BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

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LARGE-SCALE BREEDER REACTOR PROTOTYPE MECHANICAL PUMP - CONCEPTUAL DESIGN STUDY HOT LEG

PREPARED UNDER CONTRACT 67002
FOR
GENERAL ELECTRIC FAST BREEDER
REACTOR DEPARTMENT
LMFBR PUMP DEVELOPMENT PROGRAM

TASK I

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DISCLAIMER

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1. ABSTRACT

Due to the extensive nature of this study, the report is presented as a series of small reports. This is in part due to the use of different departments within Byron Jackson and outside consultants. Taking advantage of the letter report requirement, the formal numbering system was used only in portions of the study.

The complete design analysis is placed in a separate section. The drawings and tabulations are in the back portion of the report. Other topics are enumerated and located as shown in the table of contents.

Each numbered topic or report has its own page numbers.

2. INTRODUCTION

The available NPSH is the key characteristic in the design of the hot leg pump concept. The aim is to select the highest speed that will make the NPSH. This will give the smallest pump and probably the most economical one. Based on past experience, two concepts were considered.

The first one was a single stage double suction pump similar to Clinch River, but at a slower speed suitable for the available NPSH.

The second one was a two stage double suction pump, with the impellers in parallel, similar to the Large Scale Breeder cold leg concept. With two stages in parallel, the pump speed can be increased. This will make the pump smaller.

Time and expense limited the in-depth study to one concept. Looking at the many factors involved, and the evaluation as shown in the section titled "Concept Selection", the single stage pump concept was chosen.

3. SUMMARY

This pump study does not represent a complete detail design. It does go into enough depth to insure that the basic pump configurations can meet the system requirements as outlined in the Specification Document. Enough information also had to be generated to enable pricing, manufacturing, and design analysis.

A review of this complete report indicates that the design is feasible. The size is within present day capability, as to fabrication and manufacture. The design is based on experience gained to date from the Clinch River Project.

The performance test of the prototype with water is an important part of the work involved. The size of this hot leg pump places this test within the range of existing equipment.

Development needs are listed separately. The building and testing of a small scale model for the hydraulic portion of the pump and the bearings is highly recommended. Seal development could also be done separately, as recommended in the development needs.

4. EVALUATION FACTORS

TABLE 1 FACTORS INFLUENCING PUMP DESIGN

HYDRAULIC PARAMETERS	1. FLOW RATE 2. TOTAL DEVELOPED HEAD 3. NET POSITIVE SUCTION HEAD 4. PUMP EFFICIENCY 5. OPERATING MODES AND RANGE
OPERATING ENVIRONMENT	1. FLUID PROPERTIES 2. AMBIENT CONDITIONS 3. SYSTEM IMPOSED CONDITIONS
PHYSICAL REQUIREMENTS	1. WEIGHT AND SPACE 2. RADIATION SHIELDING 3. CONTAINMENT
SERVICE REQUIREMENTS	1. DESIGN LIFE 2. RELIABILITY 3. AVAILABILITY
STRESS REQUIREMENTS	1. DYNAMICS 2. NOZZLE LOADS 3. SEISMIC RESPONSE 4. BEARINGS

5. PUMP CONCEPT

5.1 Single Stage

Pump Operating Requirements:

Capacity or flow	77,000 GPM
Head	500 Ft.
NPSH available at inlet nozzle	30 Ft.

The low NPSH of 30 ft. will be the determining factor in selecting the pump speed. Also, a double suction impeller will be used. From experience and tests, a suction specific speed of 13,500 to 16,000 has been found to give satisfactory performance for a double suction impeller. With this information, enough data is available to calculate the pump speed required.

$$S_s = \frac{N\sqrt{Q}}{(NPSH)^{3/4}} = 13{,}500$$ (conservative for a double suction impeller)

S_s = Suction specific speed

N = Pump speed, RPM

Q = Capacity or flow in GPM

$NPSH$ = Net positive suction head in ft.

$$N = \frac{13{,}500 \times (30)^{3/4}}{\sqrt{77{,}000}} = 620 \text{ RPM}$$

For 620 RPM, the specific speed, N_s, is

$$N_s = \frac{620\sqrt{77{,}000}}{(500)^{3/4}} = 1{,}630$$

With this information, it is possible to design the required hydraulics or select a satisfactory model from existing pump test files.

5.2 Two Stages

Double Suction Impellers

Two stages in parallel

Flow per stage $\frac{77,000}{2}$ = 38,500 GPM.

Head (ea.stage) $\frac{500}{1}$ = 500 Ft.

Look at Clinch River PC-32488

$$Q = 25,000 \times \frac{900}{1116} \times (1.24)^3 = 38,500 \text{ GPM}$$

$$H = 500 \times \left(\frac{900}{1116}\right)^2 \times (1.24)^2 = 500 \text{ Ft.}$$

This pump at 900 RPM is a 1.24 factor of Clinch River.

Impeller Dia. 1.24 x 39 = 48.5"

For 30 Ft. NPSH:

$$S_s = \frac{900 \times \sqrt{38,500}}{(30)3/4} = 11,000$$

This is satisfactory and conservative.

The outer tank would be about 95" O.D.

Construction of this two stage concept would be similar to the large scale breeder. The difference in length would probably eliminate the top bearing, as well as top shaft flanged connection.

Nozzle location could be made to suit specification or field requirements.

PUMP CONCEPTS

(WITH COMPARISON TO CLINCH RIVER)

PUMP TYPE	NUMBER OF STAGES			SPEED R.P.M.	IMPELLER DIA.	TANK SIZE TYPE	COLUMN SIZE DIA.	BEARINGS	STATIC SEAL	NOZZLE LOCATION	SHAFT SIZE AT IMP.	PUMP CASE	REMARKS
	FIRST	SECOND	THIRD										
CLINCH RIVER SINGLE STAGE (FOR COMPARISON ONLY)	DOUBLE SUCTION	——	——	1116	39"	90" SPHERE	64" TAPERED	2-16" 1-SLV.	AT PUMP END	SUCTION AT BOTTOM HORIZ. DISCH. AT BOTTOM HORIZ.	$10\frac{3}{4}$" HOLLOW	VOLUTE	
LARGE SCALE (HOT LEG) SINGLE STAGE	DOUBLE SUCTION	——	——	620	68"	140" SPHERE	99"	2-24" AT IMP.	AT PUMP END	SUCTION AT BOTTOM HORIZ. DISCH. AT BOTTOM HORIZ.	23" HOLLOW	VOLUTE	
LARGE SCALE (HOT LEG) TWO STAGES IN PARALLEL	DOUBLE SUCTION	DOUBLE SUCTION	——	900	48.5"	95" CYLINDER	52"	2-20" AT IMP.	AT TOP END	SUCTION AT BOTTOM VERTICAL DISCH. AT TOP HORIZ.	13" HOLLOW	VOLUTE	

6. PUMP CONCEPT SELECTION

The pump specification was reviewed carefully, noting particularly the operating requirements. Next, the main objectives were listed, so that the study would cover them adequately. It was noted that these pumps are located in the hot leg side of the loops.

As suggested in the specification, experience gained in the Clinch River Project and the recent study of the Large Scale Cold Leg Conceptual Design will be utilized here.

Comparing the available NPSH of 30 ft. for this study with 53 feet available for Clinch River, indicated that the speed of 1116 RPM for Clinch River would have to be reduced to maintain an acceptable suction specific speed, S_s. It was found that for a double suction impeller the speed would have to be about 600 RPM.

Doing the same thing for two stages in parallel, similar to the Large Scale Breeder cold leg pump, it was found that a satisfactory S_s could be attained with a speed of about 900 RPM. This would be a smaller pump, but two stages. The Clinch River hydraulics were used for a model factor. With the change in speed, this two stage pump at 900 RPM would be approximately a 1.24 factor of Clinch River.

The model selected for the hydraulics of the single stage is a pump that has been built and thoroughly tested. To take advantage of the peak performance, the speed should be 620 RPM. For comparison purpose only, this pump would be about a 1.64 factor of Clinch River.

A decision now had to be made whether to select a single stage or two stages in parallel for this study. There was not enough time to go into great depth with two concepts, particularly the design analysis. Enough work had to be done, however, to make the proper selection. The experience gained from Clinch River and the Large Scale Breeder Study helped in keeping this work down to a minimum.

In general, a single stage is preferred over a two stage pump. The volute and suction castings are simplified. The internal passages and returns of the two stage pump would not be required for a single stage, thus simplifying the pattern and casting. Also, the pump end is of course shorter, as is the span between the two bearings at the pump end. However, the speed has to be lower, and the pump larger. The size increase is in the approximate ratio of 1.3 to 1 at the pump end only. Shaft size remains the same. Actual lengths do not differ greatly. The upper end of the pumps including seal size and driver mount can be about the same. This led to the selection of the single stage double suction pump for the concept to be pursued.

Model Selection:

Select a model with the same specific speed and suction specific speed as the prototype at approximately the best efficiency point. A 14 x 14 x 17A DVSR pump with a double suction impeller satisfies the requirements. The pump has been tested, and the performance shown on T-30778-1. The model factor, F, is found to be 3.88. Applying the model factor to the test curve, and correcting for the change in speed, will generate a performance curve for the prototype pump.

$$Q_P = F^3 \times \frac{N_P}{N_M} \times Q_M$$

$$H_P = F^2 \times \left(\frac{N_P}{N_M}\right)^2 \times H_M$$

where P designates the prototype and M the model.

Using a prototype motor speed of 620 RPM, and the model performance curve T-30778-1, the prototype operating requirements can be checked and additional points calculated to show the predicted performance. Leakage losses in the prototype pump will be higher than those in the model. This will result in a drop in efficiency for the prototype.

The NPSH for the prototype is derived in much the same manner as the head, Hp.

To obtain the prototype pump operating conditions, a point on the model curve having the same specific speed is selected. The factor and speed change are then applied in the following manner. The model was tested at 4460 RPM. Select a point on the curve of the model test of 9500 GPM and 1720 Ft. for a check of the model factor.

$$Q_p = (3.88)^3 \times \frac{620}{4460} \times 9500 = 77{,}000 \text{ GPM}$$

$$H_p = (3.88)^2 \times \left(\frac{620}{4460}\right)^2 \times 1720 = 500 \text{ Ft.}$$

This is a good check on the model factor.

The impeller diameter would be 3.88 x 17-1/8 = 66½". Add about 2½% for safety, making the impeller 68" dia.

VOLUTE

The volute casing for the prototype pump is similar to the one used on Clinch River. It will be a triple volute. Areas and shapes selected are based on areas of the model. The model factor must of course be used to give required areas for the prototype.

The model had a double volute. A triple volute is used on the prototype to keep the size smaller, because the total area will be distributed over an additional section. The total flow area of the prototype with the triple volute will be equal to that of the model double volute, increased by the proper use of the model factor.

The volute casing will be described in more detail in the key features.

7. KEY FEATURES

7.1 IMPELLER

As has been explained in the pump concept selection, the low available NPSH of 30 feet led to the use of a double suction impeller. In a double suction impeller, liquid is introduced at both sides. This gives a large eye area, with low inlet velocity to attain the limited available NPSH. The two inlets virtually double the inlet area of a single suction impeller.

A double suction impeller has another desirable feature. It is symmetrical about its centerline, reducing axial thrust to a minimum. There are bound to be some inequalities which will produce small amounts of axial thrust which are largely unpredictable. These will be taken by the thrust bearing in the motor.

7.2 VOLUTE CASING

The impeller discharges the liquid into a single or multiple channel of gradually increasing area called a volute. Velocity is partly converted into pressure by the pump volute before it leaves the pump at the descharge nozzle.

Based on experience from Clinch River, this pump concept has a multiple volute. Each volute has three openings leading to the discharge nozzle. Multiple volutes have been used previously for sodium pump applications.

The suction nozzle flares out as it reaches the volute and feeds both sides of the double suction impeller. The suction and discharge are separated by the close clearance formed by the tapered seal, thus keeping leakage losses to a minimum.

7.3 SUCTION PIECES

The suction pieces guide the flow to the inlet sides of the double suction impeller. They also form the stationary part of the wear rings that keep leakage losses down to a minimum from discharge to suction.

In addition, the hydrostatic bearings are integral parts of the suction pieces. The bearing pockets are formed by ribs that are part of the casting. The bearing end rings are separate pieces welded to the casting. The bore of these rings has a colmonoy hardfacing. The small size of these rings simplifies the overlay procedure. The bearing pockets are made deep for better thermal distribution. The bearing pockets are pressurized thru holes coming from the discharge side of the pump. The returns are basically to the suction side.

7.4 INTERNAL TAPER FIT FOR LEA"AGE CONTROL

The design of this pump includes a tapered ajustable clearance to minimize leakage or circulation from discharge pressure back to suction. In terms of pump head at design flow, this could be a discharge head of 500 feet to a suction head of 30 feet. The leakage or circulation affects performance in the form of additional power due to the loss in pump efficiency. It is a design objective, therefore, to keep this leakage down to a minimum flow. The leakage from discharge to suction takes place only around the periphery of the discharge nozzle. The leakage path is short only in the vertical position and keeps increasing in length, reaching a maximum in the horizontal position. Since leakage is a function of pressure, clearance, and length, the design controls clearance and length of the leakage path. This will keep the leakage or recirculation to a minimum.

In a sodium environment it is not practical to use a gasket. The large size and required freedom of movement dictate an opening with a static clearance. The size also discourages the normal straight female bore and male turn to obtain a small clearance, since an adverse tolerance stack-up would make the clearance greater than desired. Size of the parts makes it very difficult to manufacture and inspect to close tolerances.

The leakage or recirculation flow is a function of the pressure drop and the clearance. Since we cannot alter the pressure drop, we chose to control the clearance. This led to a female and male taper where the clearance could be controlled by axial movement between them. The turn and bore would not require close tolerances, as long as the tapers were essentially close to each other.

For this pump concept, the tank sealing cone has the female taper and remains in the fixed position. The matching taper is on the volute and is adjustable axially to give the desired clearance. An arrangement consisting of a series of threaded fasteners is used to move the volute axially relative to the tank sealing cone. Another set of cap screws or studs is used to maintain this setting.

Assuming matching tapers, the clearance can be set to a pre-determined value by the relative axial movement between the two parts. The two parts can be allowed to engage lightly for the zero clearance. With the taper in inches on diameter per foot of length known, the desired clearance on diameter can readily be converted to axial movement. This can be measured accurately, and locked in position, thus assuring the diametral clearance.

The use of a taper versus a straight bore enables:

(1) Use of more liberal tolerances on each piece with no sacrifice of close clearance.

(2) An adjustable diametral clearance.

(3) A means of setting a pre-determined clearance.

(4) The engagement of the taper to determine the diametral clearance.

7.5 HYDROSTATIC BEARING

This pump concept requires a bearing in the liquid being pumped. It must be reliable and designed for a long life. For this application, a hydrostatic bearing was selected that depends on flow and pressure. It is similar to Clinch River, but larger. The bearing is self-contained, taking flow and pressure from the discharge side of the pump and returning to suction.

Two bearings are used for this concept, one below the impeller and the other above. This allows the use of smaller bearings, and reduces the bending moments. It places the radial load produced by the impeller between the two bearings, a desirable feature.

This pressurized bearing is basically a non-contact centering device, as shown schematically on Fig. 1. The bearing has a series of pockets located radially around the rotating journal. These pockets have orifice holes that are exposed to pump discharge pressure. Ideally, with the journal in the center of its clearance, the pressure is the same in all of the pockets. Under radial load, the journal moves off center towards one of the pockets. The clearance is decreased, the flow is also decreased, and the pressure is increased. The clearance on the opposite pocket is increased, the flow is also increased, and the pressure is decreased. This combination of events takes the radial load and has a tendency to keep the journal centered. The load is taken by the pressure and during normal operation avoids contact between the journal and the bearing. For slower speeds, the available pressure is lower, but the radial load is also lower, so the bearing will function normally. This description is an over-simplification. The bearing must be sized properly, clearances and orifice sizes are critical, and the returns must be adequate.

Bearings of this design have been used on a previous sodium pump and on many light water applications. There is no apparent difference in the functioning of the bearing.

A start-up, the journal may contact the bearing surface. This in itself is not harmful. To give a harder bearing surface and insure long life, the journal and the end rings of the bearing are hardfaced with colmonoy. This is described in the special material section.

Bearing of this type have been tested with water at Byron Jackson. They have also been used successfully on sodium pumps in the field. The size selected is now being used on pumps for light water reactors.

If a model for this pump is built, as will be recommended, the bearings could be tested in the model.

HYDROSTATIC BEARING (FUNCTIONAL SCHEMATIC)
BASICALLY A NON-CONTACT CENTERING DEVICE

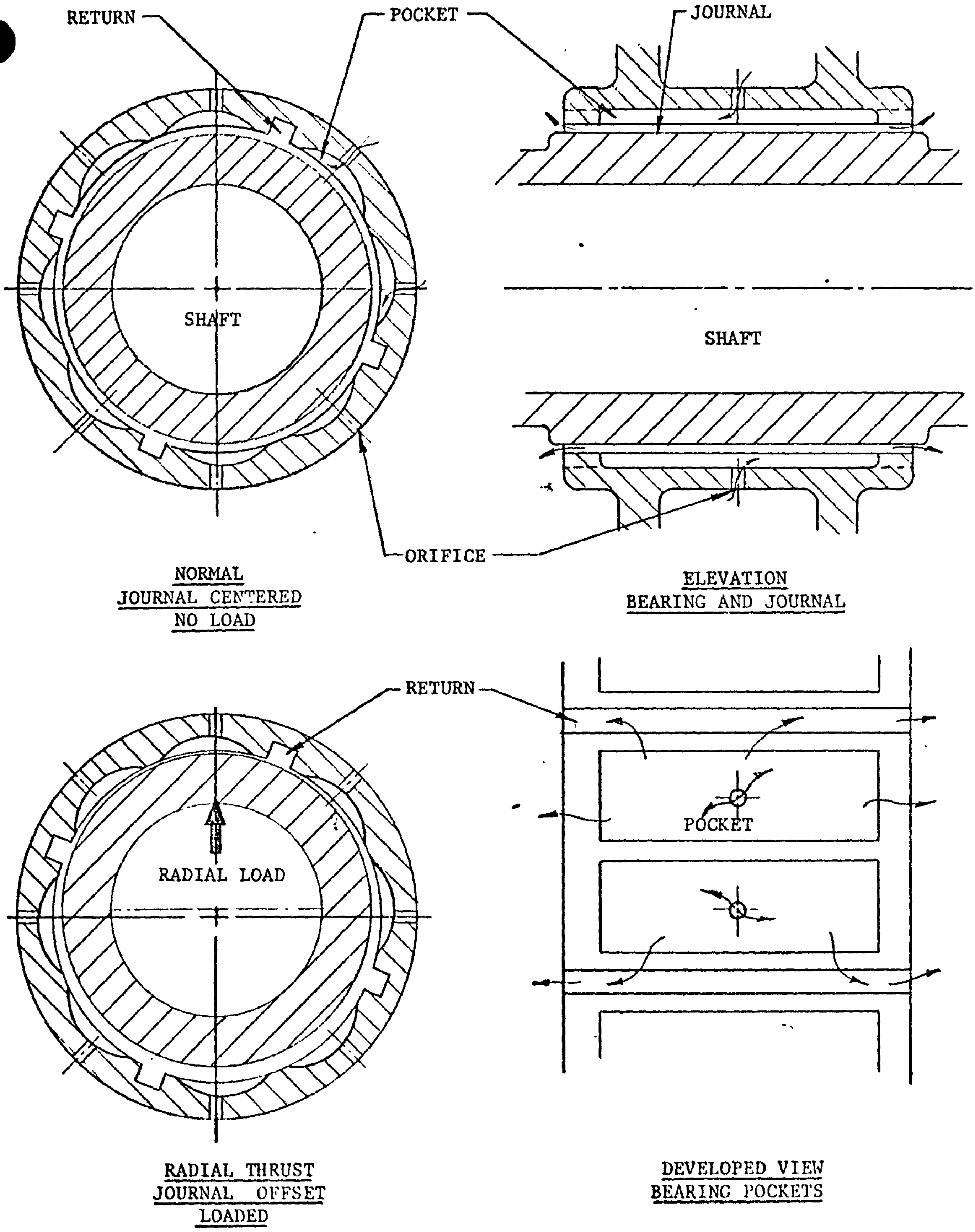


FIG. 1. HYDROSTATIC BEARING

7.6 PUMP SHAFT

The shaft is hollow except in the cold region at the top. A hollow shaft will give maximum rigidity with minimum weight. It will also give better stability under thermal transients than a solid shaft.

The lower portion of the shaft is integral with the impeller and bearing journals. It is also desirable to make the complete pump rotating assembly in one piece. Machining and grinding operations on the shaft journals, and impeller wear rings can be done with one setting. This will give accurate concentricity and squareness to close tolerances. It will also permit balancing of the complete rotating assembly as one unit.

If the shaft becomes too long for machining in one piece, it will be necessary to make it in two pieces. The coupling will consist of two welding neck type flanges bolted together. Centering of the two parts will be accomplished by a close male and female fit or with fitted bolts. The bolting will be sufficient to consider the coupling a rigid joint.

The lower portion of the hollow shaft at the impeller and journals will be machined all over, to obtain uniform wall thickness for thermal reasons and to give good balance. The small tubular section will be of a selected quality with uniform wall thickness.

The top of the shaft is solid, welded to the tubular portion. The transition from a hollow shaft to a solid shaft is made well above the hot sodium level.

A stub shaft at the top is in effect, a pump coupling, a spacer, and a mounting device for the mechanical seals.

The stub shaft is coupled to the main shaft by means of a self-centering spline. The spline "teeth" take the torque. The axial load tending to separate the spline is taken by a large fastener or a series of small ones.

The seals are mounted on the stub shaft in such a manner that they can be removed from the pump for inspection or replacement without disturbing the motor. This is a desirable feature even though the seal is designed to give a very long and satisfactory life.

The seals could be mounted on a sleeve that is removable from the top end of the shaft, thus eliminating the stub shaft. A pump coupling would be required, as well as a coupling spacer to provide axial access for seal removal. This would add an extra piece and an additional bolted joint. It would increase the height of the driver mount thus making the bearing span greater. The advantages favor the stub shaft and self-centering spline, as shown on the selected concept.

SHAFT STRESS

A. At coupling - Solid shaft use H.P. at runout

$$BHP = \frac{100,000 \times 435}{3,960 \times 0.76} \times 0.83$$

$$= 11,997 \text{ HP}$$

Shaft Dia. = 10" n = 620 RPM

$$S = \frac{321,000 \times HP}{n\ d^3}$$

$$= \frac{321,000 \times 11,997}{620 \times 10^3} = 6,211 \text{ PSI}$$

Stress Allowable = 20,000 PSI for ASTM A-182 Gr. F304 at 300° (ASME Sect. III)

B. At column - Hollow shaft (Hot)

d_o = 12" outside dia.

d_i = 9" inside dia.

$$S = \frac{321,000 \times HP \times d_o}{n \times (d_o^4 - d_i^4)}$$

$$S = \frac{321,000 \times 11,997 \times 12}{620 \times (12^4 - 9^4)} = 5,258 \text{ PSI}$$

Stress allowable = 11,600 PSI for ASTM A-182 Gr. F304 at 950° F (ASME Sect. III)

C. At impeller - Hollow shaft (Hot)

d_o = 23" outside dia.

d_i = 20" inside dia.

$$S = \frac{321{,}000 \times HP \times d_o}{n \times (d_o^4 - d_i^4)}$$

$$= \frac{321{,}000 \times 11{,}997 \times 23}{620 \times (23^4 - 20^4)} = 1{,}192$$

Therefore the torsional shear stress at every section of the shaft is well below the allowable stress value for the material used at the appropriate temperature.

7.7 Axial Thrust Requirements

The pump does not have a bearing for carrying an axial thrust load. It depends on the thrust bearing used in the motor. The motor supplier must design the bearing to carry the additional load required by the pump.

The weight of the pump rotating element is estimated to be 17,700 lbs. This is of course an axial downthrust, static and at all speeds.

Double suction impellers are theoretically balanced axially. However, actually an axial thrust is possible. This is due to slight differences in pressure on the upper and lower sides of the shrouds. This is at least in part due to the difference in actual running position between the rotating shroud and the adjacent stationary wall for the two sides of the impeller. The axial thrust due to this condition could be 19,000 lbs. in either direction. This is with the pump running at full speed. Since this load is a function of pressure, it is also a function of speed. The down-thrust requirement would be lower for the slow speeds. The motor thrust bearing is therefore sized to take the highest load required by the pump.

The upthrust due to pressure at the impeller may be high enough to overcome the dead weight of the pump rotating element. It is anticipated that the motor thrust bearing will be designed to take some load in that direction. This is for protection due to sudden unforeseen thrust surges either by the pump or the motor rotor.

The coupling from the pump shaft to the motor shaft is provided by the pump supplier. It is of the rigid type, since it carries the axial thrust load. Special requirements on the motor shaft extension are made to suit the pump supplier's design.

Provision is generally made to enable some alignment adjustment between the motor and pump shafts. Size and required accuracy make the usual rabbect fit impractical. Some form of doweling in position or fitted face keys to set and maintain alignment will be required.

7.8 Interface with Motor

The motor, though not furnished by the pump supplier, is nevertheless an integral part of the pump motor assembly. Besides supplying the power required to drive the pump, it contains the thrust bearing. The weight of the pump rotating element, as well as any axial hydraulic thrust, is taken by the thrust bearing. A summary showing the axial thrust requirements of the pump under all operating conditions is shown on the table of weights and thrust.

Utilizing the motor thrust bearing to carry the axial pump thrust eliminates the need for one in the pump. This avoids the duplication of a lubrication and cooling system. Since the seal would normally be located below the bearing were it in the pump, it would be much more difficult to remove the seal. It is highly desirable to utilize the motor thrust bearing to carry the additional axial thrust load required by the pump.

If a non-reverse is required for the pump-motor assembly, it should be placed in the motor. Also, if a flywheel is required, it could easily be placed in the motor. The reasons are similar to those enumerated for the thrust bearing.

The interface from motor to pump must be resolved mutually. The squareness and concentricity between the motor shaft and its mounting surface to the pump must be held to close tolerance. Provision shall be made for inspection to the prescribed limits.

8. SEAL AND SEAL HEAT EXCHANGER

8.1 SEALS

1.0 OBJECT

The objective of this study is to provide a conceptual design for the shaft seals in the liquid metal recirculating pump for Development Hot Leg Large Scale Breeder Reactor Plant.

2.0 SHAFT SEAL CONSTRUCTION AND OPERATION

2.1 Details and components of the shaft seal assembly are shown on assembly drawing 1E-3996.

2.2 The shaft seal assembly is based upon a modular design in which the mechanical seals, components of the seal cooling system, the pump stub shaft, the pumping ring and the seal flanges and housing are assembled as a complete cartridge. This entire package and the lubricating oil for the mechanical seals provide the sealing system of the argon cover gas.

2.3 The modular design permits removal of the assembly from the pump without removing the driver.

2.4 The cartridge is equipped with a cooling system which removes heat generated primarily by the seals at the interfaces. The cooling system consists of the two pumping rings, the heat exchanger, and the fan. When the shaft rotates, the pumping rings develop pressure differentials which result in circulation of oil past the seals and through the heat exchanger. The externally mounted fan forces ambient air, which acts as the coolant, through the tubes of the exchanger.

2.5 Plates mounted on the upper flange are inserted into the sleeve section of the pumping ring when the cartridge is out of the pump. These plates are locking devices which prevent axial movement in the seals and locate the seals to establish correct spring compression. The plates are removed after the cartridge is installed and the motor is coupled to the shaft.

2.6 Refer to sectional drawing 1C-3571 and material of construction sheet SK-0002-102 for identification and location of parts in the cartridge.

3.0 SEAL ELEMENTS

The basic seal assembly with identified components is shown in drawing 1T-15011. The functions of the major components are described as follows:

3.1 Rotating Faces: The rotating face is a gasket mounted, pin driven component made of carbon graphite. The front surface is precision lapped and polished and mated against the stationary face to provide the dynamic sealing interface. Static sealing of the rotating face to the retaining and locating component is provided by the rotating face seat gasket. The turning force is transmitted from the shaft by the rotating face drive pin.

3.2 Stationary Face: The stationary face has a stainless steel body with a hard metallic overlay which is precision lapped and polished to provide the sealing surface with the rotating face.

3.3 U-Cup: The U-Cup is an elastomer gasket which provides static sealing of stationary face.

3.4 U-Cup Follower: The U-Cup follower is a stainless steel component which maintains the position of the U-Cup gasket within the stationary face and also transmits the spring load to the stationary face.

3.5 Stationary Face Retainer: The stationary face retainer is a stainless steel component which axially holds the stationary face during assembly and disassembly operations. After the assembly is complete, the retainer serves to lock the stationary face against rotation through the stationary face drive key.

3.6 Coil Spring: Coil springs are used to maintain interface contact under zero pressure conditions and also to supplement the interface loading by hydraulic pressure.

4.0 BASIC DESIGN CONSIDERATIONS

The basic design configuration and features of the shaft seal are a result of experience and application of mechanical shaft seals in both the laboratory and in the field. The primary design considerations are described as follows:

4.1 Rotating Face

4.1.1 The rotating face is made of carbon graphite and is designed to implement the "balance" of the shaft seal.

4.1.2 Rotating the component made of carbon provides greater dissipation of heat generated by the seals than if the carbon were the stationary face.

4.1.3 The compressive strength of the rotating face is more than adequate for the oil pressures maintained in the seal cavities. The face is designed for applications up to 1000 psi in pressure.

4.1.4 Since the carbon sealing face area is important in maintaining the seal balance and the carbon material is subject to wear over a period of time, the length of the face section is significant in establishing the service life of the seal. The initial face length is designed to be 0.10 of an inch and the allowable value of total wear is .05 of an inch. The maximum allowable average wear rate for 20,000 hours or 2-1/4 years is 2.5×10^{-6} inches per hour.

4.1.5 Because the seal is designed to provide a low leakage, the sealing surfaces are subjected to a marginal lubricating film. This condition requires that the wearing member be made of a material which can maintain its integrity under this boundary lubricating condition. Experience has shown that carbon graphite satisfies this requirement best, so that this selection was made for the rotating face.

4.1.6 Besides being used in many other mechanical seal applications (including shaft seals in all Byron Jackson primary reactor coolant recirculating pumps), carbon material is used as the rotating face for the oil seal in the liquid sodium recirculating pumps supplied to Kawasaki Heavy Industries (K.H.I.). The seals in this application were tested and qualified in the Byron Jackson Seal Testing Laboratory.

4.1.7 General physical properties of the carbon material are shown in the Appendix.

4.1.8 No problems are anticipated with the carbon operating with some contact with the argon gas. The pr perly vented and pressurized seal depends on the oil for its lubricating film so that the exposure to the argon occurring on external areas does not affect seal operation. Because of the inert nature of the argon the carbon is not susceptible to chemical or corrosive attack by the gas.

4.1.9 The flatness and finish of the sealing surface is very important in the performance of the seal. Flatness of the surface is obtained by lapping and is specified to be two helium light bands total or less. The surface finish is specified to be 10 microinches, root mean square (RMS).

4.2 Stationary Face

4.2.1 The stationary face is mounted on the balance diameter and is designed to load against the rotating face by the action of the hydraulic force and the coil springs.

