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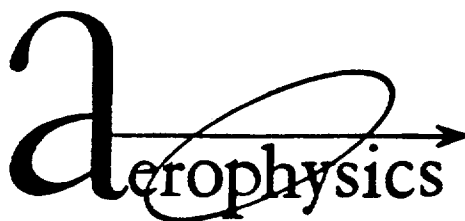
TECHNOLOGY REVIEW AND  
ASSESSMENT

*81*

OF THE ROTATING DIFFUSER (RD)

FAN FOR CURRENT SES APPLICATIONS

**PRELIMINARY**



**AEROPHYSICS COMPANY**  
**Washington, D.C.**

1982



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## SUMMARY

This document is the final technical report of the "Technical Review and Assessment of the Rotating Diffuser Fan for Current SES Applications." The report documents rotating diffuser fan technology for design use by the Surface Effect Ship program.

The study focuses on aerodynamic performance, manufacturing and structural fan considerations, as related to the SES system. Moreover, RD fan experience and performance, and the RD fan's place in the total lift system are considered. Figures and appendices provide background information considered useful to this documentation.

It should be noted that Aerophysics has an established reputation as primary lift fan designer for the SES program. The RD fan appears to be the ideal candidate design best-suited to the 1982 series of SES designs.

## INTRODUCTION

The Aerophysics company has developed Rotating Diffuser (RD) Fans for the lift systems for currently-proposed Surface Effect Ships (SES). Since 1962, these fans have been applied to transportation, industrial and marine services. Illustrations of such usage are found in the Appendices of this report. (See Appendices I, II and III).

The RD fan was first proposed for use in air cushion vehicles in 1962 studies which demonstrated the RD advantages over other lift fans. An RD fan was designed in 1970 as a backup for the 100-B SES lift fan system; this was not used due to time and resources constraints. However, this RD fan was considered the lift fan back-up candidate for the 3K SES program.

The search for a lift fan fulfilling basic needs of an ocean-going SES was conducted at the Naval Ship Research and Development Center (NSRDC). The fans studied and tested were of assorted sizes and configurations, produced by various manufacturers and of centrifugal, mixed-flow, jet flap and rotating diffuser types. A few were capable of satisfactory static performance at lower pressures and under limited conditions. None could meet the stringent demands of sea-going duty as well as the RD fans proposed by the Aerophysics Company. Such fans had been tested at sea as part of the fan-boiler propulsion system of the most important French maritime ship, the France. (See Appendix).

This RD fan, developed by Etablissements Neu (NEU Company) of Lille, France in the 1950's has been applied successfully to steel blast furnaces, sulfuric acid processing, pollution control and as mentioned above, marine propulsion. Neu has carefully documented its efficiency and durability in pressures up to 1,200 psf and diameters up to 12 feet.

Neu is the major manufacturer of RD industrial fans, while Aerophysics is the only firm applying RD fans to lift systems. As the sole U.S. licensee of Neu, Aerophysics has produced and code-tested model RD fans. It has developed and tested inlet guide vanes (IGV's), designed and dynamically tested models with IGV's, run static performance tests of full-scale fans, characterized industrial fans and in 1980-81, made fabrication designs and specimen construction of full-scale titanium RD fans with producibility studies for the 3K SES programs.

Thus, Aerophysics has determined that the basic lift fan system characteristics essential to the extended operation of an SES at sea exist in the RD fan.

Tests performed at the David Taylor Model Basin, and by Neu have determined a fan efficiency above 80 percent over a wide range of flows. Use of inlet guide vanes produces a constant exhaust plenum pressure despite continuous volume fluctuations caused by wave pumping, thus ensuring smooth movement. Extensive experience has proven the endurance of its design. Moreover, noise is generally 15 to 20 decibels lower than in other fans of similar performance.

This report seeks to analyze, correlate and document Aerophysics' efforts and provide a cohesive study of RD fan technology related to SES application.



## 2. THE RD FAN AND ITS PLACE IN SES SYSTEM TECHNOLOGY

### 2.1 General Statement of the Aerodynamic and Structural Features of the RD System

The Aerophysics Company became interested in the application of rotating diffuser fans for air cushion or captured air bubble vehicles in 1961. Such fans have been marketed internationally for industrial uses since the early 1950's by the Neu Company of Lille, France.

The RD fans have three major advantages in this application: high aerodynamic efficiency; a static pressure capacity curve with a negative slope all the way to shut off and a very rugged inherent design.

Studies by Aerophysics under the sponsorship of the U.S. Army and U.S. Navy from 1963 to 1967 and later studies have documented these advantages. A program completed for the Army in 1964 covered aerodynamic, structural fabrication and design aspects. In 1966, a prototype RD fan was strain-gage tested for the Army. In 1970, Aerophysics designed a backup fan for the SES 100B lift fan and in 1980, a full-scale lift fan for the Navy/Rohr 3K Surface Effect Ship.

As a result of work performed on RD fans by Neu and Aerophysics, this fan is a fully developed unit whose performance, strength and weight can be accurately predicted, and whose aerodynamic characteristics have been carefully studied.

In brief, centrifugal fans with rotating diffuser impellers have two important aerodynamic advantages:

- high efficiency over a wide operating zone and
- a very stable characteristic curve.



The principal reason for these advantages is that most of the conversions of the velocity pressure (kinetic energy) at the exit from the impeller blades to static pressure (potential energy) takes place under controlled expansion in the rotating diffuser. Furthermore, the uneven velocity profile at the outlet from the blades is considerably evened out by the diffuser.

In addition to these main advantages, the RD-type impeller has additional benefits for SES application:

- very flat volume/pressure curve enabling one fan to give 85 percent of the output of two identical fans operating in parallel.
- very good stability at low flow rates
- possibility for increasing the capacity in situ without changing the impellers
- very steady aerodynamic operation allowing two fans to run parallel without any surge problems
- flat radial blades which give maximum self-cleaning and abrasion resistance.

The RD fan is of the family of shrouded centrifugal fan wheels. It is aerodynamically and structurally unique for the following reasons: The air first enters the inlet eye under the action of the inducer blading; it is turned in the blade passage areas where energy is added to it, then diffused between the RD diffuser rings. (See Figure 2.1.1).

The key feature is the rotating fan exit. Its main purpose is to expand the exhaust gases, thus recovering their kinetic energy in the form of additional pressure rise.

In addition to the pressure recovery of the system, many other useful effects are noted. One is noise reduction. The decrease in delivery air velocity which occurs at the blade tips causes a corresponding reduc-

tion in noise, especially at the blade passage frequencies. The volute requires less expansion or spiral development than conventional systems. Because much of the flow is diffused in the wheel rather than in the volute, the noise producing volute cut-off can be opened up considerably. (See Figure 2.1.2).

The RD concept reduces the discharge of air velocities at the edge of the diffuser. With much of the diffusion confined to the wheel and with the exhaust velocities reduced, the means of air collection, in this case the volute, can be comparatively compact. Single, double or multiple delivery volutes are very effective.

In SES applications, the RD fan can operate directly without volute discharging into a free plenum area. (See Chapter 6.2 on page ).

The RD fan can be arranged as either a single inlet single width wheel (SISW, see Figure 2.1.3) or as a double inlet double width wheel (DIDW). The capacity can be doubled by mounting two single systems back to back to create the DIDW (See Figure 2.1.4).

Complete pressure and flow control can be obtained by the use of wheel, volute and inlet guide vane (IGV) geometries. Their dimensions may be adjusted to fit any performance, structural, sizing and mechanical requirements of the lift fan system by using information derived from Aerophysics tests.

As stated, RD fan technology has been fully proven and developed over 30 years of invention, testing, application and service. As a structure, the RD wheel is extremely robust. It is typically used under extreme conditions of heat, corrosion and performance in heavy industry. (See Appendix I).

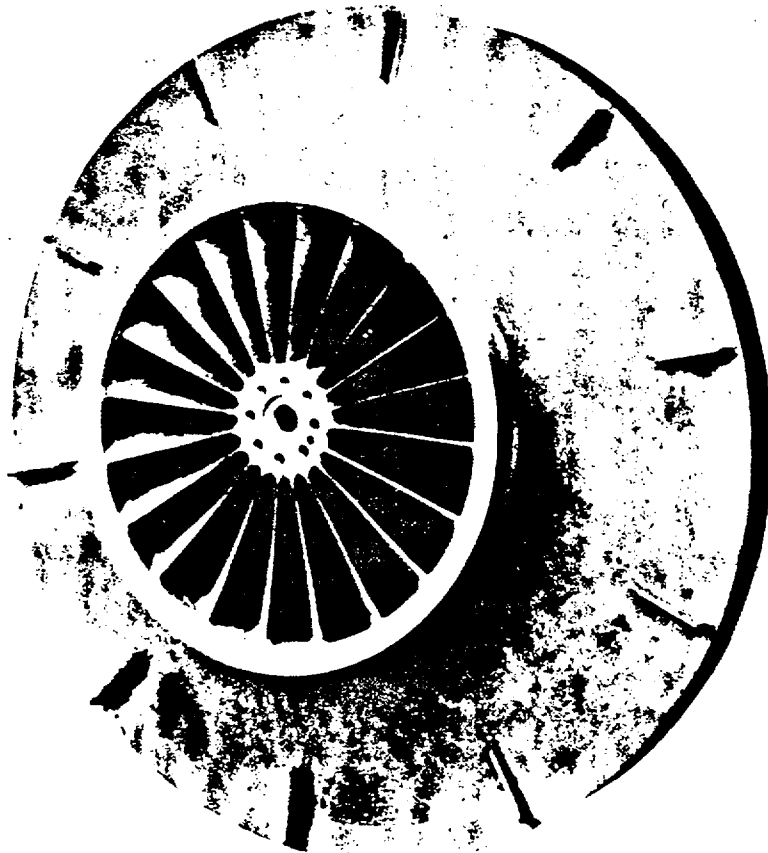


Fig. 2.1.1.

*Rotating diffuser impeller showing the inlet eye and blade leading edges. The sides of the impeller are fitted with patented anti-surge vanes.*

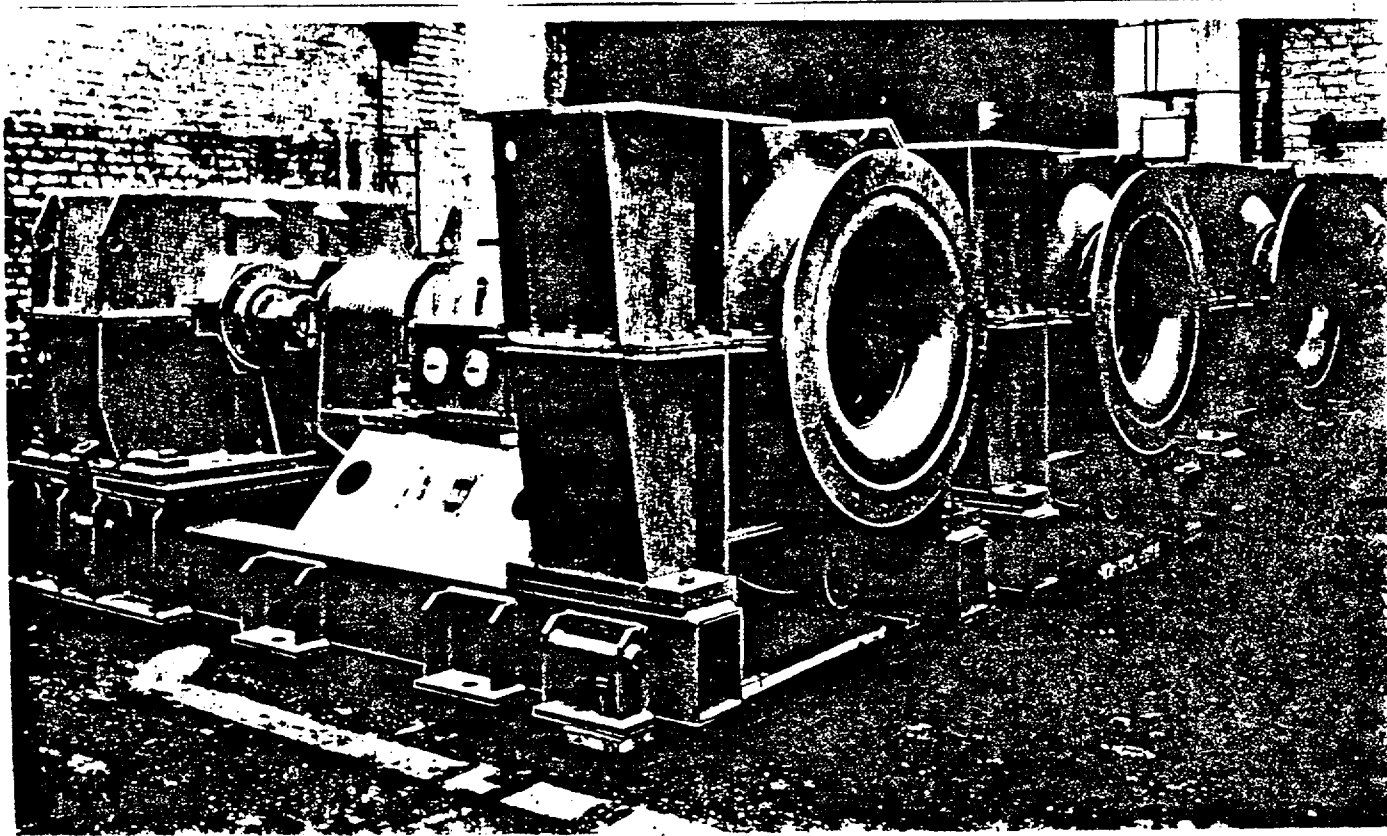
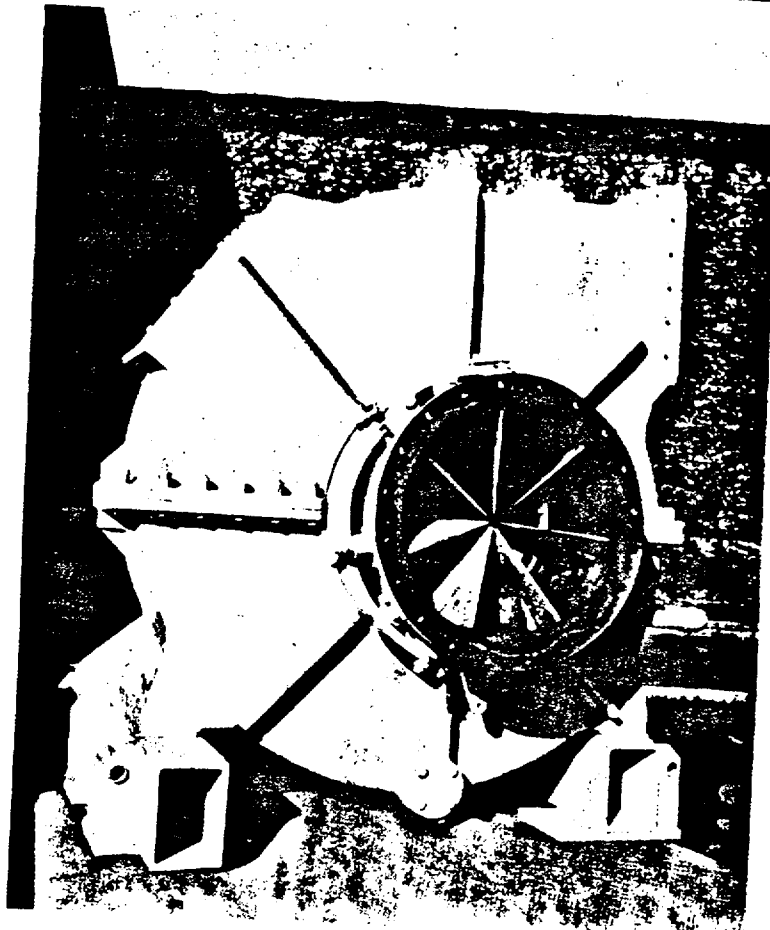


Fig. 2.1.2.

*Single stage air blower No. 70 series 0.55/1.09  
Speed 5,500 rev/min  
Flow rate 36000 m<sup>3</sup>/h (22800 cfm)  
Pressure 4600 mm H<sub>2</sub>O (180 in W.G.).*

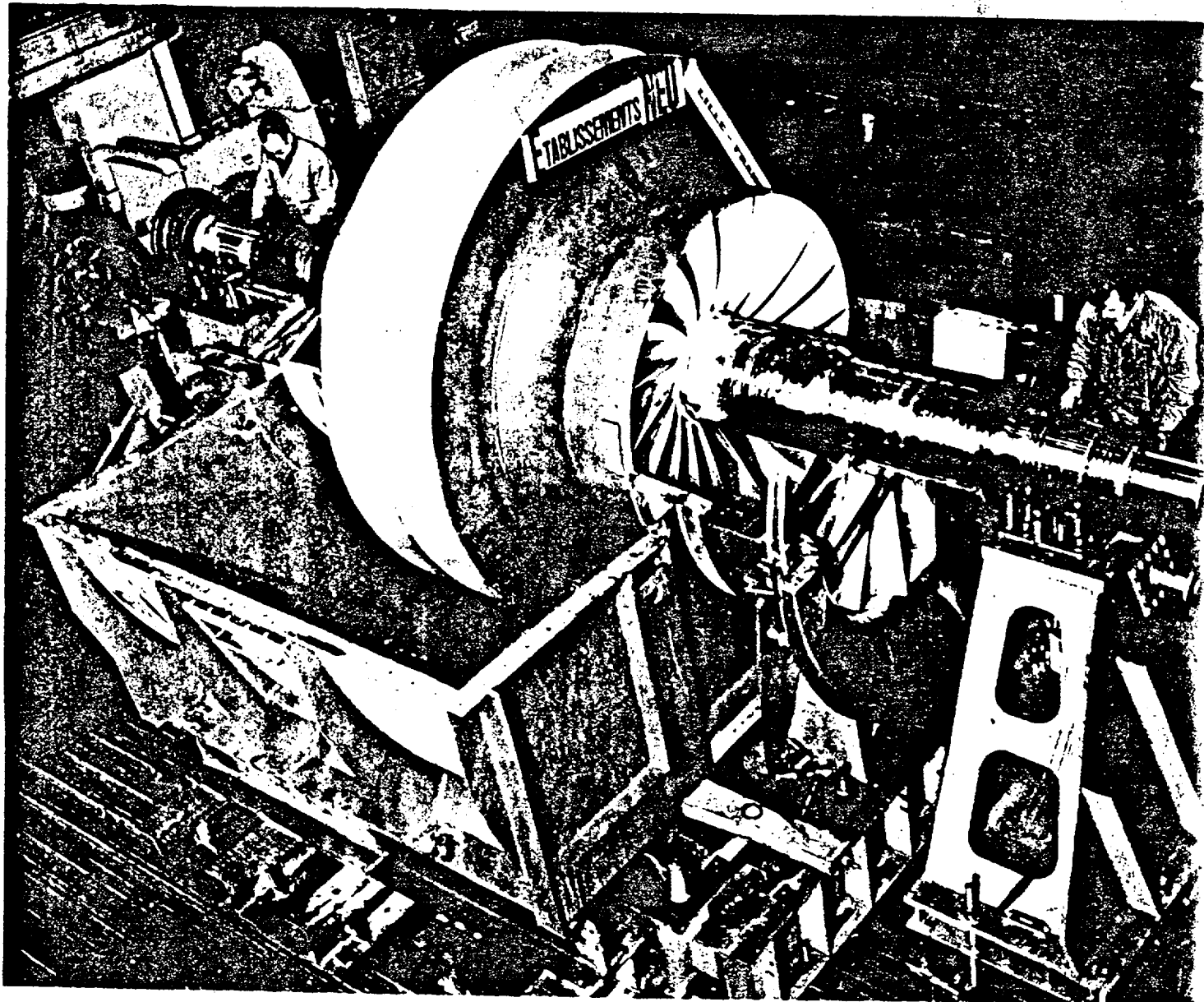


*Blower for air - SO<sub>4</sub> + SO<sub>2</sub> destined  
for a sulphuric acid plant  
Speed 3000 rev/min  
The casing is in spheroidal graphite iron  
type "Ni Resist"  
The inlet guide vanes are sealed to prevent leakage.*



*Air blower No. 220 series 0.46/1.09  
Speed 1 500 rev/min  
The casing is of welded construction  
This photo shows the blower mounted on our  
test bench.*

**Fig. 2.1.3**



*N° 240 - 0.7/1.3 Rotary diffuser forced draught fan for a 600 MW section of an Electricity generating station at Porcheville. The fan is shown during assembly prior to testing at the E.D.F. test bed, Saint-Denis.*

Fig. 2.1.4.

The structural elements of the RD wheel were shown in Figure 2.1.1. The RD impeller is a shrouded rotor with its blades acting as full depth spokes radiating from the hub to the shroud surface. For a given weight, an RD wheel will be stronger than other wheel systems because of its spoke-like radial blades connecting shroud, back plate and hub into a unified structure.

The leading edge of the inducer is radially developed from the hub to the shroud. The highly rigid structure of the RD fan is very solid. Anti-surge vanes permit stable pressure down to conditions of near zero flow.

## 2.2 Application of RD SES Lift Systems and the Comparative Limitations of Other Types

The RD wheel with its development and service experience is an excellent choice for the SES lift system. It is particularly unique in that it has useful cushion pressures ranging from 250 to 850 psf total. In this range, it operates in an area of total efficiency in excess of 80 percent, with tip speeds of less than 700 ft. per second and a stable flow-pressure down to shutoff. It can be made at a weight and in materials required for shipboard service. Industrial-type (Type L) RD units are available which deliver 1,100 psf at a tip speed of 790 ft. per second with only one stage.

The outstanding characteristic of the RD is its ability to deliver the higher cushion pressure at lower tip speeds. The lower tip speeds ensure a superior or stronger wheel structure with lower stress levels.

In Figures 2.2.1 through 2.2.3, the pressure and flow regimes at various horsepower levels of several competing systems are shown. In contrast, Figure 2.2.4 presents the performance boundaries of various Neu RD fan systems collected into a single performance regime. The boundaries are usually determined by tip speed and structural considerations.

In making a comparative examination of the RD as compared to other types of wheel and impeller systems, we have noted the following: The ACV and SES fans in Figure 2.2.1 produced pressure rises which barely reach  $150 \text{ lb/ft}^2$ . Such low cushion pressures have been adequate for the small-scale cushion craft built to date.

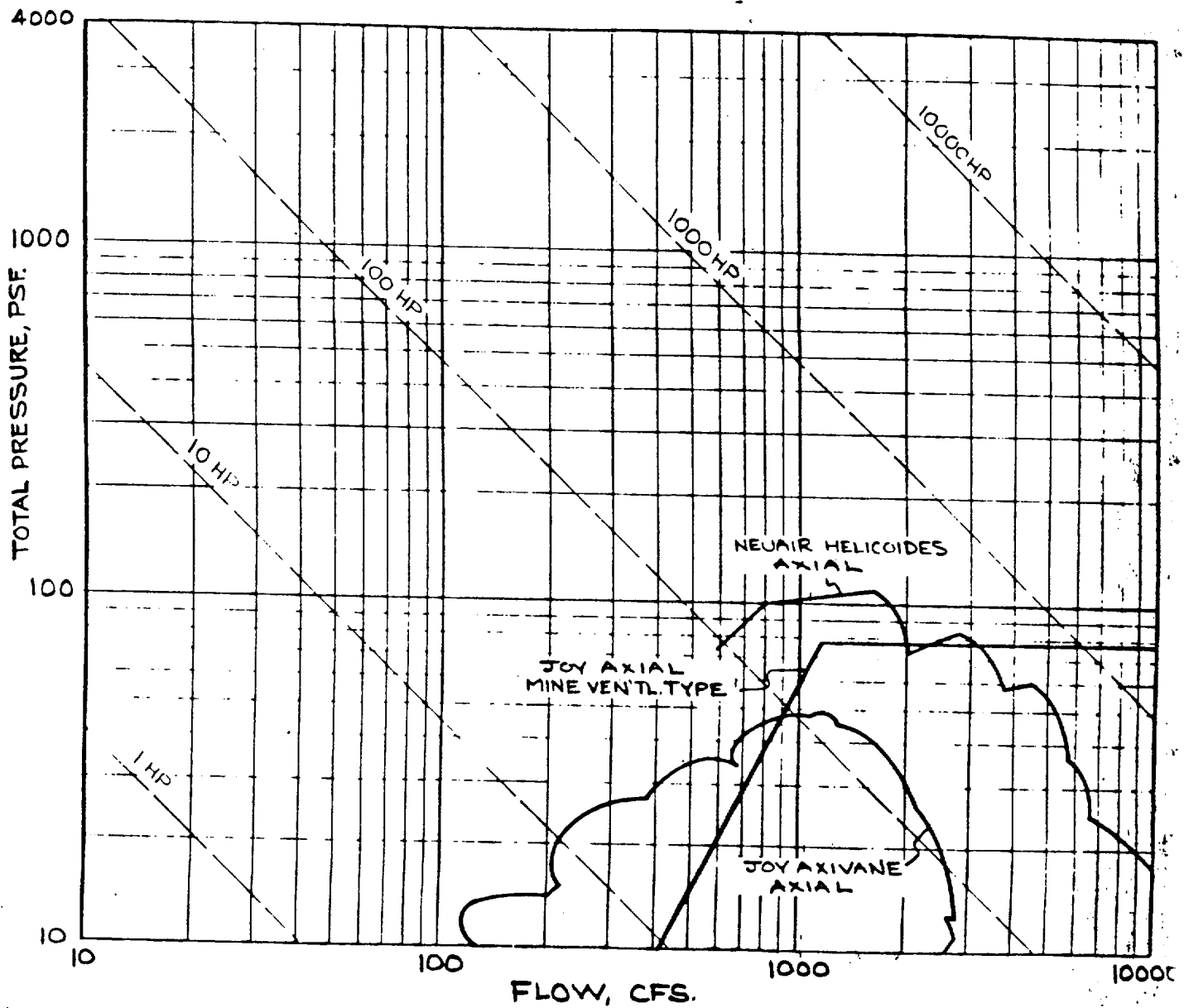
However, much higher cushion pressures become necessary, for example, in a  $325 \text{ lb./sq.ft.}$  1,500 ton surface effect ship. The aforementioned centrifugal fans would have to be staged in order to produce an adequate pressure rise. This would reduce overall system efficiency, increase weight and increase space requirements. In addition, certain fans such as the JEFF(A) have cracked under high stress.

Common axial and centrifugal fans are not an alternative, as shown in Figures 2.2.2 and 2.2.3. Their performance is in the same range and suffers from the same constraints as the ACV and SES fans.

Figure 2.2.4 shows that only the RD fan can produce the pressures necessary for a large-scale SES in a single stage. Moreover, the same basic design maintaining low tip speeds and high endurance, can be used for a wide range of vehicle sizes, including the largest proposed SES.

Another problem common to lift fan systems in SES use is their inability to provide adequate ride control. These fans, when subjected to the sinusoidally ranging backpressures of wave pumping cannot compensate in time. The resulting variable plenum pressure develops ~~poor~~ ship ride.

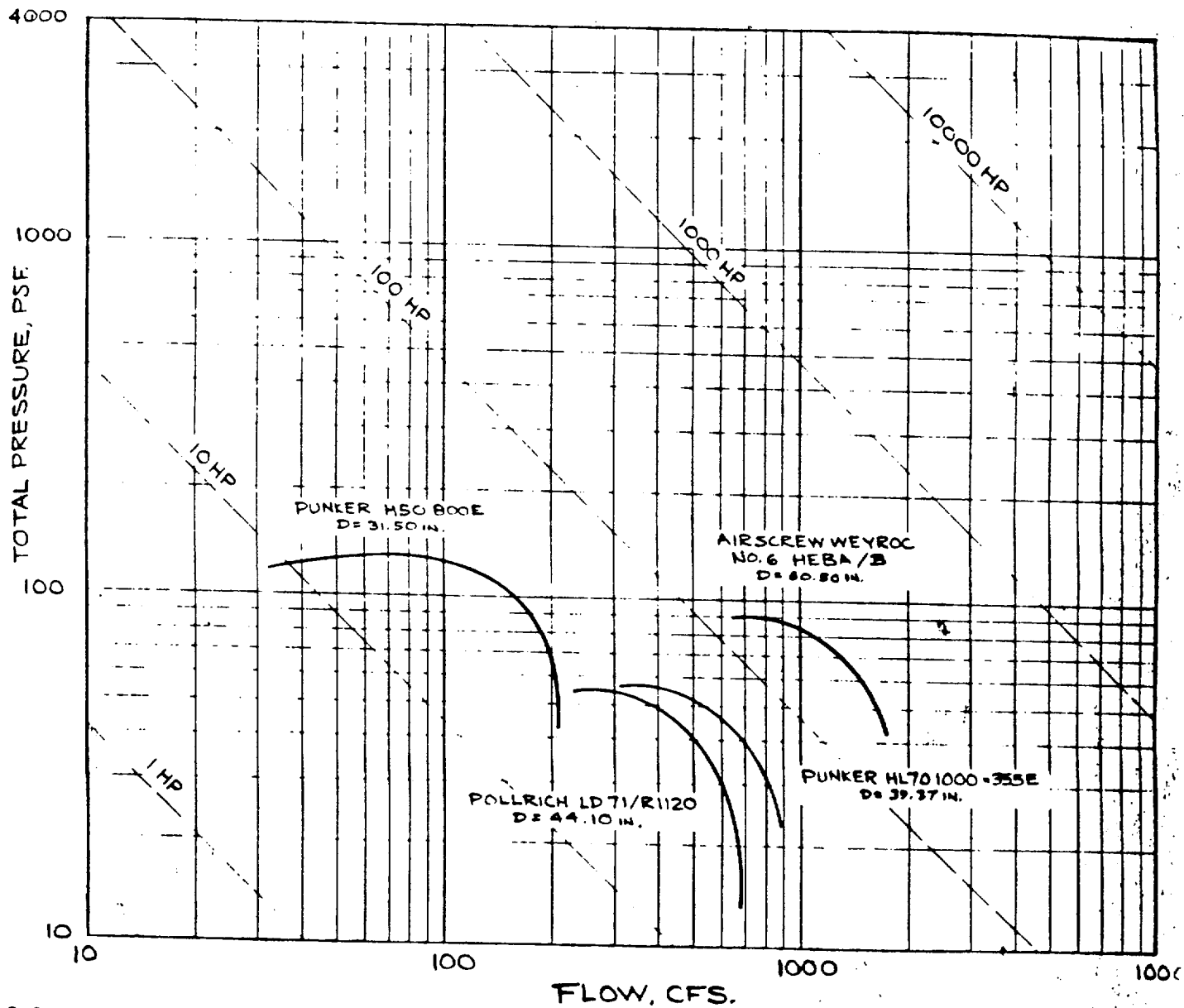
As will be demonstrated in section 6.1 of this report, the RD fan combined with a set of inlet guide vanes can virtually eliminate the adverse plenum effects of wave pumping. For surface effect ships designed to travel long distances on open sea, this is a very significant advantage.



REPRESENTATIVE PERFORMANCE SINGLE STAGE INDUSTRIAL TYPE AXIAL BLOWERS.

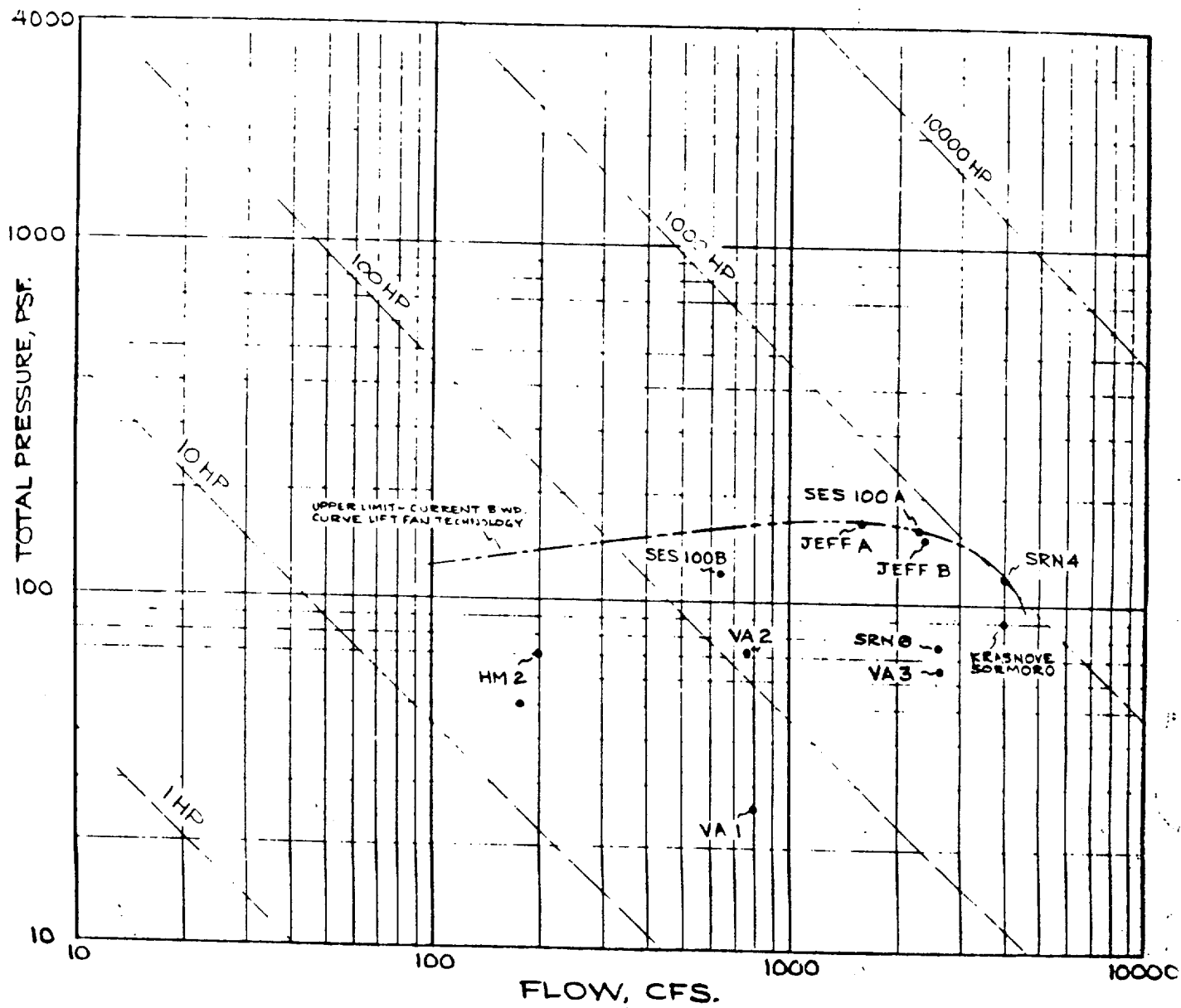
FIG. 2.2.1





CURRENT EUROPEAN HIGH EFFICIENCY BACKWARD CURVE SWSI WHEELS FOR ACV ADAPTATION  
COMMERCIAL-INDUSTRIAL TYPES

FIG. 2.2.2.



RECENT ACV/SES LIFT FANS. BACKWARD CURVE CENTRIFUGAL WHEELS

FIG. 2.2.3.

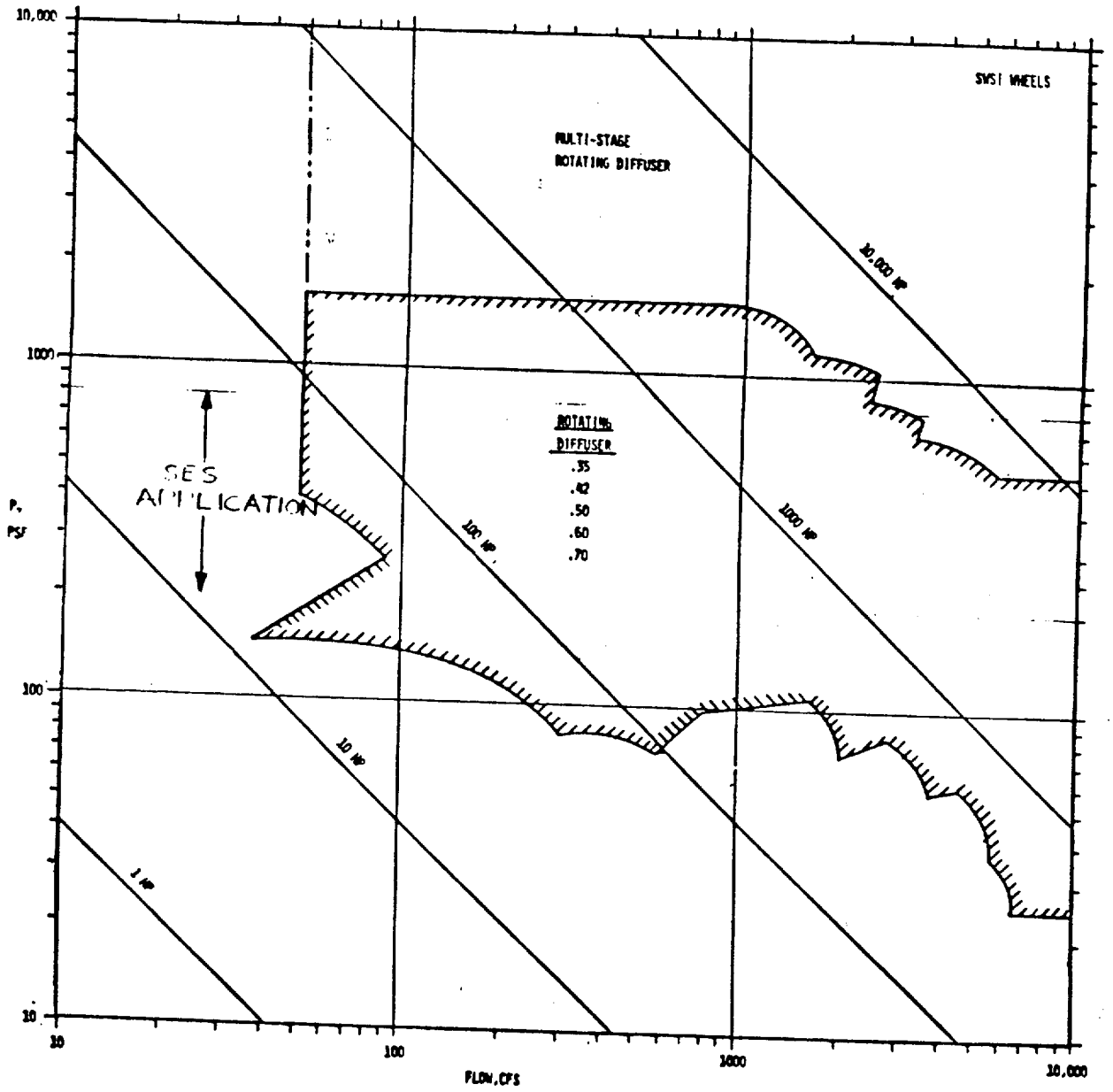


Fig. 2.2.4.

### 3. EXPERIENCE WITH THE RD FAN

#### Industrial Background and Service Applications

Since its development thirty years ago, the RD fan has been applied to a wide variety of tasks throughout the world. Its high performance, durability and adaptability to many functions make the RD invaluable to certain tasks which no other fan can perform.

RD fans are used in the iron and steel industry as blowers for blast furnaces and cooler towers, as well as gas compressors. They are used in natural gas production as air blowers and gas compressors. In the chemical industry, they are used as blowers for air and air-SO<sub>2</sub> mixtures. RD fans are employed as air blowers for other processes, including pollution control, water purification, food manufacture and electrical power generation.

Since the RD fan is designed to operate under such adverse conditions, the quality of workmanship in the fan assures rugged and long-range service. In addition, an on-going research program has produced improvements in efficiency, pressure rise and capacity. (See Appendix I).

Most RD fans have been installed at overseas sites. It is only since 1974 that a joint Aerophysics - Neu group, called Neuhair, has been selling the RD fan in the United States and Canada. A list of some local industrial examples is presented in Appendix IV along with material on previous SES applications, tests and studies. (See Appendices V through VIII).

#### 4. RD LIFT FAN PERFORMANCE

##### 4.1 General Aerodynamic Behavior of the RD Fan

Most single stage fan systems impart the greater part of their mechanical energy to air in the form of high velocity flow, at low pressure rise. While this may fulfill the volumetric requirements of an SES lift fan system, it does not satisfy the high pressure needs of large, heavy ships.

The RD fan diffuses the high speed air being expelled at its outer radius into lower speed, higher pressure flow. It accomplishes this without the inefficient losses incurred by the volute-diffusion configuration of other systems. The RD is a "mixed flow" fan in which a significant portion of the forced air is diffused within the rotating wheel structure. Air is forced into the fan by the axial-flow "impeller" portion of the blades, located at the fan inlet. The flow is then directed along the blades until it becomes fully radial at the blade tips. The higher velocity air at the tips is diffused through the radial portion of the shroud beyond the blades to regain kinetic energy in the form of pressure rise.

In a typical shrouded centrifugal fan, with fixed diffuser, flow separation occurs along blades and shroud walls. A boundary layer of slow moving air is formed along the walls, increasing as it progresses downstream. This constricts the flow between the surfaces, thus increasing its velocity. The difference between the velocity at the walls and in the center causes flow disturbances and the ensuing losses of efficiency.

In an RD fan, flow separation also occurs along the blades and shroud walls. However, the flow disturbances are absorbed in the rotating diffuser section of the fan. The boundary layer formation along the

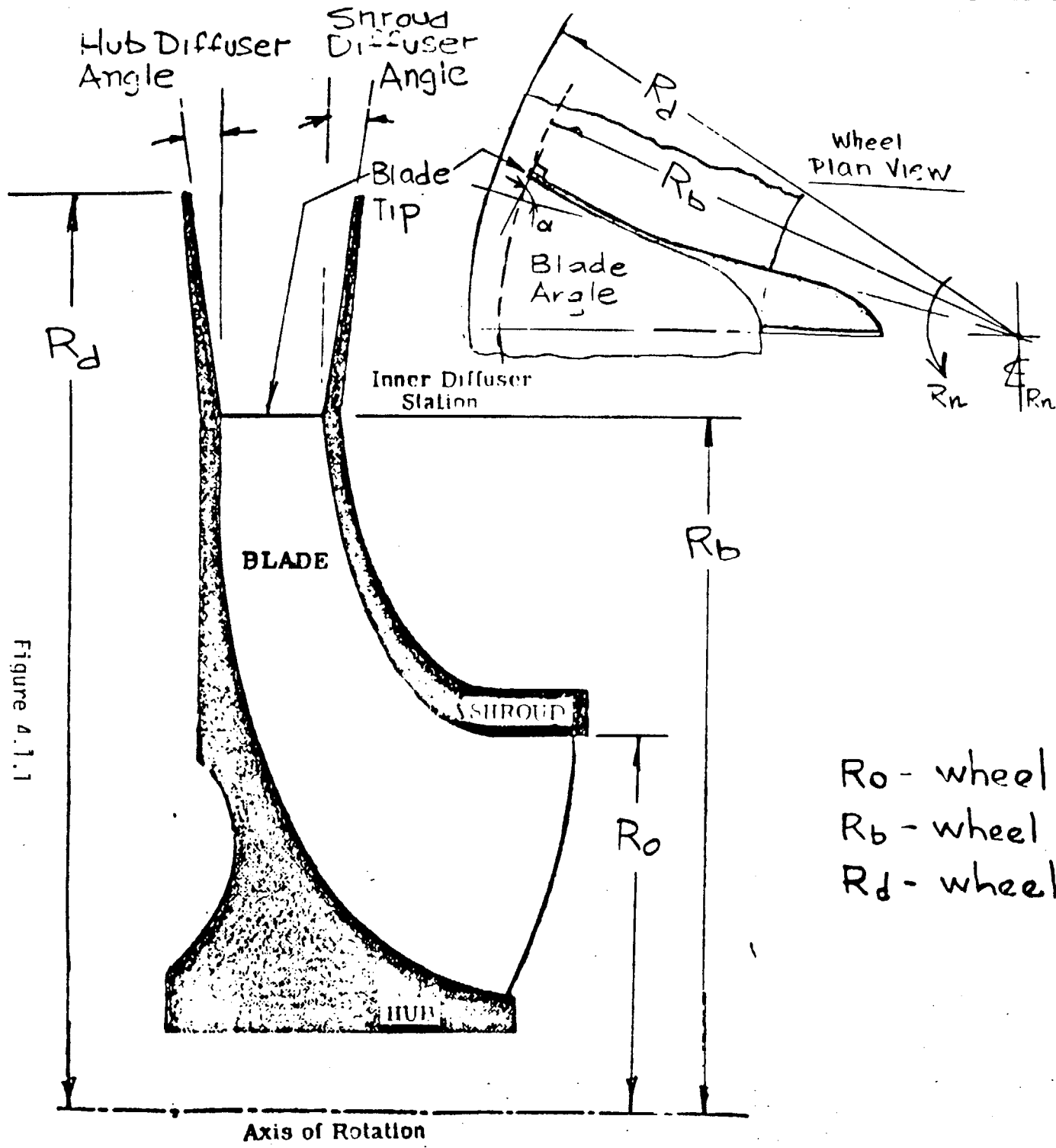
shroud and hub is reduced, thus increasing the effective flow area. The flow rate is steadier, with less difference in velocity between the center and the walls. Thus, turbulence is greatly reduced and efficiency increased.

In order to mate the fan with the rest of the lift system, a special duct leads to the inlet and a volute from the exhausts. The inlet duct may contain the inlet guide vanes, which are used for varying either the pressure rise or the flow of the fan, depending on which is needed. While a good deal of air diffusion occurs within the RD, the final expansion is completed in the volute. The volute also directs the flow to its final destination in the plenum area of the ship.

Due to complex flow patterns of air through the RD fan, the performance cannot be predicted by any theoretical means. It must be determined experimentally. It does vary, however, within the various fan parameters shown in Figure 4.1.1.

The NEU Company measured the performance of fans at various dimensional parameters and created charts for predicting the performance of any possible configuration. Figure 4.1.2 is a typical performance chart displaying the values of the non-dimensional coefficients of flow ( $C_d$ ), pressure ( $C_{pt}$ ) and reduced orifice ( $O$ ) for any fan with a blade angle as large as  $90^\circ$  and a diffuser ratio of 1.3.

From charts like this and others gauging different parameters, it is possible to determine the most efficient fan for a desired flow rate and pressure range. An analysis of the influence of parameter dimensions follows.



FAN WHEEL  
PARAMETERS

PRIMARY:

- Blade diameter  $2R_b$
- Hub-tip ratio  $R_o/R_b$
- Diffuser ratio  $R_d/R_b$
- Blade angle  $\alpha$

Sample designations:

307 - 0.55 / 1.3

↑ Blade diameter (centimeters)  
↑ Hub-tip ratio  
↑ Diffuser ratio

SECONDARY:

- Diffuser angles
- Number of blades

$R_o$  - wheel inlet eye radius  
 $R_b$  - wheel blade outer radius  
 $R_d$  - wheel outer radius

Figure 4.1.1

12

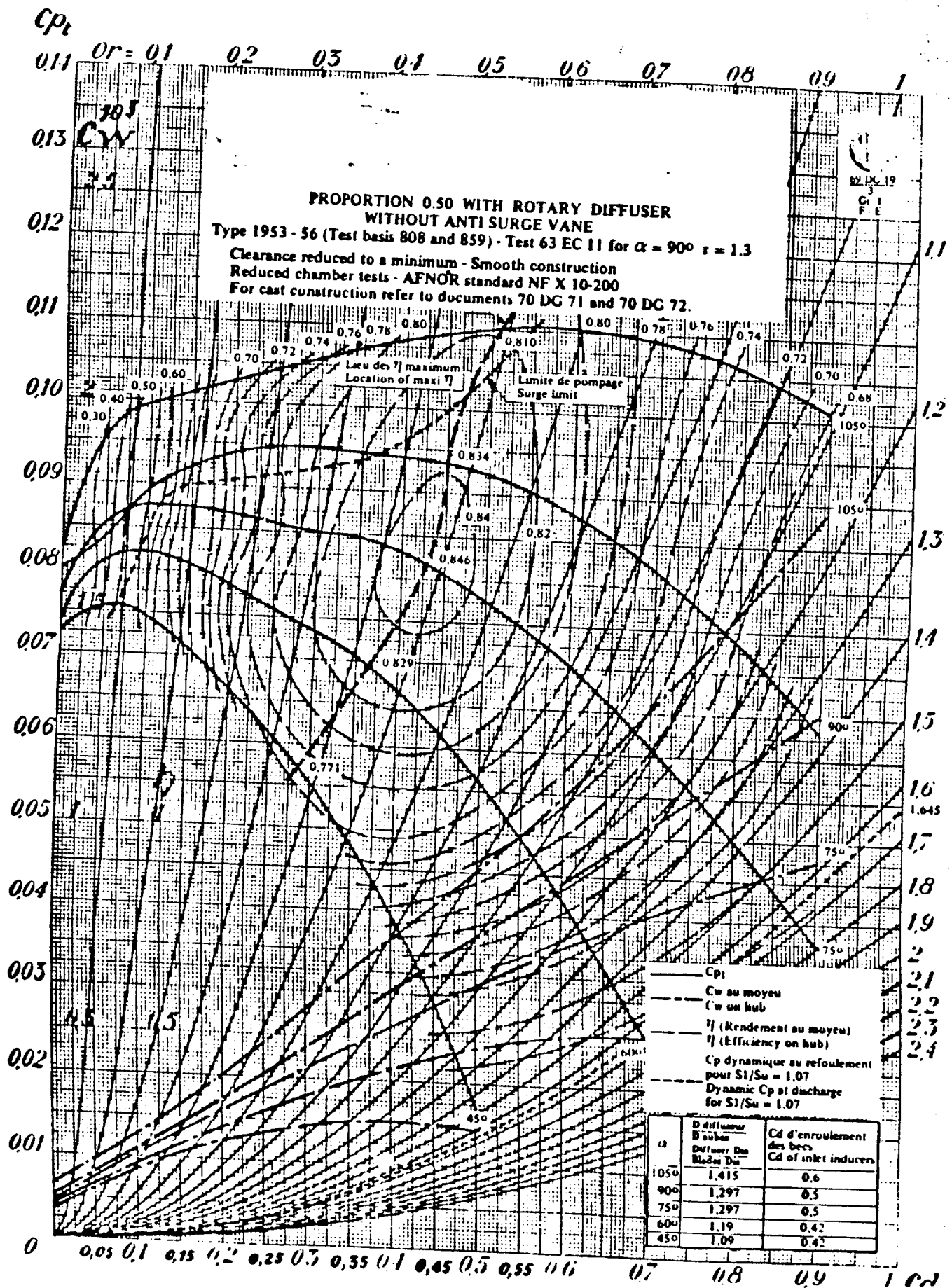


Figure 4.1.2



## 4.2 Effect of RD Wheel Geometry on Pressure/Flow and Efficiency

An RD fan is designated by a model number that characterizes its significant geometric parameters. Specifically, an example is Model Number 200-0.65-1.30-67.5°.

