800	3,800	28,500	93,700	415,000
Indirect Field Costs	5,500		9,500	6,000	21,000
Labor Benefits			14,000		14,000
TOTAL FIELD COST	\$298,300		\$52,000	\$99,700	\$450,000

SUMMARY

<u>Reciprocating</u>		<u>Centrifugal</u>
Total Field Cost	\$450,000	\$735,000
Delivery	12 months	14 months
Horsepower	175	200
Utility Penalty (3.5¢/KW-Hr)	\$1750/yr.	\$6,600/yr.
Maintenance	15 x 20 (each)	12 x 22 (including console)

The above study was done in 1975 for a Canadian grass roots refinery. Based on the economics, the reciprocating machine would be the obvious choice. UOP has ample operating experience with reciprocating compressors for this service which presents some difficulties for the reciprocating machine, namely a potentially large variation in flow, a dirty and wet gas and, in this case, a very harsh climate. The client elected to spend the extra money and buy the centrifugal compressor based on his desire to avoid reciprocating machinery.

COMPRESSOR CASE STUDY II
MEDIUM CENTRIFUGAL VS. RECIPROCATING

SERVICE - Platforming Net Gas Booster

DRIVER - Motor

Comparison is between one 100% motor/gear driven two casing centrifugal compressor train and two 60% single stage multi-cylinder slow speed motor driven reciprocating compressors.

GAS COMPOSITION (MOL PER CENT)

Hydrogen	85.27
Methane	3.72
Ethane	3.60
Propane	2.92
i-Butane	0.85
n-Butane	1.11
i-Pentane	0.62
n-Pentane	0.35
Hexanes ⁺	<u>1.56</u>
	100.00

MOLECULAR WEIGHT 7.84

Note: Molecular weight may vary from approximately 6.6 to 9.3

PROCESS REQUIREMENTS

MMSCFD	46.40
ACFM	4315
T ₁ , °F	100
P ₁ , PSIA	118.2
P ₂ , PSIA	318.9
P	200.7
COMP. RATIO	2.7
C _p /C _v	1.316
ZAVE	1.00
M, #/MIN	666
Polytropic Head, Ft.	130,000
Centrifugal BHP	(110% of normal process duty) 3900
Reciprocating BHP	(60% of normal process duty) Each 1780

COST ANALYSIS FOR ONE CENTRIFUGAL COMPRESSOR

MATERIAL

Equipment Setting	931,600
Piping	66,294
Civil	15,618
Structural Steel	0
Instrumentation	40,751
Electrical	29,817
Insulation	3,009
Paint	<u>331</u>
	1,087,418

LABOR

6231 Manhours	76,520
TOTAL INSTALLED COST	1,163,900
(Total Installed Cost w/Stm. Turb. Drive is 1,440,200)	

COST ANALYSIS FOR TWO RECIPROCATING COMPRESSORS

MATERIAL

Equipment and Setting	730,000
Piping	118,056
Civil	33,716
Structural Steel	0
Instrumentation	33,462
Electrical	2,500
Insulation	6,132
Paint	<u>522</u>
	924,388

LABOR

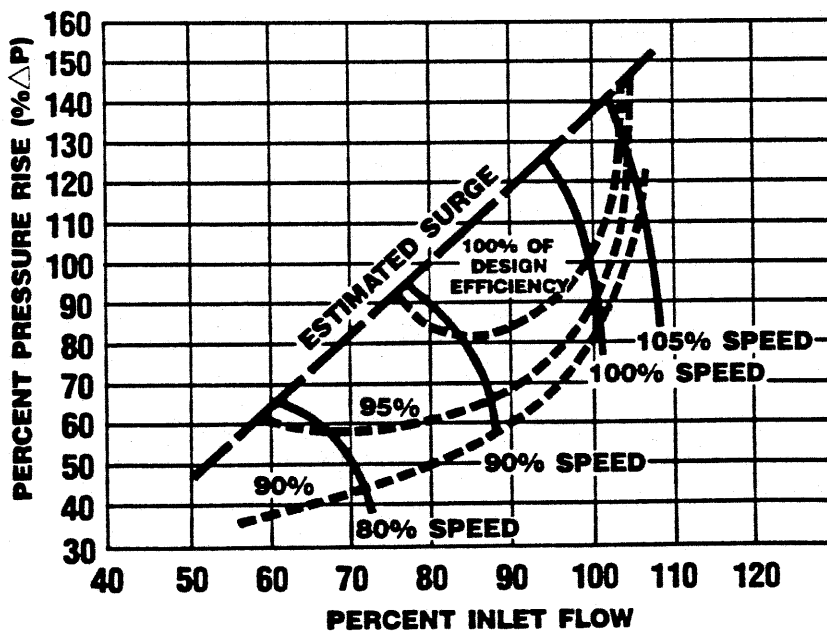
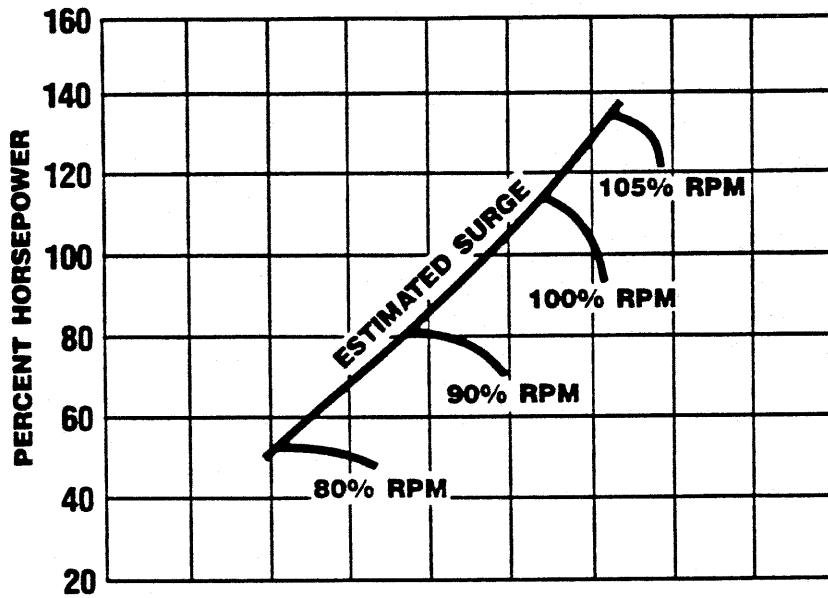
7444 Manhours	89,432
TOTAL INSTALLED COST	1,013,800

SUMMARY

	<u>Reciprocating</u>	<u>Centrifugal</u>
Total Field Cost	1,013,800	1,163,900
Delivery	12 months	12 months
Horsepower	3,560	3,900
Utility Penalty (3.5¢/KW-Hr)		77,766
Maintenance	\$18,000/yr	
Plot Area	20 x 30 (Each)	14 x 35

FIGURE 16

VARIABLE SPEED PERFORMANCE



Based on the above economics, it is a close call and the decision must be made in consideration of other factors. For the Platformer net gas booster service, a machine must have the ability to handle a range of molecular weights. Defining this range can be difficult or impossible. The compressor train must have sufficient flexibility to handle any flow and molecular weight which may result from the plant's operating severity, catalyst condition, feedstock, and octane number required, etc. A turbine driven centrifugal would have a better chance to cover all potential operating points fairly efficiently. The poorer the definition of "range of operation," the more attractive the reciprocating machine becomes. In most cases, UOP ultimately recommends reciprocating machinery for this service but, on occasion, the centrifugal has been chosen.

AXIAL COMPRESSOR BASIC THEORY

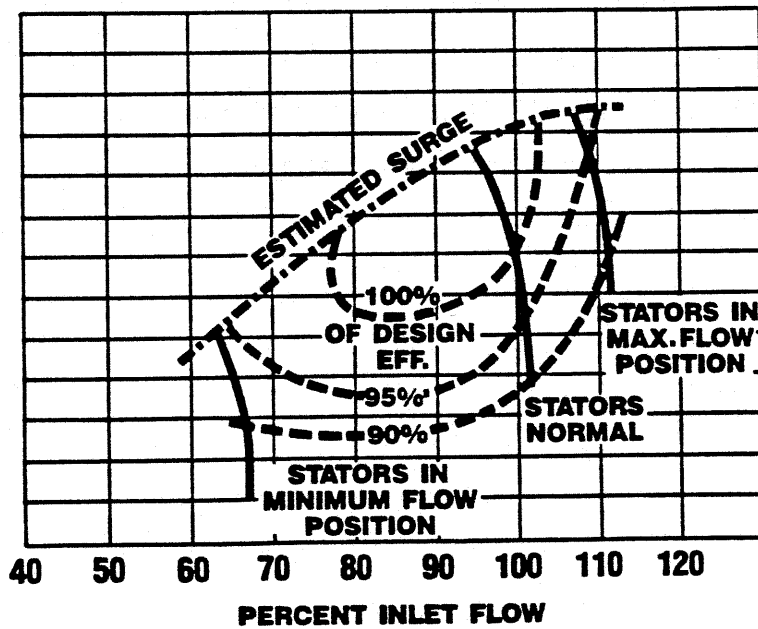
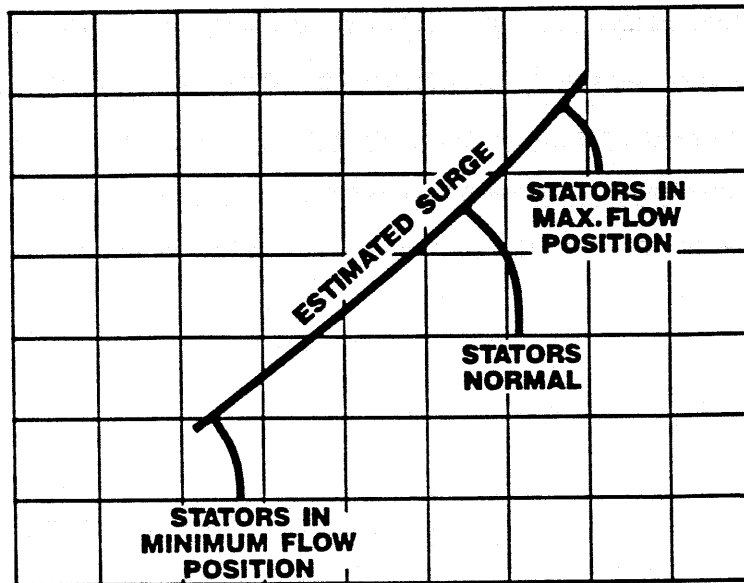
The basic compression path in an axial compressor is a polytropic path as with the centrifugal machine. The axial compressor accomplishes this through a series of rotating and stationary blades. The rotating blades in a given stage impart velocity energy on the gas and then 50 to 100% of the velocity head is converted to pressure. The remaining pressure rise, if any, occurs in the annular stage of stationary blades which follow. These stationary blades, or stators, then redirect the gas at the higher pressure to the next stage of rotating blades. An axial compressor currently should be considered only when compressing air or nitrogen. In the near future, however, sufficient research on other molecular weight gases will be completed.

AXIAL COMPRESSOR CONTROL

Axial compressors are fairly sophisticated pieces of machinery and require more sensitive and more thorough control for normal operation and surge protection than does the typical centrifugal compressor. The following are the available means of control:

FIGURE 17

VARIABLE STATOR PERFORMANCE 50% STATORS VARIABLE



1. Variable Speed - See Figure 16. As with centrifugal compressors, this is a very flexible means of control. With variable speed, an axial compressor has about the same flow flexibility and greater pressure flexibility relative to a centrifugal.
2. Variable Stators - See Figure 17. A much more efficient way of controlling a fixed speed axial compressor than suction throttling is accomplished by changing the flow path of the gas through the compressor. This is accomplished by having hydraulically actuated linkage which vary the stator blade angles and, thereby, is an efficient way of varying the performance. Normally about half of the stator stages would be movable. Sometimes variable speed and variable stators are used together to give maximum operational flexibility.
3. Bypass - Note, the snort on an FCC main air blower can allow the axial compressor to have infinite flexibility.

AXIAL COMPRESSOR ADVANTAGES

1. High Efficiency - As stated previously, the axial compressor is 6 to 8 points higher in efficiency and this can be a heavy utility advantage.
2. Smaller Relative Casing Size - Compared to a centrifugal compressor, the axial machine can handle enormous quantities of gas in a relatively small casing. For example, at approximately 100,000 acfm, the casing required for the centrifugal would be much larger than the axial casing. In this instance, the axial compressor would cost less.
3. Low Maintenance, High Reliability - Sufficient experience has been collected to prove that, with proper inlet filtering, axial machines are as reliable as centrifugals.

4. **Compatible Speed Match With Power Recovery Turbine** - In most cases, an axial blower and hot gas expander are both running closer to their optimum speeds than a centrifugal and expander would be. Putting a gear between the centrifugal and expander has been considered.
5. **Constant Volume, Wide Pressure Range** - The axial shares with the reciprocating machine a fairly steep pressure curve at a narrow volume corridor, but also has an acceptable volume range capability.
6. **Compatibility With Driver** - As with a centrifugal, an axial compressor is a nice match with a motor, steam turbine or gas turbine.
7. **Smooth Flow** - The axial shares the centrifugal's ability to provide a smooth flow for easy control and flow measurement. Most often the inlet casing throat is factory calibrated to provide a convenient and accurate flow measurement point.

AXIAL COMPRESSOR DISADVANTAGES

1. **Blade Fouling** - If the inlet filter is not doing an adequate job, the blades can fairly quickly erode in high velocity areas. A small change in blade profile will quickly decay performance.
2. **Sensitivity to Surging** - Axial compressors generally have high horsepowers, close clearances, and thin blades which soak up heat rapidly. Because of this, the surge control must be extremely sensitive. Normally the vendor provides his "black box" which will input parameters such as speed, inlet flow, differential pressure, and stator angle, etc., and quickly open the snort when a surge area has been encroached upon.