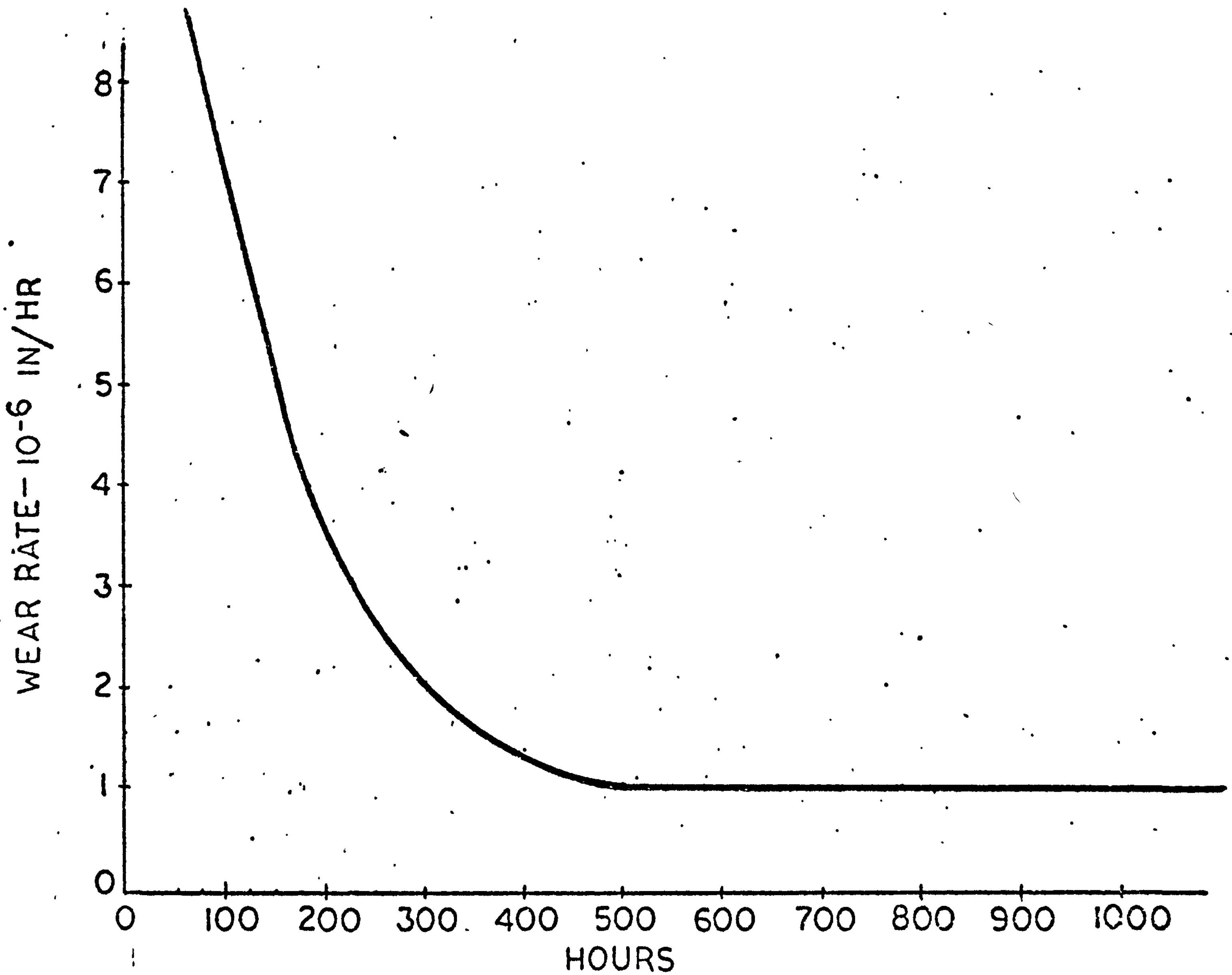
4.2.2 To facilitate the assembly and disassembly operations, the stationary face is locked axially against the coil springs by the action of the stationary face retainer as it is fastened to the flange. Installation of the rotating face completes the loading of the springs, as the stationary face is pushed back and the springs are compressed to the design working length.

4.2.3 The springs permit the stationary face to experience axial shaft movements and still maintain the sealing interface with the rotating face.

4.2.4 The stationary face is held against rotation by the stationary face drive keys which are installed in the stationary face retainer and fit into axially located keyways on the outer diameter of the stationary face.

4.2.5 The hard surface overlay that comprises the dynamic sealing surface is Stellite No. 1. This material is commonly used in shaft sealing applications and is used in the KHI oil seal and in the oil seal for the LMFBR sodium pump at the Atomics International Liquid Metal Engineering Center Pump Seal Test Facilities.

4.2.6 The hard surfaced sealing area requires a precision lapped finish with a flatness within two helium light bands and a finish of 5 microinches (RMS).



CARBON FACE VS. LENGTH OF OPERATION
WEAR RATE

FIGURE 1

4.3 Coil Springs

4.3.1 A multiple number of coil springs are used to provide a pressure of 15 to 20 psi against the rotating face.

4.3.2 The spring material is Inconel X-750 and material stress analysis is calculated using normal spring formulae. (See Appendix).

5.0 ANTICIPATED SEAL LIFE AND WEAR RATE

5.1 The life of the shaft seal depends primarily on the face area remaining on the carbon member and on the condition of the sealing interface so that leakage is maintained within acceptable values.

5.2 Wear Rate

5.2.1 Significant wear is generally associated with one or a combination of the following conditions:

5.2.1.1 Initial run-in period.

5.2.1.2 Number of start-ups and start-up conditions.

5.2.1.3 Change in sealing surface height, flatness and wear pattern from normal wear and tear under prolonged operation.

5.2.1.4 Abrasive solid foreign material in the sealed fluid.

5.2.1.5 Operation of seals in a gaseous environment (improper venting of seal cavity).

5.2.1.6 Abnormal operating conditions (pressure, and temperature, beyond specified values).

5.2.2 Correct operating procedures can eliminate the effects of the wearing conditions of Paragraphs 5.2.1.4 and 5.2.1.5. Control of abnormal operating conditions is difficult as they may occur unexpectedly. The remaining conditions contribute to the normal wear throughout the total operating period.

5.2.3 Factory tests have shown that immediately after initial start-up and for the subsequent period of about 50 hours, wear measurements are inconsistent, although generally being a relatively high value. This condition is a result of the initial surface characteristics and the wearing away of surface irregularities. After this primary run-in period the wear becomes more stable and the average wear rate, being a function of time, decreases as the period of operation progresses.

5.2.4 Calculated values of average wear rate and time of operation of carbon face oil seals for the LMFBR coolant pumps have provided data to generate the curve shown in Figure 1. This data was obtained from factory tests of mechanical shaft seals under various conditions of pressure, temperature, and speed.

5.2.5 After approximately 500 hours of operation the wear rate becomes nearly constant at 1.0×10^{-6} inches per hour. Sufficient test evidence has substantiated this wear rate up to 1000 hours of operation and one test has shown that this wear rate was valid to nearly 3000 hours.

5.2.6 The extrapolation of the wear rate to determine seal life for 10,000 hours or more may not be entirely valid, although any significant acceleration in wear rate is not anticipated until a large portion of the face length has been worn off because of normal operation. This assumption is based upon test data being substantiated by actual operation in field units where inspection of shaft seals with over 20,000 hours of operation have shown satisfactory face patterns with remaining face lengths sufficient to provide continued operation.

5.2.7 The two carbon faces tested for the KHI oil seal under similar operating conditions as the LSBRP oil seal experienced wear rates of 0.2×10^{-6} and 0.6×10^{-6} inches per hour over a test period of 629 hours.

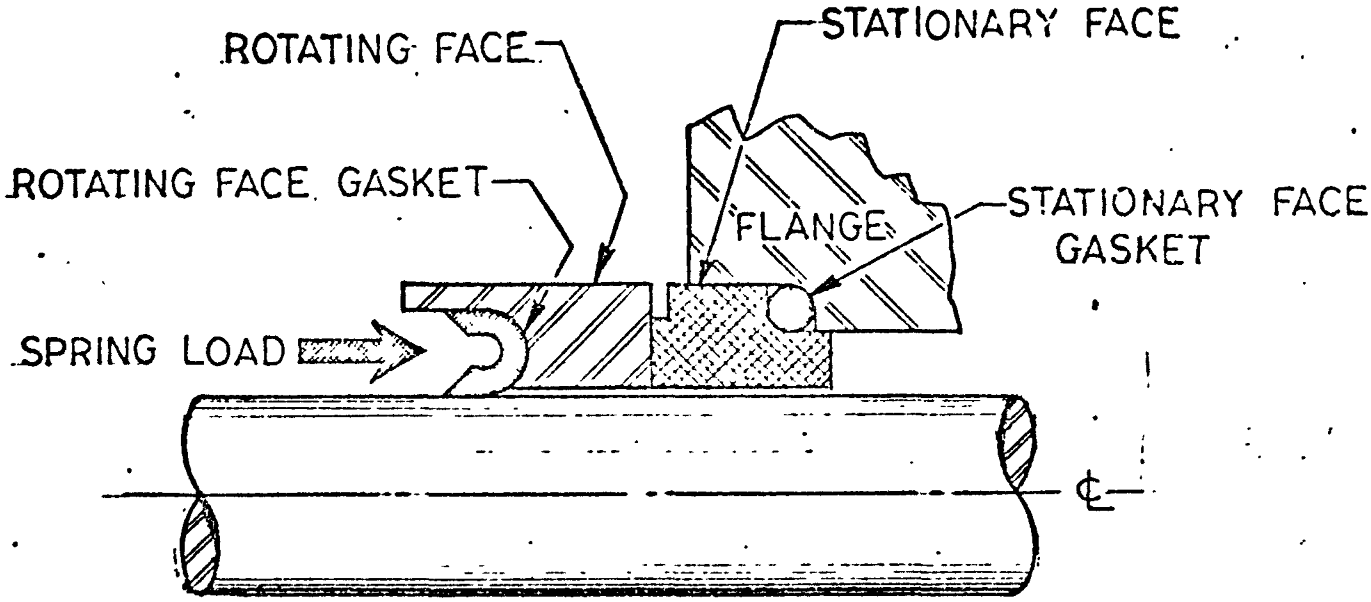
5.2.8 Based on the wear rate values of the KHI oil seal and of other shaft seals, it is anticipated that the wear rate for the LSBRP oil seal will be less than the maximum allowable average wear rate of 2.5×10^{-6} inches per hour (Para. 4.1.4)and seal life will be over 20,000 hours or 2-1/4 years.

6.0 SEAL THEORY

6.1 Face type mechanical seals, regardless of how they are constructed, are generally classified as unbalanced, where hydraulic pressure acts against the entire seal face, or balanced, where hydraulic pressure acts only on part of the seal face.

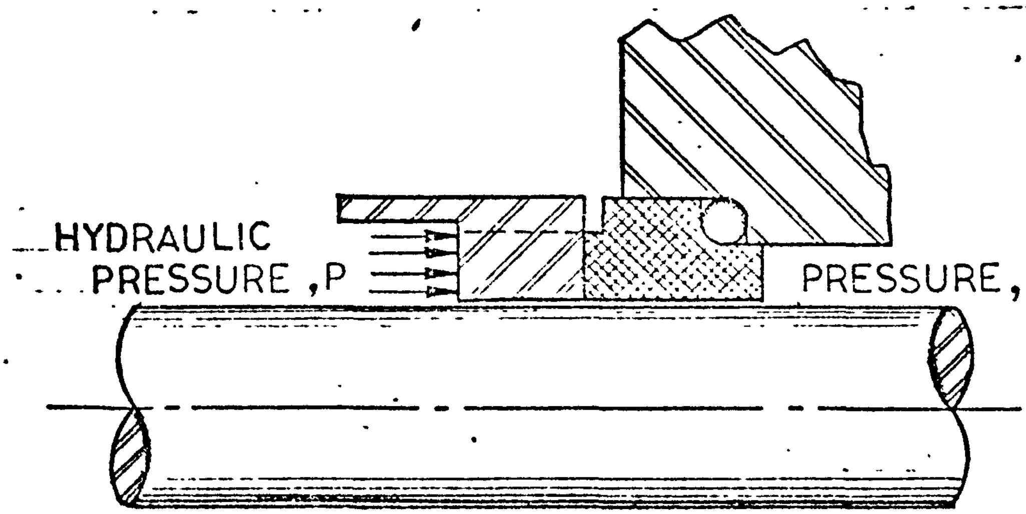
6.2 The Unbalanced Seal

6.2.1 A typical unbalanced seal configuration is shown in Figure 2. Note that the stationary face is gasketed to the flange and the rotating face is gasketed to the shaft with a spring load to hold the faces together at zero pressure and to prevent separation resulting from shaft end play and vibration.



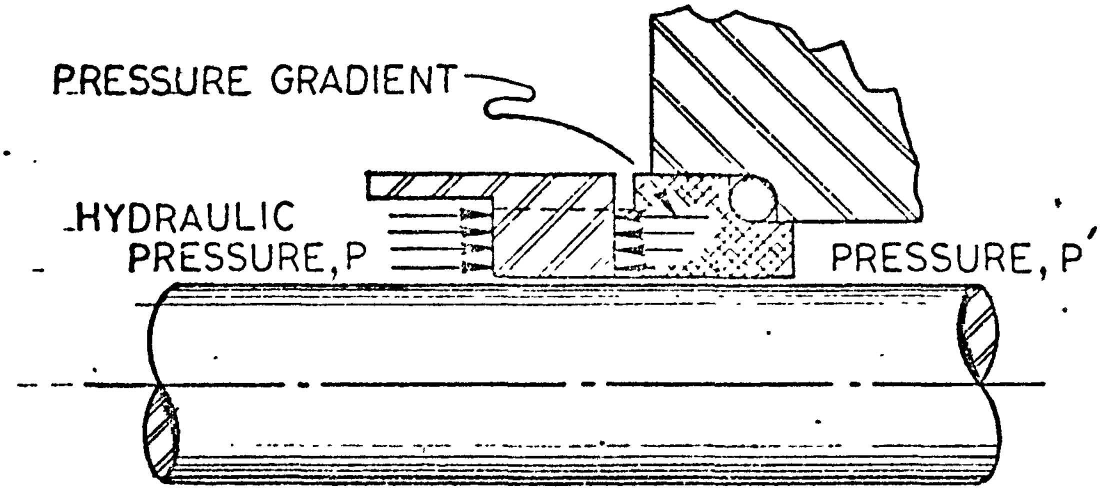
UNBALANCED SEAL CONFIGURATION

FIG. 2



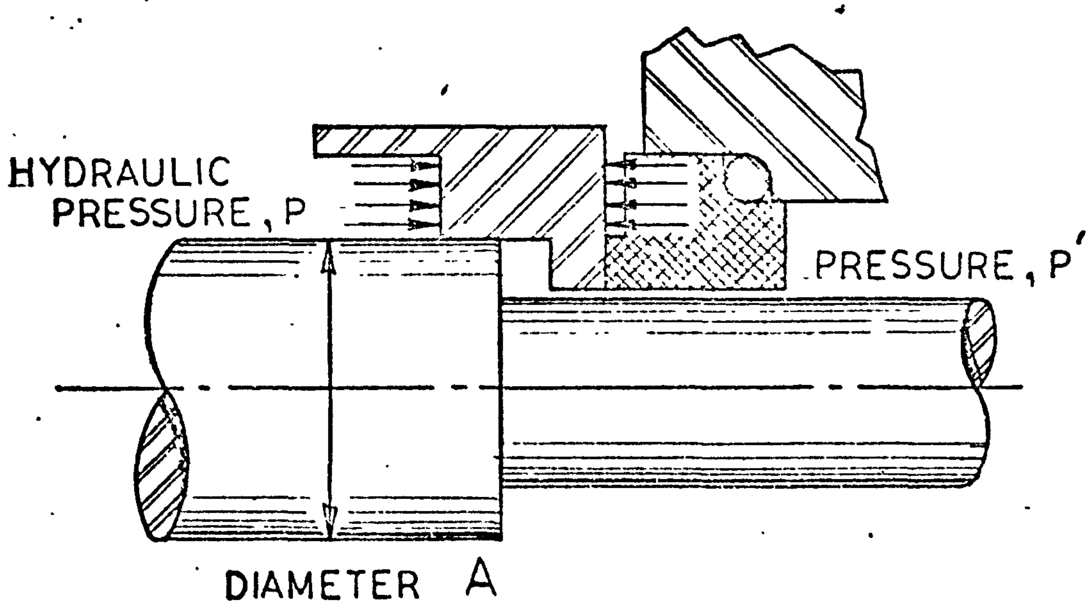
PRESSURE ACTING ON UNBALANCED SEAL FACE

FIG. 3



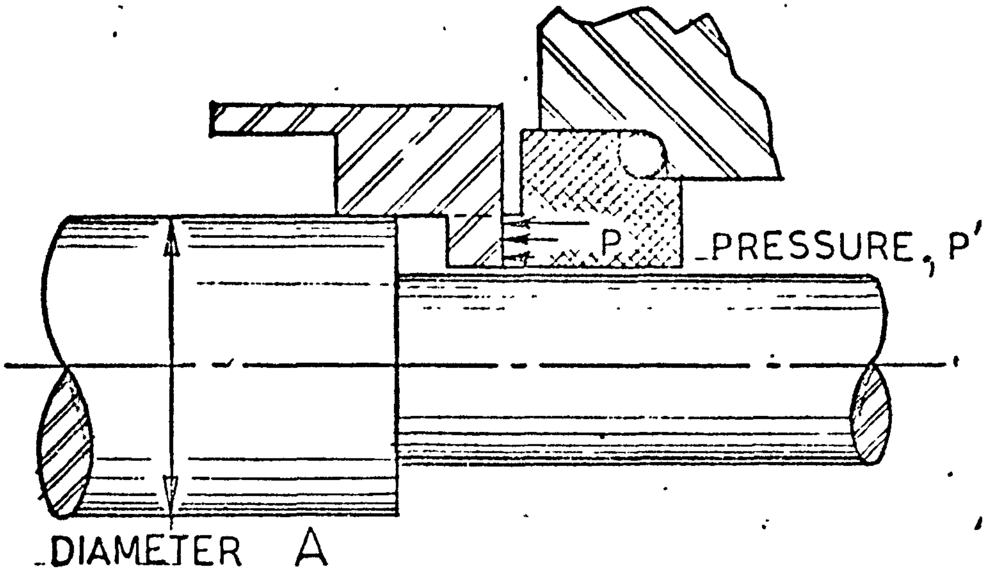
PRESSURE GRADIENT ON UNBALANCED SEAL FACE

FIG. 4



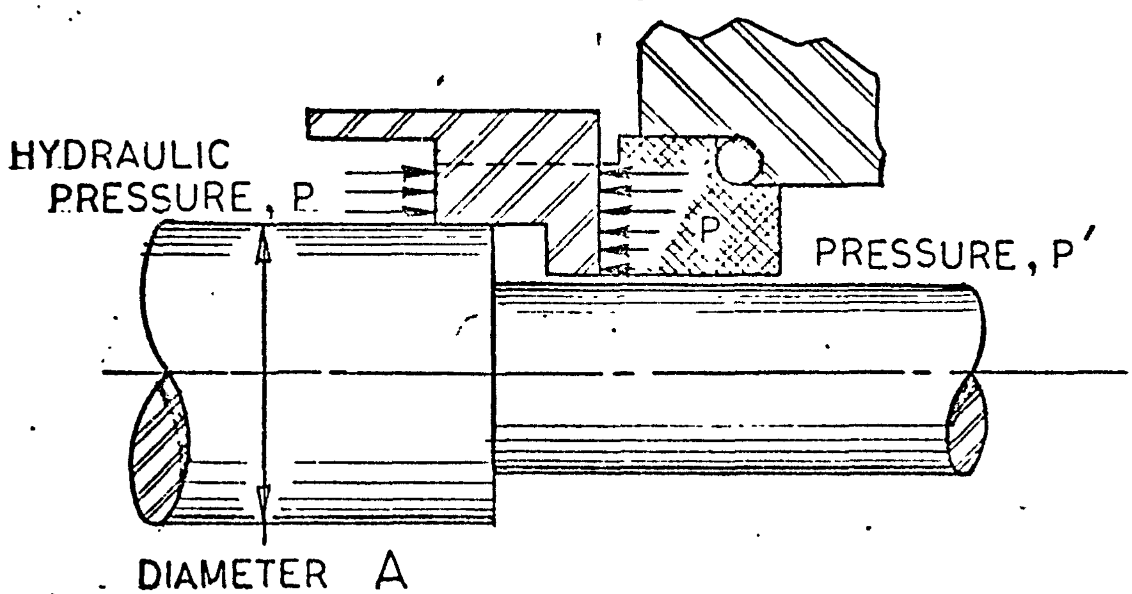
OVERBALANCED SEAL

FIG. 5



PRESSURE GRADIENT ON
OVERBALANCED SEAL

FIG. 6



BALANCED SEAL WITH
PRESSURE GRADIENT

FIG. 7

6.2.2 To consider the effects of hydraulic pressure, the seal configuration is reduced to Figure 3 in which the rotating face gasket and spring are removed. Note that the hydraulic pressure, P, acts directly against and across the entire sealing surface. This pressure forces the rotating face against the stationary face. To prevent damage to the faces from excessive heat generation, a lubricating film of fluid must be maintained between the faces. This film enters between the faces at the hydraulic pressure and exits at some lower pressure P'.

6.2.3 The lubricating film produces a pressure gradient across the faces which tends to separate the faces. As shown in Figure 4, the gradient exerts considerably less force than the opposing hydraulic pressure. A net force acting to close the face exists and this force increases as the hydraulic pressure increases. If the hydraulic pressure becomes too great, the lubricating film will not form and excessive heat will cause damage to the surfaces and the seal will fail.

6.2.4 Generally, unbalanced shaft seals perform satisfactorily up to a pressure differential (P - P') of 100 - 150 psi. Other factors such as type of sealed fluid, rotational speed, face materials, and temperature of sealed fluid may vary the pressure limits.

6.3 The Balanced Seal

6.3.1 The efficiency and life of a face type mechanical seal is greatly increased by reducing the effect of the hydraulic loading force on the sealing surface. This reduction is achieved by incorporating a design feature known as seal balance.

6.3.2 Consideration is first made of a seal which is overbalanced and is shown in Figure 5. Once again the rotating face gasket and the spring load have been deleted to show only the pressures. Note that a shoulder of diameter "A" has been included on the shaft. Diameter "A" is known as the balance diameter. In this overbalanced seal the hydraulic pressure exerts a load against the rotating face in exactly the same manner as it did on the unbalanced seal. The pressure, however, also acts on the opposite end of the face such that the area beyond the balance diameter "A" is in hydraulic equilibrium.

6.3.3 A pressure gradient from P to P' is formed as shown in Figure 6. The net result is a separating force, provided that the spring load is less than the force developed by the gradient. This seal configuration is not acceptable because it usually results in excessive leakage.

6.3.4 With balance diameter "A" maintained, a condition can be created to counteract the effects of the pressure gradient. An increase of the outside diameter (O.D.) of the stationary face beyond diameter "A" will provide an area which continues to be loaded by the hydraulic pressure. This configuration is shown in Figure 7. Areas beyond the O.D. of the stationary face are in equilibrium, and thus a resulting force acts only upon the area between diameter "A" and the stationary face O.D.

6.3.5 An increase in the loading area will tend to reduce the generation of a lubricating film (seal leakage is decreased). A decrease in the loading area will increase the effect of the separating force (seal leakage is increased). With known operating conditions the loading area is designed to provide the desired face pressure such that a lubricating film is maintained thereby achieving long life with satisfactory seal performance.

6.4 The LSBRP Oil Seal

6.4.1 Theory substantiated by both field and test experience has shown that a loading area of 60% to 80% of the sealing area provides the optimum combination of leakage and life in most shaft seal applications. The balance design value of the oil seals for the LSBRP pump falls within this range.

6.4.2 Note that in the design of the oil seals the balance diameter is located on the stationary member. This design, however, does not change the hydraulic loading effects and was established to achieve the modular concept with the greatest ease of assembly.

7.0 SEAL LEAK RATES

7.1 Balanced, face type mechanical seals for nuclear power coolant pump applications are designed for high reliability and long life. To meet this goal, operation with a finite lubricating film between the rotating and stationary faces is essential. This film must be assured throughout reasonably predictable operating conditions. The balanced face design provides the primary feature in achieving a lubricating film.

7.2 The presence of a finite film between the faces implies finite seal leakage. This leakage may be of such low value that it is not measurable without special apparatus and procedures. The term "zero leakage", therefore, indicates that the leakage is too small to be measured or visually observed.

7.3 Measurable leakage occurs and varies under many influencing factors as any condition which affects the development of the seal pressure gradient can significantly change the rate of seal leakage.

7.4 Primary factors which influence seal leakage are pressure, temperature and speed. Generally, leakage is less at low pressure, high temperature, and low speed. Transient conditions of pressure, temperature and speed may also result in variations of seal leakage. Other factors which may affect leakage are axial and radial shaft movement, seal face eccentricity, and seal face condition - flatness, and surface finish.

7.5 Leakage from seals operating under conditions of the LSBRP is in the range of zero to 10.0 cm^3/hr.

7.6 Actual leak rate predictions specifically for the LSBRP oil seal would have to be established from a full-scale component evaluation test of the seal.

8.0 SHAFT SEAL LUBRICANT AND OPERATING TEMPERATURES

8.1 The recommended lubricant for the shaft seal is Mobil SHC 824. This selection is based upon the lubricant vendor's recommendation from their testing of lubricants under conditions similar to those of the LSBRP primary pump oil seal. A test program of the seal should include operation with this oil to verify its ability to provide satisfactory lubrication under the operating conditions.

8.2 The recommended normal operating temperature is 125°F. or less. The maximum acceptable normal operating temperature is 150°F. These temperatures are recommended for sustained operation in order to achieve the minimum service period of 20,000 hours.

8.3 It should be noted that the lower the oil operating temperature, the greater will be the reliability of the seal. It is imperative, therefore, that the oil cooling system be capable of maintaining the oil temperature at or below 150°F. throughout all phases of normal operation.

8.4 Temperatures above 150°F. will not cause immediate failure, but the seal life will be significantly decreased by continuous operation at elevated temperatures.

8.5 The maximum allowable oil temperature is 250°F. based on the limitation of the elastomer gaskets. Immediate failure is not likely at this temperature, although a controlled shutdown and inspection of the shaft seals is recommended if this temperature is achieved.

APPENDIX

STRESS CALCULATIONS *

C_1, spring index
d, wire diameter
G, modulus of elasticity in shear
k, stress concentration factor due to curvature and direct shear
k_c, stress concentration factor due to curvature
k_s, stress multiplication factor due to direct shear
K, spring rate
ℓ_1, free length of spring
N, number of active coils
P, load
R, mean radius of helix
s_s, shearing stress

GIVEN: Spring O.D. = 0.625 in., $d = 0.059$ in., $G = 11 \times 10^6$ psi
$\ell_1 = 2.055$ in., $N = 11.5$, Working length = 1.125 in., Material = Inconel X-75
Solid height = 0.797 in.

TO FIND: ① Spring stress @ solid height
② Spring stress @ working length

Spring rate, $k = \dfrac{d^4 G}{64 R^3 N}$

$d^4 = (0.059)^4 = 1.212 \times 10^{-5}$
Spring mean diameter = 0.625 - 0.059 = 0.566 in.
$R = 0.566 \div 2 = 0.283$ in.
$R^3 = 0.0227$ in^3

$$k = \frac{1.212 \times 10^{-5} \times 11 \times 10^6}{64 \times 0.0227 \times 11.5}$$

$$k = 7.98 \text{ lb/in}$$

① Load at solid height, $P_{SH} = (2.055 - 0.797)\,7.98 = 10.04$ lb

Spring stress at solid height, $S_{S_{SH}} = \dfrac{k_s 16 P_{SH} R}{\pi d^3}$

$d^3 = (0.059)^3 = 2.054 \times 10^{-4}$ in^3

$k_s = \dfrac{k}{k_c}$ where $k_c = \dfrac{4C_1 - 1}{4C_1 - 4}$

*SPOTTS, DESIGN OF MACHINE ELEMENTS, Prentice-Hall, Inc., Englewood Cliffs, N.J., 1958

$$k = k_c + 0.615/C_1$$

$$C_1 = \frac{2R}{d} = \frac{2 \times 0.283}{0.059} = 9.593$$

$$k_c = \frac{4(9.593)-1}{4(9.593)-4} = \frac{37.37}{34.37} = 1.087$$

$$k = 1.087 + \frac{0.615}{9.593} = 1.151$$

$$k_s = \frac{1.151}{1.087} = 1.059$$

$$s_{s_{S.H.}} = \frac{1.059 \times 16 \times 10.04 \times 0.283}{\pi \times 2.054 \times 10^{-4}}$$

$$s_{s_{SH}} = \underline{\underline{74,600 \text{ psi}}}$$

② Load at working length, $P_{WL} = (2.055 - 1.125)7.98 = 7.42$ lb

Spring stress at working length, $s_{s_{W.L.}} = \frac{k_s \, 16 \, P_{WL} R}{\pi d^3}$

$$= \frac{1.059 \times 16 \times 7.42 \times 0.283}{\pi \times 2.054 \times 10^{-4}}$$

$$s_{s_{W.L.}} = \underline{\underline{55,100 \text{ psi}}}$$

The minimum elastic limit and maximum solid stress for Inconel compression springs made with 0.059 of an inch wire diameter is 90,000 psi. The allowable working stress for severe service is 54,000 psi.*

The calculated solid height stress is within the limit of allowable stress for the solid height and the working stressing capability is near the severe duty stress, although the actual spring service is more closely in the region of average service.

*From Machinery's Handbook, 19th Edition, Industrial Press, New York, N.Y., 1973.

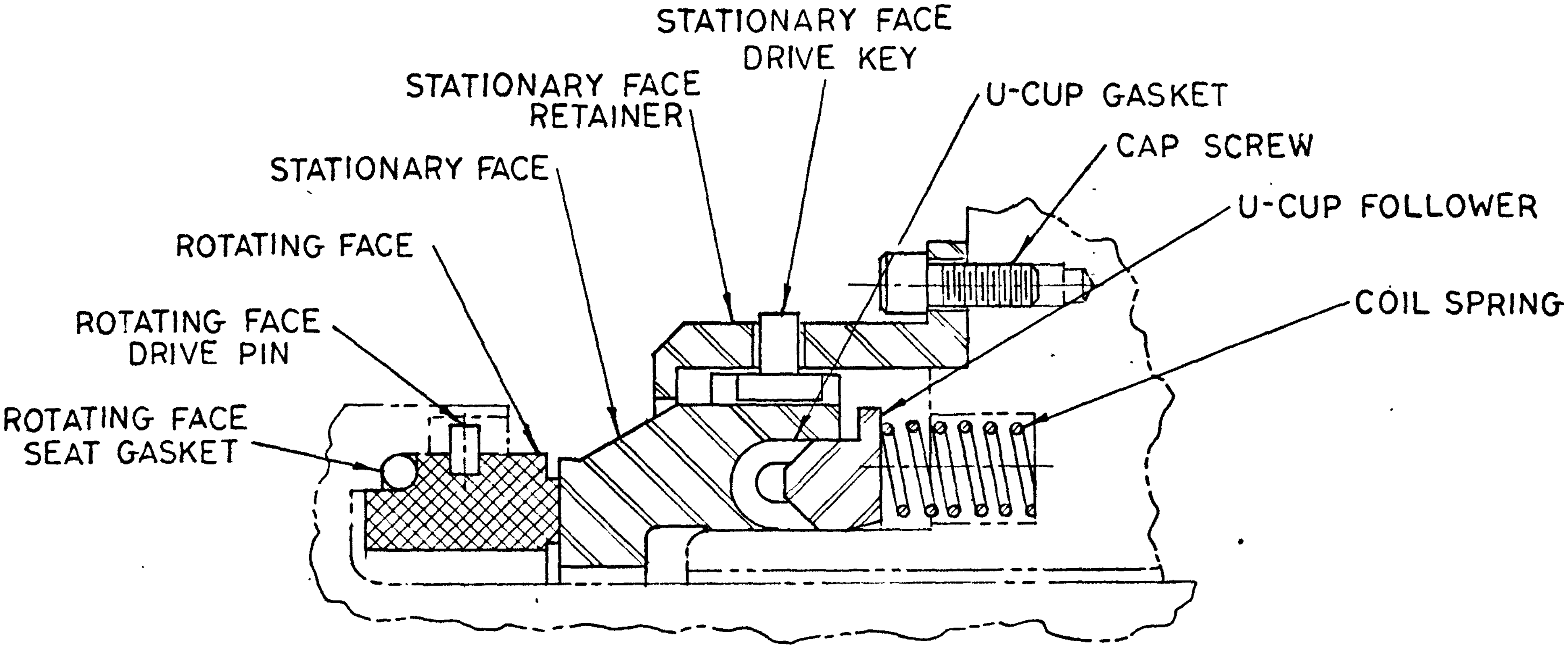
ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION 60.

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT. — 1/64
TURN — 1/64	2 TO 6 FT. — 1/32
	6 FT. AND OVER — 1/16

REVISIONS

LTR	DESCRIPTION	DATE	APP'D
θ	ORIGINAL		



UNLESS OTHERWISE SPECIFIED

DIMENSIONS ARE IN INCHES

DECIMAL TOL. .X ± .XX ± .XXX ± — ANGLE TOL. ±

√ SURFACE ROUGHNESS PER MIL — STD 10

DRILLED HOLES √ AS DRILLED
TAPPED HOLES √ AS TAPPED

DO NOT SCALE DRAWING.
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005/.015
ALL MACH. CORNERS/FILLETS .005/.020.

BJ ORDER NO.	761-G-0002	REF. NO.	
DRAWN BY	A. COLEMAN	DATE	2 SEPT. 76
CHECKED BY	L. TAKUMI	DATE	2 SEP. 76
CONTRACT NO.			
DESIGN APP'D.		DATE	
CUSTOMER APP'D.		DATE	

BORG WARNER® Byron Jackson Pump Division
BORG-WARNER CORPORATION
LOS ANGELES DIVISION

DRAWING TITLE: SEAL ASSEMBLY

CODE IDENT. NO.	DRAWING NO.	
	IT-15011	
SCALE NONE	WEIGHT	SHEET

8.2 SEAL HEAT EXCHANGER

1.0 DISCUSSION

The shaft sealing unit is equipped with an integral cooling system that is required to maintain the temperature of the seal lubricating oil at acceptable levels of operation.

Heating of the lubricant is anticipated to occur primarily from the friction resulting at the interface between the stationary and rotating faces. As much as eighty-five per cent of the total heat load is estimated to be from this source.

