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the Design and Operation of Ammonia Plants

New Delhi, India
20 - 24 January 1976

CENTRIFUGAL COMPRESSORS FOR AMMONIA PLANTS DESIGN AND OPERATION CONSIDERATIONS 1/

by

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Centrifugal compressors are the accepted present-day standard equipment for synthesis gas compression for ammonia production. This has been brought about over the past decade or so largely by an increase in plant design capacity and a reduction of required reaction pressures.

Fundamental design considerations for centrifugal compressors are briefly reviewed. Particular emphasis is placed on the analytical tools at the designer's disposal which are afforded by the electronic computer, both in the initial design stages as well as in the interpretation of data taken during test of the equipment prior to shipment. Merits as well as possible penalties of low pressure testing versus full pressure testing are discussed. Selection of compressors for typical ammonia plants are offered and design details presented.

Finally, a review of serious operational problems which have occurred is provided and means of preventing such problems are suggested.

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CENTRIFUGAL COMPRESSORS FOR AMMONIA PLANTS

DESIGN AND OPERATION CONSIDERATIONS

Abstract

Centrifugal compressors are the accepted present-day standard equipment for synthesis gas compression for ammonia production. This has been brought about over the past decade or so largely by an increase in plant design capacity and a reduction of required reaction pressures.

Fundamental design considerations for centrifugal compressors are briefly reviewed. Particular emphasis is placed on the analytical tools at the designer's disposal which are afforded by the electronic computer, both in the initial design stages as well as in the interpretation of data taken during test of the equipment prior to shipment. Merits as well as possible penalties of low pressure testing versus full pressure testing are discussed. Selection of compressors for typical ammonia plants are offered and design details presented.

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CENTRIFUGAL COMPRESSORS FOR AMMONIA PLANTS

DESIGN AND OPERATION CONSIDERATIONS

Synthesis gas compressors for ammonia plants with design capacities of 500 to 1500 tons per day are almost exclusively centrifugals. To our knowledge, no new plants in this capacity range are being built with reciprocating machinery.

The fundamental design parameter most frequently used in the evaluation of a centrifugal compressor is a term defined as specific speed N_s expressed as follows:

$$N_s = \frac{N(Q/60)^{\frac{1}{2}}}{H^{3/4}} \qquad (1)$$

Where N_s = Specific Speed
N = Speed in RPM
Q = Stage inlet flow in actual ft.^{3}/min.
H = Polytropic head in feet.

The polytropic head is related to pressure ratio or pressure rise per stage as follows:

$$H = \frac{ZRT_1}{\frac{n-1}{n}} \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \qquad (2)$$

Where H = Polytropic Head in Feet
Z = Compressibility Factor
R = Gas Constant = 1545/M
M = Molecular Weight
T_1 = Inlet Temperature ^oR
P_1 = Inlet Pressure PSIA
P_2 = Discharge Pressure PSIA
n = Polytropic Compression Exponent

The significance of specific speed N_s is that impeller or stage polytropic efficiency may be related to it as shown in Figure 1.

This plot illustrates the desirability of operating at specific speeds as high as practical to achieve high efficiencies, and hence, lower power requirements.

Re-examination of equation (1) indicates that for a given through flow, γ_s, polytropic efficiency may be increased by:

(1) Increasing speed N, or

(2) Reducing head rise H, per stage.

Either approach is limited by practical considerations. Speed is limited by driver considerations, component stresses, critical speed considerations, bearing and seal limitations, and, if used, gear capabilities.

Reducing the head rise per stage obviously results in an increase in the number of stages required to achieve the desired overall pressure rise and, hence, the practical limitations are apparent. Depending on speed, eight to ten stages per case is usual with twelve stages being about the upper limit. Hence, total number of stages required governs the number of separate compressor cases necessary, which must be limited to avoid excessive costs and complexity.

The usual design compromise is to select impellers which provide high head with high inlet flow; i.e., high head per stage in the front of the plant, and impellers with reduced head at the smaller inlet flow at the high pressure condition at the back of the compressor train. This practice results in acceptable efficiencies for all stages and at the same time, keeps total number of stages within practical limits.

Regarding the maximum number of stages that can be incorporated into a single casing, the dynamic behavior of the machine for a given speed is the most important consideration. Only in recent times has the designer become fully aware of the many variables which effect the dynamic performance of high speed rotating machinery, but more importantly, has been able to effect major changes in the dynamic behavior of such a machine with rather minor changes to certain key components such as rotor supporting bearings and shaft seals.

Oil film journal type shaft seals are in universal use on ammonia compressors for separating feed and recycle gas from the atmosphere. These seals along with the journal bearings and the attendant hydrodynamic oil film characteristics of both, have a marked influence on the behavior of the rotor. Equally important is the mounting of seals and bearings in their respective housings.

By way of illustration, Figure 2 shows the change in dynamic behavior of a typical ten-stage rotor that can be achieved with different bearing characteristics. Both predicted and tested values are shown.

Use of the electronic computer with appropriate programming permits the designer to analyze a vast number of bearing and seal combinations to ensure satisfactory dynamic performance of any given rotor system.

At the discretion of the designer, unbalances of predetermined values can be located in a random fashion at various stations along the rotor shaft, and the effect on rotor performance over the speed range of interest can be determined. On the other hand, the designer may elect to pre-establish both unbalance and location for analysis.

For the most part, virtually all compressors for ammonia plants operate above the first critical speed of the rotor system and it is essential that acceptable rotor deflection is assured at speeds near this critical speed, since the rotor must go through this speed to reach both design operation, and to take the plant off stream.

Usually rotor unbalance is not a major problem, since it can be corrected during the manufacturing stage. If the rotor becomes unbalanced after some period of operation due to non-uniform deposits or excessive erosion or corrosion, then corrective action must be taken.

Until recently, the designer was limited to evaluating the dynamic behavior of the rotor systems as influenced by unbalance only. Currently, more sophisticated techniques are available which permit the designer to evaluate rotor response as affected by other disturbing forces such as "aerodynamic cross coupling" and "internal friction damping". Such forces can lead to rotor instability which may result in sub synchronous vibration. The potential hazard to avoid if such sub synchronous vibration cannot be entirely eliminated is to ensure that the frequency of this type of vibration is not at or near the system critical speed. Sub synchronous vibration being considered here should not be confused with so-called "half frequency whirl" or "oil film whip" which is sometimes brought about by plain journal bearings or oil film seals. This phenomenon has been almost entirely eliminated by use of tilt pad bearings and more recently, by tilt pad seals. The latter is an innovation conceived and developed by the author's company, so far as is known. A typical tilt pad seal design is shown in Figure 3.

In considering rotor instability, it is difficult to give physical meaning to such terms as "aerodynamic cross coupling" or "internal friction damping". One may reasonably assume that "aerodynamic cross coupling" is associated with the fluid dynamic forces acting on the rotor, whereas "internal friction damping" is brought about by the mechanical construction of the rotor and is influenced by shrink fits of impellers and sleeves on the rotor shaft. Even so, it is difficult for the designer to evaluate and assign meaningful values to these forces for a particular rotor design which is to operate in a given environment. The designer must, therefore, depend heavily on experience or resort to full pressure testing for confirmation of his analysis.

Full load and full pressure testing is the most conservative approach, but this obviously is more costly and frequently cannot be justified for this reason. Justification of such testing is particularly difficult when design conditions are similar to successful plants and compressors already in service. However, with new applications where the required equipment necessitates much extrapolation of experience, full load testing is often indicated.

In connection with such testing, the author's company has facilities available for testing to 10,000 psig and 30,000 horsepower. Thus, compressor trains consisting of two or more compressor cases and any associated gearing can be tested at field conditions if the client desires.

Figure 4 shows a view of a contract compressor and gear under test at a discharge pressure of 5500 psia and 22,250 horsepower. Compressor speed is 9600 RPM. This particular train is used for gas reinjection in a North American oilfield. It represents one train of several being produced for this service.

As a minimum, all compressors produced by Dresser Clark are mechanically tested. The majority, of course, are tested at pressures less than design because of cost, as already noted. Hydraulic performance testing is also performed if the customer desires. Figure 5 shows one of our larger fabricated units on test. This machine is now in service handling stack gas. Design capacity is 230,547 ACFM and pressure ratio is 1.5/1. Compressor speed is 2900 RPM and brake horsepower is 5500. This machine has dual inlet connections with two stages in each section.

Data from both mechanical and hydraulic tests are automatically recorded and stored on tape for playback at will on our computer complex which is provided and used exclusively for test purposes. Figure 6 shows the test control room and computer installation.

To discuss compressor design details and construction, we have selected two typical ammonia plants. Figures 7 and 8 show schematically the synthesis gas compressor trains for a 1000 ton per day plant and a 1500 ton per day plant. The 1000 ton per day plant uses a three-casing train as shown in Figure 7, and the 1500 ton per day plant is served by a two-casing compressor train, as indicated in Figure 8. Aside from capacity, the primary difference in the two plants is the final discharge pressure which, as shown, is 4500 psia for the 1000 ton plant, compared to 3250 psia for the 1500 ton plant. The design conditions shown are intended to be typical and should not be construed as an illustration of any specific plant.

In both plants, the first or low pressure casing is our Model 463B5/5 unit. This is a ten-stage machine with a back to back impeller arrangement. Figure 9 is a cross-section view showing major components. This is a vertically split design with the casing being a simple forged steel cylinder. Heads or end closures are also steel forgings. Heads are positioned and retained by shear rings which take the hydrostatic end loads produced by internal casing pressure. The shear ring design, as illustrated in Figure 10, is a unique Dresser Clark development covered by U. S. Patent 3,552,789. The axial end load on the shear ring resulting from internal pressure on the head is necessarily eccentric with the casing reaction since the outside diameter of the head must be slightly less than the casing bore. This eccentricity results in a torsional or twisting moment on the shear ring with corresponding high torsional stress and deflections unless prevented.

As Figure 10 illustrates, the unique design geometry is such that these eccentric axial loads induce equal and opposite force acting in a radial direction. This induced couple balances the eccentric end load system, resulting in essentially a state of "pure shear" in the shear ring without attendant rotation. Sealing between casing and heads is accomplished by elastomeric "O" rings and companion backup rings. This head and casing design is simple and reliable. It is now in use on over 100 units now in service.

Nozzles are cast steel welded to the casing. The design pressure rating of this unit is 2000 psi. The complete machine is shown in Figure 11.

The ten stage rotor consists of two sections of five impellers each. Impellers in the two sections are oriented back to back and axial impeller thrust is essentially balanced. Hence, no balance piston is required. Any small residual thrust is carried by the double acting tilt pad thrust bearing with equal load carrying capacity in either direction.

Impellers are of welded construction. Depending on flow capacity, they may be of either three piece or two piece construction. The high capacity three piece design consists of a number of formed vanes welded to a separate pre-machined disc and cover. This assembly is stress relieved, inspected, heat treated and final machined to the proper contour. Impeller material is usually AISI 4330 or 410 stainless steel and welding is done by a semi-automatic MIG process with appropriate filler material for the base metal. Figure 12 shows a finished welded impeller of the three piece type construction.

Two piece welded impellers are constructed by machining the vanes as an integral part of the cover and welding the vanes to the disc in one of two ways. For moderate capacity impellers, the vanes are welded to the disc by a double fillet weld between the vanes and disc again using the semi-automatic MIG process. For low capacity impellers where the vane height is small compared to the impeller diameter, slots are machined in the disc to match and align with the vanes on the cover, and the weld is made between vane and disc. The machined slots are completely filled with weld metal. Figure 13 shows a finished impeller fabricated by the slot welded method, and Figure 14 shows the matching cover with integral vanes.

Impellers as just described are the main rotor components. The shaft, impeller spacers, and associated hardware follow conventional standard practice. Shaft material is usually AISI 4330 carbon steel, although other materials are acceptable. Figure 15 is a view of the completed rotor.

The diaphragms or stator components, which form the diffusers and return flow channels between stages, are ductile iron castings. For low pressure applications, ordinary grey iron castings may be used.

Interstage labyrinth seals at the impeller eye and shaft are aluminum as standard, although other materials are available.

End shaft seals between the gas and atmosphere are oil film bushing types. Seal oil supply pressure is maintained at approximately 5 psi above gas pressure by overhead reservoirs about 15 feet above the sealing elevations. Figure 16 illustrates a typical seal installation.

Journal bearings are of the tilting pad type as is the double acting thrust bearing.

The second casing in the 1000 ton per day plant is our Model 272B5/5 unit. This is also a ten stage machine with two sections of five impellers each arranged back to back. Figure 17 shows a cross section of this machine.

For the most part, essential design features of this unit are the same as already described for the 463B, with the exception of pressure rating which has been increased to 5500 psi. Also milled flat areas are provided for piping connections rather than nozzles. Figure 18 is a view of a completed 272B.