The first number is the blade tip diameter of the wheel in centimeters. The second is the hub to tip ratio, or the ratio of the radius of the inlet eye to the blade tip radius. The third is the rotating diffuser radius ratio, i. e., the overall diameter of the wheel to the blade tip diameter. The fourth is the blade angle at the tip. (See Figure 4.1.1.)

For SES applications, a hub tip ratio in the range .55 to .65 has been found to be preferable. This parameter controls the air handling capacity of the flow capability of a fan of given diameter. This is, of course, because the inlet eye area is the controlled area for the flow through the fan. (See Figure 4.1.1.)

A hub to tip ratio (also commonly referred to as the RD fan "proportion") of the order of 0.65 has been found most desirable for SES applications or similar uses calling for operation over a wide range of pressure and/or flow.

As has been noted, SES applications call for operation over a wide range of pressures and flows with a performance curve that remains stable to shut-off. It has been found that the latter can be achieved consistently only with blade angles less than 90 degrees. The optimum from this point of view is probably a blade angle of 67.5 degrees.

Unfortunately, blades with this much divergence from a purely radial orientation are subject to quite high bending moments. A compromise between performance and blade strength suggests the

RD FAN 200-0.65-61.5-1.30

RPM: 1442

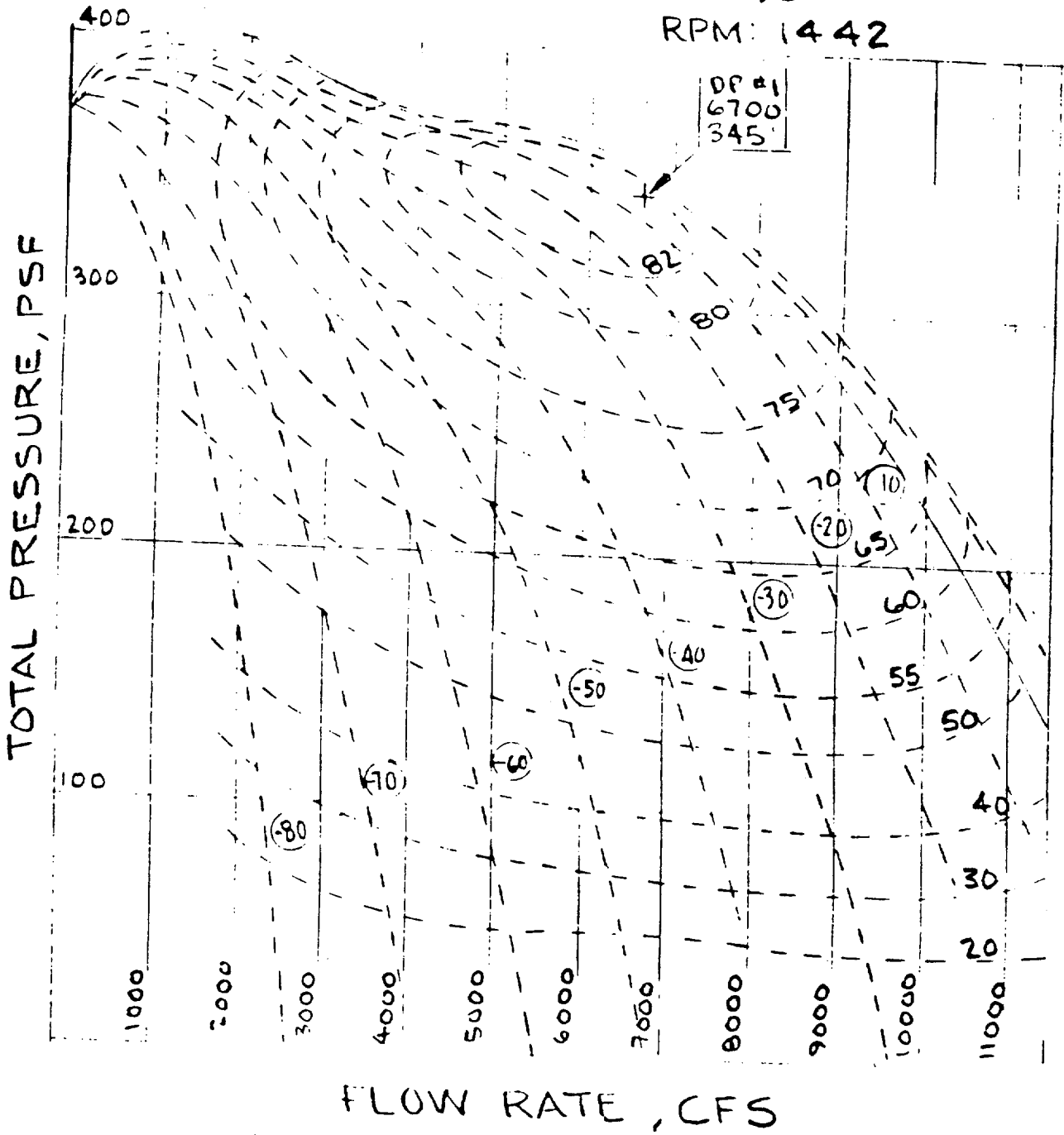


Figure: 4.2.1.

RD FAN 200-0.65-61.5-1.30

RPM: 1746

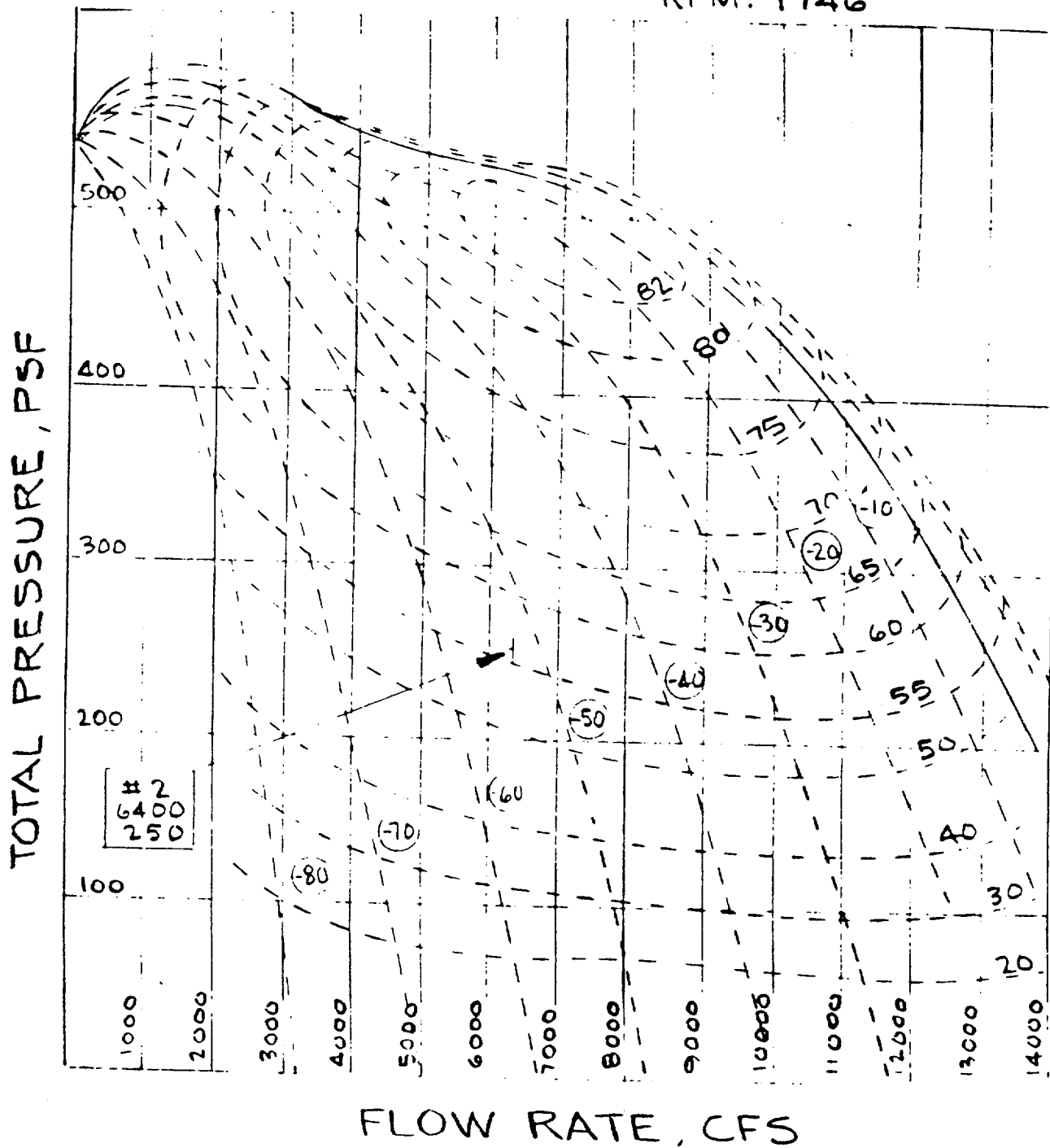


Fig. 4.2.2

use of a blade angle of 70 degrees. A diffuser ratio of 1.3 is preferred from the performance point of view (both peak pressure and efficiency). At a ratio of 1.2, the stresses are lower, without significant sacrifice of pressure and efficiency.

Normally for a fan of a given model number, the only control is the rotational speed. For SES applications in particular, this type of control is inadequate. Therefore, inlet guide vanes of some variety are necessary to obtain the flexibility to operate at the numerous combinations of pressure/flow dictated by the variations of both ship weight and speed. (See Figure 4.2.1 and 4.2.2.)

In the first case, one operating point, 6700 cfs and 345 psf, is shown on the normal fan operating curve, i. e., guide vanes wide open. In this case, the fan is being operated at a relatively low speed, but the desired output is being obtained at near peak efficiency. Figure 4.2.2 shows the same fan operating at about 20 percent higher speed. The operating point indicated in this case is well below peak pressure and efficiency. To reach this lower pressure, the guide vanes have been closed more than 50 degrees. This is a rather extreme case that does, however, show the effectiveness of the inlet guide vanes (IGVs) as controls in static performance.

#### 4.3 Effect of the Inlet Guide Vane System on Ride Control

Based on the August 1980 DTNSRDC Report of John M. Durkin and the attendant testing, the following conclusions were drawn:

- . The aerodynamic behavior of the RD fan, when subject to variable back pressure due to wave pumping, is smooth and repeatable.
- . The RD is an effective "buffer" between the variable pressure air cushion and the fan. Other fans tested in a similar manner had

greater pressure flow hysteresis. Because of these characteristics, other fans did not achieve the smooth ride that RDs do with the installation of guide vanes.

. Active dynamic tests showed the ability of axial flow inlet guide vanes to maintain constant pressure in the SES cushion, in spite of wave pumping. Inlet guide vanes can provide an effective ride control system for any surface effect ship.

The tests conducted by Mr. Durkin were done on a single inlet, single width fan of a 0.73 proportion with a set of axial flow inlet guide vanes. Subsequently, Aerophysics has built a double width, double inlet model of the fan proposed for the 3KSES program. This DWDI fan has a proportion of 0.65, and two types of guide vanes, axial and radial, have been built for this fan. In spite of the increased distance between the radial flow guide vanes and the fan inlet, this system was found to be equally effective in control of plenum pressure in the simulated wave pumping tests.

As a result of this test, a third type inlet is being tested - the three-vaned damper caisson - which also has its vanes at a distance from the fan inlet. This third guide vane system is mechanically simpler and, hence, would be less expensive and possibly more reliable than either the axial or radial systems. Should it prove as effective in simulated ride control testing, this inlet system would be the configuration of choice for the full-scale SES applications.

None of these tests of inlet guide vanes in closed loop arrangements for ride control have simulated in full an SES lift system. That is, the test configurations have not represented realistically the ducting arrangements that will be necessary in an actual ship installation. For example, the installation proposed for the 3KSES includes ducts of more than 100 feet from the aft fan to the aft cushion seal,

duct was used between the forward fan and forward seal. The center or cushion fan had very little duct.

Using inlet guide vanes to respond to pressure variations in the cushion produced by wave pumping has been effective in each of the test arrangements discussed. However, it cannot be concluded from these tests that a single controller could perform satisfactorily with ducting arrangements as different as proposed for that ship. This is, of course, a judgment only. None of the test configurations contained provision for variation of the duct length as a test parameter. Extension of the ride control simulation tests would be appropriate before attempting to finalize a lift system design, particularly a design that will rely on inlet guide vanes for ride control. In fact, an investigation of the effectiveness of volute exhaust flow control would be of interest. In this application, a simple valve is placed within the volute discharge - cushion intake feed air duct. Modulation of this gate type valve should be most effective in controlling the perturbations of the cushion pressure. In planning further dynamic tests of a lift fan system, it would be most desirable to consider the requirements for the design and construction of a full-scale "black box" that will be needed to close the loop between the cushion pressure and the inlet guide vane system.

The vent valve had been proposed for use with other fans in earlier SES applications. However, tests with the RD fan strongly suggest vent valves will not be necessary.

Figure 4.3.1 describes the axial flow inlet guide vane as tested by Aerophysics in 1980 and referenced in this report. Figure 4.3.2 shows the radial inlet guide vane system installed with the same model fan. Testing of this arrangement was completed by Aerophysics in early 1982, a reference of which appears in this report.

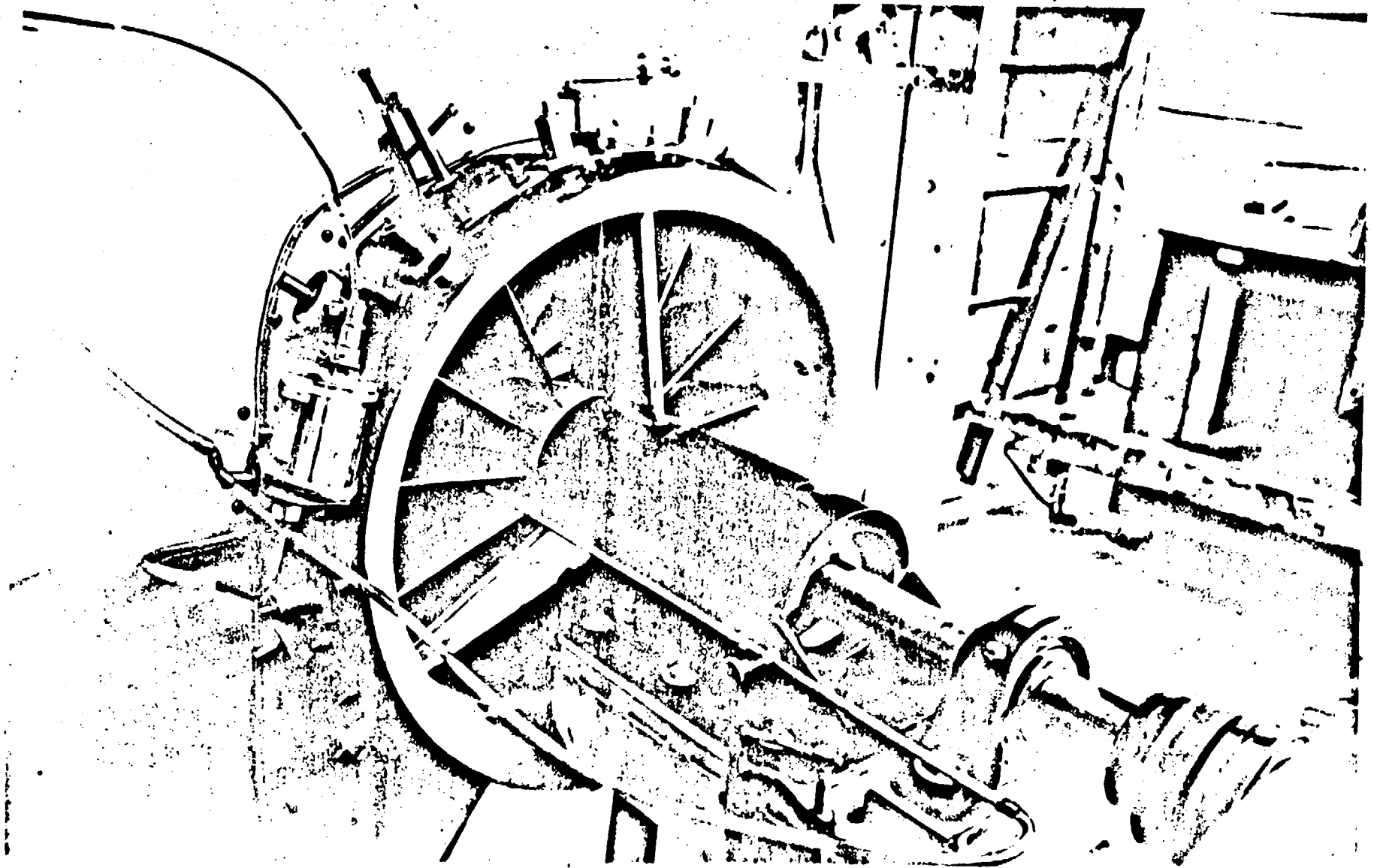


FIGURE 4.3.1: Axial Flow Inlet Guide Vane

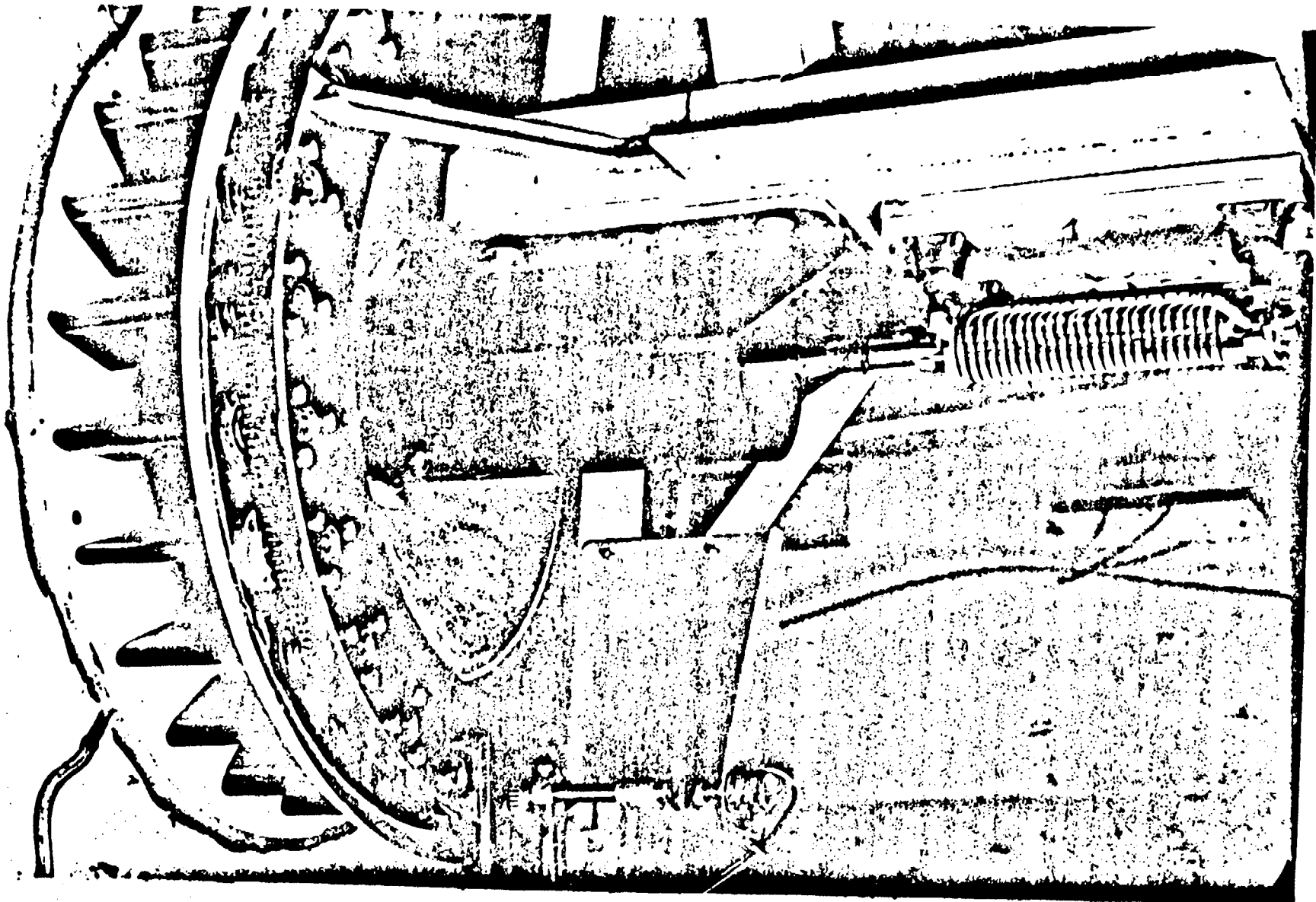


FIGURE 4.3.2: Radial Inlet Guide Vane System



## 5. MANUFACTURING AND STRUCTURAL RD FAN CONSIDERATIONS

### 5.1 Mechanical Description of the RD FAN

The RD fan is a mixed flow fan, whose impeller combines the actions of the centrifugal and axial flow impellers in a single fan stage. The unique design feature of this fan is the extension of an inner (backplate) and outer shroud beyond the blade trailing edges, facilitating further pressure recovery before the air is discharged into the volute.

The RD fan for the SES program is of a design operating since 1960. The wheel is a centrifugal discharge impeller with an integral axial inducer system. As noted, the outer shroud as well as the backplate, may extend as much as 30 percent beyond the blade trailing edges to create the rotating diffuser passage.

Single surface blades are made from flat sheets. Such blades are easy to make, to install, to service and are long-lasting. The diffuser rings may be arranged in a parallel manner, but in some designs, they may be angled outward a few degrees. (See Figures 5.1.1, 5.1.2 and 5.1.3).

In some design studies, the backplate may be of two elements. A backplate inner portion, considered as a radial extension of the hub may be a tapered disc welded to the hub shaft. The air passages between the blades are completed by addition of an inner shroud fairing surface on a contoured central hub. The inner shroud or hub surface is the inner boundary of the flow channel.

The inner shroud or backplate may be a common part to each of the double inlet, double width units. An alternate is to mount two

single inlet, single width units back-to-back on a common shaft, permitting thus to double the airflow for a given pressure rise.

The outer shroud, which may vary in thickness, is welded to the blades. The outward extension of the shroud forms the upper surface of the radial diffuser.

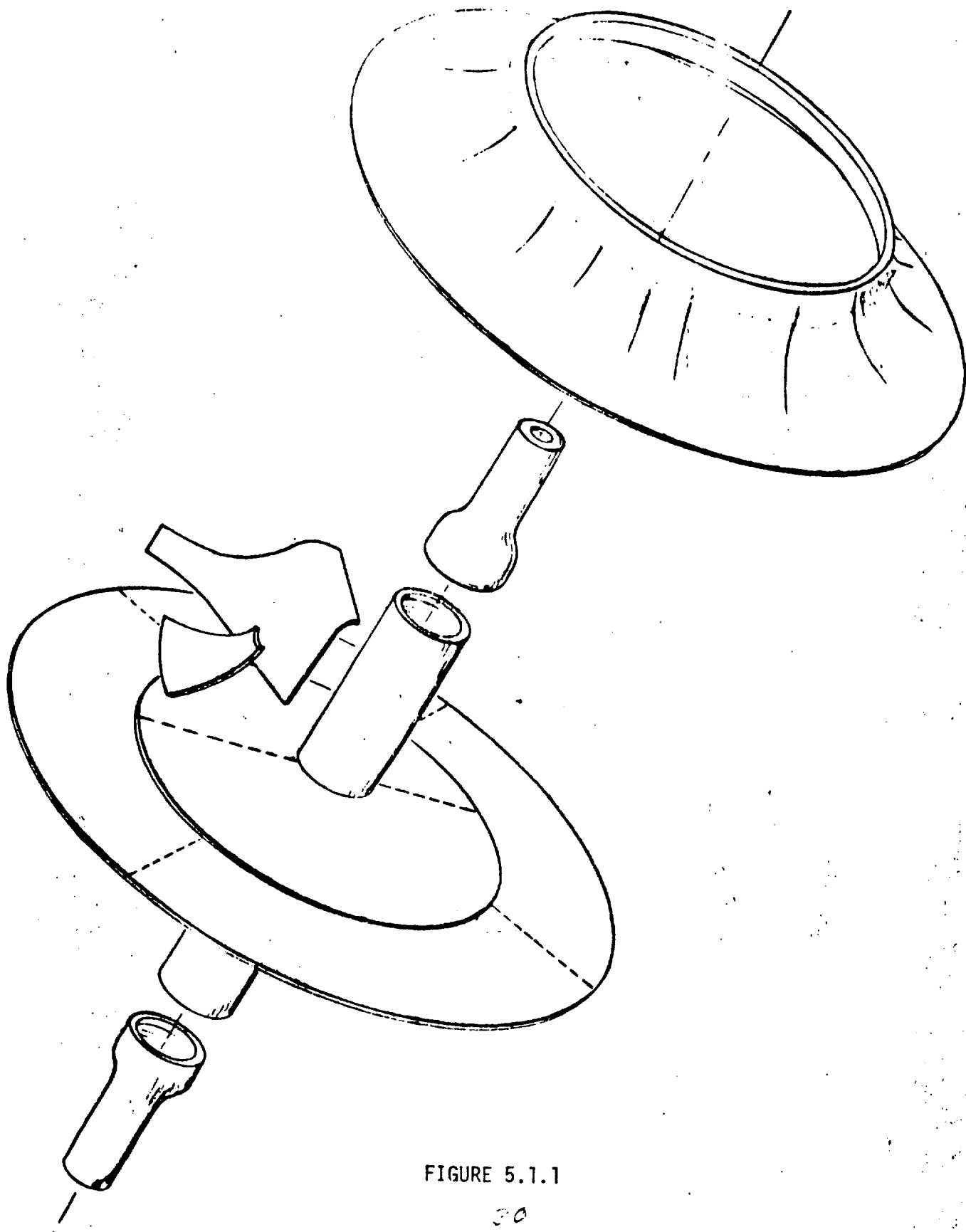
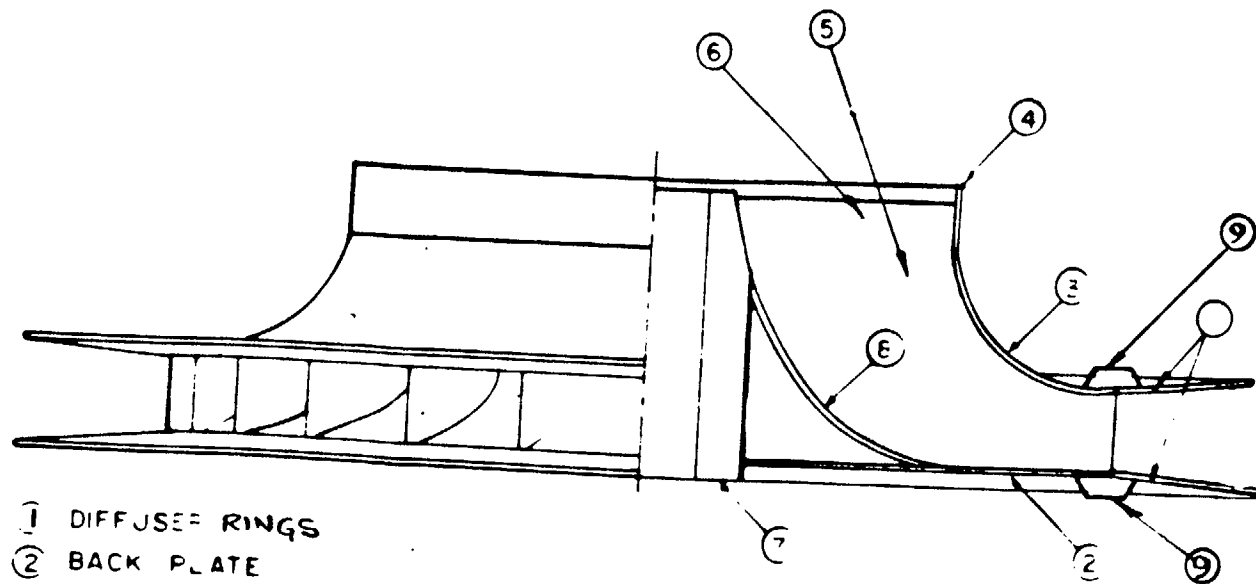


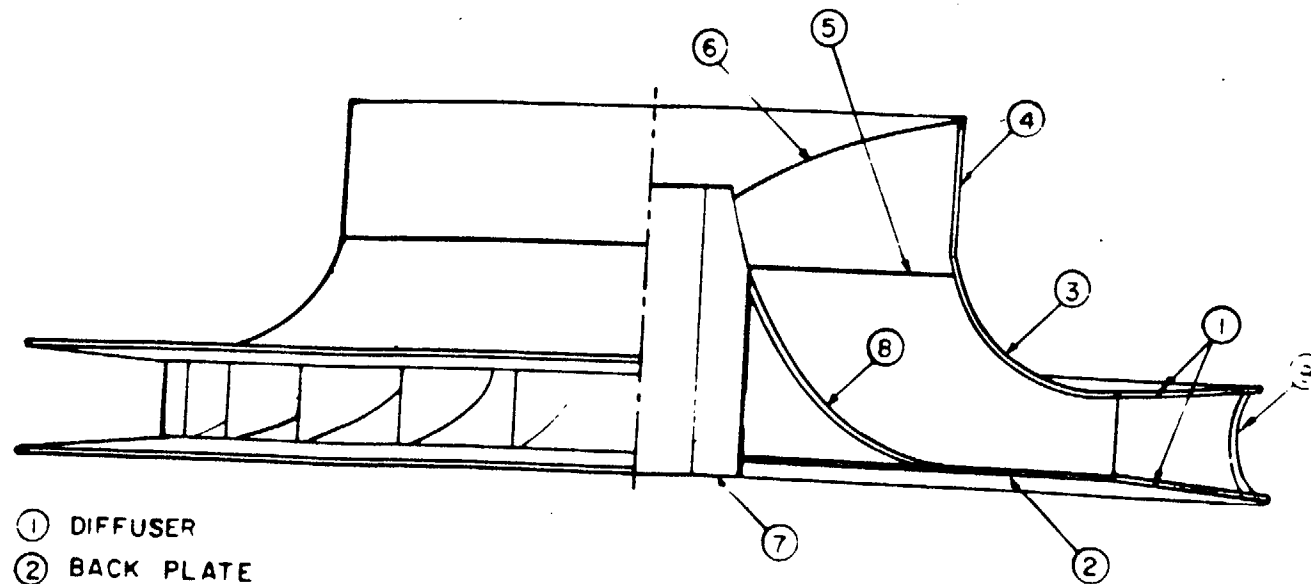
FIGURE 5.1.1



- ① DIFFUSER RINGS
- ② BACK PLATE
- ③ CONE
- ④ INLET CYLINDER
- ⑤ BLADE
- ⑥ INDUCER
- ⑦ HUB
- ⑧ FAIRING
- ⑨ ANTI-SURGE VANE

- ① ③ ④ FRONT SHROUD
- ① ② BACK SHROUD

Figure : Impeller Nomenclature  
Figure 5.1.2.



- ① DIFFUSER
- ② BACK PLATE
- ③ CONE
- ④ INLET CYLINDER
- ⑤ BLADE
- ⑥ INDUCER
- ⑦ HUB
- ⑧ FAIRING
- ⑨ MODULATING VANE

- ① ③ ④ FRONT SHROUD
- ① ② BACK SHROUD

FIGURE 5.1.3.

## 5.2 Material Selection and Fabrication Limitations

Industrial RD wheels are manufactured in various standard metals, such as, high strength carbon steel, stainless steel, alloy aluminum; in small high speed compressors, cast titanium is used. Basic industrial wheel material is chosen usually, to provide a balanced service between:

- high tip speeds, high stresses and usually high pressures
- durability to resist destructive effects of abrasion and/or corrosion
- endurance of a long service life.

Marine SES requirements are similar in that the wheels must:

- develop higher pressures at high tip speeds
- resist the effects of the marine corrosive experience
- be of such a weight that the ship total weight requirement is not overburdened
- endure a long mission life between service or overhaul times.

Our preliminary studies show that for the ship design in which the wheel and lift system wheel is a small fraction of the total ship weight, it is well to use a wheel and lift system with the following material selections criteria:

- Wheel is generally to be made by industrial fabrication techniques.
- Wheel is to be built off from a central forged hub of the industrial type.
- Wheel is to be of a built-up sheet element welded structure.
- Wheel is to be of a commercially-available high strength alloy steel. USS T<sub>1</sub> alloy steel or its commercial counterpart would be the material of choice.
- Hub forging must have mechanical characteristics at least equal to those of the plate T<sub>1</sub> elements and have like corrosion resistant properties.
- Wheel shaft, whether keyed or welded integral to the hub,

- should also be of T<sub>1</sub> or its equivalent.
- Surrounding elements of the wheel, such as the volute box, inlet bellmouth IGV's, inlet caisson and the like, can be fabricated out of the same parent metal as the rest of the ship - or if weight should be a problem, an aluminum alloy such as S456 AL.
  - Choice of constructional alloy high strength "T<sub>1</sub>" is an excellent choice, because of its excellent proven corrosion-resistant properties. There must be further study and consideration of giving the steel elements of the wheel and shaft a protective coating against further corrosion problems. One such technique is the application of the "metallized" surface treatment.

The excellent corrosion resistance of pure zinc and aluminum are well-known. Metallizing\* provides an ideal method of applying such coatings at high speed, low cost, to any desired thickness - frequently a very important consideration because the corrosion-resistant life of such a coating is directly in proportion to the thickness of the metal applied.

\*The term "metallizing" is used to describe that type of flame-spraying process which involves the use of metal in wire form. The wire is drawn through the gun and nozzle by a pair of powered feed rolls. Here the wire is continually melted in the oxygen-fuel-gas flame and atomized by a compressed air blast which carries the metal particles to the previously-prepared surface. The individual particles mesh to produce a coating of the desired metal. This meshing action is still not completely understood, but the effect is apparently due to a combination of mechanical interlocking and cementation of the oxides formed during the passage of the particles from the gun nozzle to the sprayed surface.

USS "T<sub>1</sub>" and "T<sub>1</sub>" type C Steels have 4 to 6 times the atmospheric corrosion resistance of structural carbon steel. To achieve the benefits of the atmospheric corrosion resistance of bare "T-1" Steels, it is necessary that proper design, fabrication and erection practices be observed.

A general description of this remarkable steel, its mechanical properties and a brief description of its fabrication and welding requirements can be made, as follows:

The "T<sub>1</sub>" Steels are a group of quenched and tempered constructional alloy steels with an attractive combination of advantages and characteristics. The most important are high yield strength (about 3 times that of structural carbon steel), weldability and good toughness at low atmospheric temperatures. Designed for a wide range of structural uses, as well as for machinery and equipment, these constructional alloy steels offer a selection to help approach the optimum in strength, toughness, corrosion resistance, impact-abrasion resistance, and long-term economy. The high-yield-strength steels, as a group, have a lower tolerance than lower-strength structural steels for zones of high stress concentration at design details and weld imperfections. Therefore, to obtain maximum advantage of the characteristics of the "T<sub>1</sub>" Steels, it is necessary in their application that their higher yield strength be accompanied by refinements in design, workmanship, and inspection.

"T<sub>1</sub>" Steel . . . the pioneer grade in the quenched and tempered constructional alloy steel family . . . features an attractive combination of strength, toughness, fabricability and economy in thicknesses through 2½ inches (63 mm). The chemical composition for "T<sub>1</sub>" steel and the mechanical properties for quenched and tempered plate product are shown in the following tables:



When so required, "T<sub>1</sub>" Steel plates can be produced to:

- ASTM A511 Grade F
- ASTM A517 Grade F
- AASHTO M244 Type F
- ASTM A709 Grade 100
- ASTM A709 Grade 100W

USS "T<sub>1</sub>" Steel Chemical Composition, percent (cast or heat analysis)

C	Mn	P	S	Si	Ni
0.10/0.20	0.60/1.00	0.035 max	0.040 max	0.15/0.35	0.70/1.00
Cu	Cr	Mo	V	B	
0.15/0.50	0.40/0.65	0.40/0.60	0.03/0.08	0.0005/0.0006	

USS "T<sub>1</sub>" Steel Tensile Properties - Plate

Thickness, in. (mm)	Thru 2½ (63)	
Yield Strength, min KSI (MPa)	100(690)	
Tensile Strength, ksi (MPa)	110/130 <sup>(1)</sup> (750/900)	
	Long.	Trans.
Elongation in 2 in., (50 mm), min %	18	16
Reduction of Area, min, %		
3/4 in. (19 mm) and under	40 <sup>(3)</sup>	35 <sup>(3)</sup>
Over 3/4 in. (19 mm)	50 <sup>(2)</sup> 40 <sup>(3)</sup>	45 <sup>(2)</sup> 35 <sup>(3)</sup>

Test specimens, procedures and elongation modifications conform to ASTM specifications.

(1) 115/135 (790/930) when ordered to ASTM A517 Grade F.

(2) Measured on ½ in. (12.7 mm) diameter specimen (Fig. 5 ASTM A370).

(3) Measured on 1½ in (38 mm) wide full thickness rectangular specimen (Fig. 4 ASTM A370), which is mandatory for thicknesses ¾ in. and under.

Impact Values, min ft/lb (J), Charpy V-notch, Avg. 3 Specimens - Plate

Thickness, in. (mm)	1/16 thru 1/4 (11.1 thru 32)
Temperature, °F (°C)	-50 (-46) or higher
Min Ft-Lb (J) Long.	15 (20)
Min Ft-Lb (J) Trans.	15 (20)

Although toughness is a general characteristic of USS "T<sub>1</sub>" type A Steel, Charpy Impact values for toughness will be reported only when specified on the order. The minimum values shown in the above table may be specified, or other toughness values may be specified, subject to negotiation. Charpy impact test specimens are located in accord with ASTM specifications as near as practicable mid-way between the surface and the center of the plate thickness. Modified impact values are applicable for thicknesses under 1/16 in. (11.1 mm).

"T<sub>1</sub>" Constructional Alloy Steels are water-quenched from 1650/1750°F (900/955°C) and tempered at a minimum temperature of 1100°F (590°C). (BHN plate may be tempered at lower temperatures).

"T<sub>1</sub>" Steels can be cold-formed. Suitable bending radii and increased power must be employed because of the high strength of "T<sub>1</sub>" Steels compared to that of structural carbon steel: Suggested minimum bending radii are given in the accompanying table.

For brake press forming, the lower die span should be at least 16 times the plate thickness to avoid high localized strains that might cause fracture of the steel.

"T<sub>1</sub>" Steels - Typical Physical and Engineering Properties

The values shown below are only typical properties and are not specification requirements.	
Density, lb/cu ft (kg/m <sup>3</sup> )	490 (7850)
Electrical Resistivity, microhm cm	18 to 26
Modulus of Elasticity, Tension, psi (MPa) Compression, psi (MPa)	28 to 31 x 10 <sup>6</sup> (19 to 21 x 10 <sup>6</sup> ) 28 to 31 x 10 <sup>6</sup> (19 to 21 x 10 <sup>6</sup> )
Coefficient of Expansion, in./in./°F (mm/mm/°C) in the range of 50 to +150°F (-46 to +65°C)	6.5 x 10 <sup>-6</sup> (11.7 x 10 <sup>-6</sup> )
Shear Strength Yield Ultimate	Approx. 58% of tensile yield strength Approx. 75% of tensile strength
Fatigue Limit Stress amplitude from rotating beam, polished specimens	Approx. 50% of tensile strength

### Drop-Weight Test, NDT (Nil-Ductility-Transaction) Temperature

Steel	Thickness, in (mm) incl.	Temperature, °F (°C)
"T-1"	½ to 2½ (12.7 to 63)	-30 to -130 (-34 to -90)
"T-1" type A	½ to 1¼ (12.7 to 32)	-5 to -60 (-20 to -51)
"T-1" type B	1¾ (35)	-40 (-40)*
"T-1" type C	5 to 6 (125 to 150)	-80 to -110 (-63 to -79)†

### Explosion-Bulge Test, FTE (Fracture-Transition-Elastic) Temperature

Steel	Thickness, in (mm) incl.	Temperature, °F (°C)
"T-1"	½ to 2 (12.7 to 50)	-40 to -60 (-40 to -51)
"T-1" type A	½ to 1 (12.7 to 25)	-5 to -60 (-20 to -51)
"T-1" type B	1¾ (35)	-10 (-23)*

\*Single test results.

†Limited test results.

### "T-1" Steels Cold-Forming R<sub>dii</sub> for Plates

Thickness, Inches (mm)	Suggested Minimum Inside Radius
Thru 1 (25)	2t
Over 1 (25) thru 2 (50)	3t

**Note:** For improved formability "T-1" Steels up to 2½ in. (63 mm), incl can be ordered to a yield strength range of 100-115 ksi (690-790 MPa) with tensile strength for information only.

The "T<sub>1</sub>" Steels cannot be hot-formed without impairing their mechanical properties. If hot forming is necessary, the product must be quenched and tempered after hot forming to restore its properties. Such a product cannot be designated as one of the "T<sub>1</sub>" Steels unless this heat treatment is performed by a producer because the mechanical properties of the "T<sub>1</sub>" Steels may not have been restored.

"T<sub>1</sub>" Steels can be welded satisfactorily by all major welding processes when proper procedures are used. Some suggested welding practices for T<sub>1</sub> Steels are shown in the table on the next page.

## Suggested Welding Practices for T<sub>1</sub> Steels

Welding Process	Electrode	
Shielded Metal-Arc	E11018-M per AWS A5.5, latest edition; lower strength low-hydrogen electrodes, depending on design stress and application, may also be suitable if dried to moisture level of E11018 electrode.	
Submerged-Arc	F114-EF5-F5 per AWS A5.23, latest edition.	
Gas Metal-Arc	ER110S-1 per AWS A5.28, latest edition.	
Flux-Cored-Arc	E110T5-K3 or E110T5-K4 per AWS A5.29, latest edition.	
Suggested Minimum Preheat or Interpass Temperature, °F (°C)		
Plate Thickness in. (mm)	Produced to Published Tensile Properties	Produced to Minimum BHN Hardness Requirements
Up to ½ (12.7), incl.	50(10)	100(40)
Over ½ (12.7) to 1 (25), incl.	50(10)	150(65)
Over 1 (25) to 2 (50), incl.	150(65)	200(95)
Over 2 (50)	200(90)	250(120)

**Note 1.** Other electrodes or electrode-flux or electrode-gas combinations depositing weld metal of other classifications different in strength and toughness may be either necessary or sufficient depending on design stress and application.

**Note 2.** A preheat temperature above the minimum shown may be required for highly restrained welds.

**Note 3.** Welding steel which is at an initial temperature below 100°F (38°C) may require localized preheating to remove moisture from the steel surface.

**Note 4.** Low-hydrogen electrodes for shielded metal-arc welding, the electrodes, fluxes and/or gases for the other welding processes, as well as the weld joint, must be sufficiently low in moisture, hydrogen or hydrogen-containing contaminants that unacceptable imperfections caused by excessive hydrogen are not produced in the weld metal or heat-affected zone.

**Note 5.** No welding should be done when temperature in the workplace is below 0 F (-18°C) to minimize the possibility of poor workmanship.

**Note 6.** Welding heat input should not exceed maximum values provided in tables in "How to Weld USS 'T-1' Constructional Alloy Steels," United States Steel Corporation, latest edition.

**Note 7.** Electrodes, or electrode-flux or electrode-gas combinations, normally used to weld the "T-1" Steels have sufficient alloy content to provide welds with satisfactory corrosion resistance for unpainted structures exposed to the atmosphere. If carbon steel and some low-alloy steel weld metals are used, depending on joint design and welding procedure, such weld metal may not be satisfactory for the unpainted structures.

Further information on welding, postweld heat treatment, and gas or plasma-arc cutting is given in Reference 8.

Generally welded structures of T<sub>1</sub> Steels should not be given a postweld stress-relief heat treatment. Loss of weld-metal and heat-affected-zone toughness and stress-rupture cracking may occur as a result of such treatment.

T<sub>1</sub> Steels can be oxygen cut or plasma-arc-cut using good shop or field practices in accordance with those suggested in the AWS Handbook. Cutting of this material generally does not require preheating in thicknesses up to and including 4 in. (100 mm), but the steel temperature should not be lower than 50°F (10°C) during cutting to minimize the possibility of thermal cracking.

To avoid cracking, T<sub>1</sub> Steels must be in the quenched and tempered condition prior to any oxygen cutting, plasma-arc cutting or welding.

In the case of the smaller SES, where the lift system and the wheel weight begin to be felt as a major fraction of the ship weight, a lighter wheel and shaft weight are called for. Studies have shown that a titanium wheel and integral shaft system might be the arrangement of choice. In this approach, the major central hub forging is eliminated, and the fan blades radiate directly from the combined hub-drive-shaft tube (Fig. 5.1.1).

A titanium alloy of choice, Ti 6Al-4V is the most widely-used titanium alloy. It has been used by the Navy in other applications, and has been extensively tested by the Navy research establishment at Annapolis. The alloy is generally considered immune to corrosion in sea water environments at moderate temperature, and in the ELI (Extra Low Interstitial) composition, annealed, is resistant to stress corrosion cracking and shows only a limited corrosion fatigue degradation compared to similar tests in air.

The composition of the alloy is about 90 titanium, 5.5 to 6.5 aluminum, 3.5 to 4.5 vanadium, and about 1 other element, called interstitial element; consisting primarily of oxygen, nitrogen, carbon, and hydrogen. Additionally, a small amount of iron is usually present and is reported with the ingot chemistry. Trace amounts of other elements may

be present, but are considered unimportant and are not normally reported in chemical analysis. The extra low interstitial grade imposes restrictive limits on the five elements noted.

The ELI grade is tougher, more ductile, easier to form, and more readily welded than higher strength grades with normal interstitial content. In the annealed condition, the material normally has a strength level of 110 to 120 KSI yield and 120 to 135 KSI ultimate. Most mill products are supplied in the annealed condition.

Mill annealing usually consists of holding for one to eight hours in a furnace at 1,300 to 1,550°F, furnace cooling to 1,050°F, followed by cooling in still air. It is important to note that achieving full design properties in heavy sections (forged hub and shaft, for example) may require relaxation of the ELI restriction.

The titanium alloy Ti 6Al-4V ELI has a fatigue limit in sea water approximately, as follows:

Wrought material	90,000 psi
Cast of welded material	30,000 psi

These data were generated in Navy-sponsored programs related to cast and wrought propellers for early hydrofoil applications. Titanium is sensitive to structural notches as well as stress raisers associated with changes in material section and junctures. In addition, titanium mill products often have surface conditions that are detrimental to fatigue performance unless appropriate surface treatment is undertaken. These conditions include alpha case (oxygen enriched surface layer of low ductility) and surface grinding (used at the mill to remove surface defects from plate).

These three <sup>concerns</sup> concepts translate into design concepts applicable to the RD fan wheel:

1. Minimize welding.
2. Locate welds out of zones of change of cross section.
3. Use full penetration welds to eliminate notch effects at critical weld structure.
4. Machine or chemically pickle mill surfaces after final mill finishing.

The blades and outer section of the backplate for the RD fan are to be fabricated from hot rolled plate. Generally, the minimum thickness for hot rolling is .187 inch. Below that gage, flat roll products are defined as sheet, and most of these products are cold finished. The practical size limitations on plate at .187 inch gage for U.S. mills is about 97 inches, which is adequate for a fan. Sheet products are limited to 48 inches in one mill and 60 inches in another, and would require additional welding in the sizes required for the fan. Mill gage control on cold finished (sheet) products is substantially better than in hot finished (plate) products. This is significant because, without subsequent surface work, the weight of plate products can over-run the nominal weight by 5 to 15 percent. In addition, variance in gage on a specific plate can result in increased problems of balancing the wheel.

Problems in plate gage control can be overcome by special surface finishing, usually belt grinding or polishing followed by additional acid pickling. Dimension control of the hub-shaft and center section of the backplate will be achieved by machining.

The shroud-thickness dimensional tolerances were discussed previously and are a function of the forming method more than of the thickness of input plate. The outer periphery of the shroud would be machined after final assembly, and accurate control of both thickness and axial run-out can be obtained.