The remaining heat load is a result of energy losses from the oil recirculation pumping ring and from heat conduction along the shaft from the liquid metal.

2.0 HEAT EXCHANGER DESIGN

Previous experience in Byron Jackson liquid metal pumps has shown that acceptable oil temperatures have been maintained by a heat exchanger in which ambient air serves as the coolant for the heated oil.

Components of the oil cooling system can be seen in the Seal Cartridge Drawing 1C-3571. This heat exchanger utilizes a pumping ring (12-1-4) to circulate oil past tubes (12-1-3-5) through which air is forced by a fan on the stub shaft. Typical oil and air flows are shown in the Conceptual Seal Cartridge Design LMFBR Pump Drawing 1C-3572. The major advantage of this system is that the cooling system is always operating when the pump is in operation.

A disadvantage of this arrangement is that fan capacity and pumping ring capacity are decreased when the pump is driven at pony motor speed. At lower speed, however, the heat load from the shaft seals is also reduced, so that cooling requirements may not be significantly affected.

In the event higher air and oil flows must be maintained at low speeds, an alternate arrangement would be to use an external blower and an oil pump instead of the shaft mounted fan and pumping ring. This type of design was used for the seals in the Enrico Fermi LMFBR pump. These external units would provide continuous flows through all ranges of shaft speed, although failure of the oil pump or blower would cause loss of cooling to the seals.

Utilization of both systems (integral fan and pumping ring combined with external blower and pump) would insure cooling at all speeds and during loss of one or both of the external cooling units.

The size and number of tubes required to provide adequate heat transfer surface areas results in a seal cartridge of significant size both in weight and volume. Some reduction in size is possible by using tubes with more effective heat transfer characteristics. Tubes that merit consideration are those having a fluted configuration, finned tubing, or tubes having material on the outer diameter raised to form spine-like projections.

A heat exchanger in which the oil is circulated through a cast aluminum housing with external flat longitudinal fins has been effective in the LMFBR pump for Kawasaki Heavy Industries. This design is also a possibility for the LSBRP hot leg pump application.

3.0 FUTURE STUDY AND DEVELOPMENT

The basic cooling concept of extracting the heat load with forced air has been used in both the Enrico Fermi and Kawasaki Heavy Industries LMFBR coolant pumps. This design is also planned for the recirculating coolant pump of the Clinch River Breeder Reactor Plant (CRBRP). The concept is considered to be acceptable for the LSBRP hot leg pump, but the actual means to implement the design requires further evaluation.

Results and data from the performance of the seals in the CRBRP will be a valuable source for establishing the design of the seal and heat exchanger components for the LSBRP.

Because no data is available from a seal of the size and design anticipated for the LSBRP unit, development and testing in the area of lubricant cooling requirements are recommended to determine fan performance, oil pumping ring performance, and oil and air heat transfer characteristics.

9. TANK SODIUM LEVEL CONTROL

The liquid sodium level in the pump has to be controlled to a predetermined level. If the level variations are greater than those specified for design, damage to pump components may occur. An abnormally high level could damage the pump shaft seal and radiation shield material by high temperature and sodium contamination. An abnormally low level could result in severe temperature gradients in components normally submerged in liquid sodium. This level has to be controlled for both static and running conditions. A safe level at elevation of minus 10 feet could be chosen for the desired sodium level which is also the elevation of the overflow outlet. For static condition, a cover gas with gage pressure equal to 10 feet of liquid sodium will be applied on top of the sodium to keep it at the design level. This is because the sodium level in the reactor is at zero feet elevation. For running condition, the situation is somewhat more complex. First, at the impeller suction located at elevation minus 21 feet, the NPSH is 30 feet, which is lower than atmospheric pressure and could result in getting the cover gas into the pump volute and impeller. Therefore a close clearance leakage path is provided from discharge to suction as mentioned in Section 7.4. However, by doing this, there is also a positive leakage flow from the discharge to the pump outer tank through the close clearance leakage path. This leakage will flow out from the overflow nozzle and is estimated to be about 1200 GPM at design point. With this positive overflow system, the sodium level will be maintained at the overflow outlet level.

10. ELIMINATION OF GAS ENTRAPMENT

The available NPSH of 30 feet, less than that for Clinch River, could result in a vacuum on the pump. This makes it possible to get gas into the pump volute and impeller.

The return leakage from the upper hydrostatic bearing may not be enough to avoid this gas entrapment. An additional means must be provided to keep gas out of the pump.

For this pump concept, a close clearance leakage path is provided from discharge to suction at the top and bottom of the tapered static seal. This is somewhat like a static wear ring. The leakage at the bottom returns to suction. The leakage at the top will keep cover gas out of the pump and will go out through the overflow nozzle.

The clearances and length of the leakage paths will have to be designed to keep the leakage to a minimum and yet give ample protection. The leakage from the discharge to the pump outer tank will also require control so that the overflow is also kept to a minimum. The return leakage from the hydrostatic bearing will flow back to suction directly, therefore reducing the overflow by that amount. The positive overflow at different pump flow points can be checked during the cold water performance test and adjustment made as required.

11. PUMP SUPPORT CONCEPT

In this study of a sodium pump concept, the pump outer tank is mounted on a foundation at the top end called the support flange. The pump internals then are mounted on top of this support flange. There is a taper fit between the outer tank and pump internals at the pump end and in order to control this fit to a close radial clearance, the internals have to be able to move vertically at assembly. This is described in the key features, under internal taper fit,in this concept. The pump internals are supported by studs with a nut on either side of the clamp ring. By adjusting the two nuts located along the stud, the internals will be able to move up or down. The size and number of the studs will be determined to support the internal weight in the downward direction and the upthrust in the upward direction as shown on Figure 1. This upthrust is created by the pump discharge pressure at the bottom end of the internals. Since this upthrust will act on the outer tank, the bolts used to mount the outer tank to the foundation will also be subject to the upthrust. The side load of the pump internals will be transmitted to the foundation through the outer tank and the mounting ring.

After the pump outer tank and the pump internals are installed properly, a canopy-type seal will be welded between the internals to the outer tank, and between the outer tank to the foundation. These canopy-type seals are used to prevent any cover gas leakage to the atmosphere in the event that the "O" ring seals are damaged or develop leaks.

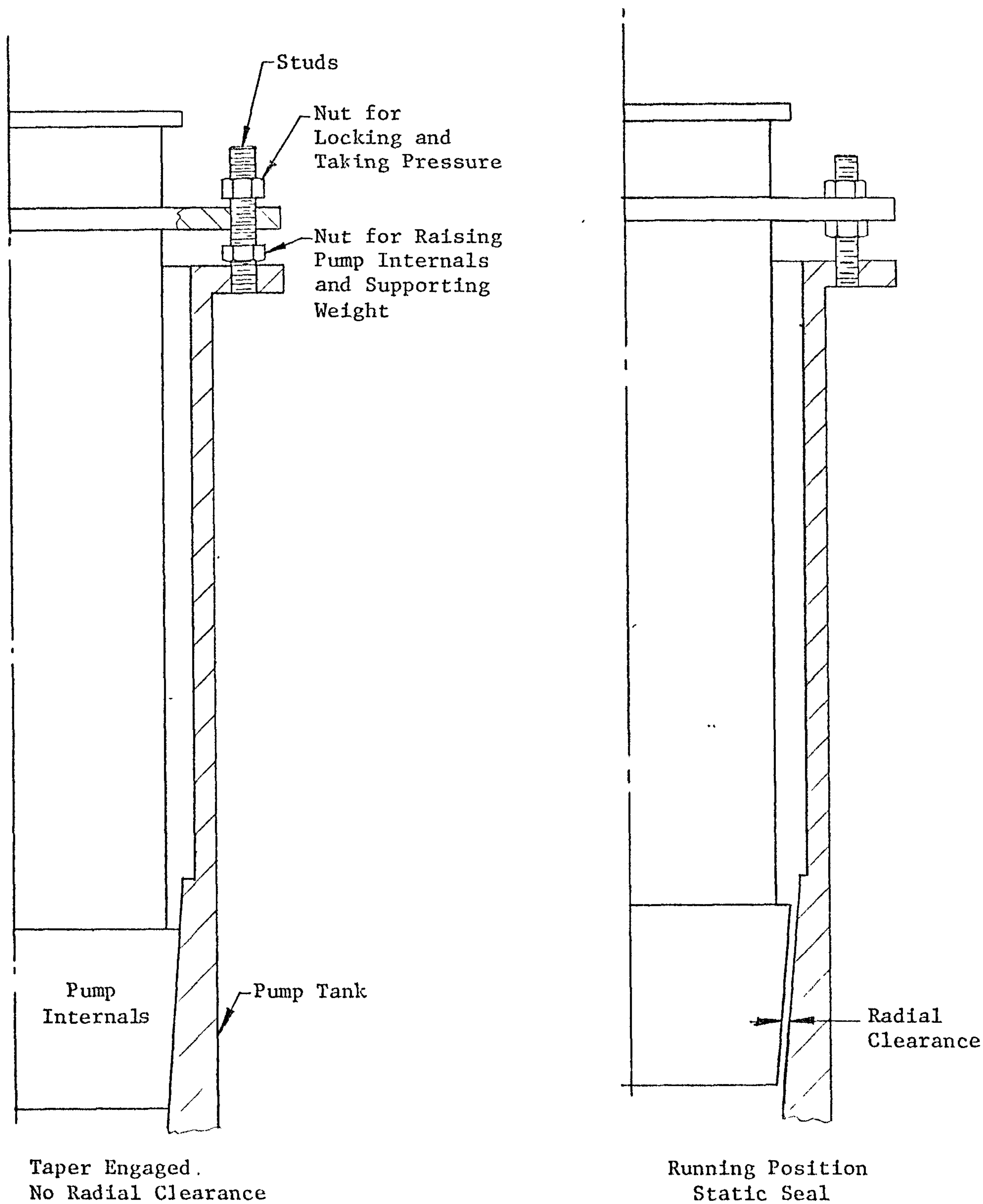


FIGURE 1 INTERNAL TAPER FIT

12. ALIGNMENT AND CRITICAL DIMENSIONS

ALIGNMENT CRITERION

The acceptable alignment criterion is one in which the manufacturing tolerances accumulate in the most adverse condition, but the rotating element will run without forcefully rubbing on any portion of the stationary parts, particularly the bore of the bearings.

METHOD OF DETERMINING MISALIGNMENT

The total misalignment is obtained by adding the manufacturing tolerances of every individual piece and the tolerances on the fits between the pieces. Therefore the manufacturing tolerance will have to be determined. In this study, the pilot diameters on each individual piece will be set to be concentric to within .005" T.I.R. Squareness between any two mounting faces on each piece is also set to be parallel within .005" T.I.R. The pilot diameters between two mating pieces will have a .006" loose fit. The pieces that will affect the alignment are the driver mount, inner tank, volute, and hydrostatic bearing. When these pieces are assembled, the accumulated misalignment is .036" from the top of the driver mount to the end of the lower hydrostatic bearing as shown on Fig. 2.

The rotating element, including motor and pump, will have the same manufacturing tolerances as before. However, when the motor is assembled to the driver mount, it is adjusted to make the motor shaft concentric to the driver mount. Therefore, the stack-up of the tolerances will consist of the motor shaft to pump shaft connection, and the shaft pilot diameter to the shaft journal diameter. Figure 3 shows how much the shaft has to move to avoid rubbing on the hydrostatic bearing when the combined motor and pump rotating element is installed in the pump. Since the total misalignment on the hydrostatic bearing is .036" from Figure 2, and there is a clearance of .030" on a side between the hydrostatic bearing and the journal, therefore, the shaft at the end of the journal will have to move only .006" from the center line of the top of the driver mount in order to fit in the hydrostatic bearing.

With this movement at the bottom journal, the shaft will move .0017" at the motor sleeve bearings. With the motor size in this study, one can expect the sleeve bearing will have about .005" clearance on a side, therefore the shaft will not rub at the motor end.

CONCLUSION

With the stack-up of tolerances due to manufacturing accumulated, the shaft is still free from rubbing the bearings. In a practical case, the changes of having all the tolerances going in one direction is remote. Therefore it can be concluded that this is a conservative approach. The tolerances selected for manufacturing would result in a favorable alignment. The shaft would stay within the bearing clearances without forced interference.

13. MATERIAL AND QUALITY ASSURANCE REQUIREMENTS

13.1 MATERIAL

Based on the evaluation of materials for elevated temperature service and a liquid sodium system, the austenitic stainless steels offer the maximum advantages for the sodium containment member and associated hardware, for maximum low service cost within realistic initial costs.

The existing state of the art includes adequate means to verify materials and facbricated joint integrity. Mechanical integrity can be obtained by sound design principles based upon experience in the design of liquid sodium pumps. Fabrication requirements is one of the most important criteria to be considered and in this area the austenitic stainless steels offer superior values.

Based on the general conclusion that austenitic stainless steels offer the maximum advantages for sodium service, more specific material call-outs can be made assuming certain forms in which the materials would be economically available, i.e., plate, forging, bar, or casting. The ASME callouts to which the major components would be procured are

Plate:	ASME SA-240 Type 316 or 304
Forging:	ASME SA-182 Type 316 or 304
Rolled Bar:	ASME SA-461 Grade 660
	ASME SA-193 Grade B7 or B8
	ASME SA-453 Grade 660 and 651
Casting:	ASME SA-613 Grade CF8 (static)
	ASME SA-613 Grade CF8 (centrifugal)
Pipe:	ASME SA-213 Type 316 (seamless)

Material specification will be based upon the requirements as set forth in the ASME Materials Specifications, Section II. Materials may be supplied to ASTM specifications provided the ASTM material is identical as to grade, composition and properties as set forth in the ASME specification. Additional special material requirements for pressure retaining and or permanent attachments to the pressure retaining boundary will be as set forth in ASME Nuclear Power Component Code, Section III, Article NB-2000. Requirements for non-pressure boundary and/or internal pump parts will be as set forth in the Design Specification and by the pump designer. Properties and additional design requirements for temperatures greater than 800°F shall be based on ASME Code Case No. 1592-7.

Manufacture

The design and manufacture of a large liquid sodium pump, must of necessity, incorporate welded construction. The importance of materials weldability in obtaining a satisfactory design at an economical cost and with a maximum degree of weldment integrity, therefore, cannot be minimized. Material weldability is no better than the weld filler used for weldments. Electrodes and filler metals should be obtained to produce a range of 4 - 9% ferrite to minimize susceptibility to microfissure cracking in the weld deposit and heat effected zone in austenitic steels.

The necessity for obtaining sodium containment equipment that will not allow leakage is a major item of importance, but one that can be obtained with existing technology of inspection media and design knowledge.

Final quality, must of necessity, be predicated on initial base metal quality. Radiographic quality castings can be obtained and internal quality verified using established techniques and qualified personnel. Surface quality can be verified by use of liquid penetrant inspection standards. The use of radiography supplemented with liquid penetrant inepction and pressure tightness confirmed by hydrostatic and helium mass spectrometer testing will provide acceptable castings for sodium containment.

Wrought material quality can be verified by use of ultrasonic and liquid penetrant inspections.

The quality of weldments for liquid metal service has to be of maximum integrity. Control of weld joints for full penetrantion and use of radiography supplemented by liquid penetrant inspection will give the maximum quality. Pressure tightness must be confirmed by hydrostatic pressure testing. Leak rates of 10^{-8} cc/sec @ 60 - 80^{o}F as confirmed by helium mass spectrometer testing are easily obtained. Leak testing at higher temperatures has not been considered not felt to be necessary. The size of equipment would create considerable testing problems not only from the standpoint of heating methods but sealing.

Sections III and IX of the ASME Boiler and Pressure Vessel Code form a basis for welding of equipment; however, it is usually necessary to amplify the welding requirements in procurement documents. Byron Jackson procedures, although based on the ASME Code are more strict for weldment in elevated temperature liquid metal service. Rather than depend on the vendor to set forth weld quality to be offered, procurement specifications should set forth minimum weld quality requirements.

13.2 Quality Assurance

The pump manufacturer shall be responsible for controlling the quality during manufacture in accordance with the requirements of ASME Nuclear Power Plant Components Code, Section III, Div. 1 to assure that all materials and parts, including subcontracting items and services conform to the requirements of the Code. The responsibility shall include performing or having performed all controls, examinations, and tests necessary to substantiate that the materials, components, parts and appurtenances confirm to the requirements of the Code.

13.3 Hydrostatic Test

The large scale breeder reactor mechanical pump is designed to ASME Boiler and Pressure Vessel Code, Section III, Class I. Therefore all the pressure containg parts must be subject to a hydrostatic test. The pressure for the hydrostatic test according to NB-6221 of ASME Section III is 1.25 times the design pressure. The design pressure for the pump tank and inlet nozzle is 30 psig, therefore the hydrostatic test pressure is 38 psig. The design pressure for the pump hydraulics and discharge nozzle is 200 psig, therefore the hydrostatic test pressure is 250 psig. All of the test parts will be blocked off by blind flanges, suitable fixtures and gaskets. Test pressure will be maintained a minimum of 15 minutes for each inch of design minimum wall thickness. Indicating pressure test gages will be used, connected directly to the part being tested. Gages will be calibrated as required by the Code. Gage range will be about double the intended maximum test pressure but in no case shall it be less than 1-1/2 nor more than four times that pressure. All hydrostatic tests will be witnessed by code inspector. All parts to be hydrostatic tested will be in finish machined condition except the connection nozzles in the outer tank. Flanges will be welded to these nozzles for hydrostatic test as shown typically on Drawing No. SK-0002-105. These flanges will also be used for the performance tests. After all tests have been completed, the test flanges will be removed and the nozzle ends prepared for welding.

14. PUMP PERFORMANCE TEST

The specification requires that these pumps shall be performance tested with cold water. These tests will be performed in accordance with the requirements of the ASME Power Test Code PTC 8.2 for Centrifugal Pumps. The motor will of course be used and tested at the same time. The pump bearings will be instrumented to check flows and pocket pressures. The seals and seal heat exchanger will be tested with the pump and motor.

The pump parts must be designed with this test in mind. For example, the nozzles on the tank will require flanges for installation in the test loop. A test stand of some kind will be required to support the pump and motor. In fact, performance testing may be second in importance only to building and assembling the pump.

Out of the many test set-ups possible, only two will be considered. One is to go below ground level in a test pit. The other is to keep the pump above ground level and support it with a structural framework.

The size and criticality of these pumps dictate that they should be tested thoroughly and accurately on water before being subjected to the Sodium test program. The more important aspects of the water tests are discussed in the following paragraphs.

Test Set-up

The significant aspects of the physical set-up include; 50 ft. overall height from inlet to top of motor, 66 ft. lifting height to install pump into barrel, 90 tons lifting weight of pump. The overall weight of the pump, barrel, and motor to be structurally supported during test is approximately 142 tons. Alternate test arrangements would be investigated in detail for cost and adequacy.

Possible alternates are proposed as follows:

(a) Below Ground Test Pit - (see sketch T-1)

This arrangement would require an excavation approximately 40 ft. wide by 100 ft. long by 30 ft. deep. The walls would be lined with reinforced concrete and covered with a supporting deck. Internal cross beams would be suitable for bracing the side walls. Three floor levels would be constructed.

The principal advantage of this plan is reducing the above ground height required for lifting out the pump. Lifting facilities would extend upward approximately 35 ft.

(b) Structural Support - Above Ground (See Sketch T-2)

Essentially the same dimensions are required, however, the extensi-e excavation would be eliminated. Space is to be provided under the structure for unloading, erecting, assembling and handling the components. Floor levels, stairway, elevator, service and safety features would be included. At this time, selection of a preferred alternate of below ground or above ground is not considered feasible. Such items as zoning ordinances, environmental impact, foundation characteristics, and other siting qualifications indicate that the final selection should be deferred.

Electrical Power

The test with cold water will require approximately 11,000 HP. This value can be reduced by permitting the loop temperature to rise or by running the test at a slightly reduced speed. For example, if a 200° F temperature is selected, the maximum required power could be reduced by 3%. This important test parameter indicates that the test site be located near adequate power facilties.

The test substation at Byron Jackson is rated at 20,000 HP and is therefore of adequate size. Some modifications may be required depending on motor characteristics. 13,000 volts can be provided from this substation which is the most probable motor voltage. New switchgear would be required.

NPSH Testing

The configuration of the loop will necessarily be limited by HPSH test requirements. The suppression tank controls suction head conditions and therefore the water level in the tank should not be higher than 10 ft. above the upper pump impeller. The NPSH available to the pump will be set by pulling a vacuum or by permitting the temperature to rise. In either case, the minimum NPSH is limited by the static level difference.

The NPSH required by the pump is to be determined by setting a constant flow rate and then gradually reducing NPSH until the performance falls off. Several points of data are recorded and plotted so that any change in performance can be detected. This process is repeated at several capacities throughout the operating range of the pump.

Suppression Tank

A suppression tank is essential for two reasons; (a) stable control of NPSH setting and (b) re-absorbing of dissolved air. The recommended size is a tank volume in gallons approximately equal to the system flow rate in gallons per minute.

Test Piping

The piping would be designed to code requirements. The 36" discharge pipe would be fabricated from 1/2" plate. Elbow, tees, and fittings can be fabricated as segmented sections. These sizes and shapes are commonly constructed in this manner. The suction pipe may be of lighter weight. Material can be carbon steel with epoxy coated interior surface. A possible alternate material would be aluminum which would also eliminate carbon steel corrosion problems.

Flow Meter

A venturi meter will be used in the discharge line. Byron Jackson has extensive background data on 36" x 24" venturi design which would insure a high level of accuracy of measurement. See attached curve T-33150.

Throttle Valves

A 36" discharge line also permits the selection of a throttle valve that is readily available. A motor-operated butterfly valve rated at 250 psi would be adequate. Other type throttling valves may also be used such as Control Components "Drag" valves which eliminate valve cavitation at high velocities. It is also possible to design the valve flow control element in a 45° elbow and install the valve outlet directly on the tank to reduce cost.

Loop Cooling and Water Treatment

Power input to the loop water is 11,000 HP or approximately 28 million BTU/HP. Most of this heat must be removed by heat exchanger, cooling tower, or other suitable means. Evaporative cooling would require 100 gpm of makeup water. Direct cycling of test loop water to the cooling tower would cause a rapid build-up of minerals of most available tap waters and is not recommended. A heat exchanger should be provided as the interface between the test loop and the cooling tower. The secondary loop would then absorb the make-up water.

Sizing of the heat exchanger depends on the desired loop operating temperature. The expected design temperature would probably not be less than 150°F but should not exceed 200°F.

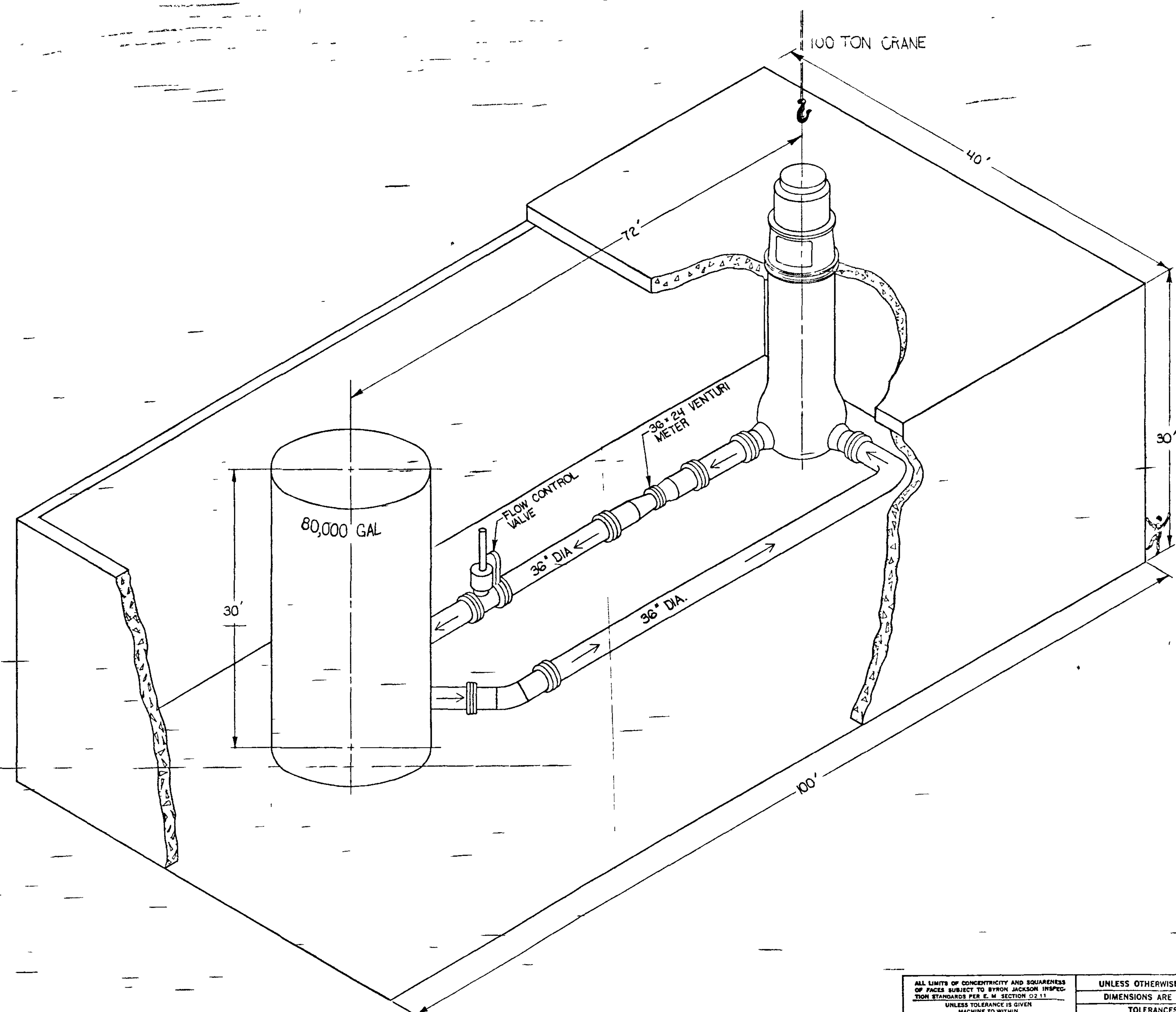
De-ionized water may or may not be desired. Generally, austenitic stainless steels are not damaged by typical tap waters at these temperatures, however, some level of treatment is recommended to reduce pump cleaning problems after test. Possibly a combination of partial demineralization plus addition of a suitable inhibitor would be adequate.

EQUIPMENT LIST

A tabulation of equipment required is as follows:

- Test Bay - Crane
- De-ionizer
- CCI Throttling Valve
- 180° Return - 36" Ø
- 36" Flanges (12)
- 36" Pipe 60 ft.
- 36" 90° El (1)
- Heat Exchanger (at least 2)
- Vacuum Pump
- Cooling Tower
- Cooling Tower Pump
- Power Source & Switchgear
- Instrumentation
- 36 x 24 Venturi (1)

REVISIONS					
LTR	REV BY	DESCRIPTION	DATE	CHK'D	APPROVED
Ø		INITIAL ISSUE			



ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION 02.11

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT. — 1/64
TURN — 1/64	2 TO 6 FT — 1/32
	6 FT AND OVER — 1/16

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UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS XX ± XXX ± XXXX ± ANGLES ±

✓ SURFACE ROUGHNESS PER M L STD 10

DRILLED HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005 / .015
ALL MACHINED CORNERS OR FILLETS .005 / .020

DISTRIBUTION	ARG	AUST	CAN	HOL	JAPAN	L A	MEX	TULSA	ORIGINATOR
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BJ ORDER NO 761-G-0002	REF NO
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CHECKED BY	DATE
DESIGN APP'D BY	DATE
TITLE	
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BORG WARNER® BYRON JACKSON®
BORG-WARNER CORPORATION

DRAWING TITLE TEST PIT-BELOW GROUND LARGE SCALE SODIUM PUMP-HOT LEG

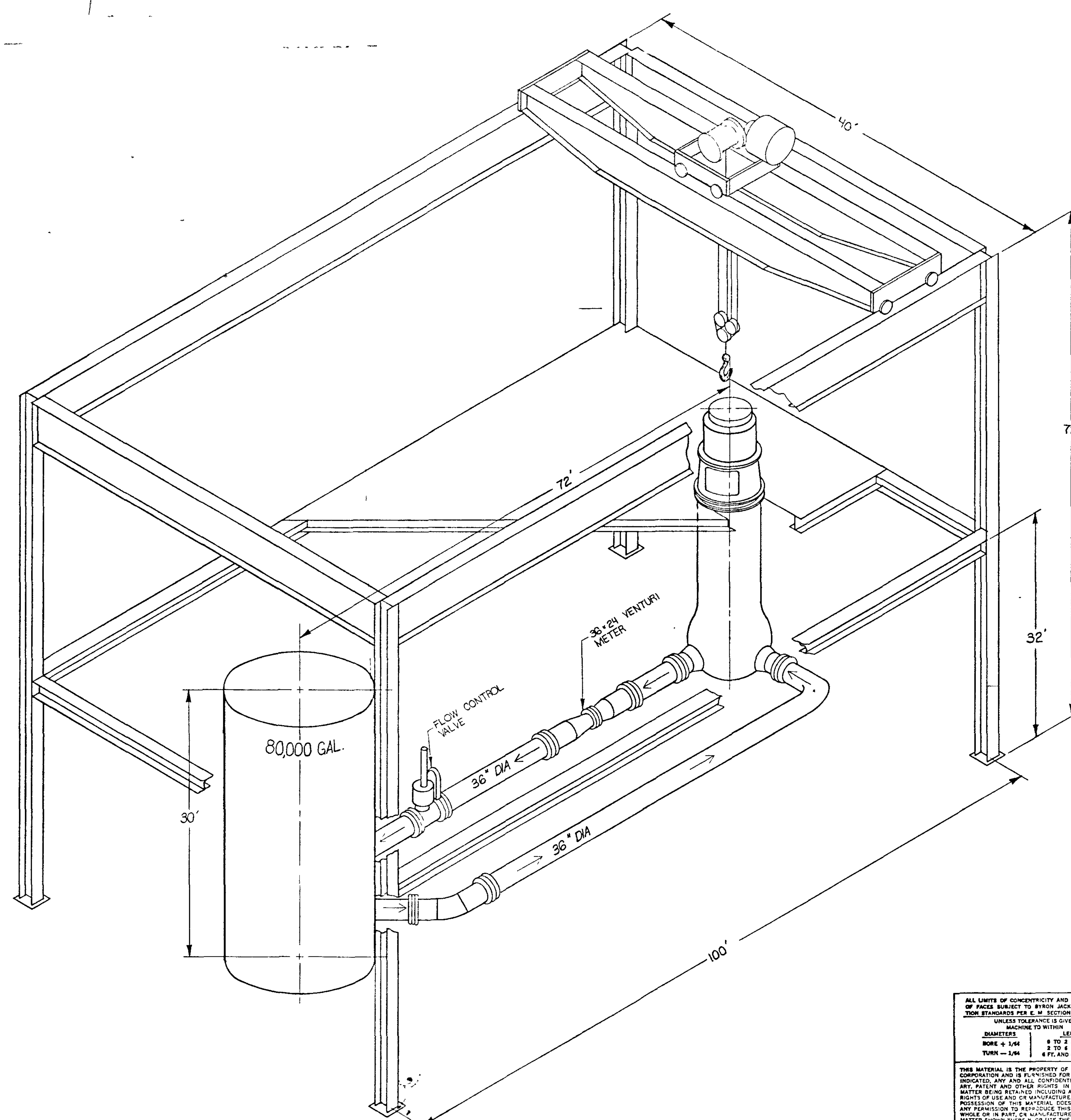
CONTRACT NO

CODE IDENT. NO.	DRAWING NO	REV
	T-1	Ø

SCALE 1/8" = 1'-0" | WEIGHT | SHEET

AEF

DIETERICH-POST CLEARPRINT



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LTR	REV BY	DESCRIPTION	DATE	CHK'D	APPROVED
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ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION 02.11

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT. — 1/64
TURN — 1/64	2 TO 6 FT. — 1/32
	6 FT. AND OVER — 1/16

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UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES

TOLERANCES ON

DECIMALS — XX ± — XXX ± — XXXX ±
ANGLES ±

✓ SURFACE ROUGHNESS PER MIL STD - 10

DRILLED HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005 / .015
ALL MACHINED CORNERS OR FILLETS .005 / .020

BJ ORDER NO 761-G 0002	REF NO
DRAWN BY S GLISSON	DATE SEPT. 76
CHECKED BY	DATE
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TITLE	
CUSTOMER APP'R	

BORG WARNER® BYRON JACKSON®
BORG WARNER CORPORATION

DRAWING TITLE
STRUCTURAL SUPPORT FOR TEST
ABOVE GROUND
LARGE SCALE SODIUM PUMP-HOT LEG

CONTRACT NO

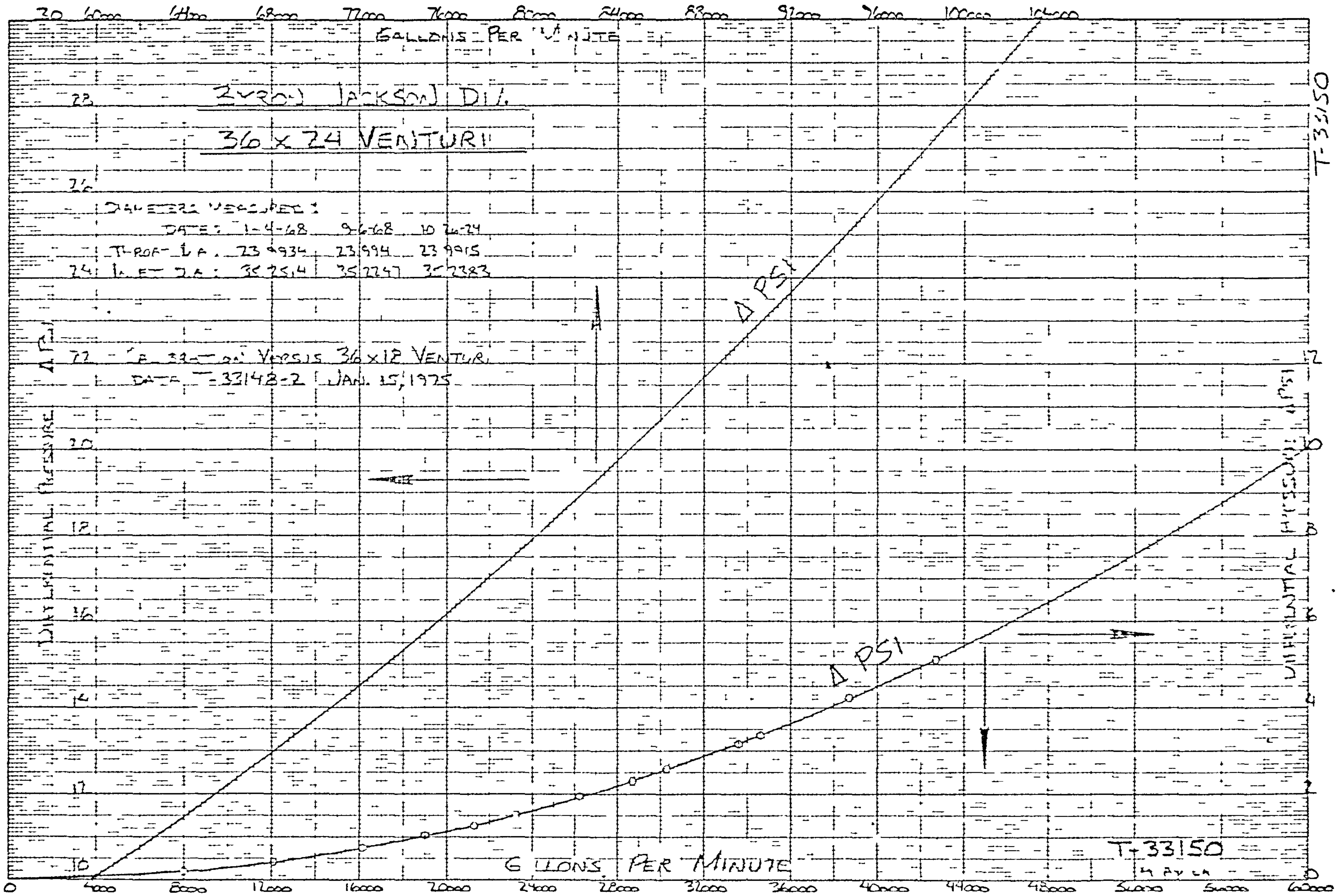
CODE IDENT. NO.	DRAWING NO.	REV.
	T-2	Ø

SCALE	WEIGHT	SHEET
1/8" = 1'-0"		

DISTRIBUTION	ARG	AUST	CAN	HOL	JAPAN	L.A.	MEX	TULSA	ORIGINATOR
DATE									

T.N.O. CLASSIFICATION

AEF



GALLONS PER MINUTE
36 x 24 VENTURI
DATE: 1-4-68 9-6-68
THROAT DIA. 23.9934 23.994 23.9915
DATA T-33148-2 JAN. 15, 1975
Δ PSI
Δ PSI
DIFFERENTIAL PRESSURE
T-33150

15. INTERFACING AND AUXILIARY REQUIREMENTS

15.1 Interfacing

A. Pump to plant interfaces

This is a list of pump to plant interfaces with some of the basic conditions which must be satisfied at each interface.

