June 1967

PROBLEMS IN CRYOGENIC PUMP DESIGN FOR SPACE APPLICATION

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INTRODUCTION

Rocket engine development by its nature is very expensive. At the present time no single program exists for which expenses and necessary time to completion can be predicted reliably. Many phenomena, even experienced in components which have been used for many years, are not fully understood. This is true in spite of the fact that already over one-thousand hours space flight have been made successfully. Experience shows that it is impossible to predict all problem areas at the start of a program and that each program reveals new problems, mainly due to new requirements. Within the frame of this paper a critical discussion of some design concepts, as applied to cryogenic pump design, will be presented. These selected problems will, however, be restricted to those which are caused by the use of cryogenic fluids thus affecting the design of a turbopump. It will be shown that in order to arrive at a reliable turbopump configuration, several unconventional steps have to be taken which are not required for conventional pump design. These discussions will also show which areas are not yet fully understood and which special research work should be conducted in order to enhance reliability.

TURBOPUMP REQUIREMENTS

The turbopump is often referred to as the heart of the rocket engine and it is in fact one of the most important and difficult components of a liquid rocket engine. Its function is to deliver the propellants from the tanks to the thrust chamber at a given flow rate and pressure without major variations since for a successful mission thrust variations from engine to engine must be kept within 0.5 percent. Besides these requirements, the turbopump must be capable of starting to full power within a few seconds, must be capable of restarting in orbit, must be operational with low and high tank pressures without changing the pump head rise. Each of these requirements is of equal importance for the success of a space mission. Taking into account that these turbopumps have to operate at shaft speeds considerably in excess of those known for commercial applications because they have to supply high pressures with a minimum of weight and envelope, and that it is desirable to minimize auxiliary means (such as tank pressurization systems, bearing lube systems, etc.), it becomes clear that pumping cryogenic fluids is a difficult and completely new task.

It is the purpose of the following discussions to shed some light on these problems.

HYDRODYNAMIC CONSIDERATIONS

The hydrodynamic design of the fluid passages of pumps has been under study for many years. A large amount of experience was accumulated and used to refine the analytical methods. There are, however, major areas which distinguish cryogenic pumps for rocket engines from any commercial pump. The easiest way to illustrate these problems is by asking which operating conditions are imposed on the turbopump of a rocket engine during start, steady state, shut off and restart in orbit. The requirement to start the engine within two to four seconds entails the control problem of achieving a gradual thrust buildup. Gradual thrust buildup minimizes structural problems within the vehicle and eases the pump cavitation problem. Other mechanical problems during the transient and steady state period are bearing axial loads, pressure and temperature environment for the rotating seals, rotordynamics and temperature gradients in the housing with their inherent effect on stress and vibrations.

During main stage operation a smooth and stably operating pump is desired. Flow oscillations at pump inlet and discharge are of major concern - especially for pumps generating high discharge pressure, combined with minimum tank pressures that is low NPSH. Pump oscillation at the inlet are of major concern for the "Pogo" problem; meanwhile, pump discharge pressure oscillations are a problem with respect to the structural integrity, vibration level and eventually even combustion instability, which may be triggered by the flow oscillations transmitted through the feed system to the thrust chamber.

The third area which may influence cryogenic pump design is the requirement to restart the engine in orbit. This, according to the planned mission, may be required at any time; a few seconds, or up to many hours after the vehicle has been placed into orbit. The problem arising from this condition is caused by the temperature distribution throughout the turbopump. Reliable restart requires a properly temperature-conditioned pump and system; however, shortly after the engine has been shutdown and the vehicle placed into orbit, the heat stored in the turbine will soak back into the pump. The liquid in the pump will vaporize and even the liquid in the inlet duct will slowly be converted to vapor. After many hours of coasting in orbit, the turbopump will have some average temperature around which the actual temperature oscillates, being higher while passing through the sun, being lower while passing through the shadow of the earth. Calculations, which are naturally highly dependent upon the assumptions made, show that conditions similar to that illustrated in Figure 1 can be expected. From this we can conclude that after a certain time in orbit, no liquid hydrogen or oxygen will be in the pumps or inlet pipes. Restarting under these conditions, therefore, requires special procedures with special controls. The problem can be eased somewhat by selecting the right materials, using internal and external insulation and unconventional start sequences.

As can be seen from this brief discussion, restarting a pump fed engine is a very complex and difficult problem. Within the frame of this paper, however, only those restart problems will be discussed which influence the turbopump design.

Pump Impeller Design

Figure 2 shows two possible impeller designs, both of which meet the performance required by the engine, such as flowrate and pump head rise. In Figure 2a, an open-faced impeller is shown; meanwhile, in Figure 2b a shrouded impeller is used. For the design of Figure 2a, a flow coefficient at pump inlet ($\phi = \frac{U}{C}$) of 0.3 was selected; meanwhile, in Figure 2b, $\phi = 0.1$ was chosen. Consequently, the essential features which distinguish the two designs are the pump inlet flow coefficient and the shrouded or unshrouded impeller. The superior design for rocket engine application is the one shown in Figure 2b. The reason becomes clear if we analyze the conditions under which the pump has to operate. Pumping cryogenic fluids to high pressures makes the control of the clearance between impeller and housing very difficult, since housing deflections occur as a function of internal pressures, the temperature effect being superimposed. A controlled clearance, however, is desirable from the efficiency and performance repeatability point of view from pump to pump. A shroud avoids these difficulties since no closely controlled impeller positioning to hold the clearance is required. Housing deflections and deformations due to pressure and temperature become less important and have only to be considered from the stress point of view. An additional advantage of the shrouded impeller is the reduction of the axial thrust, a feature especially important for bearings operating in LOX or in LH_2. Such bearings cannot withstand high axial loads and high speeds at the same time. From the reliability point of view a shrouded impeller has the advantage that impeller rubbing against the housing is avoided completely, which is a necessary design requirement when LOX is being pumped.

The second feature of importance is the selection of a low pump inlet flow coefficient (Reference 1 and 2). In order to understand the effect of the flow coefficient on suction performance we have to go back to the basic theory which relates suction performance to the flow coefficient

$$\frac{S_{max}}{\sqrt{1-\xi^2}} = \frac{8150}{\phi_{opt}} \left(\frac{1-2\ \phi_{opt}^{\ 2}}{3} \right)^{3/4}$$

Where: S_{max} the maximum suction specific speed

ξ the ratio of inducer hub to tip diameter and ϕ the flow coefficient is. Figure 3 shows this relationship graphically.

Up to a certain flow coefficient the experimental results agree very well with the theoretical prediction. However, the theory does not agree with the experimental results when the flow coefficient becomes too small. The reason is the blockage effect due to physical blade thickness which, when the blade angle becomes too flat, restricts the flow passage area too much. Under these circumstances the obtainable suction performance will even deteriorate. Some systematic tests investigating the effect of blade geometry on suction performance have verified these theoretical predictions. Figure 4 shows as an example the results of such a test series. The obtainable cavitation number,

$$k = \frac{p_s - p_v}{\frac{w^2}{2g}}$$ where p_s = static pressure, p_v = vapor pressure, w = relative

velocity; is shown as a function of the incidence angle which is the difference between flow angle and blade angle. The flow coefficient and the blade angle are shown as parameter. As can be seen, a grid of straight lines is generated by the investigated inducer family which describes completely the performance of the inducer type studied and which allows to select the flow coefficient and inlet blade angle for a given requirement. It must, however, be emphasized that the plot obtained from this test series is only good for the inducer family investigated. A different design for instance with a different number of blades will shift the grid to the right if more blades or to the left if less blades are used. This shift is again mainly caused by the blockage generated by the physical dimensions of the blades. The fact, however, remains that the portion of the graph in which the lines of constant flow coefficient and constant blade angle are straight, indicate the range in which suction performance is not affected by the physical blockage of the blades. If the blockage effect impairs suction performance, the straight lines representing constant flow coefficient and constant blade angle become curved shifting the flow coefficients to higher cavitation numbers (see Figure 4). From the graph it can be seen that the selected incidence angle, that is the angle which is formed by the blade angle and the theoretical flow direction at the inlet, has the largest influence on obtainable suction performance. For all practical purposes, a flow coefficient of $\phi = 0.1$ will result in a good suction performance. The incidence angle should be selected to about 0.4 times the blade angle, the blade angle being measured normal to the axis of rotation. These findings are in agreement with the basic theory (Reference 2) which also predicts a ratio of approximately 0.4.

Returning to Figure 2 it is now evident why, for rocket engine application, the design shown in Figure 2b should be selected.

THE UNSTEADY FORCES

Low Frequency Pump Inlet Flow Oscillations

Two types of unsteady forces are of concern: The low frequency pump inlet flow oscillations and the high frequency pump discharge flow oscillation. Low frequency inlet flow oscillations of varying strength are always generated when higher suction performance inducers are used. These oscillations are a function of inducer geometry, inlet flow conditions, such as pressure and temperature, and the properties of the liquid to be pumped. They can, therefore, become significant at any point during the flight, whenever resonance conditions are just right to amplify this effect. If the frequency of the flow oscillations for instance tunes with the natural frequency of the structure, such an amplification can be dangerous for a manned flight. This problem was not recognized until actual flights were undertaken, in the course of which difficulties appeared, jeopardizing the purpose of the flight. As a consequence, considerable research efforts were initiated in both areas: 1) To understand the phenomenon and, 2) to find means to reduce the inducer generated oscillations. In spite of these efforts, no complete understanding of this problem exists as yet. Neither the cause of these low frequency unsteady forces generated in a cavitating inducer, nor their effect on the vehicle itself are fully explained. However, from observations made during water tunnel tests it can be said that the inhermntly unsteady phenomenon of cavitation is the basic reason for these unsteady forces. Considerable effort was expended to find an inducer design which minimizes these oscillations. No reliable cure was found and this field is still wide open for research work. Basically, these investigations showed that any attempt to avoid the low frequency flow oscillations resulted in a reduction of obtainable suction performance. The basic behavior of a conventional inducer is illustrated in Figure 5 through 7, which show the test results obtained from two inducers A and B. In the design of inducer A a higher incidence angle was used than in the one of the inducer B. As can be seen from these test results, an oscillation reduction is experienced with lower incidence angles and with flow coefficients higher than design flow coefficient. Figure 5 and 6 depicts two flow coefficients for comparison of the two inducer designs.

At the present time the following conclusions can be drawn:

1. Solidity, that is the ratio of cord length over blade spacing, is important. A low solidity, for instance $\delta = 1$, will produce violent flow oscillations because the cavity may not collapse inside the blade passage under all operating conditions. Increasing the solidity δ to 1.5 to 2 will reduce the violence of the disturbance considerably - the flow will become more stable. The oscillations, however, are still there.

2. The amplitudes of the flow oscillations increase approximately linear with increasing inducer tip speed.

3. Flow coefficients lower than design tend to increase the oscillation amplitude because the incidence angle is increasing (Figure 7).