Although many exotic welding processes could be considered, the most widely-used method of producing satisfactory quality welds is the gas-tungsten-arc process. In this process, a welding arc between a non-consumable tungsten electrode and the workpiece is maintained, with bare filler metal added to the weld puddle. The weld area is protected by an inert gas from the welding torch. Argon is most widely used, although argon and helium mixtures can be used also.

Titanium is a reactive metal which means it will readily combine chemically with nearly every element except the inert gases. Normally, the surface of titanium is covered with a tenacious self-healing surface oxide film that is the real source of the corrosion resistance of the material. If the surface is scratched in air or water environment, the oxide quickly re-forms. At elevated temperatures (nominally above 1,200°F), the surface oxide is dissolved by the metal and the oxygen slowly diffuses into the surface, causing surface embrittlement. Long-time exposure, such as occurs in hot working of mill products produces a very heavy oxygen rich area commonly called "alpha case."

When titanium is molten, it readily absorbs oxygen, nitrogen, and other elements from the atmosphere. In the molten state, particularly in a highly-agitated weld puddle, these contaminating elements quickly disperse throughout the molten zone causing high levels of interstitial and contaminating elements. Embrittlement, loss of corrosion resistance and other undesirable effects result.

Consequently, when welding titanium, the material must be clean, and all molten and hot surfaces (over 1,200°F) must be protected by an inert gas blanket. Because a heavy visible surface oxide forms quickly in the temperature range of 1,200°F to 800°F, the titanium surface is usually protected by inert gas to a temperature where little visible surface oxide forms.



The principal problems with the gas-tungsten-arc process is that it is slow and thermally inefficient. We can live with the process speed, but the thermal inefficiency results in relatively high welding-induced distortion, low elastic modulus and rapid loss of strength even at moderate temperatures.

The primary concept in adapting design for titanium welding is provision of access to both the front and back sides of every weld joint. This is important because proper inert gas shielding is readily accomplished when the root of a weld is accessible. Lack of access is accommodated by purging where reliance is placed on monitoring the purity of the purging gas.

In addition to weld shielding, inspection and verification of non-contamination, weld repairs and ease of obtaining full penetration welds with smooth surface contours is enhanced. Often as important, future inspection and repair of welds is also made possible.

Some areas within the smaller RD wheels may present welding problems, because of accessibility to the weld joints inside the air passageways. The full welded backplate has been designed to partially overcome this problem. Removable inner shroud fairing panels provide adequate access to all welds both for manufacturing and subsequent inspection, in the hub-tube-to-blade fan configuration.

We recommend thermally stress-relieving the completed fan unit to improve dimensional stability for final hub and seal ring machining, to reduce the possibility of in-service distortion. These distortions result from cyclic service loading. This stress relieving improves fatigue performance in the finished wheel.

Stress-relieving of the unalloyed and Ti 6Al-4VELI alloy can be accomplished in conventional electric or gas fired (oxidizing atmosphere)

furnaces at a temperature of about 950°F for the unalloyed and 1,150°F to 1,200°F for the Ti 6Al-4V ELI furnace cooled at a controlled rate to 500°F and air cooled to ambient.

Fan wheel balance will be achieved by various manufacturing controls as well as final dynamic balancing.

Parts like blades and inner shrouds will be matched for weight and installed to balance about the centerline. Parts like the hub-shaft and backplate will be accurately pre-machined. The shrouds will be static balanced and pre-machined. Careful control of radial positioning of blades and accurate alignment of the shrouds will also be used.

Finally, the outside diameters of the shrouds and backplates will be trued when the shaft is finish machined. Dynamic balancing of the fan at normal operating speeds is a routine procedure. Consideration of weight addition or removal for balancing is still a necessity. Location for addition of weights or other provision for removing balancing material must be made.

Several basic inspection techniques are applicable to the RD fan wheel design, depending on the configuration of the area being checked. Wherever possible, at least two techniques should be used on each area:

1. Visual inspection
2. Liquid penetrant inspection
3. Ultrasonic inspection
4. Film radiographic inspection.

The ultrasonic and radiographic inspection methods tend themselves to periodic "shop" or "drydock" inspections. However, with the elimination of partial penetration welds and other structural discontinuities below

the surface, visual and penetrant inspection may be entirely adequate for inspection in the field. There should be periodic visual service inspection and liquid penetrant inspection of selected areas, on a monthly basis. It is important to identify, in the inspection manual and instructions, the specific points in the design where problems would occur first. Usually these areas are in welds where stresses are high or section changes occur. Although other critical inspection locations can be identified by detailed analysis, the welds between the fan blades and shrouds, particularly at the blade inlet and outlet tips, are good examples of areas where detailed inspection would probably be required. A description of the piece and part fabrication of the RD fan wheel elements, in Titanium, is found in Reference 8.

### 5.3 Stress Analysis

For purposes of preliminary design, as well as prototype model and full-size wheel fabrication, Aerophysics uses a procedure of stress analysis described by Deutsch in Reference 11. This is a method of stress analysis for shrouded discs and wheels, which has been refined, extended and applied considerably from its original presentation. Formerly, wheels of simple configuration and operating at low stress levels could be treated by the method of Stodola, as a loaded profile disc. The higher tip speed, stress levels and mechanical geometrics of the RD wheels now make use of the basic Deutch procedure.

The Deutsch method of solution is based upon the availability of a high-speed programmed digital computer. In setting up the input for the calculations, it has been possible to keep the technical knowledge required to a minimum. The data necessary for the calculations have been arranged in such a manner as to enable a draftsman to fill out the required input sheets, thereby freeing the stress analysts for other work. The system to be solved is one of a redundant structure which is analyzed by an iterative energy-balance method. All measurements of the convergence of the successive approximation are pre-programmed into the procedure, and the computer set to recycle as required to obtain the desired accuracy.

The problem is complicated by the fact that the shrouded wheel is an indeterminate structure. An energy-balance method that properly apportions deformation energy between the hub and shroud must be used. In addition, an exact solution is impossible, and a computer solution using an iterative scheme must be obtained. Additional refinements to the basic analysis have been made to handle structural problems concerning local and detailed design geometry refinements, as well as to handle unique stress problems particular to the RD wheel. For example,

means are included to handle local stresses particular to the addition of the rotating diffuser rings. A procedure of "fictitious densities" - a means to identify local wheel design additions - greatly enhances the range of the original procedure.

A prerequisite for this system of calculation is the availability of a method of determining radial and tangential stresses in profiled discs by a finite difference analysis. Such a method should include provision for varying the material density, coefficient of thermal expansion, Poisson's ratio, Young's modulus, and operating temperature at each station. The use of the system of calculation as given by Manson<sup>2</sup> is recommended.

As the types of materials used by each manufacturer are generally limited to a few alloys, the input data can be simplified. Thus a predetermined table of physical properties can be established for each material, and the input data need only indicate the material code. Young's modulus, shear modulus, density, Poisson's ratio, and coefficient of thermal expansion can then be fed into the calculations automatically.

In calculating stresses produced by rotation, the shape of the wheel is first established. The profile of a simple disc may be mathematically determined so as to give a desired stress pattern during rotation by application of the method proposed by Tumarkin. Use of this analysis would be a good approximation for the desired hub shape. However, the application of this method by an experienced designer is not warranted in the case of the shrouded wheel. In general, minimum thicknesses are desired for the outer portion of the shroud and hub, while all sections should either increase in thickness as they approach the center of rotation, or at least nowhere decrease in thickness. Hub contour is largely determined by the mechanical requirements of the wheel drive. With a little experience, a suitable contour can be assumed, the stress calculated, and any readjustment of the profile quickly made.

The nomenclature and identification of the various elements entering into the analysis of the disc are given in Fig. 5.3.1. In preparing to use this method of calculation, a focus point is chosen so that rays drawn from this point will intersect both the shroud and hub as nearly normal to their surfaces as is possible. These intersection points establish the location of the sections used to determine the stresses acting in the blades, hub and shroud. Sections along the rays are given like station numbers in the blades, hub and shroud. By means of the finite-difference method, a conventional stress analysis is run for: (1) the hub of the wheel alone; (2) the shroud of the wheel alone; (3) the hub loaded with the full weight of the blades; and (4) the shroud loaded with the full weight of the blades.

The designer graphically fashions the RD wheel around this "ray" system, by picking off actual paper dimensions from layout wheel design. He enters these dimensions into the computer and interacts the board design with the stress analysis output. A peak stress at the mid-point of the blade tip may occur in some designs. This additional stress is not calculated by the Deutsch program. This stress is the result of the non-radial orientation of the outboard section of the blade. Strain gage measurements during model tests may indicate significant stresses at the blade tip. The valves may be calculated by treating the outer portion of the blade as a beam fixed at both ends with a uniform loading equal to the tangential component of the centrifugal force. The stress as measured in model tests, together with the known characteristics of the model and the conditions of the test, may be used to determine a fixity factor applicable to the blade as a beam. This factor may be used in calculating the stress for the full-scale wheel. In some designs, it may become necessary to increase the thickness of the blade tip to obtain an acceptable stress level.

The refined Deutsch procedure serves well for current preliminary wheel design. The advent of the finite element stress analysis is now considered for application to future RD wheel design. This approach is considered essential to obtain a more accurate evaluation of the stresses in the wheel, particularly in the vicinities of the welds. One proposed approach in applying this technology is presented here.

It is assumed that a single geometric configuration would be analyzed at two operating steady state wheel speeds representing design and overspeed conditions respectively.

A detailed geometric drawing of the rotor structure would be required, including:

- . The general axisymmetric cross section of the rotor.
- . The detailed description of the diffuser vane including thickness, twist, camber, etc., at a minimum of seven radial positions. When intermediate geometry is required for finite element definition, a linear interpolation would be performed.
- . A detailed description of the interface joints of the structure including specific weld geometry. Where weld tolerance exists, the minimum section would be used in the analytical model.
- . The analysis would assume linear elastic behavior; therefore no plasticity effects will be included.
- . The material would be assumed to be isotropic throughout with equal material properties in the longitudinal (rolling) and transverse direction of the plate.

For completeness, two approaches would be presented. However, one covers the entire rotor stress analysis including the detailed three dimensional steady state stress and frequency analysis of the diffuser vane, and the other covers the steady state stress and frequency analysis of the diffuser vane. (This assumes a prior input of detailed boundary condition.) These two approaches would be:

- . Complete rotor steady state stress and diffuser fan frequency analysis. The technical approach proposed for the analysis of the entire rotor assembly would be: Analyze axisymmetric rotor structure using two dimensional finite element computer program, as follows:
  - Prepare axisymmetric model with blades assumed radial to provide blade loading.
  - Run model at maximum RPM.
  - Printout steady state stresses and displacements.
  - Plot stresses and displacements.
  - Compare stresses with material properties.
  - Review data and make recommendations on possible design changes, if required, prior to the analysis of the blade. Design-computer iteration.

The results of this analysis would yield accurate radial and tangential stresses and displacements in the axisymmetric rotor structure, provided that the bending loads in the actual three dimensional blade are not excessive in terms of producing large out-of-plane loads. The stresses in the vanes have physical significance only if the blades are radial with no bending. These values will only be used as a reference in assessing the total centrifugal load of the blade. The blade needs to be modeled as part of the axisymmetric structure to provide the proper loading to the rotor structure.



- . Conduct three dimensional finite element stress analysis of the diffuser vane.
  - Identify the blade-shroud segment to be analyzed.
  - Model the actual 3-D vane configuration and shroud segments (if required).
  - Prepare material property related input for program.
  - Provide boundary conditions at the rotor interfaces consistent with the axisymmetric 2-D analysis.
  - Plot model and debug prior to running program.
  - Run model at design and maximum operating conditions.
  - Obtain steady state vibrations modes.
  - Plot iso-stress contour.
  - Review overall analysis and results.
  - Provide conclusions and recommendations on the analysis results.
  - Prepare final report.

- . Diffuser vane steady state stress and vibration analysis.

This analysis would follow the same rotational as the blade. 3-D finite element analysis is required, except that no axisymmetric analysis will be conducted. But detailed displacement and/or loading boundary conditions will be required.

Let us now outline briefly a description of the computer program required for the analysis. Axisymmetric program description:

This program permits the efficient and accurate analysis of the three-dimensional axisymmetric stress distribution in solids of revolution. In addition, 2-D finite elements handling plane stress or plane strain problems can be handled separately or in combination with the axisymmetric option. The program handles: general mechanical, thermal or body loading; non-homogenous material properties. The computer output will

include principal and effective stresses and corresponding nodal displacements.

#### Finite element program description:

The program will use a parametric, 3-D, finite element computer program capable of analyzing 3-dimensional anisotropic continua and particularly gas turbine engine blades and vanes. The program accounts for the inertia forces of rotation and vibration. The program gives deflections and stresses in both stationary and rotating bodies. In addition, with proper modification to the computer model, it will calculate the lowest eight modes and corresponding natural frequencies at a specified speed of rotation. The program utilizes an eight noded isoparametric box element of practically any shape. Each box has 33 degrees of freedom: 24 corresponding to the three motions at each of the eight nodes; and nine internally eliminated to minimize strain energy. The material properties would be 3-D-anisotropic. Thermal stresses would also be computed. Distributed pressures or point forces can apply external load to the structure. Boundary conditions can be applied in the form of displacements or loads in any of the mutually-perpendicular directions.

In addition to the requirement of having detailed geometry for the entire rotor structure, a definition of blade air loading and rotor temperature conditions should be provided if the levels are significant enough to be included in the analysis.

Stress analysis by 3-D photoelastic models is regarded by many as the most powerful of all stress analysis techniques, as it allows complete analysis - both external and internal - of a component or structure. Photoelastic analysis is often used to obtain basic design data, such as, stress concentration factors and the determination of stress trajectories. Unlike the previous Deutsch and finite element methods, the 3-D photoelastic technique is not a design-to-analysis iterative technique. It is essentially a one design, one duty point analysis.

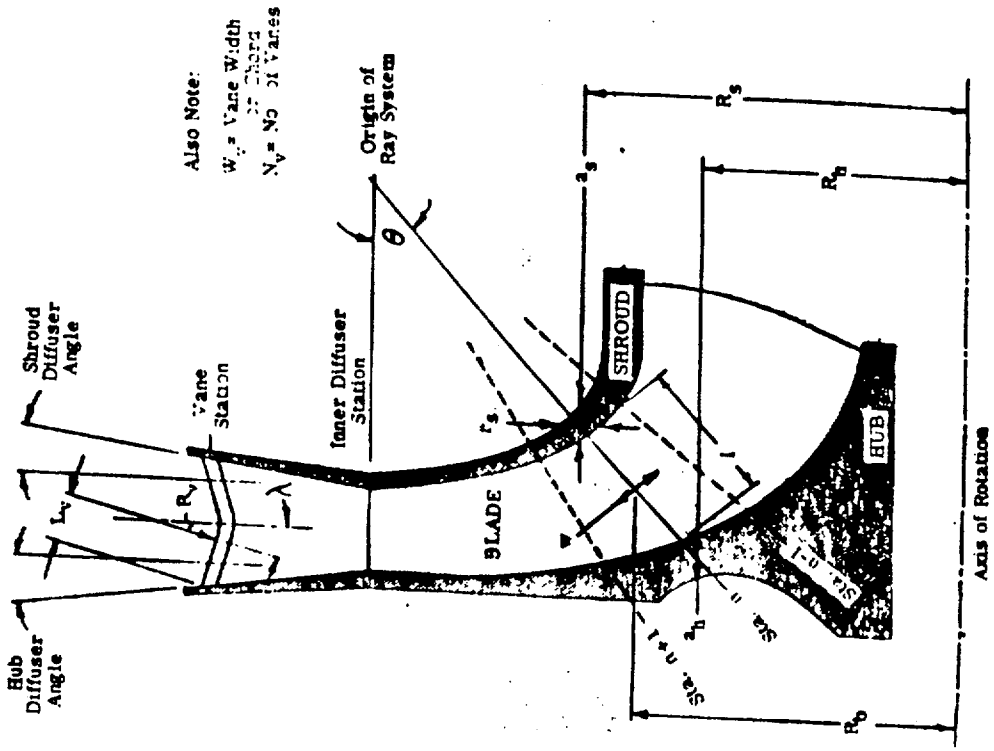
The procedure is, as follows:

- A mold of an exact-scale model of the RD design wheel is fabricated for an exact model pattern.
- From this pattern, a plastic (epoxy) model wheel is cast. This test wheel must exactly represent all of the mechanical and geometric characteristics of the design wheel. As an approximation, a DWDI wheel could be made from two SWSI model wheels cemented back-to-back.
- The model will be then subjected to a spinning/stress freezing cycle simulating centrifugal loads only. The wheel will be totally enclosed to prevent aerodynamic loads during the stress freezing cycle. The heated test wheel is allowed to cool down slowly to room temperature.
- The photoelastic stress pattern is now "frozen in" the model and will remain so indefinitely, provided the temperature is kept near ambient.
- The model may now be physically sliced in any desired number of planes. Cutting will not disturb the frozen stress pattern.

- The frozen patterns of interest may now be photoelastically examined.
- Photoelasticity measures stress and strain by detecting changes in the indexes of refraction light passing through the photoelastic material. The principle is based on the fact that polarized light, passing through a transparent plastic under strain, will split into two polarized beams, which travel in the planes of the principal strains. These beams will have different velocities, and the resulting shift is easily converted into useful stress and strain measurements with modern instrumentation. Direct surface-stress-concentration features of the wheel can now be determined.

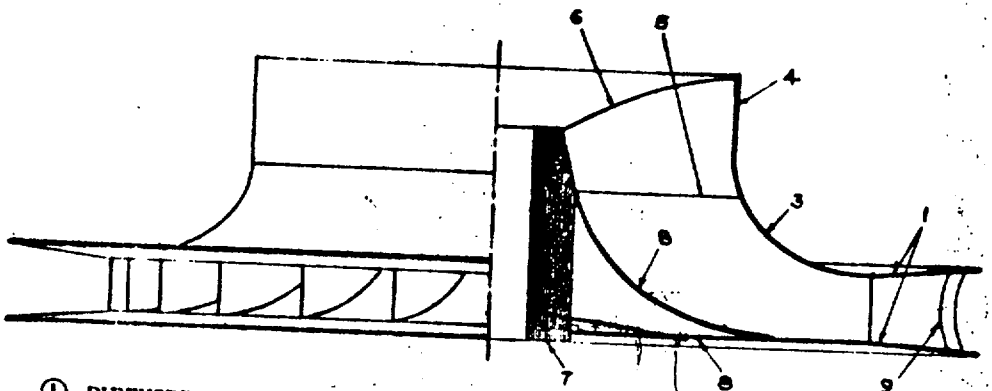
To use this technique as a design-analysis-redesign iterative tool would require that the model wheel be recast (reworked mold) with revised detail geometry and design modifications, an untimely and expensive process.

The model RD wheel is best used as a final stress check means after application of one or both of the previous analytical techniques.



Also Note:  
 $W_v$  = Vane Width  
 $C_c$  = Chord  
 $N_v$  = No. of Vanes

Figure 2. Symbols.



- ① DIFFUSER
- ② BACKPLATE (BACK DIFFUSER)
- ③ CONE
- ④ INLET CYLINDER
- ⑤ BLADE
- ⑥ INDUCER
- ⑦ HUB
- ⑧ FAIRING
- ⑨ MODULATING VANES

Note:  
 ①③④ FRONT SHROUD  
 ①② BACK SHROUD  
 ⑦⑧ INNER SHROUD

Above components may be made as complete units.

Figure 5.3.1: Impeller Nomenclature.

#### 5.4 Weight Estimation of the Wheel

Aerophysics has devised a very practical means of determining wheel and lift system weights. The lift system considered here only includes the complete wheel, wheel shaft, volute, shaft bearings, pedestal, mountings and immediate skid-foundation structure.

The major prediction method does not include the weight of any kind of inlet guide vane system intake air ducting, engine and engine drive components.

The prediction method which is presented in a log-log form in Figure 5.4.1 is synthesized from previous, actual, SCV, commercial and industrial known and available fan wheels and complete volutes with hardware. This actual experience of the wheel and system geometry and weight relationships, when plotted on the log-log graph of Figure 5.5.1, can be seen to fall in characteristic straight lines. Therefore, these relationships can be presented in the following simple formulas:

when:  $D_B$  = Wheel Blade Diameter, ft.  
 $W_1$  = SWSI Wheel CR System Weight, Lbs.

For SWSI Wheels: RD

$$W = 20 D_B^{2.67} \quad \text{Industrial Type in Steel. Small Inlet Eye \& Small Diffuser Rings.}$$

$$W = 31 D_B^{2.74} \quad \text{Industrial Type in Steel. Large Inlet Eye \& Large Diffuser Rings}$$

$$W = 2.58 D_B^{2.67} \quad \text{Light Weight Aluminum Type}$$

Backward Curved Commercial Type, High Efficiency

$$W = 5.75 D_B^3 \quad \text{Punker, Steel}$$

$$W = 11.5 D_B^{2.42} \quad \text{Heba A \& B, Steel}$$

$$W = 5.3 D_B^{2.42} \quad \text{Heba, Aluminum}$$

$$W = 1.98 D_B^{2.42} \quad \text{Bicycle Wheel Type. Dowty Roto1 Mixed R.P. Metal Construction}$$

Backward Curved Commercial Type, High Eff. (cont'd)

$$W = D_B^3 \text{ BHC Prediction}$$

Note: Wt. of DWDI Wheel = 2 x Wt. SWSI.

For Systems - Including Wheel, Shaft, Bearings, Volute and Framing  
but not IGV's

Backward Curve Industrial Type

$$W = 85 D_B^{2.34} \text{ SWSI, Buffalo Forge, BL Type}$$

$$W = 102 D_B^{2.53} \text{ DWDI, Buffalo Forge, BL Type}$$

RD For SES

$$W = 88.6 D_B^{2.66} \text{ DWDI Steel RD Wheel in Aluminum Volute \& Structure.}$$

Points to consider in this broad weight estimation arrangement are  
the following:

All Points & Curves are for Steel SWSI Wheels Unless  
Otherwise Noted.

Weight of SWSI RD Wheels Presented

Range of Industrial Type RD Wheel Weight Determined  
Boundary of Light Weight Aluminum Alloy RD Fan Determined  
Changes in Wheel Alloy-Material & Construction Tech-  
niques may be Estimated Between These Two  
Extreme Boundaries.

Available Commercial & Industrial Type Backward Curved  
Wheels and SWSI & DWDI Systems Shown.

Developed ACV/SES Lift Fan Weights Presented.

Predicted DWDI SES Lift Fan System Weight (Without IGV's)  
- Steel Wheel & Aluminum Structure-Volute - Estimated.

Of the greatest interest in this presentation to the naval architect, for his preliminary sizing and weight estimates, is the realistic RD lift fan system presentation of weights. Of immediate application is the SES lift system curve representing the DWDI, steel RD wheel with aluminum volute and structure, with a relationship

$$W = 88.6 D_B^{2.66}$$

Of great interest to the designer of the SWSI RD wheel is the technique which allows him to add the components to ascertain the total wheel weight. This is presented in Figures 5.4.2, 3, 4, 5, 6, 7 and 5.4.8. Using these curves, several of the local wheel geometry characteristics must be considered as factors in determining the final detailed component/part weight. These factors are also presented in the fields of the aforementioned Figures. When using these Figures, it should be noted that the characteristic weight curves (and diameter of gyration curve) are plotted on a logarithmic scale, which for amplification is folded back on itself.

The source of these curves is from actual industrial practice. These have been shaped by actual fabrication and practice.

From practice application of these curves, the following table should be used. Note that, as a practical matter and unlike the system weight curves, these curves are presented in metric form.



DETAILED WEIGHT DETERMINATION OF SWSI RD WHEEL  
BY ESTIMATION OF ELEMENTAL PARTS

- ELEMENT WEIGHT IN KILOS/ DIAMETER OF GYRATION (KG.M<sup>2</sup>)  
FOUND FROM SEMI-LOG CURVES - (FOLDED BACK. LOG-LOG CYCLES)
- WEIGHTS SHOWN ARE FOR STEEL ELEMENTS. OTHER MATERIALS  
MAY BE ESTIMATED BY MULTIPLYING BY THE DENSITY RATIO ( $\frac{\rho}{\rho_{STL}}$ )
- "WEIGHT", ON ABSCISSA, MUST BE MULTIPLIED BY SEVERAL  
CORRECTION FACTORS FOR:
  - PROPORTION,  $D_I/D_B$  - TABULATED
  - BLADE TRAILING EDGE ANGLE,  $\beta_2$  - ASSUMED
  - LOCAL MATERIAL THICKNESS,  $T_{MM}$  - ASSUMED
  - NUMBER OF BLADES,  $N$  - ASSUMED

• FOR EXAMPLE:

WEIGHT OF BACKPLATE =  $KG \times T_{MM} \times F_w$

WEIGHT OF OUTER SHROUD =  $KG \times T_{MM} \times F_{w1}$

WEIGHT OF DIFFUSER RINGS (2) =  $KG \times T_{MM} \times F_{w1}$

WEIGHT OF INLET CYLINDER =  $KG \times T_{MM} \times F_{w1}$

WEIGHT OF BLADES W/O INDUCER =  $KG \times T_{MM} \times F_{w1} \times N$

WEIGHT OF INDUCERS =  $KG \times T_{MM} \times F_{w1} \times F_{w2} \times N$

WEIGHT OF HUB = KG (AS SHOWN)

THEN -

WEIGHT OF SWSI RD WHEEL WITHOUT SHAFT:

(WT., BACKPLATE) + (WT., OUTER SHROUD) +

(WT., DIFFUSER RINGS) + (WT., INDUCERS) + (WT., HUB)

MULTIPLY THIS WT. x 2 FOR DWDI WHEEL

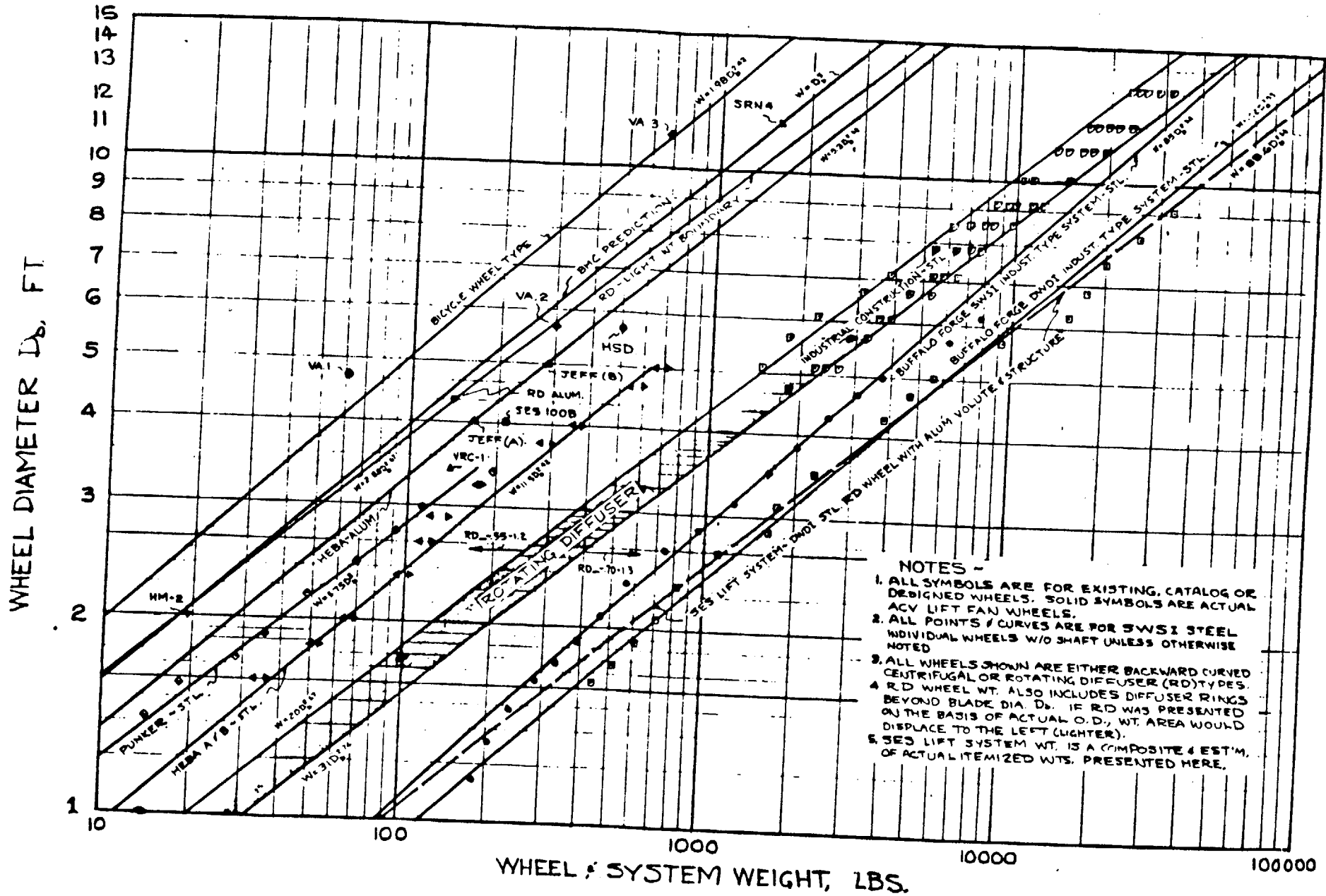
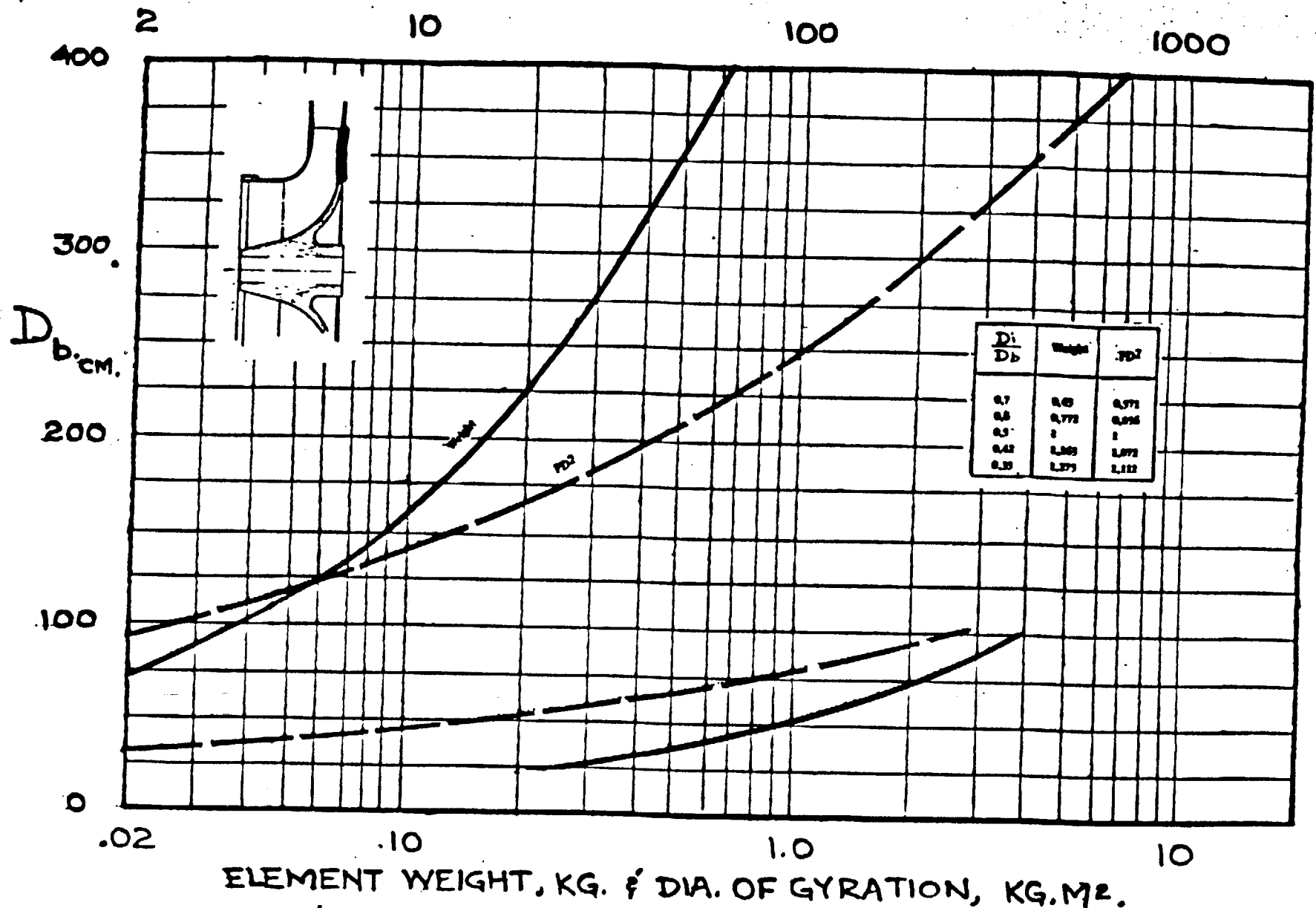


Fig. 5.4.1

- NOTES -**
1. ALL SYMBOLS ARE FOR EXISTING, CATALOG OR DESIGNED WHEELS. SOLID SYMBOLS ARE ACTUAL ACV LIFT FAN WHEELS.
  2. ALL POINTS / CURVES ARE FOR SWSI STEEL INDIVIDUAL WHEELS W/O SHAFT UNLESS OTHERWISE NOTED
  3. ALL WHEELS SHOWN ARE EITHER BACKWARD CURVED CENTRIFUGAL OR ROTATING DIFFUSER (RD) TYPES.
  4. RD WHEEL WT. ALSO INCLUDES DIFFUSER RINGS BEYOND BLADE DIA.  $D_b$ . IF RD WAS PRESENTED ON THE BASIS OF ACTUAL O.D., WT. AREA WOULD DISPLACE TO THE LEFT (LIGHTER).
  5. SES LIFT SYSTEM WT. IS A COMPOSITE & EST'M. OF ACTUAL ITEMIZED WTS. PRESENTED HERE.



ELEMENT WEIGHT, KG. & DIA. OF GYRATION, KG.M².

FIG. 5.4.2: BACKPLATE WEIGHT AND DIA. OF GYRATION.

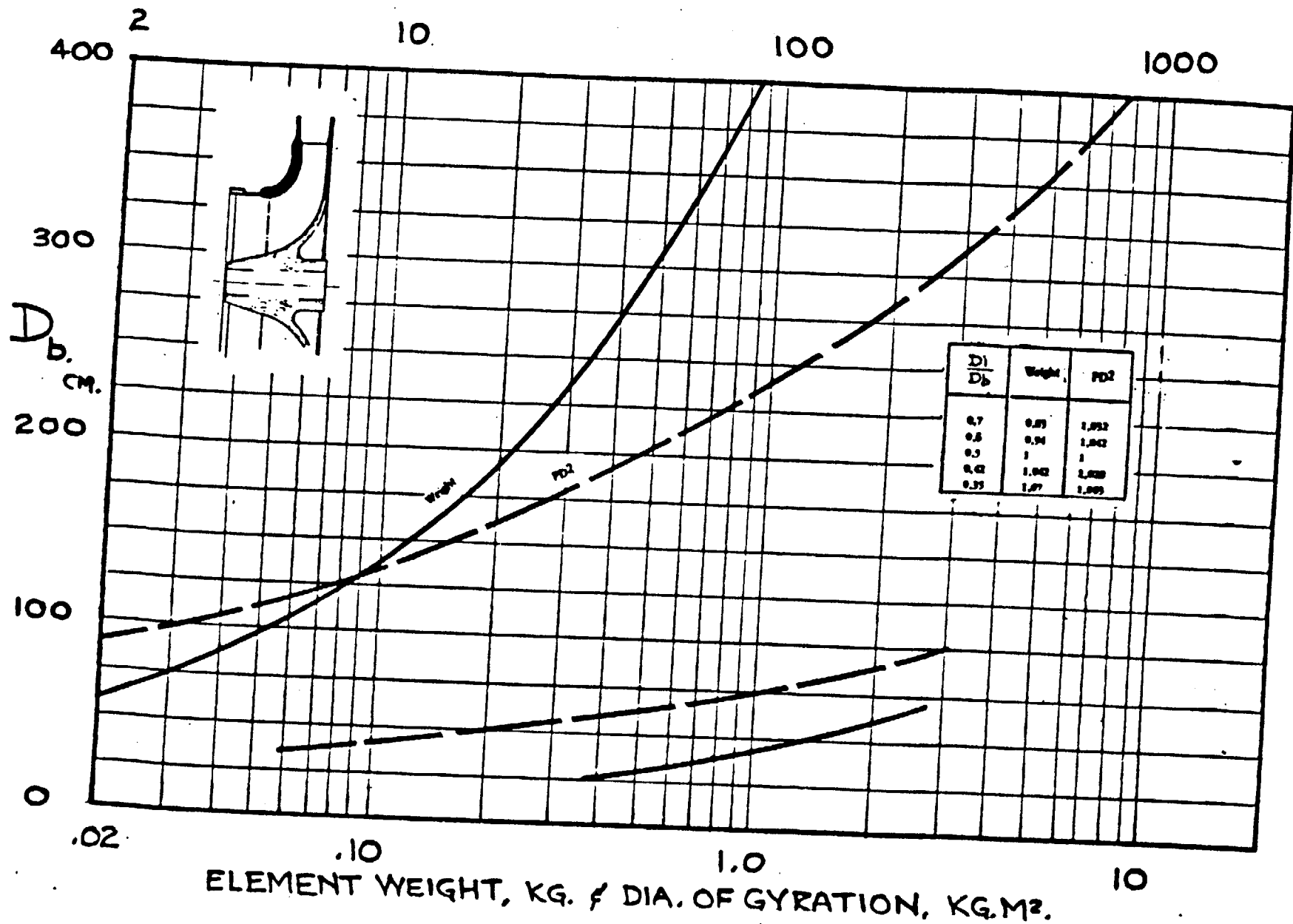
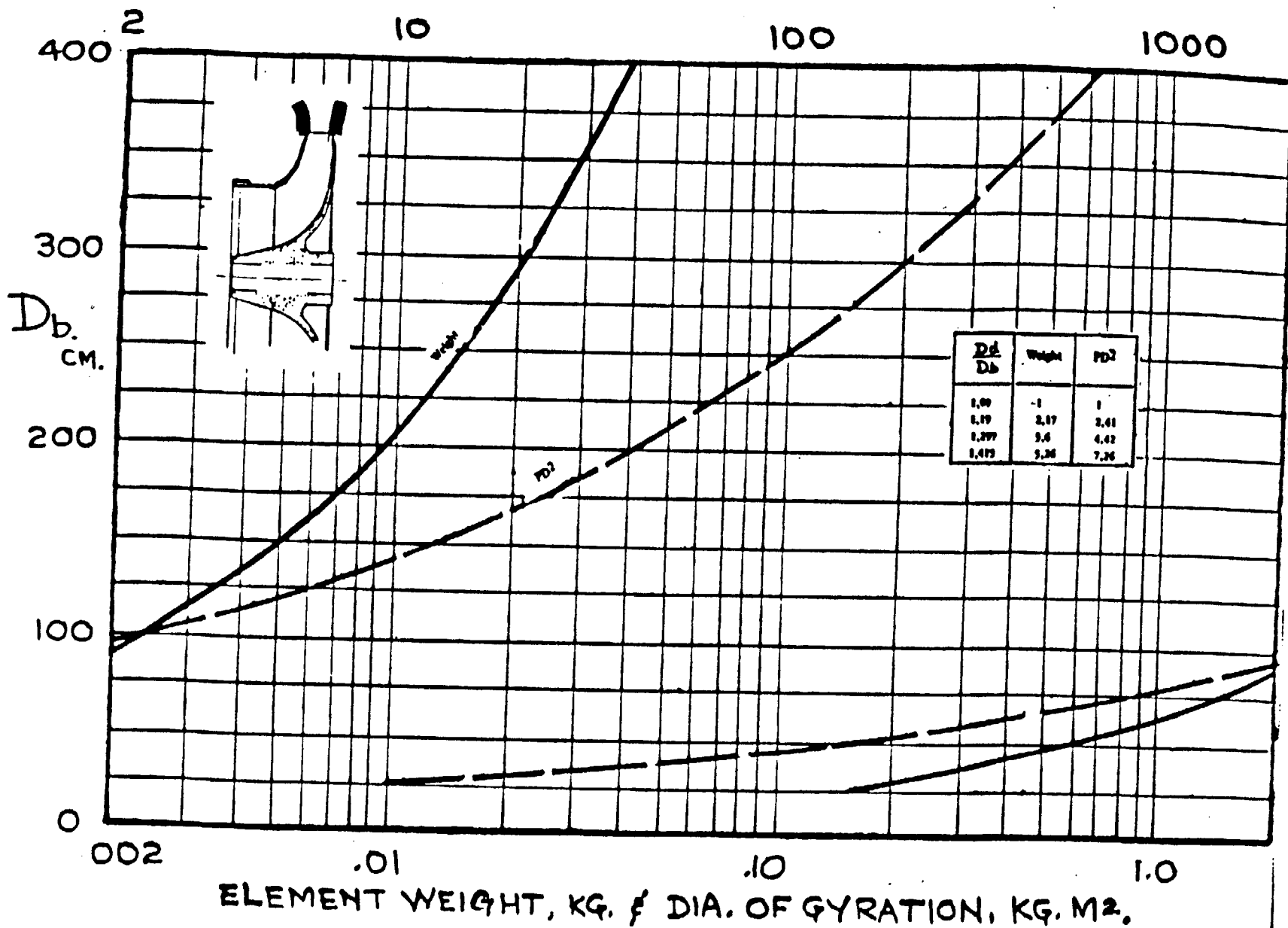


Fig. 5.4.3

FIG. 5.5.3: OUTERSHROUD WEIGHT & DIA. OF GYRATION.



ELEMENT WEIGHT, KG. & DIA. OF GYRATION, KG. M<sup>2</sup>.

FIG. 5.5.4: BOTH ROTATING DIFFUSER RINGS WEIGHT & DIA. OF GYRATION

Fig. 5.4.4

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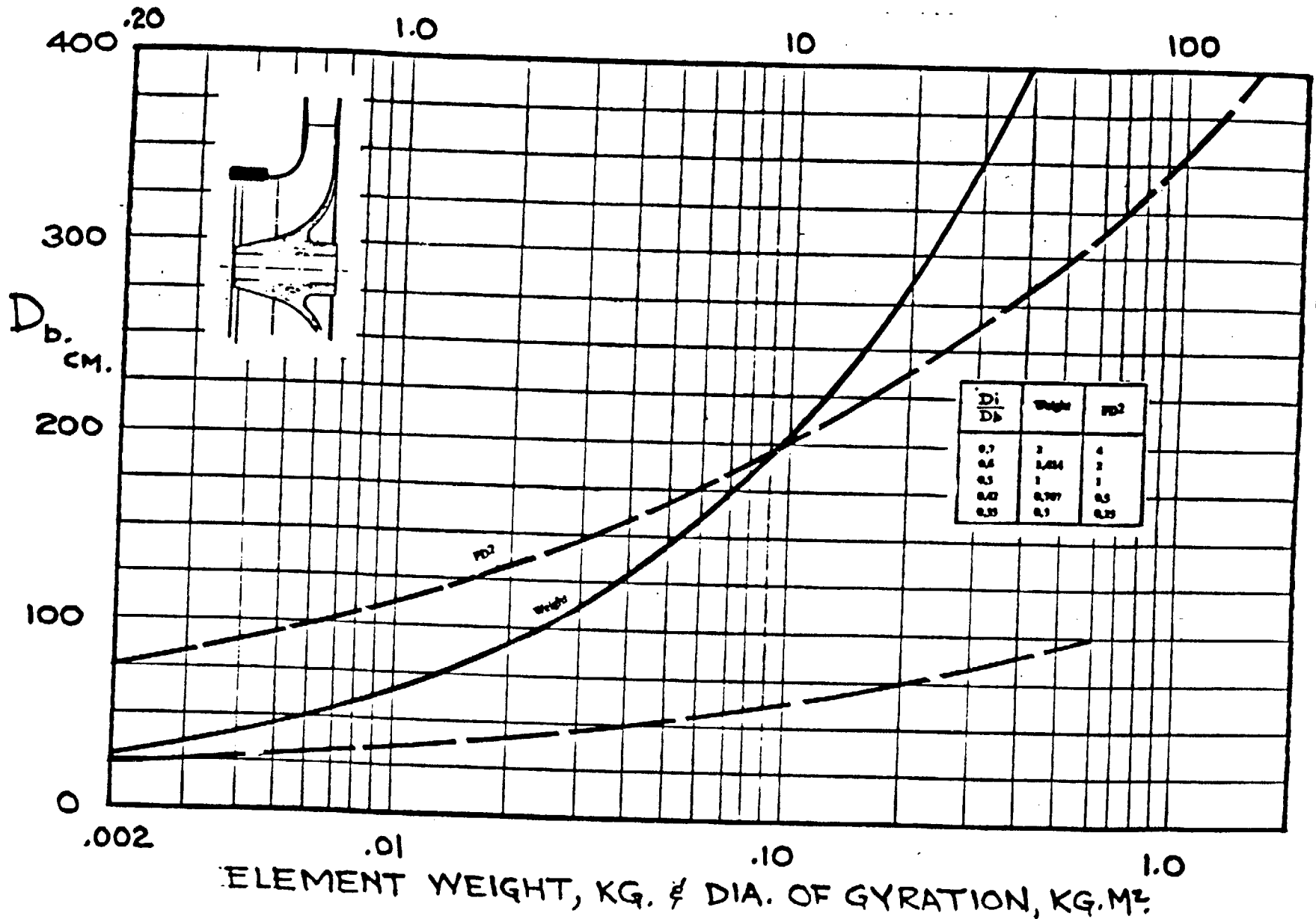


Fig. 5.4.5

ELEMENT WEIGHT, KG. & DIA. OF GYRATION, KG.M<sup>2</sup>.

FIG. 5.4.5: INLET CYLINDER WEIGHT & DIA. OF GYRATION.