The second and last casing in the 1500 ton per day plant is a 373B unit. In this machine, eight feed gas stages are arranged back to back with a single recycle stage. The back to back impeller arrangement provides more offsetting thrust than might at first be apparent even though there are eight feed gas stages and only one recycle stage. The recycle impeller is of much greater flow capacity than the feed gas impellers and hence, the unbalanced area is greater; therefore, more offsetting thrust. Nevertheless, two balance pistons are used to achieve independent thrust compensation for the feed gas and recycle sections. This feature is a distinct advantage and safeguard when operating at off design conditions such as plant startup or emergency tripout.

Figure 19 is a cross section of this machine. Depending on recycle pressure rise requirements, a two-stage recycle section is also available. Case pressure rating is 4000 psi. Construction details are similar to the two machines just described.

The final casing is the recycle unit for the 1000 ton per day plant. This is our Model 171B design which has a case pressure rating of 6500 psi. Figure 20 is a cross section of this unit Here six feed gas impellers are arranged back to back with a single recycle stage. Again, two recycle stages are available if required, and as before, double balance pistons are provided. Figure 21 is a view of the complete machine, and Figure 22 is a view of the rotor.

Considering now, operation of centrifugal compressor systems, it is generally accepted that rotating machinery is inherently safe and reliable and gives long trouble-free life. Experience indicates this to be particularly true for process gas compressors of the rotating type. There are, however, certain rules and safeguards that must be applied and followed to fully realize these advantages.

First, centrifugal compressors are relatively high speed machines, and hence, susceptible to damage by any solid foreign object that may be injested during operation. Unfortunately, the primary damage caused by such a mishap may in itself be relatively minor, but the ensuing rotor unbalance may result in severe rotor deflections and violent internal rubbing with generally disastrous results. Frequently, if such an accident occurs, shaft end seals are destroyed and a fire results. Obviously, this is an extreme situation, but nevertheless one which must be recognized as a possibility.

With regard to the hazards of foreign objects, plant startup and shakedown is perhaps the most critical time in the life of the compressor. During plant erection and equipment installation, it is essential to practice good housekeeping and constantly maintain vigilance to ensure that unwanted debris does not find its way into the compressor or process piping and other connected equipment. Experience has shown that a variety of objects including hand tools, bolts, nuts, welding electrode ends, welding backup rings, and the like do find their way into the equipment.

One safeguard against such objects and one which Dresser Clark strongly recommends, is inlet screens at all compressor inlets. These screens, in our opinion, are not only essential for startup and shakedown, but highly desirable at all times. Of course the screens must be cleaned from time to time to avoid high pressure loss and flow starvation which in the extreme case, could lead to compressor surge.

By way of illustration, Figure 23 shows a piece of threaded tie rod 1/2 inch in diameter by 2-1/2 inches long, which was ingested into the first stage of a typical syn gas compressor. Figure 24 shows the tie rod piece locked in the eye of the impeller and Figure 25 shows the damaged rotor. Fortunately, in this example, relatively minor damage occurred. End seals and bearings continued to function during rundown from tripout and damage was confined to the rotor and labyrinth seals.

Figure 26 is a schematic view of a typical inlet screen. Note a pressure gage is mounted to observe any increase in pressure loss and a cleanout access is provided. The conical shape of the screen results in maximum free flow area.

It should be recognized that the supporting frame for the screen as well as the screen itself must be relatively rugged to ensure that this method of protection does not become part of the problem.

As discussed, startup is perhaps the most hazardous period for the compressors, but damage from foreign objects is not necessarily limited to this time. Many Auxiliary pieces of equipment must be installed in or made a part of the compressor and piping system and such equipment may be a prime suspect as a source of tramp metal.

Figure 27 shows an impeller with the cover removed and illustrates vane damage caused by a part of a ball valve rachet. Figure 28 shows the recycle impeller of a nine-stage compressor damaged by injesting the seat of a six-inch ball valve. Figure 29 shows a slug of steel perfectly fitted into the channel inlet of a recycle impeller. This object was found at the inlet of a downstream intercooler. Parts of demister screens from upstream condensate separators have frequently been found in first stage impellers.

Depending on plant design and layout, process gas compressors are frequently supplied with main gas connections downward. A popular misconception is that this arrangement provides protection against injesting foreign objects into the compressors. It is true that this arrangement is less susceptible to having objects fall into the compressor during installation and plant erection, but in operation, no significant difference exists when compared to an upward connected system. For typical gas densities and velocities encountered in practice, objects of significant size and weight are easily transported by the gas stream. Reference 1 presents a detailed analysis of the dynamics of this phenomenon.

Another source of major potential damage is excessive liquid entering the compressors. Reference 1 also discusses at length the source of such liquid and the destructive internal forces that can be developed by the impellers. Liquid, even in small quantities, can also lead to secondary problems of corrosion, erosion, and fouling of the gas passages.

In final summary and conclusion, it is essential that the compressor plant be treated as an entire system. Careful engineering attention must be devoted to all components serving the system. Structural integrity of auxiliaries such as coolers, condensers, valves and the like must be assured to realize full reliability potential. Close engineering scrutiny and prudent evaluation during the selection and design phase of the plant equipment will be paid off many times over.

LIST OF ILLUSTRATIONS

FIG. 1

POLYTROPIC EFFICIENCY
vs
SPECIFIC SPEED (Ns)

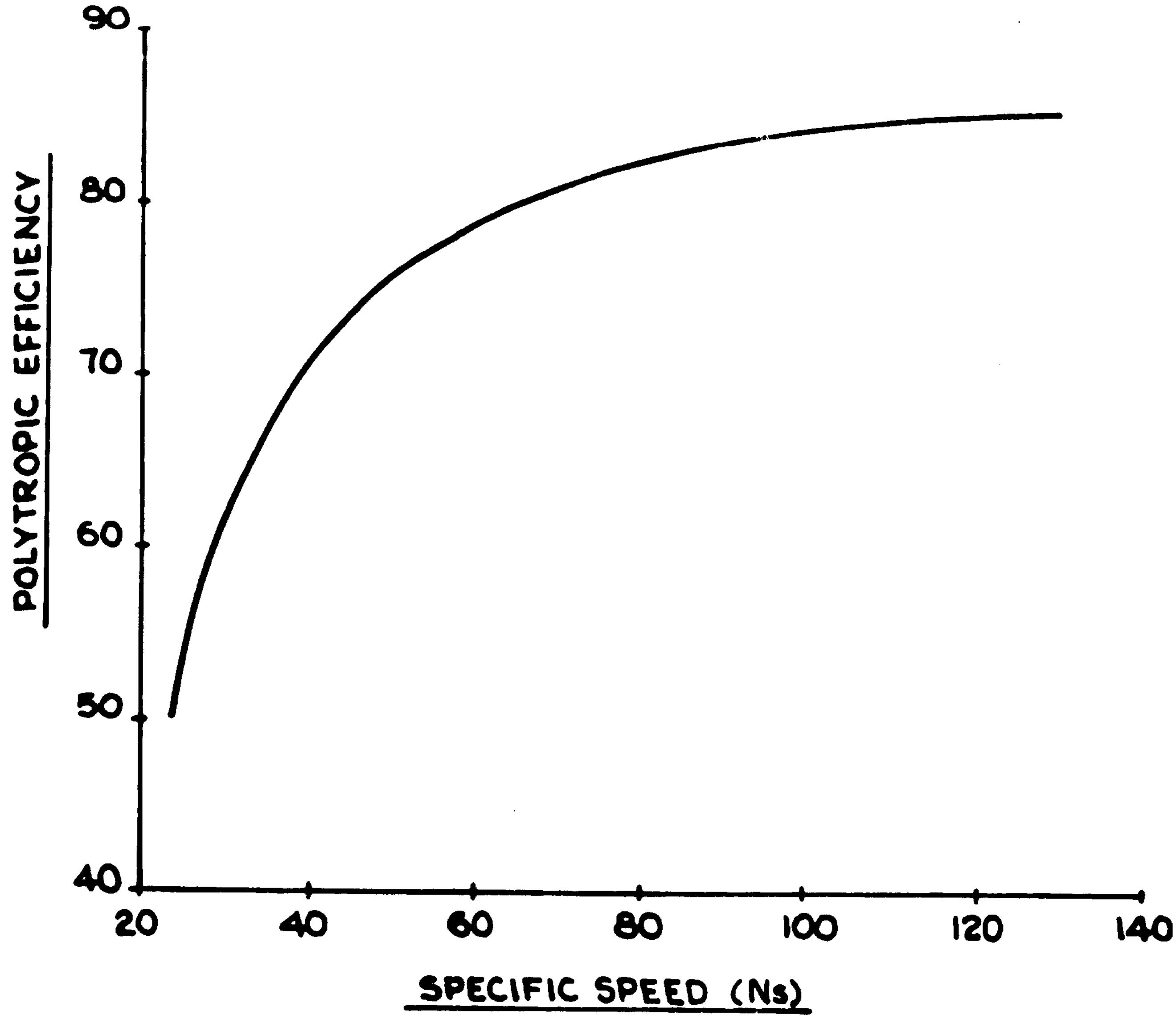
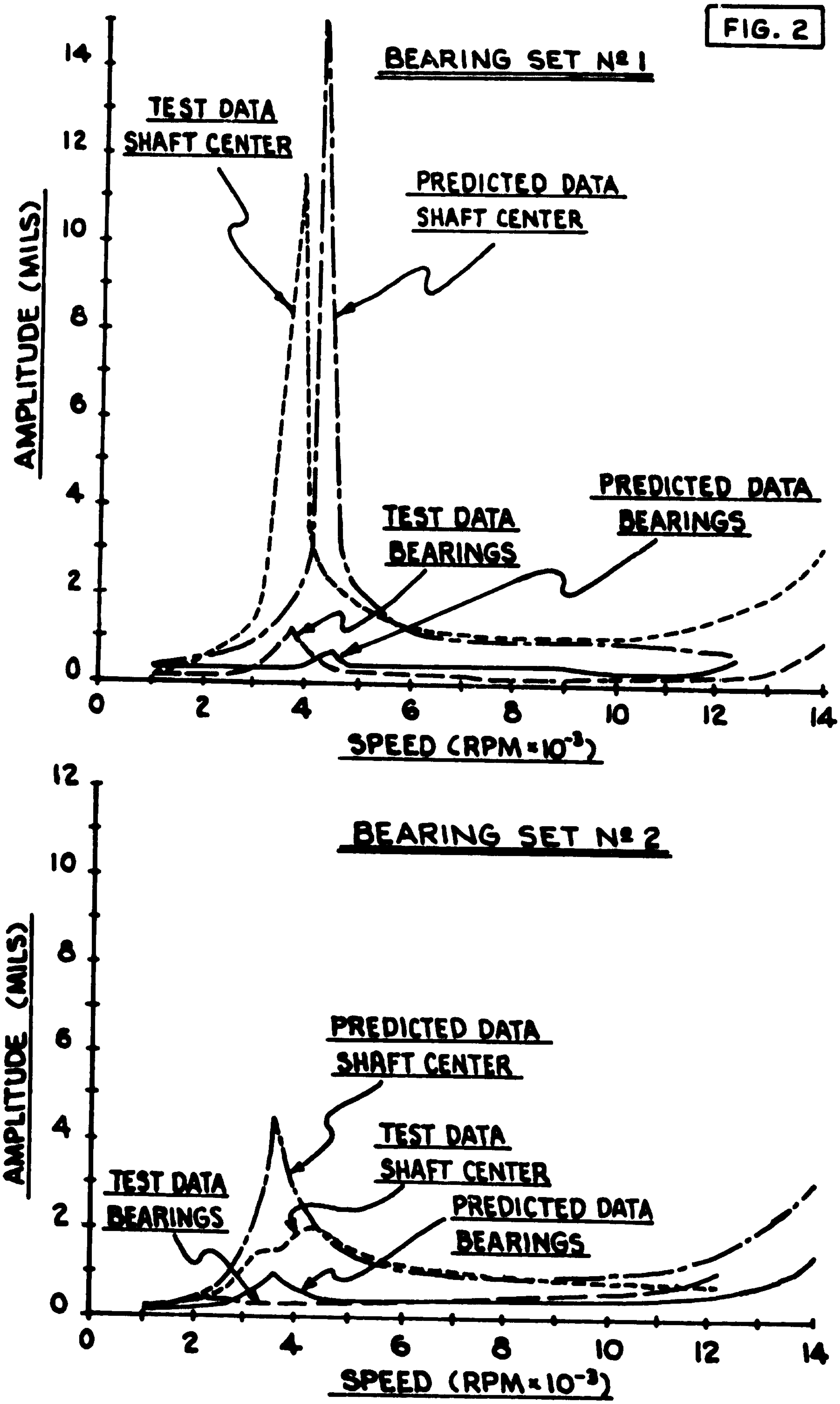


FIG. 2


BEARING SET Nº 1
TEST DATA
SHAFT CENTER
PREDICTED DATA
SHAFT CENTER
TEST DATA
BEARINGS
PREDICTED DATA
BEARINGS
AMPLITUDE (MILS)
0
2
4
6
8
10
12
14
SPEED (RPM×10⁻³)
BEARING SET Nº 2
PREDICTED DATA
SHAFT CENTER
TEST DATA
SHAFT CENTER
TEST DATA
BEARINGS
PREDICTED DATA
BEARINGS
AMPLITUDE (MILS)
SPEED (RPM×10⁻³)