4. From inducer to inducer no repeatable trend could be detected with changing NPSH.

An additional difficulty to understand, explain and predict is the phenomenon that arises from the fact that conclusions based on experimental programs using water as the test fluid will alter when cryogenic fluids are being pumped. It is well known that suction performance of a given inducer changes considerably with fluid properties (Reference Figures 3 and 4) because the shape and size of the cavity is a function of these properties. Since low frequency oscillations are also a function of the cavity size, it is clear that changing properties will influence the low frequency oscillations. This is also an area in which more research is needed.

<u>High Frequency Pump Discharge Flow Oscillations</u>

The second type of unsteady forces experienced in high pressure pumps are the pump discharge oscillations. Since pressure requirements for rocket engine turbomachinery are constantly increasing, the unsteady forces are gaining more and more importance. With the increasing flow velocities, it must be expected that the mechanical stresses resulting from the unsteady forces must be taken into account. All flow or density variations will cause unsteady flow phenomena. The effect of these variations can easily be visualized when viewed from a stationary system. The circumferential variations, normally described as circulation, are inherently connected with the transmission of circumferential forces by the rotating impeller. Downstream of any rotating vane an unsteady traveling wake interacts with the stationary vane system as illustrated in Figure 8. It can easily be seen that the effect of these wakes on the stationary object will increase the closer the stationary vane system is to the rotating system and the higher the workload or blade loading of the individual blade will be. It is also clear that the generated frequency is a function of the number of rotating vanes. We, therefore, have a possibility to avoid resonance between the forcing function and the natural frequency of the pump discharge pipe system by selecting the appropriate number of blades.

Another phenomenon to be considered is the wave reinforcement. As mentioned above, the discharge oscillations are a function of the strength of the traveling wake passing by a stationary vane system. This vane system can be a vaned diffuser or two tongues of a double exit volute (Figure 9). In determining the number of blades, it is important to avoid wave reinforcement by allowing the two waves generated at the two tongues and traveling with acoustic velocities to meet each other in phase at some place downstream in the discharge pipe. This phenomenon is illustrated in Figure 9. The wake of vane 1 interacts with the tongue at point A and a wave travels with acoustic velocity in the time t_3 to point C. In the meantime, the wake from vane 2 interacts after the time t_1 with the tongue at point B and reaches C after a time t_2. If $t_1 + t_2 = t_3$, wave reinforcement occurs. It can easily be seen that such a phenomenon can be corrected by changing the number of blades or position of the tongues.

For the design of the pump, it is also important to know how strong these oscillations are transmitted throughout the pump. For reliability reasons it is, for instance, essential to have the rotating shaft seal in an environment which treats the seal as gently as possible. Measurements made on a pump, as shown in Figure 10, revealed that the predominant frequency can be found throughout the pump. However, the amplitude is a function of the local static pressure and the operating conditions. In other words, the amplitudes decrease and increase with the static pressure using the amplitude measured in the volute as reference. Figure 11 and 12 illustrate as an example the amplitudes found during these tests at the locations a and b. It is interesting to note that the amplitudes found for the design point (Figure 11) are considerably smaller than the ones for the off-design considerations (Figure 12), which underlines the importance of a correct pump impeller design. It follows, furthermore, that in designing a pump, an arrangement has to be selected which guarantees a low static pressure in the seal environment and in all locations in which parts susceptible to fatigue are installed. The lower limit of this static pressure is set by the vapor pressure since it is not desirable to generate cavitating conditions in these areas. Summarizing it can be said that in order to reduce the amplitudes of the inherently present pump discharge oscillations the hydrodynamic design of the impeller should be based on the following recommendations:

1. Use only moderate blade loadings, avoid eddies in the blade passages

2. Avoid resonance phenomena in the discharge pipe system

3. Avoid wave reinforcement

4. Reduce the static pressure in areas sensitive to flow oscillations.

MECHANICAL CONSIDERATIONS

In this chapter some mechanical problems will be discussed which are unique for cryogenic pump design because they are caused by the fluids to be pumped. The discussion will be limited to LOX-RP1 and LOX-LH_2 systems since they are the most important for space flights.

Rotating Seal Arrangement

Figure 13 shows a bearing and seal arrangement of a LOX pumpl In this case the bearing is separated from the LOX pump by a seal package so that it may be lubricated by a separate lubrication system. The seal package consists of two rotating (A and C) and one shaft riding seal (B) which is purged in order to assure safe operation even if one rotating seal fails. As a consequence, a space consuming design results with a large overhang of the LOX impeller entailing critical speed and clearance control difficulties. Such a LOX pump arrangement is dangerous since rubbing between the rotating and stationary parts due to shaft deflections may trigger an explosion. If, however, a LOX lubricated bearing located as close as possible to the impeller is used and the seal package is installed downstream of this bearing, the critical speed and impeller clearance problems are avoided (Figure 14, Reference 5). Such an arrangement enhances the reliability of the turbopump.

The seal package as shown in Figures 13 and 14 represent the ideal reliable solution. However, in order to be effective, a careful installation is important. To avoid any leakage flow along the shaft or along the housing seal package interface, gaskets and other static seals, such as O-rings, have to be installed. It is not sufficient to use metal-to-metal seal surfaces.

The highest cryogenic static sealing reliability is achieved by use of flange gaskets or other compression seals, deflected by a clamping force between flat surfaces. Wherever the temperature permits it, plastic gaskets are recommended. Multiple stackups should also be avoided. The ideal solution is to clamp each mating ring against a shaft shoulder and assure, by using a gasket between mating ring and shoulder, a leakage-free installation.

Bearings and Axial Thrust

The design concept described above is only possible if bearings can be used which are lubricated with the propellants to be pumped. This has been established and experimentally verified for many years, (Reference 6). A minimum fatigue life of ten hours is satisfactory for rocket engine use. To achieve optimum performance, correct fits of inner races on the shaft and outer races in the housing are extremely important to prevent race creeping and fretting, it is normal practice to provide a tight fit of the rotating race against its mounting. It also must be taken into account that the room temperature dimensions of the bearings and shafts will be considerably altered at operating conditions which is an important consideration when materials of differing thermal expansions are mated. Summarizing it can be said that pumping cryogenic fluids becomes less complex if bearings are being used which are lubricated with these fluids. The designer is free to select seal arrangements which are best suited to overcome rotordynamic difficulties. Unfortunately, however, bearings lubricated with LOX or LH_2 cannot withstand high loads and high speeds. It is, therefore, essential that the axial thrust is controlled. The analystical determination of the axial thrust is relatively easy, the result, however, is full of uncertainties. An exact prediction for all operating conditions: Start, steady state, shutdown, changing clearances and cavitation, etc., make it impossible to predict the axial thrust accurately. Therefore, it is recommended to use a self-compensating balance piston. Figure 15 (a and b) shows schematically two commonly used configurations. As loads are developed in the turbopump, the shaft tends to move, opening or closing the control orifices. For example, when the shaft moves to the right, the pressure on the right side of the balance piston will increase tending to restore it to its original position. Such a system relieves the bearings from the load under all operating conditions.

TURBOPUMP RESTART AND TEMPERATURE DISTRIBUTION

Start and restart of a turbopump is a difficult problem. It is naturally no problem if quality propellant can be supplied at pump inlet. Such a requirement; however, entails complex auxiliary chill and pressurization systems which are undesirable. To understand this better, the swallowing capacity of the pump during chill down may be discussed. This capacity is limited by the vapor-generating process mainly in the inducer, caused by both cavitation and boiling during the chill-down period. During this time a strong heat transfer occurs inside the pump at all wetted surfaces and a certain amount of vapor is generated such that the latent heat of evaporation times the rate of vaporation equals the rate of heat transfer to the liquid. The vapor is, of course, produced right at the surface of the flow passage and probably distinct zones of vapor and liquid will form.

For a stationary, trapped amount of liquid, this image may be quite correct. However, for a flowing fluid the best description may be to consider a homogeneous mixture of liquid and vapor bubbles flowing along with uniform velocity. Under these conditions, the swallowing capacity of the pump is determined by the head breakdown in the inducer due to cavitation; that is vapor formation in the passage. This mechanism is not too well understood; however, it may be related to an acoustic shock phenomenon, because the acoustic velocity of a bubble mixture is low having values in the vicinity of the velocities used in the flow passages (Reference 4).

From the discussion above it becomes clear that the swallowing capacity of a pump will be affected by the vapor generated per unit time and it can be concluded that the selection of material will diminish or aggravate the vapor generation per unit. In other words, material having a slow heat conduction will be cooled down fast at the surface where low temperatures are needed; meanwhile, the temperature of the core will still be relatively high. Under these conditions, certainly less vapor is generated per unit time thus helping the pump to swallow the mixture. Unfortunately, today no material exists which would provide significant advantages over the commonly used. In Figure 16 some experimental results are summarized showing the cool-down time of three different materials. As can be seen Titanium has the slowest chill time, approximately 50% longer than the normally used aluminum. Titanium has, therefore, some advantage over aluminum. Naturally, also the warm-up period will be stretched out, which is of benefit if a restart of the engine is undertaken within a few minutes after the shut down.

Another major problem area affecting the turbopump design is to insulate the heat stored in the turbine from the pump preserving at the same time the mechanical integrity and minimizing temperature gradients which cause excessive stress. No perfect solution exists at the present time. There will always be some heat leak from the turbine to the pump by way of the shaft én the housing. A possible solution to isolate the pump housing from the turbine housing is illustrated in Figure 17. The two housings are connected and aligned with radial pins. The heat flux in th s case is minimized. Other solutions are naturally also possible; such as, for instance, a flange connection where the two flanges are separated by a heat barrier-type gasket.

SUMMARY

The goal of this report was to identify and discuss problem areas which are unique in cryogenic pumps used for spase travel. These problems affect the design of the pump impellar, the seal,bbearing, and the housing arrangement. A major problem affecting the design of the rotating elements are the unsteady forces - flow oscillations - which may endanger the planned space mission.

It is shown that in order to arrive at a reliable pump, a shrouded impeller and bearings lubricated with the fluid to be pumped and located close to the oxidizer pump should be used. Based on test results, recommendations are included concerning the unsteady forces. The effect of inducer and impeller design on these flow oscillations is presented.