1. Motor to Pump

The motor mounting flange has to be compatible with the mating flange on the driver mount. The motor shaft extension has to satisfy the pump coupling requirements. The thrust bearing in the motor shall carry the axial thrust load required by the pump. Dimensions and tolerances pertaining to the motor to pump interface shall be agreed upon mutually.

2. Nozzles

The nozzles on the pump have to be made to suit field requirements. The size, location, and weld preparation details must satisfy the specification requirements.

3. Pump Mounting

This refers primarily to the interface between the pump mounting flange and the pump support in the field. The opening required for the pump and the location of the fasteners must be satisfied at this interface. There is also a requirement for a seal weld from the pump mounting flange to the field pump support. Provision must be made for a helium leak test of this closure after installation.

4. Instrumentation

All of the instrument terminal ends can be considered as pump to plant interfacing. The sodium level probe, oil liquid level probes, and temperature sensors are good examples. The type of instrument, location, and required connection will require interface control.

15.2 AUXILIARY REQUIREMENTS

15.2.1. Motor

A driver is required in this conceptual design study of a large scale breeder pump. Drive assembly capabilities and design requirements are of major important to the selection, design, and application of the pump and driver assembly.

The driver shall be an electrical motor with all the necessary auxiliary equipment and instrumentation. It is for a vertical application, with the pump and driver connected by a rigid, flanged coupling. The pump axial thrust is carried by the motor.

The design requirements are shown in outline form.

Rated pump power requirements will range from 170 to 11,000 brake horsepower.

The speed shall be 620 RPM with a 4 to 1 turn down ratio. The speed change shall be by control signals.

A pony motor drive is required. This shall be an auxiliary electric motor to drive the pump at approximately 9% speed or 54 RPM when the main motor drive is turned off.

The pump will require an axial downthrust of motor, 36,700 lbs. which must be carried by the motor thrust bearing.

The pump requires a single direction of rotation. There is no requirement for a device to preclude shaft rotation in a reverse direction.

There is no requirement for a flywheel in the motor.

Current design in commercial use shall be used where possible.

The motor located in the reactor containment building shall be of a minimum size and weight.

The motor shall give reliable operation with a minimum number of scheduled maintenance tasks.

The motor shall be tested separately and with the pump.

The interface between motor and pump will be resolved mutually. The shaft extension and method of centering, as well as manufacturing tolerances for mounting dimensions, requires a joint design effort.

MOTOR SIZE

The performance curve will show the brake horsepower required over the entire curve. The nominal size of the variable speed motor required at the design flow will be:

$$HP = \frac{Q \times H}{3960 \times Eff.} \times Sp.\ Gr.$$

where Eff. = pump efficiency

Sp.Gr. = specific gravity of sodium at the design temperature

$$HP = \frac{77{,}000 \times 500}{3960 \times 0.80} \times 0.83 = 10{,}087$$

For the design flow with sodium at 400° F, specific gravity of 0.90

$$HP = \frac{77{,}000 \times 500}{3960 \times 0.80} \times 0.90 = 10{,}938$$

Nominal motor size, 11,000 HP.

15.2.2 Recommended Instrumentation, Pump and Motor

Recommended instrumentation includes the measurement and/or indication of pressure, temperature, flow, position, speed, vibration and power. Only functional requirements are enumerated.

Pressure should be measured at appropriate locations in the sodium system and in all lube oil and cooling systems to monitor proper function and indicate potential failure.

Temperature should be measured at appropriate locations in all lube oil and cooling systems to monitor proper function and indicate potential failure. Temperature should also be measured in the following locations:

Pump shaft seal

Motor shaft radial bearings

Thrust bearing

Drive motor windings

Liquid rheostat

Flow should be measured at appropriate locations in all lube oil and cooling systems to monitor proper function and indicate potential failure. Because accurate measurements of sodium flow rate are difficult to obtain, flow measurement is not recommended as an indication of failure. Adequate indication of failure or potential failure can be obtained by other means.

Position should be measured for the following components:

(1) Liquid rheostat electrodes

(2) Liquid rheostat electrolyte level

(3) Pump barrel sodium level

(4) Pump shaft rotating assembly, at motor shaft

Speed should be measured on the pump rotating assembly.

Vibration should be measured on the pump rotating assembly and also in the following locations:

(1) Pump shaft seal

(2) Motor shaft radial bearings

(3) Thrust bearing

Power should be measured to the following components:

(1) Liquid rheostat

(2) Drive motor

(3) Pony motor

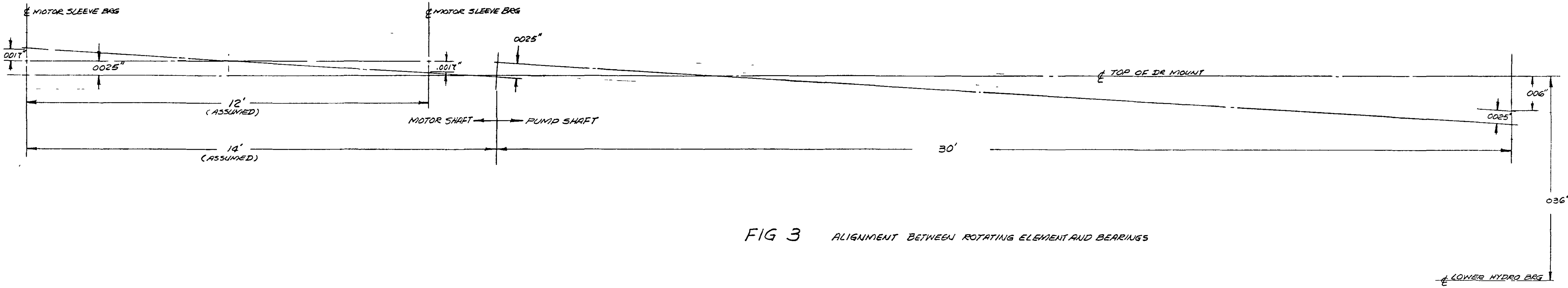


FIG 3 ALIGNMENT BETWEEN ROTATING ELEMENT AND BEARINGS

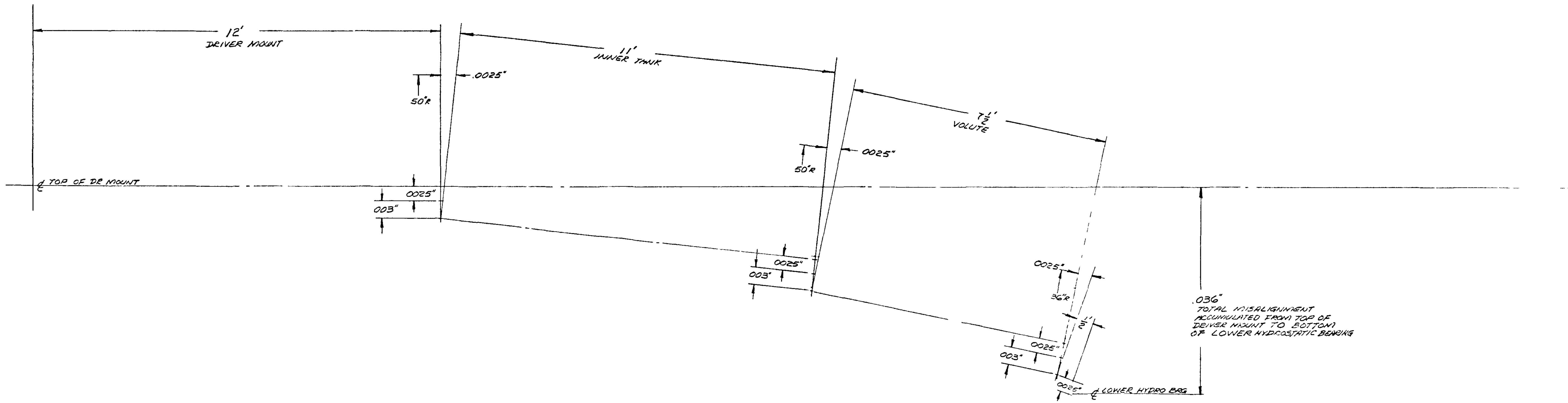


FIG 2 TOLERANCE STACK-UP

GRAPHICAL ALIGNMENT STUDY

16. MANUFACTURABILITY

This is a preliminary assessment of the manufacturability for the pump concept. It takes into account the effects of pump size, length, machining, alignment, casting capability, and assembly. Six of the areas which have been considered are described below.

1. Outer Tank Machining

The lower portion (sphere assembly) will be fabricated and finish machined similar to the Clinch River units.

The upper portion will then be attached to the sphere in two steps. A one step attachment is not feasible due to the 17 ft length - not compatible to the available equipment for nozzle weld preparation machining.

Due to the 10 ft diameter of the upper flange it is proposed to finish machine this portion utilizing special equipment, similar to the Patkay proposal for the large scale cold leg Breeder pump. Special line boring bar and star feed facing tools will be required.

2. Rotating Element

It is proposed that the rotating element be designed as a two piece assembly for ease of manufacture.

The one piece element concept must be fabricated by adding on one piece at a time followed with a machining operation for balance, true-up, and weld prep. This will require a very costly capital outlay for a 70" long bed lathe. Even so, the task would be time consuming.

Included in this proposal is a request for funding of the 70" used short bed lathe, for machining of the lower portion impeller section of a two piece element concept, similar to the large Breeder pump submitted by the Patkay group. The two sections would be joined together as two separately balanced assemblies. Runout checks would be made on roller stands for displacement, whereby corrections could be made at the joint if required. It is anticipated that correction requirements would entail altering one of the mating faces for perpendicularity to centering.

3. Assembly

The internals of the unit will be assembled in the same manner as the Clinch River pumps, that is, vertically in a hydraulically controlled pit. Present crane lift capacity and hook height clearance will not permit assembly of pump internals into tank at the Vernon facility.

A new pit as expansion of existing pit will have to be provided at an estimated cost of $200,000. Lift capacity must be at least 100 tons, preferably 150 tons. It must be noted here that this does not include assembly of internals into tank.

4. Equipment Requirement

Requirements are based on available equipment at this time. It is not assumed that Clinch River requirements or large Breeder Pump requirements will be provided and utilized where possible for the subject hot leg pumps.

5. Performance Test

Requirements for a prototype water test are covered elsewhere in this study. Also shown are the alternate test set-up requirements. Based on the Patkay proposal for the cold leg pump, and taking into consideration the difference in size, the complete cost for testing will be $1,140,000.

6. Upgrade Applicable Areas to Conform to RTD Specs.

Mercury Vapor lamp upgrading was not considered for the Clinch River project. Therefore, required funding for this project must be provided in the event of any Breeder program at an estimated cost of $25,000.

In summary, this conceptual pump size, length, and weight will require special consideration throughout the casting, fabrication, machining, assembling, and testing processes. The design as shown can be manufactured with a high degree of quality assurance utilizing equipment and technology available to industry today.

MACHINE TOOL REQUIREMENTS

PART	PART WT.	EXISTING EQUIPMENT	REQUIRED EQUIPMENT	COST
Outer Tank 14' swing x 11'	101,700	1. Schiess UBM 2. Mitsubishi HBM	1. 28" Line Boring & Facing Bar	$30,000
Inner Barrel 9' swing x 11-1/2'	20,600	1. Schiess UBM	-o-	-o-
Suction Pc 10' swing x 2'	26,000	1. Gray King UBM	-o-	-o-
Volute Assy 57" swing x 92"	47,000	1. 108" Bullard UBM	1. Grinder Adaptor to 108"	10,000
Rot. Assy 69" x 29'	17,700	1. 86" Bullard UBM	1. Balancing Equip. 2. 70" Eng. Lathe (used)	100,000 150,000
Reflector & Insulation Assy	26,000	1. 108" Bullard UBM	-o-	-0-
Driver Mount	40, 000	1. Gray King	-o-	-o-
Assembly			Assembly Pit 14' x 14' x 25' x 150 ton	$200,000

17. DEVELOPMENT NEEDS

17.1 Model Pump and Test

To obtain the desired information about the hydraulic performance of this pump with a minimum expenditure of time and money, it is recommended that a model of the pump be built and tested. To arrive at the head and capacity of the prototype from the model tests, affinity laws are used. Cavitation conditions of the model are best approached if the pump is tested at the same head and submergence as the prototype.

The physical size of the model could be about one-third the size of the prototype. The specific speed of the model must be the same as that of the prototype. The operating speed of the model must satisfy this condition, and should give the same head as the prototype.

Building and testing the model is part of the general development of this new type of pump, its behavior, and particularly its NPSH characteristic. The manufacturing and assembling of the model would be highly beneficial. It could be made for the use of additional instrumentation that would not be practical in the prototype.

The design could be improved in some areas as the patterns are being built for castings. Foundries could benefit from making the smaller model castings, using the experience thus gained for the prototype. The so-called "bugs" could be ironed out when the model is being built and assembled.

Models are built generally to obtain the desired information about the hydraulics performance of a given type of pump. In this case the bearings could also be modeled and their performance checked. Other information could be gained that would help in the performance test set-up and test of the prototype.

It would of course be ideal to use a variable speed motor. Since availability of such a driver is remote, at least one additional speed could be tested with a gear reducer.

PUMP MODEL FOR TEST

Model requirements

Model shall have same specific speed, N_s, as prototype.

Model shall have same head as prototype.

Model shall be one third, 1/3, the physical size of the prototype.

Model Speed

$$H_M = \left(\frac{1}{3}\right)^2 \times \left(\frac{N_M}{N_M}\right)^2$$

N_M = Model speed, N_P = Prototype speed

for a head of 500 Ft., $\frac{H_M}{H_P} = 1$

$$N_M = 3 \times N_P = 1{,}800 \text{ RPM}$$

Model Flow Rate (Capacity)

$$Q_M = \left(\frac{1}{3}\right)^3 \times \frac{1800}{600} \times 127{,}000 = 14{,}100 \text{ GPM}$$

Motor Size

$$HP = \frac{14{,}100 \times 500}{3960 \times 0.80} \times 1.0 = 2{,}220 \text{ HP}$$

Say about a 2500 HP vertical motor

Model Specific Speed

$$N_s = \frac{1800 \times 7050}{(500)^{3/4}} = 1470$$

This is the same as the prototype

17.2 Seal and Heat Exchanger for Seal

The heat generated by the seal must be carried away to give the desired long life. We have selected to circulate a light oil through the seal. The oil is then air cooled.

Determining the size of the fan, number and size of tube and fins, and other variables that will provide proper cooling for the oil being circulated will required further development. A full size test for the seals and heat exchanger is definitely recommended.

The seal cartridge must be removable without disturbing the motor. It is desirable to make the heat exchanger an integral part of the seal cartidge. This will require careful planning.

The seal should be removable as a cartridge - or complete assembly. This requires either a spacer or a stub shaft. Additional study could explore these alternatives.

The size of the shaft for this pump concept requires a new size seal. It is larger than the seal required for Clinch River.

This will require a development program including design, analysis and testing. The seals should be tested in a seal tester with its own variable speed driver. They could then be available when needed for the prototype pump test.

17.3 Stress Analysis of the Hydraulic Assembly

The assessment of potentially critical high stress areas indicates that several regions of the hydraulic assembly should be subjected to a detailed stress analsyis.

This analysis should investigate the overall thermal response of the hydraulic assembly in addition to determining stresses at critical locations.

It is thus recommended that a three dimensional shell model of the hydraulic assembly be constructed and analyzed. The cyclic symmetry that the structure exhibits about the vertical centerline will permit the analysis to be conducted on only a partial model and yet accurately predict overall thermal response.

17.4 Stress Analysis of the Tapered Seal Interface.

The effects of increased leakage and tank stresses due to the displacement of the inner support structure within the outer tank should be further evaluated using a three dimensional shell model of the pump in the region of the discharge penetration.

This study should determine the effective contact area between the two structures, in-plane shell stresses, and predicted clearance are under all design loads.

17.5 Pump Mounting

The method of supporting this pump must be designed jointly by the pump supplier and the user. The pump tank will include the necessary supports to match the field support structure. Carrying the dead weight of the pump and motor assembly is only part of the requirement. Thermal transients, shielding, and canopy-type seal welds must also be taken into consideration. The primary pump tank and the supporting structure shall be designed to permit a final seal weld which will provide a continuous seal with the vault liner.

The clamping ring securing the pump tank flange will have to be large enough to fit over the motor mounting flange.

The final seal weld between the pump tank flange and the field supporting structure must be helium leak tested.

All of the above indicates that further detail design is required in this area.

18. PROBLEM AREAS AND CONCERNS

18.1 Welding

This specification requires that pump parts which are internal to the containment boundary shall be designed to meet the intent of Section III of the ASME Code for Class 1 Components.

It is difficult to comply with this requirement in some areas. To obtain Class 1 welds, many of the places where plate could be used will require forgings and additional machining to obtain welds that can be radiographed.

Some of the internal attachments should not be designed to this requirement. We have taken exception particularly in the hydrostatic bearing and journal areas. This is all structural and in some areas not subject to pressure differential. The journal has been designed so that it can be removed by grinding out small welds. The hardfacing could more readily be applied to a sleeve that would then be welded to two rings on the shaft. The end rings for the bearing could also be hardfaced and then welded to the large casting containing the integral bearing.

This problem area can be solved, but complicates the design, perhaps unnecessarily.

18.2 Hydrotesting and Helium Testing

The work involved in non-destructuve testing would not be apparent in the study of this concept, therefore it is included in this section. Handling alone, after major parts are fabricated and machined, becomes a major undertaking.

Some of the larger parts will have to be designed with this requirement in mind.

Block off plates, fasteners, and seals will require additional design work. Some of the original design may be altered to make possible this non-destructive type test.

Some places, such as suction areas of the pump castings, flow areas of impellers, and others, should not be subject to these tests.

The necessity for acceptance tests is not questioned. Requirements can be met.

18.3 Performance Testing

The performance test for the prototype pump is discussed under component verification tests. This is perhaps a major portion of the complete pump requirements. The performance test itself can be accomplished with no anticipated problems.

The size of the pump and motor and the configuration of the pump, combine to make the test set-up and space requirement a possible problem area. At least it requires additional study. The area or space requirement available now may not be available when the pump will be tested. The first choice for the test site is of course at present facilities.

Electrical power and switch-gear to suit the motor will be required. Crane facilities to handle the pump sub-assemblies and the large, heavy motor will be needed at the test site. A large supply tank will be required, to give an accurate, reliable NPSH test. Much structural work will be required.

Because of the complexity of this test and its impact on time and cost, it is placed in this section of possible concern area.

18.4 Low NPSH available.

The NPSH available for this pump concept is 30 feet. This is below atmospheric pressure and makes it possible to get gas or vapor into the impeller. There could be cavitation and of course possible dmage to the bearings.

To avoid this possibility, a positive leakage is required from the discharge size of the pump. Some of this leakage could be returned to suction, but much of it would go to the overflow connection. This is a continuous loss at the pump design speed only, but represents a drop in pump efficiency.

The systems engineers should evaluate this low NPSH condition against the additional pump losses imposed on the pump. Perhaps some compromise could be reached.

BYRON JACKSON REPORT TCF-1058-STR

ASSESSMENT AND EVALUATION OF POTENTIAL HIGH STRESS AREAS IN LSBRP (HOT LEG) PUMP ASSEMBLY

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
SUNNYVALE, CALIFORNIA
CHANGE ORDER NO.1 TO LETTER CONTRACT 67002
BJ JOB NO. 761-G-0002

PREPARED BY Yoshio [signature] DATE 6 oct. 1976

CHECKED BY [signature] DATE 8 Oct 1976

APPROVED BY [signature] DATE 11 OCT 76

Sept. 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

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1.0 SCOPE

This report discusses a preliminary stress evaluation on a conceptual design of a primary coolant pump for a Large Scale Breeder Reactor. It is not intended to be a detailed analysis but only to identify the most critically stressed areas, determine stress levels for these areas, and to make recommendations to bring any unacceptable stress levels into conformance with specification requirements.

2.0 SUMMARY OF RESULTS

The most potentially critical area of the pump was determined to be the controlled clearance region located at the outer tank sphere-cylinder junction. This area was analyzed in detail as described later in this report. The results indicate that there are no insurmountable stress problems inherent in the design presented by Ref. 10.1. More specific results are discussed in the section of the report dealing with this critical area.

3.0 METHOD OF EVALUATION

The evaluation described in this report was performed in several specific steps which are enumerated generally below.

a) Loading condition definition

b) Review of Pump Structure for Function and Loading

c) Categorization of Regions of Pump Structure into potential high stress areas and detailed design considerations.

d) Assessment of criticality of potential high stressed areas.

e) Detailed Analysis of the most critical areas to determine if any insurmountable stress problems exist in the conceptual design.

f) Results and recommendations.

It should be noted that this report only evaluates the conceptual design. Regions for which the design can be modified with minimal effect on the basic design concept were left for evaluation in the detailed design phase of the project.

4.0 LOADING CONDITION DEFINITION

Loads acting on the conceptual design unit consist of thermal transients, pressure, driver torque, and externally generated nozzle loads.

4.1 Thermal Effects

The entire outer tank of this unit is considered to be insulated. Any thermal effects will therefore be due solely to thermal transients occurring in the liquid sodium. These thermal stresses can be categorized into two basic categories, thermal gradient and thermal discontinuity stresses. The magnitude of thermal gradient stress depends strictly on the thermal gradient through the section which is in turn dependent on the exposure to transients and on heat sink effects. Thus, areas where the transients are mitigated and areas where both surfaces are exposed to the transients are considered to be areas of low stress due to thermal gradients.

Thermal discontinuity stresses are those caused by differences in the average temperature through the cross section between adjacent areas. The thermal gradients causing these stresses are therefore not gradients through the wall but, rather, along the wall. They are due primarily to differences in cross section which cause thermal discontinuities or to heat sink effects of thicker cross sections. Such sections are considered to be areas of high stress and are even more highly stressed when thermal gradient stresses maximize in the same area.

4.2 Pressure Effects

The unit considered in this report is subject to three basic pressures, atmospheric, suction, and discharge. The cover gas pressure is only slightly above atmospheric and is thus considered to be the same as atmospheric. Only inbalances between these pressures are considered to produce significant pressure stresses.

4.3 Driver Torque

The driver torque is developed within the motor and transmitted through the shafting to the impellers. Thus this loading affects only those portions of the unit which are directly involved with the shafting.

4.4 Nozzle Loads

The nozzle loads are externally generated and are applied to the suction and discharge nozzles by the external piping. These are categorized as deadweight loads, thermal expansion loads and OBE seismic loads. These loads are tabulated in Reference 10.2.

5.0 REVIEW OF PUMP STRUCTURE FOR FUNCTION AND LOADING

Various subregions of the unit were next evaluated structurally and possible loading sources were enumerated. This was done on a piece by piece basis except as otherwise noted.

5.1 Outer Tank

The first major component of the unit as shown in Reference 10.1 to be considered was the outer tank which consists of a cylinder joined to a spherical head at the lower end. The pump support flange is located at the top of the cylindrical section. The tank wall is penetrated by an overflow connection located ten feet below the lower face of the lower face of the support flange, and a suction and a discharge nozzle located on the horizontal centerline of the sphere at ninety degrees to one another. In the area of the nozzles and the cylinder-sphere junction, there is a tapered section that is thicker than the remainder of the tank. The purpose of this tapered section is to provide a controlled clearance through which discharge pressure in the sphere is broken down to suction pressure in the cylindrical portion of the tank and in the suction nozzle. This controlled clearance holds the leakage between these regions of differing pressures within acceptable limits.

The purpose of the outer tank is to contain the liquid sodium used as pumpage. All the sodium within the cylindrical portion of the tank is at suction pressure and that in the spherical portion is at discharge pressure except for the suction nozzle itself.

The outer tank is subjected to pressure loading, thermal transient loading and to nozzle loads. It is not subjected to any loading whatever as a result of motor torque. Pressure loads above the overflow level are due solely to the pressure of the cover gas which is extremely small. Below the overflow level, with the exception of the discharge nozzle and sphere, the only pressure loading is due to the liquid level within the tank itself. The design pressure for the entire tank is 30 psi as specified by Reference 10.2.

Thermal loading will be primarily a thermal gradient through the thickness due to the essentially uniform tank wall. Thermal transients will have a greater effect below the sodium surface, due to the contact of the sodium with the outer tank. Above the sodium there will be a longitudinal thermal gradient extending to the pump support flange.

Finally, nozzle loads act at both the suction and discharge nozzles. The main load path is through the wall of the outer tank between the two nozzles. Any unbalanced nozzle loads will be transmitted through the upper portion of the tank, through the pump support flange to the anchor bolts and the foundation.

5.2 Motor Mount and Radiation Shield

The next major piece of the unit is the motor mount and radiation shield. This is bolted to the pump support flange and is not subjected to any of the four defined loadings. This piece is meant to support the pump drive motor and to provide a protective radiation shield where the pump penetrates the permanent radiation shield of the plant. It also supports the labyrinth shaft seal.

5.3 Inner Structure Support

The inner structure support consists essentially of a cylinder flanged at both ends. Its purpose is to provide support for the hydraulic assembly and the thermal insulation and reflector assembly.

The inner structure support is bolted to the lower surface of the radiation shield. The lower end of this structure is below the sodium level. It is subjected only to pressure and thermal loads. Both internal and external pressures are suction pressure and hence this loading is essentially zero. The thermal loading is minimal due to the uniform cross section, the lack of discontinuities, and mitigation of thermal transients due to mixing with the static sodium volumes except at the lower flange, the outside diameter of which is part of the controlled clearance boundary.

5.4 Hydraulic Assembly

Suspended from the inner structure support is the hydraulic assembly. This is the heart of the pump and consists of a volute piece and two suction pieces. This assembly encloses the double suction impeller, and includes two hydrostatic bearings. Except for the inlet passages, the hydraulic assembly is balanced insofar as pressure loadings are concerned and thermal loadings act on most surface walls of each piece simultaneously insofar as it is possible.

5.5 Motor

The drive motor is mounted on top of the motor support stand. It is furnished and designed by others and is not included in this study. It is mentioned here only in the interest of completeness. The motor is an electric motor with a vertical shaft. It provides the mechanical energy that drives the pump and absorbs any axial thrust generated in the pump. It is connected to the pump shaft by a rigid coupling.

5.6 Pump Shaft

The pump shaft is the means by which the mechanical energy developed in the motor is transmitted between the motor and the impeller in the hydraulic assembly. Due to the length and manufacturing limitations, the shaft is constructed of several individual sections welded together to form an integral shaft.

In essence, most of the pump shaft is a hollow tube. Only the upper end of the pump shaft is solid rather than hollow to provide radiation shielding.

The pump shaft is subject to motor torque loadings, to thermal loading and to pressure loading (suction pressure over the entire inside surface, and discharge pressure over part of the outside surface with suction pressure over the remainder).

Within the hydraulic assembly, the shaft is comprised of the impeller, a lower hydrostatic bearing journal, an upper hydrostatic bearing journal and whatever intermediate sections are required to maintain proper spacing between them. This section of shafting is loaded in the same manner as the other portions of the pump shaft. The two hydrostatic bearing journals are discussed in the next section of this report.

5.7 Hydrostatic Bearings

There are two hydrostatic bearings within this pump which have the purpose of maintaining alignment and carrying radial loads. Both hydrostatic bearings are supported by the hydraulic assembly. The journal of each hydrostatic bearing is mounted on a hollow portion of the pump shaft. Thus the journal is larger than the pump shaft with an annular space between the journal and the outer periphery of the shaft. In each case this annular area is connected to the interior of the pump shaft by perforations in the shaft wall.

The external surface of the journal is subjected to a pressure somewhere between suction and discharge pressure due to the pressurization of the various pockets of the bearing from a discharge pressure source through orifices. Sodium flow is therefore from the discharge pressure source through the orifices to the journal surface and then along the surface to an area of suction pressure. The bearings are therefore subjected to thermal transient loading, pressure loading and motor torque loading.

6.0 CATEGORIZATION OF REGIONS OF THE PUMP STRUCTURE

Each of the various regions discussed in the previous section were then classified into one of two categories:

a) Potentially high stressed area

b) Detailed design consideration

If the particular area being considered is categorized as being for Detailed Design consideration, it indicates that modifications can be made during the detailed design phase of the project that will permit the requirements of Reference 10.2 to be met without altering the basic design concept depicted by Reference 10.1. In general, welds and bolted flanges immersed in liquid sodium can possibly result in thermal problems. The placement of welds and the sizing and bolting arrangements of flanges do not affect the basic design concept and hence are deemed items for consideration during the detailed design phase.

The subsequent categorization does not evaluate strain and deformation limits since these are only significant in regions of high primary or secondary stress. The effect of combined creep and fatigue damage and of creep ratcheting are therefore, evaluated only in the detailed analyses of the potentially high stressed areas.