3. **Sophisticated Technology** - Although they are highly reliable, when analysis of the gas path is required to determine what is going wrong, experts are required. The basic rotor and bearing support system is fairly standard.
4. **Cost** - Axial compressors can cost about \$200,000 to \$500,000 more than a comparable centrifugal compressor.
5. **Handling Hydrocarbons** - To date, there has been very little experience in handling gases other than air and nitrogen. The confirmation that other gases behave in a predictable manner with no new considerations will open some new doors. Propane dehydrogenation is a process which is waiting for this development.

COMPRESSOR STUDY III

LARGE CENTRIFUGAL VS. AXIAL

SERVICE - FCC Main Air Blower

DRIVER - Electric Motor

Comparison is between one 100% steam turbine centrifugal compressor and one 100% steam turbine driven axial compressor.

GAS COMPOSITION

50% Relative Humidity

100% Atmospheric Air

28.6 Molecular Weight

PROCESS REQUIREMENTS

MMSCFD	108.77
ACFM	81262
T ₁ , °F	88
P ₁ , PSIA	14.4
P ₂ , PSIA	50.7
P	36.3
COMP. RATIO	3.521
C _p /C _v	1.4
ZAVE	1.0
#/MIN	5693
Polytropic Head, Ft.	46386
Centrifugal BHP	10400
Axial BHP	9800

COST ANALYSIS FOR ONE CENTRIFUGAL AIR BLOWER

MATERIAL

Equipment and Setting	\$3,401,500
Piping	253,740
Civil	47,186
Instrumentation	99,547
Electrical	1,365
Insulation	2,430
Paint	<u>4,212</u>
	\$3,809,976

LABOR

18762 Manhours	\$228,177
TOTAL INSTALLED COST	\$4,038,200

COST ANALYSIS FOR ONE AXIAL AIR BLOWER

MATERIAL

Equipment and Setting	\$3,665,000
Piping	232,500
Civil	39,000
Instrumentation	124,000
Electrical	1,400
Insulation	2,100
Paint	<u>3,500</u>
	\$4,067,500

LABOR

17500 Manhours	\$ 212,800
TOTAL INSTALLED COST	\$4,280,300

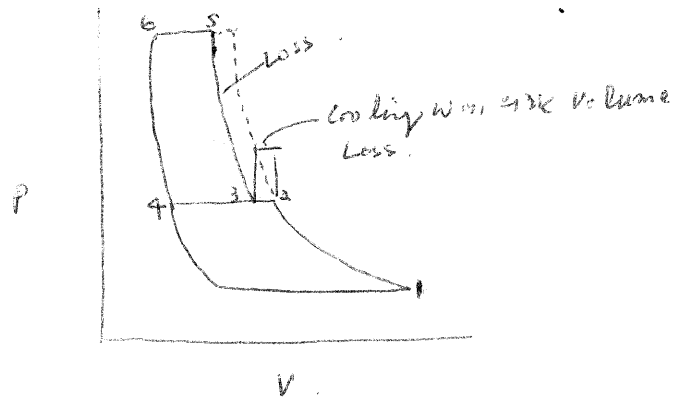
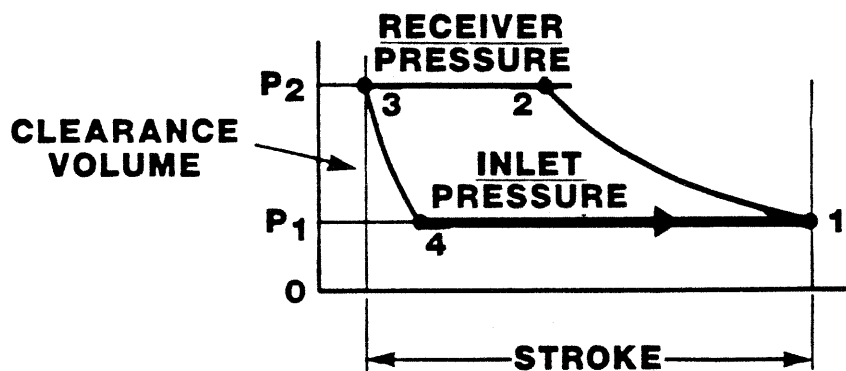


FIGURE 11
INTAKE

TWO STAGE COMPRESSOR



UOP 537A-22

Inside material : peek ~~←~~ plastic material
 { corrosion in 75% (corrosion resistance)
 High Temp in 75%
 peek
 change in 25% in 25%

One of the fundamentals in designing a reciprocating compressor for a given application is consideration of the volumetric efficiency which is equal to:

$$\text{Volumetric Efficiency} = \frac{\text{Actual Capacity}}{\text{Piston Displacement}}$$

Volumetric Efficiency (%) = 100 - (r^{1/k} - 1) c - L where:

r = compression ratio

k = C_p/C_v

c = clearance in the cylinder

L = practical correction for particular application
(typically 37%)

The clearance is equal to all of the volume remaining at the end of the cylinder when the piston is at the end of its stroke. As the number of valves increases, the amount of clearance increases, resulting in a lower volumetric efficiency.

To compute horsepower required for a given reciprocating compressor application, one could calculate the adiabatic horsepower and correct for the appropriate inefficiencies.

$$\text{BHP} = \left[\frac{ZRT (r^{k-1}/k - 1)(k/k - 1)}{3300mN_{AD}} \right]$$

However, since the valve pressure drops, frictional losses and inlet gas preheating, etc., should be considered, this approach has been substituted with a greatly simplified one. Empirical BHP versus compression ratio and specific heat ratio curves have been developed which can be used to easily determine the horsepower requirement for a given application.

Reciprocating compressors are inherently low volume, high compression ratio machines. When the acfm is approximately 2000 or more, a centrifugal compressor should be considered, even if two casings are required due to high compression ratio and/or low molecular weights. Reciprocating machines should be limited to approximately 300°F discharge when handling hydrocarbon gases and about 250°F discharge when handling significant quantities of oxygen.

RECIPROCATING COMPRESSOR CONTROL

There are basically three types of control which are commonly employed when using reciprocating compressors, namely:

1. **Suction Valve Unloaders** - One end of a given cylinder is completely unloaded when the suction valves (or separate unloading valves) are manually or automatically held open. Therefore, a two cylinder machine with double acting cylinders could be unloaded to 75, 50, 25, or 0% of rated machine capacity. Note that every two cylinder machine does not have this capability due to resultant torsional affects and bearing lubrication considerations. If unloading capability is desired, this requirement must be considered during selection and design of the machinery. UOP requires reciprocating machines to come equipped with suction valve unloaders on a fully unloaded start-up.
2. **Cylinder Pockets** - Fixed and variable pockets are available. These devices change the effective clearance (and therefore volumetric efficiency) and are viable for services with approximately 2:1 compression ratios and greater. Variable pockets are only used occasionally and at low pressure due to poor field experience at higher pressures and on wet or dirty gases. Almost all of our booster gas services use fixed pockets which are sized to accommodate a reduced capacity operation.

3. **Bypass** - After the above two means of reducing capacity have been considered, any additional gas may be bypassed. This means, of course, no saving on horsepower. Many UOP processes utilize bypass control to hold pressure on the vessel which the compressor is drawing gas from. Note that having multiple machines for a given service is a convenient way of accomplishing a turndown capacity requirement.

- Spillback -

Although variable speed has been used for capacity variation on occasion, this is not a safe and reliable way of operating a reciprocating compressor train. Due to the numerous critical speeds and large amount of pulsating energy, it is too easy to destroy the driver, possible gear reducer or compressor.

RECIPROCATING COMPRESSOR ADVANTAGES

1. **High Compression Ratios** - Due to the positive displacement characteristic, reciprocating compressors are capable of extremely high compression ratios and pressure levels. They are limited by the resultant discharge temperature (best approximated by assuming adiabatic compression) and mechanical strength of the drive train and cylinder. Polyethylene machines are today compressing this material to 50,000 psig, at which point the fluid behaves more like a liquid than a gas.
2. **Constant Volume, Variable Pressure** - As has been illustrated in the figure showing characteristic performance curves, a reciprocating compressor can be advantageous in its ability to deliver a fairly constant volume over a wide pressure range. However, due to increased slippage, inlet gas preheating, valve efficiency, and volumetric efficiency effects at higher compression ratios, less volume will be moved.
3. **Molecular Weight Flexibility** - Whether a low molecular weight material such as hydrogen or an extremely high molecular weight material is employed, this machine can achieve high compression ratios within the discharge temperature and mechanical

limitations. This has proven to be beneficial when handling a service which has a fairly wide molecular weight range due to diversity of feed stocks, plant operational flexibility, and catalyst activity changes, etc.

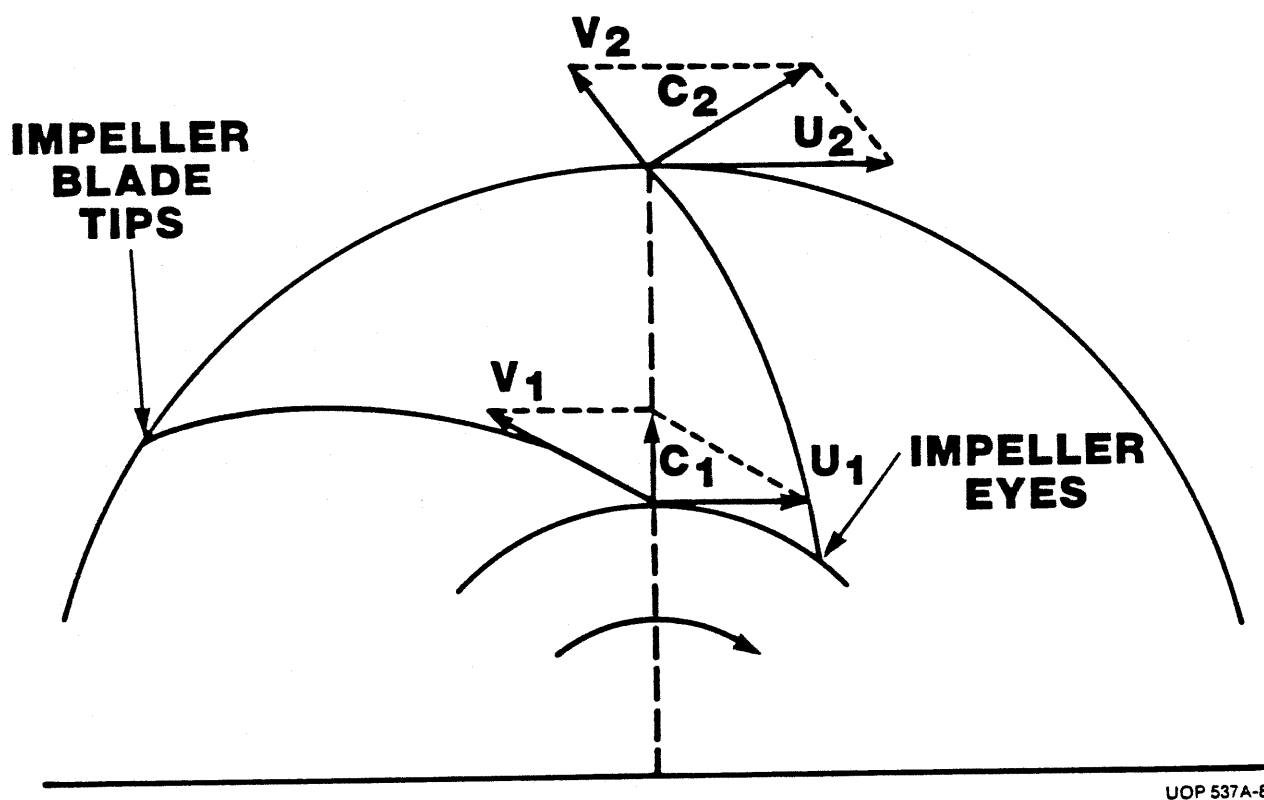
4. Basic Technology - Reciprocating compressors require sophisticated manufacturing and design techniques to be successful. However, once built and installed properly, they are more easily understood and maintained than the dynamic machines.
5. High Efficiency - Up until a few years ago, most users had an unflinching desire to use dynamic machines wherever possible. In today's world of high kilowatt hours, reciprocating compressors are winning a few more battles when the utilities are evaluated. The efficiency advantage is greater at lower flows and for services requiring a wide molecular weight range.

RECIPROCATING COMPRESSOR DISADVANTAGES

1. Foundation and Piping Requirements - Depending on the resultant unbalanced forces for a given application, the mass of concrete required for the foundation can be considerable. Due to the pulsation on the suction and discharge of the machine, pulsation suppression devices are required to dampen the pressure wave amplitudes which are damaging to the compressor valves and connecting piping. Piping must be sufficiently supported to avoid fatigue failures. At higher pressures, higher horsepowers, more stages of compression, and multiple machines, the analysis grows more complex and an analogue study is used to insure a satisfactory piping design.
2. Pulsating Flow - In addition to the complexity added to the piping design, the pulsating flow aggravates flow measurement. Also, the pulsations can be transmitted to other parts of the process. Exchangers in reciprocating compressor circuits must be designed in consideration of the potentially damaging pulse waves.

3. **Vulnerable to Dirt and Liquid** - Packing, rings, and valves are extremely sensitive to dirt and liquid. Prior to start-up, the gas circuit is acidized to remove mill scale and other dirt. Suction vessels and inlet lines are designed to minimize liquid entrainment and condensation.
4. **Maintenance** - This is an area which the user usually has a full appreciation for and requires multiple machines to improve plant on stream efficiency. A reasonably good rule of thumb for maintenance costs is \$35 per BHP per year.
5. **Plot Area** - These compressors usually take up a considerable area. There are more machines, and each one can be fairly large, requiring substantial clearances for maintenance.
6. **Large Volume Capability** - For very large volumes, the size, number and cost of cylinders makes these compressors unattractive.
7. **Lubrication Contamination** - While non-lubricated compressors have certainly earned their place, at high pressures and high compression ratios, the maintenance requirements can become atrocious. For some processes that do not permit lubricating oils, due to contamination effects, the non-lube alternative can be equally unattractive from a reliability standpoint.
8. **Torsional Implications** - Due to the characteristic pulsating torque for a reciprocating machine, the drive train and electrical circuit have to be protected. A flywheel is one way of smoothing out the torque demand. Coupling, gears, motors, and steam turbine have to be selected with careful consideration given to the particular compressor and its torsional characteristics.