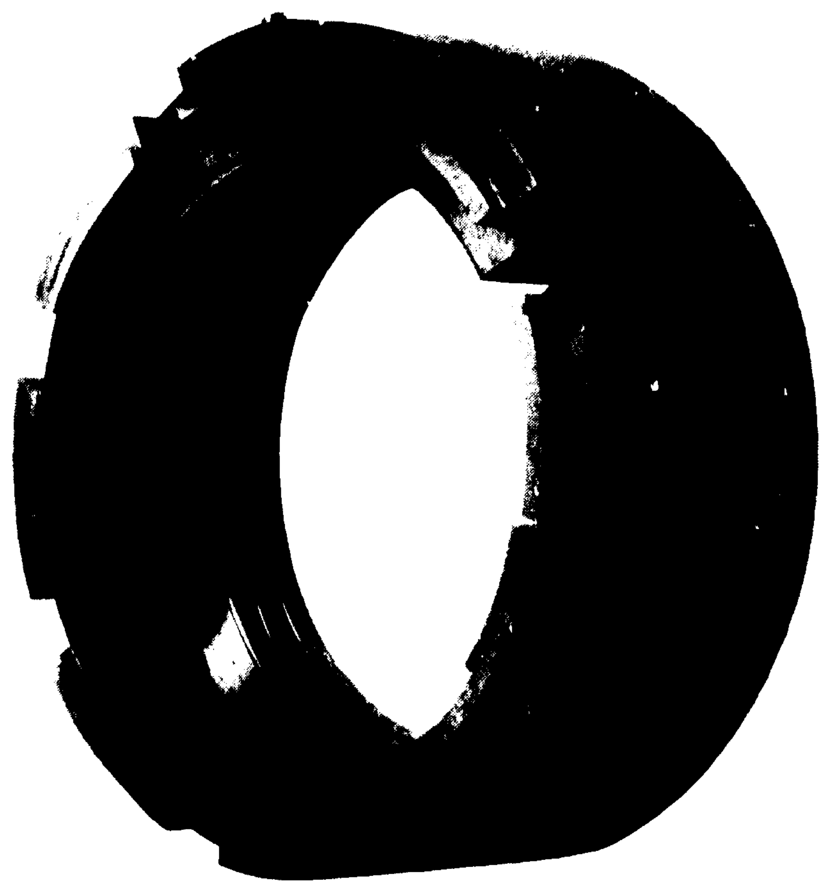


Figure 3 - Tilt Pad Seal

Figure 4 - Full Load & Pressure Test Barrel Compressor

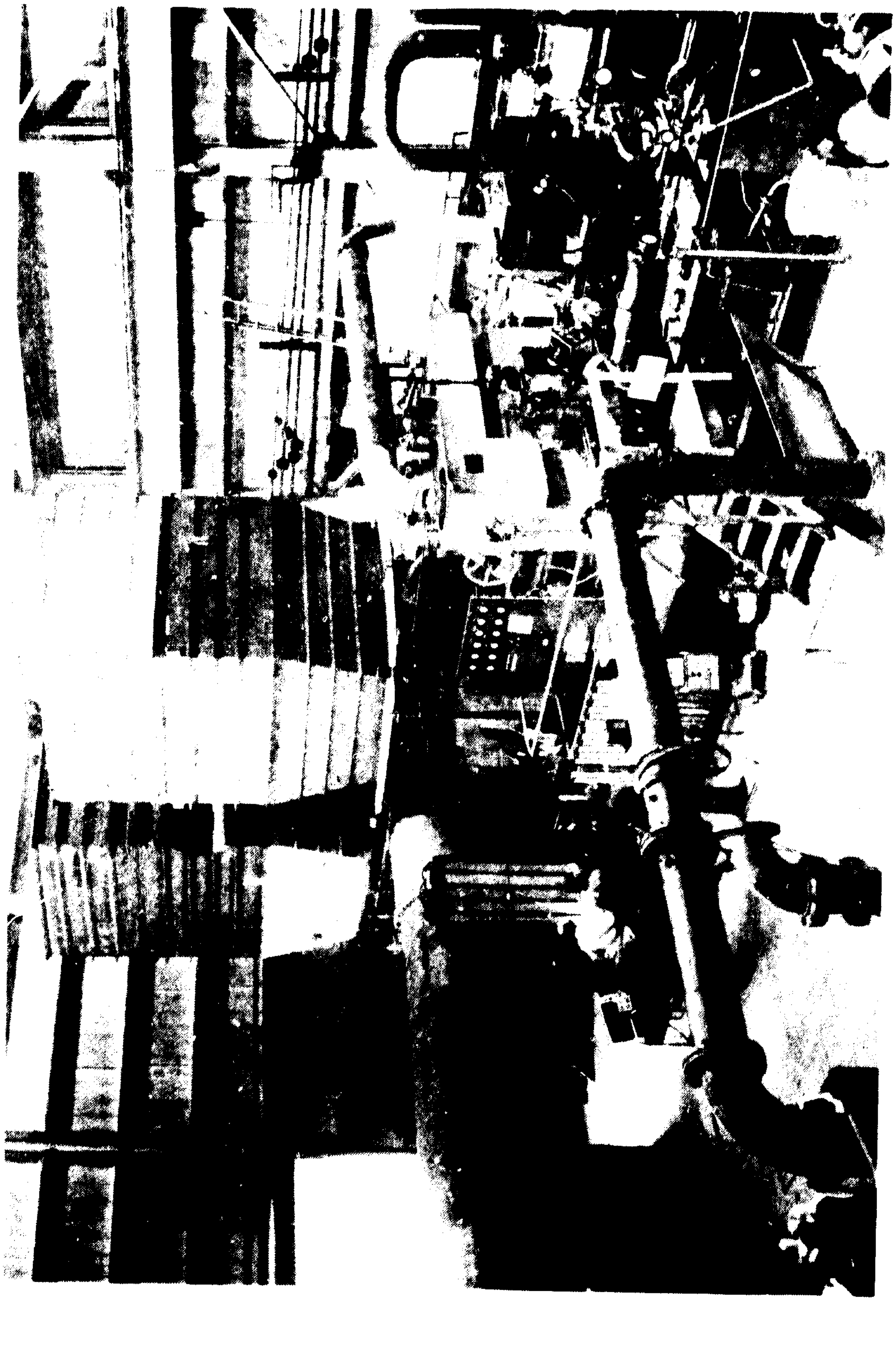


Figure 5 - Booster Compressor on Test

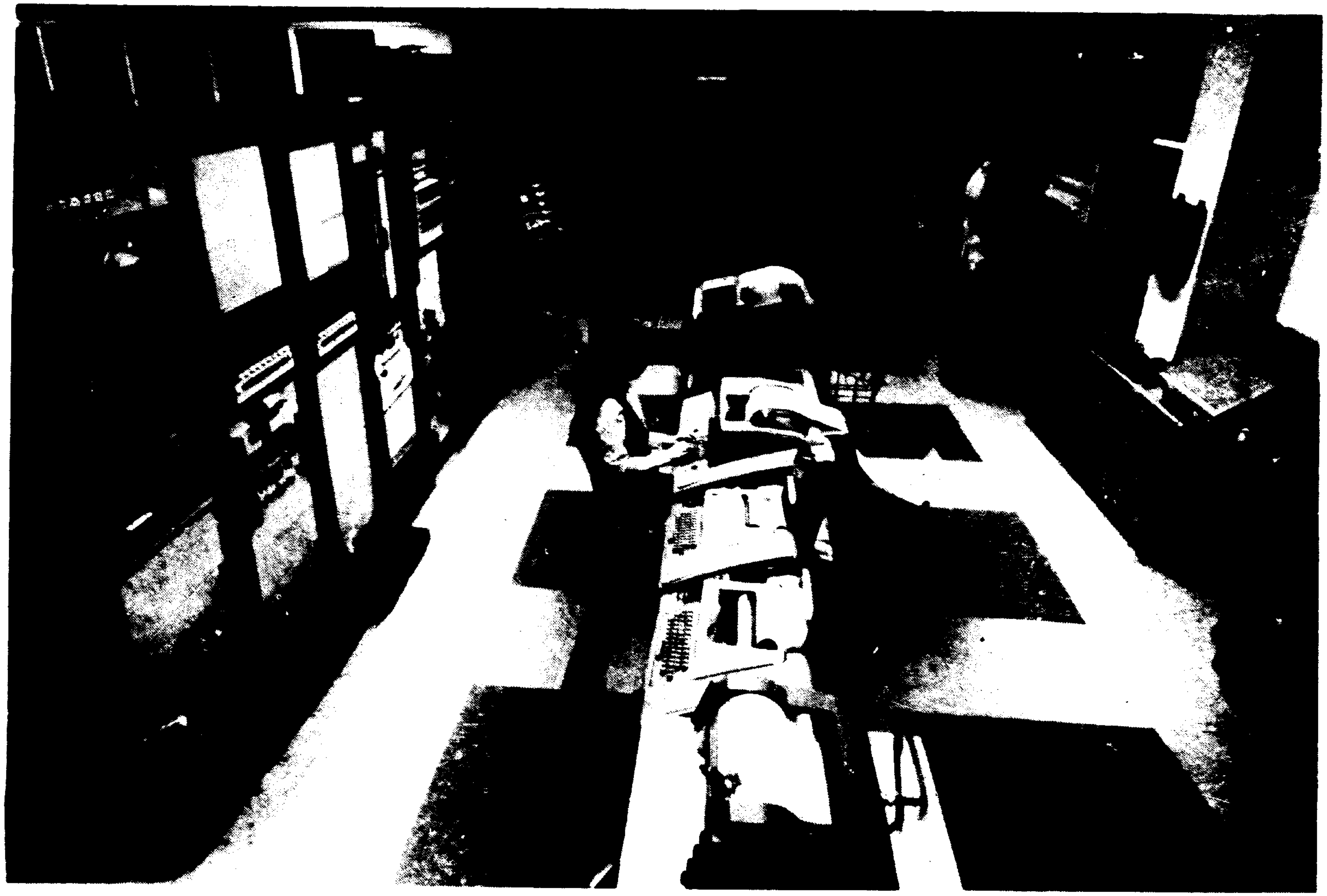
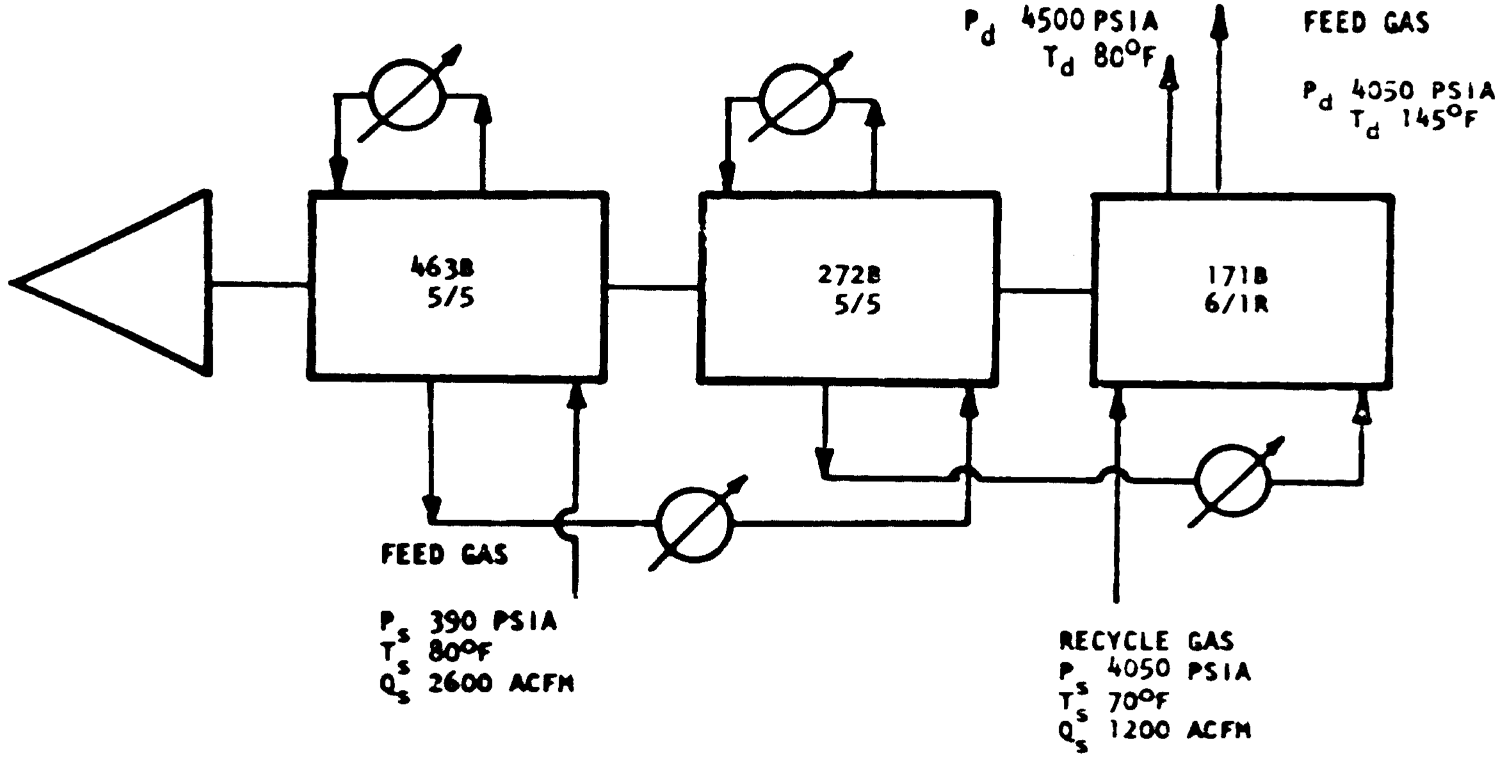