6.1 Outer Tank

The outer tank is a structure of relatively uniform thickness. In general, the thickness of the cylindrical and spherical portions of the tank can be altered to accommodate thermal, pressure and nozzle loads without any effect on the design concept. Hence the basic tank is deemed to involve detailed design consideration only. Specific areas, which are part of the outer tank structure, where this general categorization may not apply are individually discussed below.

6.1.1 Pump Support Flange

The pump support flange together with the anchor bolting and foundation is the reaction point for any unbalanced nozzle loads. It is also the sole support of the entire unit including driver. As such, it is likely to be a highly stressed area. On the other hand, any modification in the design of this flange has no effect whatever on the design concept and hence this is designated for detailed design consideration only.

6.1.2 Overflow Connection

This connection is subject to thermal loading, a minimum of pressure loading and unbalanced nozzle loads. There are discontinuities in this area which tend to increase the stress level. However, these are relatively small and tend to keep such increases to a minimum. The actual design of this connection can be modified as necessary to meet any requirements without altering the conceptual design of the unit.

This connection is therefore designated for detailed design consideration only.

6.1.3 Cylinder-Sphere Junction

The reinforcement at the junction of the cylinder and sphere is an area of gross structural discontinuity subject to both pressure and thermal loads. Due to the controlled clearance, it is categorized as an area of potentially high stress and discussed further in Section 7.3.

6.1.4 Discharge Nozzle

This nozzle involves thermal, pressure, and nozzle loads. The structural discontinuities are among the largest in the unit. This is therefore an area of potentially high stress and will be further discussed in Section 7.1 of this report.

6.1.5 Suction Nozzle

The same reasoning that was applied to the discharge nozzle applies to this nozzle. It is therefore similarly categorized for further discussion in Section 7.2 despite the fact that the structural discontinuities here are less severe than for the discharge nozzle.

6.2 Motor Mount and Radiation Shield

In general, this component is not subjected to any external load. The specific design can vary greatly without affecting the conceptual design and the entire component, including the labyrinth seal it supports, is designated for Detailed Design consideration only.

6.3 Inner Structure Support

The inner structure support supports the thermal insulation and reflector assembly and is essentially free from structural discontinuities. Since the design of the inner structure support as well as the thermal insulation and reflector assembly can be greatly modified without affecting the basic design concept, these items are considered to be in the Detailed Design Category.

The lower flange of the inner structure support is bolted to the hydraulic assembly. Both of these flanges introduce a structural and thermal discontinuity which can result in deformation and high stress levels. In addition, the periphery of these flanges, form one boundary of the controlled clearance region of the pump. The inner support lower flange is therefore categorized as an area of potentially high stress and is discussed further in Section 7.3

6.4 Hydraulic Assembly

Due to the thermal and pressure load balance inherent in the present design, the hydraulic assembly, excluding the hydrostatic bearings, is deemed, generally, to involve detail design considerations only. It is apparent that there are a large number of discontinuities in this assembly but the transient effects are mitigated by the inherent thermal balance and by the symmetry of the component.

Within the hydraulic assembly, there are several areas for concern which are discussed individually in the following paragraphs.

6.4.1 Lower Suction Piece

The lower suction piece and volute piece are joined with a bolted flanged joint. There are a number of relatively large sections in this area with resulting high thermal gradient and discontinuity stresses. While these stresses are minimized by the mitigation of the transient due to mixing in the spherical tank area and by exposure of all possible surfaces to the transient effects, the stresses may still be high enough to cause problems. The geometry of the flange and bolting has already been categorized as a detail design consideration but it is an area of concern.

6.4.2 Upper Suction Piece

The upper suction piece is joined to the volute with a bolted joint similar to that discussed in Section 6.4.1. As a result, the same considerations are applicable as were pointed out there.

6.4.3 Clearances

The hydraulic assembly, by virtue of the fact it is basically a centrifugal pump, must have relatively close running clearances. Accordingly, overall deformation must be evaluated whether it be due to creep effects because of the relatively high operating temperatures or due to deflection from other loads. This overall deformation can be considered cause for concern.

6.5 Motor

The motor is not included in this study as was stated in Section 5.5. The connection between the motor shaft and pump shaft is a rigid coupling which does not affect the design concept and can therefore be relegated to the detailed design phase of the project.

6.6 Pump Shaft

The pump shaft, other than in the areas where the hydrostatic bearings are located is of uniform cross section and is symmetrical. It can be readily modified to meet the requirements of Reference 10.2 and is therefore deemed to involve detailed design considerations only.

6.7 Hydrostatic Bearings

In view of the structural discontinuities and close running clearances present in each of the bearing areas, these areas are considered to be potentially high stressed areas and are reviewed further in Section 7.4.

6.8 Summary of Potential High Stress Areas

The preceding portions of this section have reviewed all portions of the conceptual design as depicted by Reference 10.1. Most of the areas reviewed were categorized as detailed design phase considerations. The remaining six regions were categorized as potential high stress areas which would require additional review and analysis. These six areas are summarized in the following list and are discussed further in Section 7.0

1. Discharge Nozzle Area
2. Suction Nozzle Area
3. Cylinder to Sphere Junction
4. Hydraulic Assembly
5. Lower Hydrostatic Bearing
6. Upper Hydrostatic Bearing

7.0 ASSESSMENT OF CRITICALITY OF POTENTIAL HIGH STRESS AREAS

The six potentially high stressed areas determined in the preceding section of this report were then reviewed for degree of criticality. That review is described in more detail in the remaining portions of this section.

7.1 Discharge Nozzle

Section 6.1.4 pointed out that this nozzle is subjected to pressure loads, nozzle loads and thermal loads. Therefore both primary and secondary stresses exist in this nozzle. Pressure loads are among the highest in the pump since they are due to a discharge pressure of 200 psi. Nozzle loads are superimposed on the pressure loading thus increasing the primary stress levels. Thermal stresses are also expected to be high since the nozzle is directly affected by the thermal transients and because of the structural discontinuities. This area will therefore be subject to the effects of creep ratcheting.

Inasmuch as the shape of the discharge nozzle has not yet undergone any final design iterations, no detailed analysis is attempted. It is noted, however, that high stresses are anticipated in this region and further development work is therefore recommended.

7.2 Suction Nozzle

Geometrically the suction nozzle shown in Reference 12.1 is similar to the discharge nozzle for the Clinch River pumps as analyzed in Reference 10.3. However, the loadings are somewhat different. The pressure inside the conical section and the nozzle itself were at discharge pressure levels in the Clinch River units whereas the conceptual design nozzle is subjected to suction pressure. Across the face of the seal area, the pressure gradient is in the reverse direction from that considered by Reference 10.3. Finally, the area between the outer surface of the conical section and the inner surface of the sphere is under discharge pressure in the conceptual design while for Clinch River, this was a suction pressure area.

The nozzle and inner surface of the conical section are affected by the full thermal transients while the external surface of the conical section and inner surface of the sphere are affected by mitigated transients becuase of the low velocity flows in this area.

The temperature change due to transients applied to the nozzle will be more rapid in the area around the Clinch River unit nozzle than in the nozzle under consideration because the source of flow into this region is the nozzle itself whereas in the unit being considered, the source of flow is the other nozzle. Therefore any temperature change must be propagated further before any effect can be noticed. Accordingly, the thermal stresses in the spherical portion of the nozzle are much less in this nozzle than they are in the Clinch River unit. This fact and the lower temperature of the unit being considered here reduce creep effects significantly in this area.

The effect of nozzle loads will pass through the spherical portion of the outer tank rather than through the conical portion of the nozzle. This is true for both the CRBRP and conceptual design nozzles.

Reference 10.3 indicates that for primary loading, the transition region of the nozzle (from nozzle dimensions to pipe dimensions) is the most critical area. These stresses are due to pressure and nozzle loads. The lower pressure level in the unit being considered results in lower stress levels due to pressure. Superimposing nozzle loads of approximately the same magnitude results in overall primary stress levels below those of the Clinch River unit. Since the thermal transients to be applied are smaller than those applied on the Clinch River unit, the secondary stresses will also be smaller. Thus the most critical area of the Clinch River nozzle is more severely loaded than the nozzle being considered.

On the other hand, due to the slower temperature change around the outer surface of the conical section, thermal stresses in the conical portion will be higher in the nozzle being considered than those encountered in the Clinch River nozzle. These thermal stresses must also be considered along with the pressure loads.

It was pointed out earlier in this section that nozzle loads have no effect on the conical section of the nozzle.

The Clinch River nozzle pressure tends to expand the entire conical section. In addition, the pressure gradient across the seal area face is higher at the inner edge of the seal area. This gradient load combines with the expanding tendency at the narrow end of the conical section in the same direction. In the nozzle depicted by Reference 10.1, the nozzle pressure tends to contract the conical section and the pressure gradient load, being higher at the outer edge of the seal area has a greater effect than the gradient on the Clinch River unit. Accordingly, at the narrow end of the conical section the effects of these two loads tend to oppose each other and reduce the net pressure load.

At the wide end of the conical section, the pressure effects in the nozzle under consideration are additive and hence are higher. When combined with the increased thermal effects already defined, this may become an area where the stress level reaches the critical level. This is then an area which requires additional investigation.

7.3 Cylinder to Sphere Junction

The junction between the cylinder and sphere in the outer tank is another area of potentially high stress. It is a portion of the controlled clearance area. This is a region where the fluid flow through the annulus between the inner support flange and lower outer tank causes the inner structure to deflect relative to the outer tank. The magnitude of the reaction in the outer tank governs the primary stress level in this particular region. Any increase in the primary stress level results in a corresponding decrease in the allowable secondary stress level. Since this area is subject to thermal shock due to the transients, the secondary stress level which is permitted is an important consideration. Relative deflections in this region also cause an increase in the gap between the inner structure and outer tank with resulting increased leakage which is detrimental to pump hydraulic performance. This junction is thus considered a potentially critical region.

7.4 Hydrostatic Bearings

There are two hydrostatic bearings in the conceptual design unit. Each is subjected to pressure, thermal transient, and motor torque loadings. From the standpoint of the motor torque loading, the full motor torque, less shafting losses, must be transmitted through the upper bearing. Most of this torque loading is used by the impellers to impart velocity to the sodium so that the torque loading on the lower bearing is minimal. Pressure loadings on both bearings are the same. Stress changes due to thermal transients are more severe on the upper bearing than on the lower bearing so that the upper bearing is more critical than the lower bearing. These bearings are similar in design to those analyzed in Reference 10.4. The results of analyses indicated no critical areas in the bearings but the effects of minor differences in configuration should be investigated.

7.5 Hydraulic Assembly

The hydraulic assembly is basically categorized as a detailed design consideration. It contains two bolted joints which have also been categorized as detailed design considerations. The two suction pieces provide the support for the hydrostatic bearings. Because the sodium volume maintained at suction pressure below the lower suction piece is much smaller than the sodium volume at suction pressure above the upper suction piece, the sodium temperature will change more rapidly below the lower suction piece as a result of a thermal transient than the temperature above the upper suction piece. The upper suction piece will therefore be subjected to higher thermal gradients than the lower suction piece. On the other hand, the bolted joint at the lower suction piece involves much more mass and the heat sink effects at this joint will tend to cause higher stresses than in the upper bolted joint.

Increased leakage due to thermal distortions along the sealing surface will also be encountered in the hydraulic assembly.

A large number of discontinuities are inherent in the design of any such hydraulic assembly and thermal effects can only be minimized. All of these items tend to place the hydraulic assembly high on the list of areas to be analyzed in the detailed design phase of this project.

7.6 Relative Degrees of Criticality

Based on the above discussions, it is felt that for the six regions of potential high stress, the following is the order of criticality with the first on the list being most critical.

1. Discharge Nozzle Area
2. Cylinder to Sphere Junction
3. Hydraulic Assembly
4. Suction Nozzle Area
5. Upper Hydrostatic Bearing
6. Lower Hydrostatic Bearing

The most critical area of this conceptual design unit is the discharge nozzle area for which further optimization is required before any significant analysis is performed.

The second most critical area, the cylinder to sphere junction, was analyzed and further discussion on this analysis is presented in Section 8.0 of this report.

The third most critical area, the hydraulic assembly, is geometrically similar to the Clinch River hydraulic assembly. That unit is currently undergoing analysis and the results of that analysis will be applied to this unit. This makes further analysis of the hydraulic assembly for this conceptual design superfluous at the present time.

Recommendations concerning the other three areas listed above are included in Section 9.0 of this report.

8.0 CYLINDER TO SPHERE JUNCTION

The cylinder to sphere junction was determined to be the most critical area in this design concept. Accordingly, this area was subjected to a detailed analysis which was performed by Basic Technology, Inc. The final report on this analysis is included as Addendum A to this report.

Addendum A indicates that no part of the cylinder to sphere junction can be considered a potential problem area due either to creep ratcheting or creep fatigue.

8.1 General Description of Analysis Performed

In general, the analysis presented in Addendum A was performed using the ANSYS computer program. It involved both heat transfer analysis and stress analysis prior to evaluation.

8.1.1 Heat Transfer Analysis

The PP-3UT and PP-2ET transients were analyzed to provide a time history for each. These were examined and the times of peak gradients used to establish four thermal maps for later use in the stress analysis.

8.1.2 Stress Analysis

The same model was used for both the heat transfer and stress analyses. For the stress analysis, thermal loadings as described by the four thermal maps were imposed as well as deadweight and pressure loads.

8.1.3 Code Evaluation

The stress results were superimposed as necessary for code evaluation. To simplify data reduction, only primary stresses, creep ratcheting and creep fatigue strain were evaluated. Comparison of these values with the code specified limits indicated a completely satisfactory design.

9.0 CONCLUSIONS AND RECOMMENDATIONS

The preceding sections of this report identified and ranked six critical areas of the pump according to the degree of criticality. One of the three most critical areas was analyzed and found to be satisfactory. Another of these three most critical designs was found to be in the early stages of optimization and analysis was deferred until completion of one or more further design iterations. The third most critical area is similar geometrically to the Clinch River unit and this area of that unit is currently being analyzed. The results of that analysis are expected to be applicable to this conceptual design.

Two of the remaining three items are similar in design to two that were analyzed in Reference 10.4. The results of those analyses indicated no identifiable problems but recommended further analysis after the design is finalized.

The final item which was identified as being an area of potentially high stress is the suction nozzle. This nozzle is similar to the discharge nozzle for Clinch River which has been analyzed. The differences in loading conditions necessitate further analysis and such analysis is recommended. In addition, it is recommended that particular attention be paid to the stress levels in the conical portion of this nozzle.

Assessing these critical areas leads to the conclusion that no insurmountable stress problems are included in this conceptual design. However, it will be necessary to apply the results of work on the Clinch River unit to this conceptual design during the detailed design phase of this project.

10.0 REFERENCES

10.1 Byron Jackson Conceptual Design Drawing (3 sheets) 1F-8084 Rev. 0 dated 18 Aug. 1976.

10.2 General Electric Specification 22A 3994 Rev. 8 dated 18 June 1976.

10.3 Byron Jackson Report TCF-1053-STR "Screening Analysis of Pump Tank Nozzle Using 2-D Axisymmetric Finite Element Model W.P. 7321" dated 12 January 1976.

10.4 Addenda B & C to Byron Jackson Report TCF-1057-STR "Assessment and Evaluation of Potential High Stress Areas in LSBRP Pump Assembly" June 1976.

ADDENDUM A

LARGE SCALE LIQUID SODIUM PUMP
CYLINDER-TO-SPHERE JUNCTION ANALYSIS

FOR

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BY

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October 8, 1976

BTI77401

1.0 SCOPE OF WORK

This report presents a thermal-hydraulic and a structural analysis of the cylinder-to-sphere junction in the casing of a proposed large-scale sodium pump for a commercial LMFBR.

The junction to be analyzed is shown in Byron Jackson's layout drawing No. IF-8084 and No. IF-8085 (see Figs. 1.1 and 1.2). It connects the upper and lower tanks of a large-scale sodium pump which is to be placed in the hot leg of a LMFBR primary loop. Except for the suction and discharge nozzles, the lower pump tank is part of a perfect sphere. Reinforcement in the junction to the upper cylinder provides structural support to the pump casing under primary mechanical loads and thermal transient conditions.

The analysis includes, first of all, a detailed thermal-hydraulic finite-element analysis of the region around the cylinder-to-sphere junction, including the fluid flow as well as heat connection and conduction through the walls adjacent to the junction. Secondly, the results of the thermal-hydraulic analysis are used to provide the thermal boundary conditions for a two-dimensional axisymmetric finite-element model of the junction. Thirdly, the temperatures obtained from the thermal analysis are used to calculate the thermal stresses in the junction with an equivalent two-dimensional elastic finite-element model.

The primary loads on the junction includes internal pressure, deadweight and unbalanced pressure loads. The clearance of the gap between the junction and the hydraulic assembly is also investigated for thermal growth restraint. The primary and secondary stresses are evaluated for possible fatigue, creep and/or ratchetting damages, in accordance with ASME Boiler and Pressure Vessel Code Section III and the Elevated Temperature Code Case 1592.

The design and operating conditions are given by General Electric Company's Specification No. 23A2126, which contains the requirements for the large scale breeder reactor plant hot leg mechanical pump conceptual design.

Temperature dependent material properties for SS316 are used throughout the analysis.

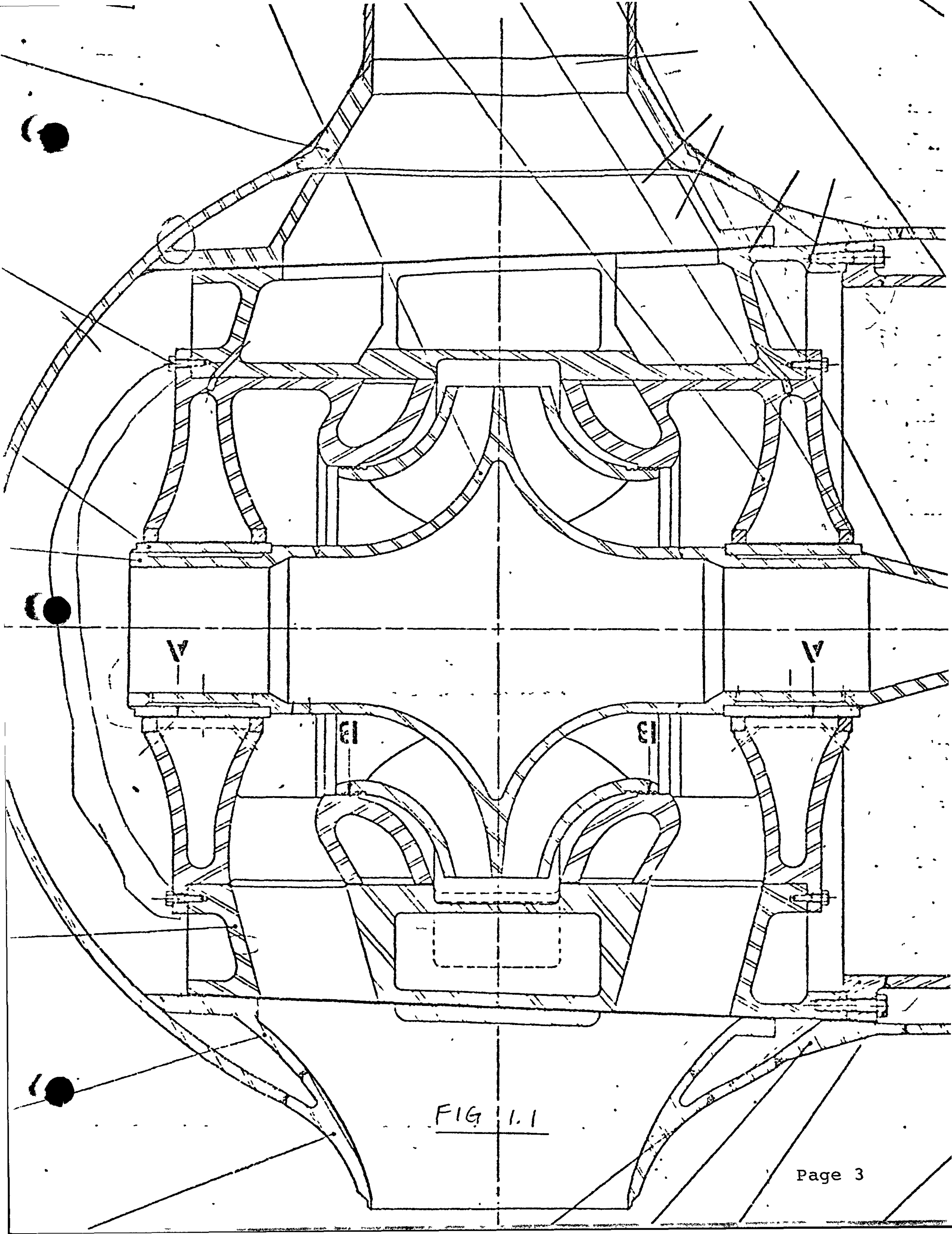
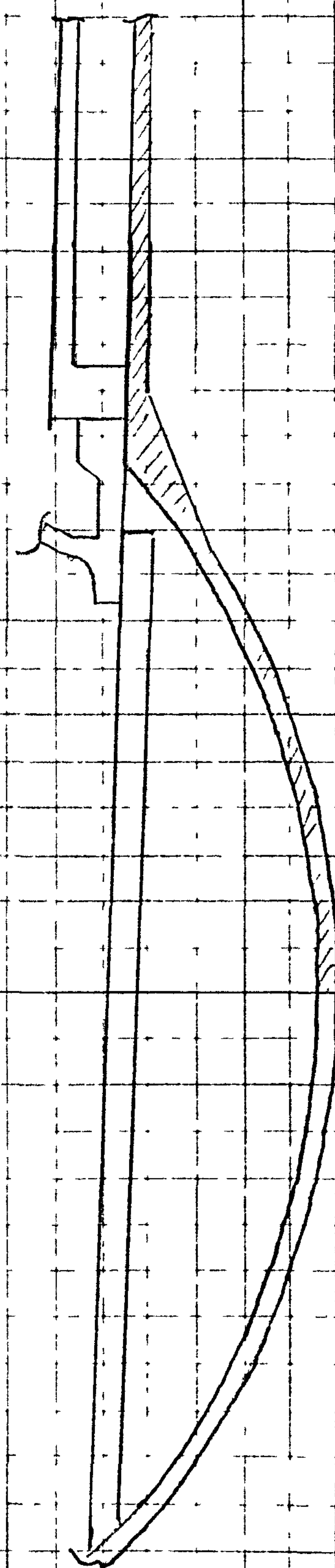


FIG 1.1

BASIC TECHNOLOGY, INC.
806 Manhattan Beach Boulevard
Manhattan Beach, California 90266

BY ______ DATE ______ SUBJECT ______ SHEET NO. 4 OF ______
CHKD. BY ______ DATE ______ JOB NO. ______

FIG. 1.2



2.0 THERMAL-HYDRAULIC ANALYSIS

An overall thermal-hydraulic model for the region of interest includes the cylinder-to-sphere junction and portion of the hydraulic assembly, as well as the nearby transient and bulk liquid sodium. This analysis is necessary for determination of the thermal transient boundary conditions at the junction, which are expected to be mitigated from the full transient in the main flow, due to mixing with leakage flows and to heat conduction through the walls. A schematic sketch of the thermal-hydraulic model is shown in Fig. 2.1. The lower annular region (region 1) represents the volume between the sealing ring and the spherical pump tank. The flow into this region (W_1) occurs at the gap between the discharge nozzle and the sealing ring discharge cone. The flow out of this region is partially back into the inlet nozzle and partially into the upper annular region (region 2) between the cylindrical pump casing and the hydraulic assembly. The flow rates into the inlet nozzle and the upper annular region are denoted by W_3 and W_2, respectively, in Fig. 2.1, where

$$W_1 = W_2 + W_3$$

The flow leaving region 2 exits with the same rate W_2 through the overflow pipe. Complete mixing is assumed in both the lower and upper annular regions.

The additional heat flows Q_1 and Q_2 into the lower and upper annular regions are primarily due to conduction through various walls of the hydraulic assembly, with some convection over the wall surfaces.

A more detailed layout of the thermal-hydraulic model with the applicable parameters, is shown in Fig. 2.2. Two sets of full transients (T_F) are included in the finite-element calculations. These are the PP2E and PP3U transients, selected because of their severity of transient rates. The temperature responses in both region 1 (T_1) and region 2 (T_2) are calculated as a function of time.

The ANSYS computer program is used for the thermal-hydraulic analysis. The volumes of fluid in regions 1 and 2 are represented by isoparametric temperature elements (STIF 55), with the temperatures coupled at the four nodes. Conduction through the walls is represented by three conduction links (STIF 32) for each wall. Connection links (STIF 34) are also used between the fluids and the adjacent wall surfaces. In addition, the thermal fluid flow elements (STIF 56) are used to represent the sodium flows.

The results of the thermal-hydraulic calculations for the PP2E and PP3U transients are shown in Tables 2.1 and 2.2, and in Figs. 2.3 and 2.4. The temperatures $T_1(t)$ and $T_2(t)$ are used as thermal boundary conditions in the next section.

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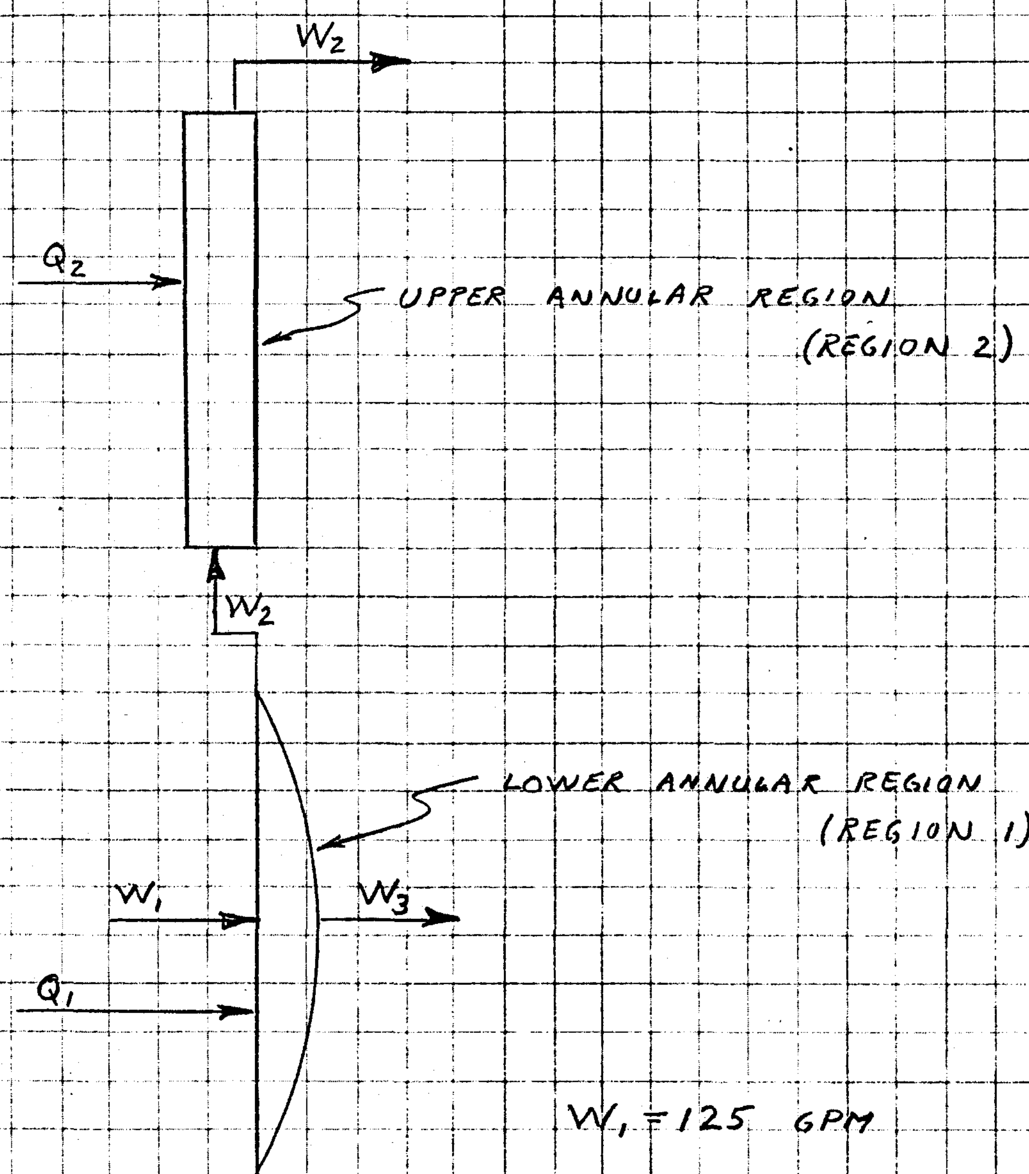
FIG. 2.1



$W_1 = 125$ GPM

$W_2 = 75$ GPM

$W_3 = 50$ GPM

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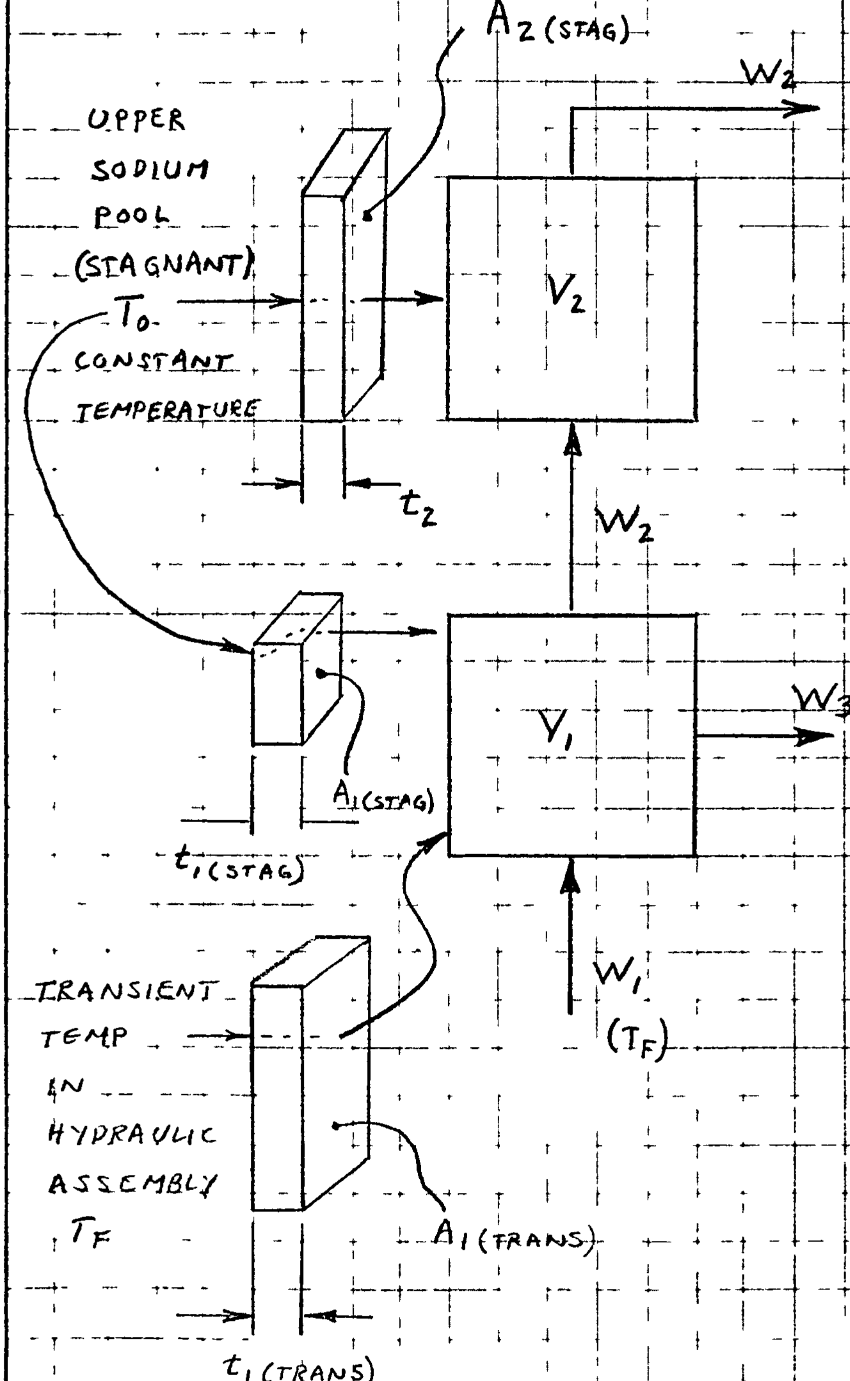
FIG. 2.2



$W_1 = 125$ GPM
$= 481.2\ IN^3/SEC$

$W_2 = 75$ GPM
$= 288.7\ IN^3/SEC$

$W_3 = 50$ GPM
$= 192.5\ IN^3/SEC.$

$V_1 = 267{,}914.\ IN^3$

$V_2 = 122{,}522\ IN^3$

$A_{1\,(TRANS)} = 20{,}281.\ IN^2$

$A_{1\,(STAG)} = 2461.\ IN^2$

$A_{2\,(STAG)} = 23{,}326.\ IN^2$

$t_{1\,(TRANS)} = 2.5$ IN

$t_{1\,(STAG)} = 2.0$ IN

$t_{2\,(STAG)} = 1.5$ IN

$T_0 = 935.0\ ^\circ F$

T_F = PP-3U TRANSIENT
or PP-2E TRANSIENT

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BY ______ DATE ______ SUBJECT ______ SHEET NO. 9 OF ______

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t	T(t)	$T_1(t)$	$T_2(t)$	$T_2 - T_1$
0	935.0	935.0	935.0	0
140	580.0	884.4	930.4	46.0
290	580.0	815.0	914.4	99.4
500	580.0	739.0	884.8	145.5
700	586.3	691.8	858.5	166.7
900	592.5	662.1	837.9	175.8
1100	598.8	644.3	823.2	178.9
1300	605.0	634.6	813.4	178.8

TABLE 2.1 TEMPERATURE BOUNDARY CONDITIONS FOR PPZET

BASIC TECHNOLOGY, INC.