FIGURE 12
CENTRIFUGAL IMPELLER
VECTORAL DIAGRAM



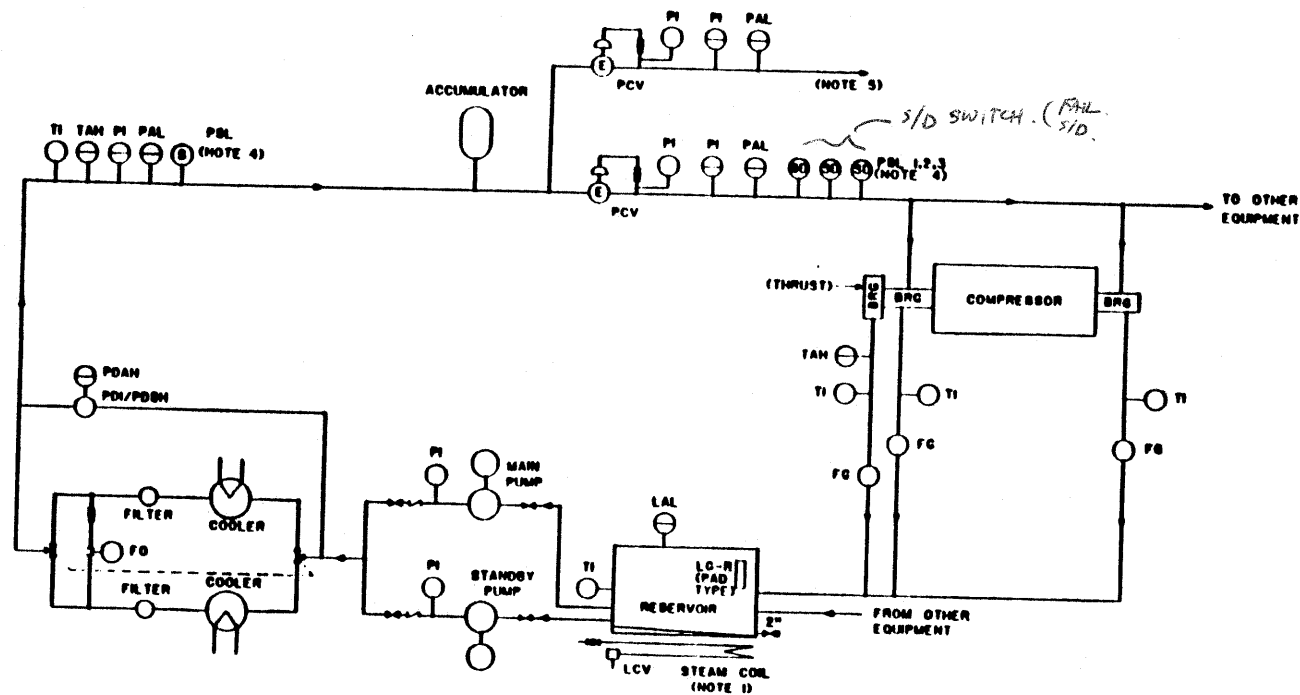
SUMMARY

	<u>Centrifugal</u>	<u>Axial</u>
Total Field Cost	\$4,038,200	\$4,280,300
Delivery	12 months	14 months
Stages	5	10
Horsepower	10400	9800
RPM	3000	5000
Utility Penalty (3.5¢/HW-Hr)	\$137,234/Yr	
Maintenance		\$30,000/Yr*
Machine Size	12 x 11 x 15	5 x 5 x 10

*Rotor differential cost over 5 years

The axial compressor is the best choice based on the above information. Most new FCC unit air blowers for units greater than about 20,000 BPSD will be axial in the future. Note the difference in size of the two machine types. Also note that the axial has twice as many stages and runs much faster. The utility cost is conservative to say the least - check what the penalty is at 6-8¢/Kw-Hr!

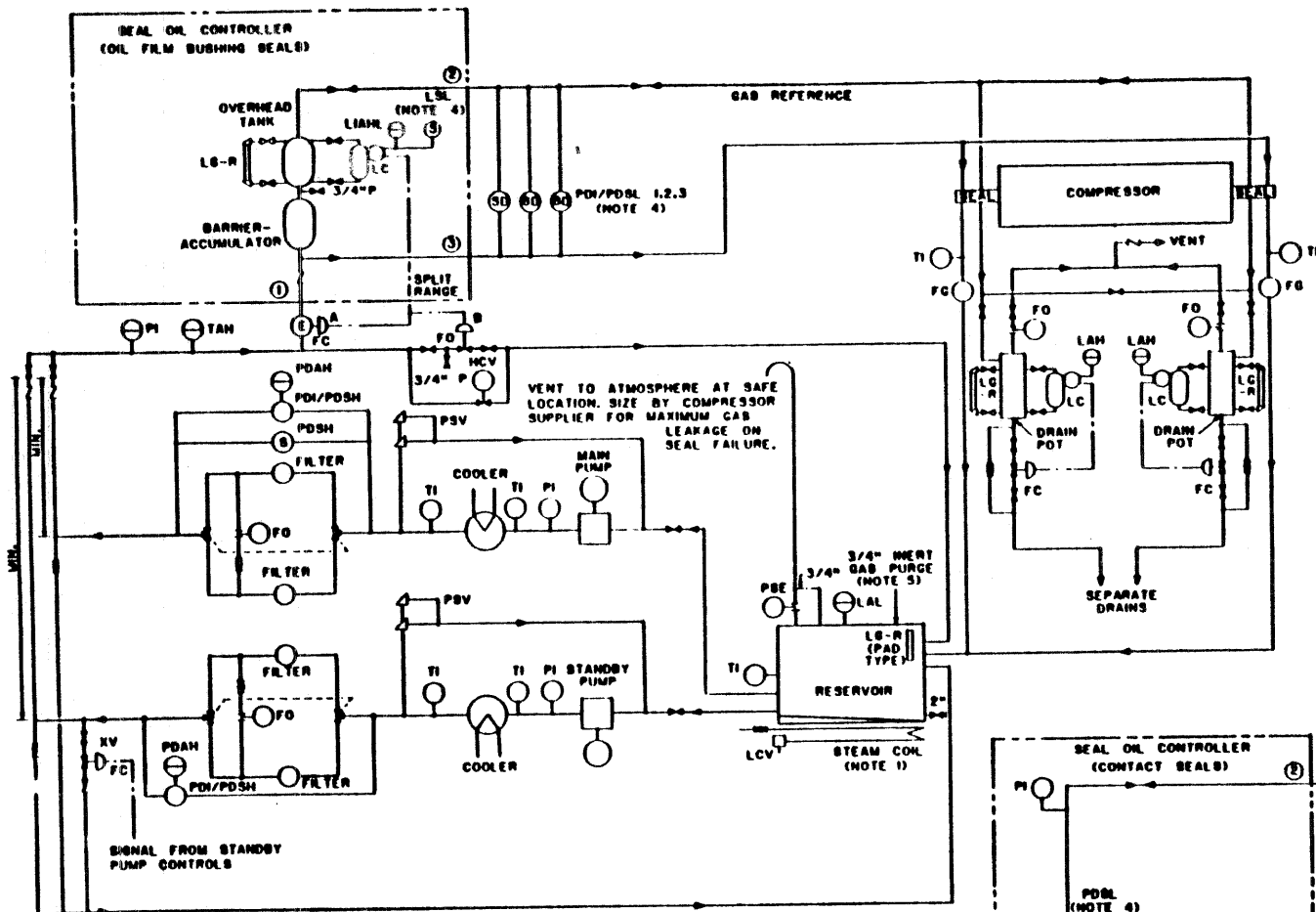
COMPRESSOR LUBE OIL SYSTEM



NOTES:

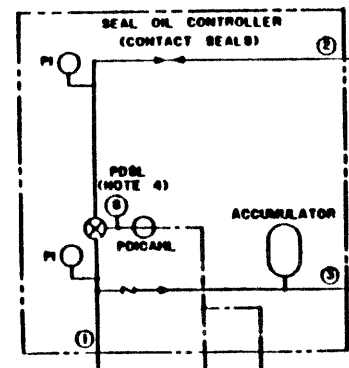
1. PROVIDE WHEN SPECIFIED-SEE PROJECT SPECIFICATION.
2. PROVIDE RELIEF VALVES AND PLUGGED DRAIN AND VENT VALVES IN PIPING AS REQUIRED.
3. PROVIDE PLUGGED DRAIN AND VENT VALVES ON COOLERS AND FILTERS.
4. SWITCHES WITH "S" IN SYMBOL START STANDBY PUMP.
SWITCHES WITH "SD" IN SYMBOL SHUT DOWN COMPRESSOR.
5. PROVIDE FOR CONTROL OIL SUPPLY OR OTHER SERVICE AS REQUIRED.

COMBINATION OR SEAL OIL SYSTEM



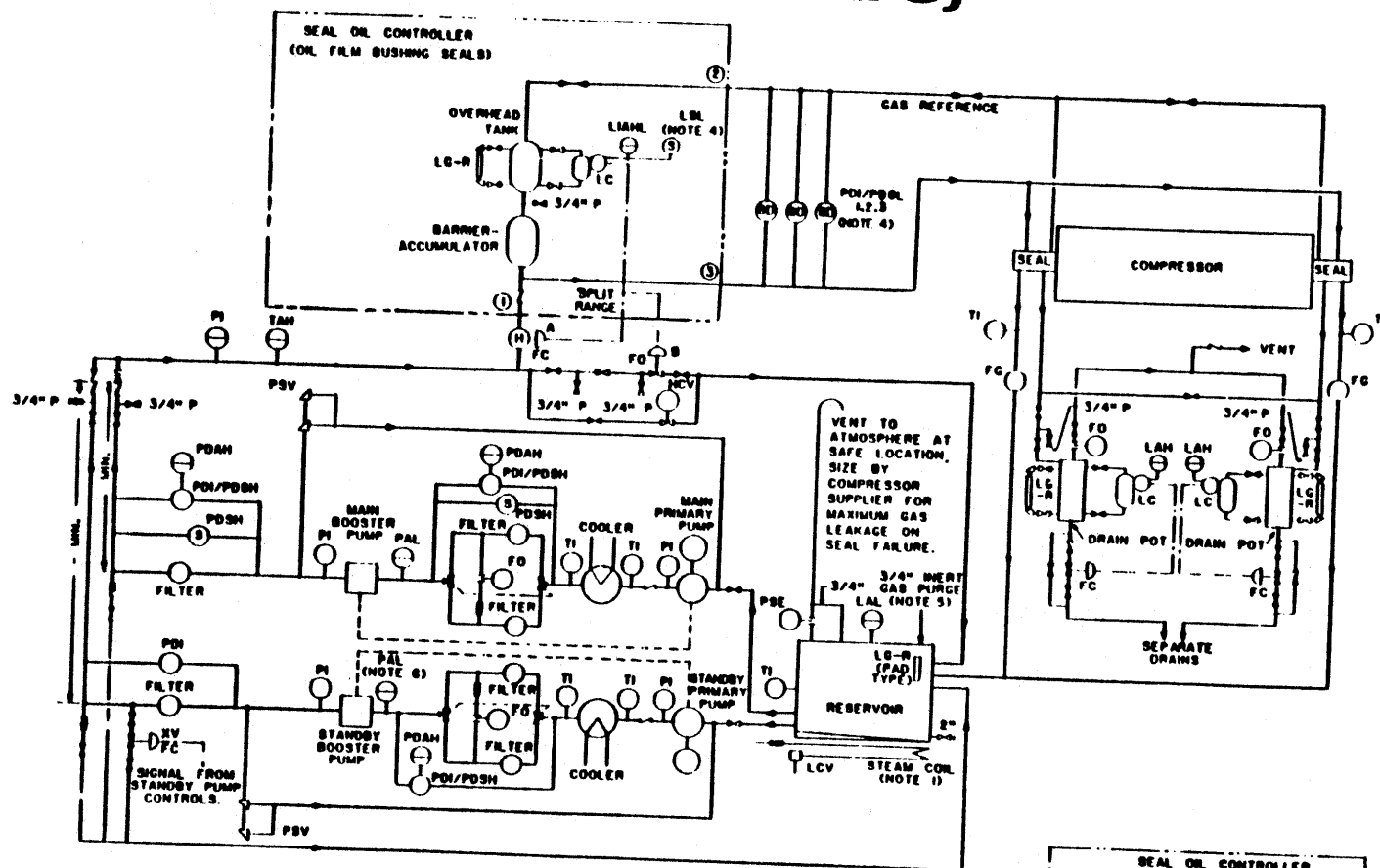
NOTES:

1. PROVIDE WHEN SPECIFIED-BEE PROJECT SPECIFICATION.
2. PROVIDE RELIEF VALVES AND PLUGGED DRAIN AND VENT VALVES IN PIPING AS REQUIRED.
3. PROVIDE PLUGGED DRAIN AND VENT VALVES ON COOLERS AND FILTERS.
4. SWITCHES WITH "S" IN SYMBOL START STANDBY PUMP.
SWITCHES WITH "SD" IN SYMBOL SHUT DOWN COMPRESSOR.
5. PROVIDE FOR COMPRESSORS HANDLING GASES CONTAINING OVER 30 MOL PERCENT HYDROGEN.