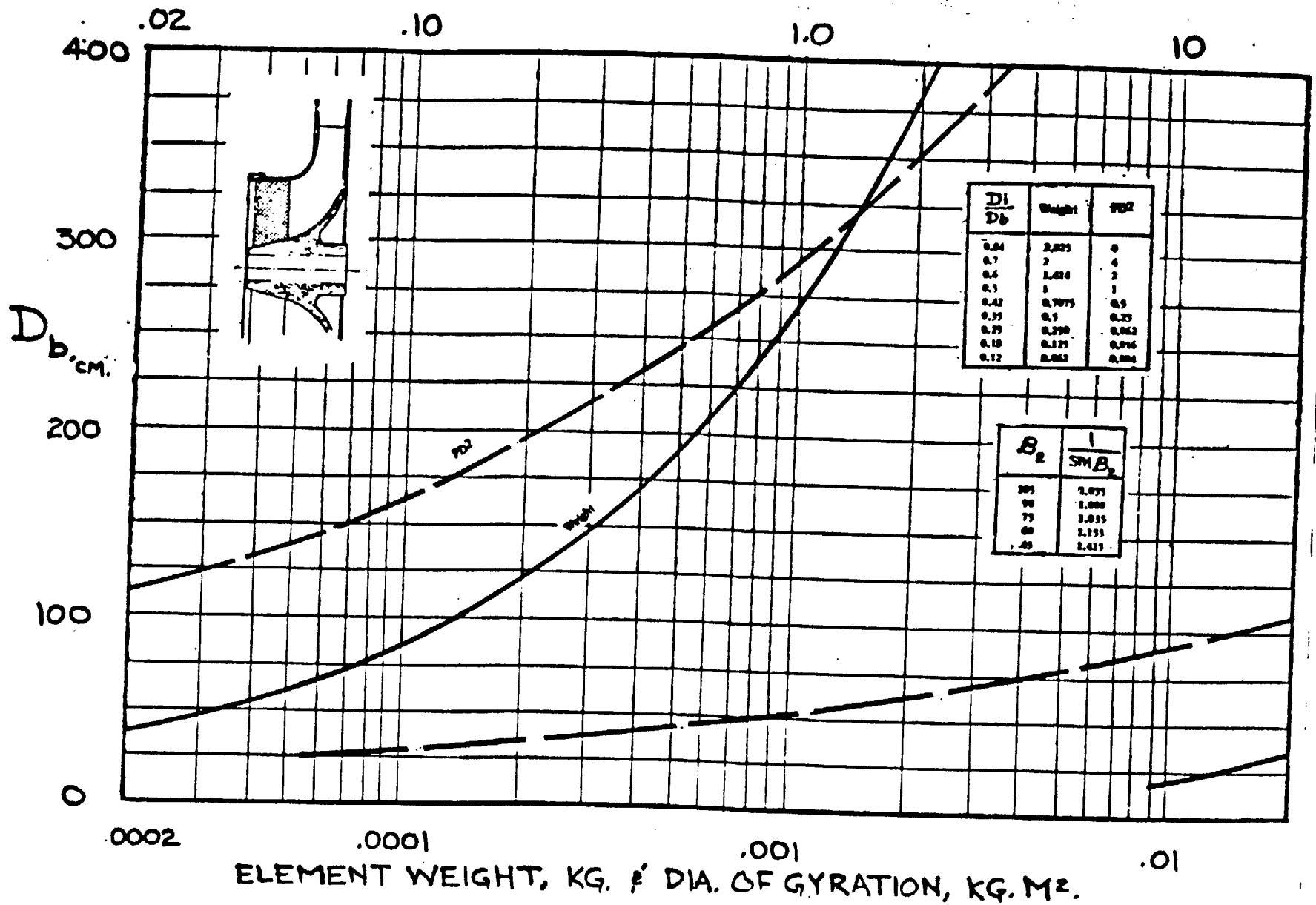
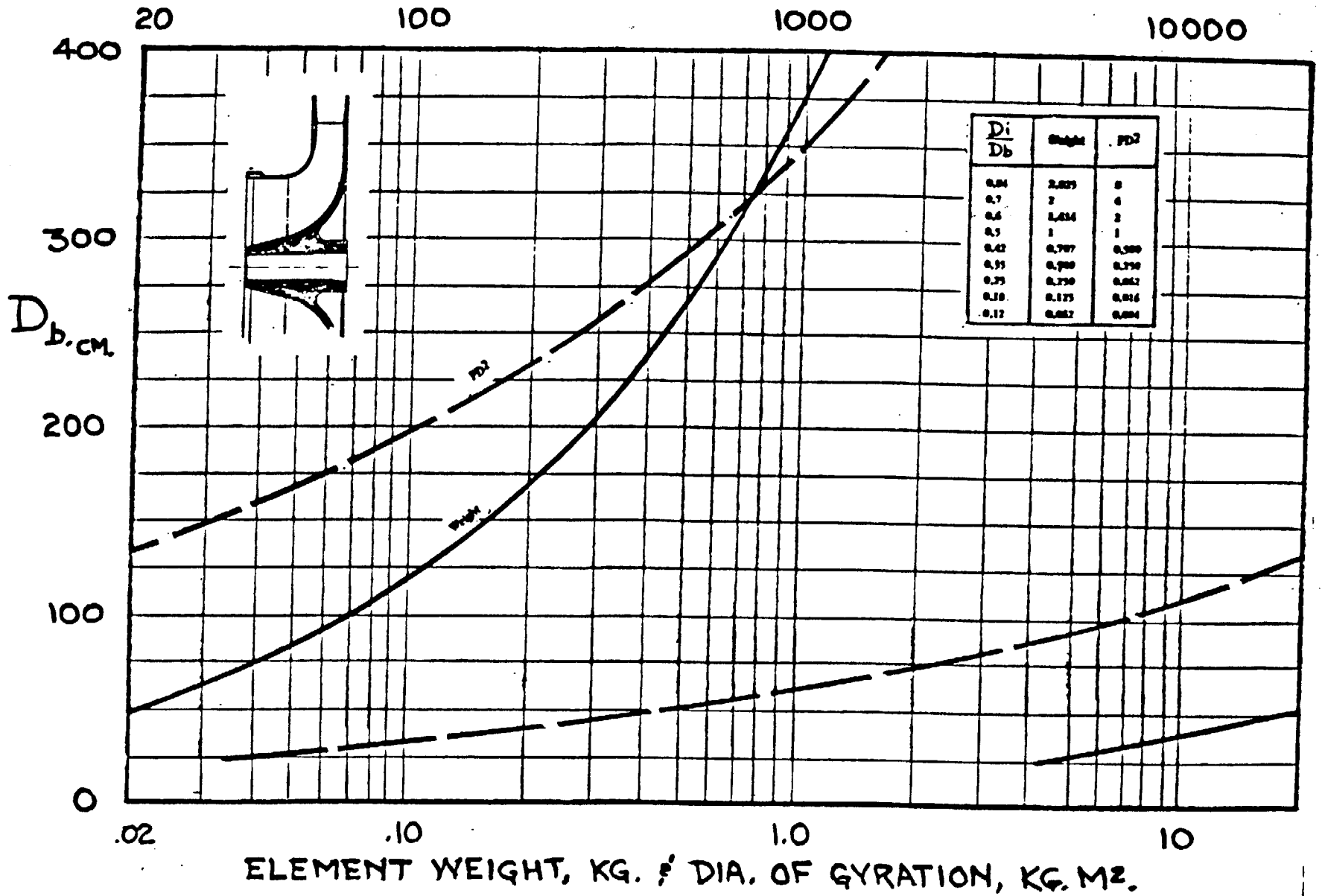


FIG. 5.5.6: INDUCER WEIGHT & DIA. OF GYRATION

Fig. 5.4.6



ELEMENT WEIGHT, KG. & DIA. OF GYRATION, KG. M<sup>2</sup>.

FIG. 5.4.7: HUB WEIGHT & DIA. OF GYRATION

Fig. 5.4.7

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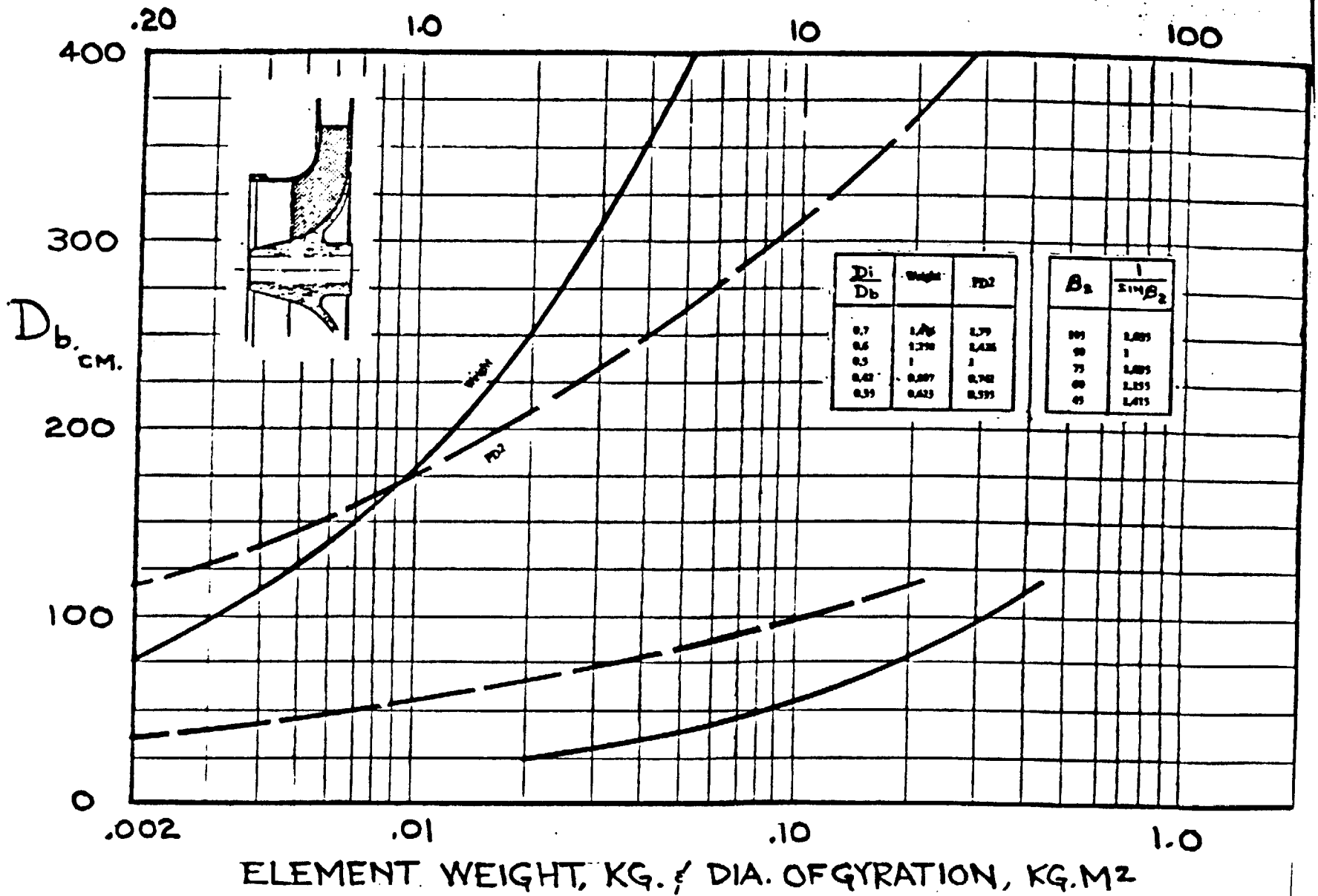


Fig. 5.4.8

FIG. 5.5.8: INDIVIDUAL BLADE (W/O INDUCER SECTION) WEIGHT & D.A. OF GYRATION

## 6. SELECTED APPLICATIONS OF THE RD FAN TO THE SES LIFT SYSTEM

### 6.1 IGV's, Dampers and Flow Control Means.

Earlier ACV's and SES have operated their lift fans without any means of lift air modulation, other than changes in shaft RPM. With the need for matching lift performance to varying load and cushion duties (static adjustment), and the need to actively and constantly keep a uniform cushion pressure over a varying wave-pumping sea (active ride control), the use of mechanical flow control has become a necessity.

Fan flow-pressure control is achieved by placing vanes, shutters, airfoils, registers or dampers just upstream or immediately downstream of a lift fan intake or exhaust, respectively.

A discussion by de la Combe of the basic airflow control of centrifugal fans is presented here in English translation from the original French version (Reference 12).

- Flow and pressure control of the RD fan

One must be able to vary the flow and the pressure of an RD that supplies air to the SES cushion, both statically and dynamically. There are five major ways to accomplish this objective:

1. Variable rotational speed of the fan

When the rotational speed of the fan varies, say from 1000 rpm to 900 rpm, as shown on Figure 6.1.1, the duty point of the fan moves along a parabola (from point  $A_1$  to point  $A_1'$  of Figure 6.1.1, for example). However, it is also possible to move the duty point  $A$  to  $A_2$  (constant flow delivery) or  $A_3$  (constant pressure).

Therefore, control using variable RPM can be very useful, but it requires a variable-speed drive or transmission.

2. Flow control using a damper

For a given airflow distribution network, the flow capacity can be changed through an "orifice" located anywhere in the network, for example, a damper on the discharge side of the fan, as shown on Figure 6.1.2. The damper can also be used as a one-way valve, to eliminate pressure surges into the fan from the cushion. Typical performance of the system is shown on Figure 6.1.3. The original duty point is shown as  $A$ , corresponding to a flow  $Q_A$ . If the flow is reduced by means of the damper to  $Q_B$ , this corresponds to the operating point  $B'$

of the operating curve. However, since the discharge circuit is not modified, the actual pressure corresponding to  $Q_B$  is  $p_B$ , on the same load curve as  $p_A$ . The new flow conditions correspond to  $(Q_B, p_B)$ . Note that the damper has "killed" the pressure  $p_B' - p_B$ . The efficiency of the fan at the new operating point B is:

$$\eta_B = \eta_{B'} \frac{p_B}{p_B'}$$

Therefore, the efficiency, using a damper to regulate flow, is fairly low.

### 3. Use of a by-pass

One can increase the capacity of a fan discharging into a given network by controllably by-passing some of the flow downstream of the fan, with possible return of the flow to the fan inlet. This means of control is rarely used on centrifugal fans.

### 4. Use of axial inlet guide vanes

In accordance with Euler's equation which characterizes the theoretical pressure rise through a centrifugal fan, the pressure-flow relationship of a fan can be significantly changed by introducing, through a set of inlet guide vanes (axial or radial), a "rerotation" of the flow.

A set of axial inlet guide vanes, shown schematically on Figure 6.1.4, consists of pie-shaped elements, usually closely fitted to each other when the vanes are closed, which each rotate about an axis perpendicular to the axis of rotation of the machine.

The simultaneous rotation of all vanes is usually achieved by a single control mechanism, as shown in figures 6.1.8 and 6.1.2.

The number of inlet guide vanes is usually a prime number, such as 7, 9 or 13, and is different from the number of blades of the fan, to avoid potential resonance. A typical performance map of a rotating diffuser fan equipped with inlet guide vanes is shown on Figure 6.1.5.

#### 5. Use of radial inlet guide vanes

A schematic of a typical radial guide vane setup is shown in Figure 6.1.5. The axis of rotation of the vanes is parallel to the axis of rotation of the machine, and the vanes are of rectangular shape and held on both sides, as shown. The vanes are located between two parallel walls, one corresponding with the inlet face of the fan and the other a circular fairing facing the inlet. Usually, the vanes are controlled by a single "collective pitch" mechanism.

Usually, one uses more vanes for radial inlet guide vanes than in the axial vane case. Since the vanes are physically farther from the fan inlet than in the axial case, their wake is somewhat damped when they reach the fan, and therefore risks of resonance are lessened. One usually employs an even number of vanes: 16, 18, 24, etc..

This type of inlet guide vanes always has a diameter much larger than that of the inlet eye of the fan or than that of the corresponding axial inlet guide vane system. It is more expensive and harder to close, but it usually has a lower pressure drop and is quieter than the axial vanes.

Both axial and radial vanes are used with RD fan. Usually, however, radial vanes are preferred.

An interesting variation in radial inlet guide vanes is shown in Figure 6.1.7 and 6.1.13, and referred to as the "3-vane damper caisson". Pre-rotation of the air entering the fan can be achieved by using a set of three collectively-linked parallel vanes mounted in a rectangular caisson. The effect of these vanes on fan performance is somewhere in-between

that of radial inlet vanes and of an ordinary damper. The advantage is lower cost and greater simplicity.

Recent model tests on the development of the RD wheel for SES lift systems apply the concepts of axial and radial IGVs. These models are described in Figures 6.1.8 - 6.1.12, and their flow and pressure performance are shown in Figures 6.1.15 and 6.1.14. This data is for steady state fixed IGV performance.

The use of the direct duct-right angle inlet caisson, with IGVs, is of very practical interest. This interest originates from the facts that the right angle caisson with IGVs:

- . is simpler in construction, requiring only a few (3-6) vanes, simply mounted and activated.
- . in the case of the DWDI system, allows the two shaft bearing centers to be brought closer to the wheel.
- . permits direct-ducted air delivery to the fan inlet and results in a smaller system installation.

Such an arrangement is often used in stationary industrial practice, and it allows for a continuous throughput of a process flow system (see Figure 6.1.15). Typical full-size static IGV performance data for a related industrial DWDI RD caisson system is shown in Figure 6.1.16.

Industrial and general research data on IGV applications to centrifugal fans is voluminous. Additional mechanical and aerodynamic schemes for flow modulation by vanes, dampers and jets is to be found in References 13, 14 and 15. Several of the described schemes appear mechanically simple and adaptable to SES lift control, but little test information is available on those arrangements, other than on the well-established axial, radial, toroidal IGV and general duct damper systems.

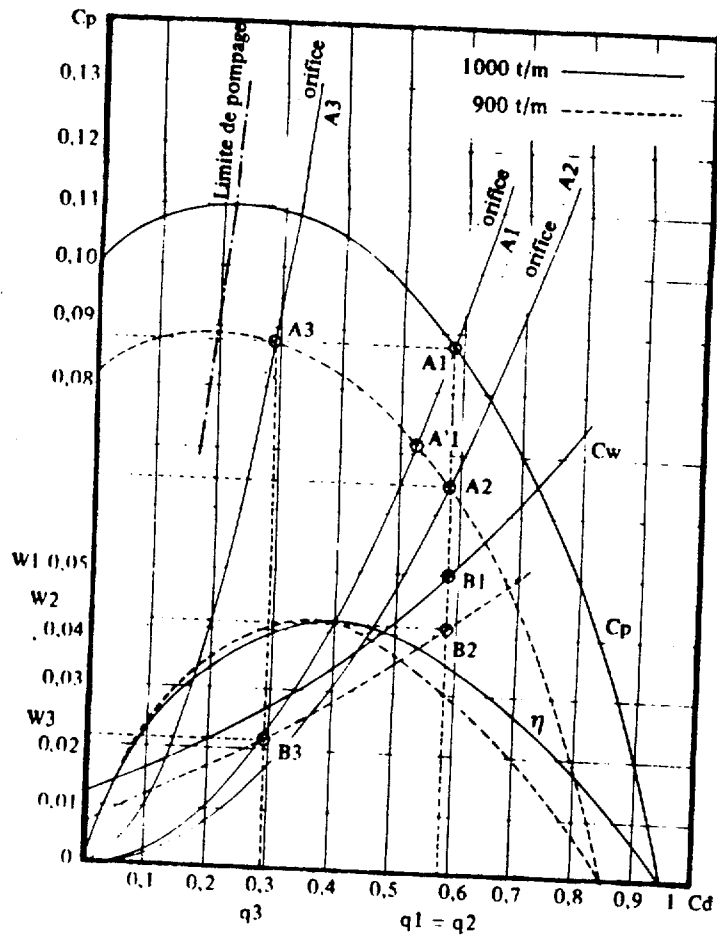


FIGURE 6.1.1. CHANGE IN CENTRIFUGAL FAN PERFORMANCE WITH CHANGE IN SHAFT RPM. NOTE CONSTANT PRESSURE UP TO THE SURGE LIMIT

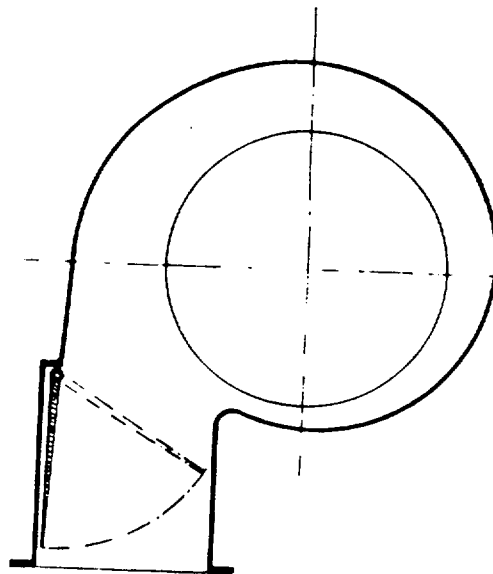


FIGURE 6.1.2. WHEEL AND VOLUTE WITH DAMPER VALVE IN EXHAUST.

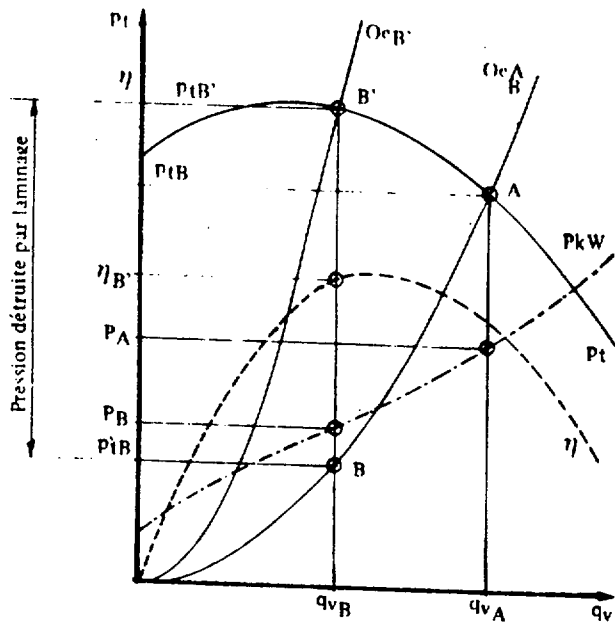


FIGURE 6.1.3. EFFECT DAMPER VALVE OF PRESSURE FLOW PERFORMANCE.

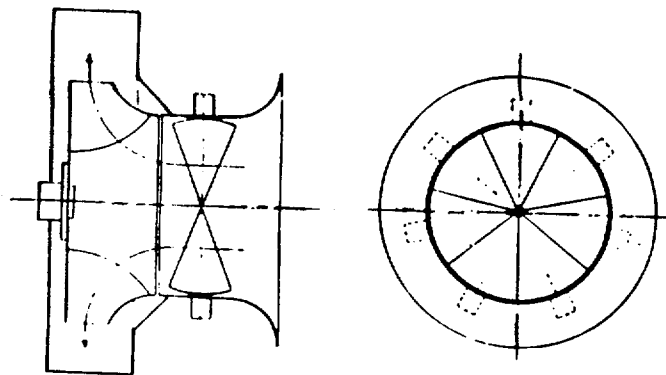


FIGURE 6.1.4. CENTRIFUGAL WHEEL WITH AXIAL INLET GUIDE VANES.



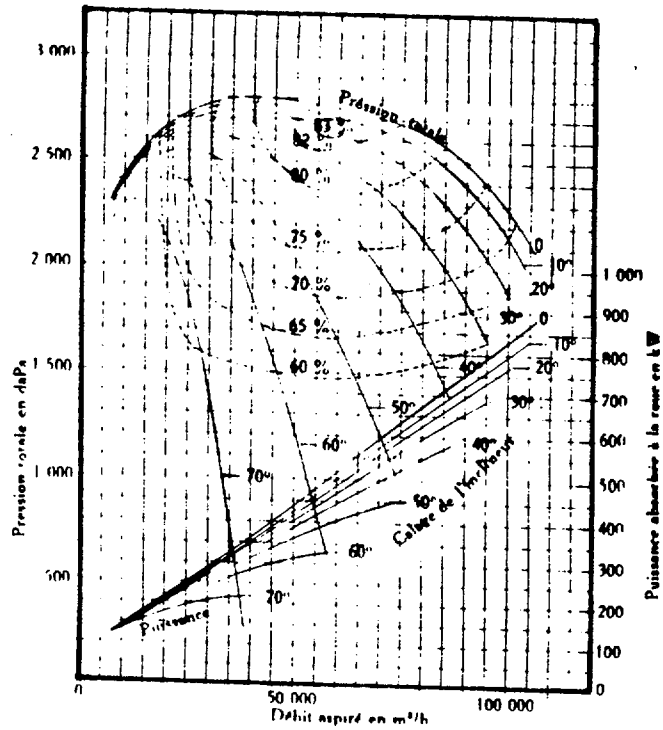


FIGURE 6.1.5. PERFORMANCE OF A RD FAN SYSTEM WITH AXIAL IGV'S - 2980 RPM.

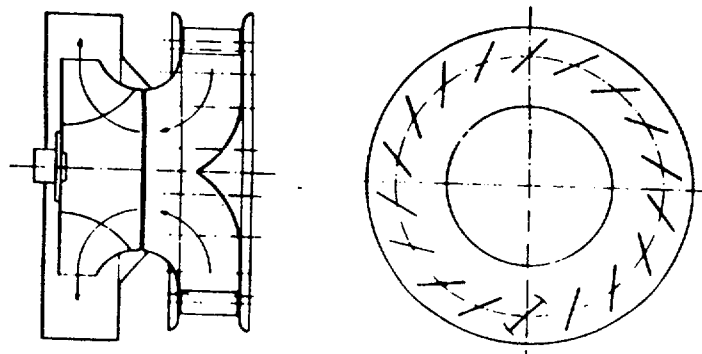


FIGURE 6.1.6. CENTRIFUGAL WHEEL WITH RADIAL INLET GUIDE VANES.

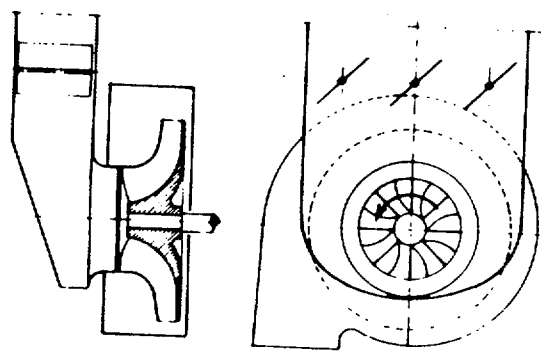


FIGURE 6.1.7. CENTRIFUGAL WHEEL WITH CONTROLABLE THREE INLET VANE RIGHT ANGLE CAISSON.

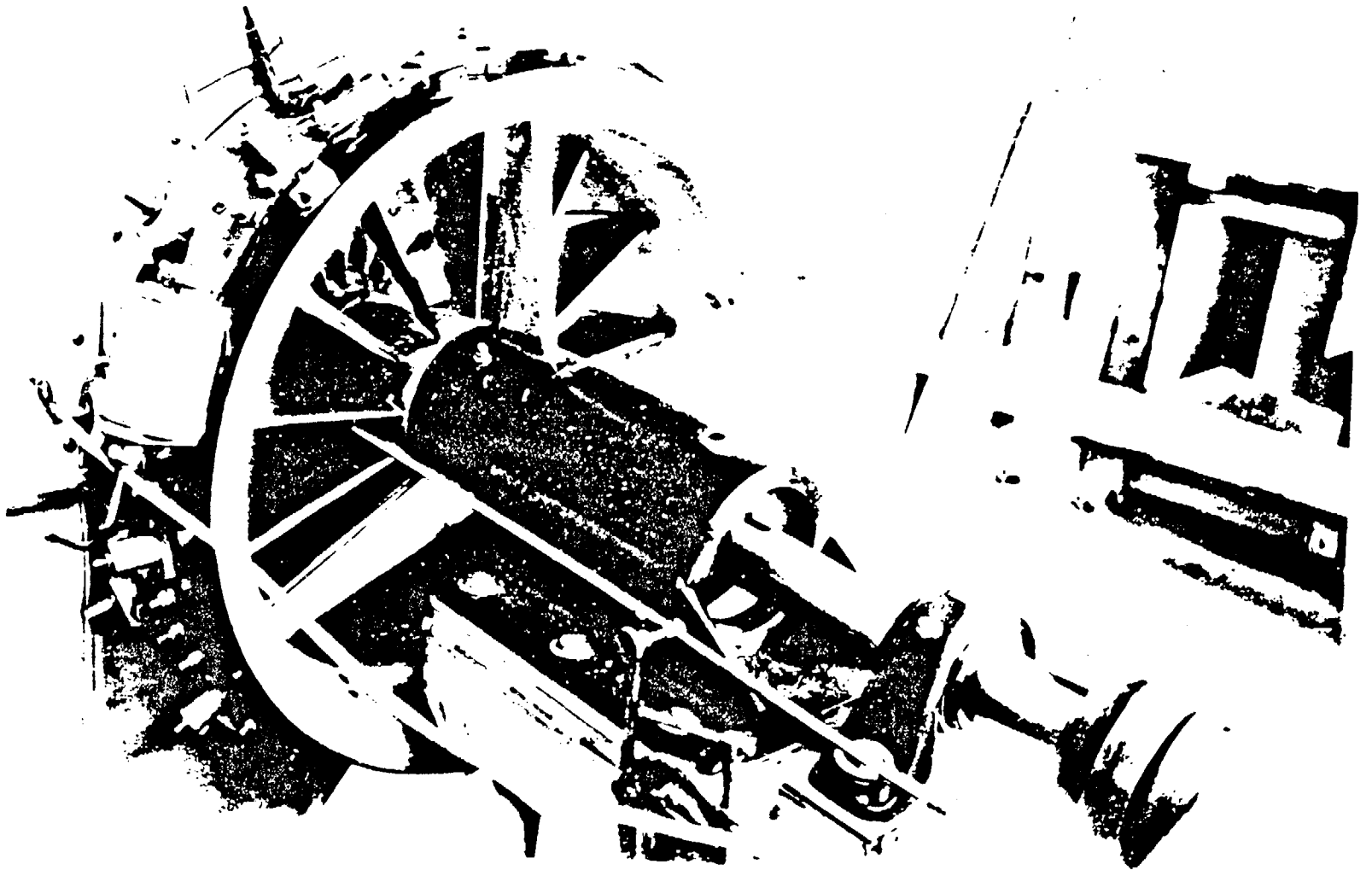


Figure 6.1.8.

Figure 6.1.9

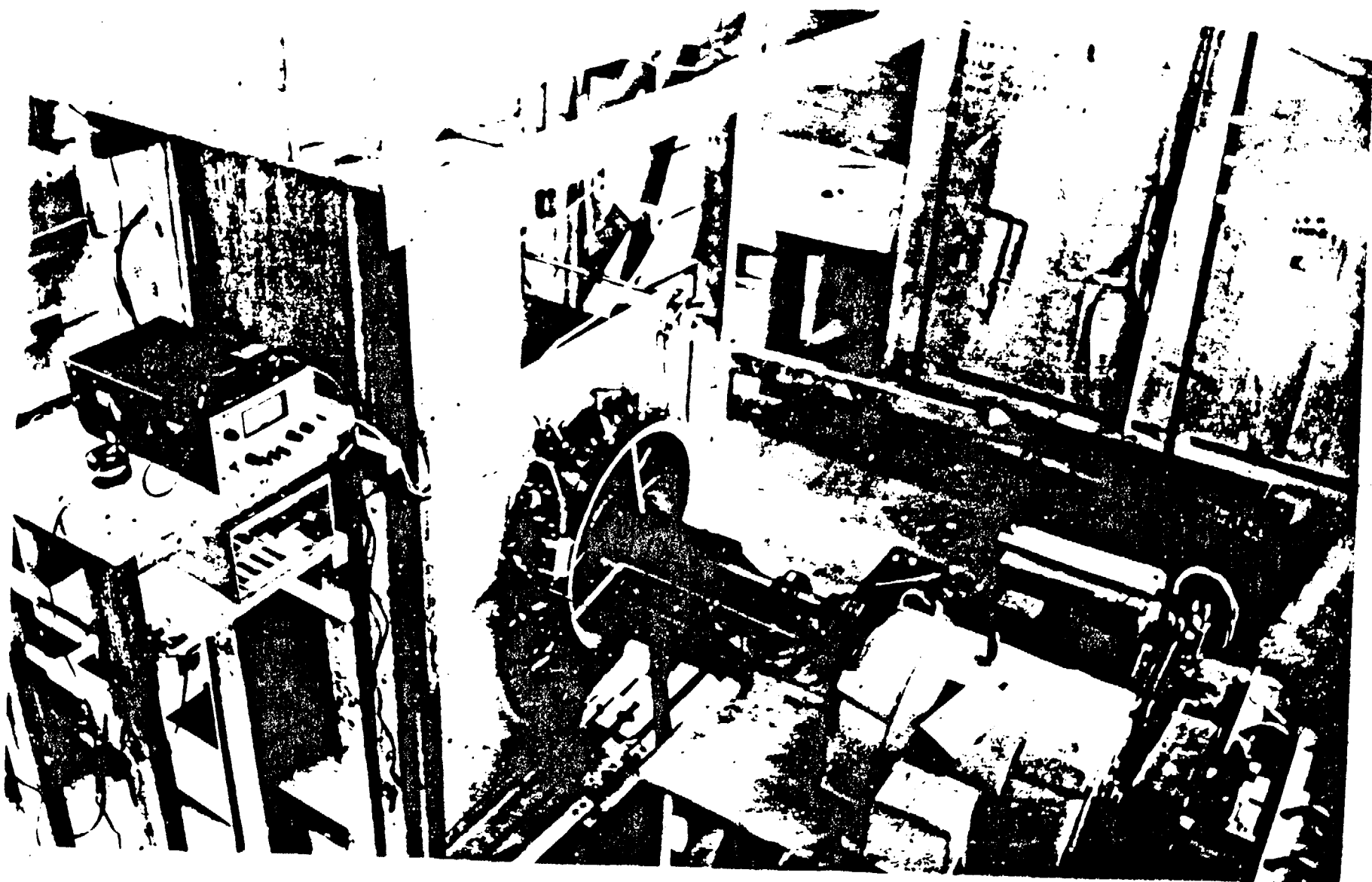
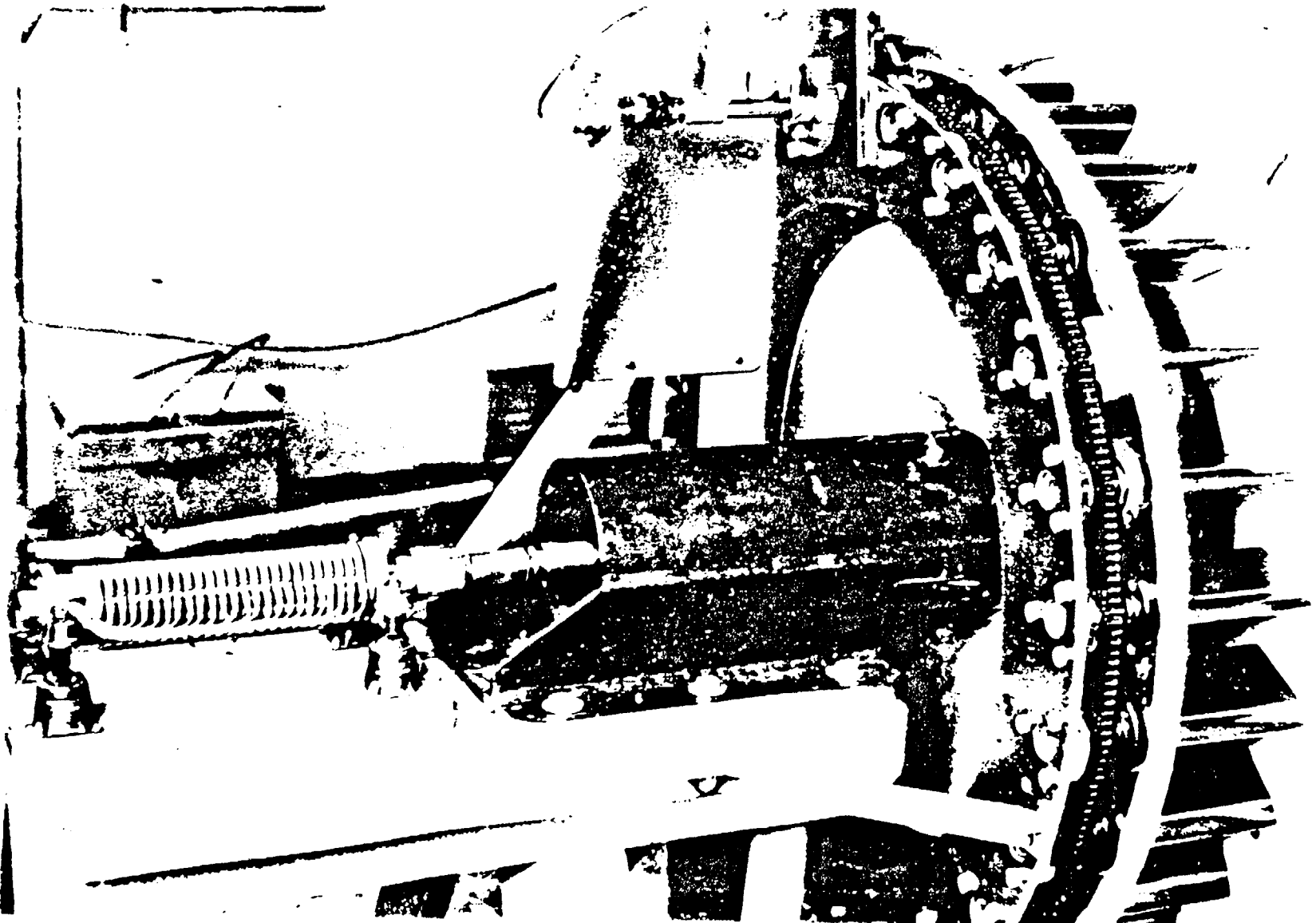
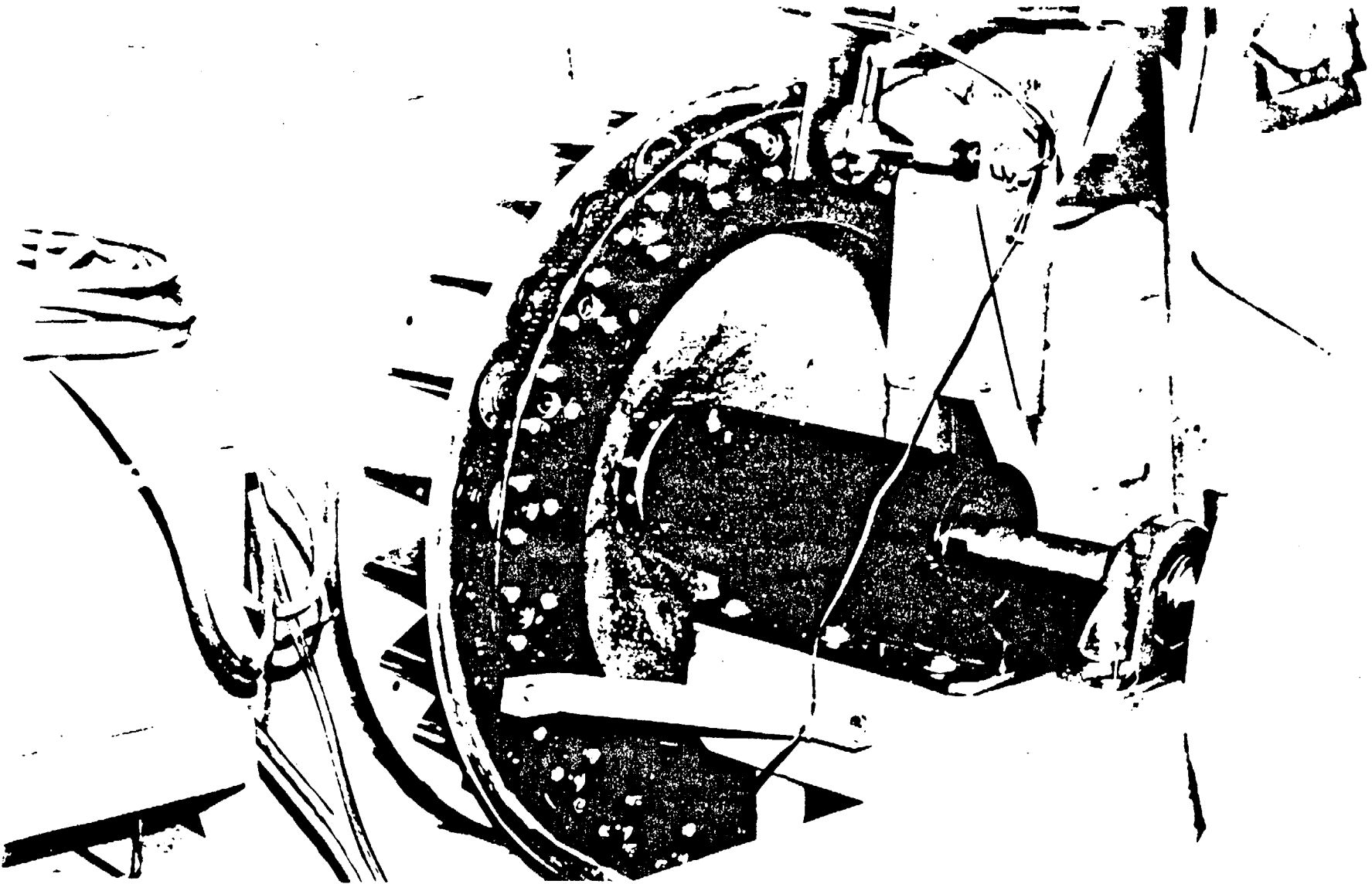


Figure 6.1.10.



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Fig. 6.1.11.



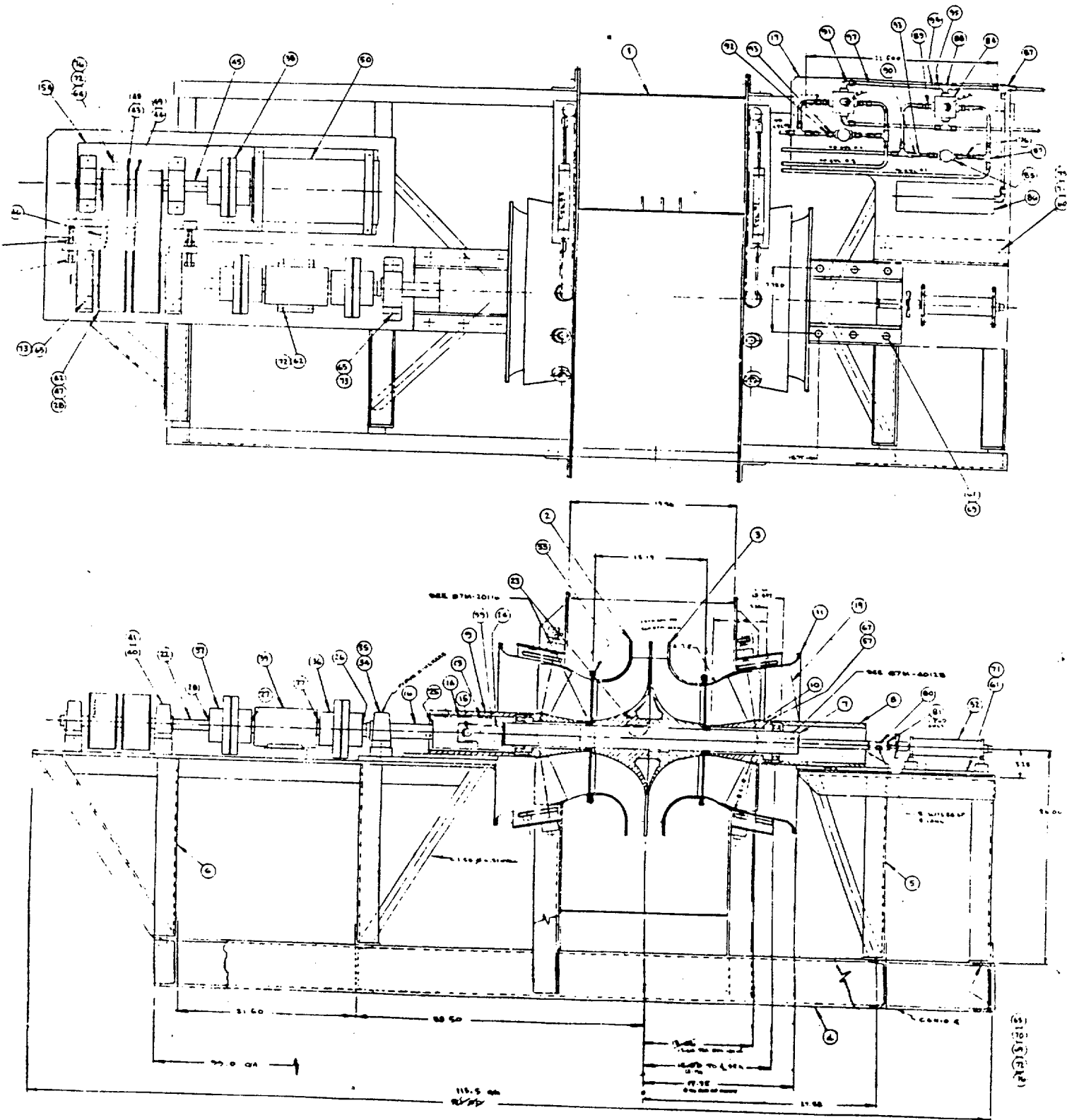
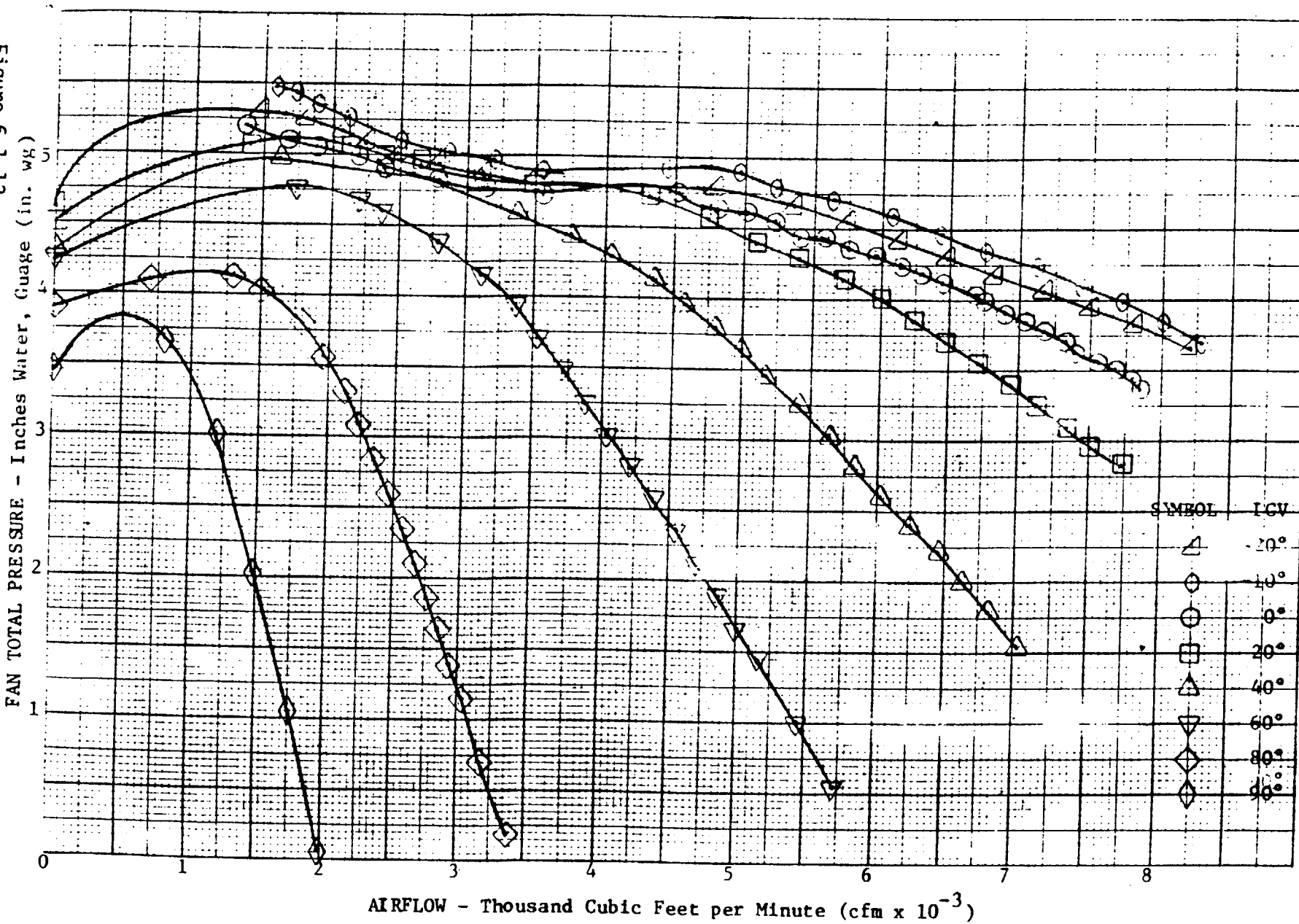


Figure 6.1.12.

STATIC TEST RESULTS - RD 50-.65-1.3-70° BK SES MODEL LIFT FAN WITH AXIAL FLOW INLET GUIDE VANES

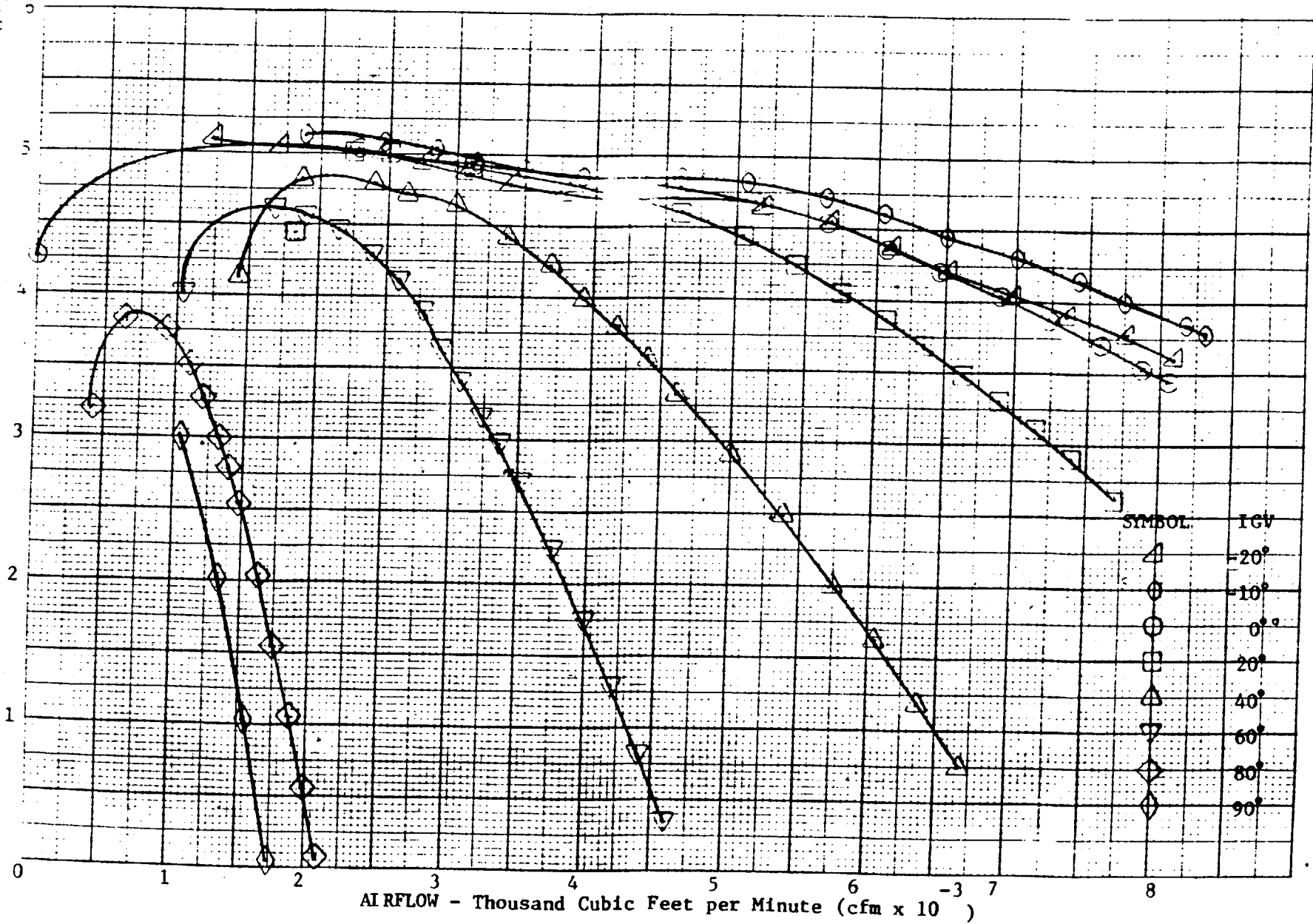
Figure 6.1.13.