BY ________ DATE ________ SUBJECT ________ SHEET NO. 10 OF ____

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t	T(t)	$T_1(t)$	$T_2(t)$	$T_2 - T_1$
0	935.0	935.0	935.0	0
50	970.0	936.5	935.1	−1.4
300	1000.0	953.8	938.1	−15.7
400	990.0	960.6	940.4	−20.2
500	820.0	955	942	−13
700	700.0	898.0	935.7	37.7
1000	635.0	798.7	905.1	106.4
1200	621.7	749.3	881.8	132.5
1400	615.0	710.2	860.8	150.6
1600	615.0	683.0	843.7	160.7
1800	615.0	664.7	830.7	166.0

TABLE 2.2 TEMPERATURE BOUNDARY CONDITIONS FOR PP3UT

BASIC TECHNOLOGY, INC.

BY________DATE________

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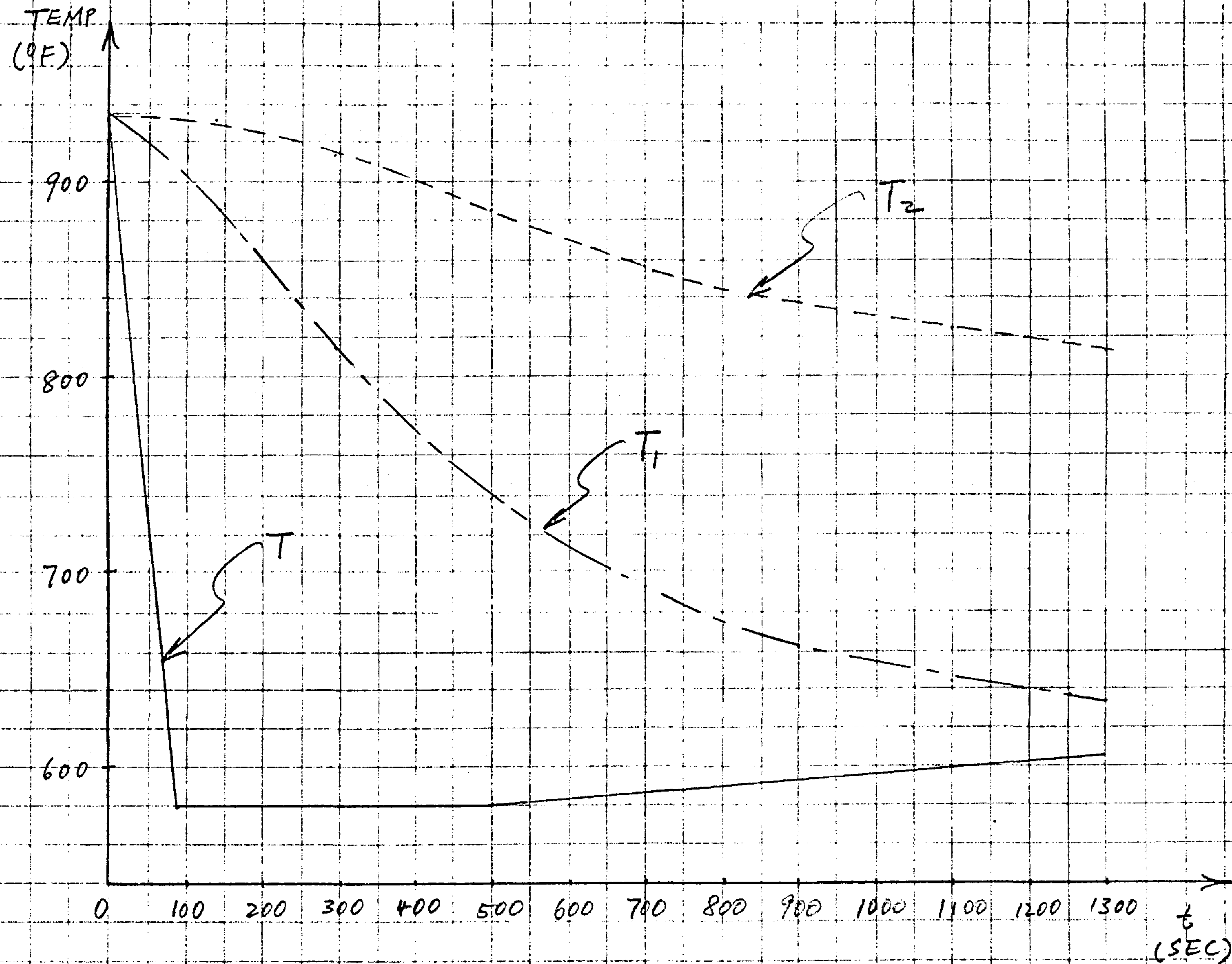

FIG 2.3 TEMPERATURE BOUNDARY CONDITIONS FOR PPZET

FIG 2.4 TEMPERATURE BOUNDARY CONDITIONS FOR PP3UT

BASIC TECHNOLOGY, INC.

BY........ DATE........
CHKD. BY........ DATE........
SUBJECT........
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JOB NO........

3.0 THERMAL ANALYSIS OF JUNCTION

A finite-element model of the cylinder-to-sphere junction has been set up using the ANSYS computer program. The model consists of 141 2-D axisymmetric isoparametric elements, with three elements in shells far away from the discontinuity area, and five or more elements in the discontinuity area. The computer plots with labelled element and node numbers are herewith attached. The lower part of the model extends to the 90°-plane of the spherical shell. Rather than modelling the lower hemisphere, its effects on the junction area are simulated with the appropriate boundary conditions. The upper part of the finite-element model includes a portion of the cylindrical shell to a point well exceeding the compensation distance of $2.5\sqrt{Rt}$. For the purpose of this analysis, the presence of the inlet and discharge nozzles is ignored in the axisymmetric model.

The most severe modified transients due to the PP2ET, from Fig. 2.3, are imposed as boundary conditions. The temperature profile $T_1(t)$ is applied to the bulk sodium which connects to the inner surface of the spherical region. The temperature profile $T_2(t)$ is applied to the sodium in the upper annular region which connects to the inner wall of the cylindrical tank. The surface adjacent to the gap between the junction and the hydraulic assembly is assumed to be insulated since heat flow in the radial direction in this area is small. A total of 38 iterations (time steps) is used for the PP2E transient, starting from the point of down transient to 1300 seconds beyond. The results of the thermal transient analysis provide the temperature at all nodes for each of the time steps. These results are saved on tape for a subsequent thermal stress analysis.

In addition to the thermal analysis of the junction area, a 1-D heat transfer analysis is also performed with the ANSYS program on the hydraulic assembly wall adjacent to the gap across from the pump tank. The result of this analysis is used, along with the thermal deflection analysis of the junction, to determine the gap closure and possible thermal growth restraint effects.

4.0 THERMAL STRESS ANALYSIS

The finite-element model for the thermal stress analysis of the junction is identical to that of the thermal model except that the thermal elements STIF 55 are replaced by the structural elements STIF 42. The boundary conditions are as follows:

1) All four uppermost nodes in the cylinder portion of the model are fixed in the axial direction;

2) All four lowermost nodes in the spherical portion of the model are coupled in the axial direction.

The temperature solutions for PP2ET at t = 290 sec and t = 700 sec are applied on the finite-element structural model to give the thermal stresses in the junction. The latter time has been chosen from considerations of both the ($T_A - T_B$) and the ($T_{surface} - \overline{T}$) effects. It appears that the membrane and bending stresses, as well as the peak stresses all reach a maximum at t = 700 sec, although the locations of the maximum differ for different types of stresses.

5.0 MECHANICAL LOADS

The mechanical loads considered in this section include internal pressure, axial force due to unbalanced pressures, radial force due to unbalanced nozzle pressure, deadweight and thermal growth restraint.

Internal Pressure

The structural finite-element model used for thermal stress analysis in Section 4.0 is also applicable to the stress analysis with internal pressure loading. The boundary conditions are the same as those in the previous section as far as the specified nodal displacements are concerned. In addition, an axial pressure loading of (- pR/2t), where p is the internal pressure in the spherical region, R and t are the radius and thickness of the sphere, must be applied on the cutoff surface of the hemisphere in order to simulate the effects of the internal pressure loading on the lower hemisphere, which is not included in the finite-element model. The suction pressure of 30 PSI is applied on the inner surface of the cylinder. The discharge pressure of 200 PSI is applied on the inside surface of the spherical shell. The discharge pressure of 200 PSI is also applied to the leakage gap region, except for the surface of the last element (No. 104) in that region, where a linear pressure drop is assumed.

A maximum stress intensity of 9.67 KSI is obtained at the lower cylinder-to-sphere junction, through elements 57-61.

Axial Unbalanced Pressure Load

Since the bottom of the spherical pump tank is exposed to the 200 PSI discharge pressure, an axial stress is developed in the pump casing due to the reaction of the unbalanced pressure at the top of the pump tank. This axial tensile stress is easily calculated to be 3.63 KSI at the thinnest part of the cross section.

Radial Unbalanced Pressure Load

The pressure differential of 170 PSI between the discharge and suction nozzles produces a lateral unbalanced force, which causes a tensile hoop stress of 1.38 KSI.

Deadweight

The axial tensile stress in the casing of 1.5-inch thickness due to the weight of the spherical pump tank is 50 PSI.

Thermal Growth Restraint

The thermal growth restraint effect is considered for PP2ET at t = 700 sec, at the lower tip of the sphere-junction discontinuity. Initially, (at t = 0 sec), there is a radial gap of 0.01 inches between the junction and the hydraulic assembly. As the transient develops, the junction may cool down faster than the hydraulic assembly, thus closing the gap and inducing stresses from restraint. The radial displacement of the junction tip (Node 99) at t = 700 sec from its initial position at 935°F (t = 0 sec) is calculated from the thermal stress-deflection analysis in Section 4.0:

$$\Delta R_{99} = .418'' - \alpha R \Delta T$$

$$= .418'' - 10.2 \times 10^{-6} \times 54.5 \times (935 - 70)$$

$$= .418'' - .481'' = -.063''$$

The radial displacement of the hydraulic assembly is calculated from the temperatures obtained from the 1-D thermal transient analysis discussed in the last paragraph of Section 3.0. The average temperature for PP2ET at t = 700 sec is 845.31°F. The radial displacement is thus

$$\Delta R_{H.A.} = \alpha R\ (845.31 - 935)$$

$$= -0.050''$$

Furthermore, the pressure loading on the junction, from the finite-element analysis, causes the tip to move out radially by 0.019 inches. Therefore, the gap remains open during thermal transient and no thermal growth restraint occurs at the lower tip.

6.0 SUMMARY OF RESULTS

The primary and secondary stresses obtained in Sections 4.0 and 5.0 are summarized here for two worst sections across the junction. These sections correspond, respectively, to the lower discontinuity region nearest to the spherical shell, and to the thickest part of the discontinuity region. The first section is expected to have the highest linearized stresses, while the second section is to have the highest peak stresses.

The primary membrane stresses (P_L) due to internal pressure are summarized in Table 6.1 for elements 57-61. The stress intensity is 9.67 KSI. Adding the stress components of other primary loads discussed in Section 5.0, the total primary stress intensity is 10.55 KSI.

The linearized secondary stresses (Q) receive contributions from both the thermal loading and the local bending stresses due to pressure loading. These are summarized in Table 6.2 for elements 57-61. The maximum value of Q occurs at the inside surface of element #57 and is equal to 18.93 KSI (neglecting any thermal growth restraint).

The peak stress intensities (F) for section through elements 77-82 are summarized in Table 6.3. The maximum peak stress of 46.8 KSI occurs at the inside surface of element #77 for PP2ET at t = 700 sec.

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LOCATION	σ_x	σ_y	τ_{xy}	σ_z	LINEARIZED STRESS INTENSITY
57	−0.89	−0.73	0.44	7.81	
58	0.10	1.33	−0.79	8.72	
59	0.50	3.59	−1.66	9.47	
60	1.76	5.29	−2.87	10.36	
61	2.64	7.68	−4.36	11.30	
AVERAGE (P_L)	0.82	3.43	−1.85	9.53	
INNER SURFACE					
$P_L + Q$	−1.27	−1.56	0.95	7.46	9.67
Q	−2.09	−4.99	2.80	−2.07	
OUTER SURFACE					
$P_L + Q$	2.91	8.42	−4.65	11.60	11.34
Q	2.09	4.99	−2.80	2.07	

TABLE 6.1 STRESSES FROM PRESSUE LOADING

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BY ________ DATE ________ SUBJECT ________ SHEET NO. 21 OF ______
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LOADING	σ_x	σ_y	τ_{xy}	σ_z
INTERNAL PRESSURE	0.82	3.43	-1.85	9.53
UNBALANCED AXIAL LOAD	—	3.63	—	—
UNBALANCED RADIAL LOAD	.04	—	—	1.38
DEADWEIGHT	—	.05	—	—
TOTAL	0.86	7.11	-1.85	10.91

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 7.62$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 0.36$$

$\therefore$ PRIMARY MEMBRANE STRESS INTENSITY = 10.55 KSI

SECTION THRU ELEMENTS 57-61

TABLE 6.1 (CONT'D)

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BY ________ DATE ________ SUBJECT ________ SHEET NO. 22 OF ______
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LOADING	σ_x	σ_y	τ_{xy}	σ_z
INTERNAL PRESSURE	0.07	3.62	-.34	5.90
UNBALANCED AXIAL LOAD	—	3.63	—	—
UNBALANCED RADIAL LOAD	.04	—	—	1.38
DEADWEIGHT	—	.05	—	—
TOTAL	0.11	7.30	-.34	7.28

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 7.32$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 0.10$$

$\therefore$ PRIMARY MEMBRANE STRESS INTENSITY = 7.22 KSI

SECTION THRU ELEMENTS 109-111

TABLE

BASIC TECHNOLOGY, INC.

BY ______ DATE ______ SUBJECT ______ SHEET NO. 23 OF ______

CHKD. BY ______ DATE ______ JOB NO. ______

LOCATION	σ_x	σ_y	τ_{xy}	σ_z	LINEARIZED STRESS INTENSITY
57	6.05	13.07	-8.77	21.87	
58	1.63	6.75	-3.04	14.92	
59	-0.29	-0.89	0.53	9.43	
60	-3.71	-5.64	4.46	5.16	
61	-4.79	-12.46	7.42	2.14	
AVERAGE	-0.22	0.17	0.12	10.70	
INNER SURFACE	6.26	15.40	-7.67	22.51	19.05
OUTER SURFACE	-6.70	-15.06	7.91	-1.11	18.72

TABLE 6.2 THERMAL STRESSES FOR

PPZET AT $t = 700$ SEC.

LOCATION	LOADING	σ_x	σ_y	τ_{xy}	σ_z	σ_1	σ_2	STRESS INTENSITY
INNER SURFACE:	THERMAL	6.26	15.40	-7.67	22.51			
	PRESSURE	-2.09	-4.99	2.80	-2.07			
	TOTAL	4.17	10.41	-4.87	20.44	1.51	13.08	18.93
OUTER SURFACE:	THERMAL	-6.70	-15.06	7.91	-1.11			
	PRESSURE	2.09	4.99	-2.80	2.07			
	TOTAL	-4.61	10.07	5.11	0.96	-13.13	-1.55	14.09

TABLE 6.2 LINEARIZED SECONDARY STRESSES

ELEMENTS NO. 57-61

LOCATION (EL. NOS. 77-82)	PEAK STRESS INTENSITY			
	PRESSURE	OTHER	THERMAL	TOTAL
77	8.58	5.06	33.16	46.80
82	8.61	5.06	14.05	27.72

TABLE 6.3 PEAK STRESS INTENSITIES

PPZET AT $t = 700$ SEC

7.0 CODE EVALUATION

The stresses summarized in Section 6.0 are compared with the criteria set forth in the Elevated Temperature Code Case 1592.

Primary Membrane Stress

The code limits for the primary stresses are

$P_m \leq S_o$

$P_L + P_b \leq 1.5\ S_o$ for design condition

and

$P_m \leq S_{mt}$

$P_L + P_b \leq 1.5\ S_m$ for normal and upset conditions

$K_t S_t$

Since $S_o \simeq S_{mt} \simeq 15.5$ KSI for the conditions applicable to the pump, it is seen that with $P_L = 10.55$ KSI, these Code criteria are satisfied.

Creep-Ratchetting

The worst section for possible ratchetting is the lower sphere-junction discontinuity because of the large primary membrane stress at that location. According to Code Case 1592, the parameters x and y are calculated:

$x = (P_L + P_b/K_t)/S_y$

For $S_y = 17.5$ KSI, and $P_L = 10.6$ KSI, $P_b = 0$, $x = 0.61$. For PP2ET, $Q_R = 19$ KSI, so

$y = Q_R/S_y = 1.08.$

From Fig. T - 1324 - 1, the z equation for regime S_1 should be used. Thus,

$$z = y + 1 - 2\sqrt{(1 - x)y} = 0.78 = \sigma_c/S_y$$

So the creep-ratchetting stress is 1.25 σ_c = 17.1 KSI. The strain from the isochronous curve is less than 0.3%. Therefore, the creep ratchetting limit by the elastic criterion of T-1324 is satisfied.

Fatigue Evaluation

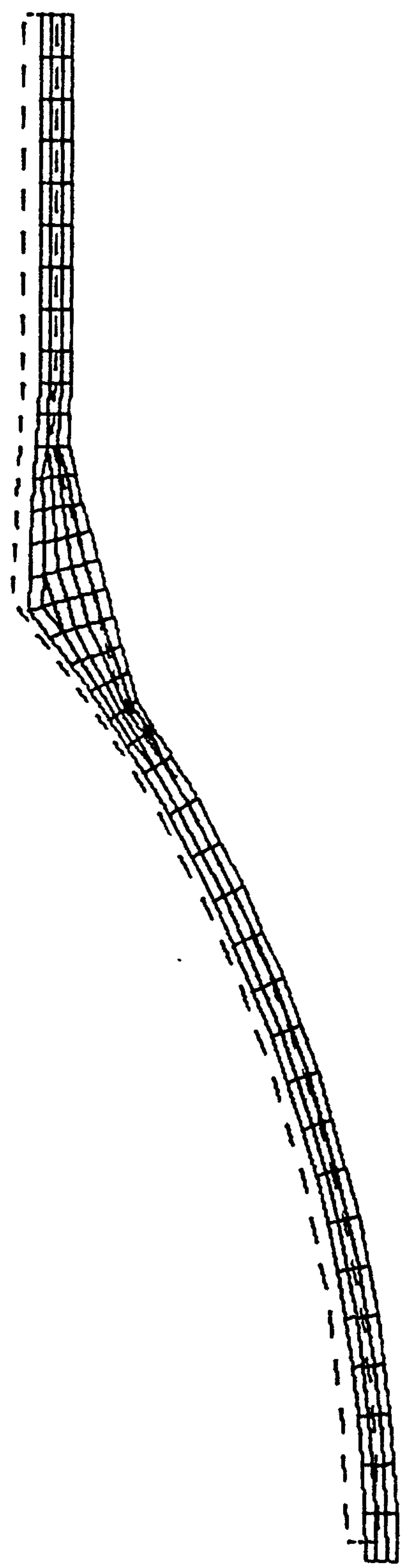
The equivalent stain range is obtained by dividing the peak stress by the Young's modules. For a peak stress of 46.8 KSI at the inside surface of thickest part of the junction at t = 700 sec for the PP2E transient, the equivalent strain range is 2.0×10^{-3}. The allowable number of cycles N_d is 2×10^4. Since the sum of all cycles including PP2ET, PP2UT, PP3UT, PP9UT, PP10UT and PP11UT is only 212, the fatigue damage is negligible.

Creep damage is also small since $S \simeq 17$ KSI and the minimum time to rupture is greater than 10^6 hours.

8.0 CONCLUSION

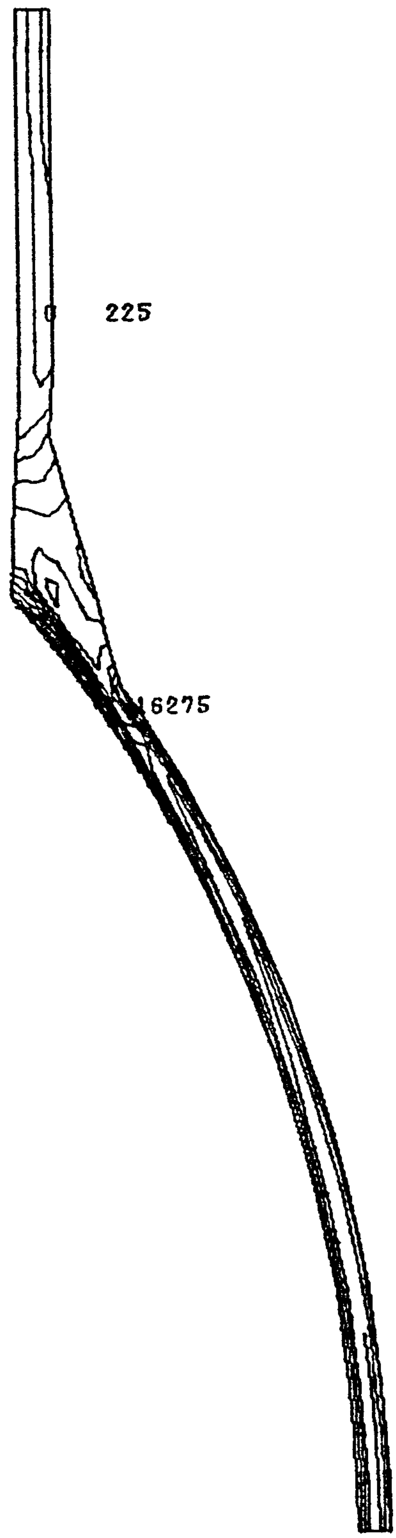
The cylinder-to-sphere junction of the large scale sodium pump has been investigated using the finite-element method. Results indicate that in spite of the large primary membrane stresses, the design passes the criteria set forth by Code Case 1592. This is largely due to the mitigated thermal transients in the region under consideration. It would improve the design of the junction if the reinforcement region be modified so that there is a smoother transition between the cylinder and the sphere with the junction. This would also reduce the primary membrane stress intensity in the discontinuity region. A more detailed thermal-hydraulic-stress analysis is recommended for this region including portion of the hydraulic assembly in order to have a more accurate calculation of the thermal growth restraint effect, when further design iterations are made in the future. In addition, it is recommended that both the inlet and discharge nozzles should be included in the study.

STEP= 1 ITERATION= 1



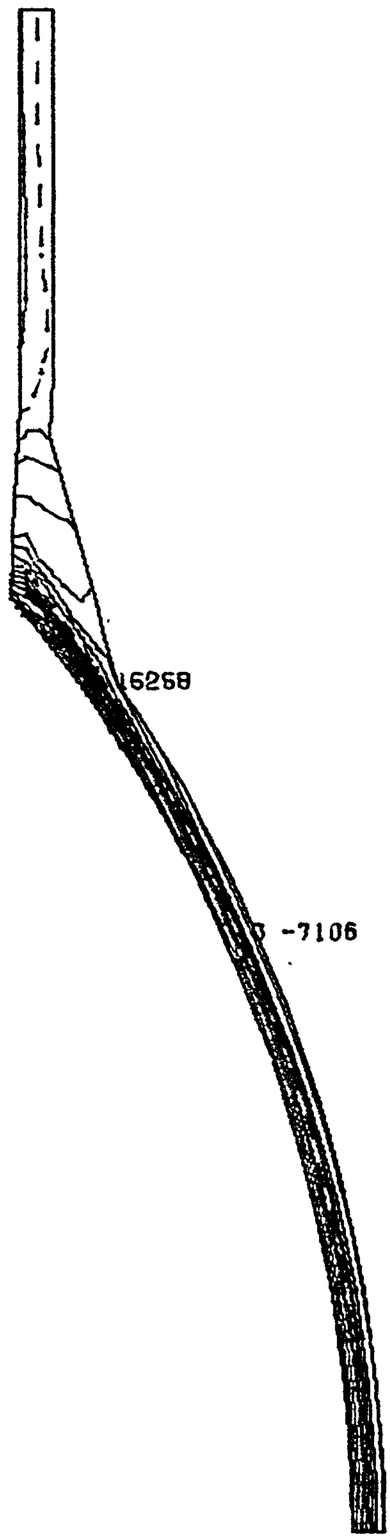
LARGE SCALE LMFBRP CYLINDER-SPHERE JUNCTION THERMAL STRESS POST PLOTS

STEP= 1 ITERATION= 1



LARGE SCALE LMFBRP CYLINDER-SPHERE JUNCTION THERMAL STRESS POST PLOTS

STEP= 1 ITERATION= 1



LARGE SCALE LMFBRP CYLINDER-SPHERE JUNCTION THERMAL STRESS POST PLOTS

BYRON JACKSON REPORT TCF-1020-DYN
TORSIONAL VIBRATION ANALYSIS
of
LSBRP HOT LEG PUMP ASSEMBLY

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
SUNNYVALE, CALIFORNIA
Change Order 1 to LETTER CONTRACT 67002
BJ JOB NO. 761-G-0002

PREPARED BY Jerel E. Ellison DATE 28 Sept. 1976

CHECKED BY [illegible] DATE 30 Sept 1976

APPROVED BY [illegible] DATE 6 oct 76

APPROVED BY Kent A [illegible] DATE 8 OCT 76

September 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

TABLE OF CONTENTS

1.0 SCOPE

This report describes the torsional analysis performed in accordance with the requirements of Reference 6.2 for the pump assembly of Reference 6.1. It includes a detailed description of the model used, the results obtained and draws appropriate conclusions.

2.0 METHOD OF ANALYSIS

The torsional analysis described in this report was conducted using the proprietary Byron Jackson computer program "TORVIB" which is based on the "Holzer" Method. References 6.3 and 6.4 contain the details of this method and of the program capability.

In general, TORVIB uses a mathematical model consisting of a number of inertial masses, separated by sections of massless shafting which provide torsional rigidity. The TORVIB model begins and ends with an inertial mass. The shafting between inertial masses can involve multiple sections of shafting of varying diameters and lengths.

The program generates successive Holzer tables for the entire model each having an angular velocity one radian per second greater than the preceding table. The initial table is formed for an angular velocity of one radian per second. The residual torques for each table are examined until two successive tables are found where the residual torque differs in sign. Linear interpolation between these two frequencies to locate the point where the residual torque is zero then determines the torsional critical frequency. The Holzer table for this frequency is then printed by the computer along with the critical frequency and the search for the next critical frequency is resumed. The program continues to determine torsional critical frequencies until all possible frequencies have been found or a frequency of 100,000 cycles per minute is reached. On special occasions, the 10^5 cpm limit may be exceeded by entering another upper limit in the input data.

The model used in this analysis is detailed in Appendix A and the resulting computer output using the normal 100,000 cycle per minute upper limit is presented in Appendix C.

3.0 RESULTS

The results of this TORVIB run were nine of the seventeen possible torsional critical frequencies. The first of these frequencies was found to be 574.16 cycles per minute. A rough manual calculation using a single disc model is presented in Appendix B, which predicts a first torsional critical frequency of 476.5 cycles per minute. This agrees with the calculated value within seventeen percent and substantiates the accuracy of the model.

The first three natural frequencies determined are shown in Table 3.1. The other six frequencies so determined exceeded 32,000 cycles per minute and are so far from any possible exciting frequency that they need not be considered. They are included in Appendix C for purposes of providing complete information.

TABLE 3.1 Torsional Natural Frequencies

Mode	Frequency (cpm)
1	574
2	13,916
3	24,776

4.0 COMPARISON WITH POSSIBLE EXCITING FREQUENCIES

The major exciting frequency developed by the pump occurs due to pressure pulses generated when impeller vanes pass volute lips (cutwaters). The impeller will be installed in the hydraulic assembly which has three volute lips angularly spaced at one hundred twenty degree intervals.

This unit utilizes a double suction impeller of five vane design. The vanes for each half of the impeller are staggered with respect to the other half which in effect makes the impeller a ten vane impeller. Thus successive vane tips are displaced from each other by thirty-six degrees.

Every twelve degrees of rotation of the pump shaft will result in a vane passing a volute lip. This will generate a pressure pulse for every twelve degrees of revolution or thirty pulses per revolution. The design rotating speed of the unit is 600 RPM. Thus the impeller will generate pressure pulses at a rate of 18,000 pulses per minute. This is the internally generated frequency that can excite the unit torsionally. The first torsional natural frequency is well below this possible exciting frequency and thus no torsional resonance is expected at this frequency.

The second and third natural frequencies straddle this possible excitation frequency. The second torsional critical is less than 78 percent of the possible exciting frequency. The third torsional critical is more than 137 percent of the exciting frequency. Both of these are far enough removed so that no resonance at this natural frequency need be considered.

It is also noted that the first mode torsional natural frequency is approximately 96 percent of the operating speed of 600 RPM. This mode is due to the impeller vibrating against the inertial mass of the motor. The other modes occur at frequencies in excess of 10,000 cpm and their proximity to possible excitation frequencies has already been discussed.

Because the driver is an electric motor, there is little likelihood that the first mode natural frequency, which is close to the motor operating speed, will ever be excited. On the other hand, the frequency of this mode can easily be increased since it is very sensitive to the impeller moment of inertia and to the torsional rigidity of the shafting as was shown in Reference 6.6.

5.0 CONCLUSIONS

The torsional vibration analysis has shown that the specification requirements can be satisfied within the existing conceptual design. An optimization of shafting and impeller parameters will be required. However this optimization is deemed to be a detailed design consideration.

6.0 REFERENCES

6.1 Byron Jackson Drawing SK-(TCF-1019-DYN-01) Rev. 0 dated 18 August 1976.

6.2 Letter from Y. Usui to J. Husmann dated 16 Sept. 1976.

6.3 Byron Jackson Report TCF-1009-PGM, "Documentation for Program TORVIB".

6.4 Handbook on Torsional Vibration, Compiled by E. J. Nestorides of the BICERA Research Laboratory published by the SYNDICS of the Cambridge University Press, 1958.

6.5 Rotor Dimension Large Scale Breeder Reactor Pump Hot Leg dated 9 September 1976, by Y. Usui.

6.6 Byron Jackson Report TCF-1017-DYN, Original Issue dated June 1976.

6.7 Letter from Y. Usui to J. Husmann dated 24 Sept. 1976.

APPENDIX A

MODEL DETAILS

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APPENDIX A

A-1.0 GENERAL

This appendix describes the model used in this Analysis. As described in the report, the model is required to begin and end with an inertial mass. Between these two inertial masses, additional inertial masses may be placed. Connections between all inertial masses are modeled as one or more sections of massless shafting (multiple sections are connected in series).

The model used for this analysis is basically described in Figure A-1 and consists of 18 inertial masses interconnected by

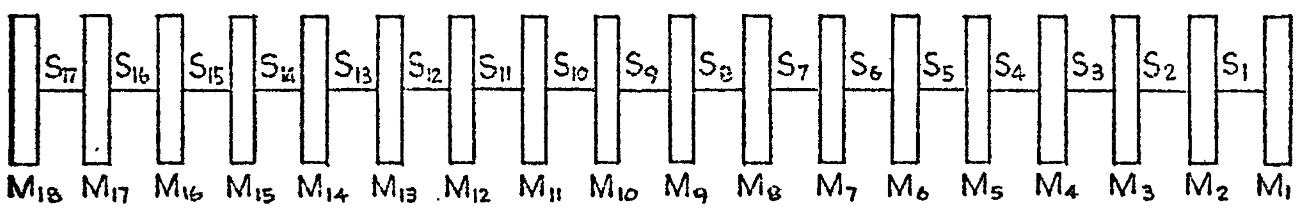


Figure A-1 TORVIB Model of Rotating Element

17 sections of shafting with torsional stiffness properties only. The identifiers shown in Fig. A-1 are Identifiers only. The values used as input to the program were derived as shown in the following two sections of this Appendix. Section A-2.0 details the derivation of the inertial mass values while Section A-3.0 shows how the torsional stiffnesses were derived. Section A-4.0 includes other calculations.