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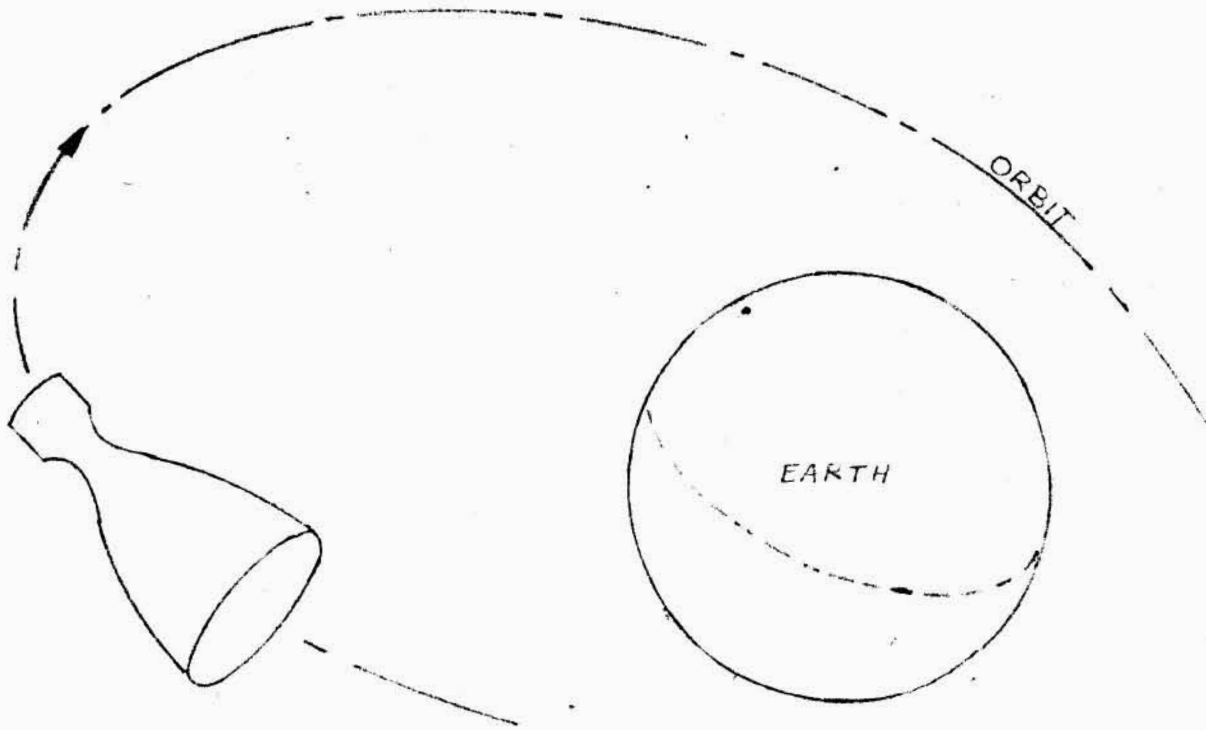
ESTIMATED AVERAGE TEMPERATURE HISTORY
DURING ORBITAL FLIGHT, (1 ORBIT = 90 MINUTES)



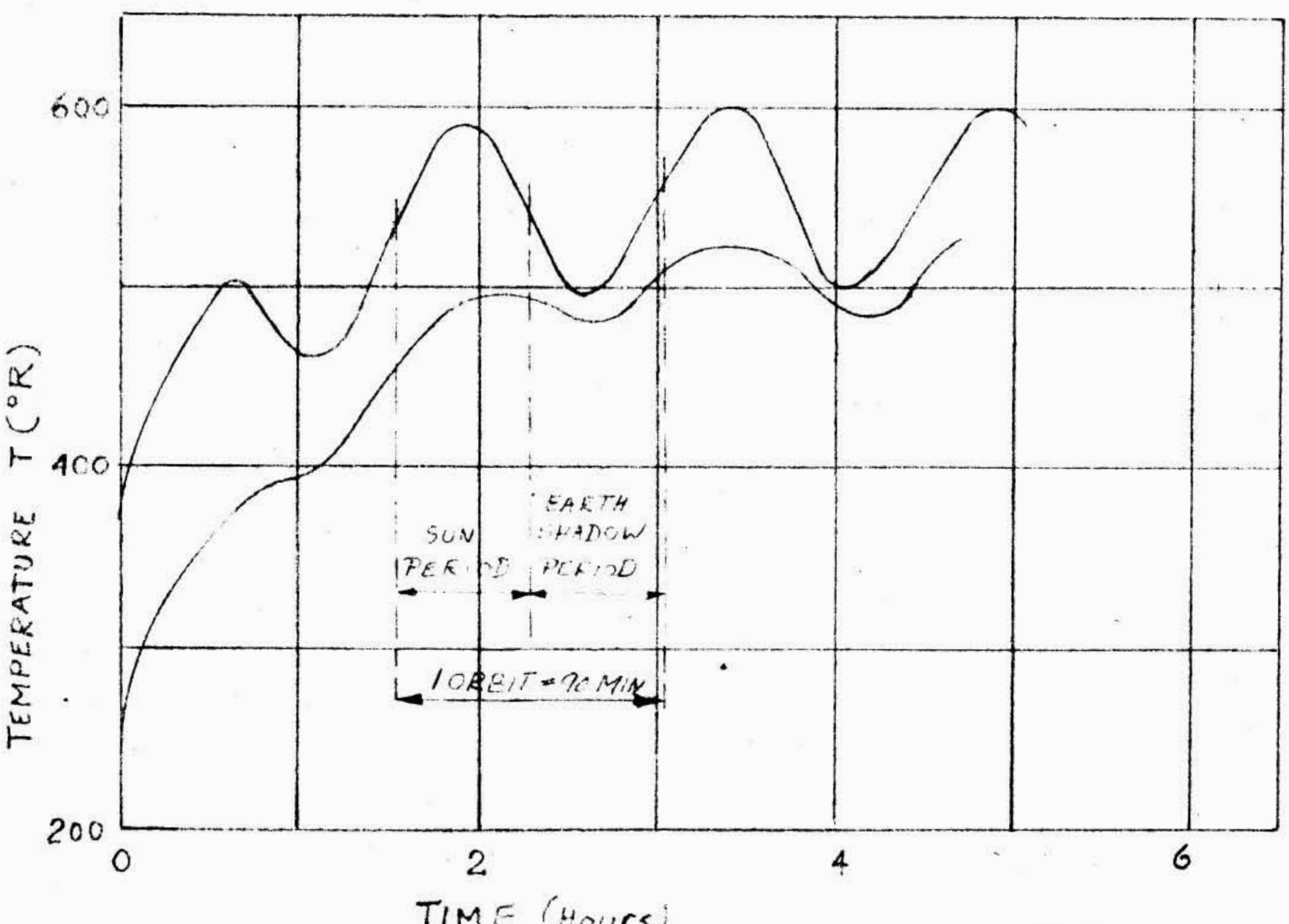
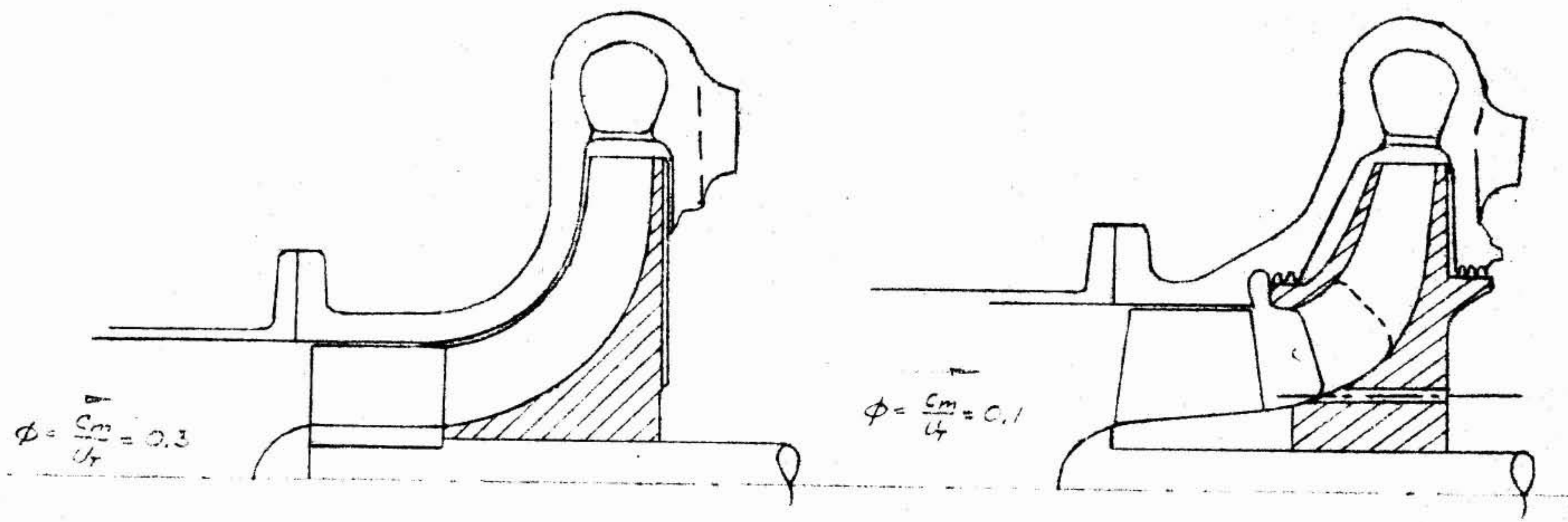


FIG. 1



2a

HIGH INLET FLOW COEFFICIENT, OPEN
FACED IMPELLER DESIGN

2b

LOW INLET FLOW COEFFICIENT
SHROUDED IMPELLER DESIGN

COMPARISON OF TWO IMPELLER DESIGN
APPROACHES.