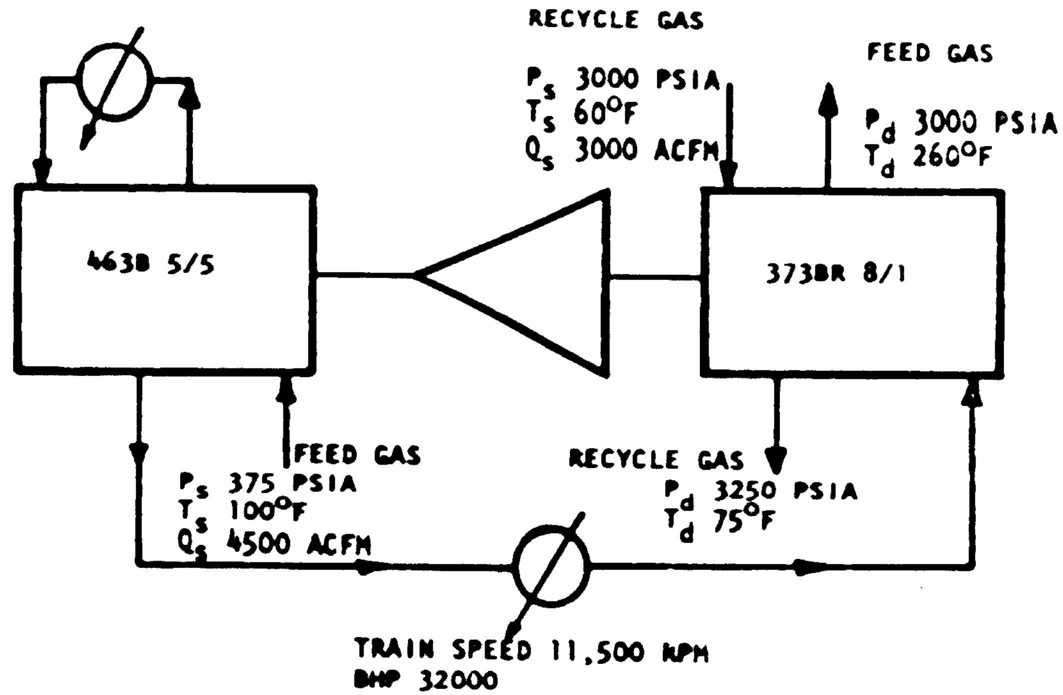
Figure 6 - Test Dept. Control Room & Computer Center



TRAIN SPEED 10,500 RPM
BHP 12,000

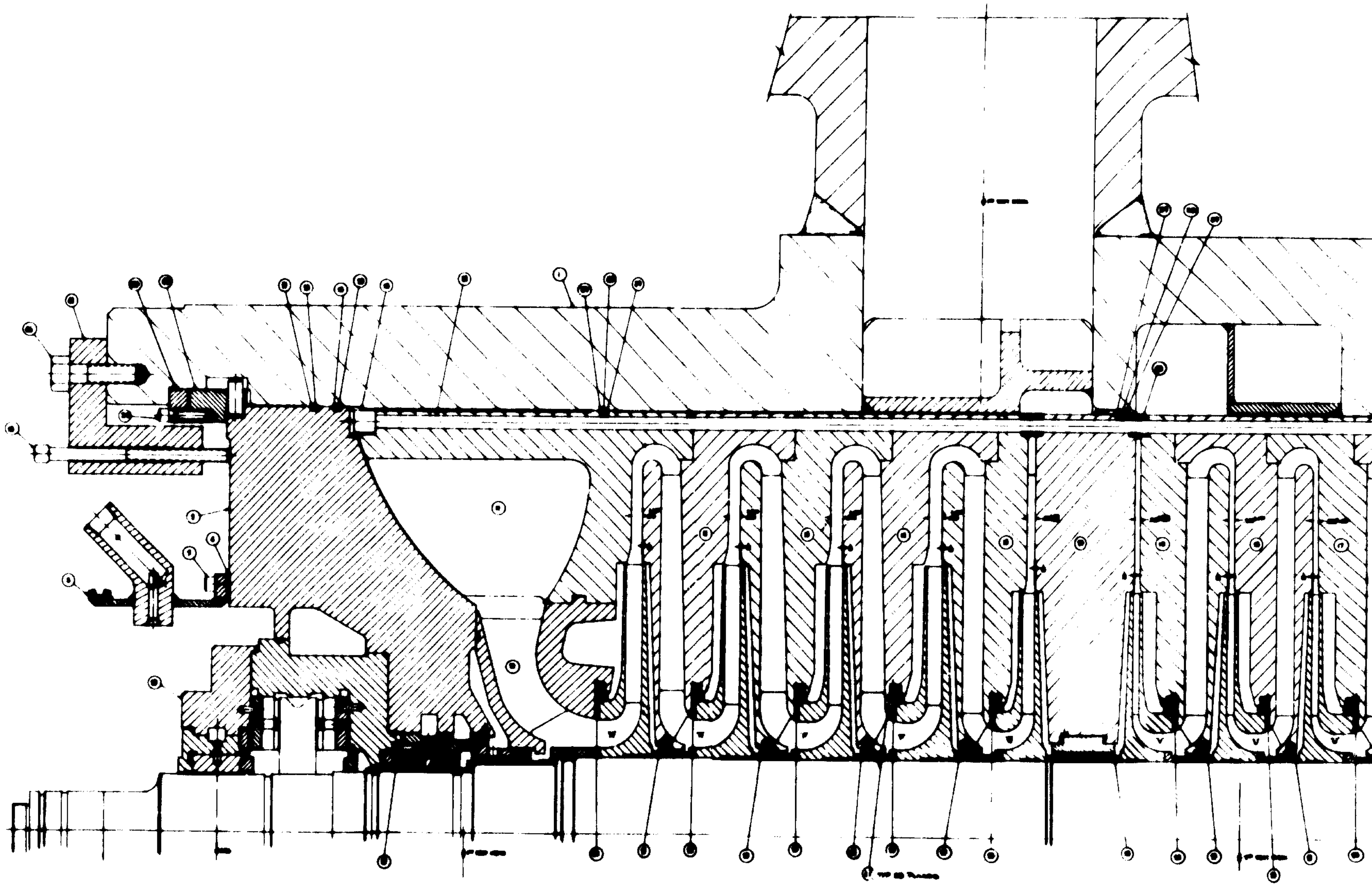
SCHEMATIC
1000 TON/DAY
AMMONIA PLANT

FIG. 7



SCHEMATIC
1500 TON/DAY
AMMONIA PLANT

FIG. 8



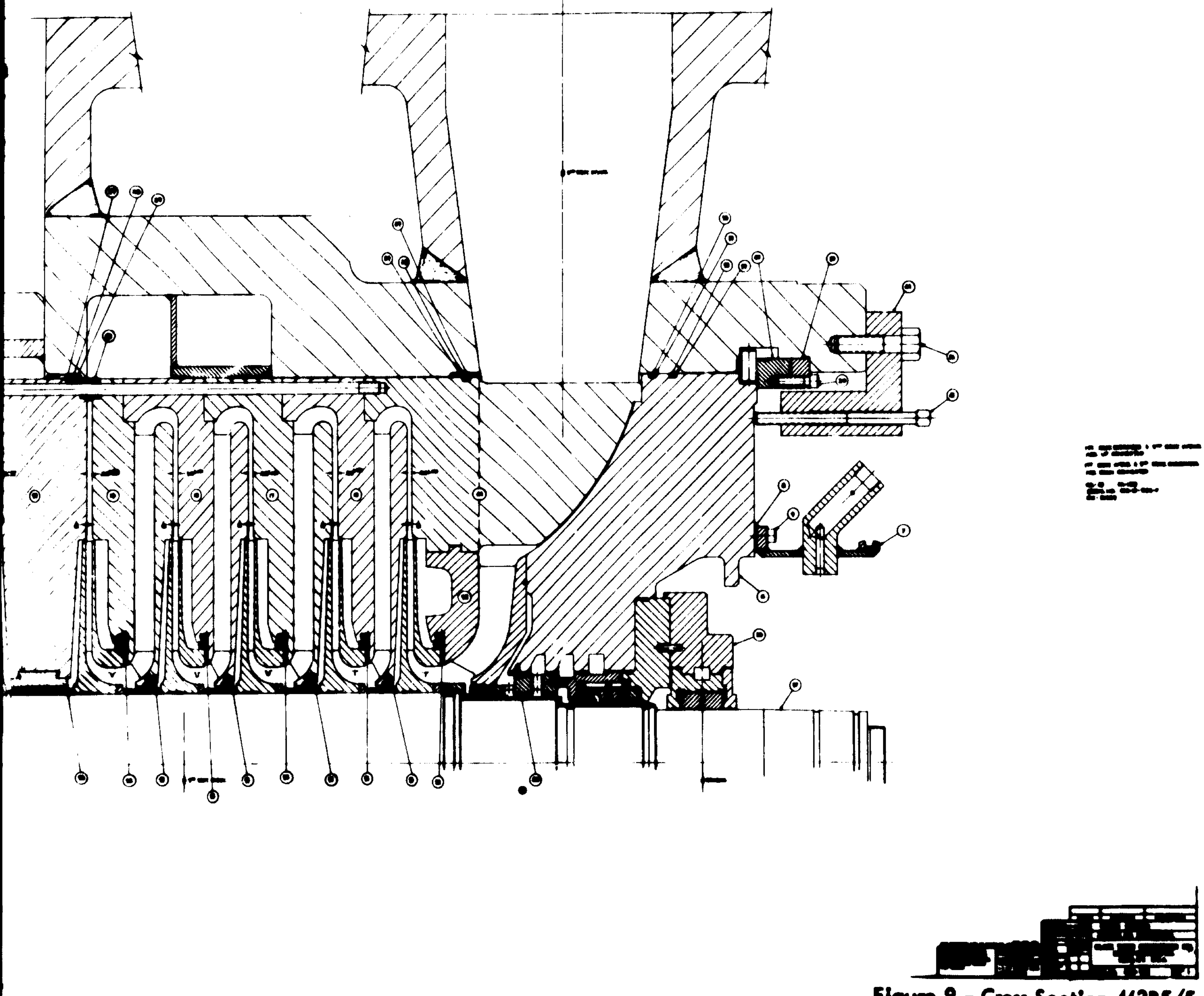


Figure 9 - Cross Section 463B5/5

SECTION 2

SHEAR RING

The pressure and reaction forces acting on the shear ring are offset by a system of equal and opposite forces forming a couple. Thus each general section of the shear ring is in equilibrium.

The shear ring is made in several segments from the standpoint of convenience of assembly. Also, smaller segments result in a lower overall stress level.

Other advantages include:

a) faster assembly and disassembly
b) elimination of precise torque requirements of bolted head designs

Clark have successfully used shear rings up to 6,500 psi levels. All medium and high pressure barrel compressors and some pipeline boosters designed in the late 60's have this feature as a standard one.