A-2.0 INERTIAL MASSES

Program TORVIB requires that the inertial masses be input in the form of polar weight moments of inertia (WK^2). The units as input are pound-inch squared.

Relating the model shown in Figure A-1 to the drawing from which it was taken (Reference 6.1) the inertial masses indicated represent the actual parts as indicated in Table A-1 which also indicates the section of the Appendix where the origin of the appropriate values may be found. Most of the moments of inertia required were calculated using a supplementary program called WTMOM which was primarily written for determination of Impeller weights and moments of inertia. The appropriate runs of this program are included in Appendix B and are referenced by number in the appropriate section

A-2.1 Motor Rotor

The moment of inertia of the motor rotor was estimated to be 18,000,000 lb-in^2. This value was provided via Reference 6.5. It is input as M18.

TABLE A-1 Model Mass Identifiers vs Pump Parts

Masses	Pump Part	Section Number
M1, M2	Lower Hydrostatic Bearing	A-2.4
M3	Impeller	A-2.2
M4, M5	Upper Hydrostatic Bearing	A-2.5
M6 thru M16 Inclusive	Hollow Shafting	A-2.3
M17	Curvic Coupling	A-2.6
M18	Motor Rotor	A-2.1

A-2.2 Impeller

The impeller was considered to be torsionally rigid between wear ring extremities. The torsional mass calculation was performed using WTMOM. The dimensions used in this run were scaled from Reference 6.1. M3 represents the impeller. Moment of Inertia due to the vanes was calculated as part of WTMOM using estimated values based on Reference 6.6 The value of the moment of inertia thus determined was 7,240,593 lb-in^2. The Computer Run is contained in Appendix B as Run B-1.

A-2.3 Hollow Shafting

The mass of the hollow shafting was calculated from the drawing dimensions (12" OD x 1.5" thick) using WTMOM. The Computer Run is contained in Appendix B as Run B-6. The results of this calculation indicate the moment of inertia of the shafting is 393.83 lb-in^2 per inch of length. The hollow portion of the shafting was divided into ten sections, six of which are 15 inches long and are located between the wide end of the internal taper at the top of the shaft and the normal sodium level at elevation 10' as shown in Ref. 6.1. The remaining four 16 inch long sections are located between the sodium level and the beginning of the external taper on the shaft. The inertial mass of any remaining portions of shafting was disregarded as being relatively insignificant. The only exceptions to this are in the case of the two hydrostatic bearings where the inertial mass of the shafting within the bearings was considered to be part of the bearings.

For each section of shafting, the inertial mass was calculated by multiplying the moment of inertia, (393.83 lb-in^2/in) by the length of the section and applying half the resulting mass at each end of the section. Thus at the junction between two sections of shafting, half the inertial mass of each adjacent section was applied. The results of these calculations are tabulated in Table A-2.

TABLE A-2. Moments of Inertia Hollow Shaft Section

Length (in)	Total Moment of Inertia ($lb\text{-}in^2$)	Moment of Inertia Applied at Each End ($lb\text{-}in^2$)
16	6301.28	3150.64
15	5907.45	2953.73

Accordingly the inertial masses input to represent the shafting were calculated as indicated in Table A-3.

TABLE A-3. Moments of Inertia Hollow Shafting

Mass Number	Inertia Above Location	Inertia Below Location	Total Shafting Inertia
M16	0	2953.7	2953.7
M15	2953.7	2953.7	5907.45
M14	2953.7	2953.7	5907.45
M13	2953.7	2953.7	5907.45
M12	2953.7	2953.7	5907.45
M11	2953.7	2953.7	5907.45
M10	2953.7	3150.6	6104.3
M9	3150.6	3150.6	6301.3
M8	3150.6	3150.6	6301.3
M7	3150.6	3150.6	6301.3
M6	3150.6	0	3150.6

A-2.4 Lower Hydrostatic Bearing

The journal of the lower hydrostatic bearing is connected only to the lower portion of the bearing. Therefore one calculation was made for lower portion and a second calculation for the upper portion of this bearing. These calculations were made using WTMOM. The calculation of the upper portion included the shaft above the bearing to the wide end of the internal taper. These runs are included in Appendix B as Run B-2 and B-3. The moments of inertia thus determined were 100873.8 lb in^2 for the lower portion and 39549.4 for the upper portion of the bearing. These were input to the program as M1 and M2 respectively.

A-2.5 Upper Hydrostatic Bearing

The journal of the upper hydrostatic bearing is connected only to the upper portion of the bearing. Therefore one calculation was made for the lower portion bearing and a second calculation for the upper portion of this bearing in the same manner as was used for the lower bearing. The calculation of the lower portion included the shaft below the bearing to the wide end of the internal taper. The calculation of the upper portion extended above the bearing to the point on the shaft where both the external and internal tapers begin. The runs are included in Appendix B as Runs B-4 and B-5. The moments of inertia determined by WTMOM for these portions were 32817.4 for the lower portion and 98723.4 for the upper portion. These values were input to TORVIB as M4 and M5 respectively.

A-2.6 Curvic Coupling

Those portions of the curvic coupling which involve the bolted flanges and the hub were considered to be torsionally rigid and contribute to the inertial mass of the system. That portion of the curvic coupling which extends to the self centering spline as depicted in Reference 6.1 was considered to provide no inertial mass.

Accordingly, the dimensions were scaled from Reference 6.1, run through WTMOM and the results were found to be 30374.2 $lb\text{-}in^2$ as shown in Run B-7 in Appendix B. This value was used as M17 in the input to TORVIB.

A-3.0 Torsional Stiffnesses

TORVIB permits torsional stiffnesses to be input in two different ways. These are after direct calculation or by means of shaft geometry. When the shaft geometry method is used for input, the program first calculates an equivalent length of 2" diameter shafting for that particular geometry. The equivalent length of successive sections of shafting are added to determine a single equivalent length from which is determined the torsional stiffness between inertial masses.

The equivalent length calculation is described in References 6.3 and 6.4. Most of the geometries included in Reference 6.1 are included in program TORVIB.

The method chosen for input of torsional stiffness to TORVIB for this analysis was the shaft geometry method. Accordingly, the sections of shafting between inertial masses were scaled from Reference 6.1 and input directly. The logic involved in the inputs for torsional stiffness is explained briefly in the following sections.

A-3.1 Hydrostatic Bearings

In the two hydrostatic bearings, the shaft torsional stiffness between the two inertial masses is considered to be the stiffness of the shaft between the two masses. Hence the torsional stiffnesses represented by S_1, and S_4 as defined by Figure A-1 were input by the use of the shaft dimensions only. These were scaled from Reference 6.1.

A-3.2 Shafting Inside the Hydraulic Assembly

The remaining shafting inside the hydraulic assembly, S_2, and S_3 as defined by Figure A-1, is represented to the computer by the geometric dimensions as scaled from Reference 6.1.

A-3.3 Hollow Shafting

Sections S_6 through S_{15} represent sections of hollow shafting. These were input to the computer program TORVIB by the use of their geometry.

A-3.4 Motor Shafting

Reference 6.4 indicates that where a shaft penetrates a coupling 1/3 the penetration should be modeled as shafting with the remaining 2/3 of the penetration considered to be torsionally rigid. The shaft penetration is ten inches from the split ring (circular key) to the end of the curvic coupling hub. Therefore 3.33" or one third of the penetration is modeled in for torsional stiffness.

The diameter of the motor shaft of Reference 6.1 disagrees with the diameter of Reference 6.5. The shaft dimensions of Reference 6.5 represent an equivalent mathematical model of the actual shaft according to verbal communication with Y. Usui. To provide compatibility of these two references it was assumed that the last 21" of motor shafting was actually 10" in diameter to provide for coupling installation.

Of the total shaft length of 52.5" shown in Reference 6.5, 31.5" is modeled with a diameter of 12.25" and 11.833" is modeled with a diameter of 10". The 11.833" section with the diameter of 10" includes 8.5" above the coupling and 3.333" as the portion of penetration to be modeled. 2.5 inches of the shaft extends below the split ring but is not considered because it is not in direct contact with the coupling.

A-3.5 Curvic Coupling to Hollow Shaft

This section of shafting is represented in Figure A-1 as S_{16}. This is a composite section made up of several different geometries. From the lower face of the flange of the curvic coupling downward, the shafting was considered to be solid at 10" diameter as shown by Reference 6.1. Scaling was used for the next two sections of 11.25 inches diameter (this was made two sections for reasons discussed in Section A-4.0). The next short section is a solid section with an external taper. The last section, immediately adjacent to M_{16}, was approximated by extending the internal taper beginning at M_{16} to the plane where the external taper begins. Above this plane, the shaft was considered to be solid. It is felt that the error introduced by this approximation is negligible for the purposes of this analysis. All of the above mentioned sections which were not assigned dimensions by Reference 6.1 were scaled from that reference.

A-3.6 Immediately above Upper Hydrostatic Bearing

The only other section of shafting is that represented by S5 in Figure A-1. This again is a composite section of shafting, made up of two separate sections. The upper section of the two is a section with a straight bore and an external taper which was input directly from the scaled dimensions. The remaining section is a section with both internal and external taper. TORVIB does not calculate this geometry and hence an equivalent length and diameter were used. This calculation is included in Appendix B as Calculation B-9.

A-4.0 OTHER CALCULATIONS

The only other items required for input to TORVIB are the moduli of elasticity of the various shaft sections. In accordance with Reference 6.7, the temperature of the sodium is 915°F., and all material below normal sodium level (Elevation 10' on Reference 6.1) was assumed to be at that temperature. The modulus of elasticity for austenitic stainless at this temperature is 22.9×10^6 psi. This value was used for all shafting between M1 and M10 which was purposely placed at this point for this reason.

All shafting material above the bottom of the foundation plate was considered to be at 200°F, with a modulus of elasticity of 27.7×10^6 psi in accordance with Reference 6.7. This point is the point at which the division between two shaft sections of 11.25 inch diameter was made as mentioned in Section A-3.5.

Between these two points, the modulus of elasticity was considered to vary linearly with length. Hence the moduli of elasticity as input were obtained by linear interpolation at the center of length of each section. The results of this interpolation are listed along with the remaining computer input and the Holzer tables at the torsional critical frequencies in Appendix C.

In addition, a preliminary check calculation was performed manually using a simplified model. For this calculation, the motor moment of inertia was assumed to be infinite, the total impeller moment of inertia was concentrated at the center of the impeller and the shafting was assumed to be uniform (12" O.D. x 9" I.D.) over the entire length.

Using this simplified model, the 1 node critical frequency was determined to be 476.5 RPM. This calculation is included in Appendix B as B-8. The purpose of this calculation was to provide some verification of the results from the program.

APPENDIX B

DETAILED CALCULATIONS

TABLE OF CONTENTS

B-1 WTMOM IMPELLER CALCULATION

```
                         RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?2,13

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (0 IF UNKNOWN)
?5,1.25,30,45,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,2,68,47,.5,2
?2,1,47,36,2,2
?3,2,36,30,2,2.5
?4,2,30,26,2.25,3
?5,2,26,24,3.5,4.5
?6,3,24,21,4.5,0
?7,1,23.5,20,5.5,5.5
?8,1,68,60,1.75,1.75
?9,2,60,54,1.75,2
?10,2,54,48,2,2.75
?11,2,48,46,3,3.75
?12,3,46,42,4,0
?13,1,45.5,42,5.75,5.75

      TOTALS

VOLUME           =    41045.313 IN3
IMPELLER WEIGHT  =    11615.824 LBS
POLAR MOMENT OF INERTIA =7240593.000 LB-IN2    OR  18738.595 LB-IN-SEC2
VANE MOMENT OF INERTIA  =3223546.875 LB-IN2
```

B-2 WTMOM LOWER BEARING LOWER PORTION CALCULATION

```
RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,3

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (0 IF UNKNOWN) .
?0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,21,17,2.5,2.5
?2,1,20.5,17,8,8
?3,1,24,21,19,19

       TOTALS

VOLUME            =    3137.663 IN3
IMPELLER WEIGHT =      887.959 LBS
POLAR MOMENT OF INERTIA = 100873.809 LB-IN2    OR    261.061 LB-IN-SEC2
VANE MOMENT OF INERTIA  =        0.    LB-IN2
```

B-3 WTMOM LOWER BEARING UPPER PORTION CALCULATION

```
                   RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,4

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (0 IF UNKNOWN)
?0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,20.5,17,8,8
?2,1,21,17,1.5,1.5
?3,1,23.5,20,2.75,2.75
?4,3,20,17,2.75,0

       TOTALS

VOLUME            =    1455.685 IN3
IMPELLER WEIGHT =       411.959 LBS
POLAR MOMENT OF INERTIA =  39549.392 LB-IN2     OR     102.353 LB-IN-SEC2
VANE MOMENT OF INERTIA  =         0.     LB-IN2
```

B-4 WTMOM UPPER BEARING LOWER PORTION CALCULATION

```
                    RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,4

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (0 IF UNKNOWN)
?0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,20,17,8,8
?2,1,21,17,1.5,1.5
?3,3,20,17,2.5,0
?4,1,23,20,2.5,2.5

       TOTALS

VOLUME            =    1241.713 IN3
IMPELLER WEIGHT =       351.405 LBS
POLAR MOMENT OF INERTIA =  32817.351 LB-IN2    OR     84.931 LB-IN-SEC2
VANE MOMENT OF INERTIA  =        0.    LB-IN2
```

B-5 WTMOM UPPER BEARING UPPER PORTION CALCULATION

```
                         RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,2_3

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (O IF UNKNOWN)
?0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,20,17,8,8
?2,1,21,17,3,3
?3,1,24,21,19,19

       TOTALS

VOLUME              =     3070.119 IN3
IMPELLER WEIGHT =      868.844 LBS
POLAR MOMENT OF INERTIA =  98723.357 LB-IN2     OR     255.495 LB-IN-SEC2
VANE MOMENT OF INERTIA  =         0.     LB-IN2
```

B-6 WTMOM 12 INCH O.D., 9 INCH I.D. SHAFT PER INCH CALCULATION

```
                    RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,1,

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO CG OF VANES (0 IF UNKNOWN)
?0,0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?1

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,12,9,1,1

    SECTION #     1
VOLUME FOR THIS SECTION         =     49.4800 IN3
WEIGHT FOR THIS SECTION         =     14.0029 LBS
MOMENT OF INERTIA THIS SECTION =    393.8302 LB-IN2


       TOTALS

VOLUME            =      49.480 IN3
IMPELLER WEIGHT =      14.003 LBS
POLAR MOMENT OF INERTIA =    393.830 LB-IN2    OR       1.019 LB-IN-SEC2
VANE MOMENT OF INERTIA  =        0.    LB-IN2
```

B-7 WTMOM CURVIC COUPLING CALCULATION

```
                        RELEASE 1.0

ENTER EITHER 1 OR 2 FOR SINGLE OR DOUBLE SUCTION, NO OF SECTIONS
?1,3

ENTER MATERIAL DENSITY IN LBS/IN^3
?.283

ENTER NUMBER OF VANES, THICKNESS, WIDTH, LENGTH, RADIAL DISTANCE
TO C-F VANES (0 IF UNKNOWN)
?0,0,0,0,0

DO YOU WANT THE VOLUME, WEIGHT AND MOMENT OF INERTIA
PRINTED FOR EACH SECTION?  0=NO,1=YES
?0

ENTER SECTION NUMBER, CODE, OD, ID, LENGTH ALONG OD AND ID
?1,1,14,10,10,10
?2,1,14,11,6,6
?3,1,20.5,10,4,4

       TOTALS

VOLUME             =    2113.505 IN3
IMPELLER WEIGHT =        598.122 LBS
POLAR MOMENT OF INERTIA =  30374.249 LB-IN2    OR     78.608 LB-IN-SEC2
VANE MOMENT OF INERTIA  =        0.    LB-IN2
```

B-8. CHECK CALCULATION FOR FIRST TORSIONAL CRITICAL

Impeller

$$I = \frac{7240593}{386.4} = 18738.595 \quad lb.\ in.sec^2$$

Shaft Stiffness

$$K = \frac{\pi}{32} \quad G \quad \frac{D_o^4 - D_i^4}{L}$$

$$= \frac{\pi}{32} \left(11.8 \times 10^6\right) \left(\frac{12^4 - 9^4}{352}\right)$$

$$= 46651144.8$$

Critical Frequency

$$\omega = \sqrt{\frac{K}{I}} = \left[\frac{46651144.8}{18738.595}\right]^{1/2} =$$

$$\omega = 49.90 \quad cps.$$

$$f = \frac{\omega}{2\pi} = \frac{49.90}{2\pi} = 7.942 \quad RPS$$

or

$$f = 60 \times 7.942 = 476.5 \quad RPM$$

B-9 EQUIVALENT LENGTH FOR TAPER ABOVE UPPER BEARING

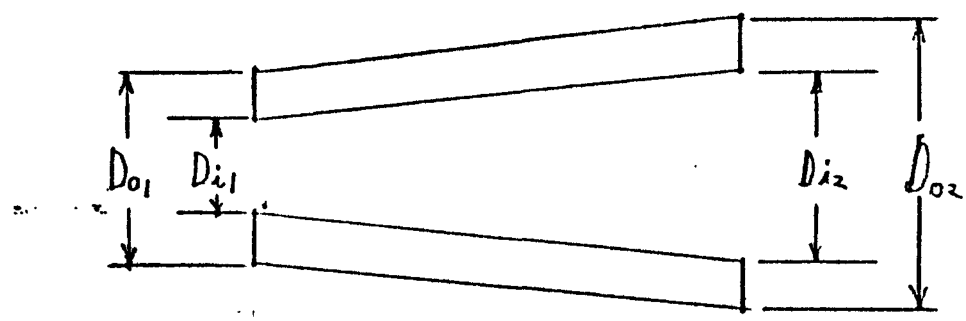
USING Ref. 6.4, p.40.

$$L_e \cong L \frac{G_e}{G} \frac{D_e^4}{3(A-B)}$$

where L_e = equivalent length
L = Length
G_e = equivalent rigidity
G = rigidity
D_e = equivalent diameter

$$A = D_{o1}^3 D_{o2}^3 / [D_{o1}^2 + D_{o1} D_{o2} + D_{o2}^2]$$

$$B = D_{i1}^3 D_{i2}^3 / [D_{i1}^2 + D_{i1} D_{i2} + D_{i2}^2]$$



Let $D_e = 16''$
$G = G_e$
$L = 15''$
$D_{o2} = 21''$
$D_{i2} = 17''$
$D_{o1} = 13''$
$D_{i1} = 9''$

$$A = 13^3\ 21^3 / [13^2 + 13(21) + 21^2] = 23042.3749$$
$$B = 9^3\ 17^3 / [9^2 + (9)(17) + 17^2] = 6848.1396$$

$$L_e = 15\,(1) \frac{16^4}{3(23042.37 - 6848.14)} = 20.23$$

APPENDIX C

TORVIB

COMPUTER RUN

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

TORVIB-R2

- INPUT DATA -

MOM INERTIA LB-IN2	TORSIONAL STIFF LB-IN/RAD E6	OD INCH	LENGTH INCH	OD END 2 IF TAPERED	ID INCH	MOD OF ELAST PSI E6
100874.0	0.000	20.500	16.000	0.000	17.000	22.90
39549.0	0.000	23.500	6.500	0.000	20.000	22.90
7240593.0	0.000	23.500	6.500	0.000	20.000	22.90
32817.0	0.000	20.000	16.000	0.000	17.000	22.90
98723.0	0.000	16.000	20.230	0.000	0.000	22.90
0.0	0.000	12.000	1.500	13.000	9.000	22.90
3150.6	0.000	12.000	16.000	0.000	9.000	22.90
6301.3	0.000	12.000	16.000	0.000	9.000	22.90
6301.3	0.000	12.000	16.000	0.000	9.000	22.90
6301.3	0.000	12.000	16.000	0.000	9.000	22.90
6104.3	0.000	12.000	15.000	0.000	9.000	23.20
5907.5	0.000	12.000	15.000	0.000	9.000	23.80
5907.5	0.000	12.000	15.000	0.000	9.000	24.40
5907.5	0.000	12.000	15.000	0.000	9.000	25.00
5907.5	0.000	12.000	15.000	0.000	9.000	25.60
5907.5	0.000	12.000	15.000	0.000	9.000	26.20
2953.7	0.000	12.000	13.000	-2.250	9.000	26.76
0.0	0.000	12.000	2.000	11.250	0.000	27.06
0.0	0.000	11.250	15.000	0.000	0.000	27.40
0.0	0.000	11.250	20.000	0.000	0.000	27.70
0.0	0.000	10.000	20.000	0.000	0.000	27.70
30374.0	0.000	10.000	11.833	0.000	0.000	27.70
0.0	0.000	12.250	31.500	0.000	0.000	27.70
18000000.0	0.000	0.000	0.000	0.000	0.000	30.00

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

1 NODE FREQUENCY IS 574.16 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.9438E+06	.1000E+01	.9438E+06	.9438E+06	.5031E+10	.1876E-03
2	.1024E+03	.3700E+06	.9998E+00	.3699E+06	.1314E+07	.1929E+11	.6811E-04
3	.1874E+05	.6774E+08	.9997E+00	.6772E+08	.6904E+08	.1929E+11	.3580E-02
4	.8493E+02	.3070E+06	.9962E+00	.3059E+06	.6934E+08	.4133E+10	.1678E-01
5	.2555E+03	.9236E+06	.9794E+00	.9046E+06	.7025E+08	.2196E+10	.3198E-01
6	.8154E+01	.2948E+05	.9474E+00	.2793E+05	.7028E+08	.7661E+09	.9174E-01
7	.1631E+02	.5895E+05	.8557E+00	.5044E+05	.7033E+08	.7661E+09	.9180E-01
8	.1631E+02	.5895E+05	.7639E+00	.4503E+05	.7037E+08	.7661E+09	.9186E-01
9	.1631E+02	.5895E+05	.6720E+00	.3962E+05	.7041E+08	.7661E+09	.9191E-01
10	.1580E+02	.5711E+05	.5801E+00	.3313E+05	.7044E+08	.8278E+09	.8509E-01
11	.1529E+02	.5527E+05	.4950E+00	.2736E+05	.7047E+08	.8493E+09	.8298E-01
12	.1529E+02	.5527E+05	.4120E+00	.2277E+05	.7049E+08	.8707E+09	.8097E-01
13	.1529E+02	.5527E+05	.3310E+00	.1830E+05	.7051E+08	.8921E+09	.7904E-01
14	.1529E+02	.5527E+05	.2520E+00	.1393E+05	.7053E+08	.9135E+09	.7721E-01
15	.1529E+02	.5527E+05	.1748E+00	.9661E+04	.7054E+08	.9349E+09	.7545E-01
16	.7644E+01	.2763E+05	.9935E-01	.2745E+04	.7054E+08	.2071E+09	.3406E+00
17	.7861E+02	.2842E+06	-.2412E+00	-.6855E+05	.7047E+08	.3981E+09	.1770E+00
18	.4658E+05	.1684E+09	-.4182E+00	-.7043E+08	.3650E+05	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

2 NODE FREQUENCY IS 13915.80 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.5544E+09	.1000E+01	.5544E+09	.5544E+09	.5031E+10	.1102E+00
2	.1024E+03	.2174E+09	.8899E+00	.1934E+09	.7478E+09	.1929E+11	.3877E-01
3	.1874E+05	.3979E+11	.8510E+00	.3387E+11	.3461E+11	.1929E+11	.1795E+01
4	.8493E+02	.1804E+09	-.9436E+00	-.1702E+09	.3444E+11	.4133E+10	.8333E+01
5	.2555E+03	.5426E+09	-.9277E+01	-.5033E+10	.2941E+11	.2196E+10	.1339E+02
6	.8154E+01	.1732E+08	-.2267E+02	-.3925E+09	.2902E+11	.7661E+09	.3788E+02
7	.1631E+02	.3463E+08	-.6054E+02	-.2097E+10	.2692E+11	.7661E+09	.3514E+02
8	.1631E+02	.3463E+08	-.9569E+02	-.3314E+10	.2361E+11	.7661E+09	.3082E+02
9	.1631E+02	.3463E+08	-.1265E+03	-.4381E+10	.1923E+11	.7661E+09	.2510E+02
10	.1580E+02	.3355E+08	-.1516E+03	-.5086E+10	.1414E+11	.8278E+09	.1708E+02
11	.1529E+02	.3247E+08	-.1687E+03	-.5476E+10	.8663E+10	.8493E+09	.1020E+02
12	.1529E+02	.3247E+08	-.1789E+03	-.5808E+10	.2856E+10	.8707E+09	.3280E+01
13	.1529E+02	.3247E+08	-.1822E+03	-.5914E+10	-.3058E+10	.8921E+09	-.3428E+01
14	.1529E+02	.3247E+08	-.1787E+03	-.5803E+10	-.8861E+10	.9135E+09	-.9701E+01
15	.1529E+02	.3247E+08	-.1690E+03	-.5488E+10	-.1435E+11	.9349E+09	-.1535E+02
16	.7644E+01	.1623E+08	-.1537E+03	-.2495E+10	-.1684E+11	.2071E+09	-.8133E+02
17	.7861E+02	.1659E+09	-.7235E+02	-.1208E+11	-.2892E+11	.3981E+09	-.7265E+02
18	.4656E+05	.9893E+11	.2923E+00	.2691E+11	-.1038E+08	.0000E+01	.0000E+01

3 NODE FREQUENCY IS 24776.30 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.1757E+10	.1000E+01	.1757E+10	.1757E+10	.5031E+10	.3493E+00
2	.1024E+03	.6890E+09	.6507E+00	.4483E+09	.2206E+10	.1929E+11	.1144E+00
3	.1874E+05	.1261E+12	.5363E+00	.6765E+11	.6986E+11	.1929E+11	.3622E+01
4	.8493E+02	.5717E+09	-.3086E+01	-.1764E+10	.6809E+11	.4133E+10	.1647E+02
5	.2555E+03	.1720E+10	-.1956E+02	-.3364E+11	.3445E+11	.2196E+10	.1569E+02
6	.8154E+01	.5439E+08	-.3525E+02	-.1935E+10	.3252E+11	.7661E+09	.4245E+02
7	.1631E+02	.1098E+09	-.7769E+02	-.8529E+10	.2399E+11	.7661E+09	.3131E+02
8	.1631E+02	.1098E+09	-.1090E+03	-.1197E+11	.1202E+11	.7661E+09	.1569E+02
9	.1631E+02	.1098E+09	-.1247E+03	-.1369E+11	-.1668E+10	.7661E+09	-.2178E+01
10	.1580E+02	.1063E+09	-.1225E+03	-.1303E+11	-.1470E+11	.8278E+09	-.1775E+02
11	.1529E+02	.1029E+09	-.1048E+03	-.1078E+11	-.2548E+11	.8493E+09	-.3000E+02
12	.1529E+02	.1029E+09	-.7476E+02	-.7694E+10	-.3317E+11	.8707E+09	-.3810E+02
13	.1529E+02	.1029E+09	-.3666E+02	-.3773E+10	-.3695E+11	.8921E+09	-.4142E+02
14	.1529E+02	.1029E+09	.4759E+01	.4898E+09	-.3646E+11	.9135E+09	-.3991E+02
15	.1529E+02	.1029E+09	.4467E+02	.4597E+10	-.3186E+11	.9349E+09	-.3408E+02
16	.7644E+01	.5146E+08	.7875E+02	.4052E+10	-.2781E+11	.2071E+09	-.1343E+03
17	.7861E+02	.5292E+09	.2130E+03	.1127E+12	.8491E+11	.3981E+09	.2133E+03
18	.4658E+05	.3136E+12	-.2703E+00	-.8491E+11	.6633E+06	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

4 NODE FREQUENCY IS 32339.57 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.2994E+10	.1000E+01	.2994E+10	.2994E+10	.5031E+10	.5951E+00
2	.1024E+03	.1174E+10	.4049E+00	.4752E+09	.3469E+10	.1929E+11	.1799E+00
3	.1874E+05	.2149E+12	.2250E+00	.4835E+11	.5182E+11	.1929E+11	.2687E+01
4	.8493E-02	.9741E+09	-.2462E+01	-.2398E+10	.4942E+11	.4133E+10	.1196E+02
5	.2555E+03	.2930E+10	-.1442E+02	-.4225E+11	.7170E+10	.2196E+10	.3264E+01
6	.8154E+01	.9352E+08	-.1768E+02	-.1654E+10	.5516E+10	.7661E+09	.7200E+01
7	.1631E+02	.1870E+09	-.2488E+02	-.4654E+10	.8620E+09	.7661E+09	.1125E+01
8	.1631E+02	.1870E+09	-.2601E+02	-.4864E+10	-.4002E+10	.7661E+09	-.5225E+01
9	.1631E+02	.1870E+09	-.2078E+02	-.3887E+10	-.7890E+10	.7661E+09	-.1030E+02
10	.1580E+02	.1812E+09	-.1048E+02	-.1900E+10	-.9789E+10	.8278E+09	-.1183E+02
11	.1529E+02	.1753E+09	.1340E+01	.2350E+09	-.9554E+10	.8493E+09	-.1125E+02
12	.1529E+02	.1753E+09	.1259E+02	.2208E+10	-.7347E+10	.8707E+09	-.8438E+01
13	.1529E+02	.1753E+09	.2103E+02	.3687E+10	-.3659E+10	.8921E+09	-.4102E+01
14	.1529E+02	.1753E+09	.2513E+02	.4407E+10	.7472E+09	.9135E+09	.8180E+00
15	.1529E+02	.1753E+09	.2431E+02	.4263E+10	.5010E+10	.9349E+09	.5359E+01
16	.7644E+01	.8767E+08	.1895E+02	.1662E+10	.6672E+10	.2071E+09	.3221E+02
17	.7861E-02	.9016E+09	-.1326E+02	-.1196E+11	-.5283E+10	.3981E+09	-.1327E+02
18	.4658E+05	.5343E+12	.9912E-02	.5296E+10	.1256E+08	.0000E+01	.0000E+01

5 NODE FREQUENCY IS 37281.75 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.3979E+10	.1000E+01	.3979E+10	.3979E+10	.5031E+10	.7910E+00
2	.1024E+03	.1560E+10	.2090E+00	.3261E+09	.4305E+10	.1929E+11	.2232E+00
3	.1874E+05	.2856E+12	-.1418E-01	-.4050E+10	.2557E+09	.1929E+11	.1326E-01
4	.8493E+02	.1295E+10	-.2743E-01	-.3551E+08	.2201E+09	.4133E+10	.5326E-01
5	.2555E+03	.3894E+10	-.8070E-01	-.3143E+09	-.9411E+08	.2196E+10	-.4285E-01
6	.8154E+01	.1243E+09	-.3785E-01	-.4704E+07	-.9882E+08	.7661E+09	-.1290E+00
7	.1631E+02	.2486E+09	.9115E-01	.2266E+08	-.7616E+08	.7661E+09	-.9942E-01
8	.1631E+02	.2486E+09	.1906E+00	.4737E+08	-.2879E+08	.7661E+09	-.3759E-01
9	.1631E+02	.2486E+09	.2282E+00	.5671E+08	.2792E+08	.7661E+09	.3644E-01
10	.1580E+02	.2408E+09	.1917E+00	.4616E+08	.7408E+08	.8278E+09	.8949E-01
11	.1529E+02	.2330E+09	.1022E+00	.2382E+08	.9790E+08	.8493E+09	.1153E+00
12	.1529E+02	.2330E+09	-.1306E-01	-.3043E+07	.9486E+08	.8707E+09	.1090E+00
13	.1529E+02	.2330E+09	-.1220E+00	-.2843E+08	.6643E+08	.8921E+09	.7446E-01
14	.1529E+02	.2330E+09	-.1965E+00	-.4578E+08	.2064E+08	.9135E+09	.2260E-01
15	.1529E-02	.2330E+09	-.2191E+00	-.5105E+08	-.3041E+08	.9349E+09	-.3252E-01
16	.7644E+01	.1165E+09	-.1865E+00	-.2174E+08	-.5214E+08	.2071E+09	-.2518E+00
17	.7861E+02	.1198E+10	.6521E-01	.7814E+08	.2599E+08	.3981E+09	.6529E-01
18	.4658E+05	.7100E+12	-.7782E-04	-.5525E+08	-.2926E+08	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