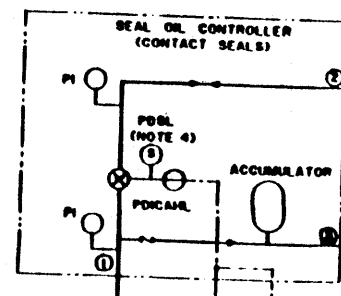
UOP 742 12

COMPRESSOR SEAL OIL SYSTEM (BOOSTER PUMPS)

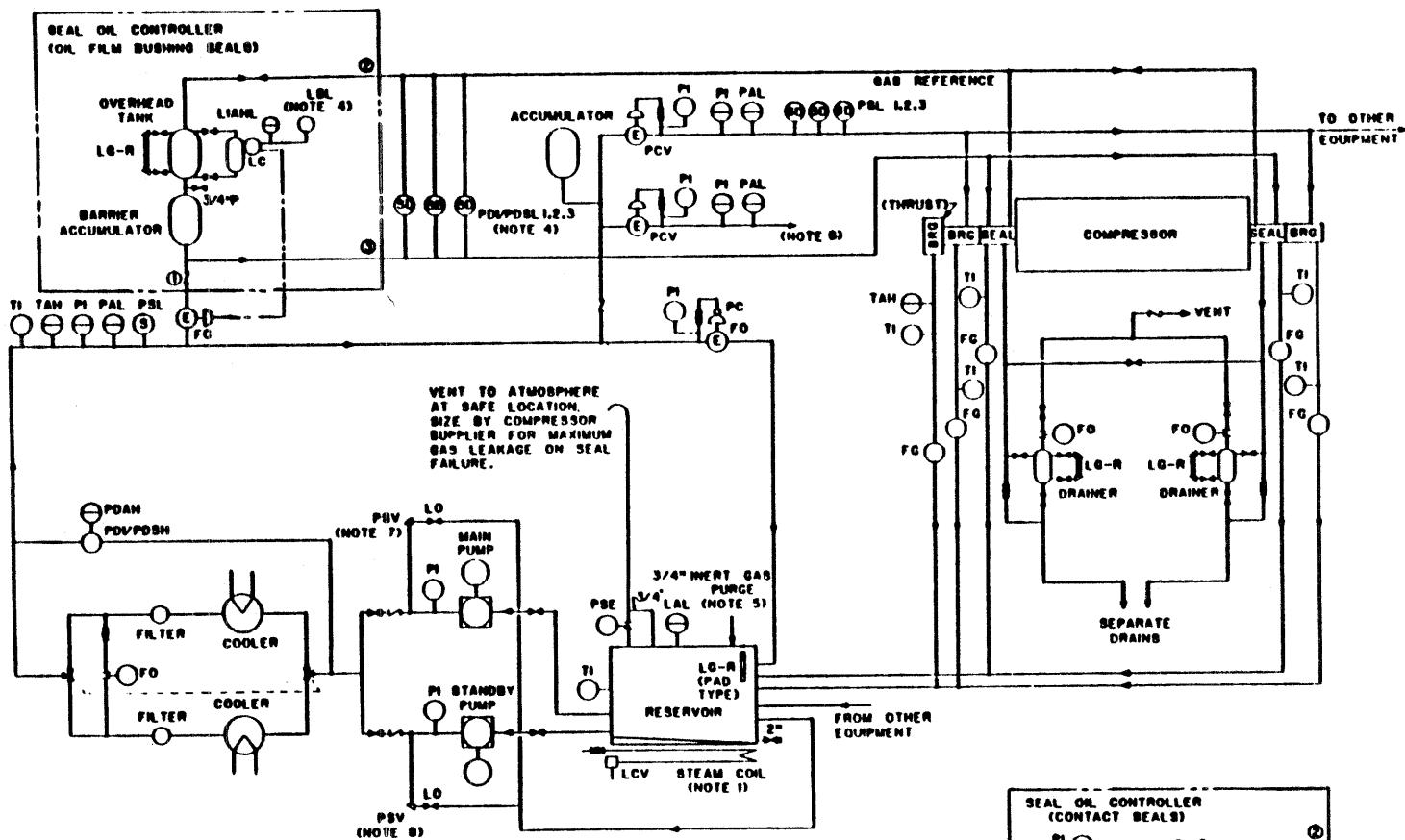


NOTES:

1. PROVIDE WHEN SPECIFIED-BEE PROJECT SPECIFICATION.
2. PROVIDE RELIEF VALVES AND PLUGGED DRAIN AND VENT VALVES IN PIPING AS REQUIRED.
3. PROVIDE PLUGGED DRAIN AND VENT VALVES ON COOLERS AND FILTERS.
4. SWITCHES WITH "S" IN SYMBOL START STANDBY PUMP
5. SWITCHES WITH "SD" IN SYMBOL SHUT DOWN COMPRESSOR
6. CONTACTS IN PUMP START CIRCUITRY GASES CONTAINING OVER 30 MOL PERCENT HYDROGEN.

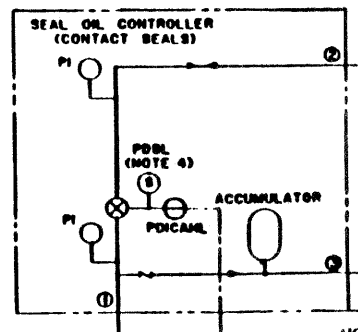


COMPRESSOR LUBE AND SEAL OIL SYSTEM



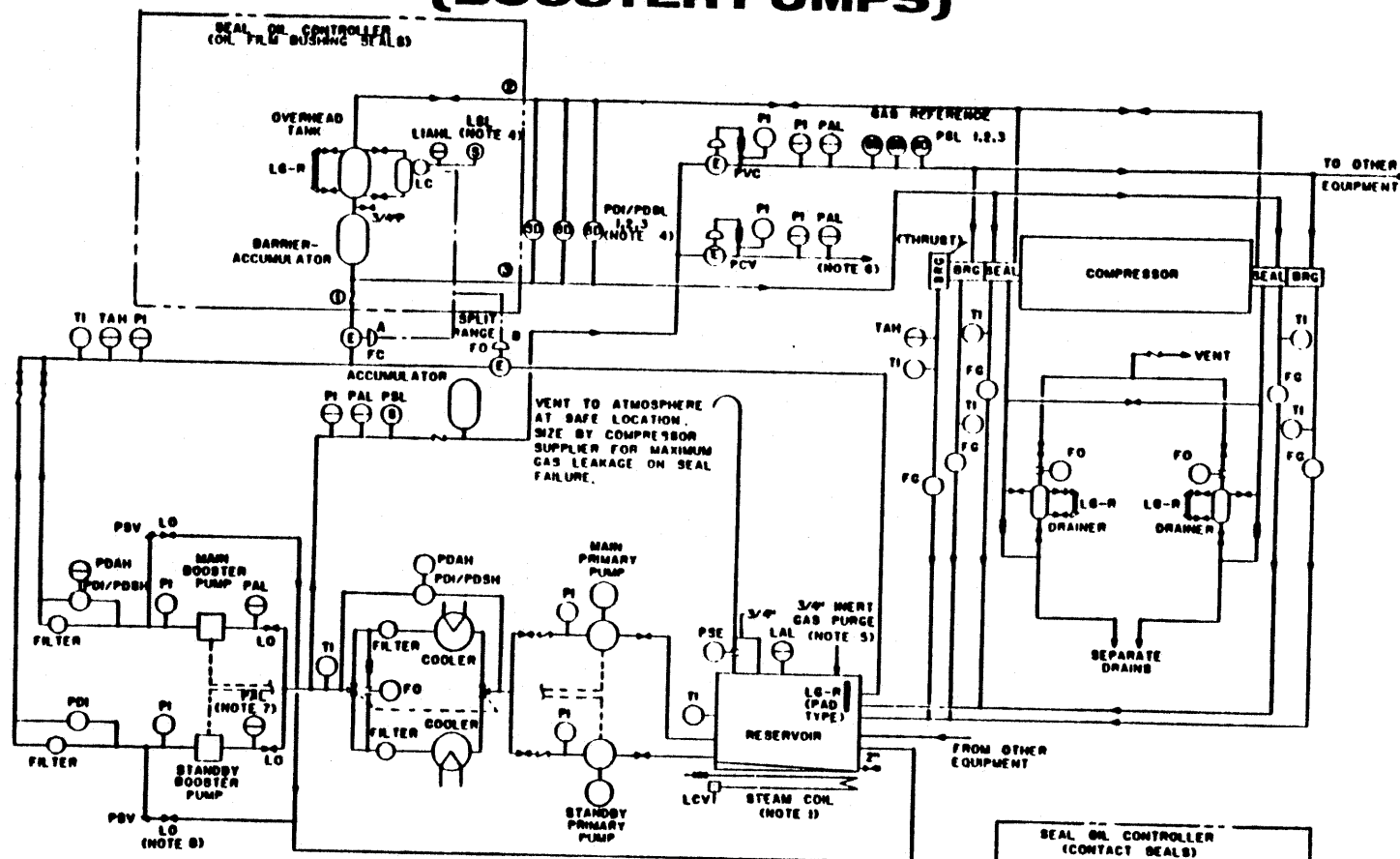
NOTES:

1. PROVIDE WHEN SPECIFIED-SEE PROJECT SPECIFICATION.
2. PROVIDE RELIEF VALVES AND PLUGGED DRAIN AND VENT VALVES IN PIPING AS REQUIRED.
3. PROVIDE PLUGGED DRAIN AND VENT VALVES ON COOLERS AND FILTERS.
4. SWITCHES WITH "S" IN SYMBOL START STANDBY PUMP.
SWITCHES WITH "SD" IN SYMBOL SHUT DOWN COMPRESSOR.
5. PROVIDE FOR COMPRESSORS HANDLING GASES CONTAINING OVER 30 MOL PERCENT HYDROGEN.
6. PROVIDE FOR CONTROL OIL SUPPLY OR OTHER SERVICE AS REQUIRED.
7. PROVIDE FOR POSITIVE DISPLACEMENT PUMPS ONLY. PROVIDE LOCKED OPEN BLOCK VALVE IF REQUIRED TO SERVICE RELIEF VALVE.



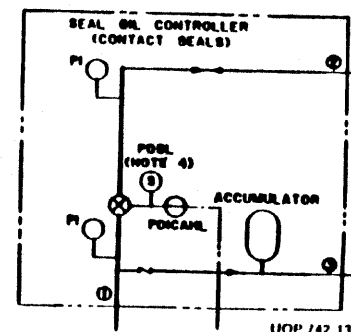
UOP 742-19

COMPRESSOR LUBE AND SEAL OIL SYSTEM (BOOSTER PUMPS)



NOTES:

1. PROVIDE WHEN SPECIES-SEE PROJECT SPECIFICATION.
2. PROVIDE RELIEF VALVES AND PLUGGED DRAIN AND VENT VALVES IN PIPING AS REQUIRED.
3. PROVIDE PLUGGED DRAIN AND VENT VALVES ON COOLERS AND FILTERS.
4. SWITCHES WITH "S" IN SYMBOL START STANDBY PUMP.
5. SWITCHES WITH "SD" IN SYMBOL SHUT DOWN COMPRESSOR.
6. PROVIDE FOR COMPRESSORS HANDLING GASES CONTAINING OVER 30 MOL PERCENT HYDROGEN.
7. PROVIDE FOR CONTROL OR SUPPLY OR OTHER SERVICE AS REQUIRED.
8. CONTACTS IN PUMP START CIRCUITRY BYPASS ALARM WHEN PUMP IS ON STANDBY
9. PROVIDE LOCKED OPEN BLOCX VALVE IF REQUIRED TO SERVICE RELIEF VALVE.



TRAINING PROGRAM
PROBLEM A
COMPLETE BHP CALCULATION FORMS

TYPE _____ COMPRESSORS		PAGE NO.	DATE
		PROJECT	BY
ITEM NO.	CENTRIFUGAL		Reynolds
SERVICE	MAIN AIR BLOWER		CRUDE GAS (LUBRICATED)
CAPACITY, 10 ⁶ SCFD (NORMAL)			
(DESIGN)	34.54		1.62
SUCTION PRESSURE, PSI	14.4		16.7
DISCHARGE PRESSURE, PSI (EST.)	51.2		93.7
SUCTION TEMPERATURE, °F	95		105
DISCHARGE TEMPERATURE, °F (EST.)			
COMPRESSION RATIO (EST.)	3.556		5.61
COMPRESSIBILITY AT SUCTION	1.0		0.98
COMPRESSIBILITY AT DISCHARGE			0.93
APPROXIMATE CP/CV	1.4		1.12
GAS MOLECULAR WEIGHT	28.49		53.05
GAS COMPOSITION, MOL PCT.		H ₂ O	6.24
		H ₂ S	0.26
	Atmospheric	C ₁	0.04
	Air	C ₂	12.00
	(Wet)	C ₃	29.31
	84% RH	iC ₄	6.71
		nC ₄	22.18
		iC ₅	5.58
		nC ₅	8.27
		C ₆ ⁺	9.41
			100.00
DRIVER TYPE			

SUBJECT		RECIPROCATING COMPRESSORS - BHP		DATE	
FOR				BY	

SYMBOL

Q_b	Million SCFD	1.62				
T_s	° Rankine	565				
P_s	PSIA	16.7				
P_d	PSIA	93.7				
r	P_d / P_s					
k	C_p / C_v	1.12				
MW	Mol. Wt.	53.05				
S.G.	Spec. Gravity					
Z_s		.98				
Z_d		.93				
$Z_{av.}$		.955				
Q_s	CFM @ Suction SCFD (14.7) (T_s) Z_s 1440 (P_s) 520					
	$\frac{k-1}{k}$					
T_d	$\frac{k-1}{(r)^k (T_s)}$					
(A)	BHP / mm @ 14.4 PSIA and Suct. Temp.					
(B)	Sp. Gr. Corr.					
(C)	$\frac{Q_b (1.02) T_s}{520}$					
(D)	Add 5% if N.L.					
(E)	$Z_{av.}$					
(F)	Gear Loss 3% (if Gear is Used)					
(G)	Moisture Factor					
BHP Req'd	(A + B) (C) (D) (E) (F) (G)					