FIG. 2

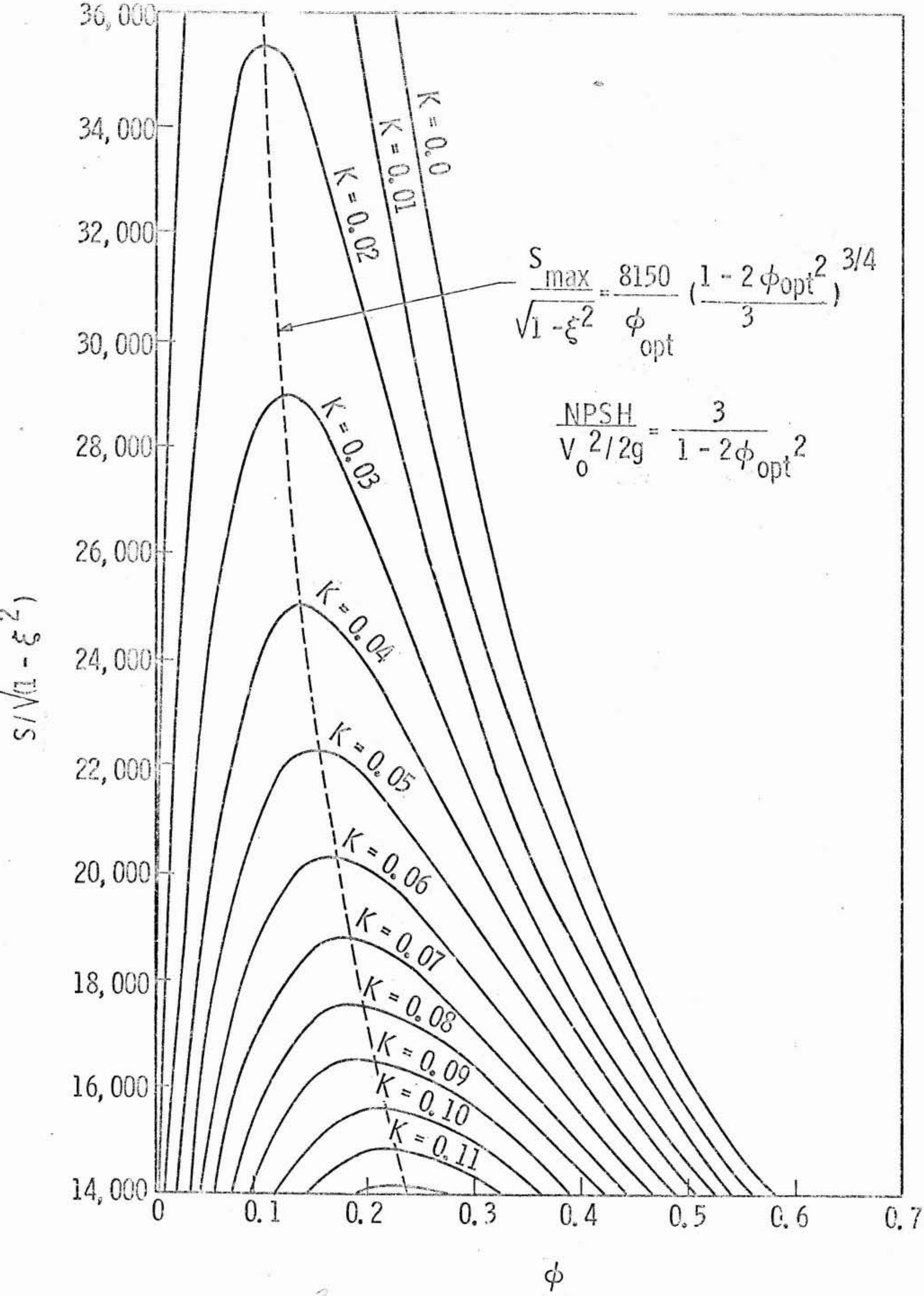


Figure 13. Theoretical Relationships Between Cavitation Parameters

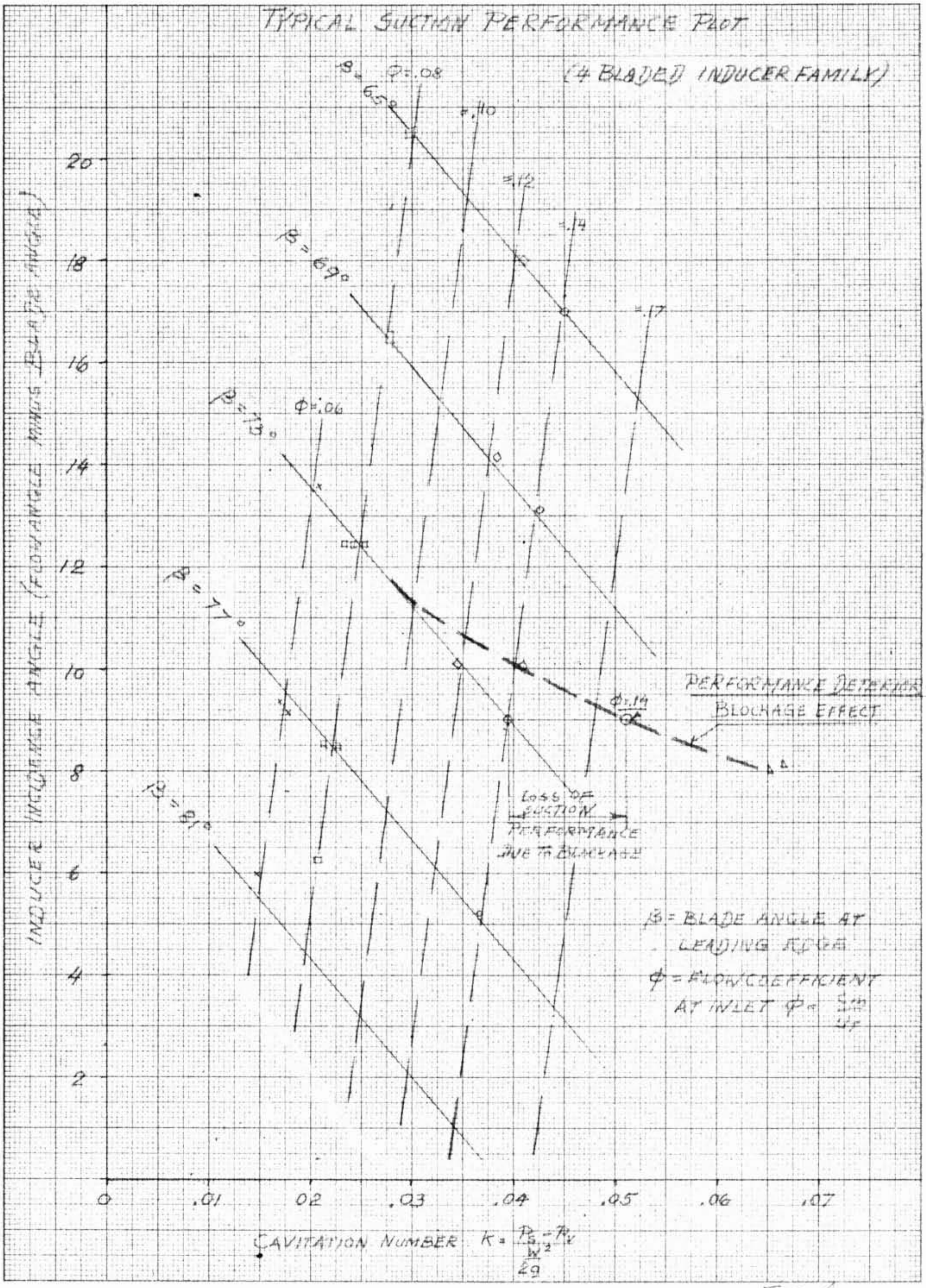

TYPICAL SUCTION PERFORMANCE PLOT
(4 BLADED INDUCER FAMILY)
β = 65°
Φ = .08
=.10
=.12
=.14
=.17
β = 69°
β = 73°
Φ = .06
β = 77°
β = 81°
Φ = .14
PERFORMANCE DETERIORATES
BLOCKAGE EFFECT
LOSS OF SUCTION PERFORMANCE DUE TO BLOCKAGE
β = BLADE ANGLE AT LEADING EDGE
Φ = FLOW COEFFICIENT AT INLET
INDUCER INCIDENCE ANGLE (FLOW ANGLE MINUS BLADE ANGLE)
CAVITATION NUMBER K = $\frac{P_S - P_V}{\frac{W^2}{2g}}$
0
.01
.02
.03
.04
.05
.06
.07
2
4
6
8
10
12
14
16
18
20