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STATIC TEST RESULTS - RD 50-.55-1.3-70° KSES MODEL LIFT FAN WITH RADIAL FLOW INLET GUIDE VANES

Figure 6.3.14. FAN TOTAL PRESSURE - Inches Water, Gauge (In. wg)



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14



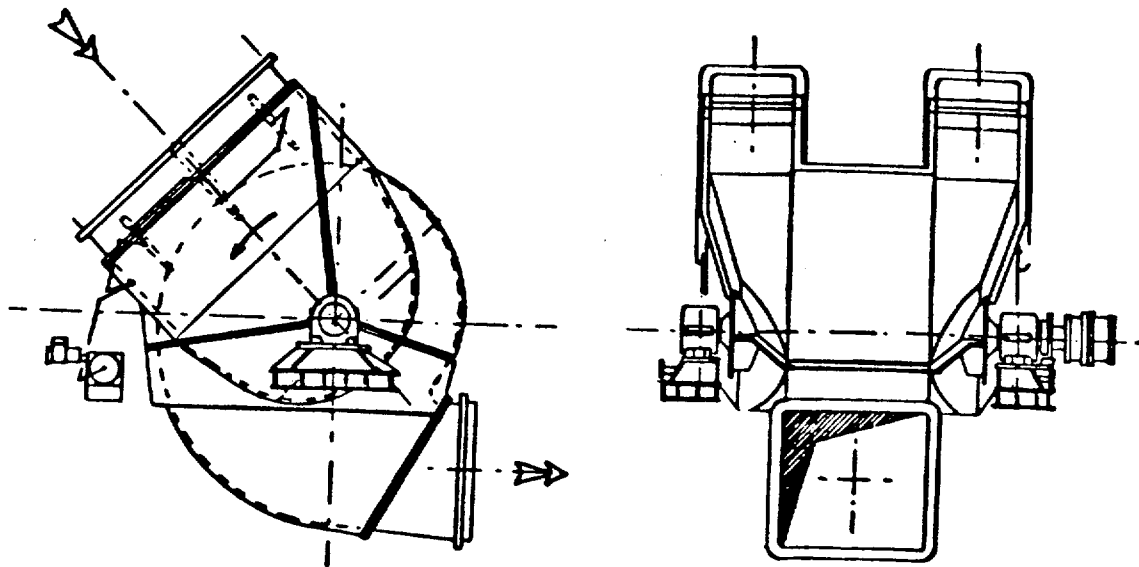


FIGURE 6.1.15. INDUSTRIAL TYPE DWDI RD SYSTEM WITH CONTROLLABLE THREE VANE INLET DAMPER IN CONTINUOUS PROCESS CLOSED DUCT ARRANGEMENT.

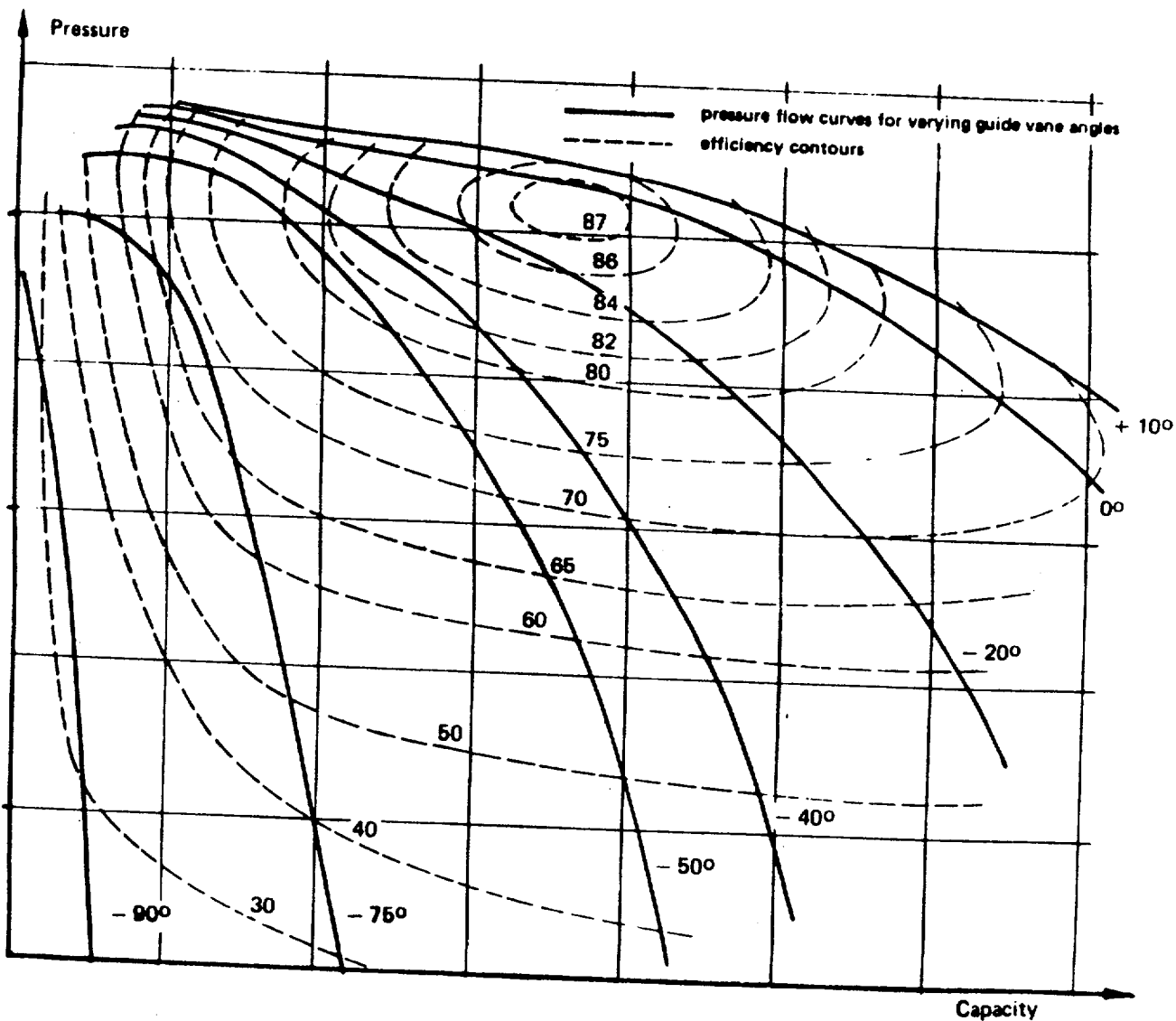
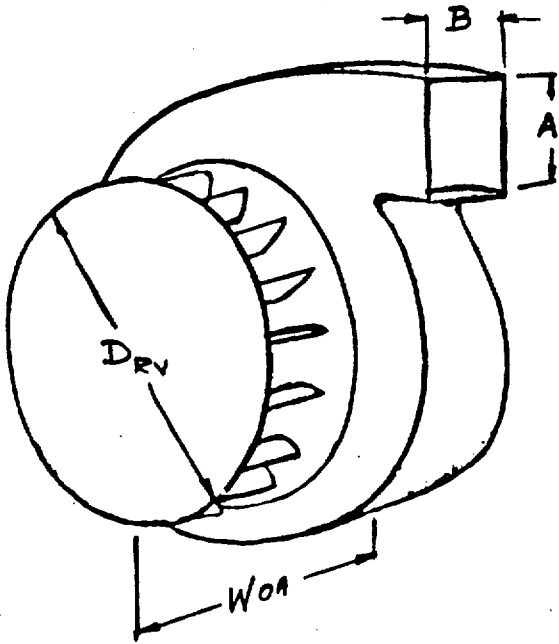


FIGURE 6.1.16. REPRESENTATIVE PRESS-FLOW MAP OF SIMILAR RD FAN SYSTEM WITH STATIC CHANGES IN IGV ANGLE.

## 6.2 Volute Design and Geometry

The RD fan operates in a standard industrial-type rectangular scroll volute. The volute continues the static pressure recovery of the wheel's rotating diffuser. The volute also serves to collect and exhaust the fan flow through a rectangular discharge opening.

For preliminary design, it is of interest to quickly determine the size and volume of RD volutes. A series of designer's curves are presented with which one may determine maximum dimensions of the volute as a function of the RD wheel diameter and diffuser ratio. (See Figures 6.2.1-6.2.4). It is important to note that the wheel diameter is presented in its characteristic metric dimension, centimeters; however, the linear dimensions of the volute are presented in inches. The diffuser ratio ( $P_R$ ), is the ratio of the wheel outer blade diameter to that of the outer diffuser (total wheel) diameter. These volute dimensions have been determined from aerodynamic tests and satisfy a broad range of general applications. Other test developments allow the basic volute size to be reduced ("tight volute") in order to fit a more limited space. Note that the curves may include the dimensions of a radial-type inlet guide vane assembly. Further note that the general volute axial length (B) is given to match the installation of a SWSI RD wheel. The axial length of the volute for a DWDI wheel now becomes 2B.



**BASIC RD VOLUTE DIMENSIONS**

W<sub>OA</sub>, WIDTH OVERALL, IN.  
 B, VOLUTE WIDTH FOR SISW WHEEL INSTALL. FOR DWDI WHEEL, DOUBLE B & W<sub>OA</sub> DIMENSION.  
 A, VOLUTE EXHAUST HEIGHT, IN.  
 D<sub>RV</sub>, O.D. RADIAL IGV INLET, IN.  
 REF. NEU  $\frac{N350}{3}$  & 59N85.

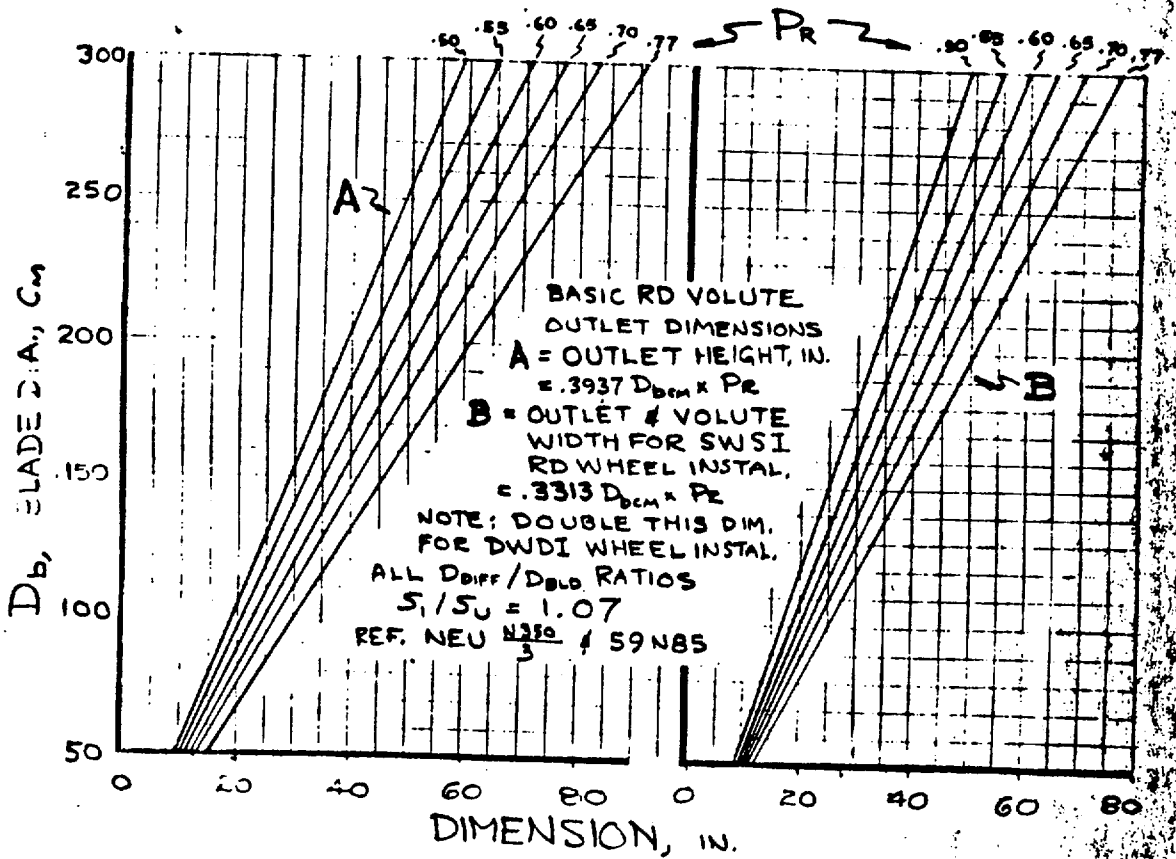


Figure 6.2.1

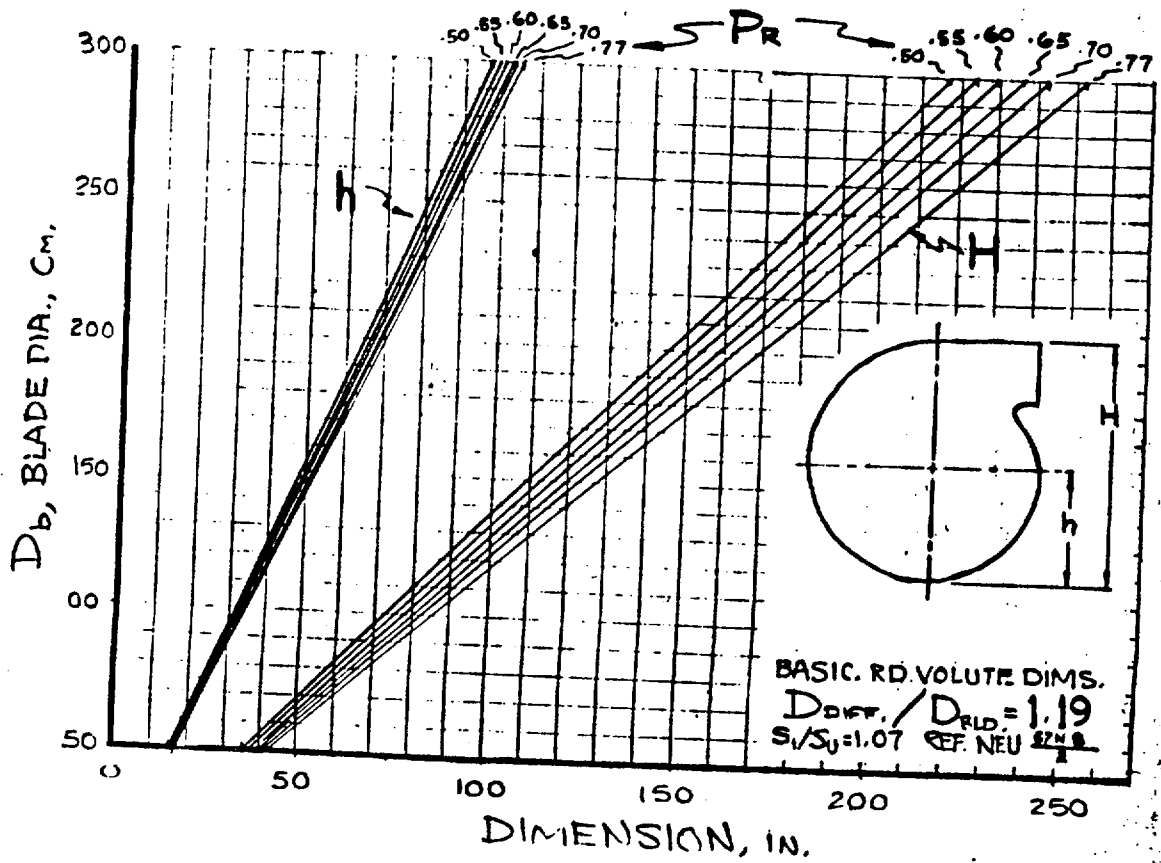
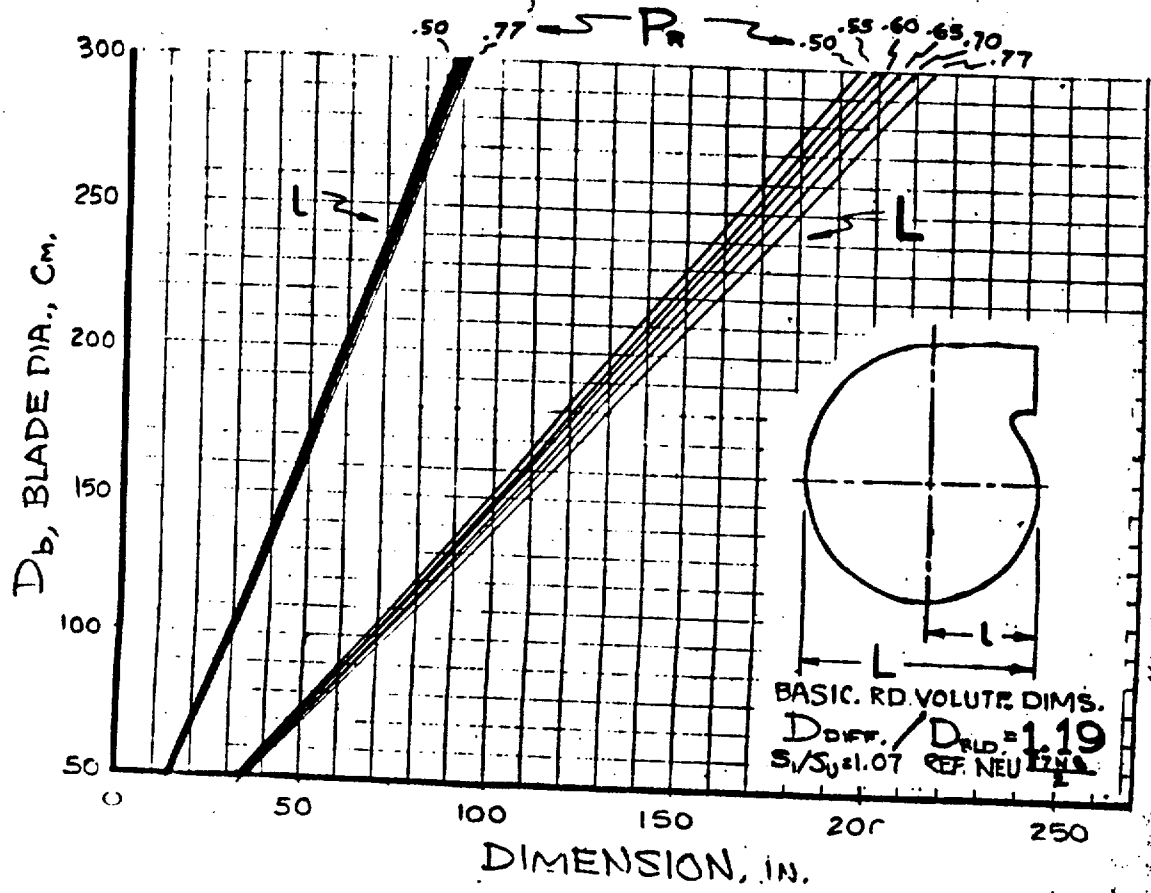


Figure 6.2.2

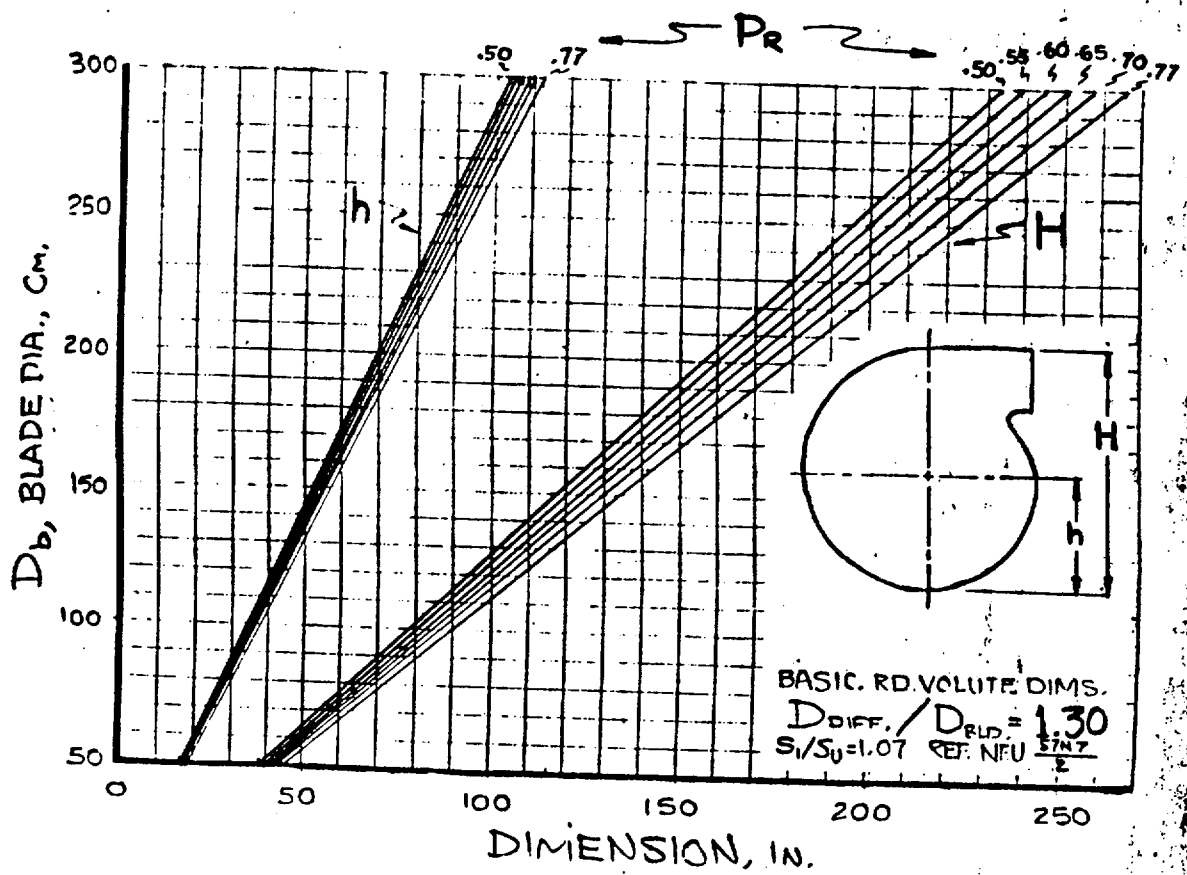
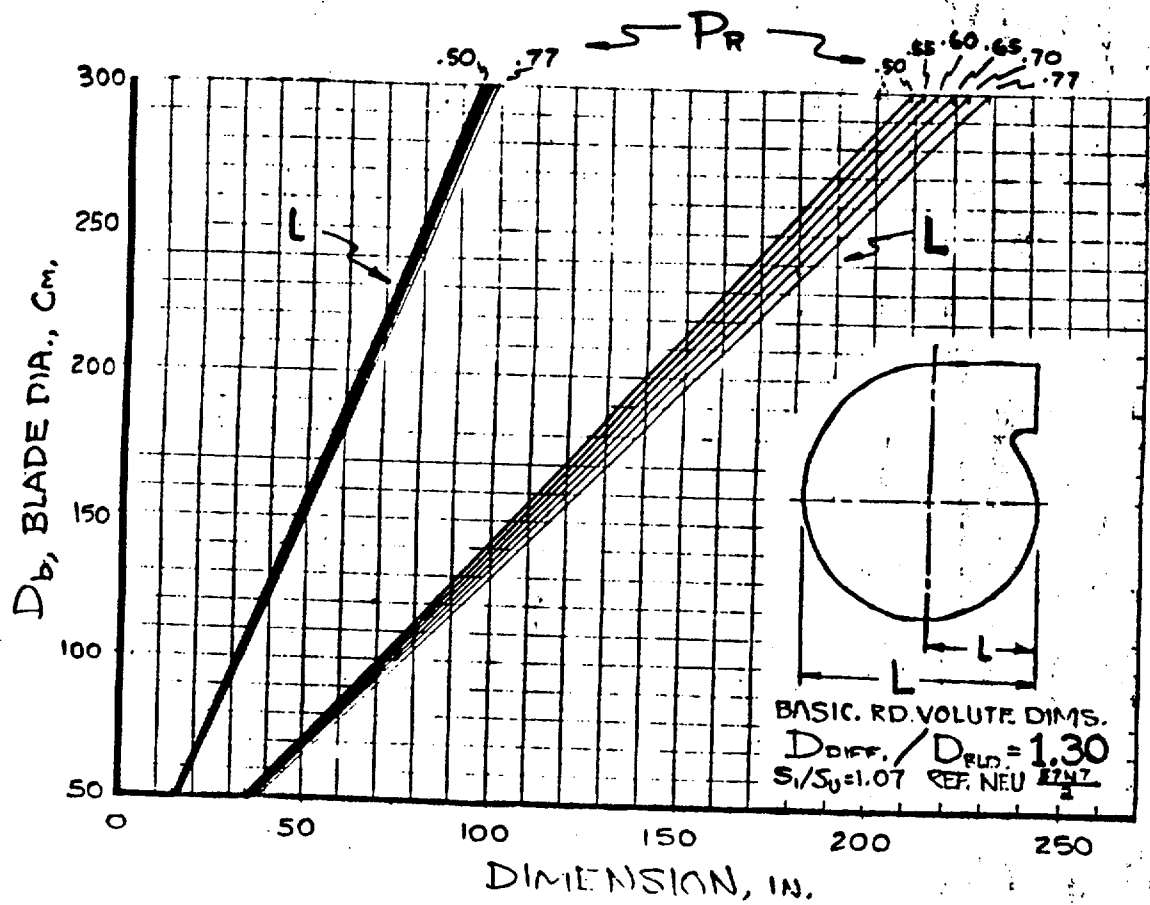


Figure 6.2.3

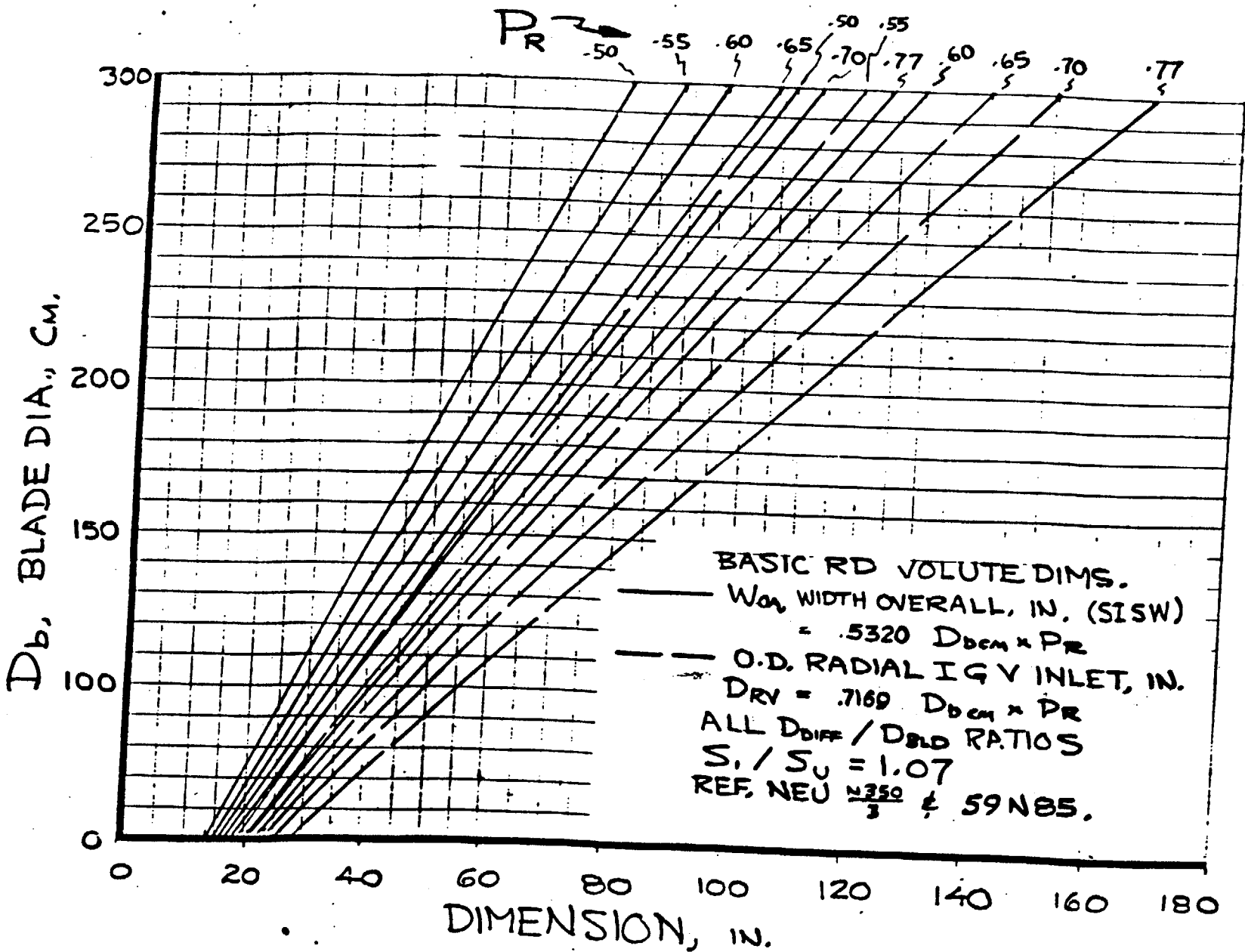


Figure 6.2.4

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### 6.3 Some Representative RD Fan SES Design Installation Studies

The practical application of the RD fan to high cushion pressure SES is presented here in three representative examples. These ships have the following cushion static pressure and powering requirements:

SHIP TYPE	FIG.	CUSHION PRESSURE, PSF, STATIC	TOTAL FLOW PER SHIP, CFS	NO. OF LIFT MODULES PER SHIP	HP AVAIL. PER ENGINE
900 LT		216	6168	6	620
280 MDC		216	6168	3	2200
MPS		514 700	35815 26290	6	7000

One of the points to be considered here is the modular lift fan engine drive unit and its flexible design application to SES marine architecture. These studies reflect several combinations and adaptations of realistic hardware selection and performance possibilities. Note, for example, how the MPS ship can be run at two cushion loadings only by the static adjustment of the IGV inlet angle.

The lift fan module would have very desirable ship-board features, such as excellent overhaul service accessibility, commonality of design, installation and ship control. Note that the units described are Diesel powered and that the larger MPS module breaks down further into two sub-units containing the DWDI lift fans.

The pressure and flow characteristics of these particular units are directly derived from actual full size wheels and volutes installations. Ship and module characteristics are tabulated as shown. (See tables and views in Figures 6.3.1-6.3.11).

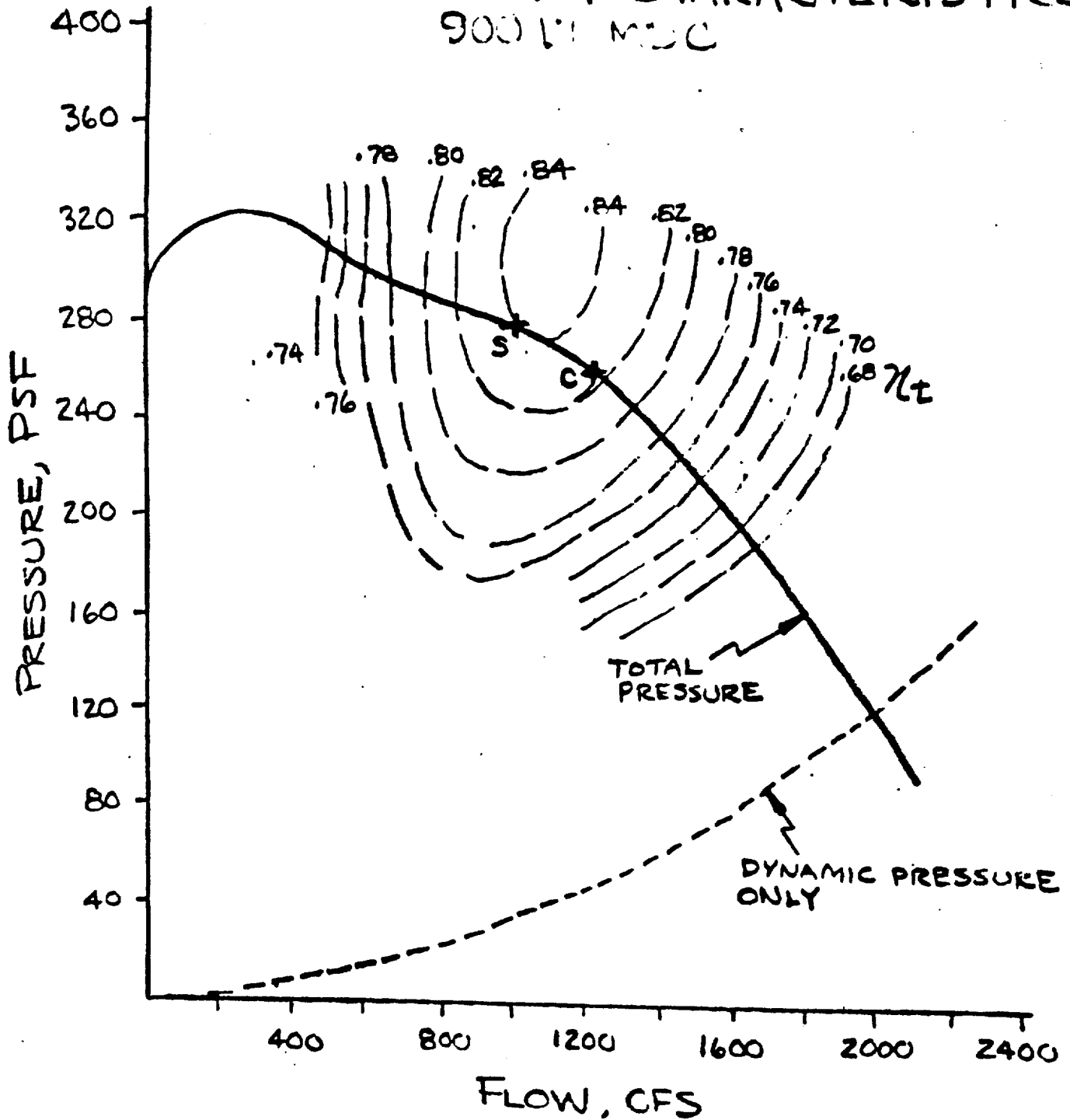
# LIFT SYSTEM PERFORMANCE-

PROJECT- 900 LT MDC.							DATE-12-1-80	
WHEEL TYPE-RD140-60-1.2-65°					COND. $\rho = .0749 \text{ LB/FT}^3$			
ENGINE-CUMMINS DIESEL VTA-1700-P, 620 HP AT 1800 RPM.								
RIDE CONTROL- OFF, DIRECT								
SUB-SYSTEM →					CUSHION		SEALS	
NUMBER OF SWSI LIFT FANS					2		4	
NUMBER OF INLETS					2		4	
HORSEPOWER AVAILABLE PER INLET					620		620	
REQ'D.	$P_s$ STATIC, PSF.				216		238	
	$P_T$ TOTAL, PSF.				256		282	
	MINIMUM FLOW PER INLET, CFS.				1100		992	
	MINIMUM FLOW PER SHIP, CFS.				6168			
AVAILABLE	$P_s$ STATIC, PSF.				216		238	
	$P_T$ TOTAL, PSF.				256		282	
	DELIVERED FLOW PER INLET, CFS.				1100		992	
	TOTAL FLOW IN CUSHION, CFS.				2200		—	
	TOTAL FLOW IN BOTH SEALS, CFS.				—		3968	
	TOTAL FLOW PER SHIP, CFS.				6168			
DUTY POINT	$C_{pT}$ TOTAL PRESSURE COEFFICIENT				.0726		.0740	
	$C_d$ FLOW COEFFICIENT/INLET				.432		.389	
	$\eta_z$ TOTAL EFFICIENCY				.825		.840	
	IGV BLADE SETTING, °				0		0	
	SLOPE OF PERFORMANCE CURVE				STABLE		STABLE	
TIP SPEED OF WHEEL; M/S, FT/SEC.					131.31 / 430.80			
RPM ENGINE - FAN					1-1			
GEAR RATIO REQUIRED					NO GEAR REQ'D.			
STD. VOLUTE DIMS. PER WHEEL INSTL. INCHES	H	L	$W_{0A}$	A	B	h	L	DRY
	109	96	45	33	28	47	41	60

Fig. 6.3.1.



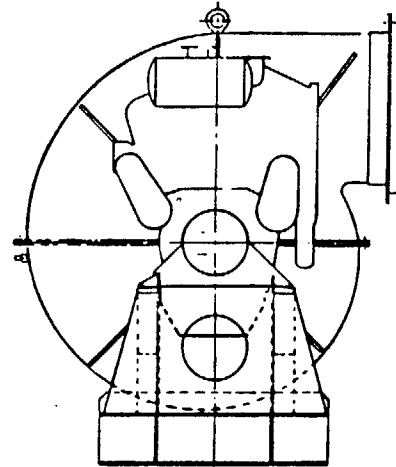
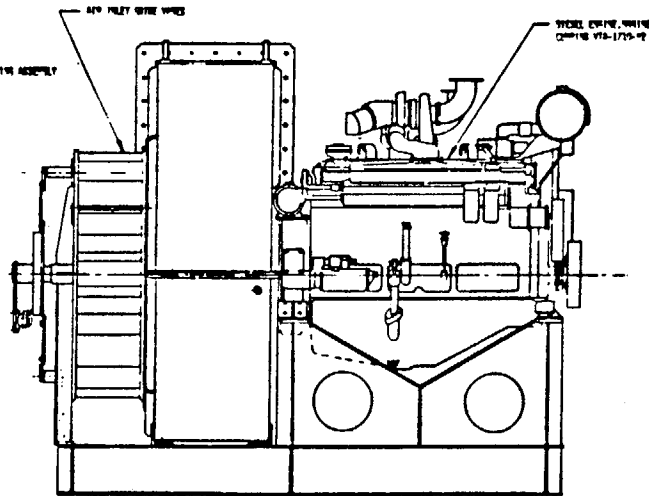
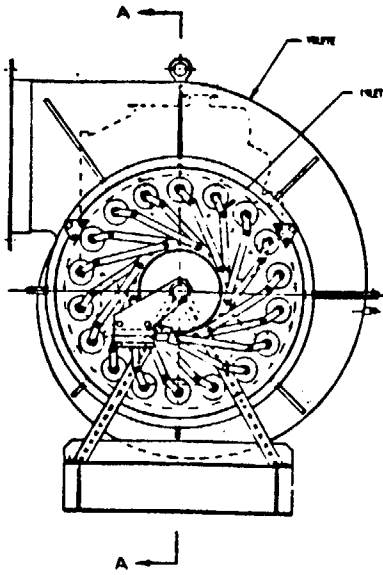
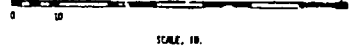
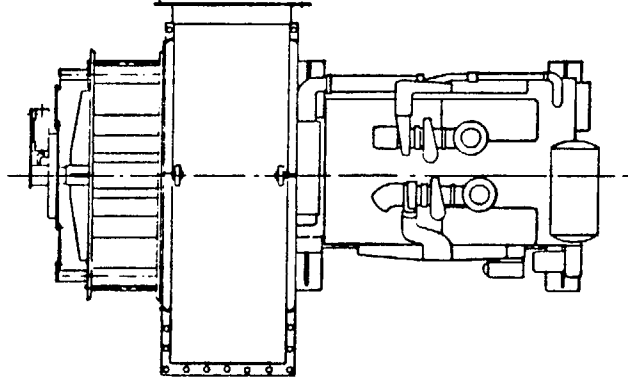
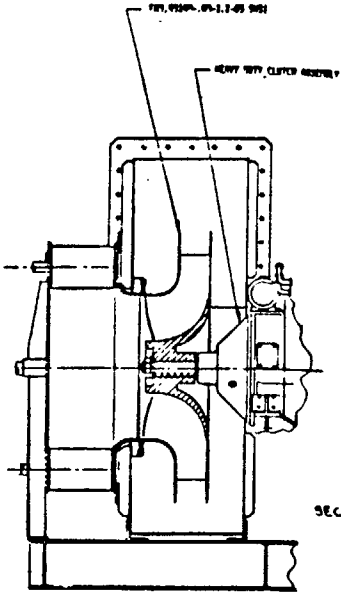
# RD LIFT FAN CHARACTERISTICS 900 L/MDC



SERIES RD 140-.60-1.2-65°, 1800 RPM  
 DIRECT DRIVE, STD. CONDITIONS, NOT  
 ADJUSTED FOR SCALE EFFECT, SWS!

Fig. 6.3.2.

Fig.: 6.3.3.



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# LIFT SYSTEM PERFORMANCE

PROJECT - 280 MDC							DATE 2-17-81	
WHEEL TYPE - RD112-65-1.3-70°					COND. 80F 50			
ENGINE - SACM 195V12CZSHR (2200R @ 1560)								
RIDE CONTROL - OFF, DIRECT								
SUB-SYSTEM →					CUSHION	SEALS		
NUMBER OF DW DI LIFT FANS					2	4		
NUMBER OF INLETS					4	8		
HORST POWER AVAILABLE PER INLET					550	550		
REQ'D.	P <sub>s</sub> STATIC, PSF.				216	238		
	P <sub>t</sub> TOTAL, PSF.				256	282		
	MINIMUM FLOW PER INLET, CFS.				550	496		
	MINIMUM FLOW PER SHIP, CFS.				6168			
AVAILABLE	P <sub>s</sub> STATIC, PSF.				272	310		
	P <sub>t</sub> TOTAL, PSF.				307	338		
	DELIVERED FLOW PER INLET, CFS.				845	750		
	TOTAL FLOW IN CUSHION, CFS.				3380	—		
	TOTAL FLOW IN BOTH SEALS, CFS.				—	6000		
	TOTAL FLOW PER SHIP, CFS.				9380			
DUTY POINT	C <sub>p</sub> TOTAL PRESSURE COEFFICIENT				.0714	.0786		
	C <sub>d</sub> FLOW COEFFICIENT/INLET				.397	.348		
	η <sub>t</sub> TOTAL EFFICIENCY				.848	.842		
	IGV BLADE SETTING, °				-20°	0°		
	SLOPE OF PERFORMANCE CURVE				STABLE	STABLE		
TIP SPEED OF WHEEL, FT./SEC.					475.88			
RPM ENGINE - FAN					1560-2474			
GEAR RATIO REQUIRED					1.586			
STD. VOLUTE DIAM. PER WHEEL INSTL. INCHES	H	L	W <sub>0A</sub>	A	B	h	L	DRV
	95	83	76	28	48	41	36	53

Fig.: 6.3.4.

Fig. 6.3.5.

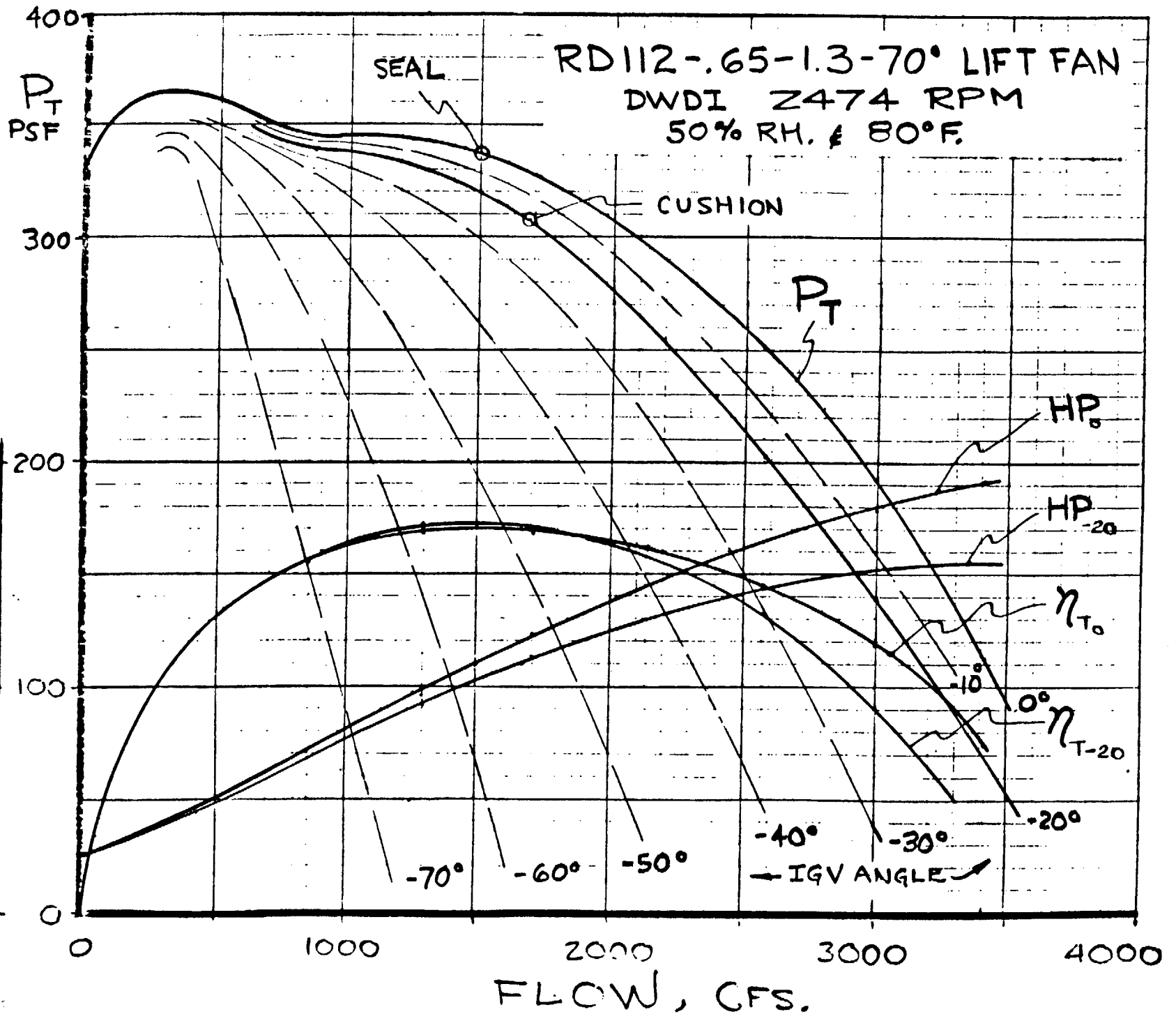
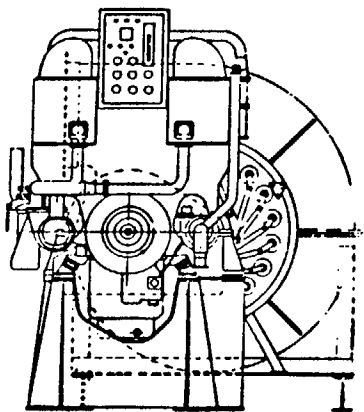
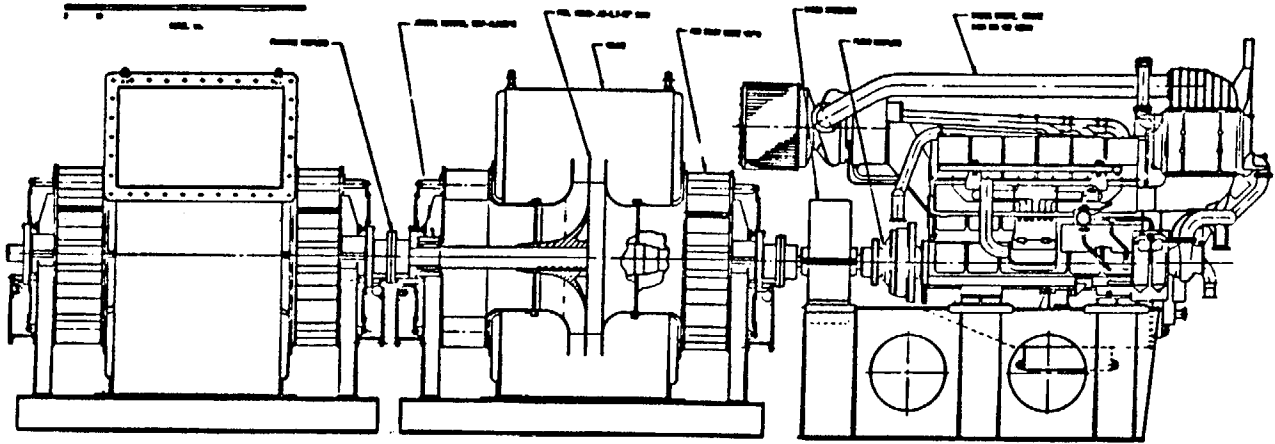
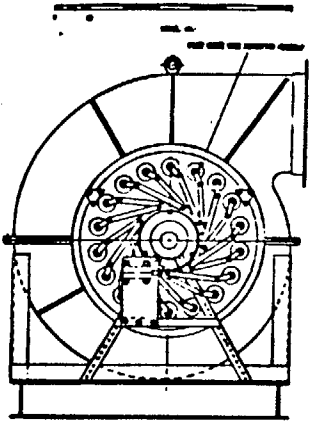


Fig. : 6.3.6.