(D) : (IF N.L. $\rightarrow$ 1.05
IF LUBRICATED $\rightarrow$ 1.0

(F) : (IF GEAR $\rightarrow$ 1.03
IF NO GEAR $\rightarrow$ 1.0

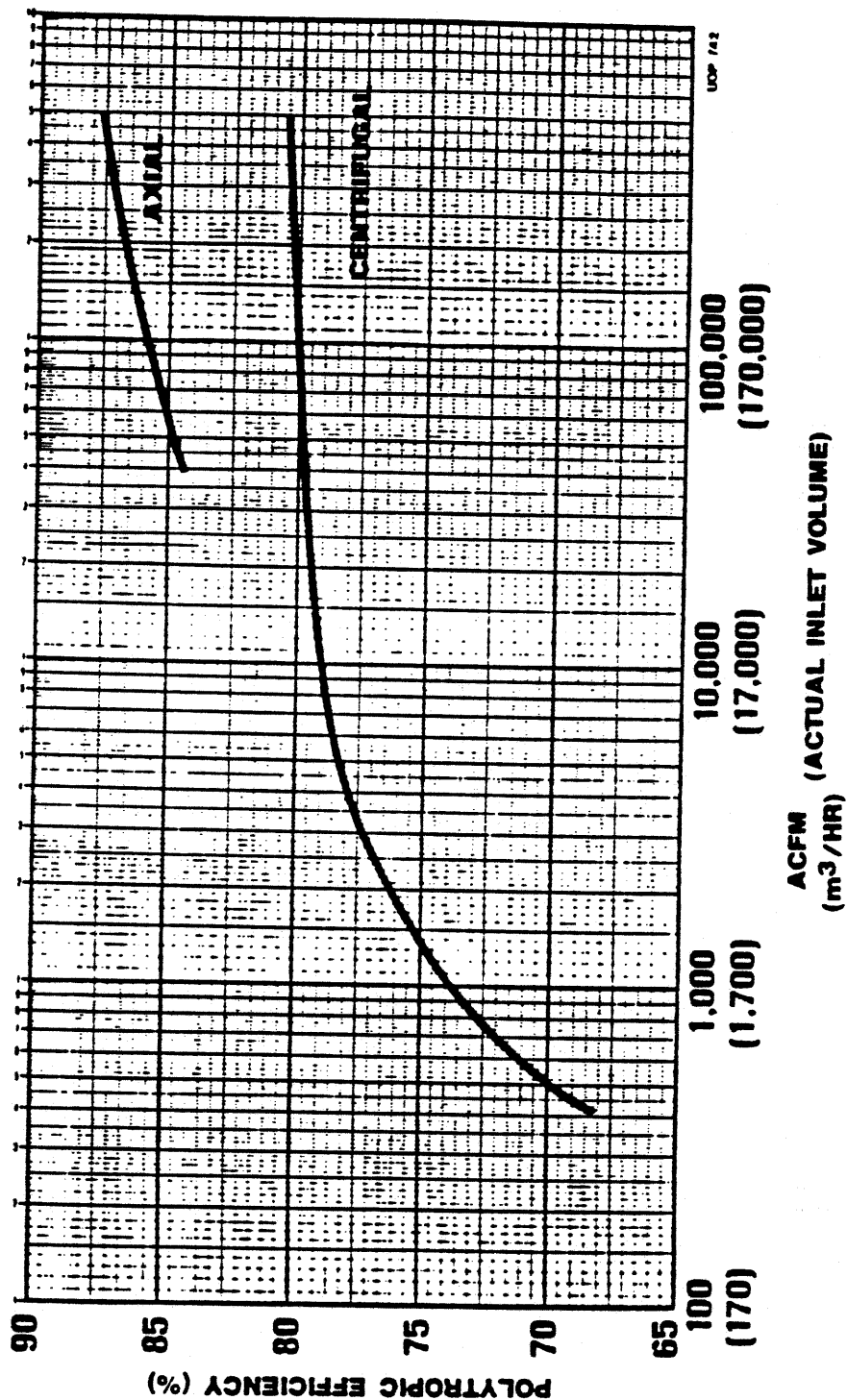
UOP17-1077 (F) : Dryer gas is usually 3% oil at 22.
10% Dryer gas at 22.5-24.

SUBJECT		CENTRIFUGAL COMPRESSORS – BHP		DATE	
FOR				BY	

SYMBOL

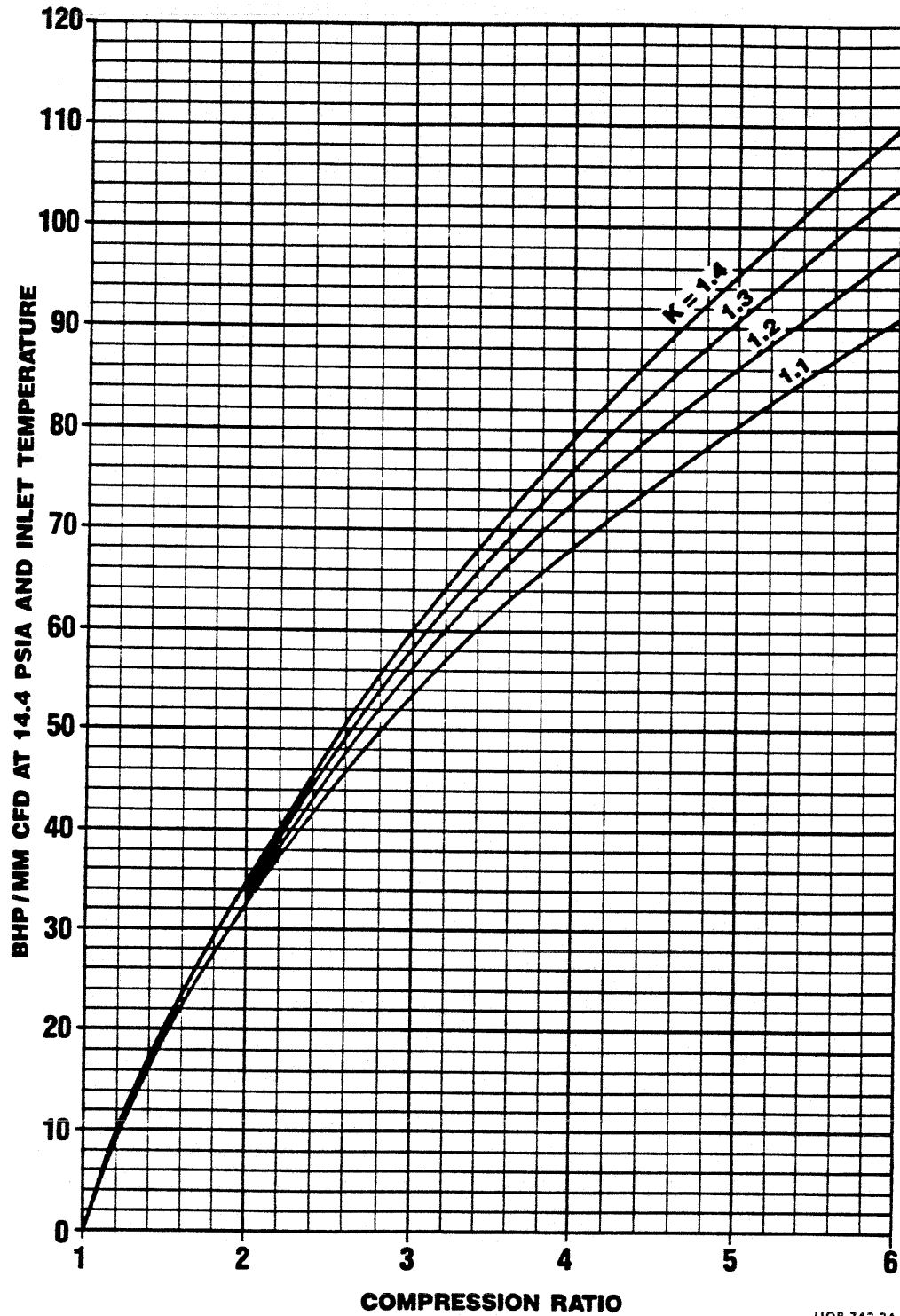
Q _b	Million SCFD	34.54				
T _s	° Rankine	555				
P _s	PSIA	14.4				
P _d	PSIA	51.2				
r	P _d / P _s					
MW	Mol. Wt.	28.49				
R	$\frac{1545}{MW}$					
k	C _p / C _v	1.4				
Q _s	CFM @ Suct.					
# / Min. WT. FLOW	$\frac{Q_b \times MW}{546000}$					
Z _s		1.0				
Z _d		1.0				
Z _{av.}		1.0				
E _h	Polytropic Efficiency					
	$\frac{k - 1}{k}$					
M	$\frac{k - 1}{k E_h}$					
	(r)M – 1					
Beta	$\frac{(r)M - 1}{M}$					
Head	Z _{av.} R T _s Beta					
GHP	$\frac{Wt. Flow \times Head}{33000 \times E_h}$					
Losses						
Frict.						
Seal	1% of GHP					
Gear	(If used) 3% of HP					
Total BHP }						
T _d	T _s (r)M					

COMPRESSOR POLYTROPIC EFFICIENCY

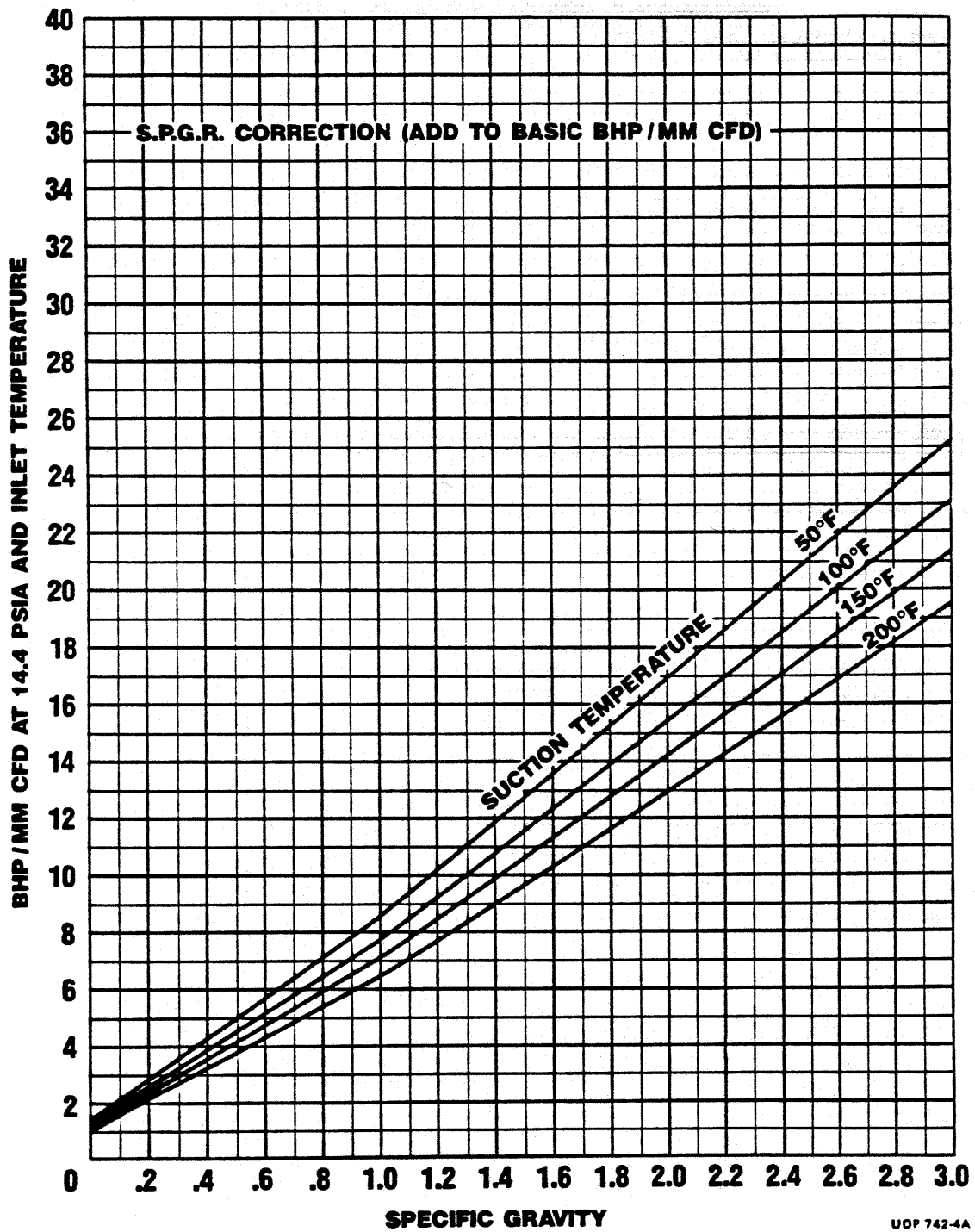


PD 1003 A
RBC 4-22-75
REV.1 MJW 2-19-82

BASIC BHP / MM CFD



S P.G R. CORRECTION



$\frac{14.7}{14.4} = 1.02$. MOST COMPRESSOR ^{inter p} 21 LOSS 0122

SAMPLE PROBLEM B

Calculate the duty requirements for both a centrifugal and reciprocating compressor for the following applications:

Service: Booster Gas

P_1	115 psiA
P_2	275 psiA
T_1	100°F
MMSCFD	10.0
MOLE WT	8.5
C_p/C_v ave.	1.30
Z_1	0.99
Z_2	1.02

Based on the following additional information, recommend which type of compressor to apply (two reciprocating or one centrifugal).