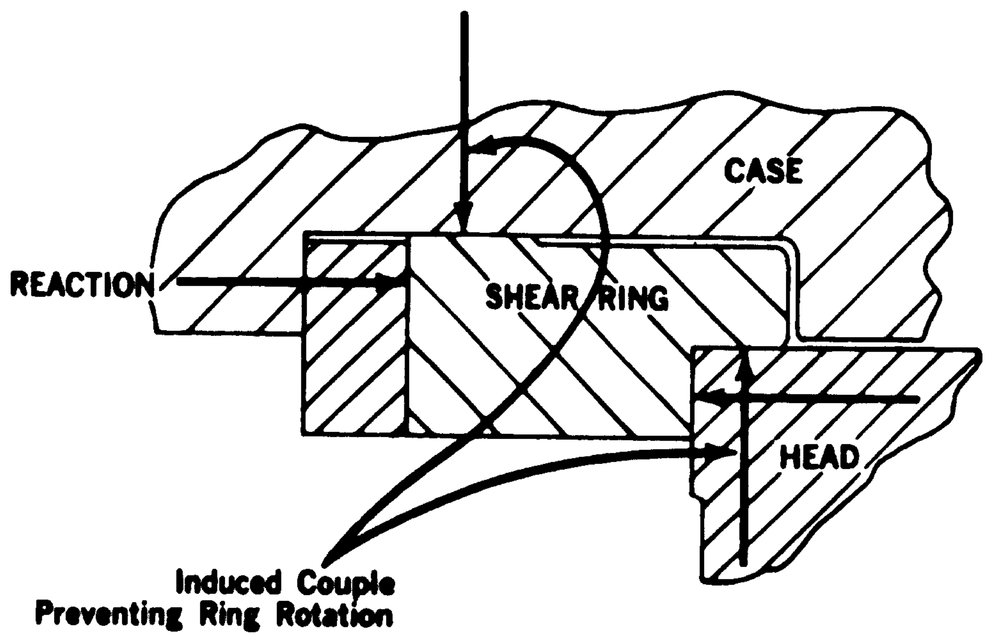


Figure 10 - Shear Ring Design

U.S. Pat. 3,552,789

Figure 11 - 463B Unit

Figure 12 - 3 Piece Welded Impeller

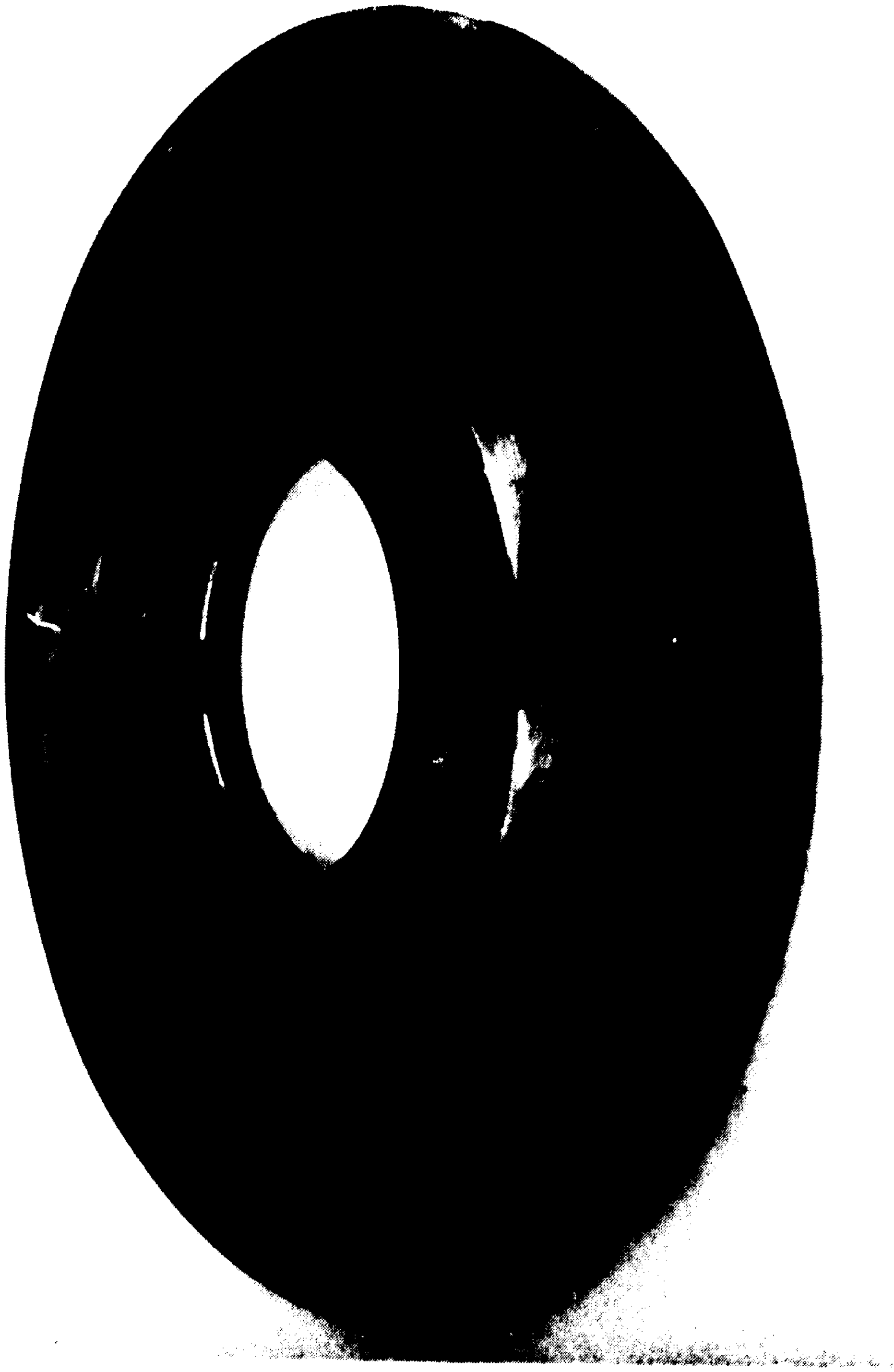


Figure 13 - Finished Slot Welded Impeller

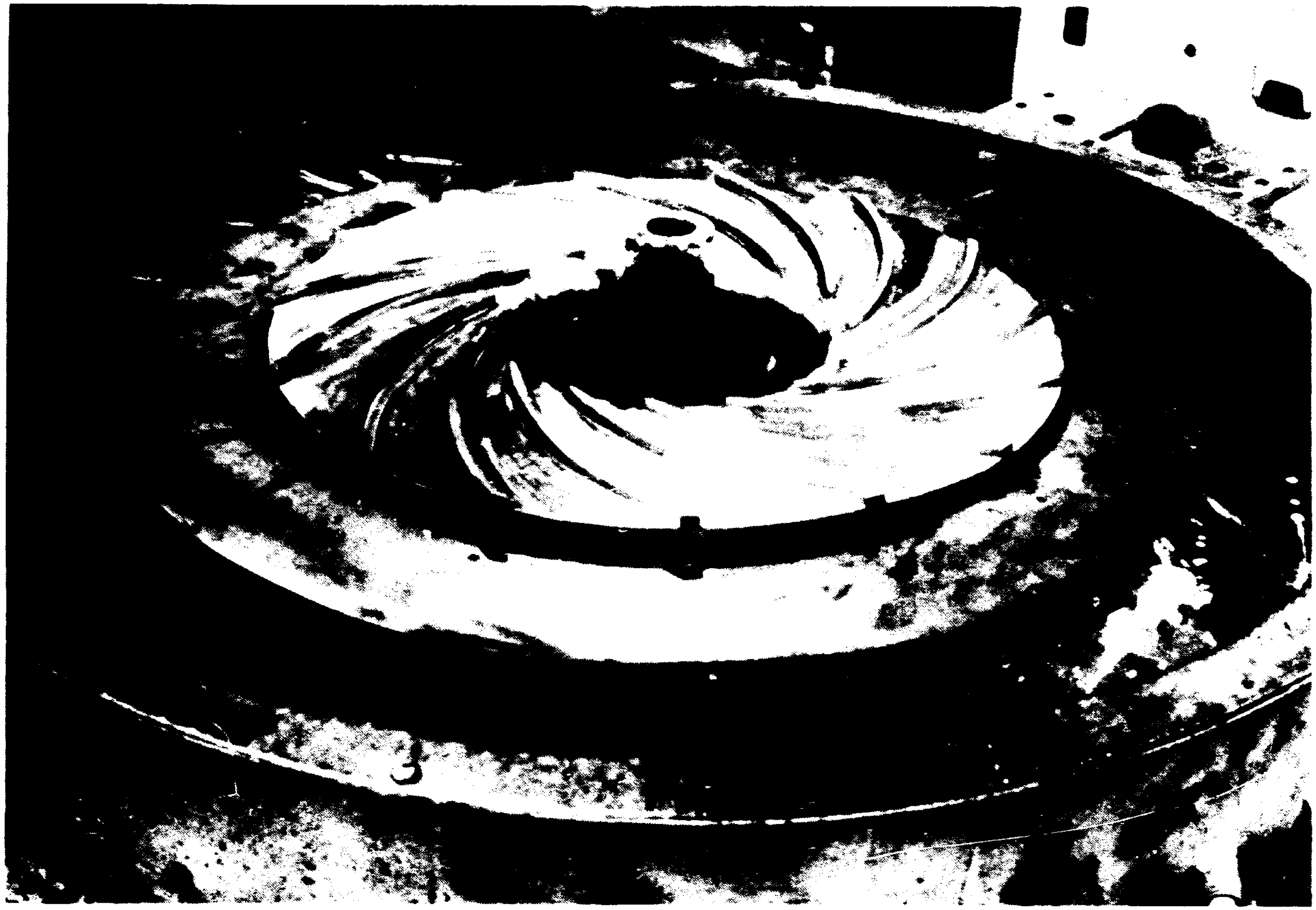


Figure 14 - Vanes Integral with Cover for Slot Welded Impeller

Figure 15 - 463B Rotor

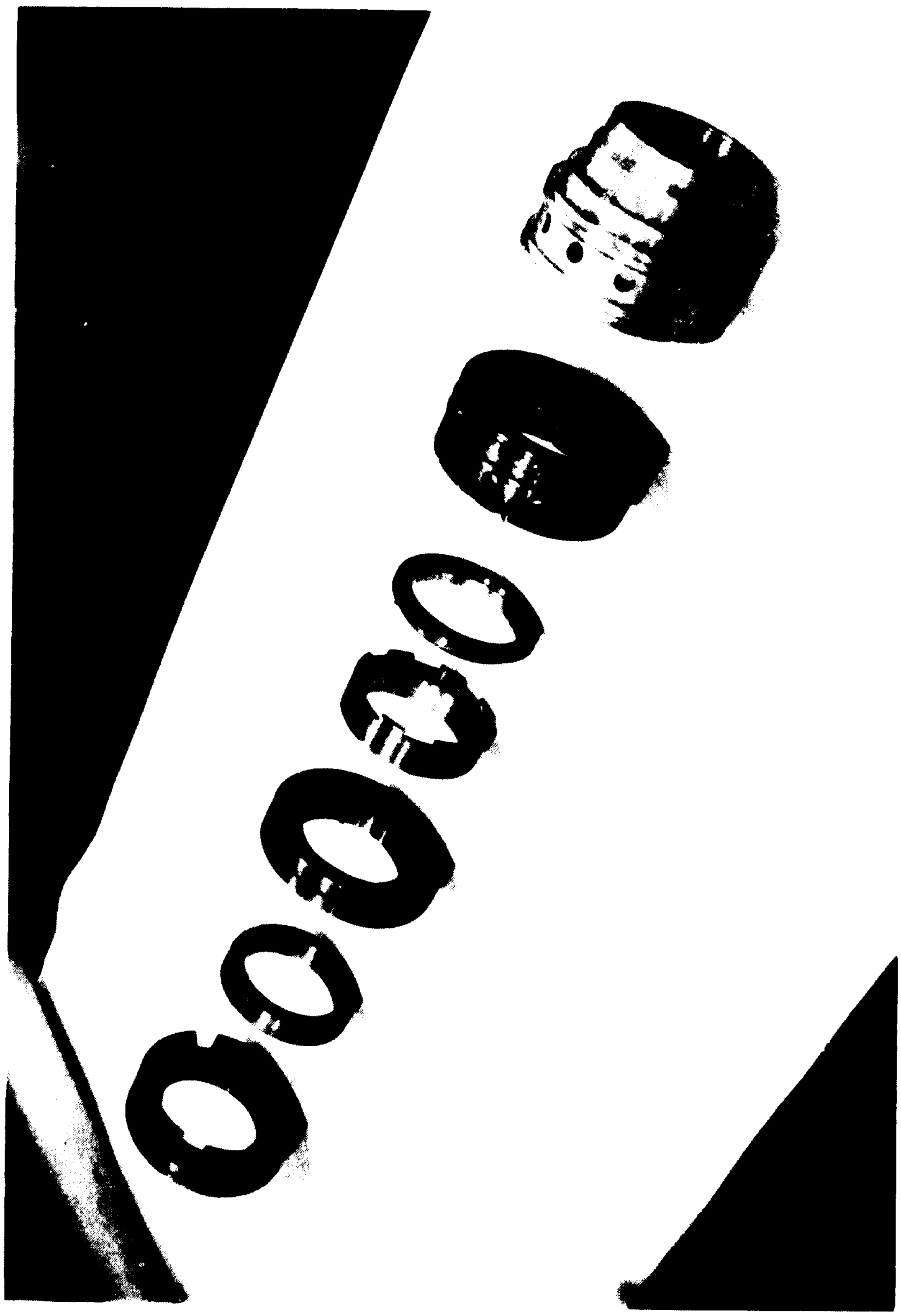
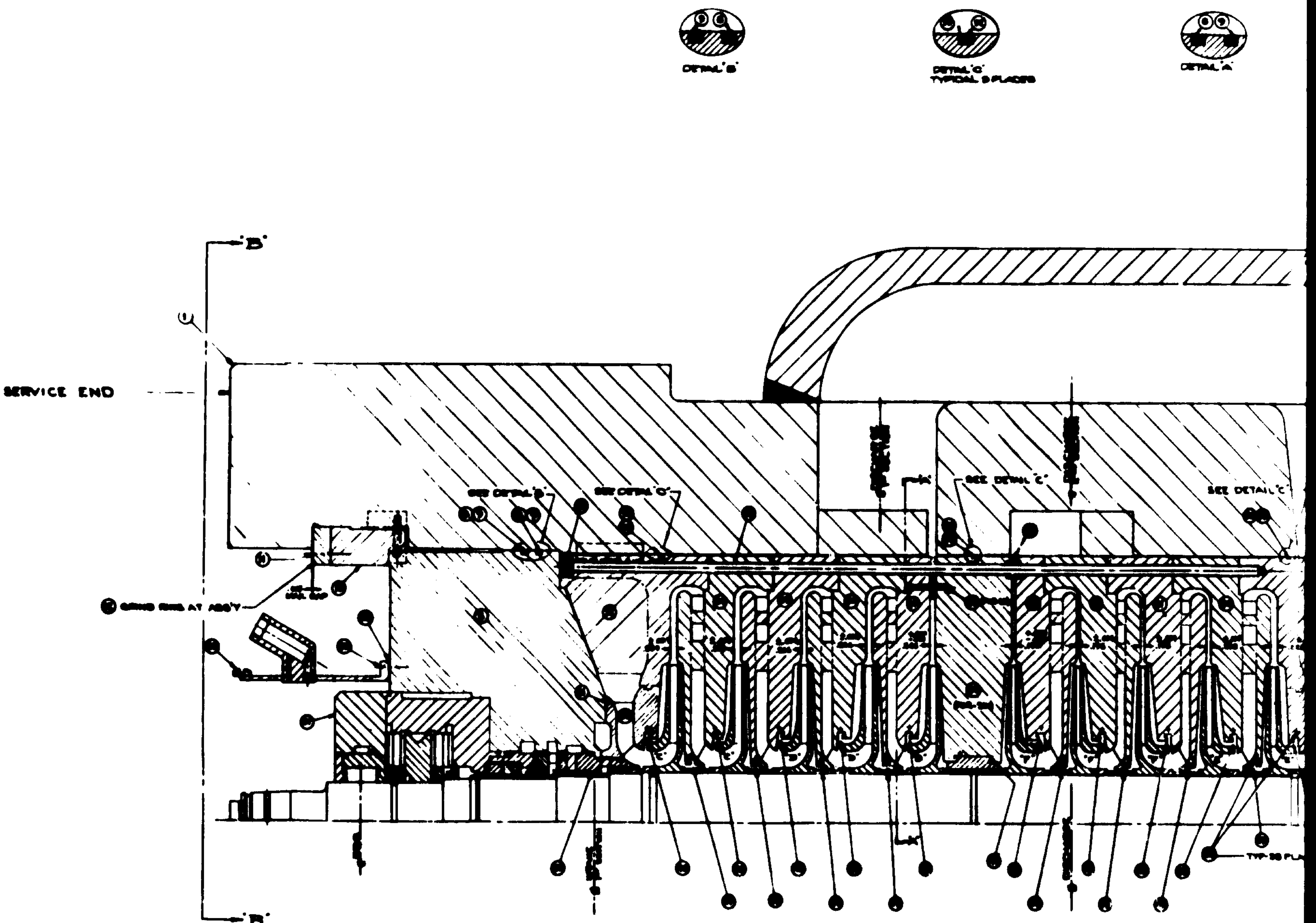