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LIFT SYSTEM PERFORMANCE

(10,000 LT)

SUB-SYSTEM	SEALS	CUSHION
NO. OF DWDI RD LIFT FANS	4	2
NO. OF INLETS	8	4
HORSE POWER AVAILABLE/INLET	3500	3500
$P_S$ STATIC, PSF	514	467
$P_T$ TOTAL, PSF	555	504
DELIVERED FLOW/INLET, CFS	2917	3120
TOTAL FLOW IN CUSHION, CFS	--	12480
TOTAL FLOW IN BOTH SEALS, CFS	23336	--
TOTAL FLOW/SHIP, CFS		35815
$C_{Pt}$ , TOTAL PRESSURE COEFFICIENT	.0970	.0881
$C_d$ , FLOW COEFFICIENT	.551	.584
$N_t$ , TOTAL EFFICIENCY	.865	.825
IGV BLADE SETTING, DEGREES	-9	-16
SLOPE OF PERFORMANCE CURVE	STABLE	STABLE
TIP SPEED, FT/SECOND		549
RPM, ENGINE - FAN		1350 - 1639
GEAR RATIO REQUIRED		1.214

Fig.: 6.3.7.

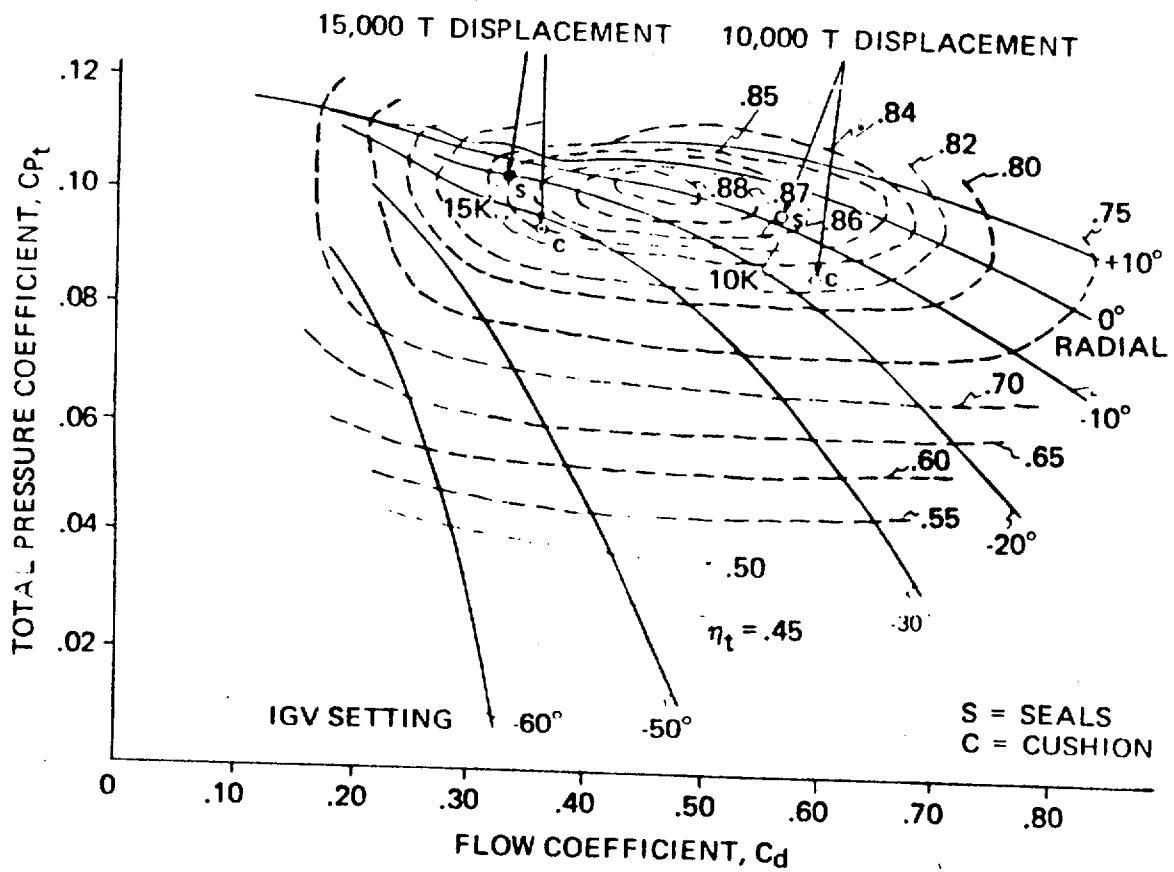
LIFT SYSTEM PERFORMANCE

(15,000 LT)

SUB-SYSTEM	SEALS	CUSHION
NO. OF DWDI RD LIFT FANS	4	2
NO. OF INLETS	8	4
HORSE POWER AVAILABLE/INLET	3500	3500
$P_S$ STATIC, PSF	700	700
$P_T$ TOTAL, PSF	808	735
DELIVERED FLOW/INLET, CFS	2191	2190
TOTAL FLOW IN CUSHION, CFS	--	8762
TOTAL FLOW IN BOTH SEALS, CFS	17528	--
TOTAL FLOW/SHIP, CFS	26290	
$C_{Pt}$ , TOTAL PRESSURE COEFFICIENT	.1050	.0955
$C_d$ , FLOW COEFFICIENT	.322	.354
$N_t$ , TOTAL EFFICIENCY	.850	.850
IGV BLADE SETTING, DEGREES	-10	-32
SLOPE OF PERFORMANCE CURVE	STABLE	STABLE
TIP SPEED, FT/SECOND	636	
RPM, ENGINE - FAN	1350 - 1900	
GEAR RATIO REQUIRED	1.407	

Fig.: 6.3.8.

# GENERAL PERFORMANCE OF FULL SIZE RD WHEEL



SERIES RD 195-.55-1.3-90°  
 80°F TEMP  
 50% REL. HUMIDITY

Fig.: 6.3.9.

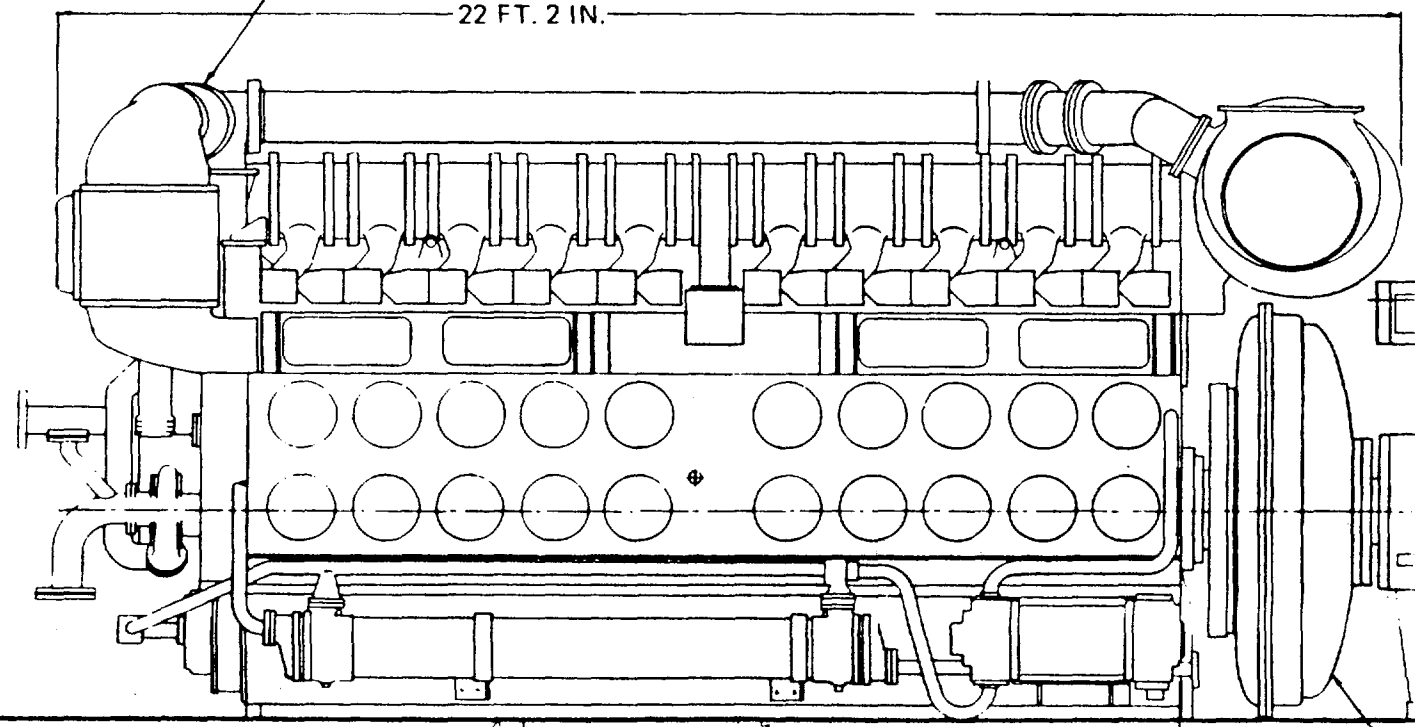


LIFT SYSTEM A

DIESEL ENGINE ~ TYPE "SACM" AGO 240 - V20-RVR  
(SOCIETE ALSACIENNE DE CONSTRUCTIONS MECHANQUES DE MULH)  
7000 CONT. H.P. @ 1350 RPM

SPEED INCREASER

22 FT. 2 IN.



37 FT. 6 IN. UNIT LENGTH OVERALL

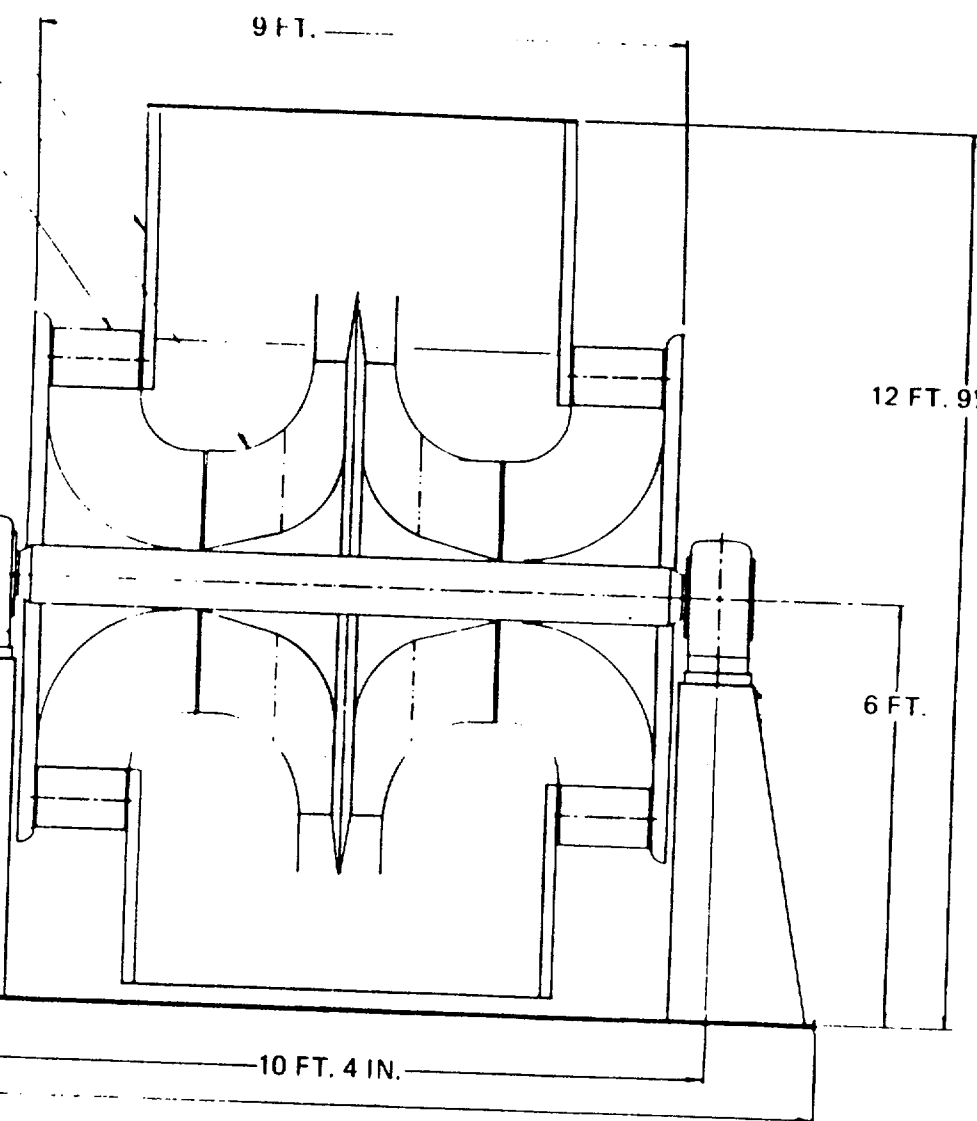
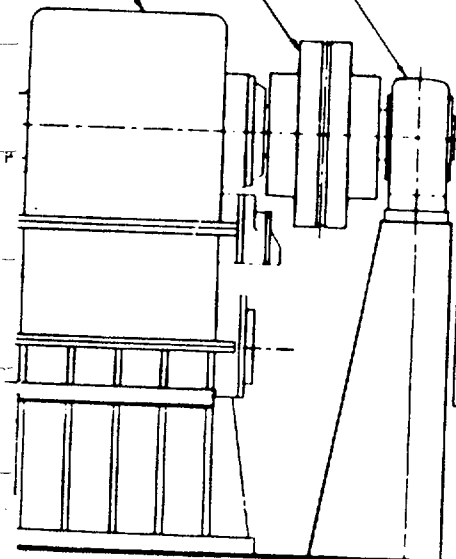
Fig. 6.3.10a

MACHINERY SET

USE)  
RD 195-.55-1.3-90° LIFT FAN—  
AIR DELIVERY  
VOLUTE  
RADIAL AIR INLET GUIDE  
VANE SYSTEM

FLEXIBLE  
COUPLING

JOURNAL  
BRG.



9 FT.

12 FT. 9½ IN.

6 FT.

2 FT. 2 IN.

10 FT. 4 IN.

FLUID COUPLING

SIDE ELEVATION

FIGURE 4-28

MPS LIFT FAN – SIDE VIEW

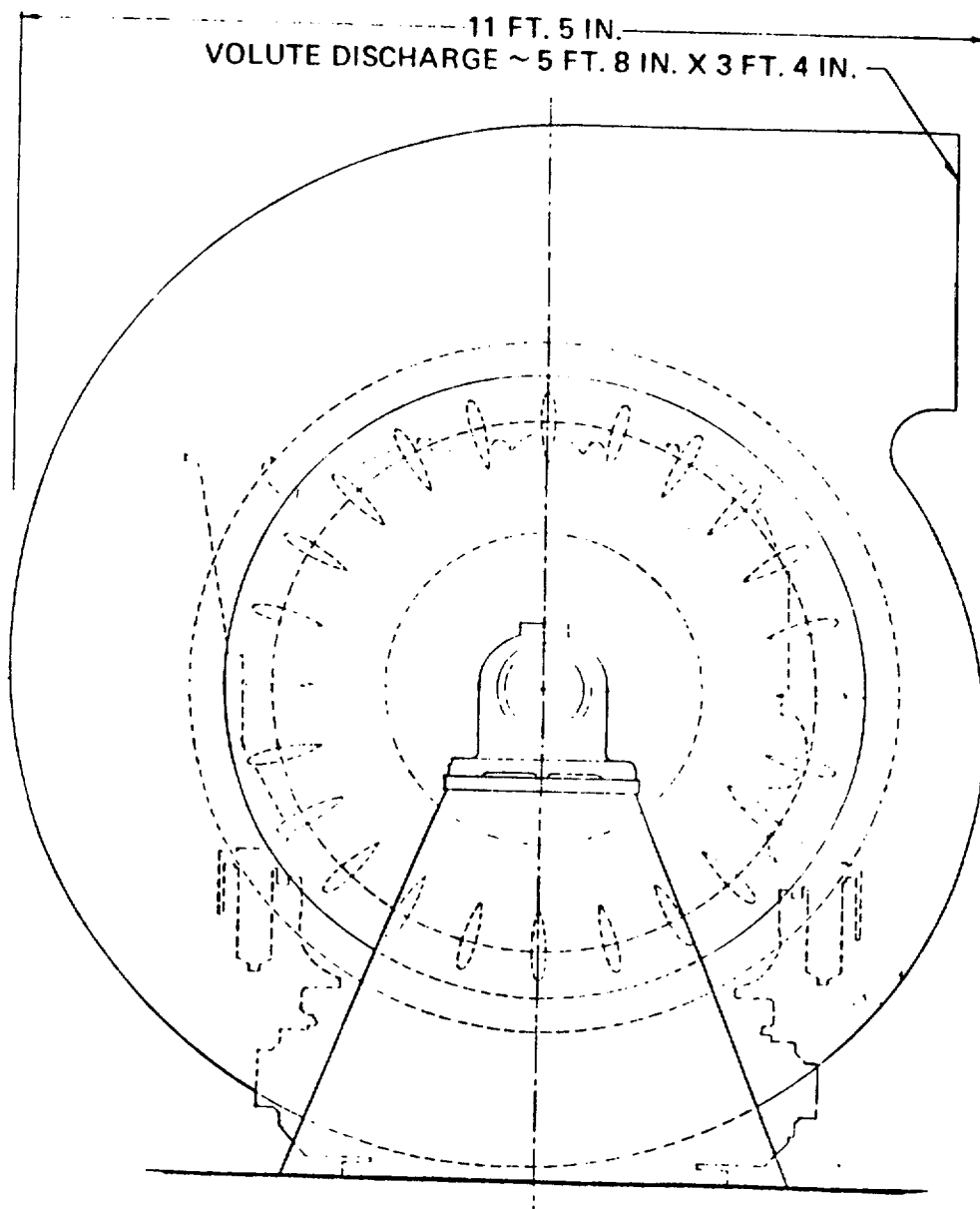


FIGURE 4-30

## 6.4 Noise and Its Control

In its December 1976 houseorgan, Courrier, Neu of Lille, France published information on noise levels of centrifugal turbomachinery. Aerophysics has tested and installed NEU fans in the U.S., and presents the following findings related to fan noise.

Centrifugal fans can be either shrouded or unshrouded wheels with backward-inclined, radial or forward-inclined blades. Their volutes can either be equipped with a diffuser or be without: the diffuser can be either a fixed structure beyond the volute or a circumferential area beyond the discharge end of the wheel. Finally, the diffuser can be axysymmetrical, in which case it will be of the vaned, stationary type. The combination of the above-mentioned elements can strongly affect the noise levels. The following points should be noted:

- . Turbulence noise increases with the speed of flow through the fan and volute. Therefore, large passage areas are beneficial in lowering noise.
- . For a given airflow capacity, low noise levels usually go together with good aerodynamic design and, hence, high aerodynamic efficiency.

An exception is siren noise, which can be very noticable even at very low energy levels. It is well-known that a sound power level of 10 watts corresponds to a sound power level, as follows:

$$L_W = 10 \text{ Log } \frac{10}{10^{-12}} = 130 \text{ dB}$$

Siren noise can be generated by fixed obstacles placed upstream of, or at, the fan inlet; their wake is picked up and amplified by the blades or the fan. Such obstacles can be bearing supports, turning vanes or pre-rotation vanes.

- . Siren noise can be generated by the interaction of the wake of the fan blades with one or more fixed obstacles downstream of the wheel. If the velocity profile of the discharged air is irregular, the same phenomenon occurs, at a much higher frequency, for a fixed vane (circumferential in a multi-staged diffuser or toroidal channel).
  
- . The RD fan significantly reduces siren noise since the exhaust flow is "diffused" between the discharge from the blades and the diffuser exhaust. The sound power level is little changed by changes in the operating point of the RD fan. The remarkably lower noise results from the higher efficiency and from the flow stability of this type of machine. (See Figures 6.4.1 - 6.4.3).
  
- . Unshrouded centrifugal-type wheels operating without bellmouth, in a big volute, called "transport wheels", are noisy. Often the vector diagram of the entrance velocity is incorrect, which results in large vortices in the volutes. However, the ability of transport wheels to handle very dusty industrial gases at high temperatures makes them desirable in thermal engineering.
  
- . Backward-inclined wheels, with thin or thick airfoils, are fairly quiet for high-efficiency regimes, if the wheel is effectively matched to the air delivery circuit.
  
- . Forward-inclined wheels have a very high pressure coefficient and operate over a wide range of airflows. This allows fans with low structural volume; however, they have no less aerodynamic noise than previously-mentioned wheels for the same capacity and pressure. Since their rotation speed is lower, their bearing noise is lower, which makes them desirable when mechanical noise is likely to be higher than aerodynamic noise, as in low-power equipment. (See Figure 6.4.4).

- . The thickness and stiffness of the walls of the volute are important in absorbing radiating noise of the fan assembly. This is why blowers and compressors which usually have thick walls do not radiate appreciable noise levels, even at higher mechanical power levels.
- . The type of bearings has an effect on the noise of the rotation equipment. For lower noise levels, journal bearings are preferable to ball bearings or roller bearings.

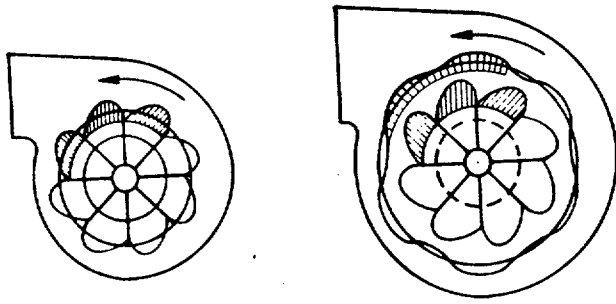
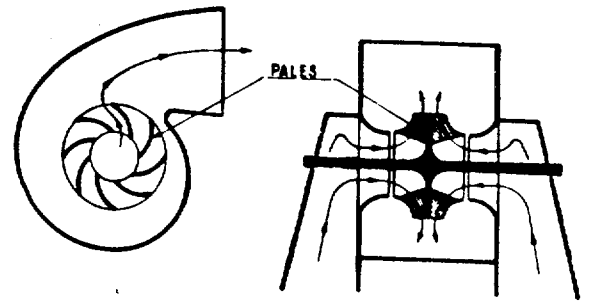


Figure 6.4.1: Schematic (left) of the airflow out of a conventional centrifugal wheel. Schematic (right) of the airflow for an RD wheel showing greater flow uniformity & hence lower siren noise.



Ventilateur centrifuge courant

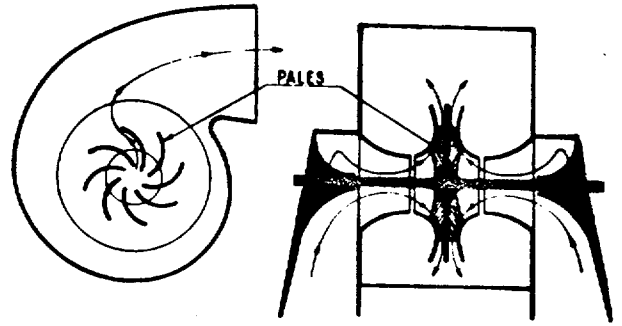


Figure 6.4.2: Section view of two DWDI fans, the lower figure being an RD-type.



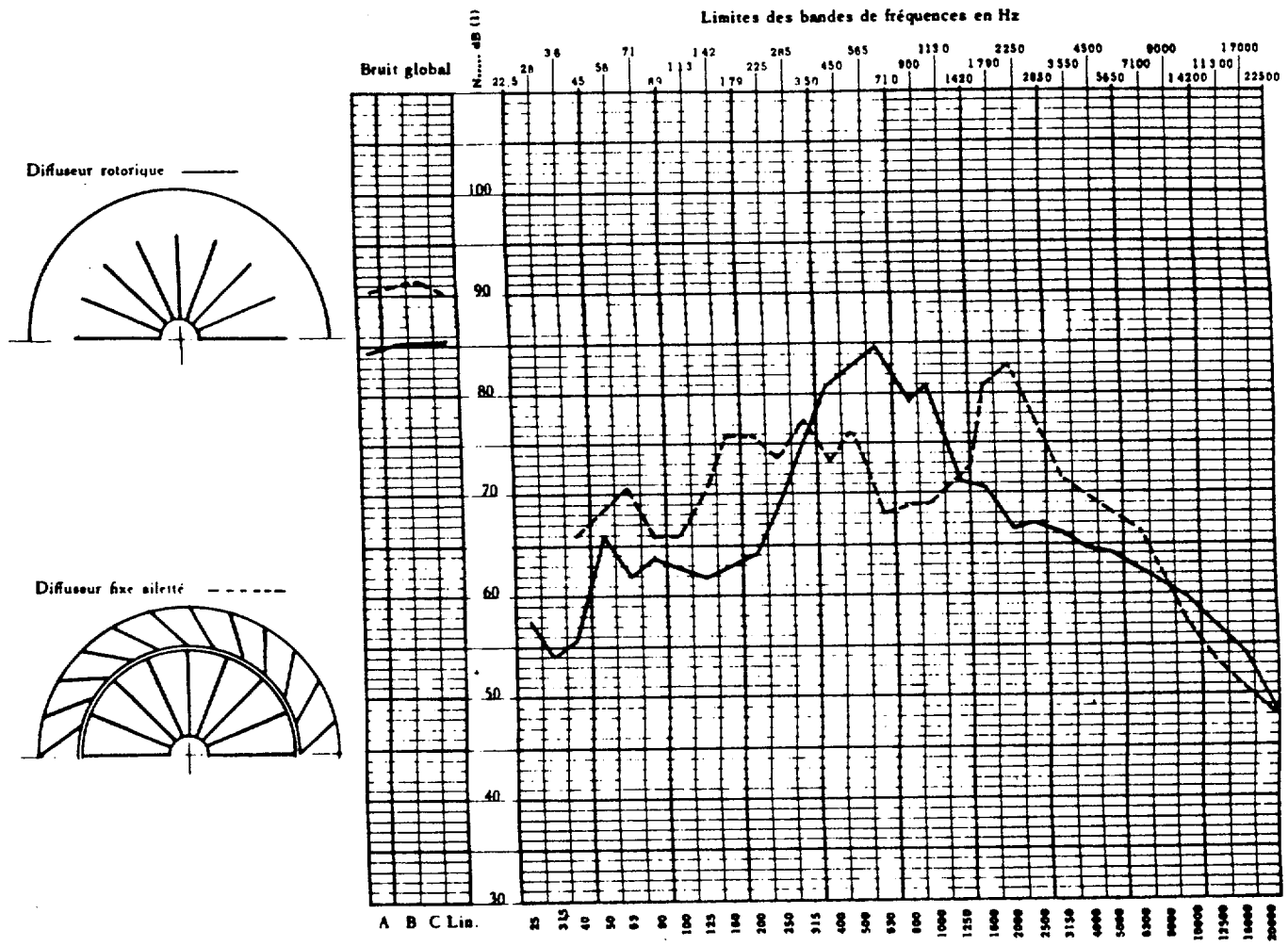


Figure 6.4.3:  
 Comparison of noise levels as a function of frequency  
 of two radial type wheels, one RD, one circumferential  
 fixed-vane diffuser.

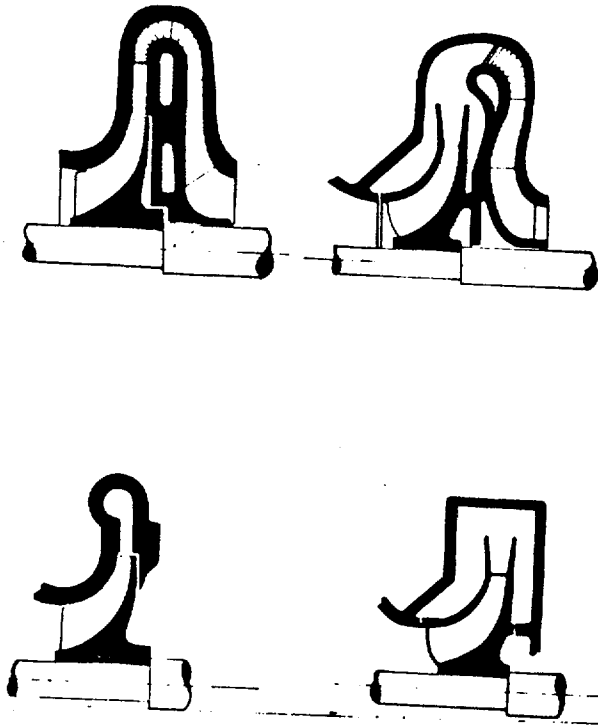


Figure 6.4.4:  
Sectional views of two types of centrifugal  
compressors (respectively with and without  
an RD), both with volute and cylindrical  
discharge with toroidal-shaped channels.

## 7. CONCLUSIONS

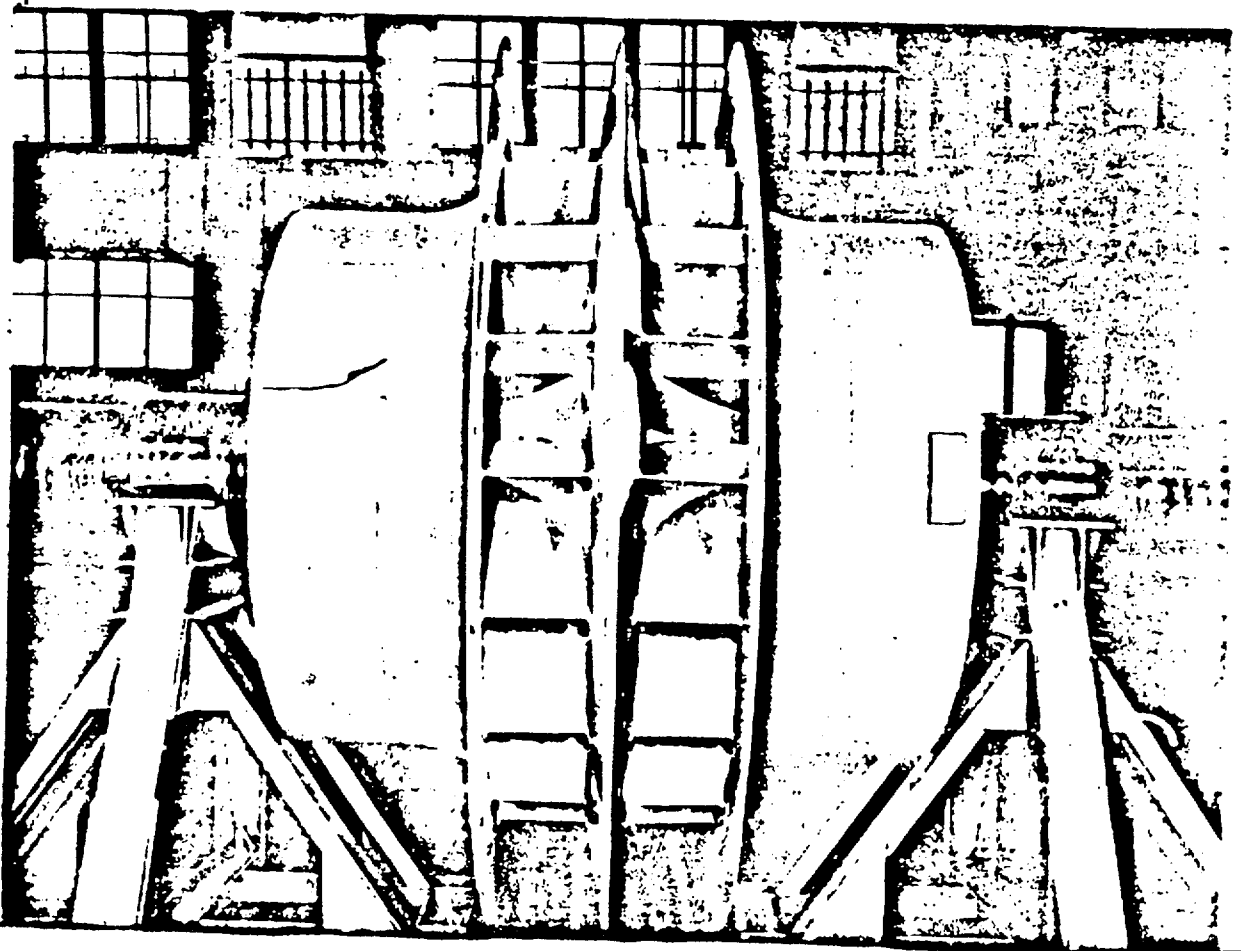
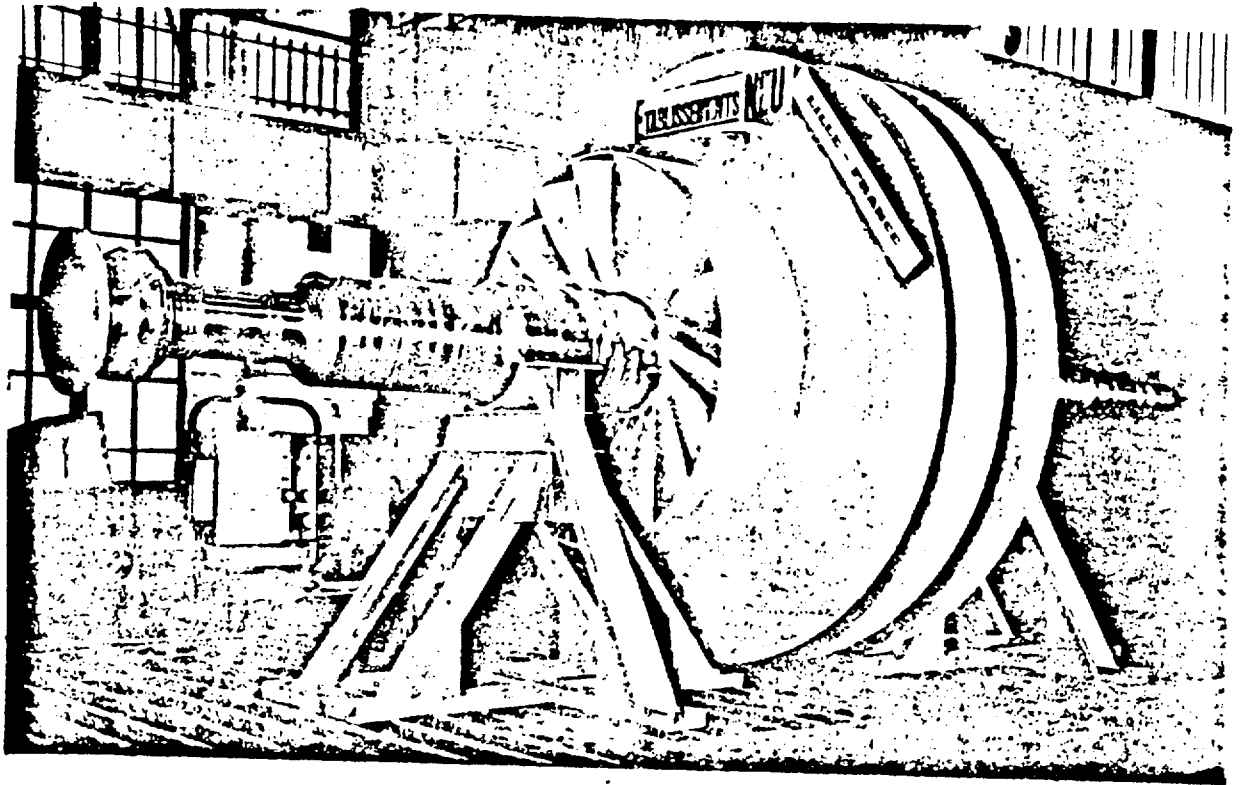
As this study shows, the RD fan is the most efficient, aerodynamically-sound and especially-structured fan for use in the 1982 series SES program. The fan has been successfully tested and applied to heavy-duty uses, including marine service. Therefore, Aerophysics concludes that the full-scale RD fan is available without technical risks as the full-scale lift fan system for forthcoming SES applications.

8. REFERENCES

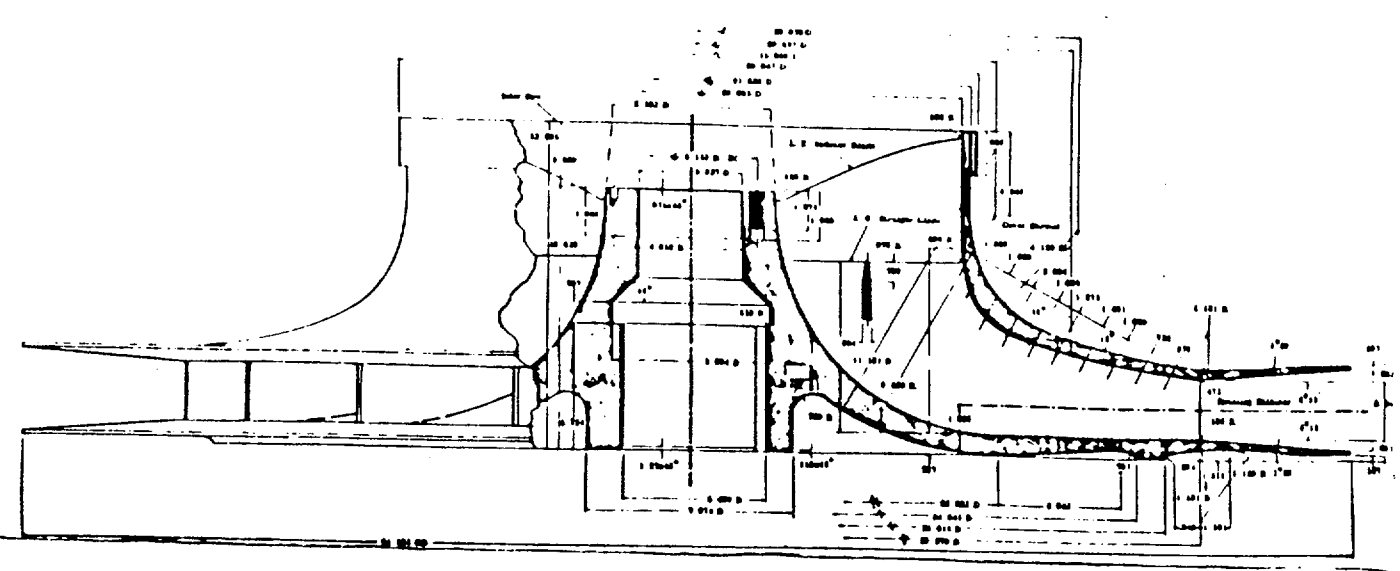
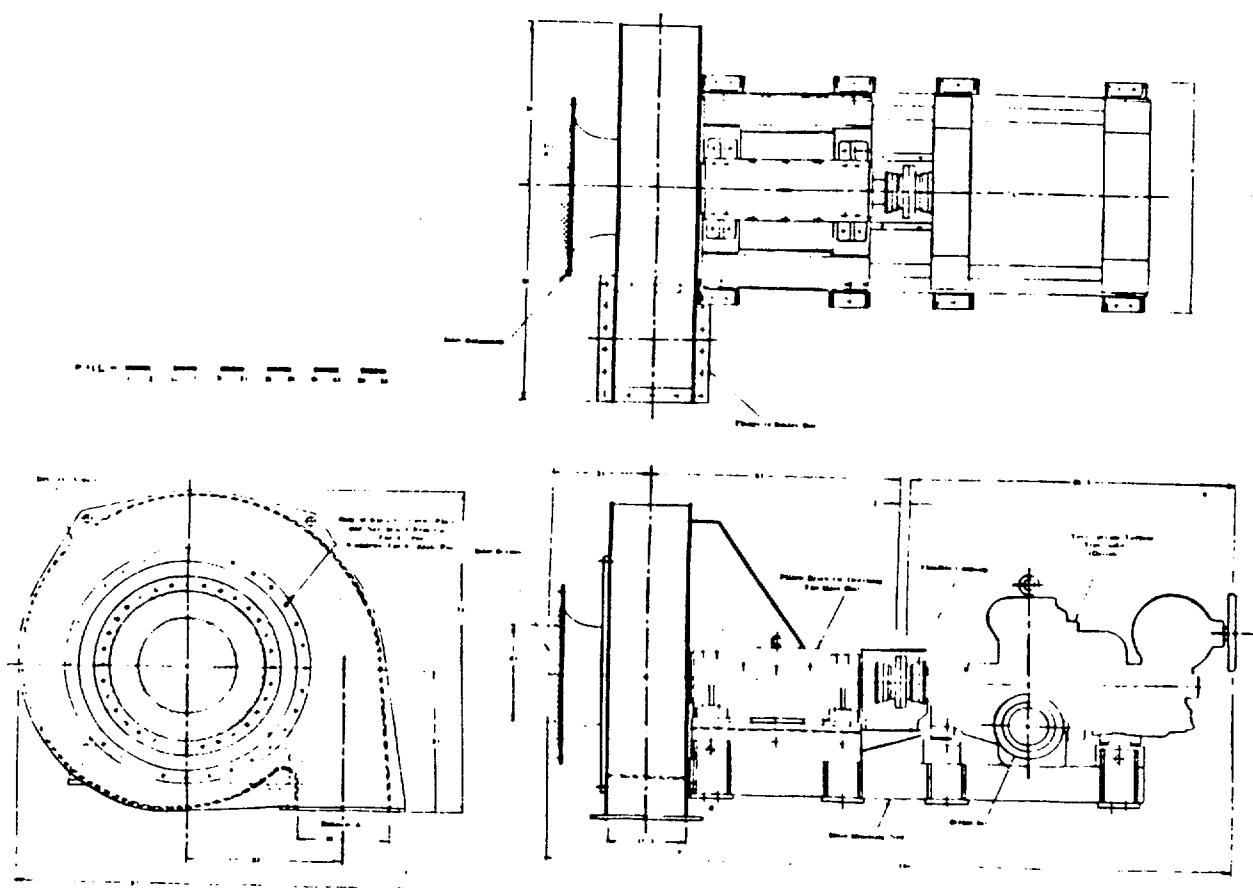
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9. APPENDICES

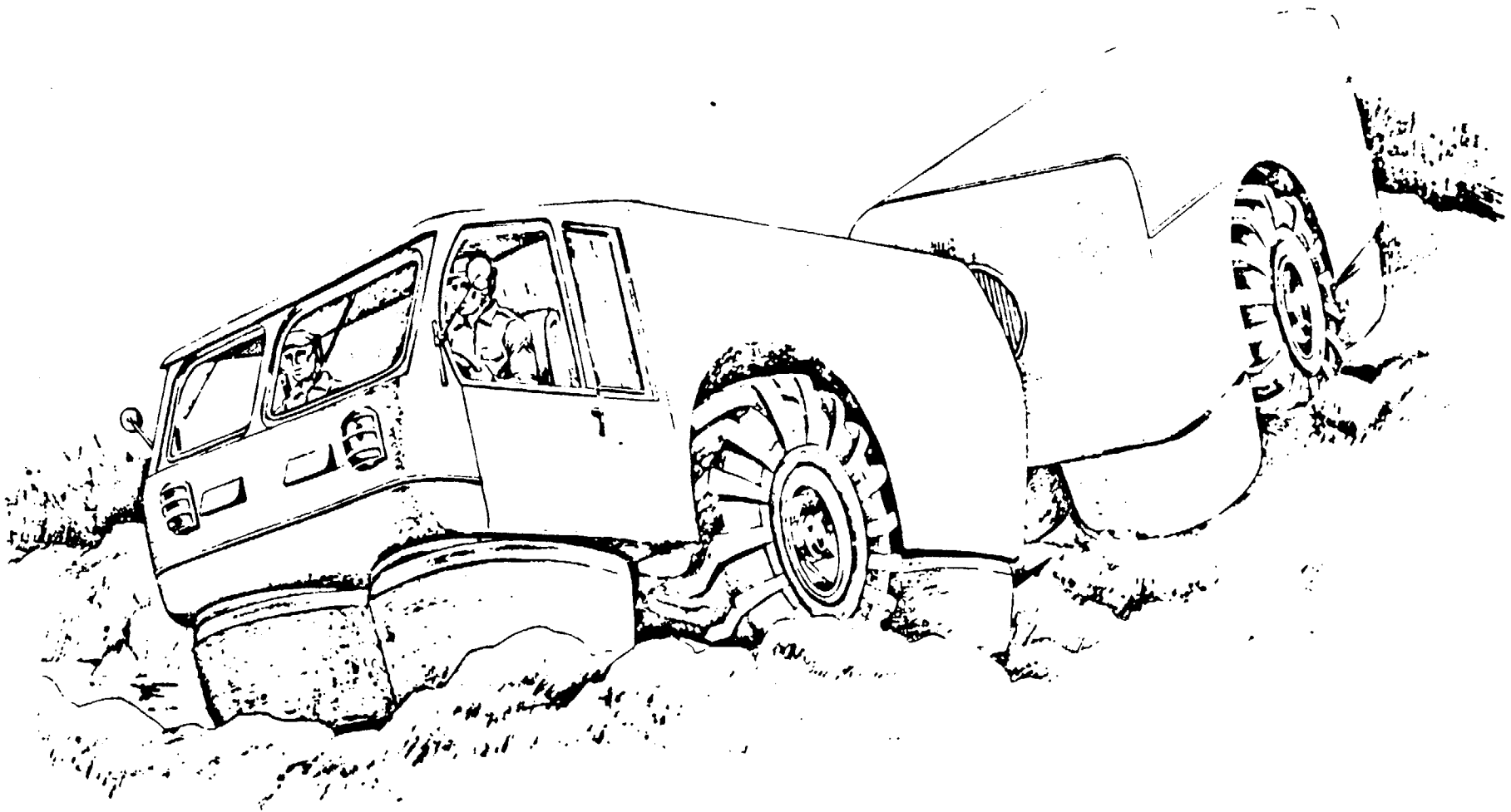


Appendix I: A Representative Industrial Service DWI  
Rotating Diffuser wheel of large diameter



Appendix II: Steam turbine-driven RD fan for forced draft marine boiler and propulsion

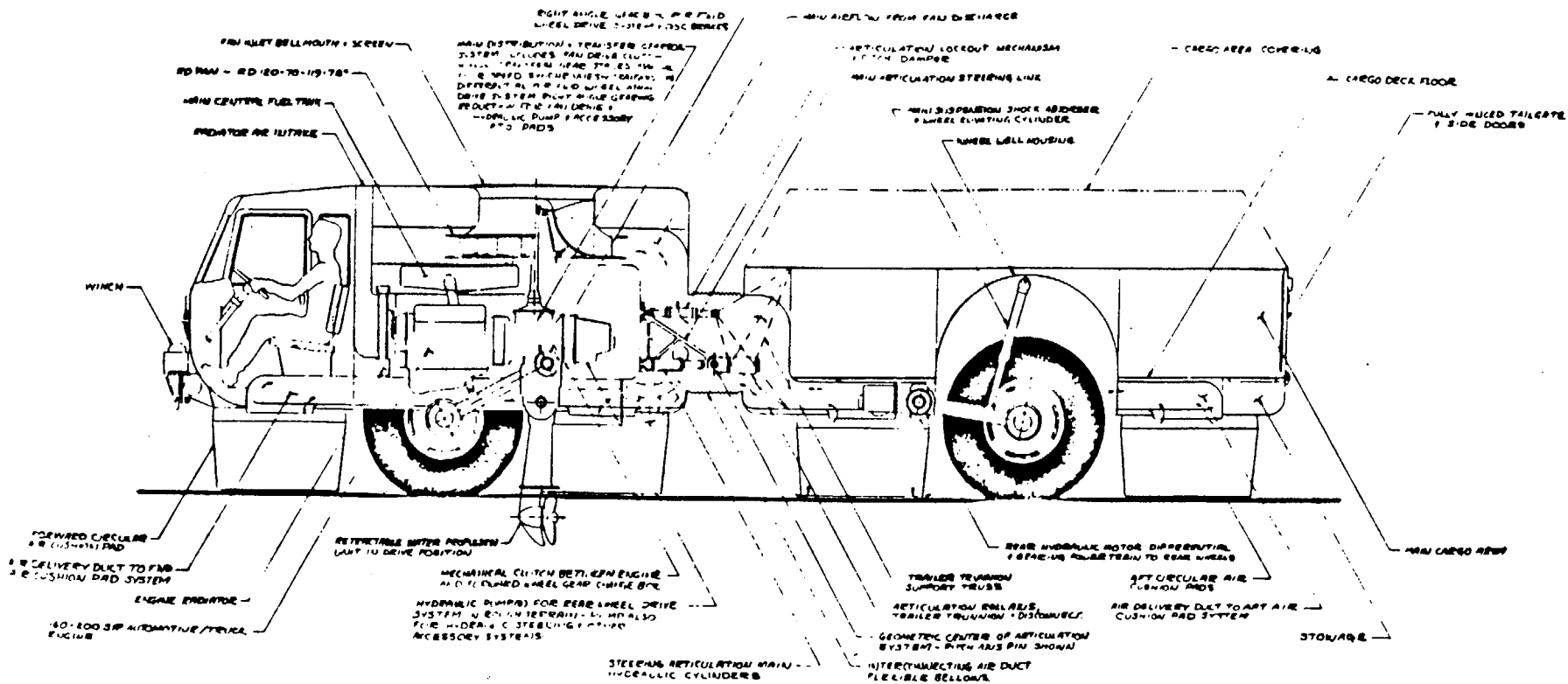




117

Appendix IIIa: Representational view of all-terrain vehicle with air cushion pad support.

811



Appendix III b: Inboard profile of all-terrain vehicle with air cushion pad support.

APPENDIX IV a

NEUAIR, INC.