Cost of electricity	5¢/KW-hr
Cost of each recip train	\$325,000
Cost of installing each recip	\$325,000
Yearly recip maintenance cost	\$20,000 per machine
Cost of centrifugal train	\$850,000
Cost of installing one centrif	\$250,000
Yearly centrif maint. cost	\$10,000

SUBJECT	CENTRIFUGAL COMPRESSORS – BHP	DATE
FOR	Sample Problem B	BY

SYMBOL

Q_b	Million SCFD	10.0				
T_s	° Rankine	560				
P_s	PSIA	115				
P_d	PSIA	275				
r	P_d / P_s					
MW	Mol. Wt.	8.5				
R	$\frac{1545}{MW}$					
k	C_p / C_v	1.30				
Q_s	CFM @ Suct.					
# / Min. WT. FLOW	$\frac{Q_b \times MW}{546000}$					
Z_s Z_d $Z_{av.}$		0.99 1.02				
E_h	Polytropic Efficiency					
	$\frac{k - 1}{k}$					
M	$\frac{k - 1}{k E_h}$					
	$(r)^M - 1$					
Beta	$\frac{(r)^M - 1}{M}$					
Head	$Z_{av.} R T_s \text{ Beta}$					
GHP	$\frac{\text{Wt. Flow} \times \text{Head}}{33000 \times E_h}$					
Losses Frict. Seal	1% of GHP					
Gear	(If used) 3% of HP					
Total BHP }						
T_d	$T_s (r)^M$					

SUBJECT	RECIPROCATING COMPRESSORS – BHP	DATE
FOR	Sample Problem B	BY

SYMBOL

Q_b	Million SCFD	10.0				
T_s	° Rankine	560				
P_s	PSIA	115				
P_d	PSIA	275				
r	P_d / P_s					
k	C_p / C_v	1.30				
MW	Mol. Wt.	8.5				
S.G.	Spec. Gravity					
Z_s Z_d $Z_{av.}$		0.99 1.02				
Q_s	CFM @ Suction SCFD (14.7) (T_s) Z_s 1440 (P_s) 520					
	$\frac{k - 1}{k}$					
T_d	$\frac{k - 1}{(r)^k (T_s)}$					
(A)	BHP / mm @ 14.4 PSIA and Suct. Temp.					
(B)	Sp. Gr. Corr.					
(C)	$\frac{Q_b (1.02) T_s}{520}$					
(D)	Add 5% if N.L.	1.05				
(E)	$Z_{av.}$					
(F)	Gear Loss 3% (if Gear is Used)	1.00				
(G)	Moisture Factor					
BHP Req'd	(A + B) (C) (D) (E) (F) (G)					

COMPRESSOR PROBLEM SOLUTIONS

PROBLEM A

CENTRIFUGAL - MAIN AIR BLOWER

$$\text{ACFM} = \frac{\text{SCFD}}{\text{minutes/day}} \times \frac{14.7 \text{ psia}}{P_s \text{ psia}} \times \frac{T_s^\circ\text{R}}{520^\circ\text{R}} \times Z_s$$

$$\text{ACFM} = \frac{34.54 \times 10^6}{1440} \times \frac{14.7}{14.4} \times \frac{555}{520} \times 1.0 = 26134 \text{ CFM @ Inlet}$$

$$\text{\#/min wt. flow} = \frac{34.54 \times 10^6}{1440} \times \frac{28.49 \text{ \#/mole}}{379.48 \text{ scf/mole}} = 1800.8 \text{ \#/min}$$

$$\frac{K-1}{K} = 0.2857 \quad \frac{0.2857}{0.795} = 0.3594 = M$$

$$(\text{comp. ratio})^M - 1 = 3.556^{0.3594} - 1 = 0.5776$$

$$\frac{0.5776}{M} = \text{Beta} = 1.607$$

$$\text{Polytropic Head} = Z_{av} R T_s B = (1.0)(1545/28.49)(555)(1.607) = 48376 \text{ ft } \frac{\text{lb}_p}{\text{lb}_m}$$

$$\text{GHP} = \frac{\text{Head} \times \text{\#/min}}{33000 \times E_h} = \frac{(48376)(1800.8)}{(33000)(0.795)} = 3321$$

$$\text{BHP} = 3321 (1.01)(1.03) = 3454 \text{ HP}$$

$$\text{Toisch} = T_s \times r^m = 555 (1.5776) = 876^\circ\text{R} = 416^\circ\text{F}$$

SUBJECT	CENTRIFUGAL COMPRESSORS – BHP	DATE
FOR	Main Air Blower	BY

SYMBOL

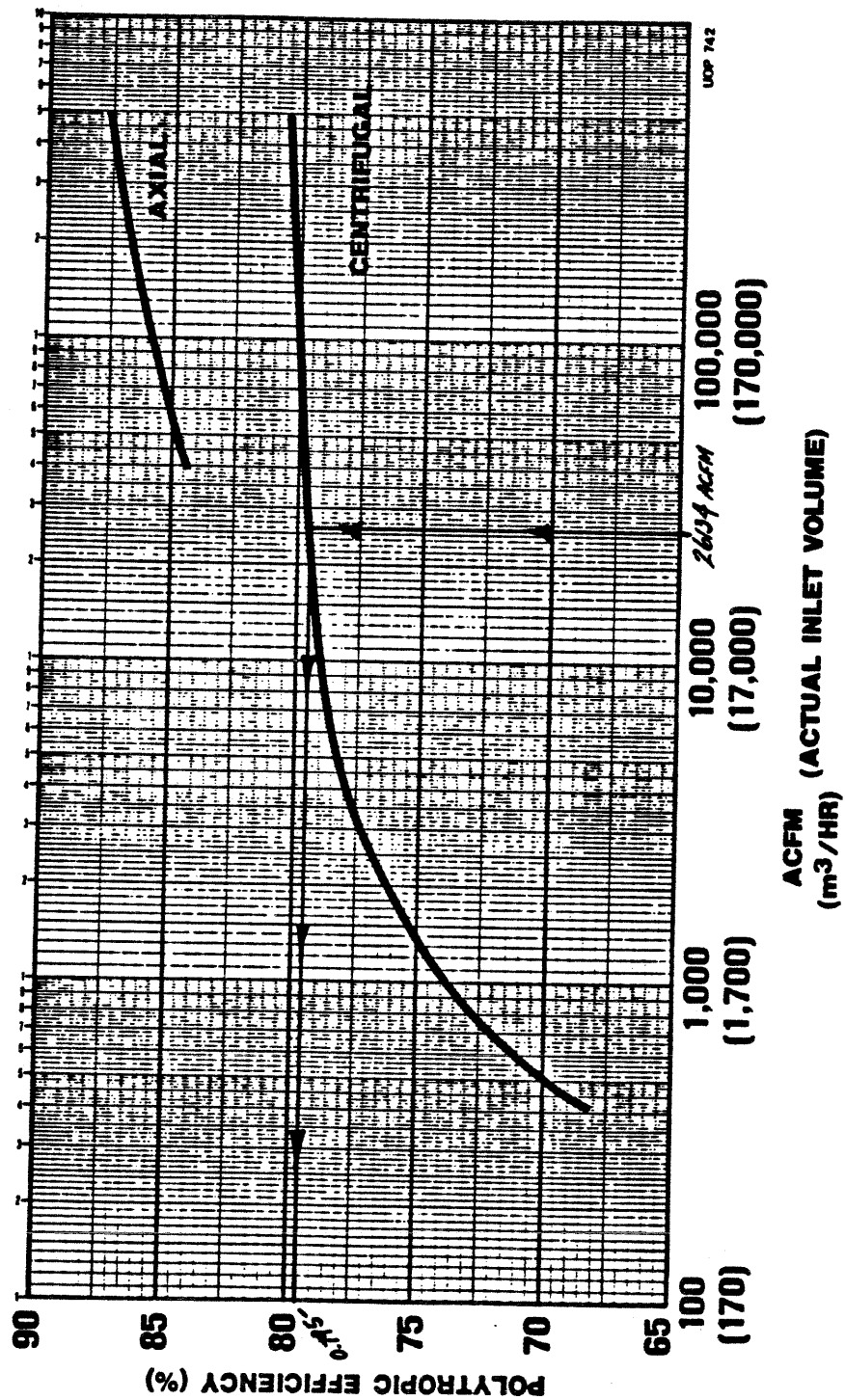
Q_b	Million SCFD	34.54				
T_s	°Rankine	555				
P_s	PSIA	14.4				
P_d	PSIA	51.2				
r	P_d / P_s	3.556				
MW	Mol. Wt.	28.49				
R	$\frac{1545}{MW}$	54.23				
k	C_p / C_v	1.4				
Q_s	CFM @ Suct.	26134				
# / Min. WT. FLOW	$\frac{Q_b \times MW}{546000}$	1800.8				
Z_s Z_d $Z_{av.}$		1.0 1.0 1.0				
E_h	Polytropic Efficiency	0.795				
	$\frac{k - 1}{k}$	0.2857				
M	$\frac{k - 1}{k E_h}$	0.3594				
	$(r)M - 1$	0.5776				
Beta	$\frac{(r)M - 1}{M}$	1.607				
Head	$Z_{av.} R T_s \text{ Beta}$	48376				
GHP	$\frac{Wt. Flow \times Head}{33000 \times E_h}$	3321				
Losses Frict. Seal	1% of GHP	33				
Gear	(If used) 3% of HP	100				
Total } BHP }		3454				
$T_d (°F)$	$T_s (r)M$	416				

SUBJECT	RECIPROCATING COMPRESSORS – BHP	DATE
FOR	CRUDE GAS	BY

SYMBOL

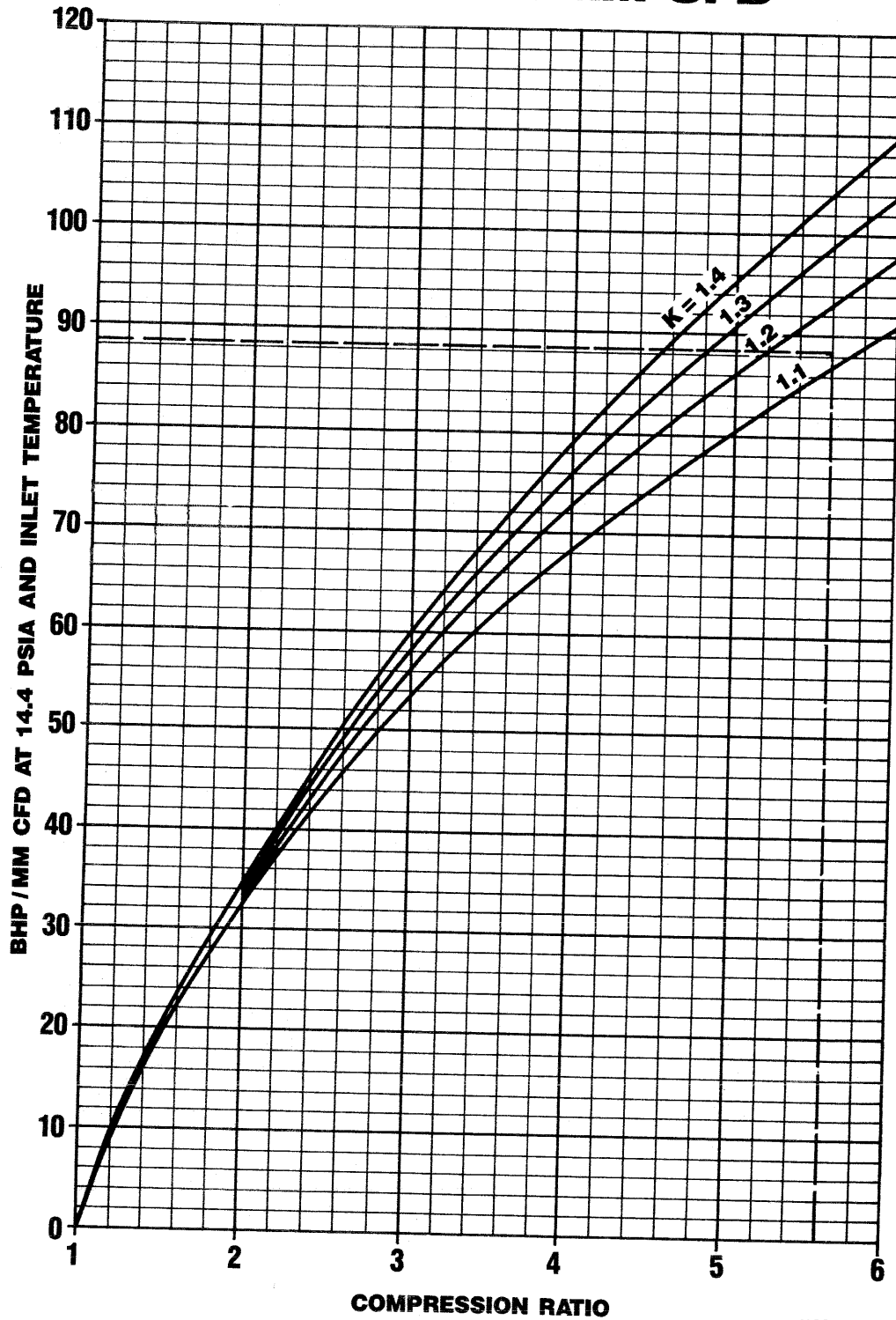
Q _b	Million SCFD	1.62				
T _s	° Rankine	565				
P _s	PSIA	16.7				
P _d	PSIA	93.7				
r	P _d / P _s	5.61				
k	C _p / C _v	1.12				
MW	Mol. Wt.	53.05				
S.G.	Spec. Gravity	1.83				
Z _s Z _d Z _{av.}		0.98 0.93 0.955				
Q _s	CFM @ Suction SCFD (14.7) (T _s) Z _s 1440 (P _s) 520	1054				
	$\frac{k-1}{k}$	0.1071				
T _d	$\frac{k-1}{(r)^k (T_s)}$	220°F				
(A)	BHP / mm @ 14.4 PSIA and Suct. Temp.	88.4				
(B)	Sp. Gr. Corr.	13.9				
(C)	$\frac{Q_b (1.02) T_s}{520}$	1.795				
(D)	Add 5% if N.L.	-				
(E)	Z _{av.}	0.955				
(F)	Gear Loss 3% (if Gear is Used)	1.03				
(G)	Moisture Factor	-				
BHP Req'd	(A + B) (C) (D) (E) (F) (G)	181				

COMPRESSOR POLYTROPIC EFFICIENCY



PD 1003 A
RBC 4-22-75
REV.1 MJW 2-19-82

BASIC BHP / MM CFD



**SAMPLE PROBLEM B
SOLUTION**

	<u>CENTRIFUGAL</u>	<u>RECIPROCATING</u>
Utilities Cost	\$223,000	\$180,700
Total Installed Cost	\$1.1 Million	\$1.3 Million
Maint. Cost	\$10,000	\$40,000

Conclusion:

Based on the above estimated costs, it can be seen that the higher efficiency of the recip is balanced by the lower maintenance costs of the centrifugal.