FIG. 4

TYPICAL PUMP INLET FLOW OSCILLATIONS

$$\frac{\phi}{\phi_{DES}} = 1.0$$

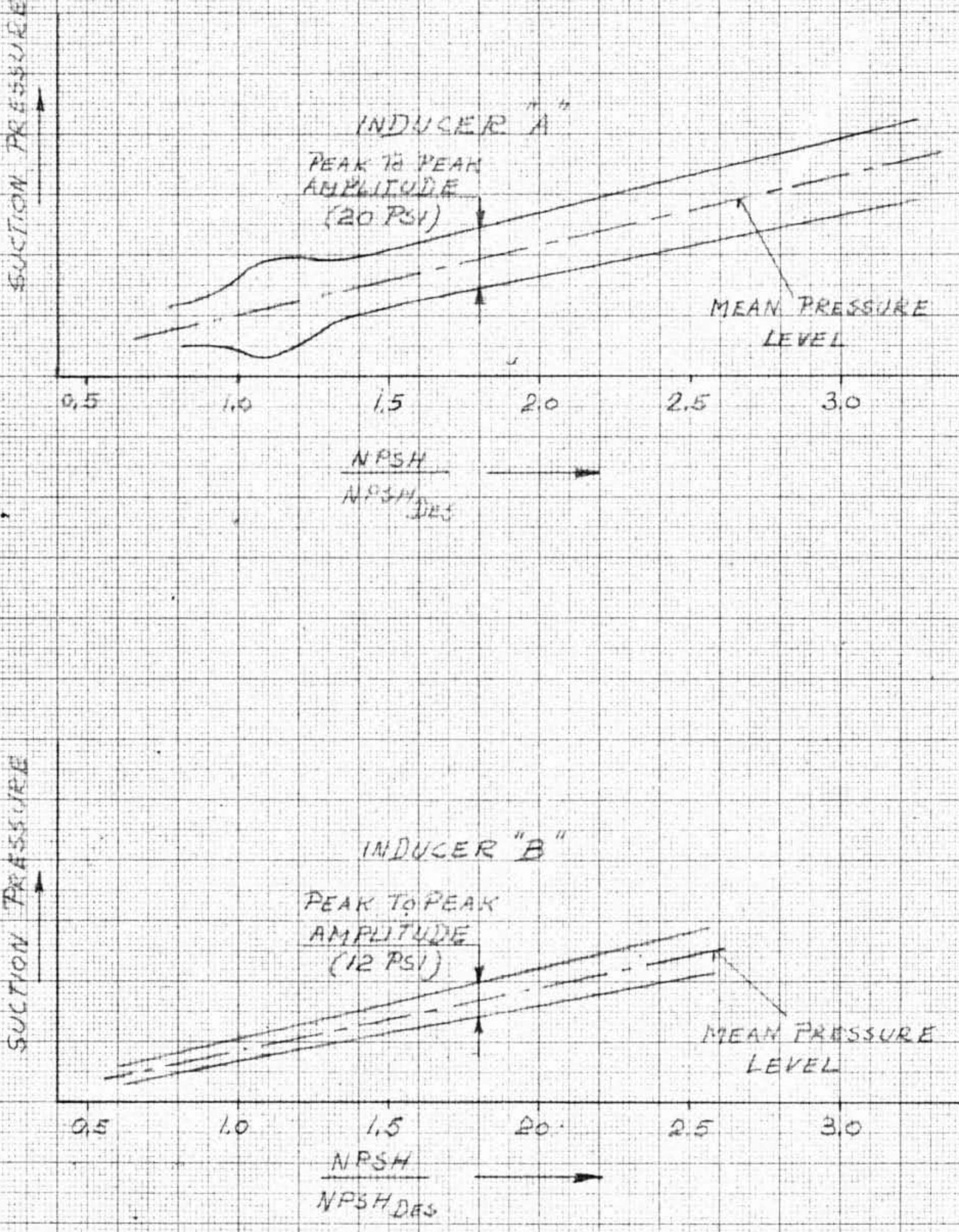


FIG. 5

TYPICAL PUMP INLET FLOW OSCILLATIONS

$$\frac{\Phi}{\Phi_{DES}} = 0.7$$

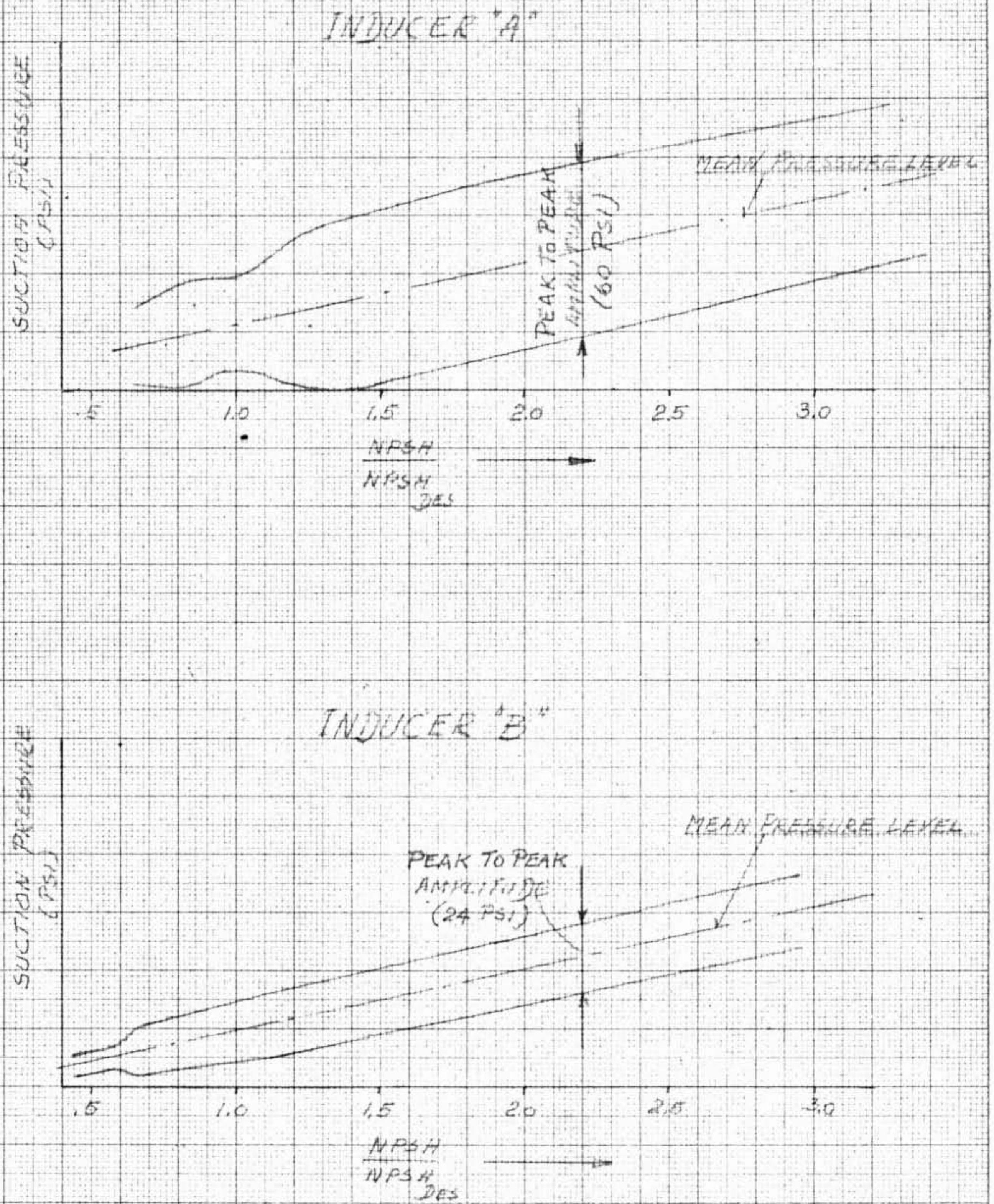


FIG. 6

TYPICAL EXAMPLE OF INLET FLOW OSCILLATIONS

— EFFECT OF INDUCER DESIGN —

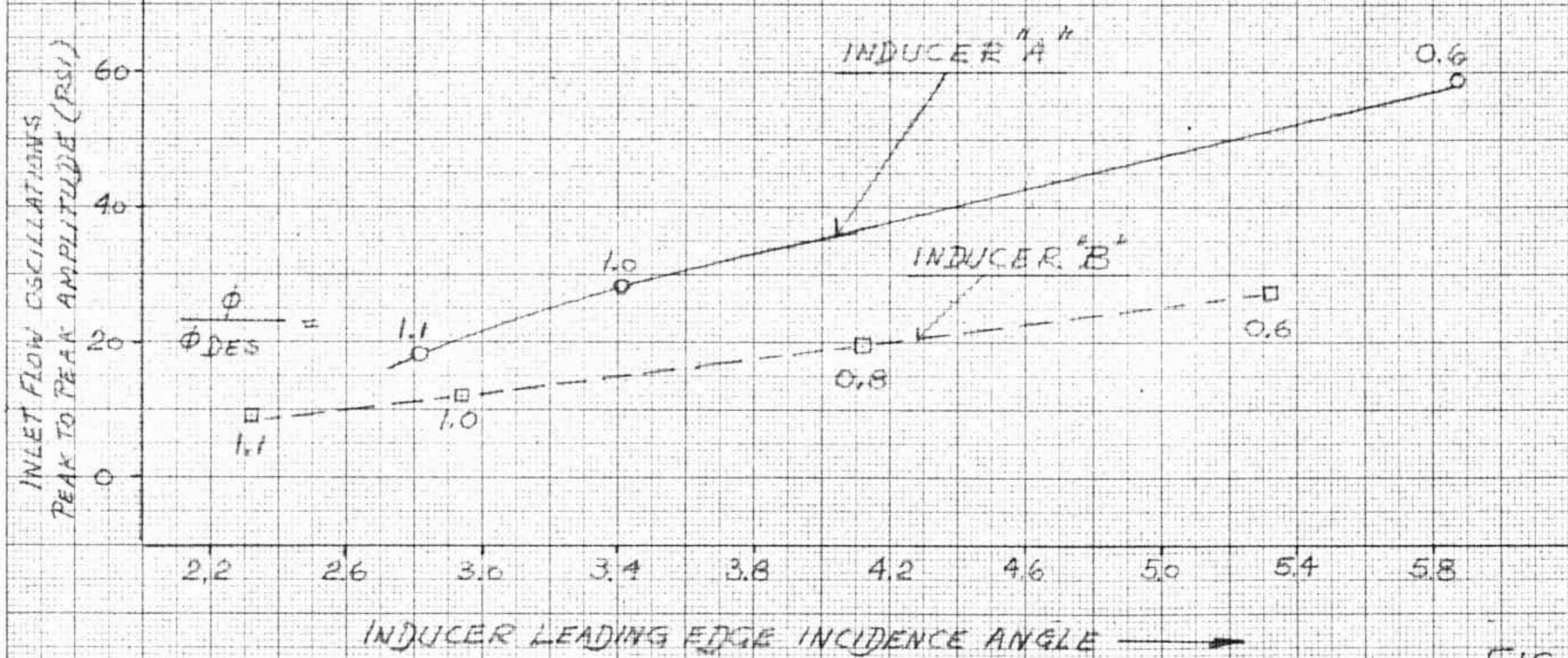


FIG. 7

ILLUSTRATION OF UNSTEADY WAKE INTERACTION
WITH STATIONARY VANE SYSTEM

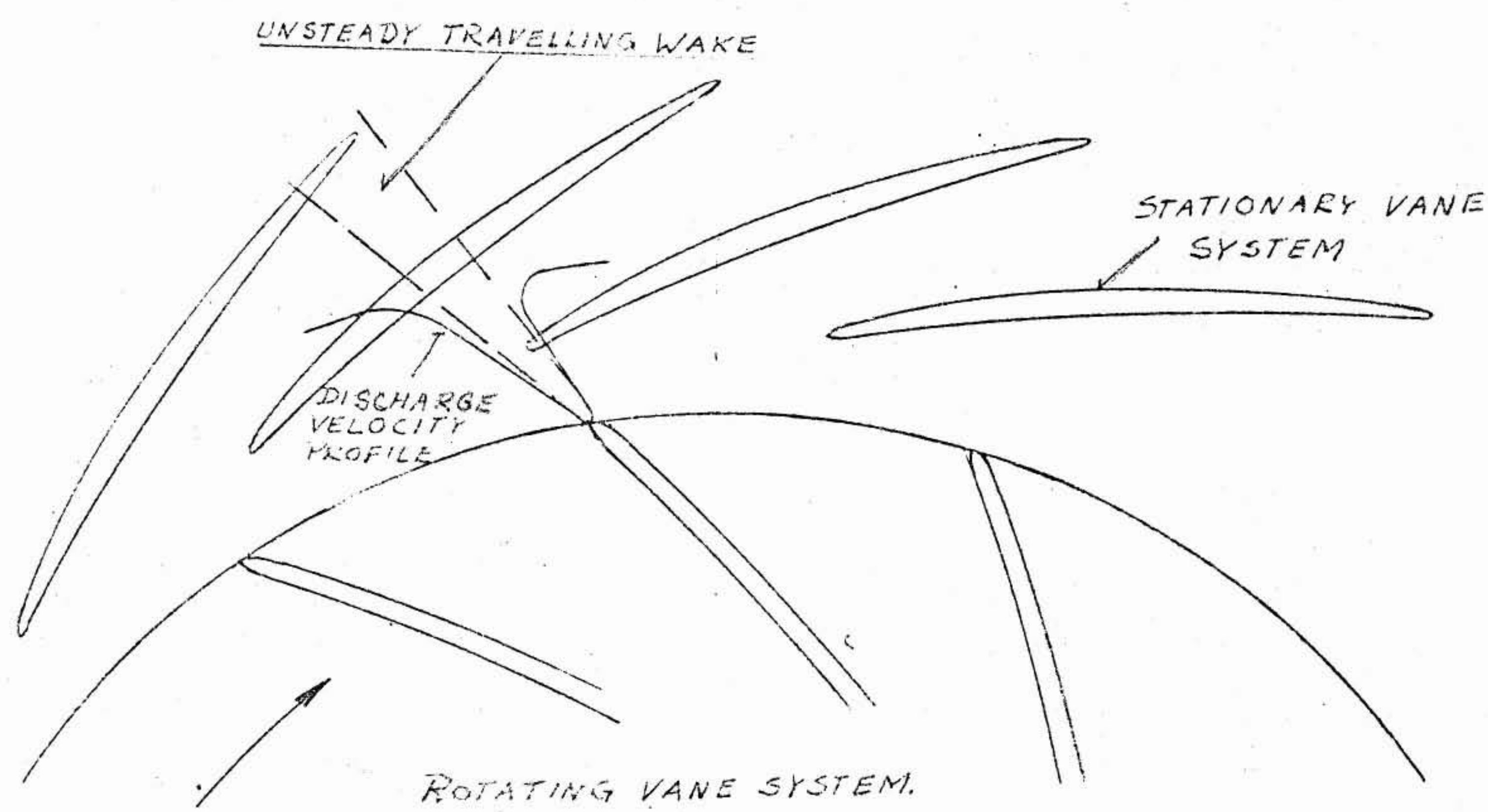


FIG. 8

ILLUSTRATION OF WAVE RE-ENFORCEMENT PHENOMENON.