References in United States and Canada

- 1974 BEKER INDUSTRIES  
Agricultural Products Corporation  
Conda, Idaho  
1 Blower Air + 0.5% SO<sub>2</sub>  
R.D. 185-0.46/1.2 SWSI overhung - axial inlet guide-vanes  
93 3/8" wheel diameter  
104,000 CFM 100 IWG 180° F 1865 BHP 1780 RPM
- 1976 DORR OLIVER  
Stamford, Connecticut  
for  
BEKER INDUSTRIES  
Conda, Idaho  
1 Blower Air  
R.D. 84-0.5/1.09 SWSI overhung  
36" wheel diameter  
34,000 CFM 7.5 PSI 70° F 1500 BHP 5467 RPM
- 1976 SCIENTIFIC DESIGN  
New York, N.Y.  
for  
HOUSTON CHEMICAL CO.  
Division of PPG Industries  
- 1 Spare Impeller Process Gas  
R.D. 23-0.5/1.3 SWSI  
11 3/4" wheel diameter  
1740 ACFM 63 PSI 99° F 612 BHP 12,235 RPM  
- 1 Spare Impeller Process Gas  
R.D. 60-0.46/1.3 SWSI  
30 3/4" wheel diameter  
8500 ACFM 59 PSI 99° F 2420 BHP 4670 RPM
- 1976 BEKER INDUSTRIES  
Conda, Idaho  
1 Spare Rotor Air + SO<sub>2</sub>  
R.D. 185-0.46/1.2 SWSI  
93 3/8" wheel diameter  
104,000 ACFM 100 IWG 180° F 1865 BHP 1780 RPM

APPENDIX IV b

Neuair, Inc.  
References  
Page Two

- 1977 BSP ENVIROTECH  
Belmont, California  
for  
San Jose Scum Incinerator  
San Jose, California  
1 R.D. 84-0.65/1.09 SWSI overhung - wheel Carpenter 20  
stator 316  
36" wheel diameter  
41,000 ACFM 2.8 PSI 150° F 560 BHP 3580 RPM
- 1977 PPG  
Pittsburgh, Pennsylvania  
for  
HOUSTON CHEMICALS  
Houston, Texas  
2 Spare Impellers, identical to the ones sold to  
Scientific Design one year before
- 1977 DAVY POWERGAS  
Lakeland, Florida  
for  
STANDARD OIL OF CALIFORNIA  
CHEVRON CHEMICAL COMPANY  
New Orleans, Louisiana  
Oil incineration  
1 R.D. 168-0.50/1.2 SWSI  
78 3/4" wheel diameter  
90,912 ACFM 68 IWG 600° F 1214 BHP 2225 RPM
- 1978 CHEVRON CHEMICAL  
Belle Chasse, Louisiana  
1 Spare Rotor for blower sold to Davy Powergas one year before
- 1978 DOFASCO  
Hamilton, Ontario  
I.D. Fan for converter  
1 S.L. 154-0.60/1.09 DWDI  
66" wheel diameter  
120,000 ACFM 20 IWG 270° F 488 BHP 1190 RPM
- 1979 KOPPERS  
Pittsburgh, Pennsylvania  
for  
The Industrial Development Board of the City of Fairfield, Alabama  
Coke oven pushing emission control - One spot cars  
2 R.D. 120-0.55/1.09 SWSI  
57 1/2" wheel diameter  
40,000 ACFM 41 IWG 151° F 320 BHP 1780 RPM

APPENDIX IV c

Neuair, Inc.  
References  
Page Three

- 1979            DRAVO CORPORATION  
                 Pittsburgh, Pennsylvania  
                 for  
                 ALLIED CHEMICAL  
                 Detroit, Michigan  
                 Coke oven pushing emission control  
                 2 UL 185-0.55/1.09 SWSI Stainless Steel  
                 79 1/2" wheel diameter  
                 92,900 ACFM    27.6 IWG    256° F    521 BHP    1180 RPM
- 1979            DRAVO CORPORATION  
                 Pittsburgh, Pennsylvania  
                 for  
                 CITIZEN GAS  
                 Indianapolis, Indiana  
                 Coke oven pushing emission control  
                 1 UL 168-0.60/1.09 DWDI  
                 73 3/8" wheel diameter  
                 146,600 ACFM    22.5 IWG    260° F    645 BHP    1180 RPM
- 1979            RILEY ENVIRONEERING  
                 Chigaco, Illinois  
                 for  
                 INTERLAKE STEEL  
                 Toledo, Ohio  
                 Coke oven pushing emission control - One spot car  
                 1 UL 130-0.50/1.10 DWDI Stainless Steel  
                 55 5/8" wheel diameter  
                 75,000 ACFM    31 IWG    156° F    462 BHP    1780 RPM

## APPENDIX V a

## VENTILATEURS DE CHAUFFE DU PAQUEBOT "FRANCE"

On a déjà tant parlé et écrit au sujet du paquebot "France" qu'il semble inutile de présenter ce navire qui constitue l'unité la plus importante de la Marine marchande française.

Rappelons simplement ses caractéristiques générales :

Longueur hors tout .....	315,66 m
Largeur maximale .....	33,70 m
Déplacement en charge .....	57 000 t
Puissance de l'appareil moteur ...	160 000 ch
Vitesse moyenne de service .....	31 nœuds
Vitesse maximale supérieure à ...	33 "

qui en font le paquebot le plus long de la flotte mondiale. Construit aux Chantiers de l'Atlantique à St-Nazaire, pour la Compagnie Générale Transatlantique, par sa conception et sa réalisation il fait honneur à l'armement et à la construction navale de notre pays qui prouvent ainsi, 25 ans après l'achèvement du prestigieux "Normandie", qu'ils se sont maintenus en tête du progrès international dans le domaine le plus complexe des techniques maritimes.

Nos Établissements se devaient de participer à une telle réalisation et, en particulier, à l'étude des ventilateurs de chauffe destinés à l'appareil propulsif, auxiliaires vitaux du navire.

Nous ne nous étendrons pas sur les multiples projets qui ont été envisagés tout au long de l'étude et de la mise au point de l'appareil propulsif, pour en arriver au matériel qui équipe en définitive "France". Alors que "Normandie" avait une propulsion turbo-électrique comprenant 29 chaudières fonctionnant avec de la vapeur à 28 kg/cm<sup>2</sup>, "France" est équipé de 8 chaudières produisant de la vapeur surchauffée à 500°C sous une pression effective de 71,5 kg/cm<sup>2</sup>. Ces 8 chaudières sont réparties en un groupe avant et un groupe arrière de chacun 4 chaudières.

Le système de ventilation est dit "de soufflage" seulement, c'est-à-dire que les ventilateurs, placés en tête de circuit, soufflent l'air nécessaire aux brûleurs à mazout jusqu'à évacuation des fumées par les cheminées. Il avait été envisagé à un certain moment de faire un système "soufflage-tirage", système qui présente le gros avantage de pouvoir régler la pression dans les circuits de fumées et par conséquent d'en minimiser les fuites éventuelles dans les compartiments chauffés (ou leurs salubrité, l'entretien facilité des autres auxiliaires, etc...).

Moyennant certaines précautions supplémentaires pour la réalisation de ces circuits de fumées, la solution du "soufflage seul" a été retenue car elle présente des avantages non négligeables comme :

- gain de poids sur les ventilateurs,
- gain d'encombrement,
- gain de puissance assez important,
- entretien plus facile (les ventilateurs de soufflage fonctionnent à la température ambiante, avec un fluide non corrosif, ni abrasif).

Les ventilateurs de chauffe sont au nombre de 16, soit 2 appareils en parallèle par chaudière.

Pratiquement, avec un ventilateur par chaudière, on peut assurer la marche normale du navire ; les deux ne sont nécessaires que pour la marche maximale. Ceci constitue un élément de sécurité qui doit permettre de faire face à des avaries éventuelles, sans perturber l'allure du paquebot.

Pour satisfaire aux impératifs de stabilité de fonctionnement, de rendement élevé, de bruyance minimum, nous ne pouvions proposer que des ventilateurs "à diffuseur rotorique" qui sont bien les appareils présentant ces qualités au plus haut degré.

Nous avons donc livré :

16 ventilateurs centrifuges "à diffuseur rotorique" n° 120, série 0.6/1.2 entraînés par des moteurs électriques à 1 800 t/mn (une seule vitesse) d'une puissance nominale unitaire de 250 ch. Le débit global des 16 groupes peut atteindre 1.330.000 m<sup>3</sup>/h mais il est plus normalement voisin de 880.000 m<sup>3</sup>/h.

Chaque ventilateur est muni, à son refoulement, d'un clapet équilibré qui se ferme automatiquement dès que la pression dans le circuit devient supérieure à celle fournie par l'appareil : ceci évite les fuites d'air importantes lorsque le 2<sup>e</sup> ventilateur en parallèle continue à fonctionner.

D'autre part, tous les ventilateurs sont asservis à un système de chauffe automatique qui permet de régler le débit d'air nécessaire en fonction du combustible à brûler pour obtenir la marche désirée du navire.

Ce réglage du débit pouvait être obtenu de plusieurs manières, mais nous avons adopté le principe qui nous ayant fait ses preuves depuis de longues années pour tous les ventilateurs de chauffe de navires, c'est-à-dire

APPENDIX V b

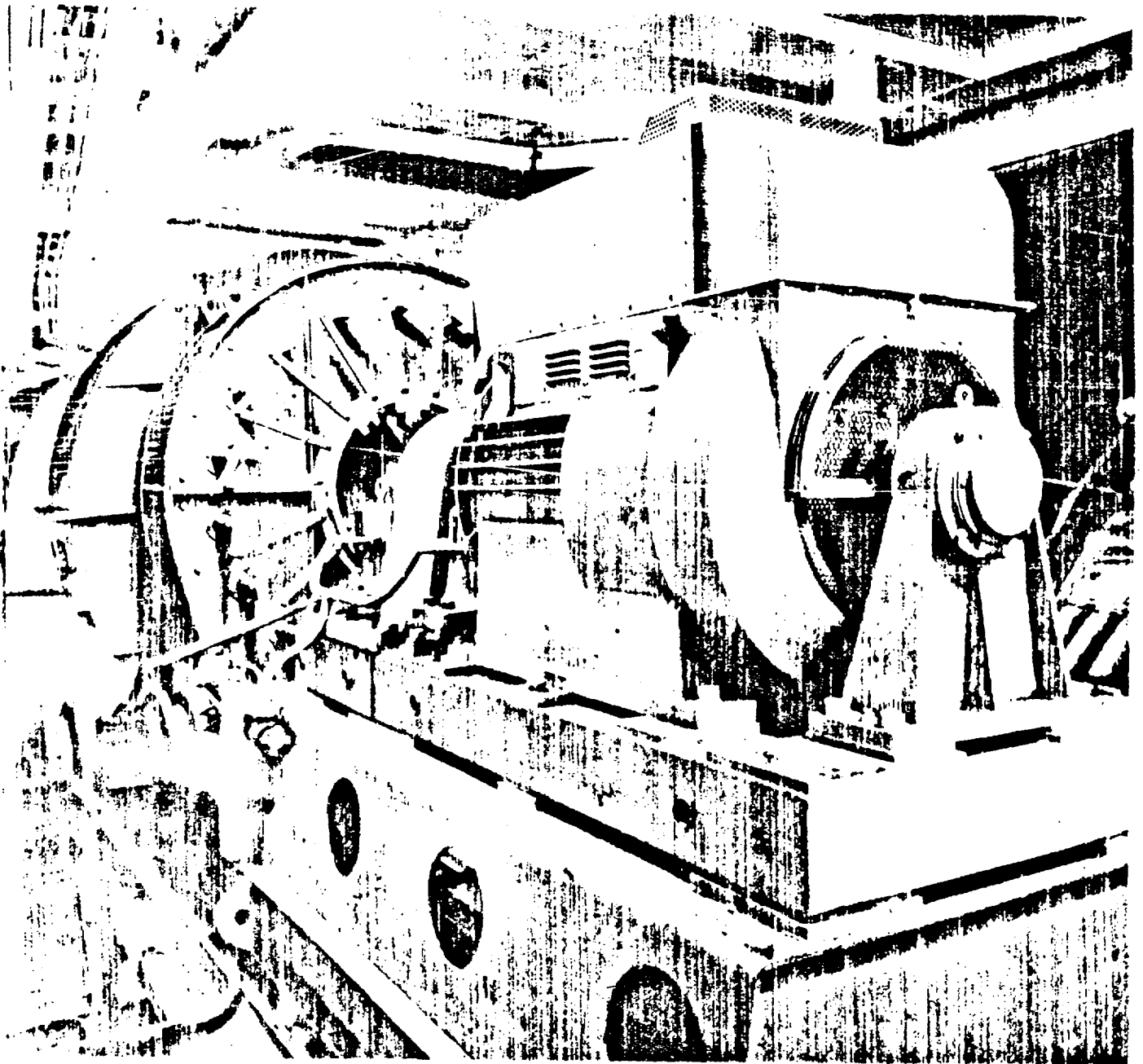


Fig. 1. — Un des 16 ventilateurs centrifuges a diffuseur rotorique n° 120, série 0,6.

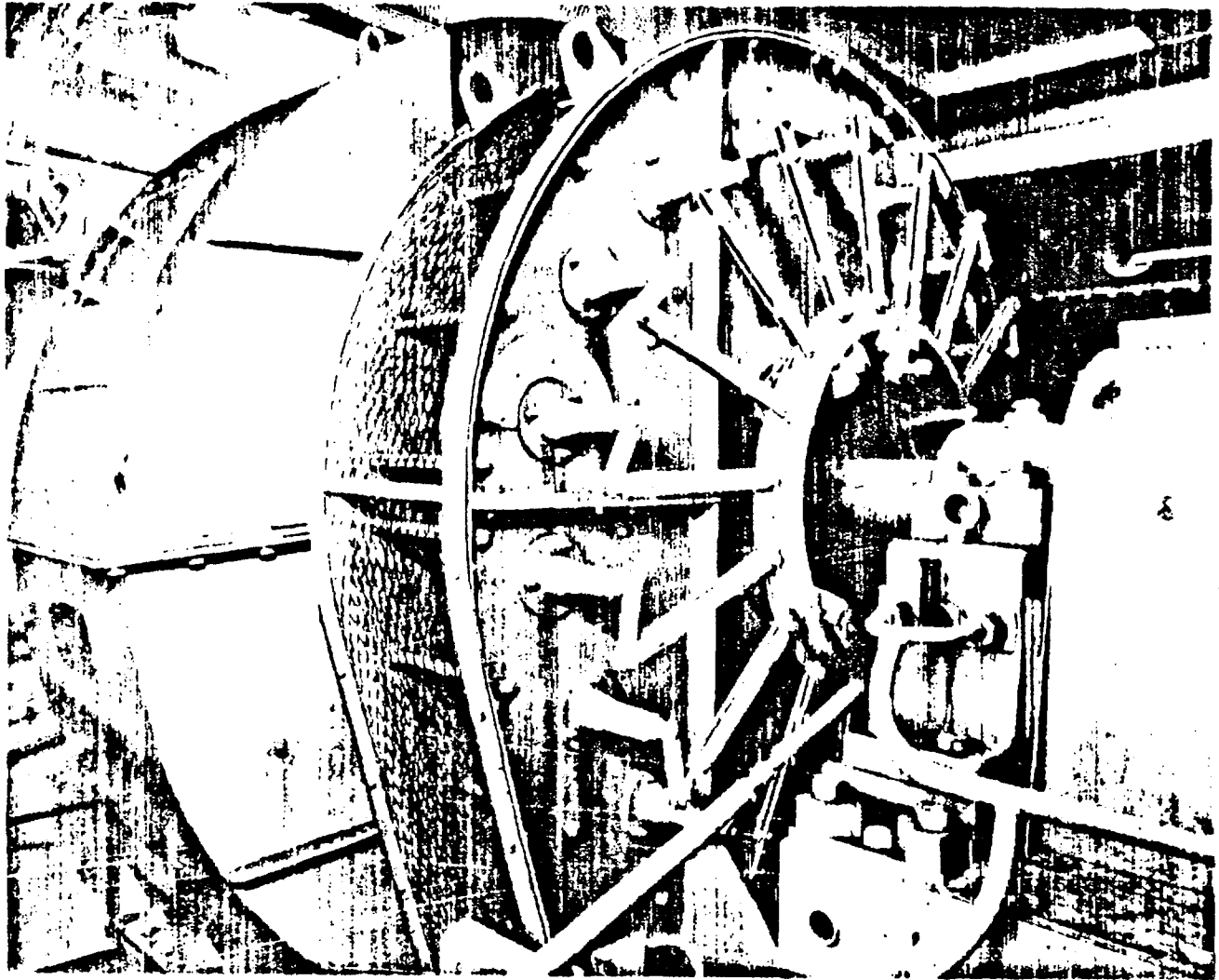


Fig. 2. Détails montrant le dispositif de commande de volets de réglage du débit.

un inclineur torque variable placé à l'aspiration du ventilateur.

Ce dispositif comprend des volets mobiles placés radialement et dont la rotation crée une giration dans l'axe de la roue : si cette giration est dans le sens inverse de la rotation de la roue, la pression fournie par le ventilateur augmente et elle diminue si la giration est créée dans le sens de rotation de la roue.

Nous pouvons ainsi obtenir tout un réseau de courbes caractéristiques permettant de réaliser théoriquement *à volonté* quel point entre le débit maximum et le débit nul. Nous disons théoriquement car, en fait, pratiquement, il est impossible de réaliser un ensemble de volets mobiles qui soit parfaitement étanche à la turbulence. Pour les ventilateurs du « France » nous avons pris les précautions particulières qui ont permis d'obtenir un débit minimum inférieur à 5 % du débit nominal, sur une plage de variation extrêmement grande et couvrant largement les besoins d'exploitation.

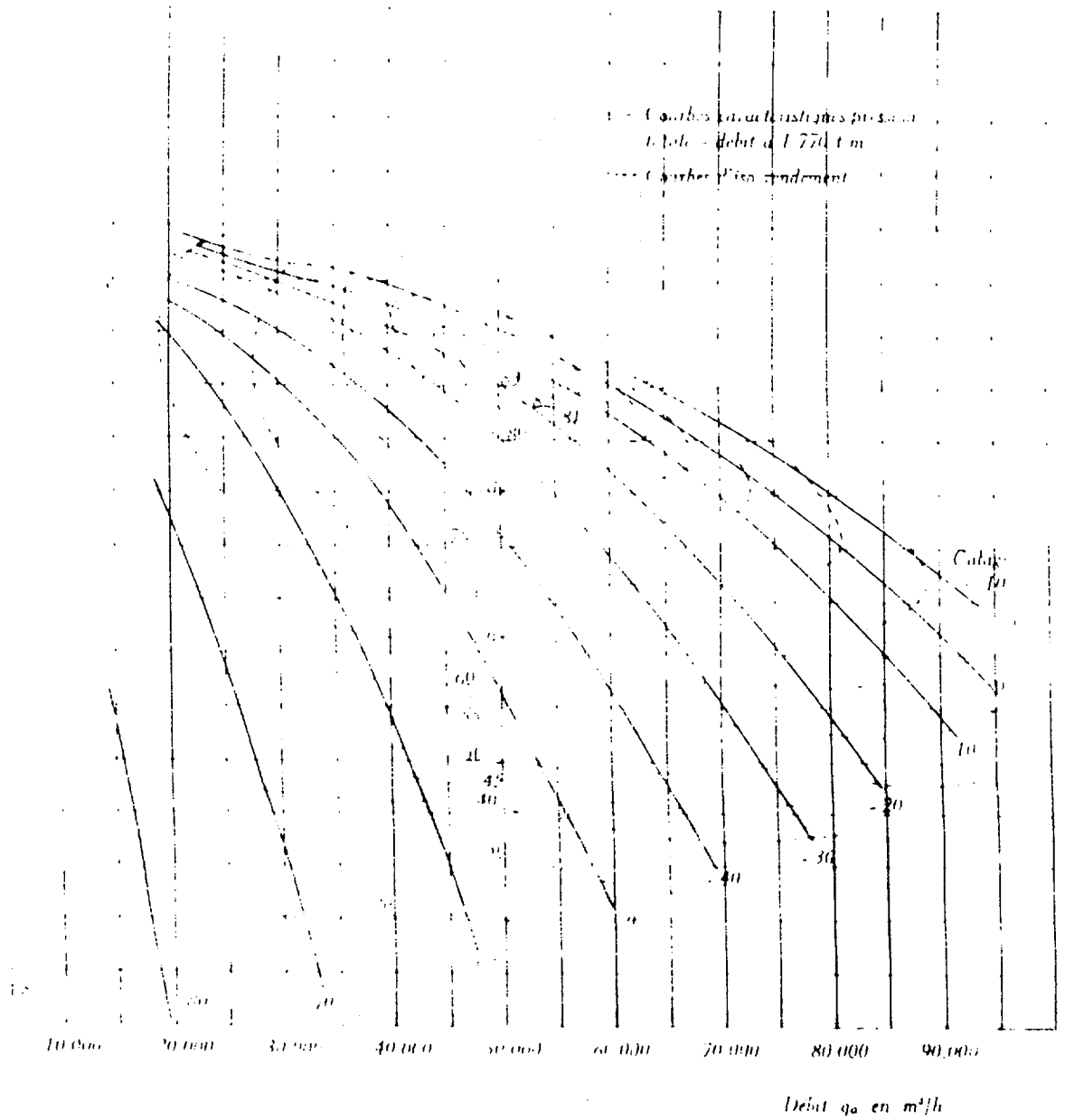
La facilité de l'entretien des groupes à bord du « France » était une des préoccupations de l'Armateur, compte tenu de l'exiguïté des locaux où ils sont installés.

Alors que les ventilateurs ont habituellement leur arbre supporté par 2 paliers à roulements, ceux du « France » présentent la particularité d'avoir un palier à roulement et un palier lisse à coussinet en 2 parties. En effet, si le palier à roulement côté opposé au manchon d'accouplement est facilement visitable et remplaçable, il n'en est pas de même de celui côté accouplement pour lequel un changement de roulement nécessite le démontage du moteur et du manchon. Pour remédier à cet inconvénient, les paliers de chaque côté de l'accouplement (celui du ventilateur et celui du moteur électrique) sont des paliers lisses dont la visite et le remplacement des coussinets en 2 parties ne nécessitent aucun démontage important. Ces paliers lisses sont à bain d'huile, à graissage par bague, avec chambres d'eau de refroidissement, compte tenu de la température ambiante relativement élevée pouvant régner dans le local.

Ces appareils, qui présentent d'excellentes qualités aérodynamiques et de robustesse, feront honneur, croyons-nous, à nos Établissements qui ont été très heureux de participer ainsi à l'une des plus belles réalisations françaises.



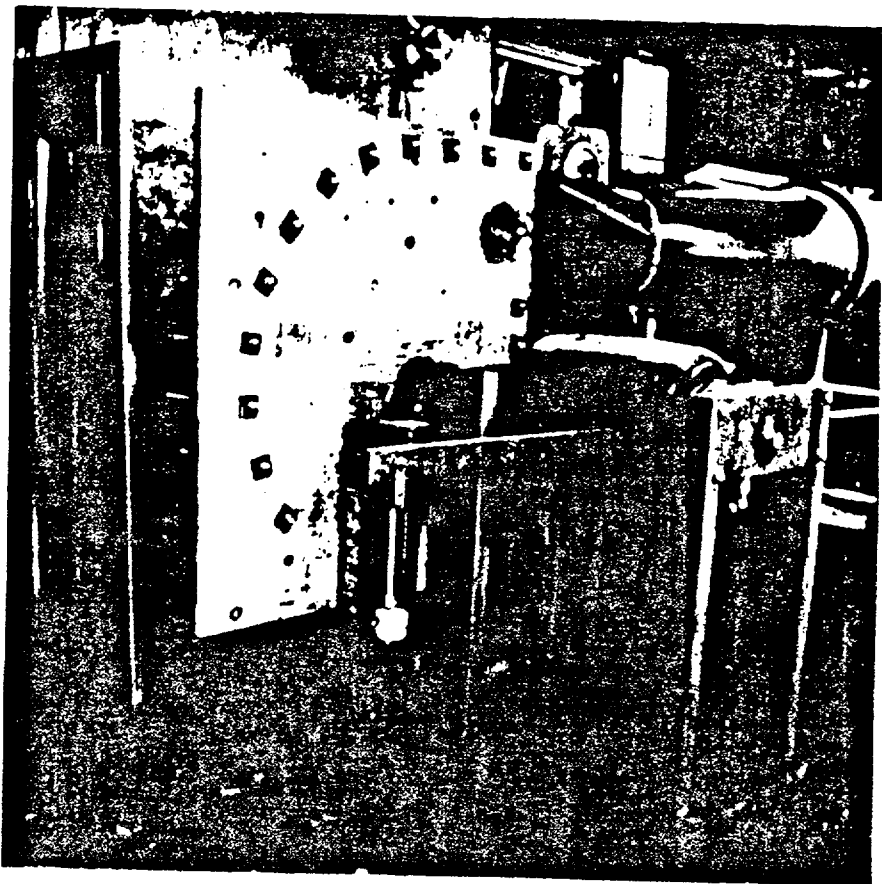
APPENDIX V d

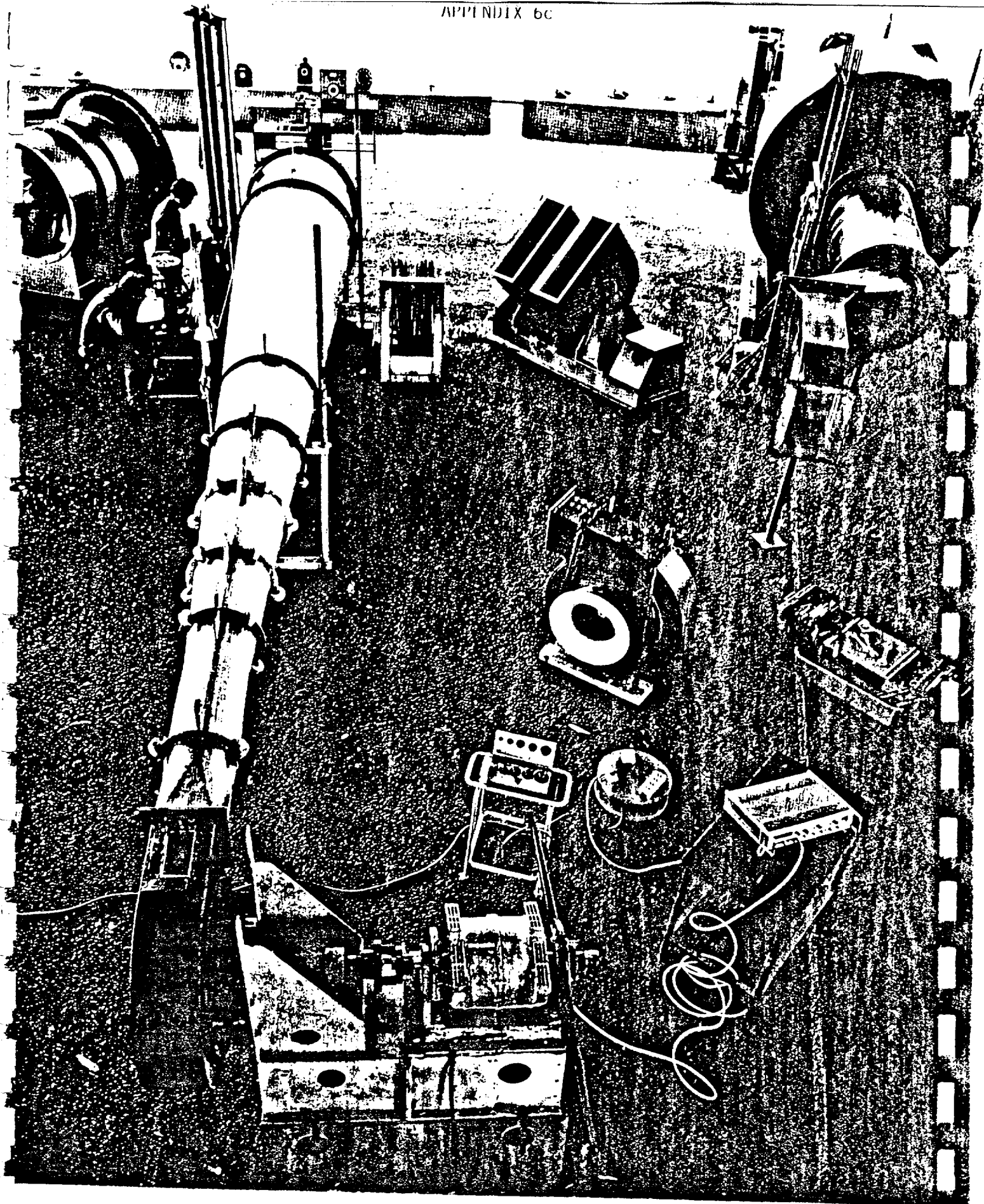


APPENDIX VI

PREVIOUS  
AEROPHYSICS AND NSRDC TESTS  
OF ACV/SES LIFT SYSTEMS

- o SWSI RD UNITS SHOWN:
  - . WITH AXIAL DYNAMIC GUIDE VANES
  - . WITH RIGHT ANGLE INLET CAISSON
  - . WITHOUT VOLUTE, DISCHARGING INTO  
PLenum WITH CIRCUMFERENTIAL  
EXHAUST VANES.





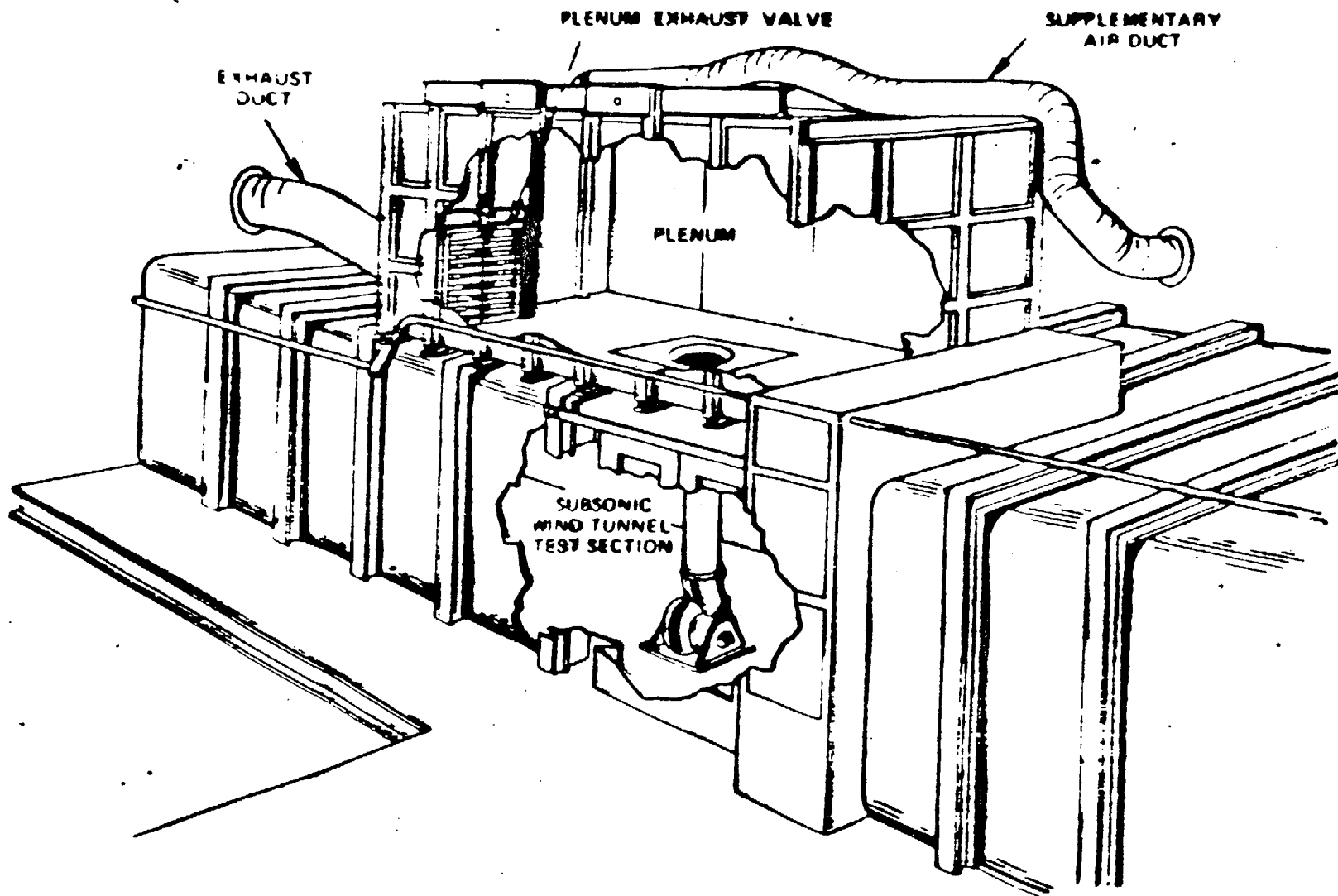
Testing a centrifugal fan at the Research Centre, Mincq en Batocul, France

## AERODYNAMIC PERFORMANCE

- HSRDC ROTATING DIFFUSER FAN MODEL TEST RESULTS.
- DISCUSSION OF FAN PERFORMANCE AERODYNAMIC COEFFICIENTS  $C_D$ ,  $C_P$ ,  $C_W$
- FINAL FULL-SCALE FAN CONFIGURATION SELECTION

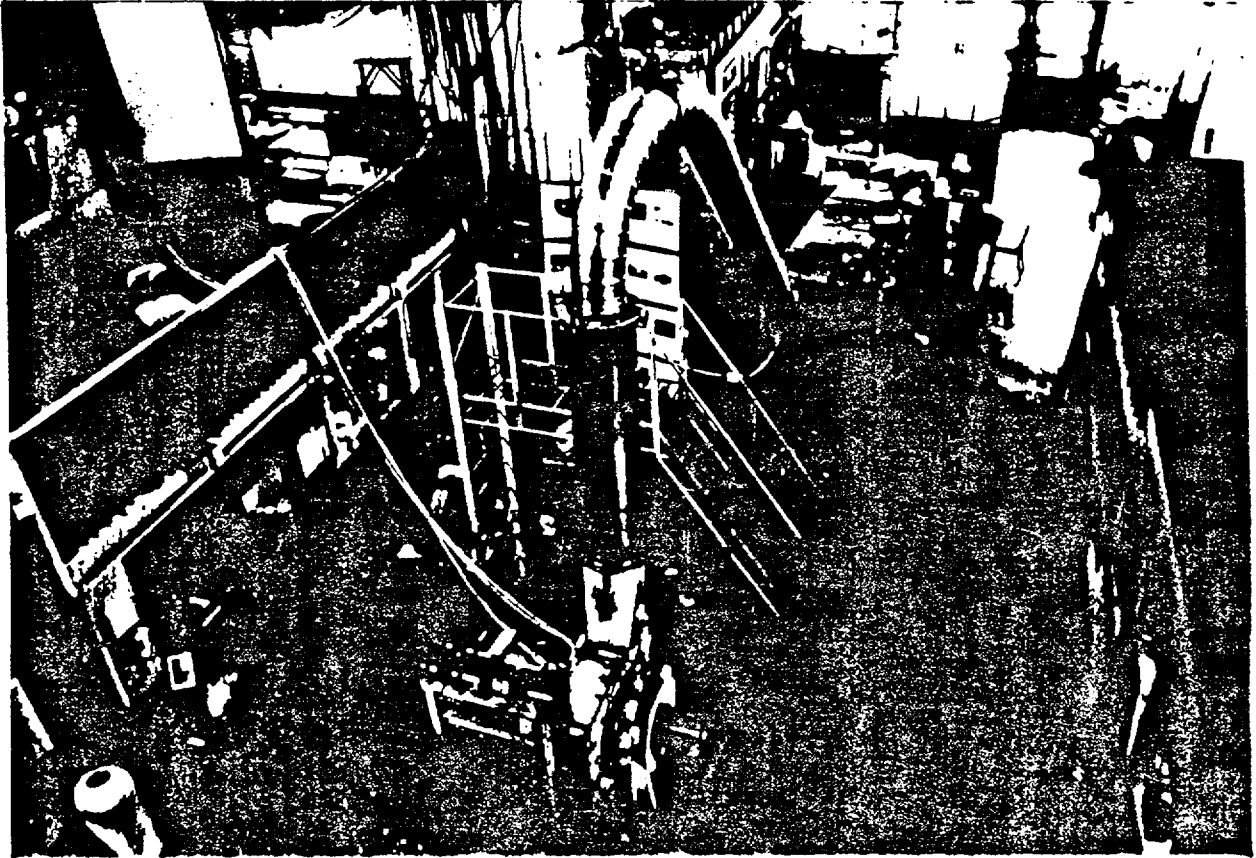
123

130

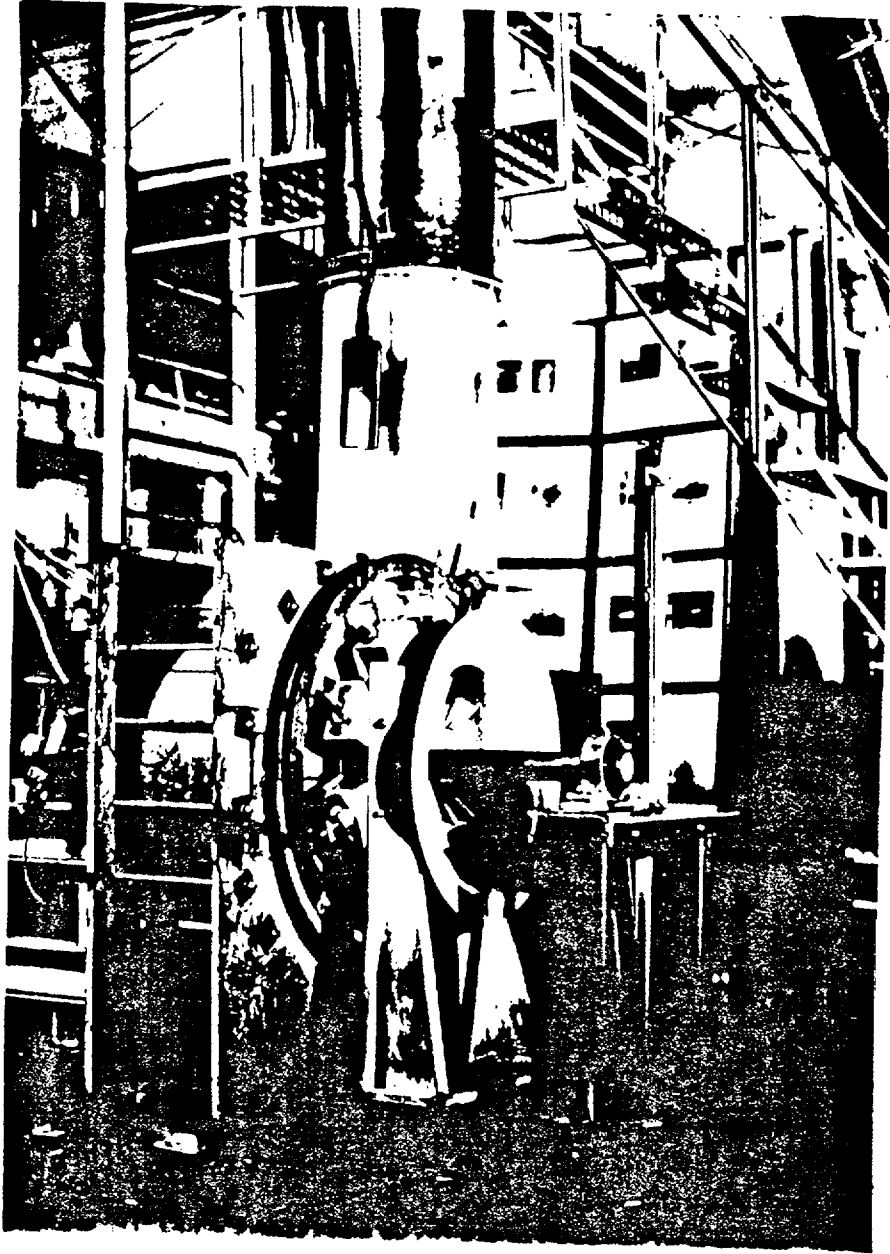


SES Lift Fan Evaluation Rig

APPENDIX VIIb



APPENDIX VIIc









APPENDIX VIIf

Figure 23 Regula Demand Flow Pressure versus Flow Rate  
 RUN 416 0.25 HZ LOW FLOW CLOSED LOOP