Although this is a relatively high head, low flow centrifugal, it would provide smoother operation and higher reliability in a smaller space.

The reciprocating compressor would be a straightforward single-stage, two cylinder machine. It would be much less sensitive to a changing mole wt.

If lube oil contamination were a concern, a turbine drive were required, or the site were arctic, the centrifugal would provide much better characteristics.

RECOMMENDATION: CENTRIFUGAL

SUBJECT	RECIPROCATING COMPRESSORS – BHP	DATE
FOR	Sample Problem B	BY

SYMBOL

Q_b	Million SCFD	10.0				
T_s	° Rankine	560				
P_s	PSIA	115				
P_d	PSIA	275				
r	P_d / P_s	2.391				
k	C_p / C_v	1.30				
MW	Mol. Wt.	8.5				
S.G.	Spec. Gravity	0.29				
Z_s Z_d $Z_{av.}$		0.99 1.02 1.005				
Q_s	CFM @ Suction $\frac{SCFD (14.7) (T_s) Z_s}{1440 (P_s) 520}$	946				
	$\frac{k - 1}{k}$	0.231				
T_d	$\frac{k - 1}{(r)^k (T_s)}$	225°F				
(A)	BHP / mm @ 14.4 PSIA and Suct. Temp.	44.6				
(B)	Sp. Gr. Corr.	3.1				
(C)	$\frac{Q_b (1.02) T_s}{520}$	10.985				
(D)	Add 5% if N.L.	1.050				
(E)	$Z_{av.}$	1.005				
(F)	Gear Loss 3% (if Gear is Used)	1.00				
(G)	Moisture Factor	1.0				
BHP Req'd	(A + B) (C) (D) (E) (F) (G)	553				

SUBJECT	CENTRIFUGAL COMPRESSORS – BHP	DATE
FOR	Sample Problem B	BY

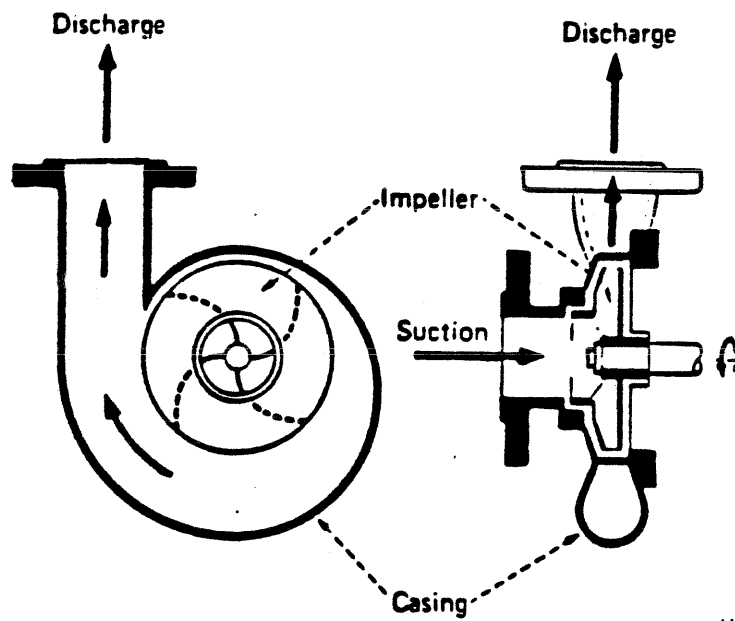
SYMBOL

Q_b	Million SCFD	10.0				
T_s	° Rankine	560				
P_s	PSIA	115				
P_d	PSIA	275				
r	P_d / P_s	2.391				
MW	Mol. Wt.	8.5				
R	$\frac{1545}{MW}$	181.77				
k	C_p / C_v	1.30				
Q_s	CFM @ Suct.	946				
# / Min. WT. FLOW	$\frac{Q_b \times MW}{546000}$	155.5				
Z_s Z_d $Z_{av.}$		0.99 1.02 1.005				
E_h	Polytropic Efficiency	0.736				
	$\frac{k - 1}{k}$	0.231				
M	$\frac{k - 1}{k E_h}$	0.313				
	$(r)^M - 1$	0.314				
Beta	$\frac{(r)^M - 1}{M}$	1.003				
Head	$Z_{av.} R T_s \text{ Beta}$	102560				
GHP	$\frac{Wt. Flow \times Head}{33000 \times E_h}$	656				
Losses Frict. Seal	1% of GHP	7				
Gear	(If used) 3% of HP	20				
Total BHP }		683				
T_d	$T_s (r)^M$	276°F				