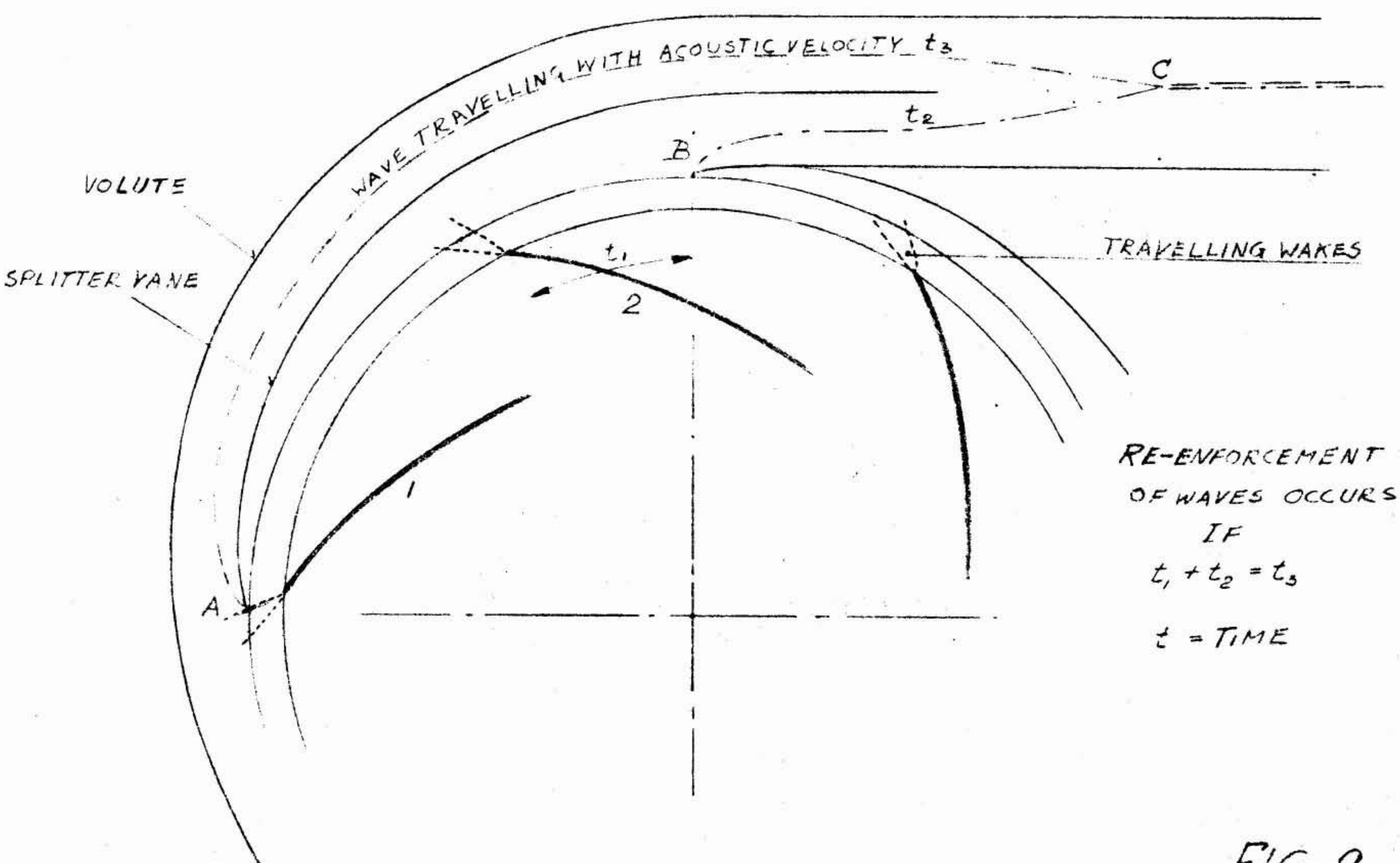
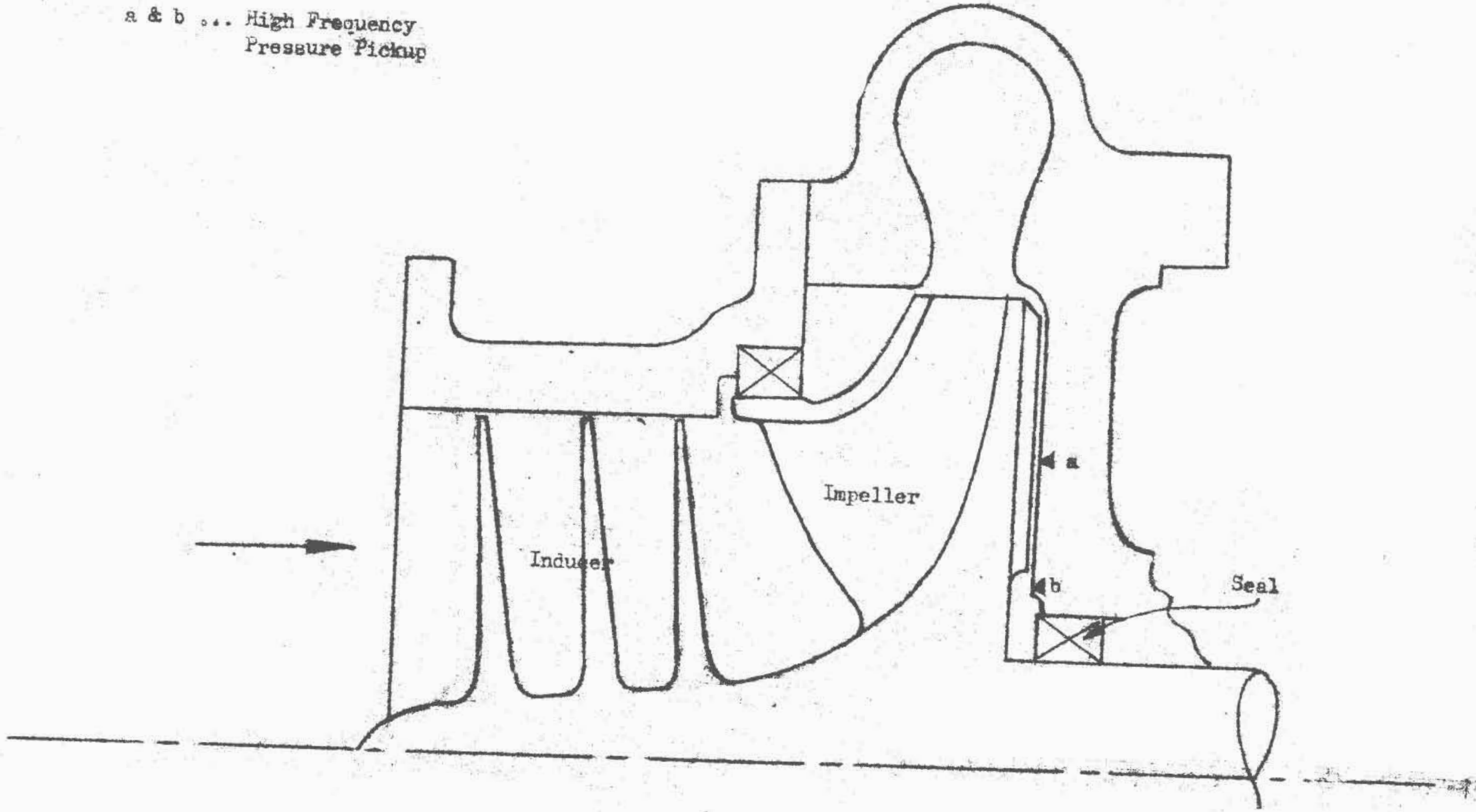