Figure 16 - Typical Oil Film Seal Assembly


DETAIL 'B'
DETAIL 'C' TYPICAL 9 PLACES
DETAIL 'A'
SERVICE END
SEE DETAIL 'B'
SEE DETAIL 'C'
SEE DETAIL 'C'
SEE DETAIL 'C'

SECTION

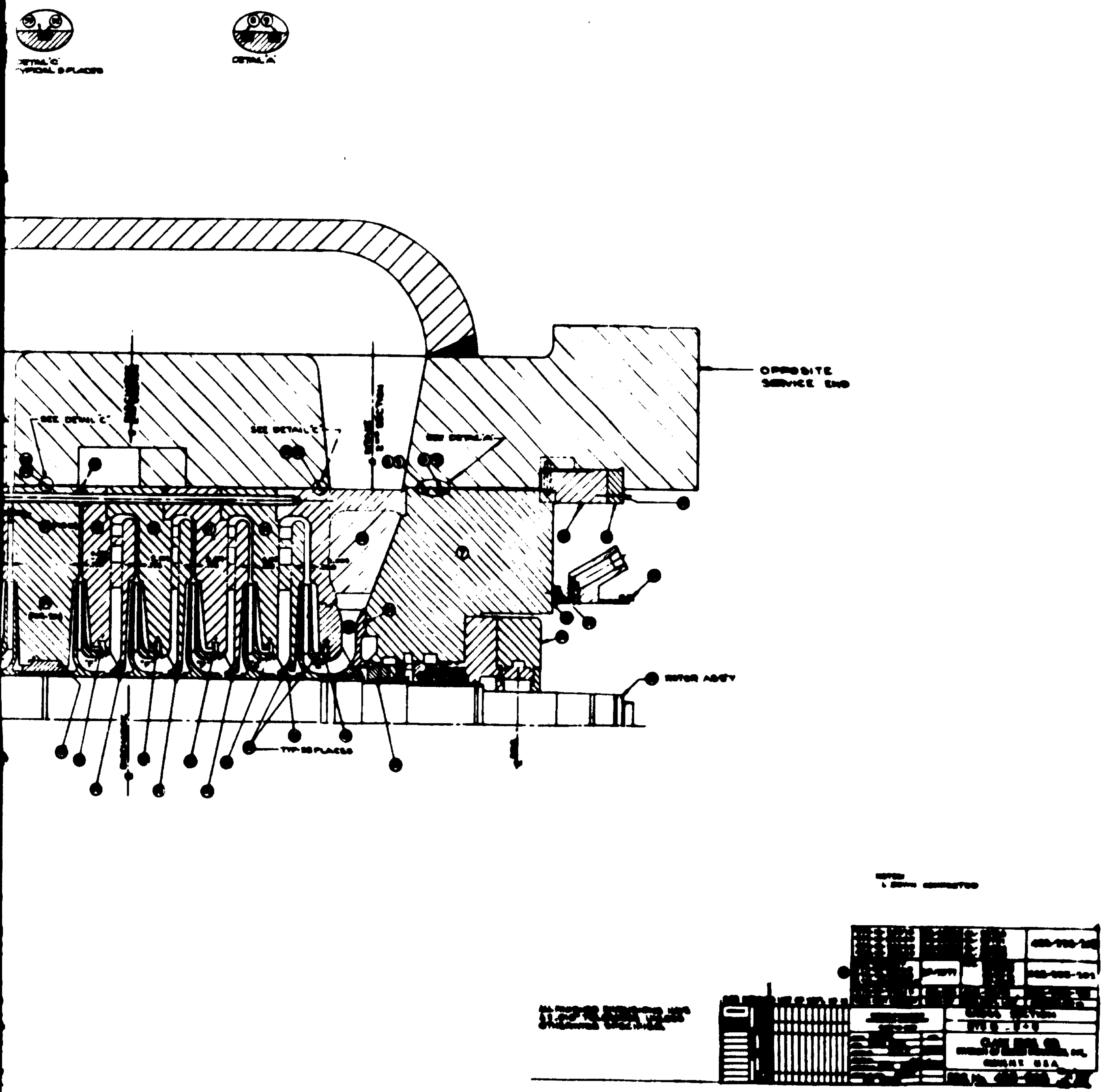


Figure 17 - Cross Section 272B

SECTION 2

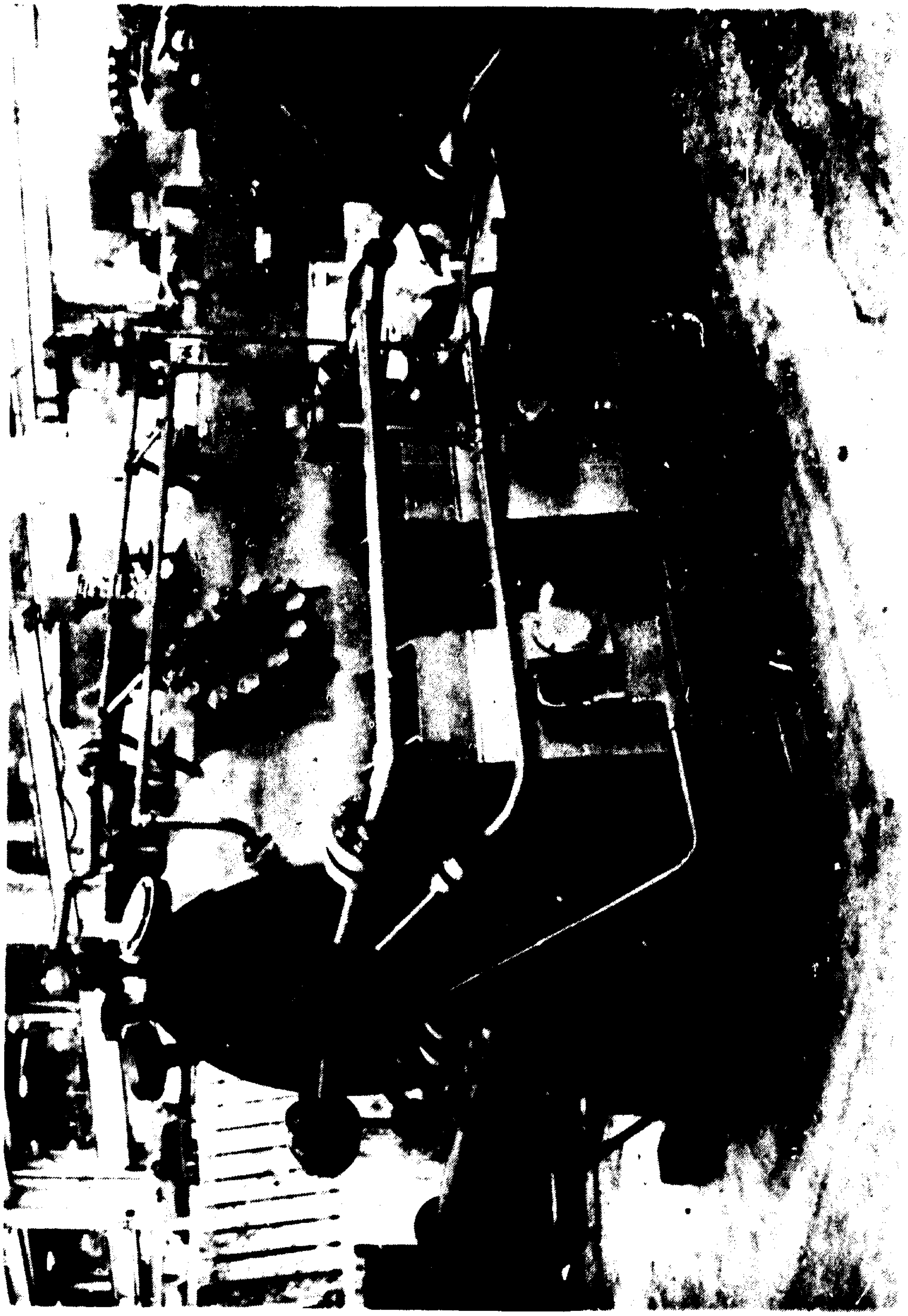
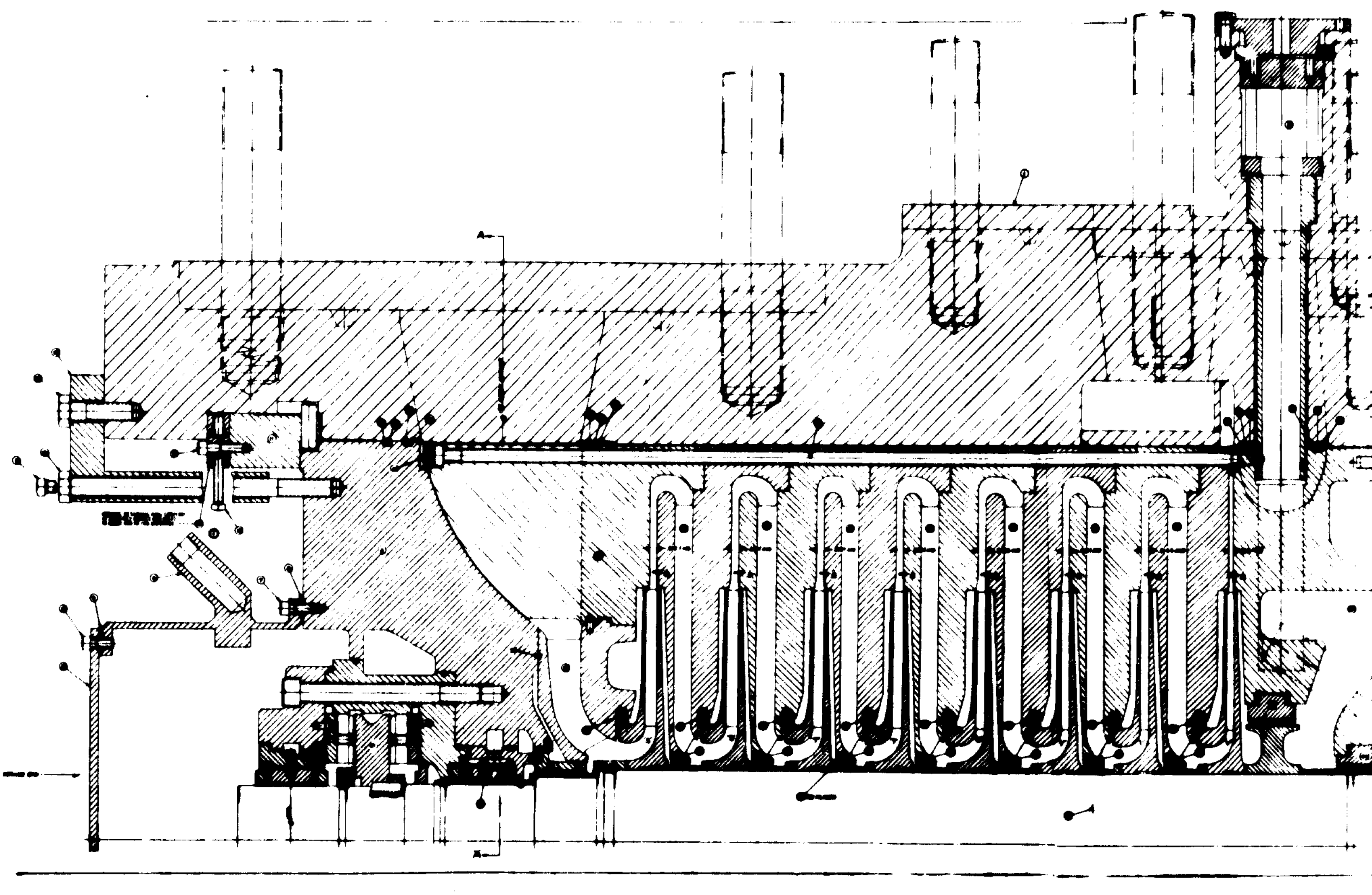