HYDRAULIC POWER RECOVERY TURBINES

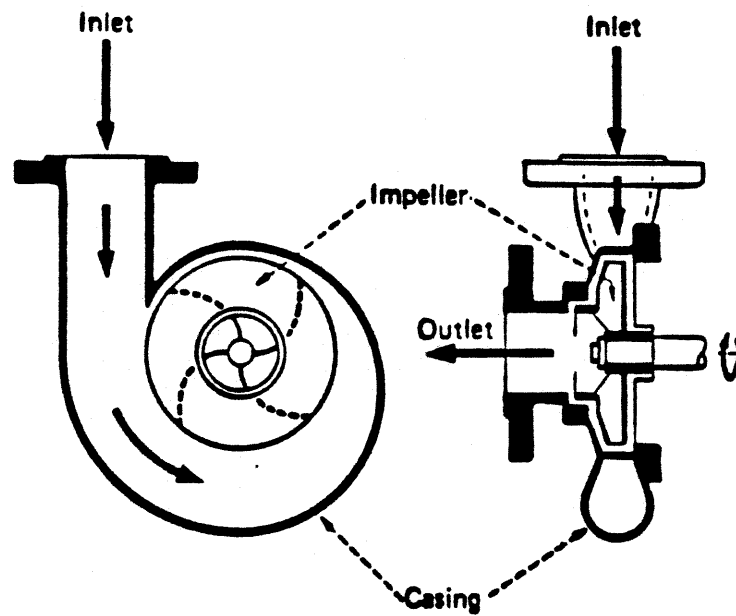
UOP 742-20

RADIAL-FLOW PUMP AS FLUID MOVER



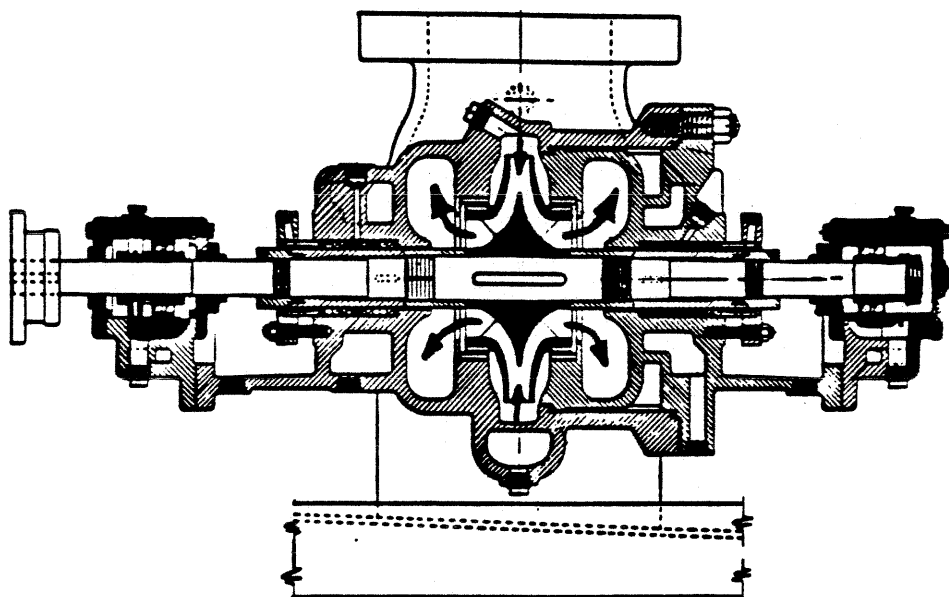
UOP 742-7

RADIAL-FLOW PUMP AS HYDRAULIC TURBINE



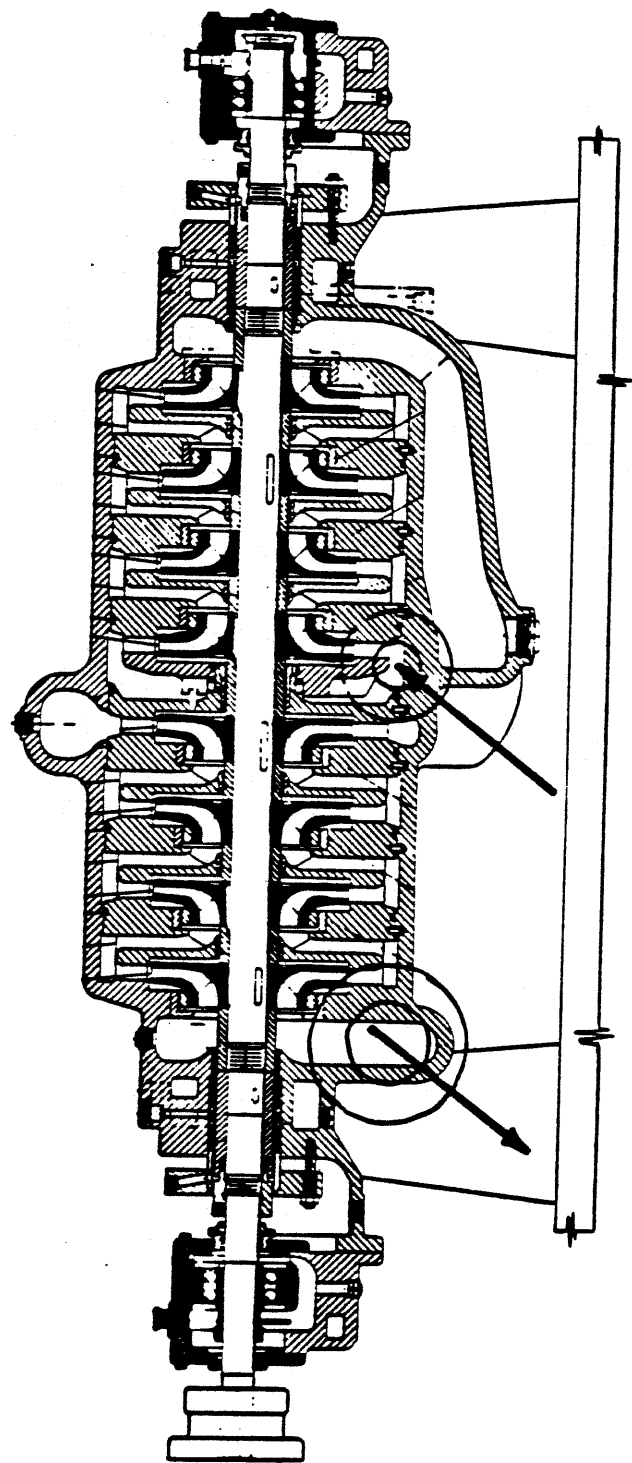
UOP 742-6

SINGLE-STAGE HYDRAULIC TURBINE



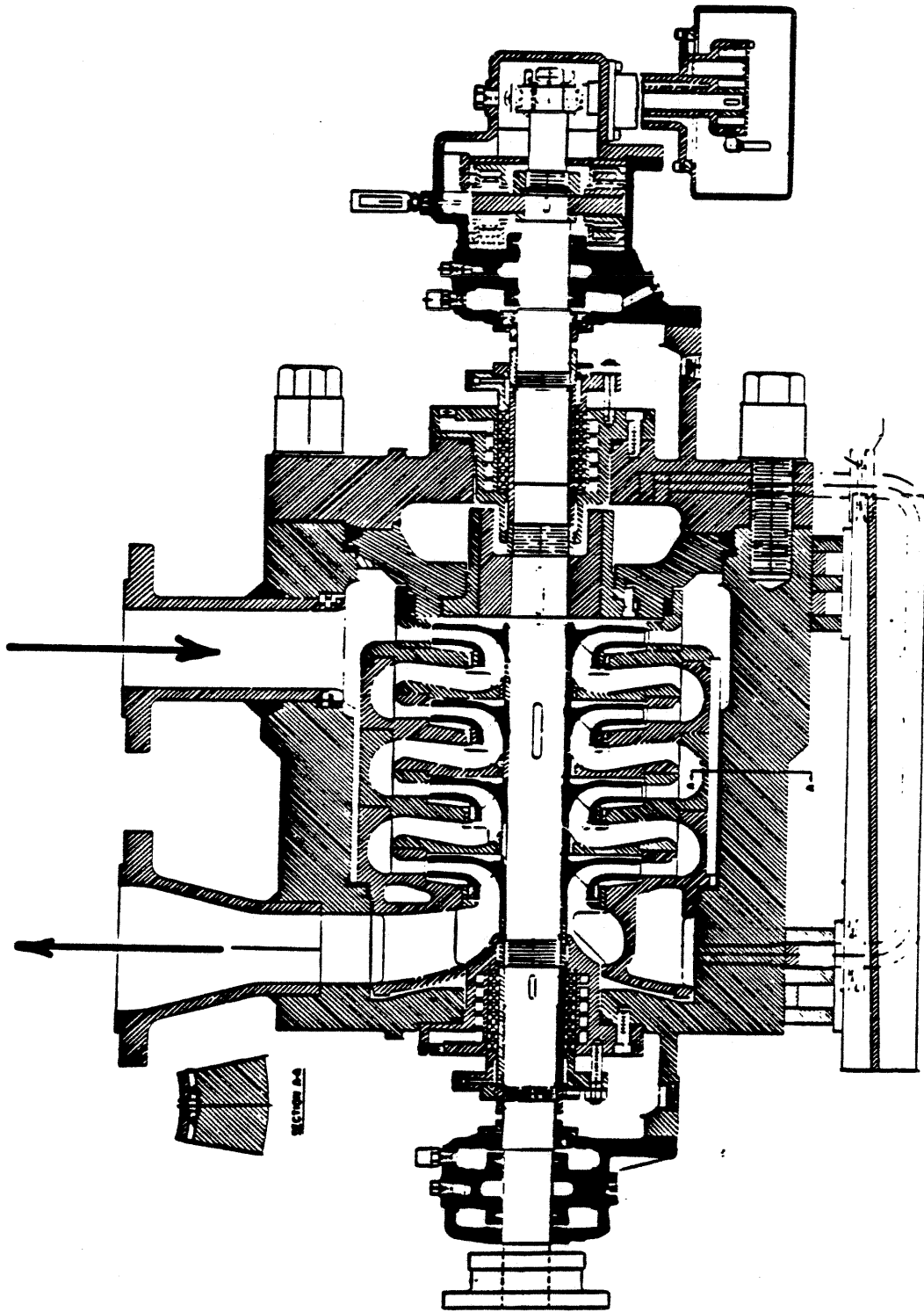
UOP 742-5

MULTI-STAGE HORIZONTALLY SPLIT CASE UNIT

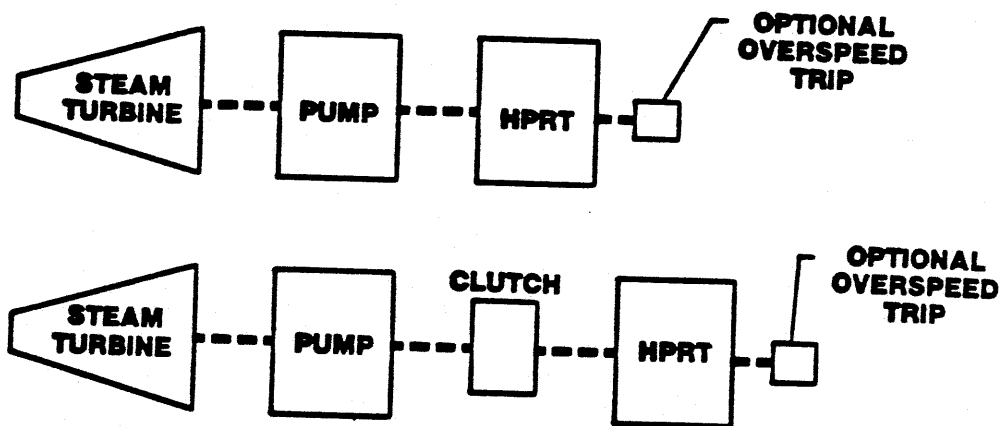


UOP 142-4

MULTI-STAGE RADIALY SPLIT CASE

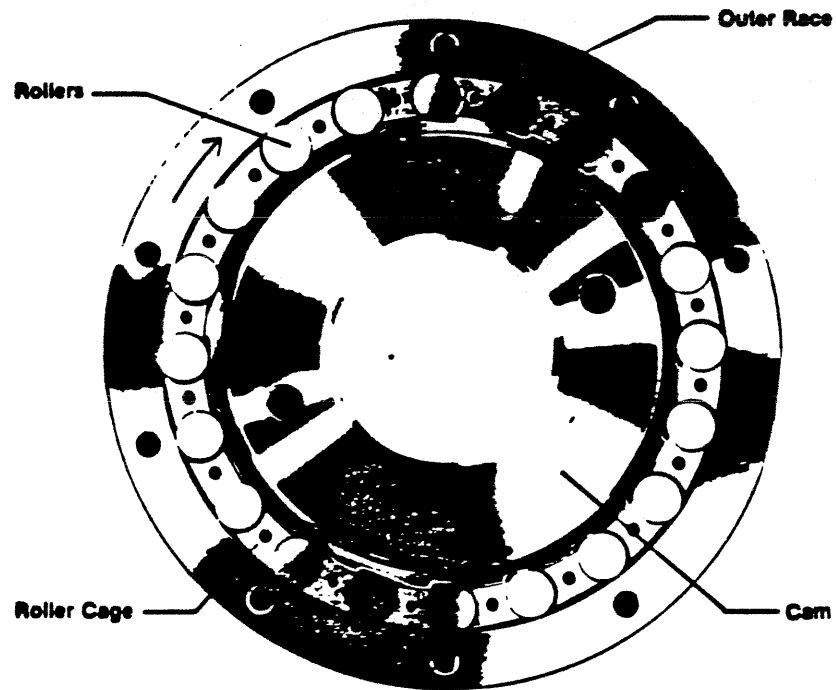


HPRT ARRANGEMENT FOR A STEAM TURBINE AUXILIARY DRIVER



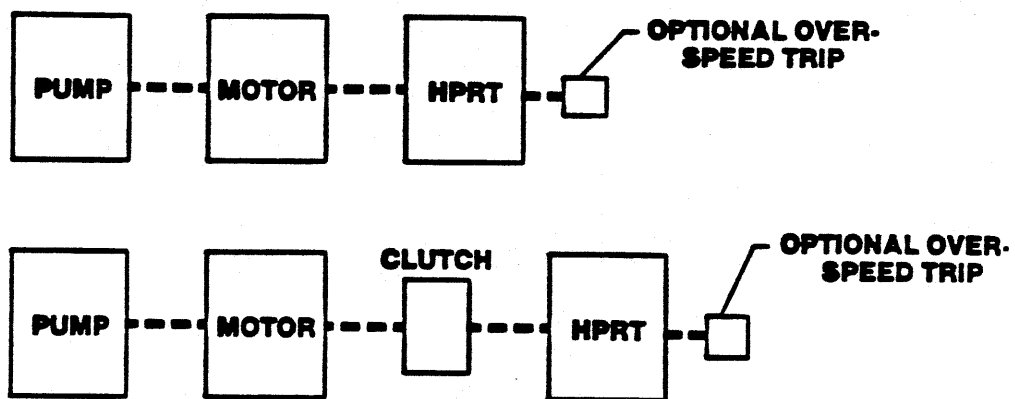
UOP 742-2

ONE-WAY CLUTCH



UOP 742-8

HPRT ARRANGEMENT FOR AUXILIARY MOTOR DRIVER



UOP 742-1

$$\text{HPRT bhp} = \frac{Q_t \times \text{TDH}_t \times e \times \text{sp.gr.}}{3960} \quad (2)$$

UOP 742-9

FOR 5¢ PER KW HR AND 3 YEARS PAYOUT BASIS:

	<u>EST'D. MIN. HP RECOVERED</u>
SINGLE STAGE	75
MULTI-STAGE AXIALLY SPLIT	100
MULTI-STAGE RADIALLY SPLIT	250

UOP 742-27

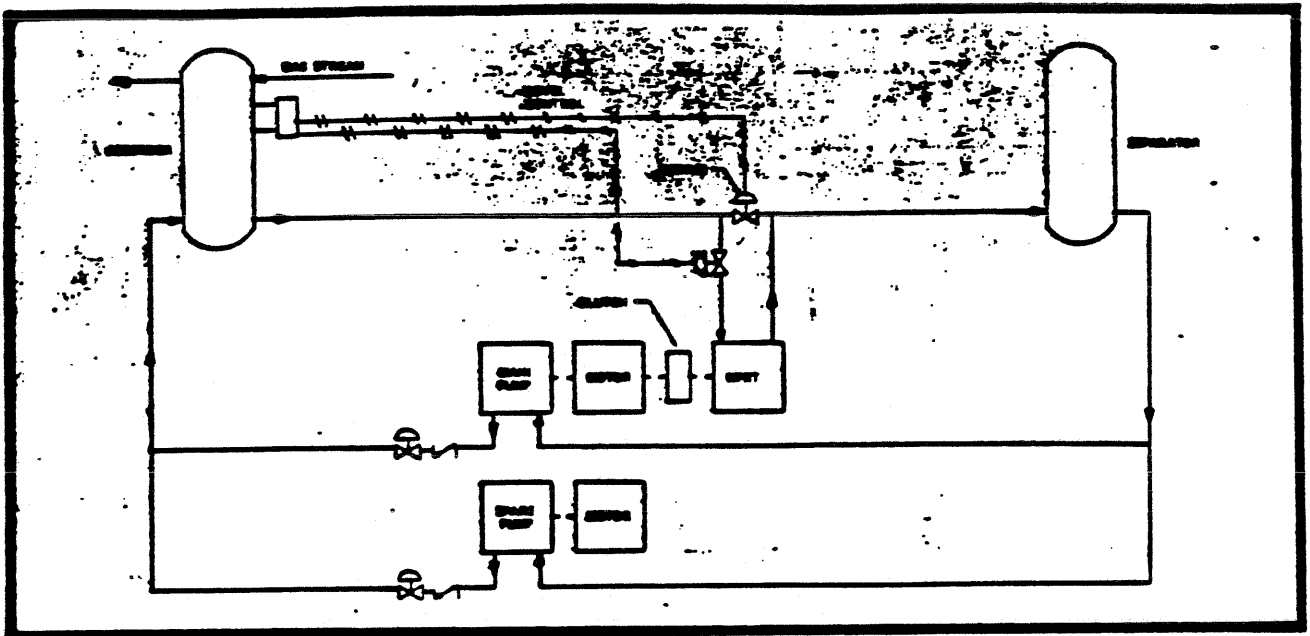


Fig. 13 A schematic drawing of an HPRT control arrangement.

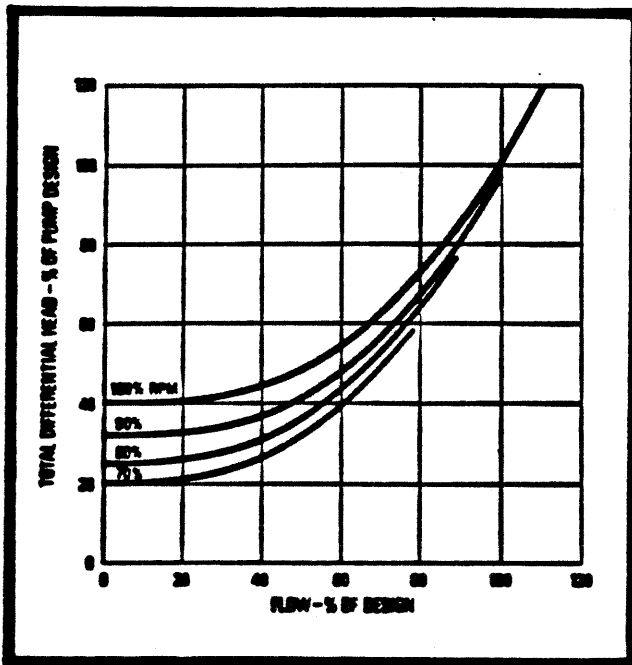


Fig. 15 Variations of HPRT Total Differential Heads (TDH) with speed.

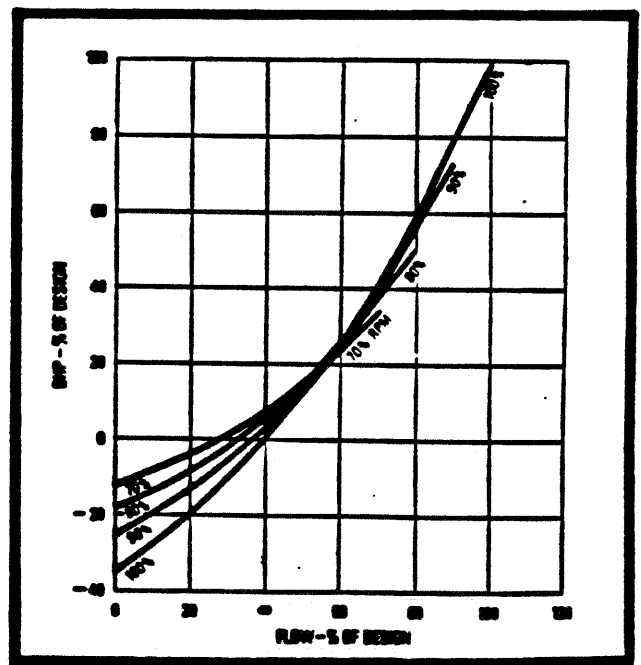


Fig. 16 Variations of HPRT brake horsepower (bhp) output with speed.

Hydraulic Power Recovery Turbine Problem I

A charge pump is driven by a motor and hydraulic power recovery turbine. Calculate the charge pump horsepower, the hpvt power recovered, and the utility savings per year for the hpvt given the following information:

Charge Pump Duty

Flow 1200 gpm
P inlet 150 Psig
P outlet 2875 Psig
T inlet 100 °F
Sp. Gr. 0.877
Eff. 81%

Hydraulic Power Recovery Turbine

Flow 1300 gpm
P inlet 2480 Psig
P outlet 250 Psig
T inlet 140 °F
Sp. Gr. 0.808
Eff. 74%

Electrical Power 5 cents per kw-hr

pump: VI-3.

Hydraulic Power Recovery Turbine Problem I Solution

Charge Pump Horsepower

$$\frac{1200 \text{ gpm} \times 2725 \text{ psi}}{1714 \times 0.81} = 2355 \text{ horsepower}$$

HPRT Horsepower Recovered

$$\frac{1300 \text{ gpm} \times 2230 \text{ psi} \times 0.74}{1714} = 1252 \text{ horsepower recovered}$$

At \$.05 per kW-Hr,

$$1252 \text{ hp} \times 0.746 \times 0.05 \times 24 \times 365 = \mathbf{\$409,000}$$

Note: The estimated incremental installed cost of the hp rt is \$600,000 including the additional valves, piping, controls, extended baseplate, and the tested hp rt. Therefore the addition of this piece of equipment would payout in approximately 18 months.