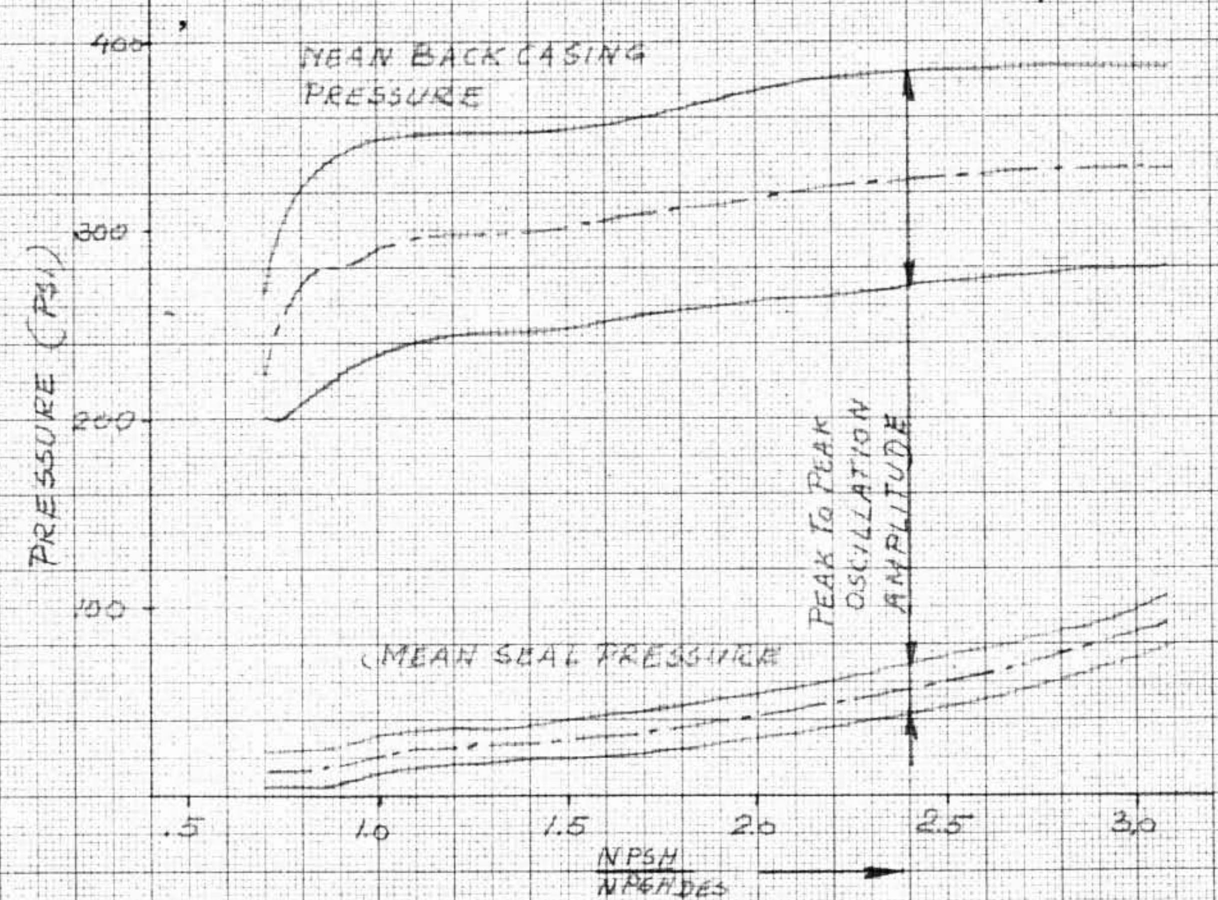
FIG. 9



Centrifugal Pump

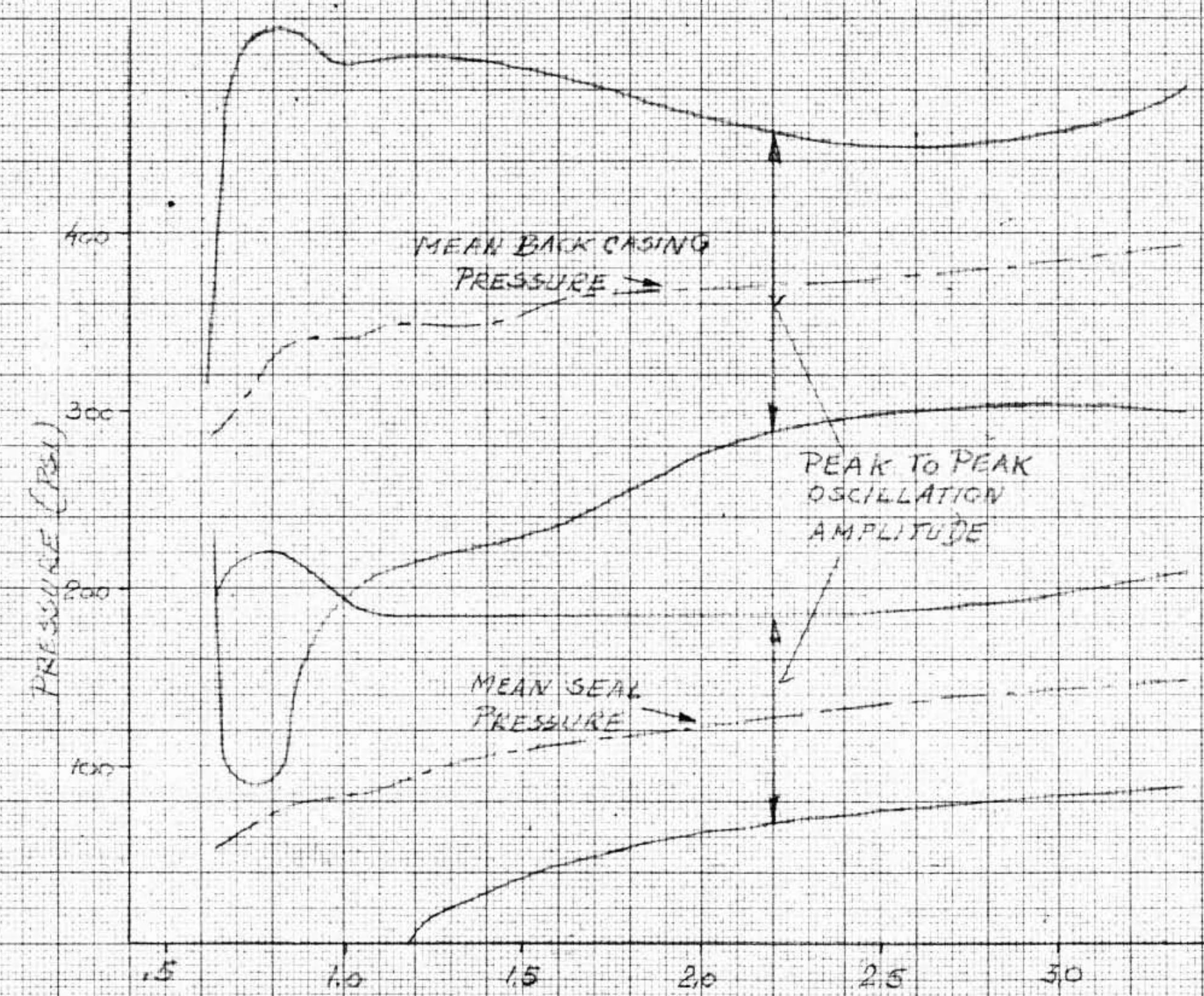
(Location of High Frequency Pressure Pickups)

FIG. 10



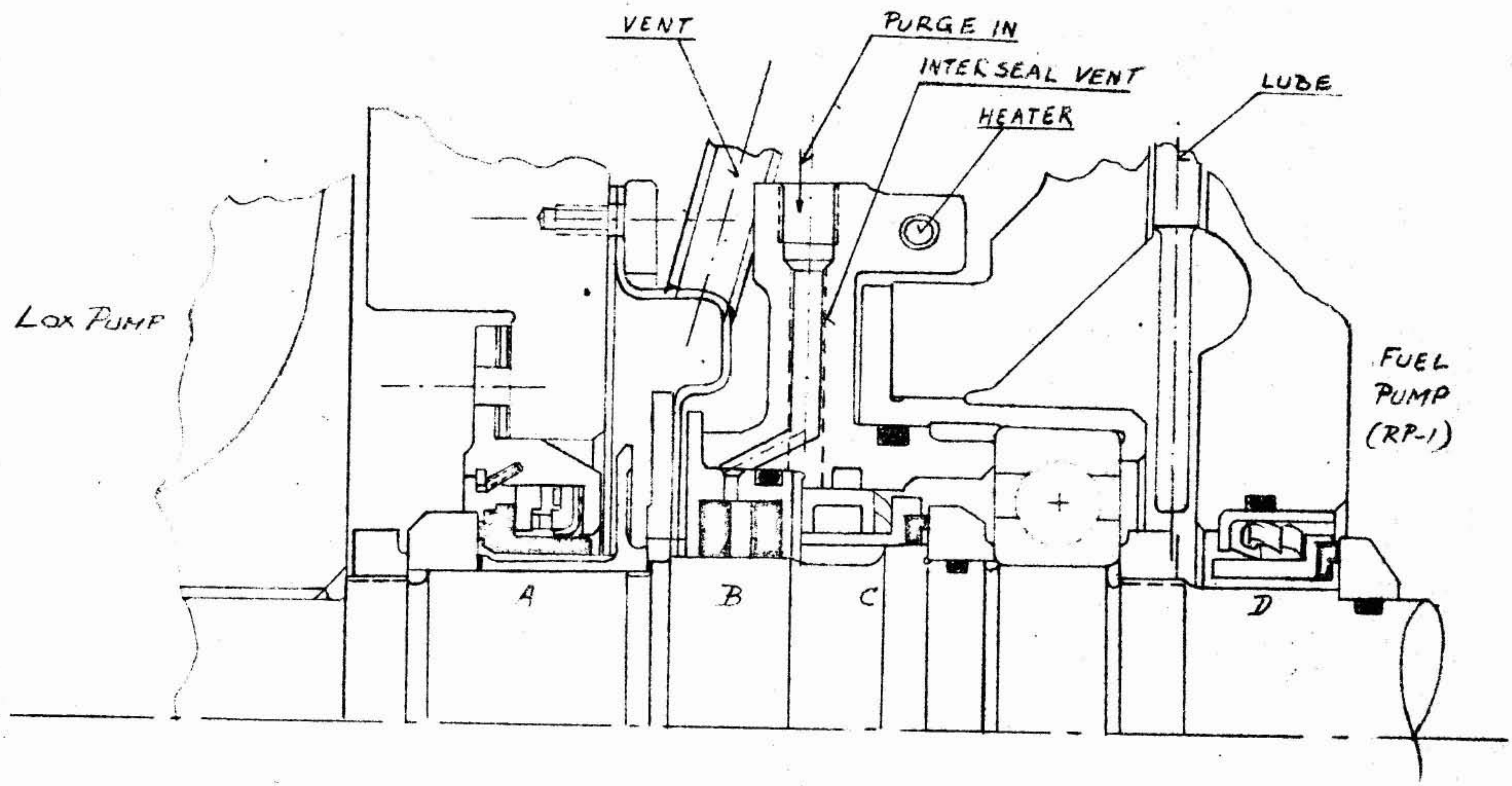
BACKCASING AND SEAL AREA OSCILLATION AMPLITUDE
(EXAMPLE FOR DESIGN CONDITIONS: $\frac{\Phi}{\Phi_{DES}} = 1.0$)

FIG. 11



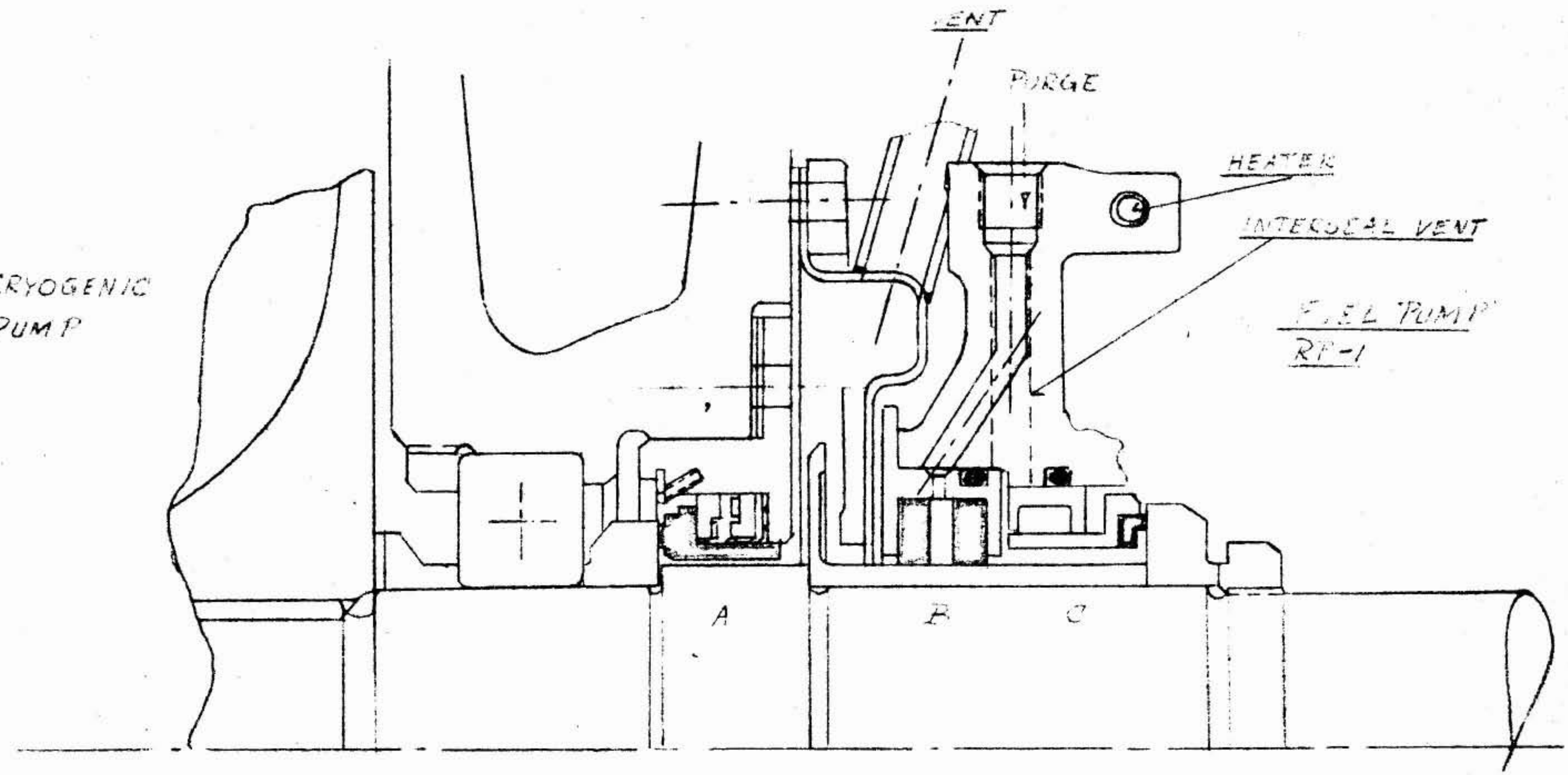
BACKCASING AND SEAL AREA OSCILLATION AMPLITUDE
(EXAMPLE FOR OFF DESIGN CONDITIONS: $\frac{\phi}{\phi_{DES}} = 0.5$)