Figure 18 - 272B Compressor



SECTION 1

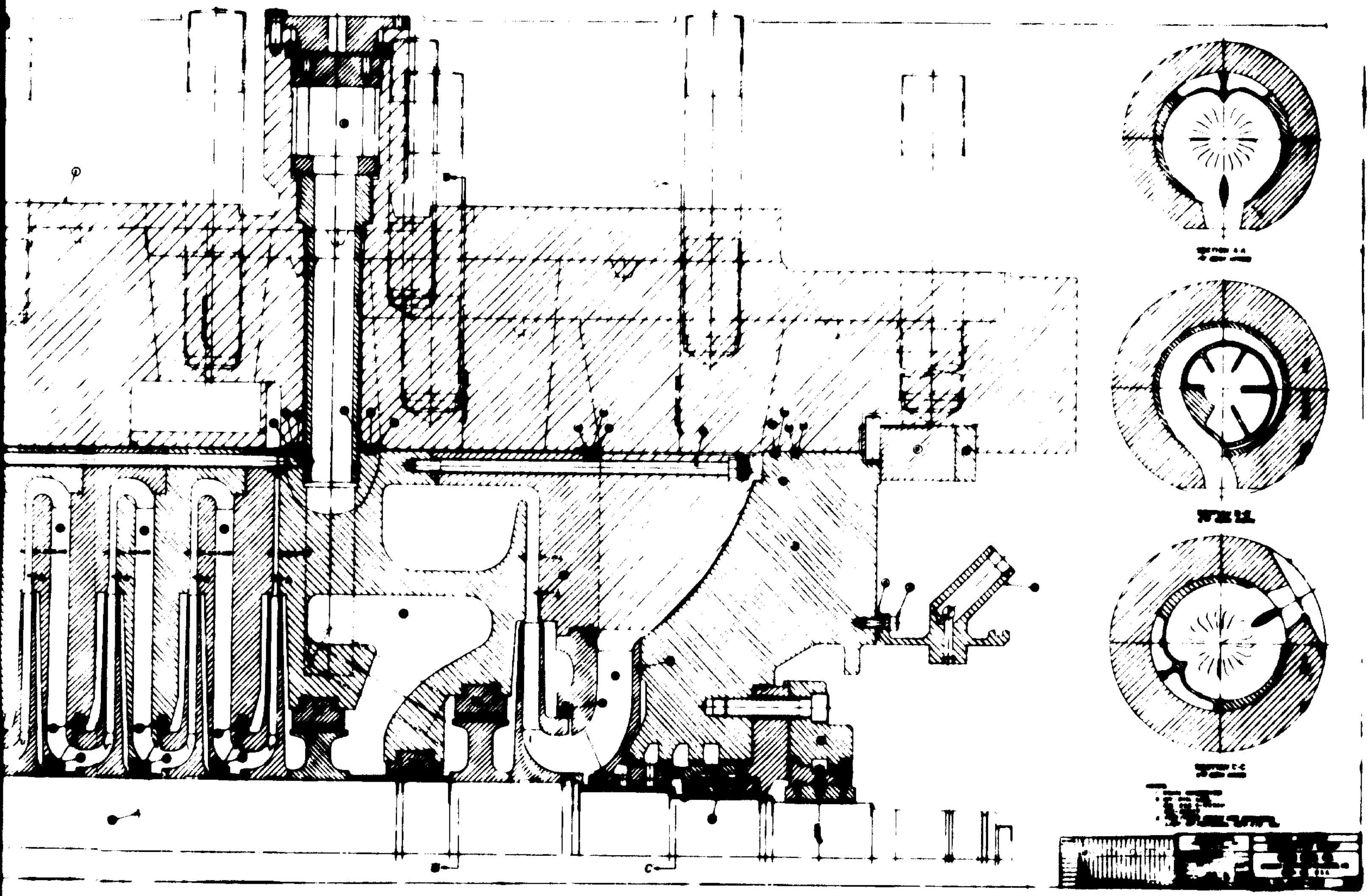
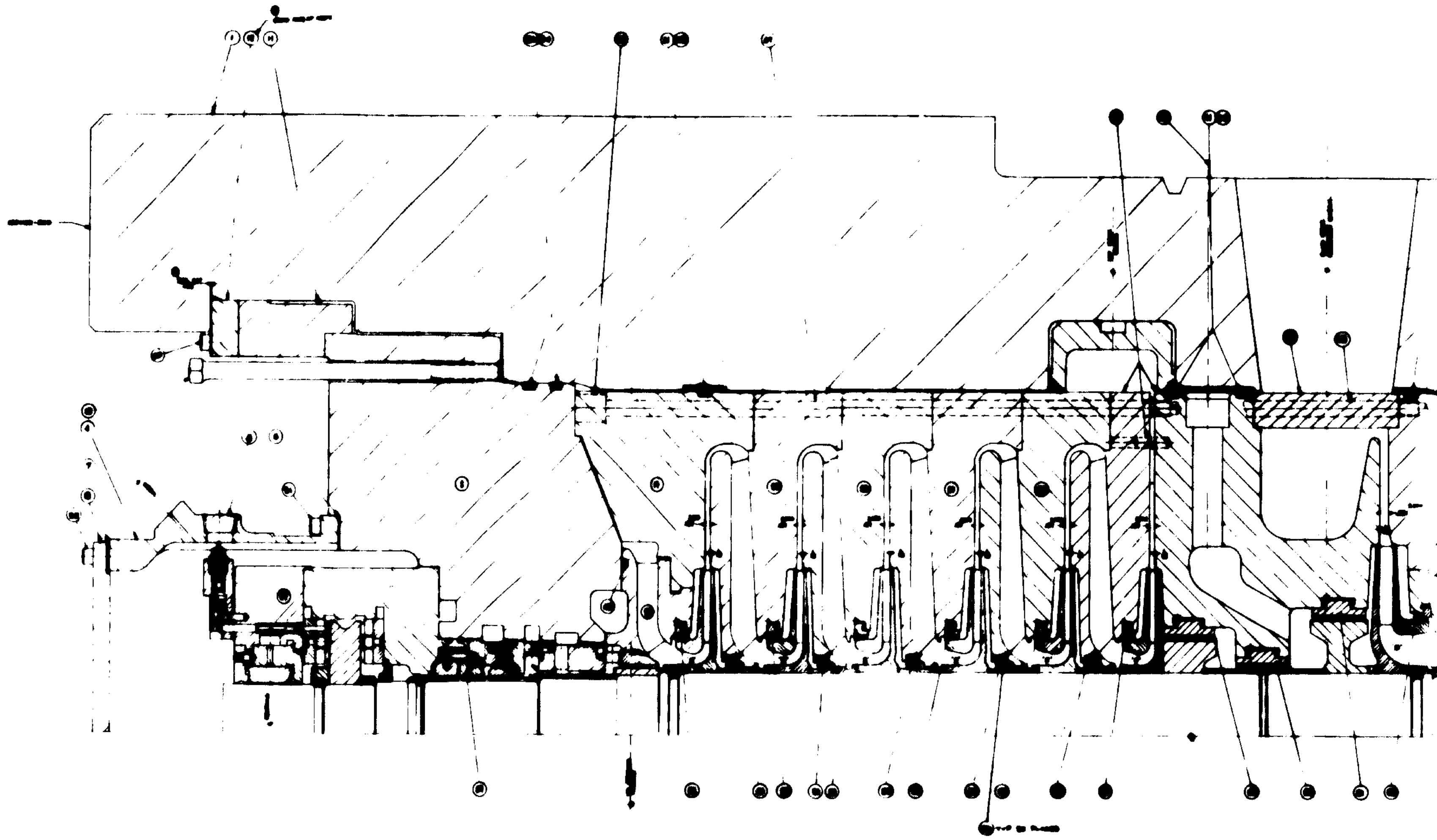


Figure 19 - 3738 Recycle Cross Section

SECTION 2



SECTION

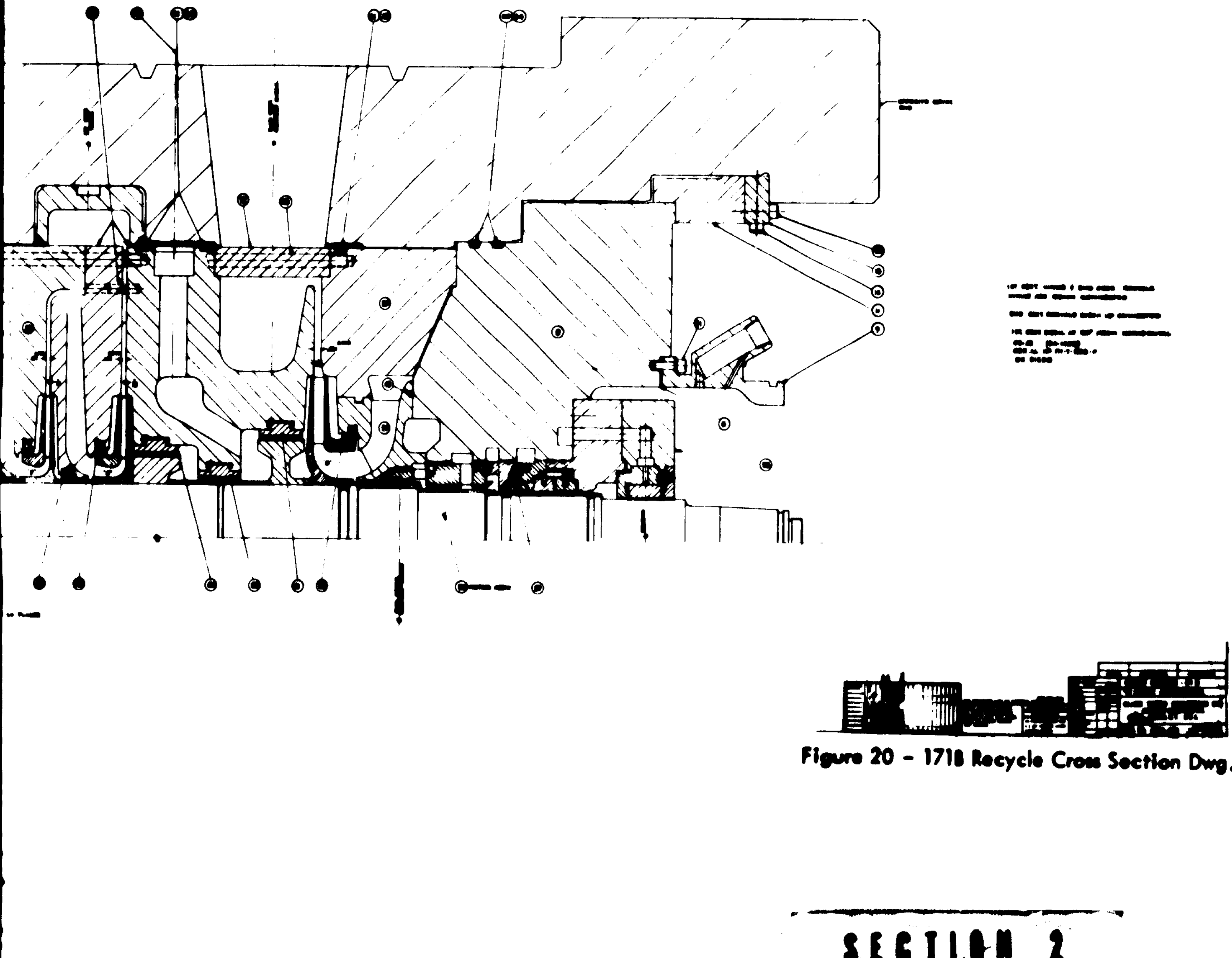


Figure 20 - 171B Recycle Cross Section Dwg.