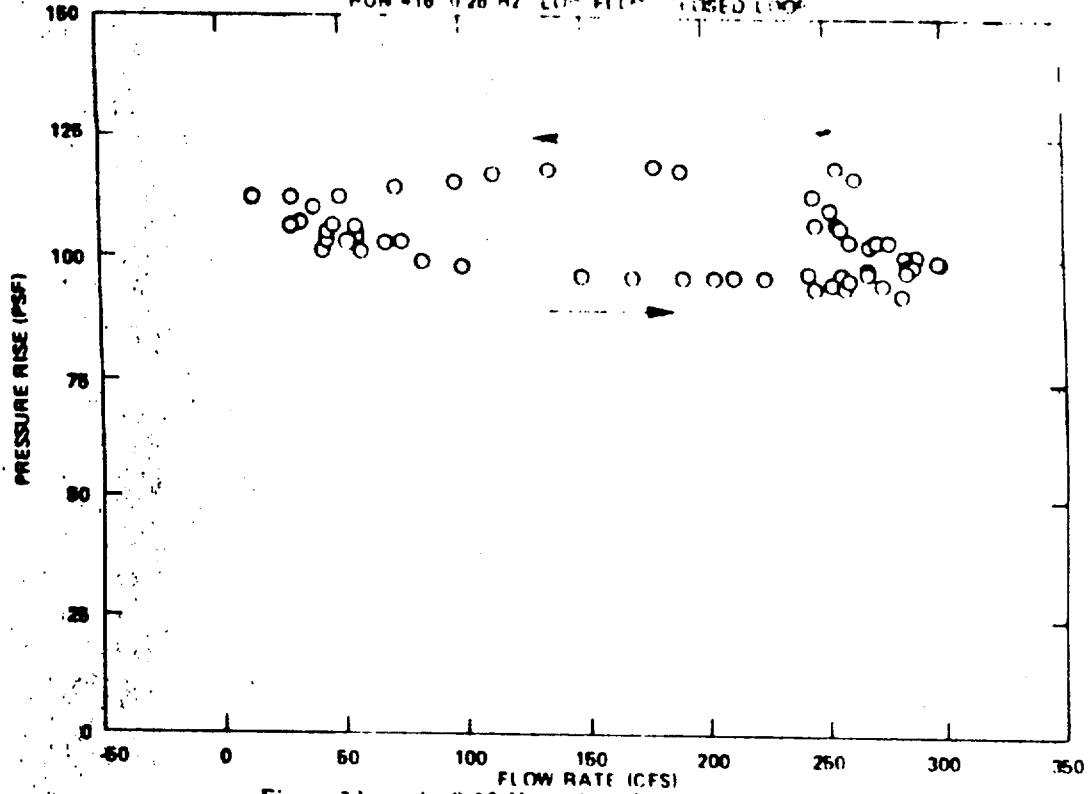


Figure 23a At 0.25 Hertz Low Flow Closed Loop

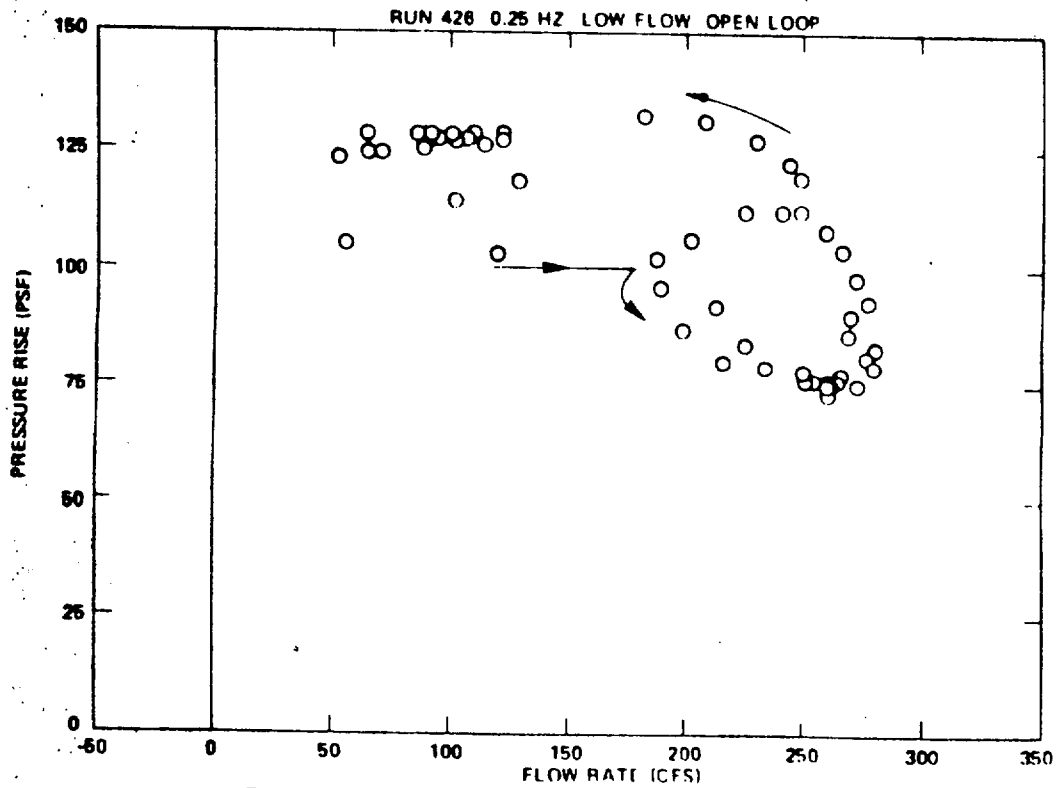


Figure 23b At 0.25 Hertz Low Flow Open Loop

PREVIOUS STATE-OF-THE-ART, FROM DTNSRDC REPORT 76-0073, JUNE 1976

APPENDIX VIIg

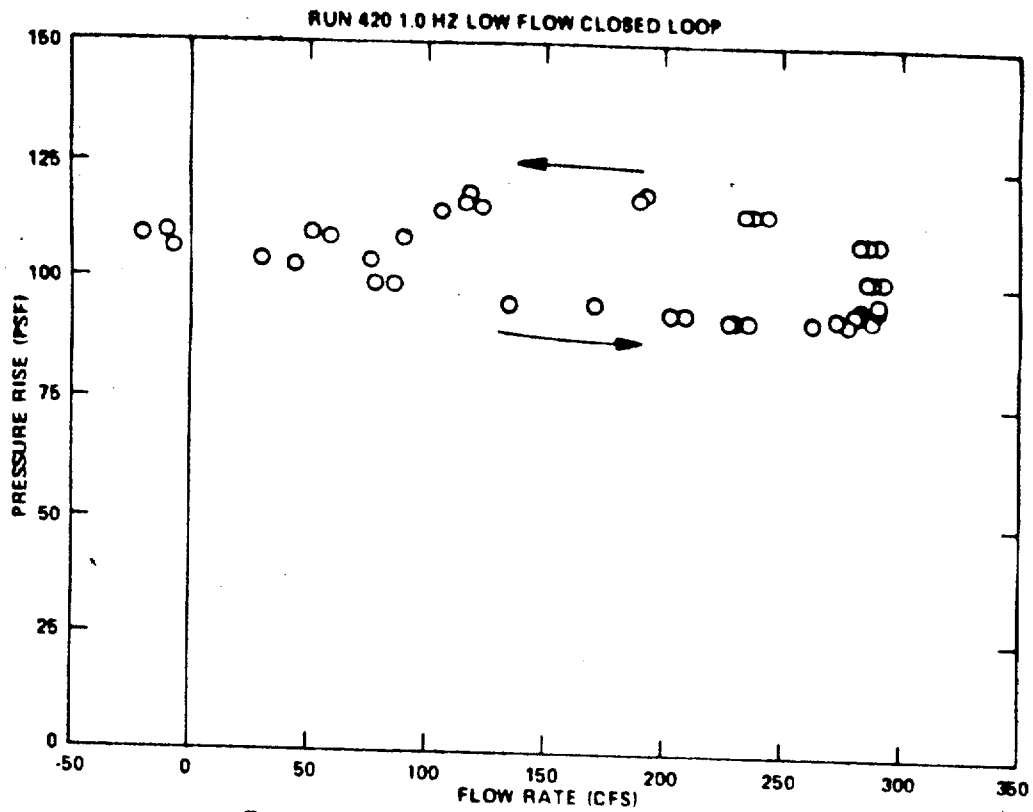


Figure 23c -- At 1 Hertz Low-Flow Closed Loop

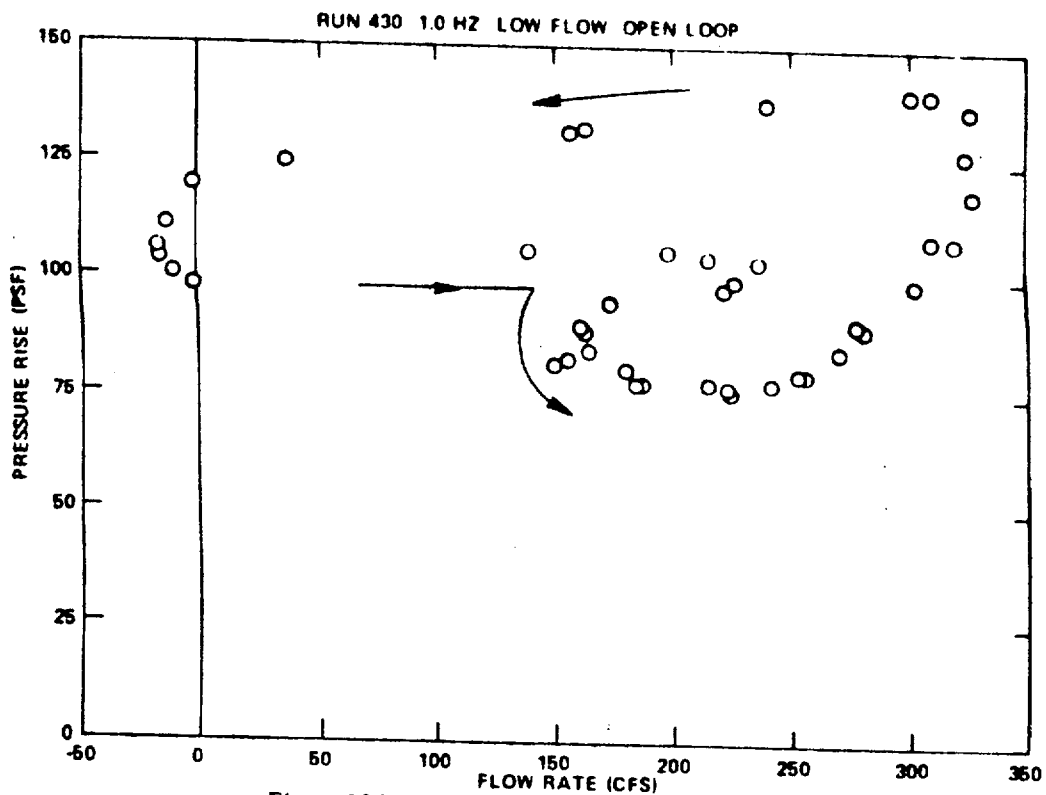
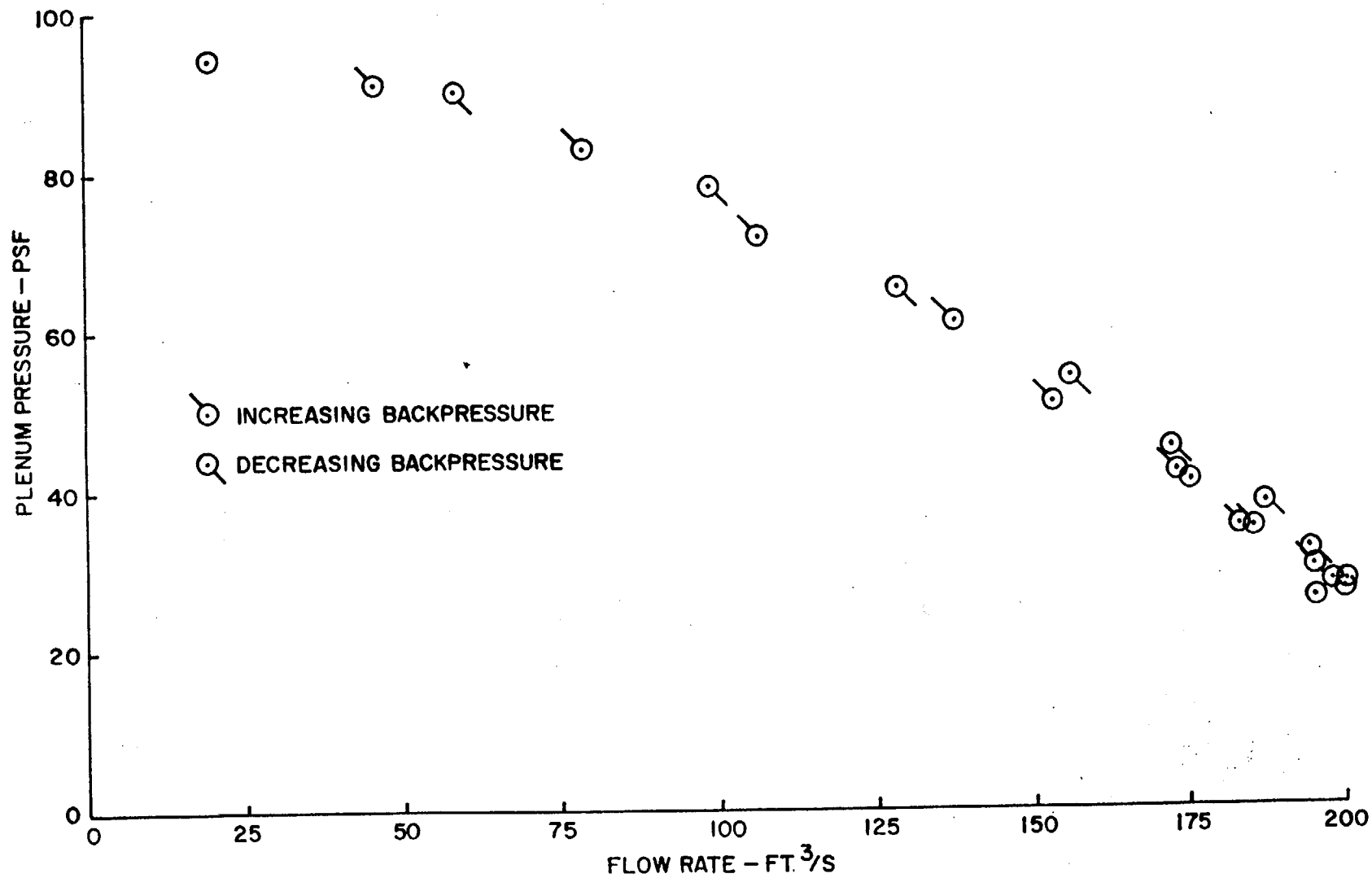
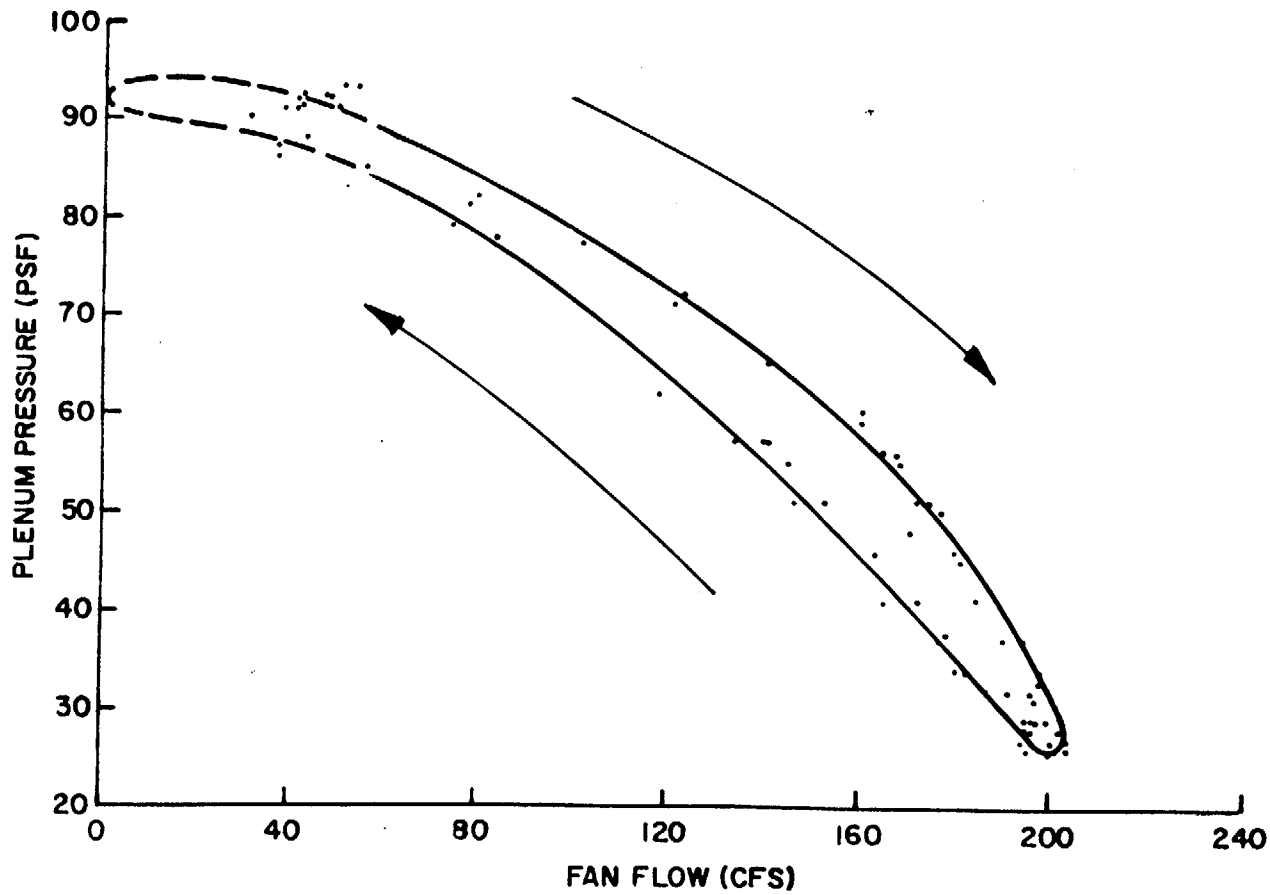


Figure 23d -- At 1 Hertz Low-Flow Open Loop

ROTATING DIFFUSER FAN MODEL TESTS NSRDC LIFT FAN  
EVALUATION RIG OPEN LOOP DYNAMIC TESTS.  
3120 PPM - 1.0 HZ BACKPRESSURE OSCILLATION



ROTATING DIFFUSER FAN MODEL TESTS NSRDC LIFT FAN  
EVALUATION RIG OPEN LOOP DYNAMIC TESTS.  
3120 PPM - .25 HZ BACKPRESS. OSCILLATION.

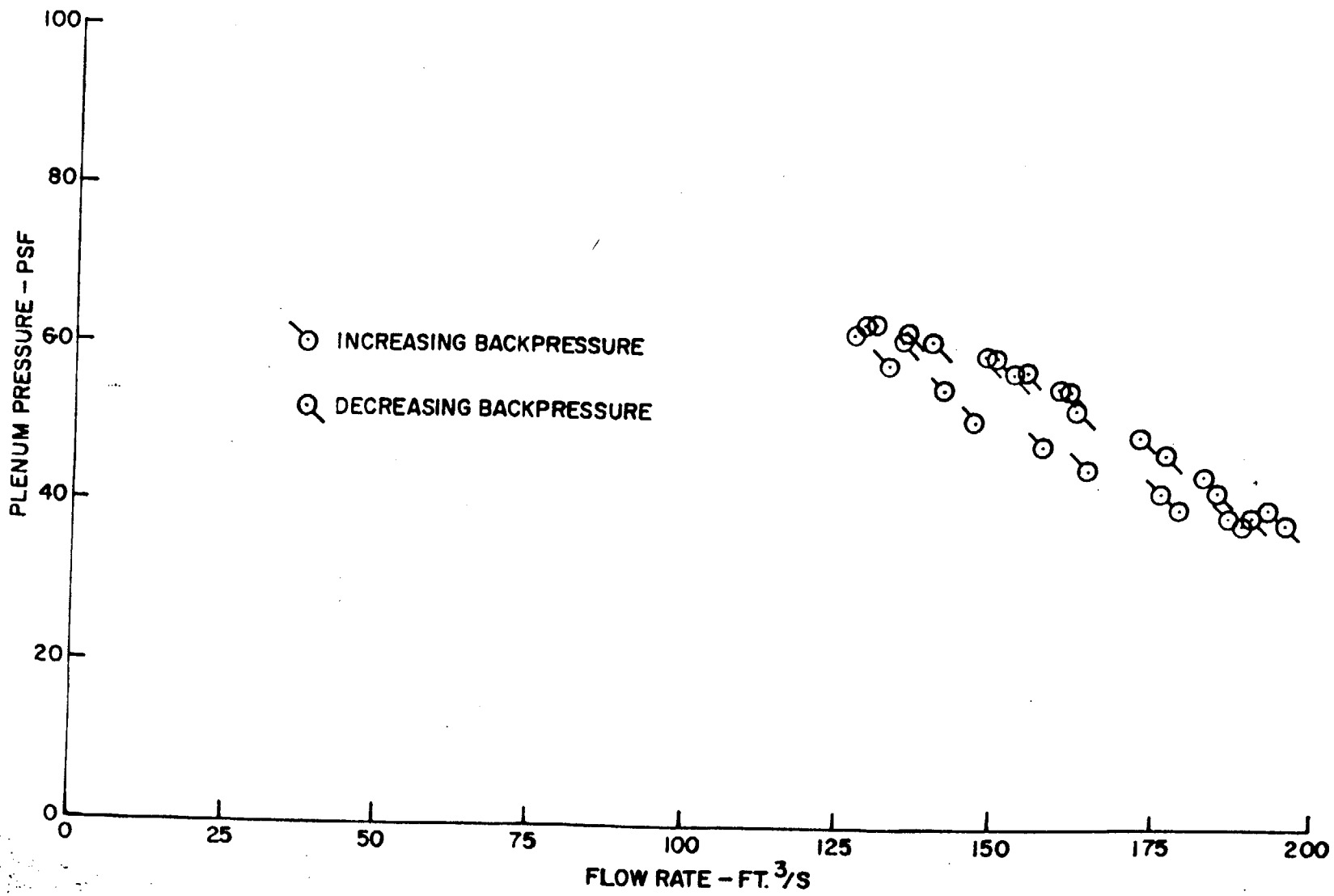


APPENDIX VII

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ROTATING DIFFUSER FAN MODEL TESTS NSRDC LIFT FAN  
EVALUATION RIG OPEN LOOP DYNAMIC TESTS  
3120 PPM, 4.0 HZ BACKPRESSURE OSCILLATION

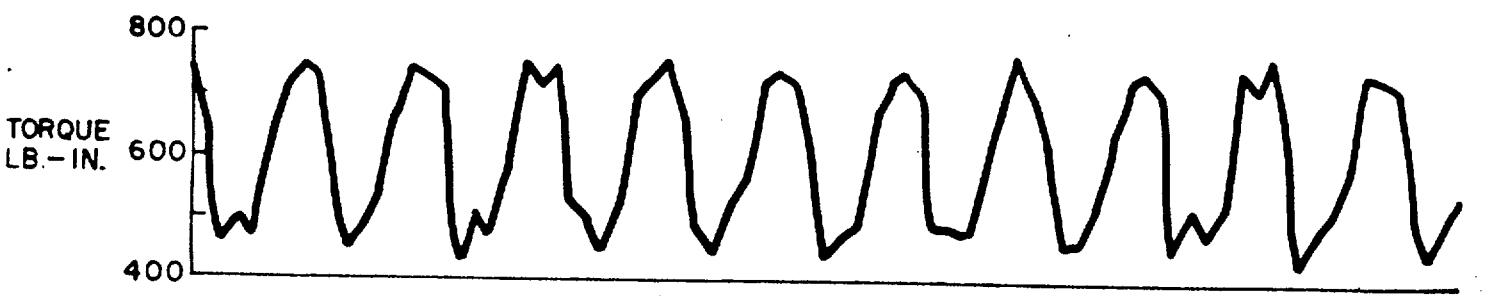
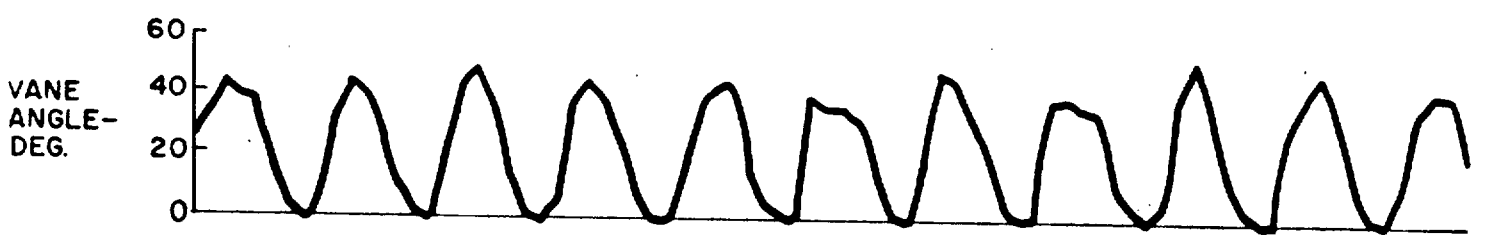
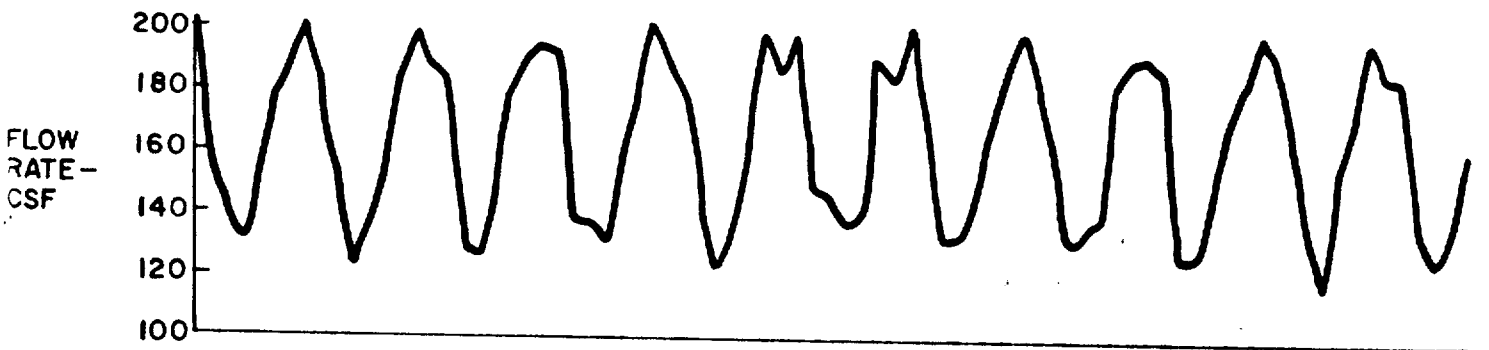
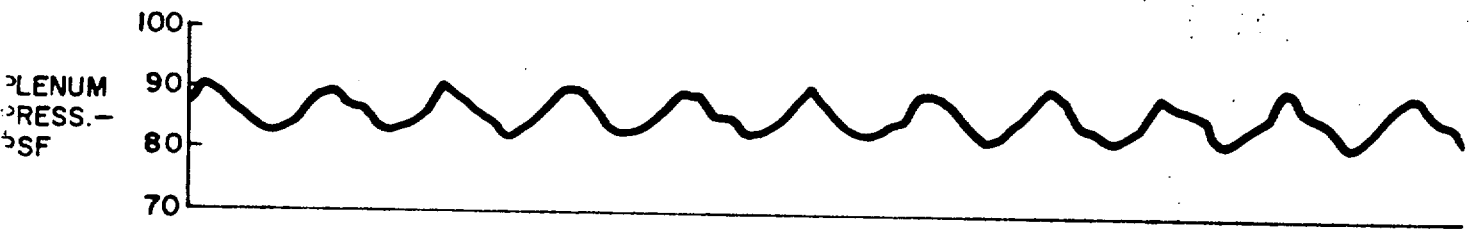
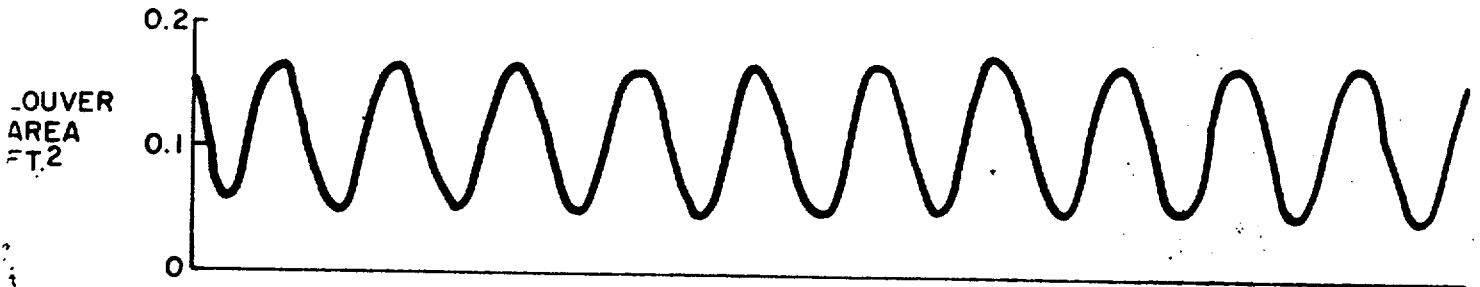
139



APPENDIX VIIJ

APPENDIX VIIK

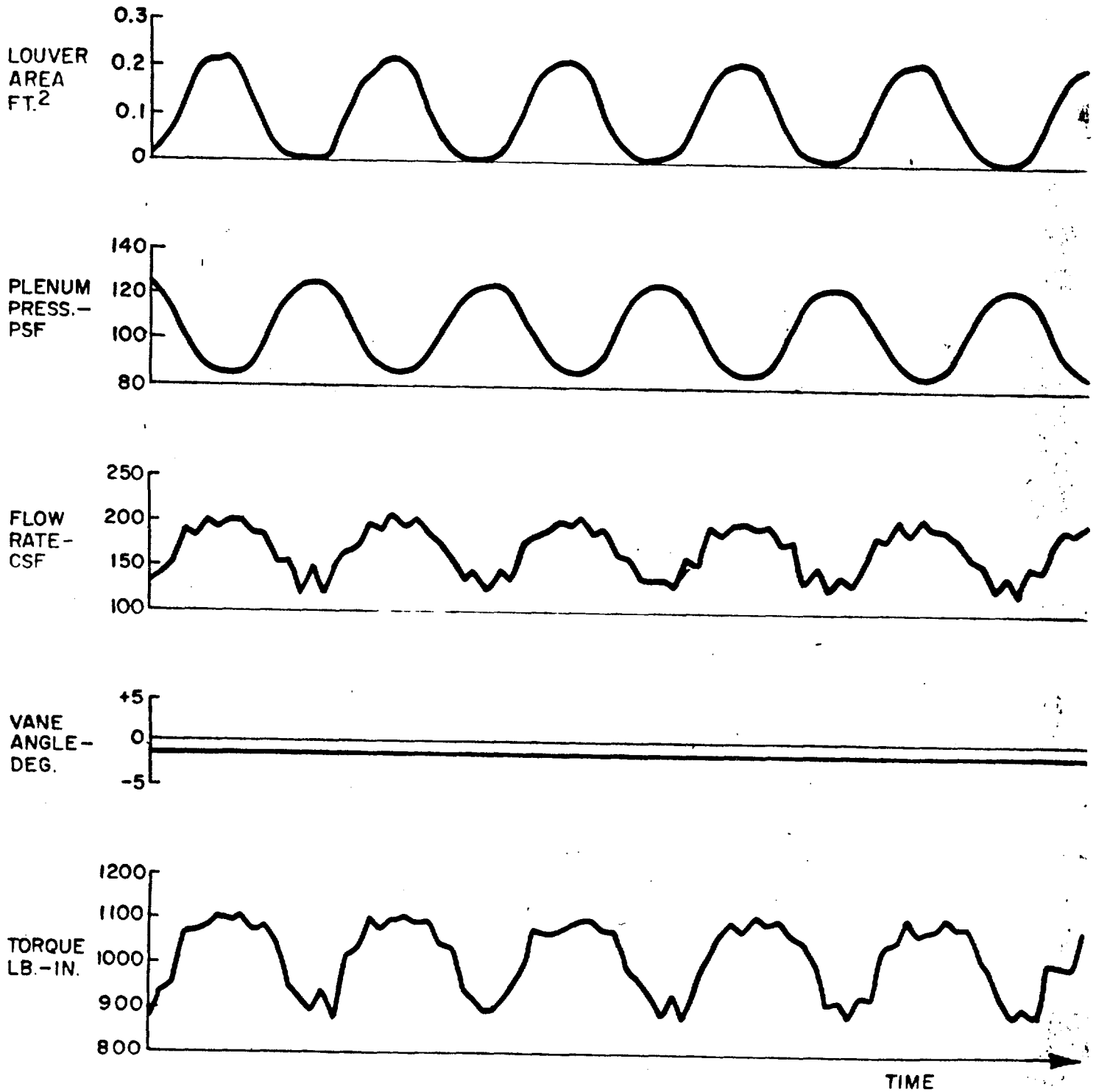
ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 569  
CLOSED LOOP TEST, 1.0 HZ





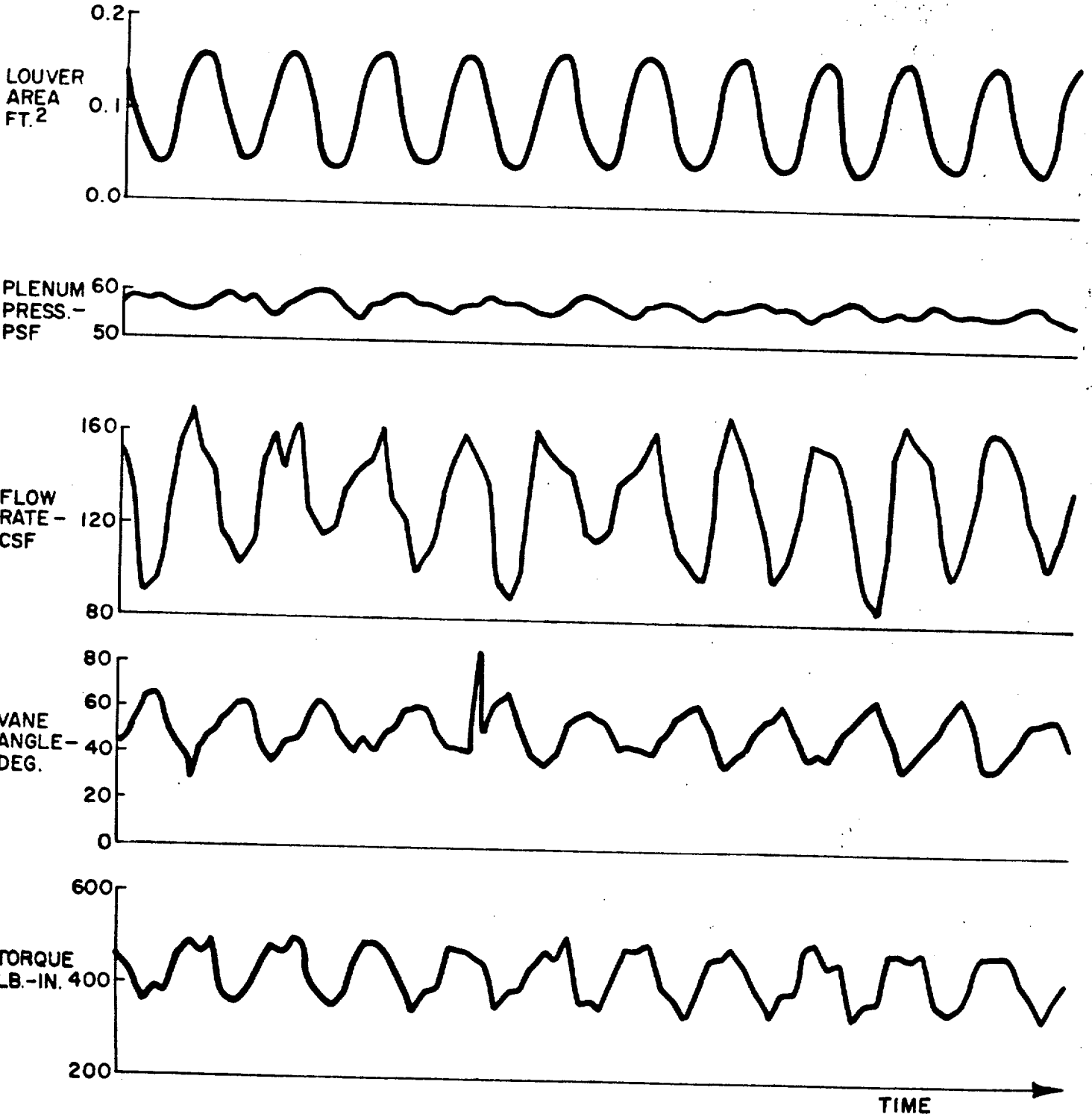
APPENDIX VIII

ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 573  
OPEN LOOP TEST, 0.5 HZ



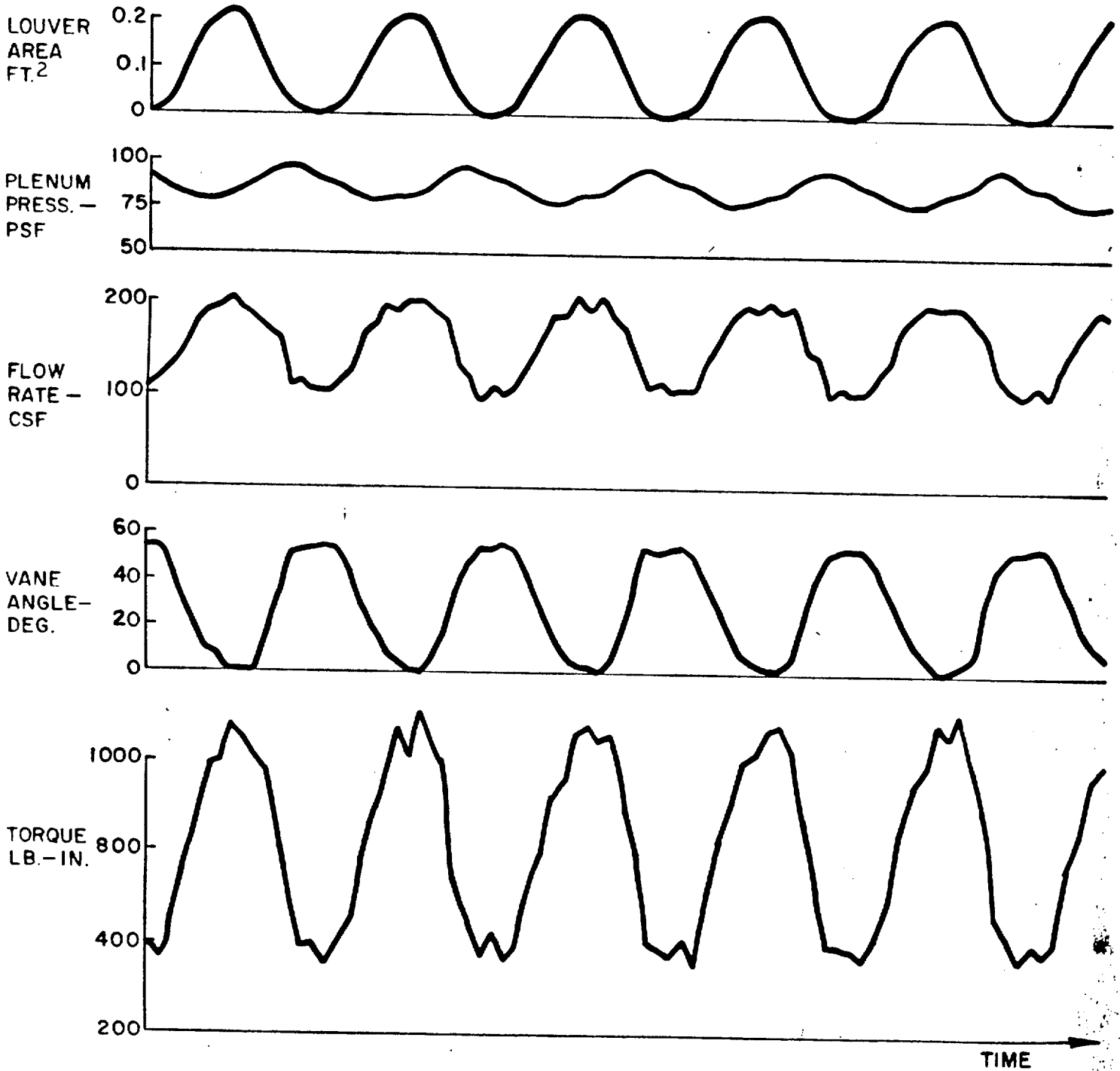
APPENDIX VIIIm

ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 566  
CLOSED LOOP TEST, 1.0 HZ



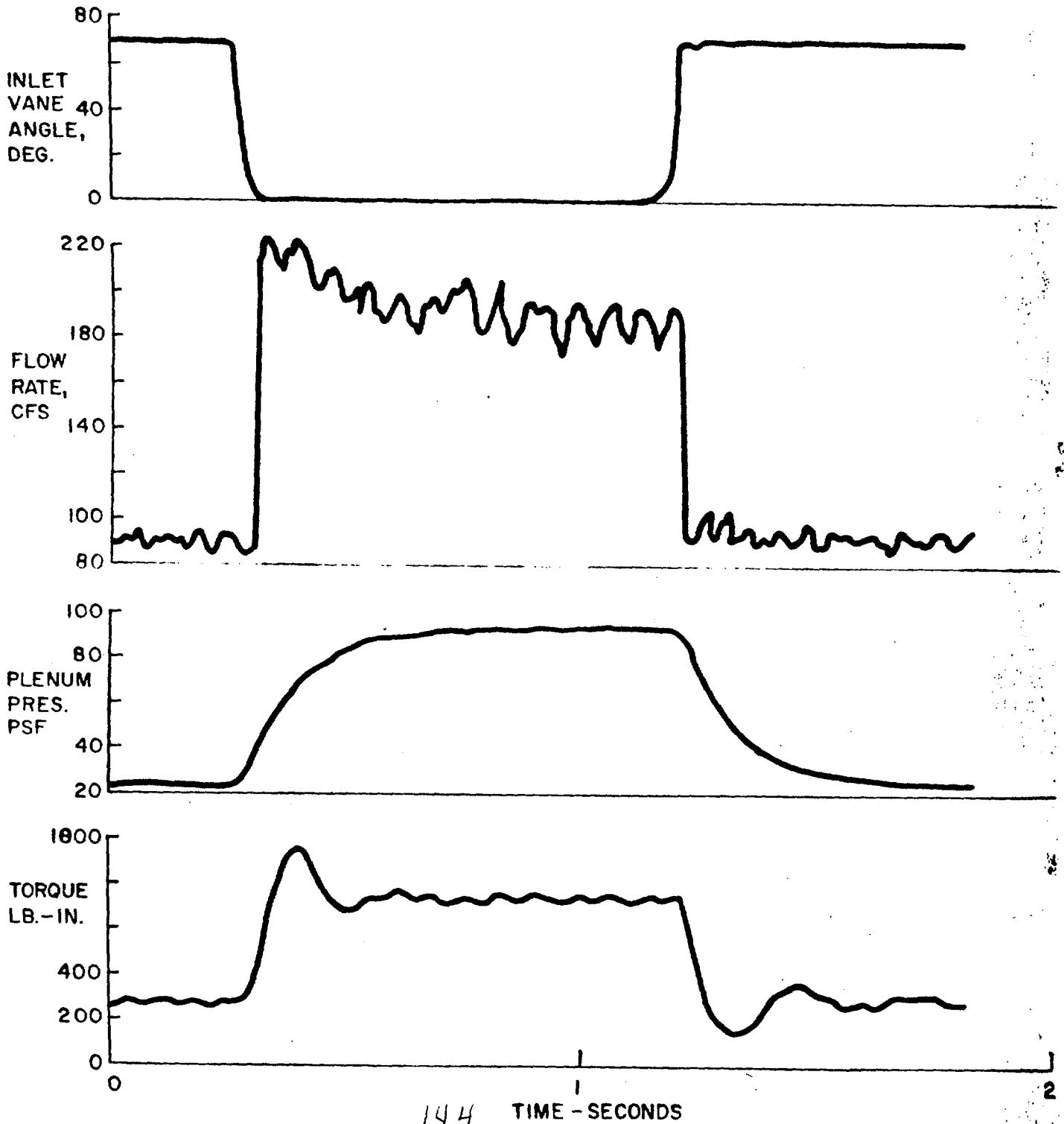
APPENDIX VI In

ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 577  
OPEN LOOP TEST, 0.5 HZ

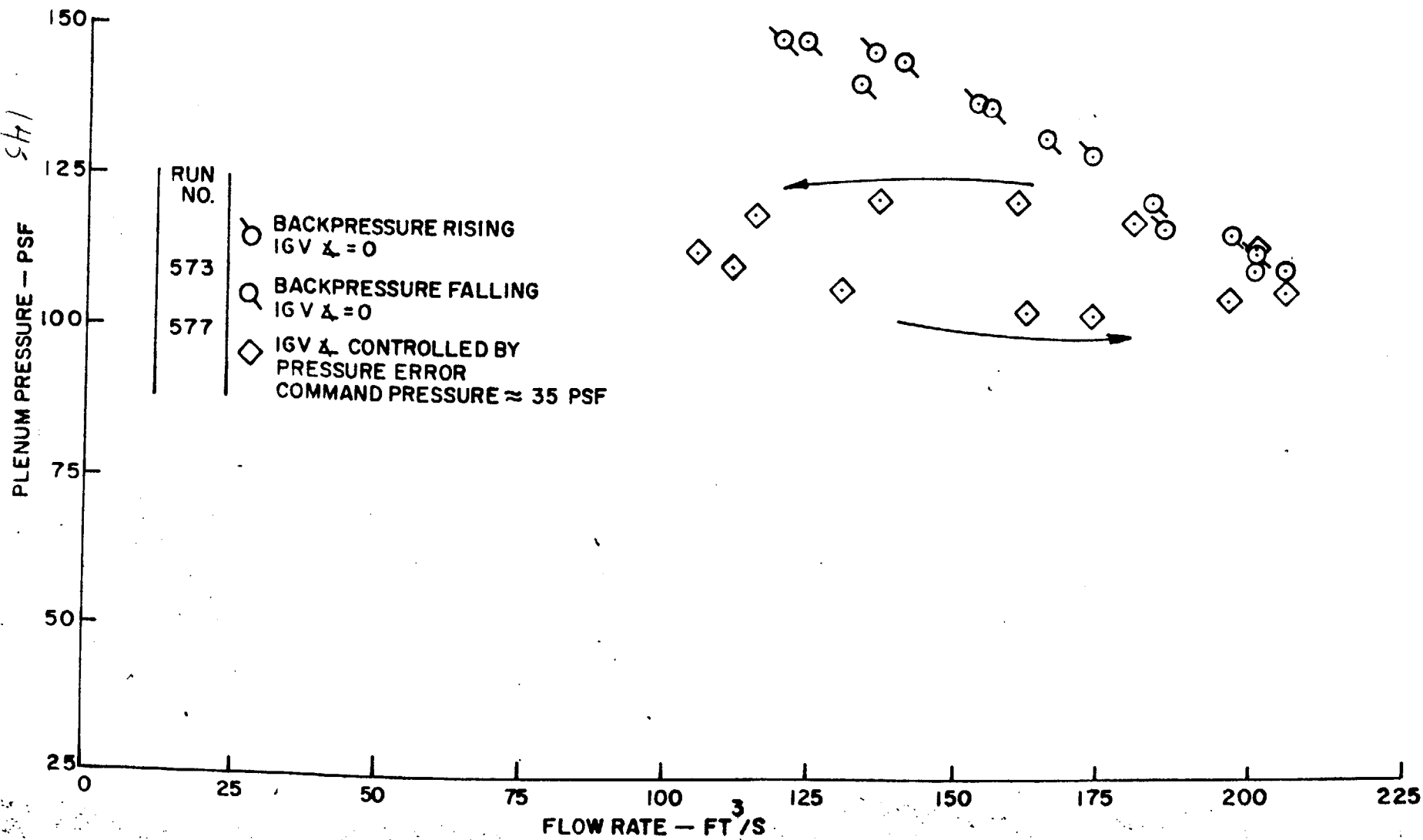


APPENDIX VIIo

ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 522, RESPONSE TO  
INLET GUIDE VANE ANGLE STEP FUNCTION, 0 DEG. TO 70 DEG.

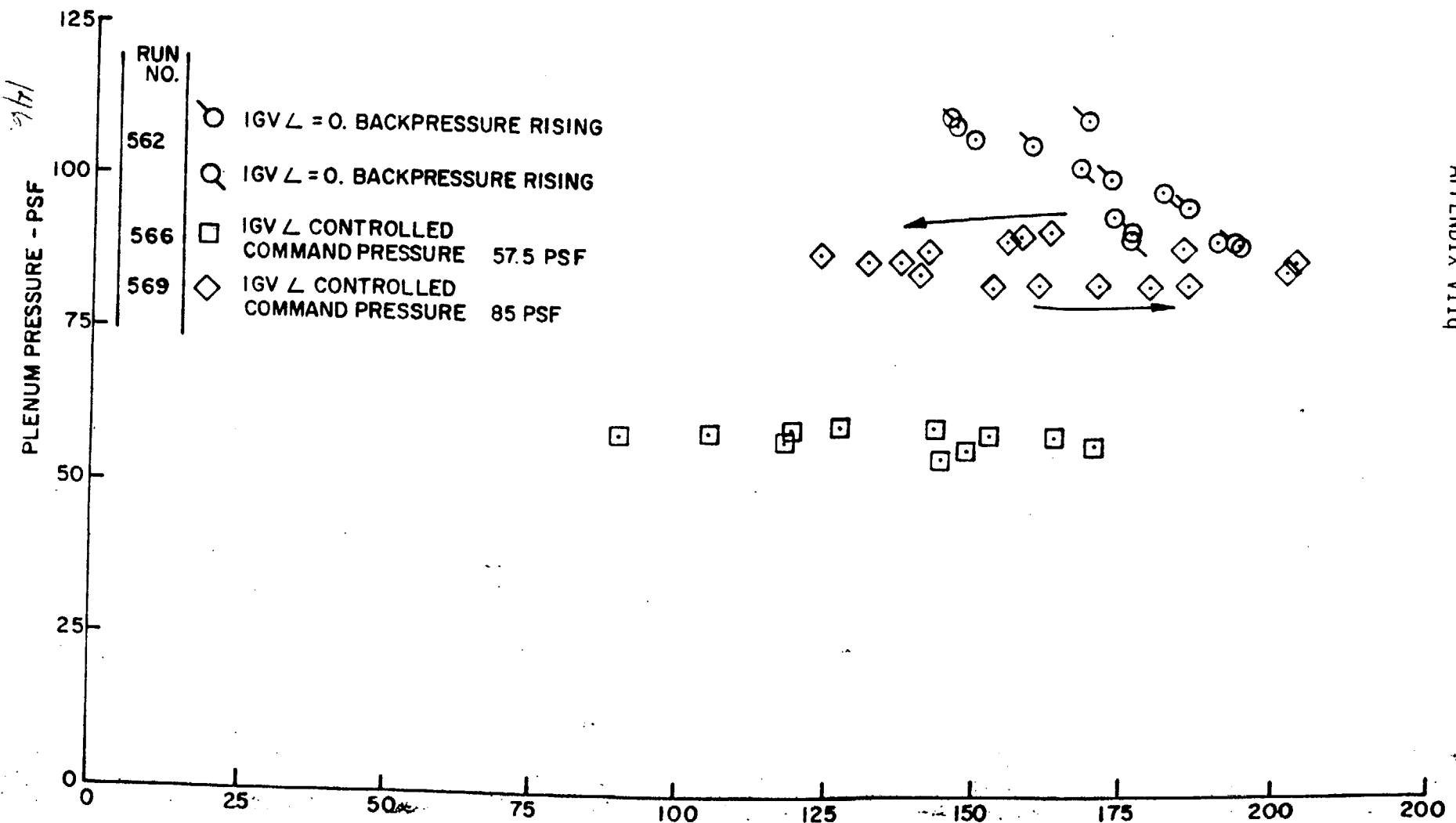


ROTATING DIFFUSER FAN MODEL TESTS NSRDC LIFT FAN EVALUATION RIG  
 OPEN LOOP AND CLOSED LOOP DYNAMIC TESTS  
 3800 RPM; 0.5 HZ BACKPRESSURE OSCILLATION, LOUVER AREA:  $0.105 \pm 0.1 \text{ FT.}^2$



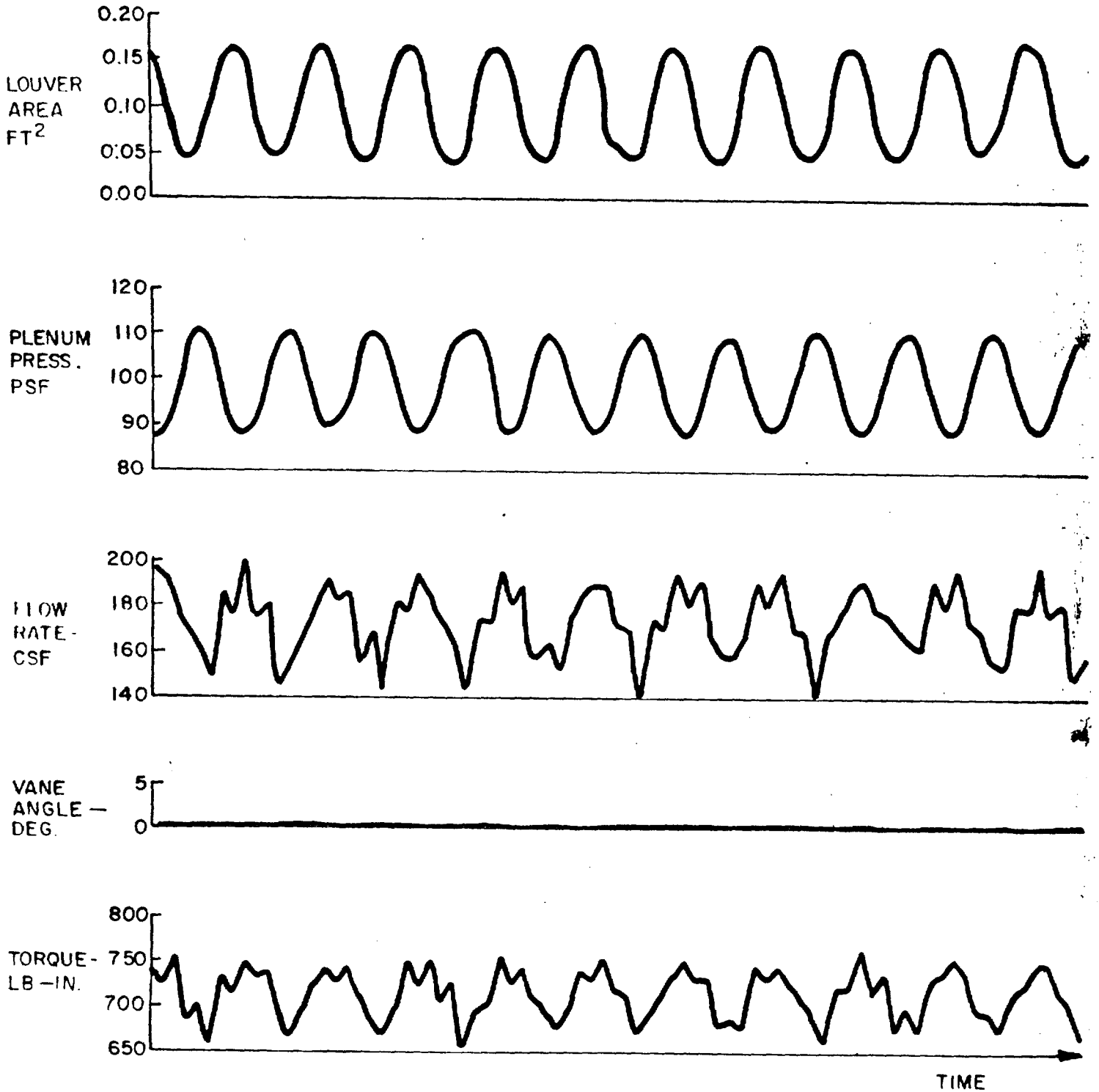
APPENDIX VIIp

ROTATING DIFFUSER FAN MODEL TESTS NSRDC LIFT FAN EVALUATION RIG  
 OPEN LOOP AND CLOSED LOOP DYNAMIC TESTS  
 3800 RPM, 1.0 HZ BACKPRESSURE OSCILLATION, LOUVER AREA:  $0.105 \pm 0.06$  FT.<sup>2</sup>



APPENDIX VIIr

ROTATING DIFFUSER FAN MODEL TESTS  
DYNAMIC TESTS, 3800 RPM, RUN NO. 562  
OPEN LOOP TEST, 1.0 HZ



## CONCLUSIONS FROM MODEL TESTS

- HIGH STEADY-STATE FAN EFFICIENCY AND HIGH EFFICIENCY OVER A WIDE RANGE OF FLOWS. SPECIFICATION EFFICIENCY OF 83% FULL-SCALE WILL BE EXCEEDED. AXIAL-FLOW INLET GUIDE VANES CAN REACH ANY POINT OF INTEREST ON THE PRESSURE-FLOW MAP.
- AERODYNAMIC BEHAVIOR OF FAN, WHEN SUBJECTED TO VARIABLE BACK PRESSURES DUE TO WAVE PUMPING, IS SMOOTH AND REPEATABLE.

148 THE ROTATING DIFFUSER IS AN EFFECTIVE "BUFFER" BETWEEN THE VARIABLE-PRESSURE AIR CUSHION AND THE FAN. PRESSURE-FLOW HYSTERESIS LOOPS ARE MINIMAL OR NON-EXISTENT. DYNAMIC MECHANISMS OF THE FLOW ARE NOT CURRENTLY UNDERSTOOD.

- ACTIVE DYNAMIC TESTS SHOW THE ABILITY OF THE AXIAL-FLOW INLET GUIDE VANES TO MAINTAIN CONSTANT PRESSURE IN THE SES CUSHION, IN SPITE OF WAVE PUMPING, AND ITS EFFECTIVENESS IS ONLY LIMITED BY THE IDIOSYNCRASIES OF THE ELECTRONIC CONTROLLER. INLET GUIDE VANES CAN PROVIDE AN EFFECTIVE RIDE CONTROL SYSTEM FOR THE 3KSES.