FIG. 12



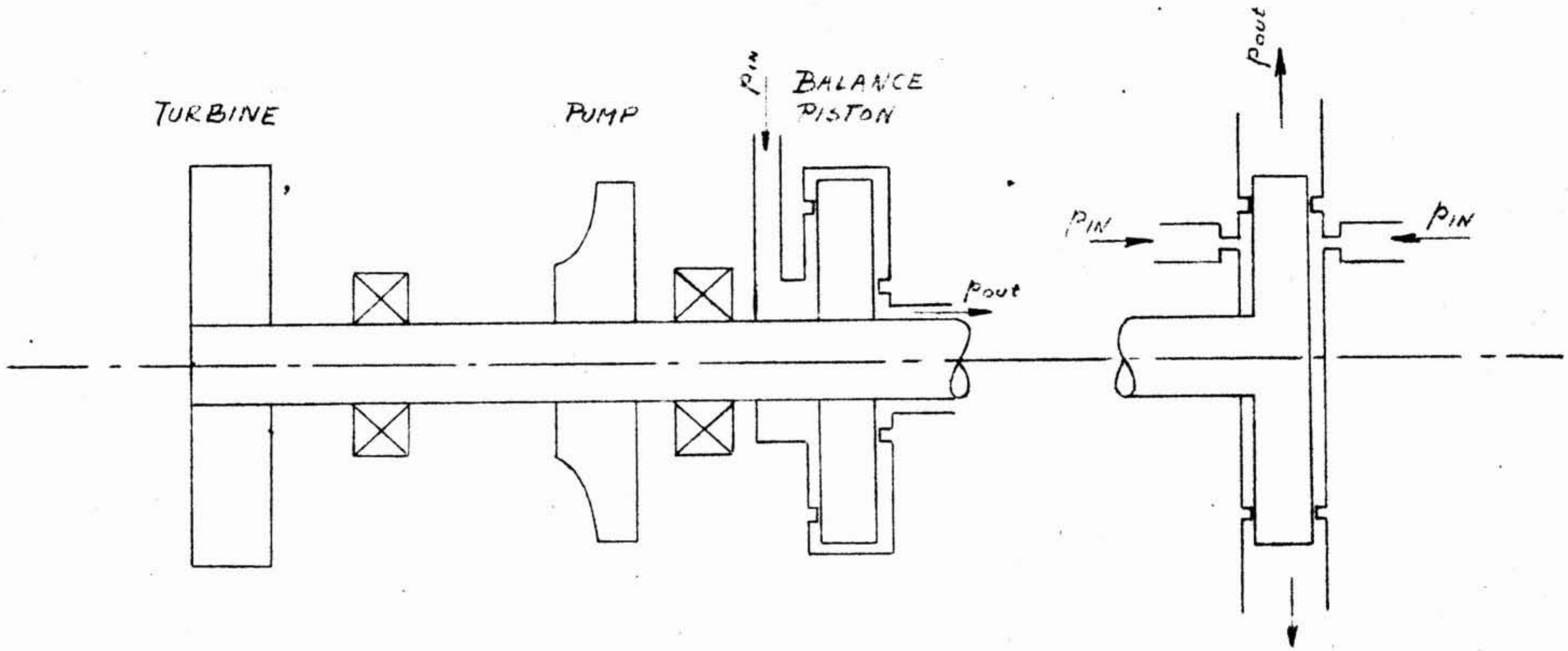
SEAL ARRANGEMENT-LOX PUMP OVERHANGING

FIG. 13



SEAL ARRANGEMENT - LOX PUMP NOT OVERHANGING

FIG. 14

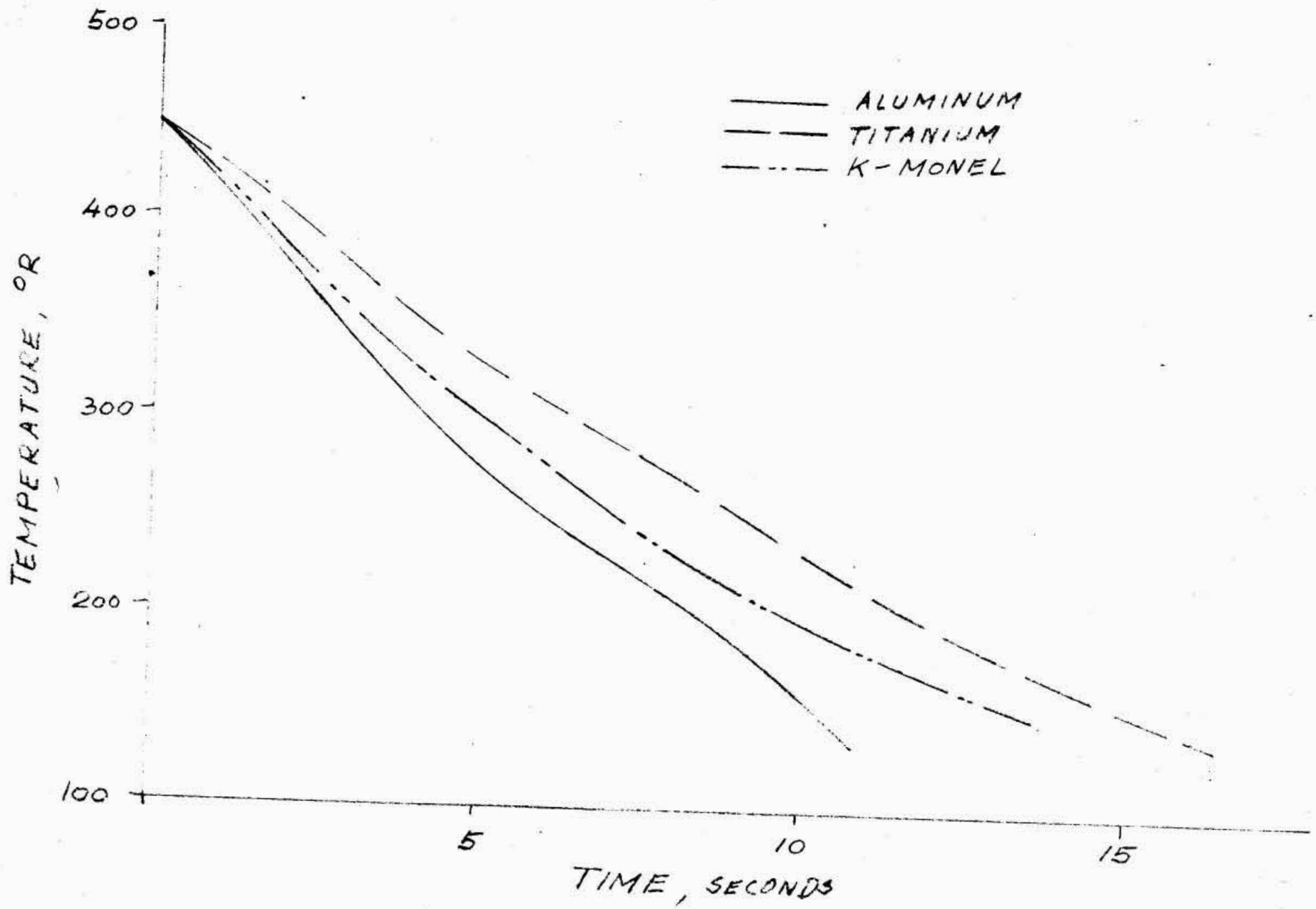


15a SERIES FLOW BALANCE PISTON

15b PARALLEL FLOW BALANCE PISTON.

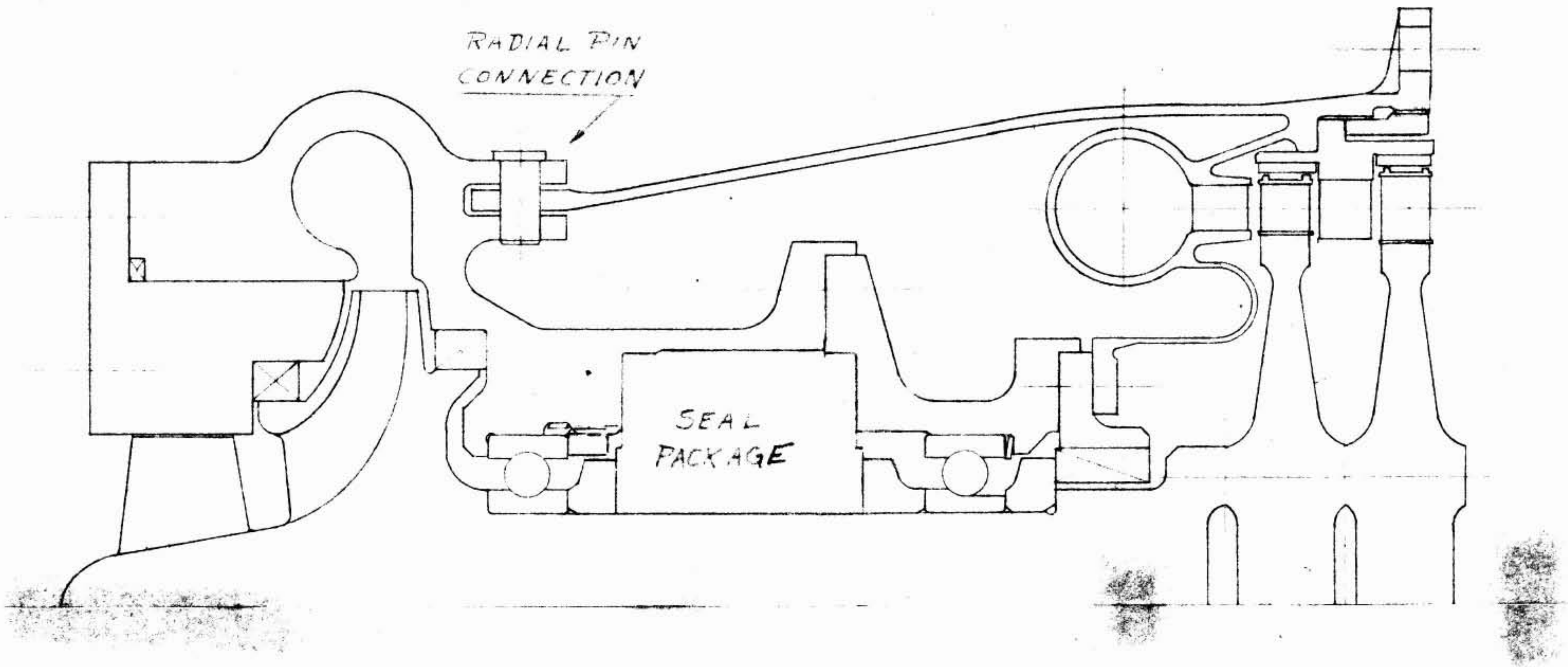
SCHEMATIC OF TWO TYPES OF SELFCOMPENSATING BALANCE PISTONS.

FIG 15



COOLDOWN OF THREE METAL TUBES IN HYDROGEN.

FIG. 16



EXAMPLE OF THERMAL ISOLATION OF TURBINE HOUSING
FROM PUMP HOUSING

FIG. 17