SECTION 2

Figure 21 - 171B Recycle Unit

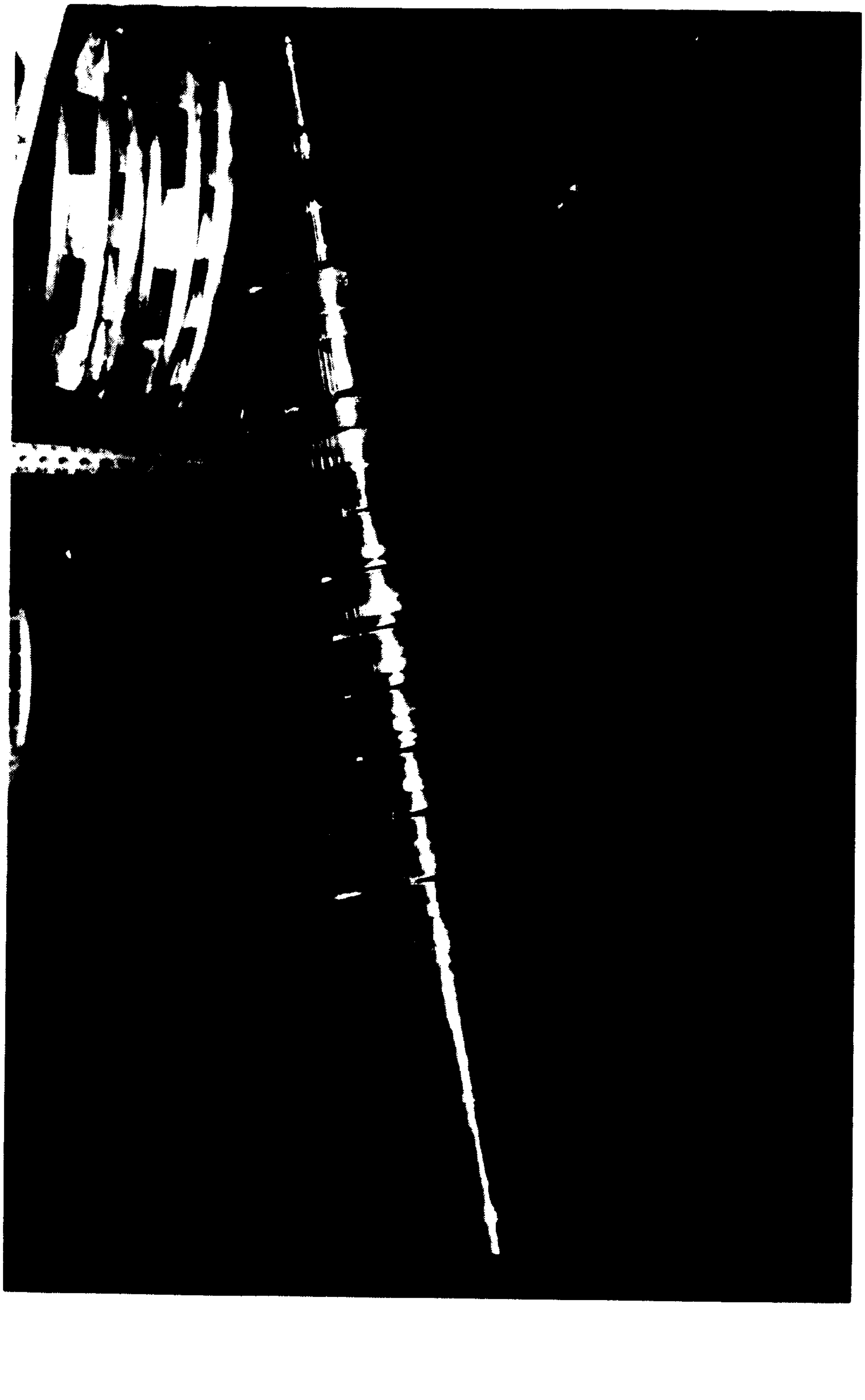


Figure 22 - 171B Rotor

Figure 23 - Photo Threaded Tie Rod

Figure 24 - Tie Rod in Eye of Impeller

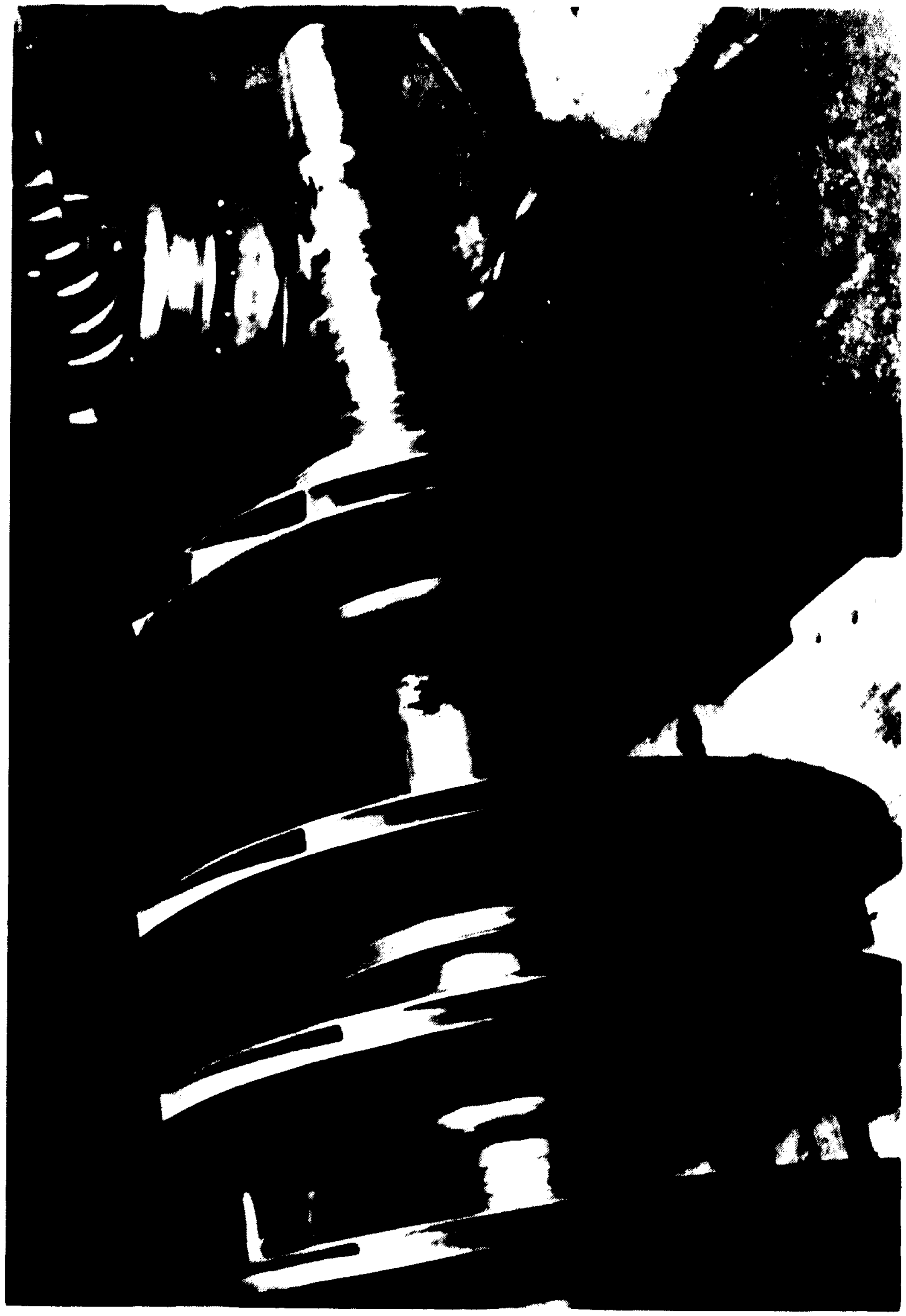


Figure 25 - Damaged Rotor

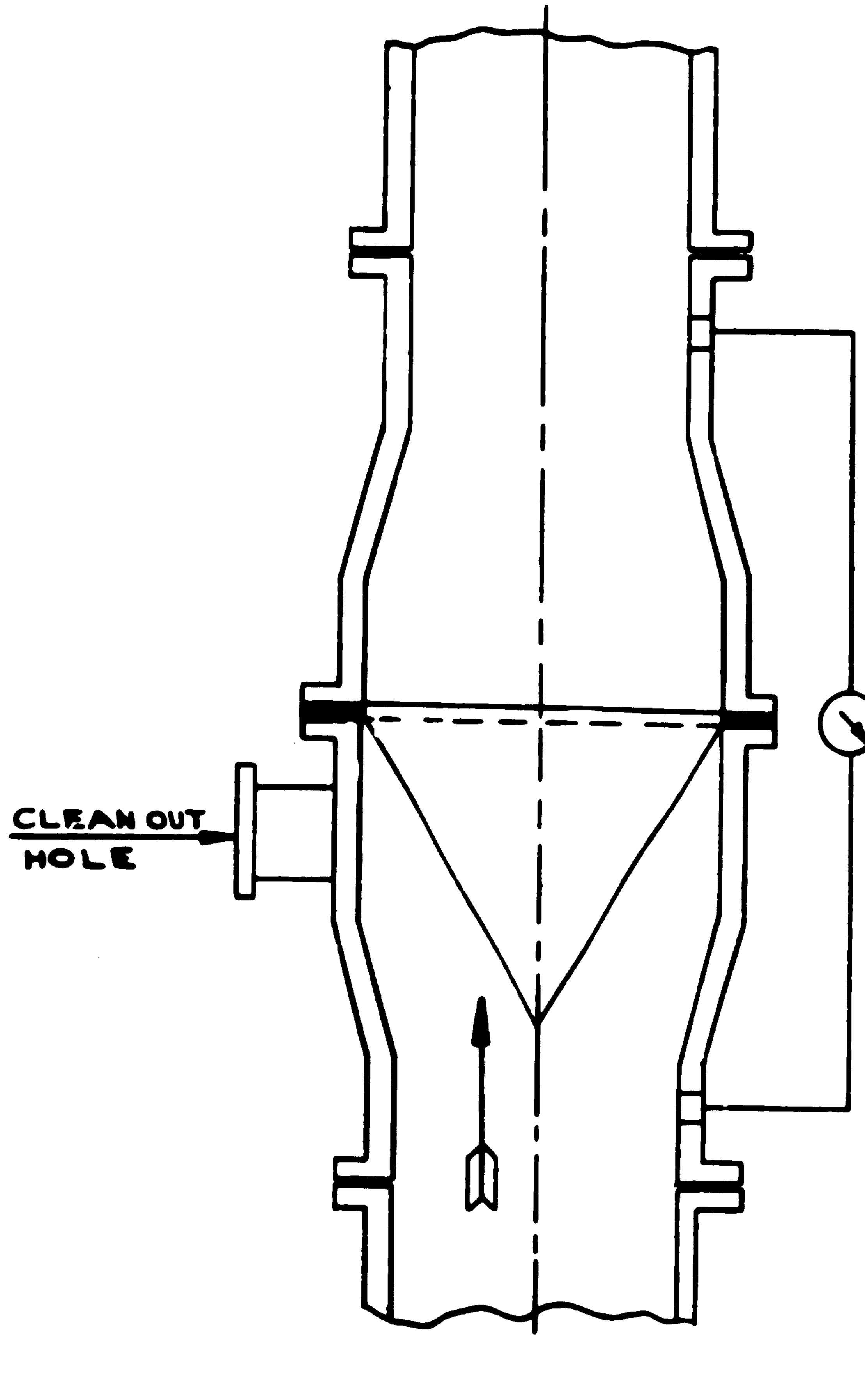


Figure 26 - Inlet Screen - Sketch

Figure 27 - Impeller with Cover Removed

Figure 28 - Damaged Recycle Impeller

Figure 29 - Photo Slug of Steel - Recycle Impeller Inlet

REFERENCES

Discussion of Mechanical Failures of Centrifugal Compressors as Experienced in High Pressure Installations.

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