6 NODE FREQUENCY IS 38476.09 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.4238E+10	.1000E+01	.4238E+10	.4238E+10	.5031E+10	.8424E+00
2	.1024E+03	.1652E+10	.1576E+00	.2618E+09	.4500E+10	.1929E+11	.2333E+00
3	.1874E+05	.3042E+12	-.7576E-01	-.2305E+11	-.1855E+11	.1929E+11	-.9617E+00
4	.8493E+02	.1379E+10	.8859E+00	.1222E+10	-.1733E+11	.4133E+10	-.4192E+01
5	.2555E+03	.4148E+10	.5078E+01	.2106E+11	.3736E+10	.2196E+10	.1701E+01
6	.8154E+01	.1324E+09	.3377E+01	.4470E+09	.4183E+10	.7661E+09	.5460E+01
7	.1631E+02	.2647E+09	-.2083E+01	-.5516E+09	.3631E+10	.7661E+09	.4740E+01
8	.1631E+02	.2647E+09	-.6824E+01	-.1807E+10	.1825E+10	.7661E+09	.2382E+01
9	.1631E+02	.2647E+09	-.9206E+01	-.2437E+10	-.6124E+09	.7661E+09	-.7994E+00
10	.1580E+02	.2565E+09	-.8407E+01	-.2156E+10	-.2768E+10	.8278E+09	-.3344E+01
11	.1529E+02	.2482E+09	-.5062E+01	-.1256E+10	-.4025E+10	.8493E+09	-.4739E+01
12	.1529E+02	.2482E+09	-.3230E+00	-.8016E+08	-.4105E+10	.8707E+09	-.4715E+01
13	.1529E+02	.2482E+09	.4392E+01	.1090E+10	-.3015E+10	.8921E+09	-.3380E+01
14	.1529E+02	.2482E+09	.7772E+01	.1929E+10	-.1086E+10	.9135E+09	-.1189E+01
15	.1529E+02	.2482E+09	.8961E+01	.2224E+10	.1138E+10	.9349E+09	.1217E+01
16	.7644E+01	.1241E+09	.7743E+01	.9609E+09	.2099E+10	.2071E+09	.1013E+02
17	.7861E+02	.1276E+10	-.2391E+01	-.3051E+10	-.9523E+09	.3981E+09	-.2392E+01
18	.4658E+05	.7563E+12	.1038E-02	.7853E+09	-.1671E+09	.0000E+01	.0000E+01

7 NODE FREQUENCY IS 54377.98 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.8465E+10	.1000E+01	.8465E+10	.8465E+10	.5031E+10	.1683E+01
2	.1024E+03	.3319E+10	-.6827E+00	-.2266E+10	.6200E+10	.1929E+11	.3214E+00
3	.1874E+05	.6076E+12	-.1004E+01	-.6101E+12	-.6039E+12	.1929E+11	-.3131E+02
4	.8493E+02	.2754E+10	.3031E+02	.8347E+11	-.5205E+12	.4133E+10	-.1259E+03
5	.2555E+03	.8285E+10	.1562E+03	.1294E+13	.7739E+12	.2196E+10	.3524E+03
6	.8154E+01	.2644E+09	-.1961E+03	-.5185E+11	.7221E+12	.7661E+09	.9425E+03
7	.1631E+02	.5288E+09	-.1139E+04	-.6021E+12	.1199E+12	.7661E+09	.1565E+03
8	.1631E+02	.5288E+09	-.1295E+04	-.6849E+12	-.5650E+12	.7661E+09	-.7375E+03
9	.1631E+02	.5288E+09	-.5577E+03	-.2949E+12	-.8599E+12	.7661E+09	-.1122E+04
10	.1580E+02	.5123E+09	.5648E+03	.2893E+12	-.5706E+12	.8278E+09	-.6892E+03
11	.1529E+02	.4958E+09	.1254E+04	.6217E+12	.5113E+11	.8493E+09	.6021E+02
12	.1529E+02	.4958E+09	.1194E+04	.5918E+12	.6430E+12	.8707E+09	.7385E+03
13	.1529E+02	.4958E+09	.4553E+03	.2257E+12	.8687E+12	.8921E+09	.9738E+03
14	.1529E+02	.4958E+09	-.5185E+03	-.2570E+12	.6117E+12	.9135E+09	.6696E+03
15	.1529E+02	.4958E-09	-.1188E+04	-.5890E+12	.2266E+11	.9349E+09	.2424E+02
16	.7644E+01	.2479E+09	-.1212E+04	-.3005E+12	-.2778E+12	.2071E+09	-.1342E+04
17	.7861E+02	.2549E+10	.1292E+03	.3293E+12	.5143E+11	.3981E+09	.1292E+03
18	.4658E+05	.1511E+13	-.1471E-01	-.2221E+11	.2922E+11	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE	PURCHASER	DATE
LARGE SCALE-HOT LEG	SODIUM PUMP	24SEPT 76

8 NODE FREQUENCY IS 72670.64 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.1512E+11	.1000E+01	.1512E+11	.1512E+11	.5031E+10	.3005E+01
2	.1024E+03	.5928E+10	-.2005E+01	-.1189E+11	.3233E+10	.1929E+11	.1676E+00
3	.1874E+05	.1085E+13	-.2173E+01	-.2358E+13	-.2355E+13	.1929E+11	-.1221E+03
4	.8493E+02	.4919E+10	.1199E+03	.5898E+12	-.1765E+13	.4133E+10	-.4270E+03
5	.2555E+03	.1480E+11	.5469E+03	.8093E+13	.6328E+13	.2196E+10	.2881E+04
6	.8154E+01	.4722E+09	-.2334E+04	-.1102E+13	.5226E+13	.7661E+09	.6821E+04
7	.1631E+02	.9444E+09	-.9155E+04	-.8647E+13	-.3421E+13	.7661E+09	-.4466E+04
8	.1631E+02	.9444E+09	-.4690E+04	-.4429E+13	-.7850E+13	.7661E+09	-.1025E+05
9	.1631E+02	.9444E+09	.5558E+04	.5249E+13	-.2601E+13	.7661E+09	-.3396E+04
10	.1580E+02	.9149E+09	.8953E+04	.8191E+13	.5590E+13	.8278E+09	.6752E+04
11	.1529E+02	.8854E+09	.2201E+04	.1949E+13	.7539E+13	.8493E+09	.8877E+04
12	.1529E+02	.8854E+09	-.6676E+04	-.5911E+13	.1628E+13	.8707E+09	.1869E+04
13	.1529E+02	.8854E+09	-.6545E+04	-.7566E+13	-.5938E+13	.8921E+09	-.6657E+04
14	.1529E+02	.8854E+09	-.1888E+04	-.1672E+13	-.7610E+13	.9135E+09	-.8331E+04
15	.1529E+02	.8854E+09	.6443E+04	.5705E+13	-.1906E+13	.9349E+09	-.2039E+04
16	.7644E+01	.4427E+09	.8482E+04	.3755E+13	.1849E+13	.2071E+09	.8927E+04
17	.7861E+02	.4552E+10	-.4450E+03	-.2026E+13	-.1772E+12	.3981E+09	-.4451E+03
18	.4658E+05	.2698E+13	.2508E-01	.6765E+11	-.1095E+12	.0000E+01	.0000E+01

9 NODE FREQUENCY IS 89772.11 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.2611E+03	.2307E+11	.1000E+01	.2307E+11	.2307E+11	.5031E+10	.4586E+01
2	.1024E+03	.9046E+10	-.3586E+01	-.3244E+11	-.9366E+10	.1929E+11	-.4856E+00
3	.1874E+05	.1656E+13	-.3100E+01	-.5135E+13	-.5144E+13	.1929E+11	-.2667E+03
4	.8493E+02	.7506E+10	.2636E+03	.1979E+13	-.3165E+13	.4133E+10	-.7658E+03
5	.2555E+03	.2258E+11	.1029E+04	.2324E+14	.2008E+14	.2196E+10	.9142E+04
6	.8154E+01	.7206E+09	-.8112E+04	-.5846E+13	.1423E+14	.7661E+09	.1858E+05
7	.1631E+02	.1441E+10	-.2569E+05	-.3847E+14	-.2424E+14	.7661E+09	-.3164E+05
8	.1631E+02	.1441E+10	.4945E+04	.7127E+13	-.1711E+14	.7661E+09	-.2233E+05
9	.1631E+02	.1441E+10	.2728E+05	.3932E+14	.2221E+14	.7661E+09	.2899E+05
10	.1530E+02	.1396E+10	-.1708E+04	-.2385E+13	.1982E+14	.8278E+09	.2394E+05
11	.1529E+02	.1351E+10	-.2565E+05	-.3466E+14	-.1484E+14	.8493E+09	-.1747E+05
12	.1529E+02	.1351E+10	-.8180E+04	-.1105E+14	-.2589E+14	.8707E+09	-.2974E+05
13	.1529E+02	.1351E+10	.2156E+05	.2913E+14	.3236E+13	.8921E+09	.3628E+04
14	.1529E+02	.1351E+10	.1793E+05	.2422E+14	.2746E+14	.9135E+09	.3006E+05
15	.1529E+02	.1351E+10	-.1213E+05	-.1639E+14	.1107E+14	.9349E+09	.1184E+05
16	.7644E+01	.6756E+09	-.2397E+05	-.1619E+14	-.5127E+13	.2071E+09	-.2475E+05
17	.7861E+02	.6947E+10	.7828E+03	.5438E+13	.3116E+12	.3981E+09	.7826E+03
18	.4658E+05	.4117E+13	.1828E+00	.7525E+12	.1064E+13	.0000E+01	.0000E+01

COMPUTATION HAS BEEN TERMINATED AT UPPER LIMIT OF 100000. RPM

BYRON JACKSON REPORT TCF-1019-DYN
MODAL CHARACTERISTICS
OF
LARGE SCALE BREEDER REACTOR HOT LEG
MECHANICAL PUMP CONCEPTUAL DESIGN

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
AEC CONTRACT AT 67002

PREPARED BY Robert Kubo DATE 27 SEP 76

CHECKED BY Yoshio Usui DATE 29 SEP 76

APPROVED BY Kent A. Huber DATE 30 SEP 76

September 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

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APPENDIX D - COMPUTER MODEL OF ENTIRE PUMP ASSEMBLY ON COMPLIANT FOUNDATION

APPENDIX E - MODE SHAPE PLOTS

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1.0 SCOPE

The purpose of this study is to identify potential areas of concern in the large scale LSBRP hot leg pump conceptual design of Reference 2.1. Specifically, the reported analyses determine whether lateral modal characteristics of the rotating element and of the assembled pump unit meet the requirements of Reference 2.2

Conclusions are based on estimated values of stiffnesses for the foundation support arrangement. The results of a concurrent study to determine more exact values for these parameters are presented, and their effect on the dynamic response is discussed.

Details of the foundation, attached piping, and driver were not available. Consequently, these analyses are based either on "typical" dimensions and weights, approximate the effect of the component with conservative mass and spring data, or use information available from the Clinch River Breeder Reactor hot leg pump. (Reference 2.13)

As required, the analysis investigates design modifications required to achieve the desired natural frequencies.

2.0 REFERENCES

2.1 Byron Jackson Engineering Drawing SK-(TCF-1019-DYN)-01

2.2 General Electric Specification 23A2126, Revision 0.

2.3 "Structural Response of CRBRP Primary and Intermediate Prototype Pumps", J. Chrostowski and L. T. Lee, J. H. Wiggins Company, Technical Report No. 75-1225-3.

2.4 Nuclear Systems Materials Handbook, Volume 1, "Design Data", Hanford Engineering Development Lab. TID 26666.

2.5 "SAP IV, A Structural Analysis Program for Static and Dynamic Response of Linear Systems", Earthquake Engineering Research Center, College of Engineering, University of California, Report No. EERC 73-11.

2.6 "Analysis of the Deformation Behavior of a Pump Tank Structure", Universal Analysis, Inc. July 15, 1974

2.7 "Lateral Modal Characteristics of CRBRP Pump Assembly", Y. Usui, Byron Jackson Report No. TCF-TM-1025.

2.8 "Design Analysis of Shafts and Beams" by R. B. Hopkins, McGraw-Hill, 1970.

2.9 MTI letter report dated 5 December 1975 to J. Ogg "Preliminary Results of Analyses of 24" x 17" and 16" x 12" Bearings".

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2.10 "Lateral Vibration Analysis - 33 x 33 x 38 DFSS Primary Coolant Recirculation Pump", Gerald Helt, Byron Jackson Report No. TCF-1013-DYN.

2.11 Byron Jackson Engineering Drawing SK-(TCF-1019-DYN)-01 (Collinear Beam Model Worksheet)

2.12 "Volume and CG Equations", E. W. Jenkins, Knolls Atomic Power Laboratory, General Electric Co., Schenectady, N.Y.

2.13 Intercompany Communication, J. Ogg (BRPD) to K. Huber

3.0 DESCRIPTION OF LSBRP PUMP

The pump is comprised of six major subassemblies, the motor rotor, the pump shaft, the motor housing, the motor mount, the internal structure (includes radiation shield, hydrostatic bearing and hydraulic assembly), and the outer tank. Figure 3.1(a) identifies these subassemblies.

The outer tank is mounted to the foundation with a clamp ring arrangement. The motor, motor mount, and pump internals are assembled and lowered into place as a complete unit. Attachment between the two structures consists of a series of jack-screws used to obtain a prescribed vertical placement of the pump internal assembly and a series of hold-down bolts.

For analysis purposes, the various components, including the hydraulic assembly, are represented by collinear shear beams with thin walled circular cross sections. Figure 3.1(b) is a simplified description of the collinear beam model used to represent the LSBRP pump assembly for the dynamic analysis.

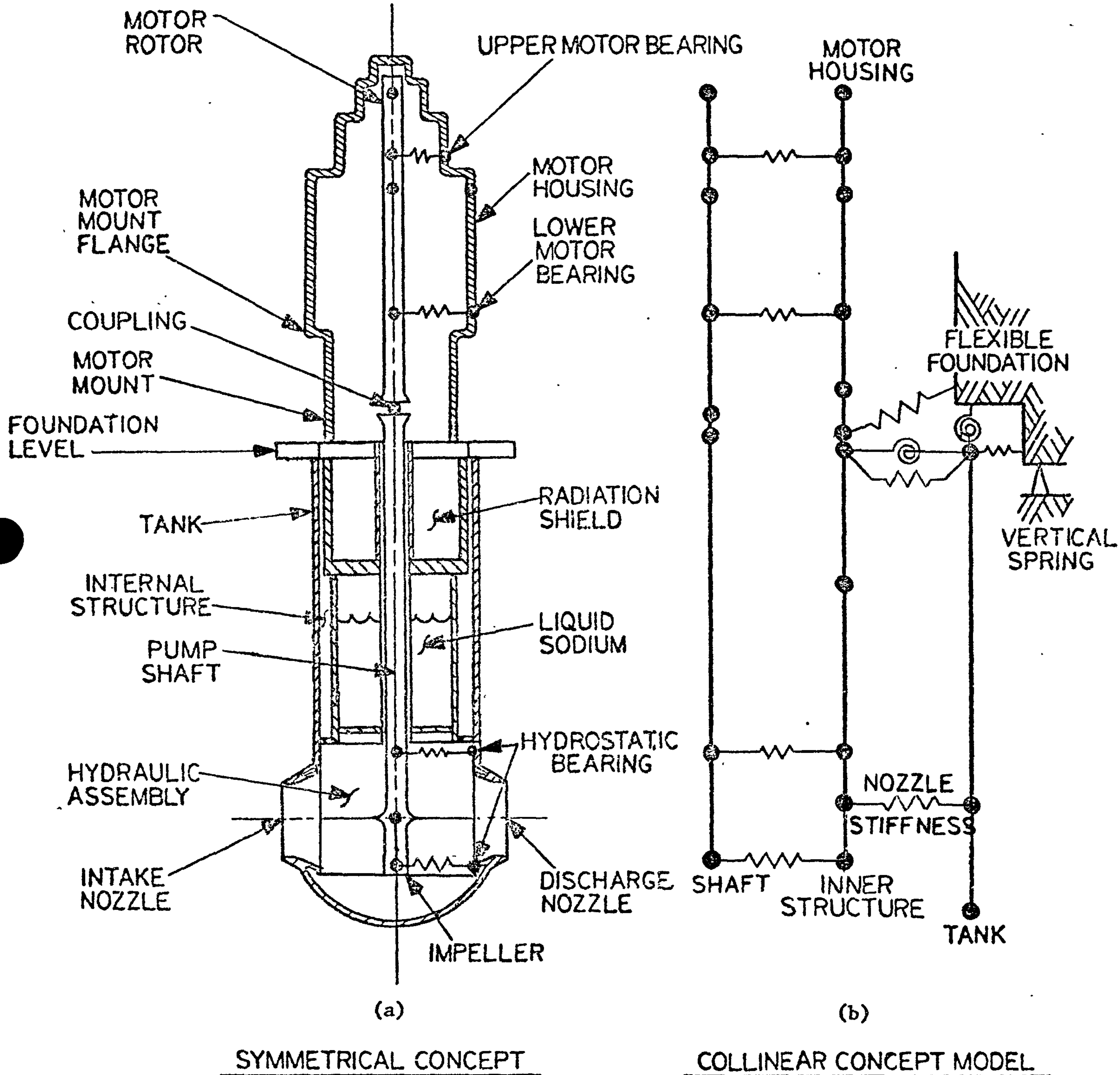


Figure 3.1 Idealization of the LSBRP Pump

4.0 STRUCTURAL MODELING

4.1 General Modeling Considerations

The modal characteristics of the LSBRP pump were determined using the colinear shear beam model shown in Figure 4.1. The analysis uses the SAP IV Structural Analysis Program, Reference 2.5, and accounts for the following effects:

- Shear deformations,
- Temperature dependent material properties
- Mass of entrained sodium, and
- Mass of attached piping.

The basic assumptions for the analysis are:

- The pump structure is axisymmetric
- Tapered portions of the structure are represented by graduated cylindrical sections,
- Only planar motion is considered (i.e., torsional motion is not allowed),
- Attached piping has zero stiffness,
- The steel balls comprising the radiation shield behave as a lumped mass,
- The shaft bearings can be treated as linear elastic springs,
- The foundation can be represented by a set of linear springs,
- No shell deformation occurs,
- Bolted joints in the shaft and inner structure components are rigid.

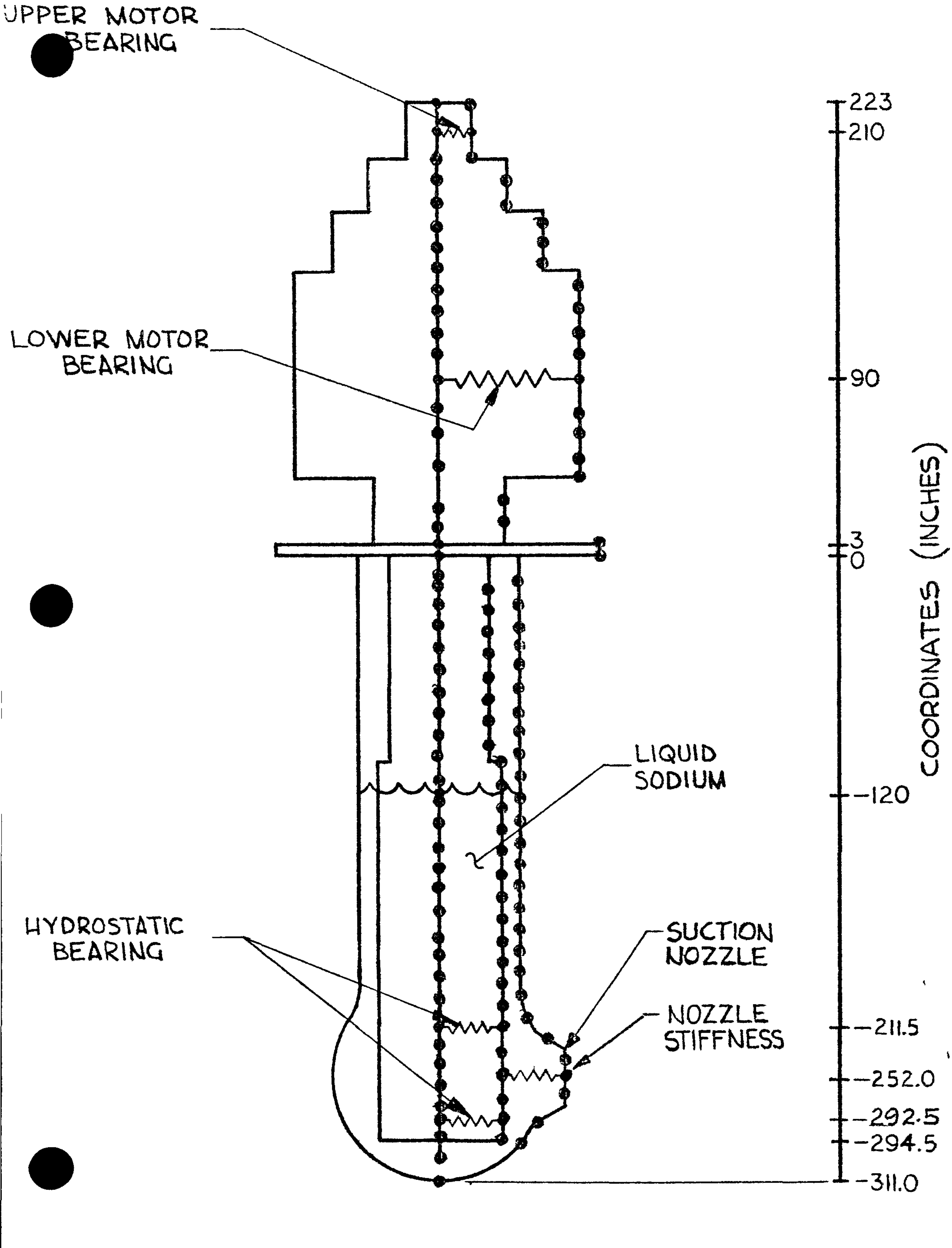


FIGURE 4.1 COLINEAR SHEAR BEAM MODEL

- The mass moment of inertia can be neglected.
- Gyroscopic effects in the rotating element can be neglected, and,
- Dynamic coupling due to the sodium between the shaft, inner structure, and tank is neglected.

The effect of lateral sloshing of the liquid sodium was not considered in the present analysis. However, Reference 2.3 presents an analysis of this effect using an equivalent mechanical system for a prototype pump of similar design. The results predict the slosh frequency to occur well below the pump operating range.

4.2 Pump Components

Figure 4.1 is an illustration of the lumped masses (nodes) used to represent the pump structure. The node locations were selected to provide a proper mass and stiffness representation along the length of the pump shaft, inner structure, and tank. The nodes were also placed to provide enough elements to define a linear temperature distribution in material located above the liquid sodium level. The origin of the coordinate system used to define the nodes is located at an elevation corresponding to the bottom of the outer tank mounting flange.

Due to the collinear model, there is no coupling between the lateral and vertical degrees of freedom. Therefore the analysis of the lateral pump modal characteristics is considered separate from the vertical by defining only a lateral displacement and an in-plane rotation at each node.

The mass and stiffness properties are calculated for each element defined between the nodes shown in Figure 4.1. For circular cylinders with inside and outside diameters of D_i and D_o, respectively, these calculations are performed by the proprietary Byron Jackson computer program INSAP using the following equations from Reference 2.8:

1. Cross Sectional Area, A;

$$A = \frac{\pi}{4} (D_o^2 - D_i^2)$$

2. Shear coefficient, K_s:

$$K_s = 6(1+\nu)[1 + (D_i/D_o)^2]^2 / \{(7+6\nu)[1 + D_i/D_o)^2]^2 + (20 + 12\nu)(D_i/D_o)^2\}$$

where Poisson's ratio, ν, is assumed to be 0.3.

3. Shear Area, A_s;

   $A_s = AK_s$

4. Weight, W;

   W = (Length * density * A) + concentrated load

5. Modulus of Elasticity, E;

   $E = E(T_{ave})$

   where T_{ave} is temperature at center of element.

Unfortunately, the program cannot process complex geometries and the shear beam properties for these components must be calculated manually. These calculations may be found in Appendix A.

The modulus of elasticity is determined as a function of temperature from Reference 2.4. A computer printout of the complete model may be found in Appendix D.

4.2.1 Pump Shaft

With the exception of the impeller and hydrostatic bearings, the mass and stiffness properties of the pump shaft can be calculated by program INSAP.

The impeller was modeled as a concentrated load acting on a shaft section of 30 inch O.D. and 27 inch I.D.

The total weight was found to be 10,000 lbs. This value was increased by 719 lbs to account for the entrained sodium. The modeled shaft section contributes 1,026.2 lbs of this total weight. Thus, the equivalent concentrated load acting at the impeller centerline is 8,514.8 lbs.

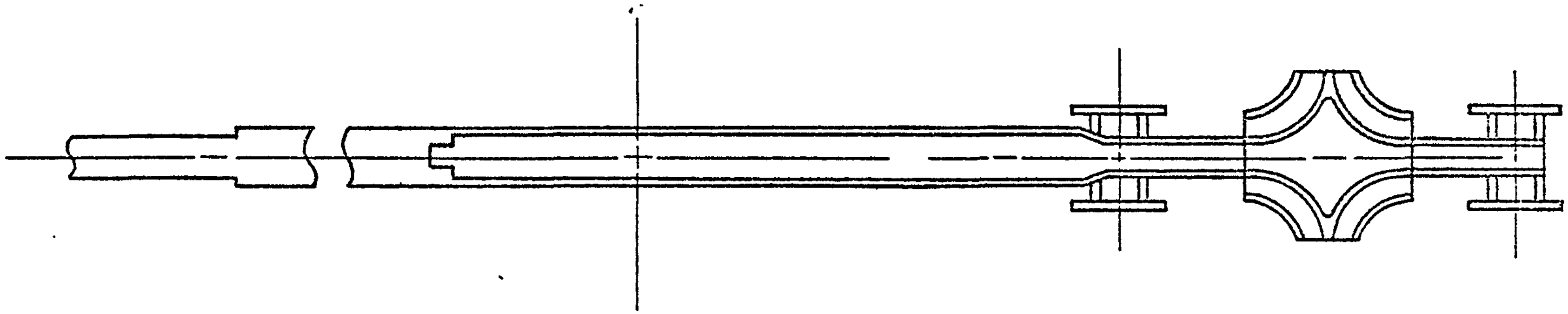
Figure 4.2 presents the parameters of the shear beams used to represent the pump shaft, impellers, and hydrostatic bearings.

4.2.2 Hydraulic Assembly

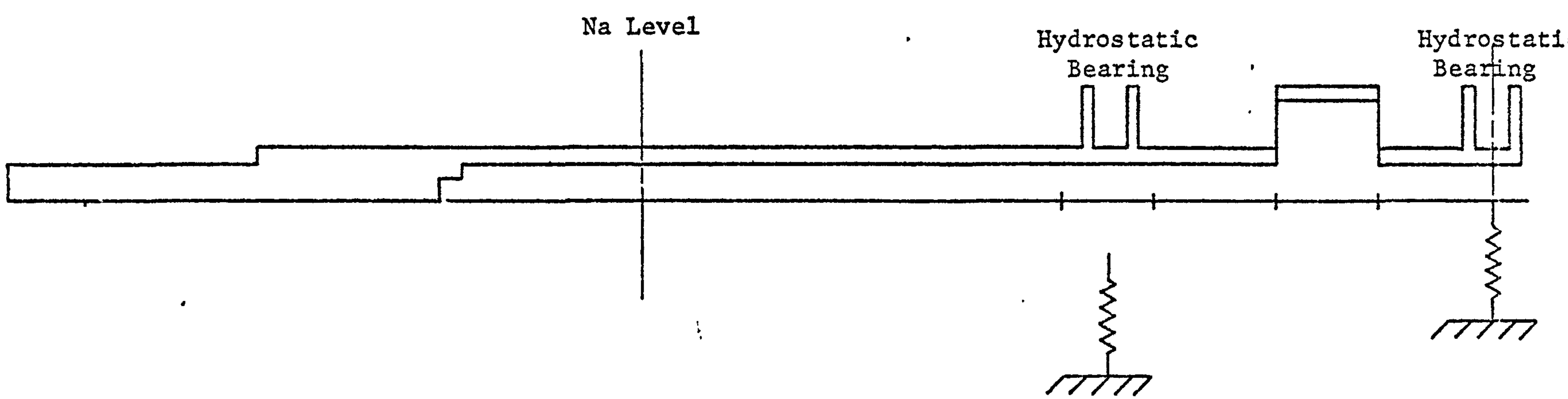
The complex geometry of the hydraulic assembly was represented as a uniform cylindrical section with an inside diameter of 101 inches and a constant thickness of 2 inches. (Reference 2.1). The weights in this section were proportioned up from a geometrically similar pump (Clinch River Breeder Reactor Hot Leg Pump), See Appendix A) and added to the model as concentrated loads. This simplified shear beam model is consistent with the method of construction. Also, the proper distribution of mass at the tip of a cantilever type structure and the stiffness characteristics at the base are of primary interest in modeling. This is because during vibrations most of the strain energy ($U = \frac{1}{2}k\Delta^2$) will be stored at the base of the structure while the tip will contribute the largest portion of the kinetic energy ($K.E. = \frac{1}{2}Mv^2$) in the system.

4.2.3 Internal Support Structure

The structural members supporting the hydraulic assembly exhibit, an axis of symmetry and the equivalent shear beam properties may be calculated using the INSAP program.



Pump Shaft
Figure 4.2(a)



Beam representaion of pump shaft
FIGURE 4.2 (b)

In the calculation the weight of the liquid sodium between the shaft and the internal structure was added to the metal weight of the structure. An additional weight was also added to account for the shot contained in the radiation shield plug. The density of the shot was taken to be 278 lbs. per cubic foot (0.161 lbs. per cubic inch).

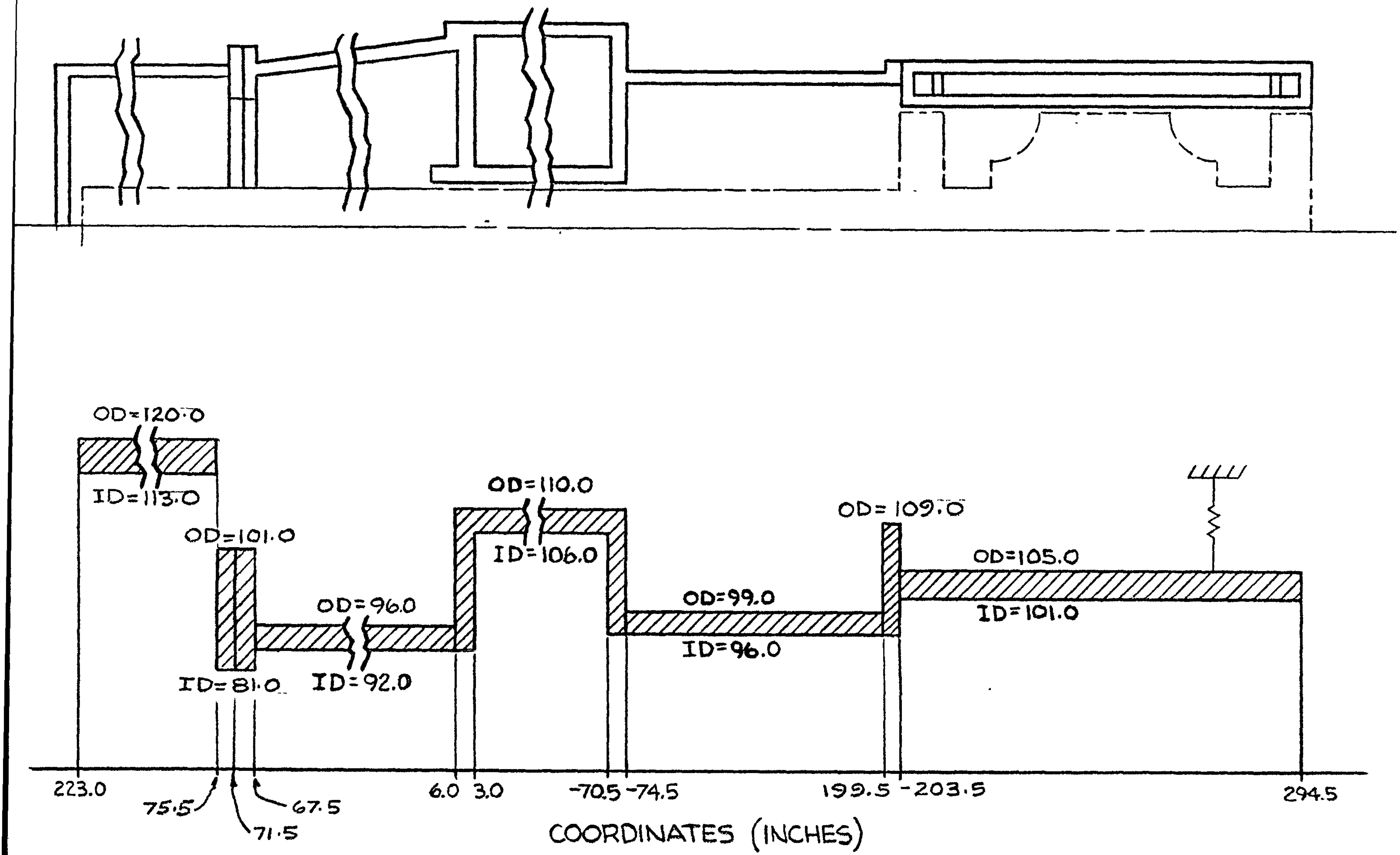
Figure 4.3 shows the equivalent beam properties used to represent the hydraulic assembly and internal support structures for the modal analysis.

4.2.4 Outer Tank

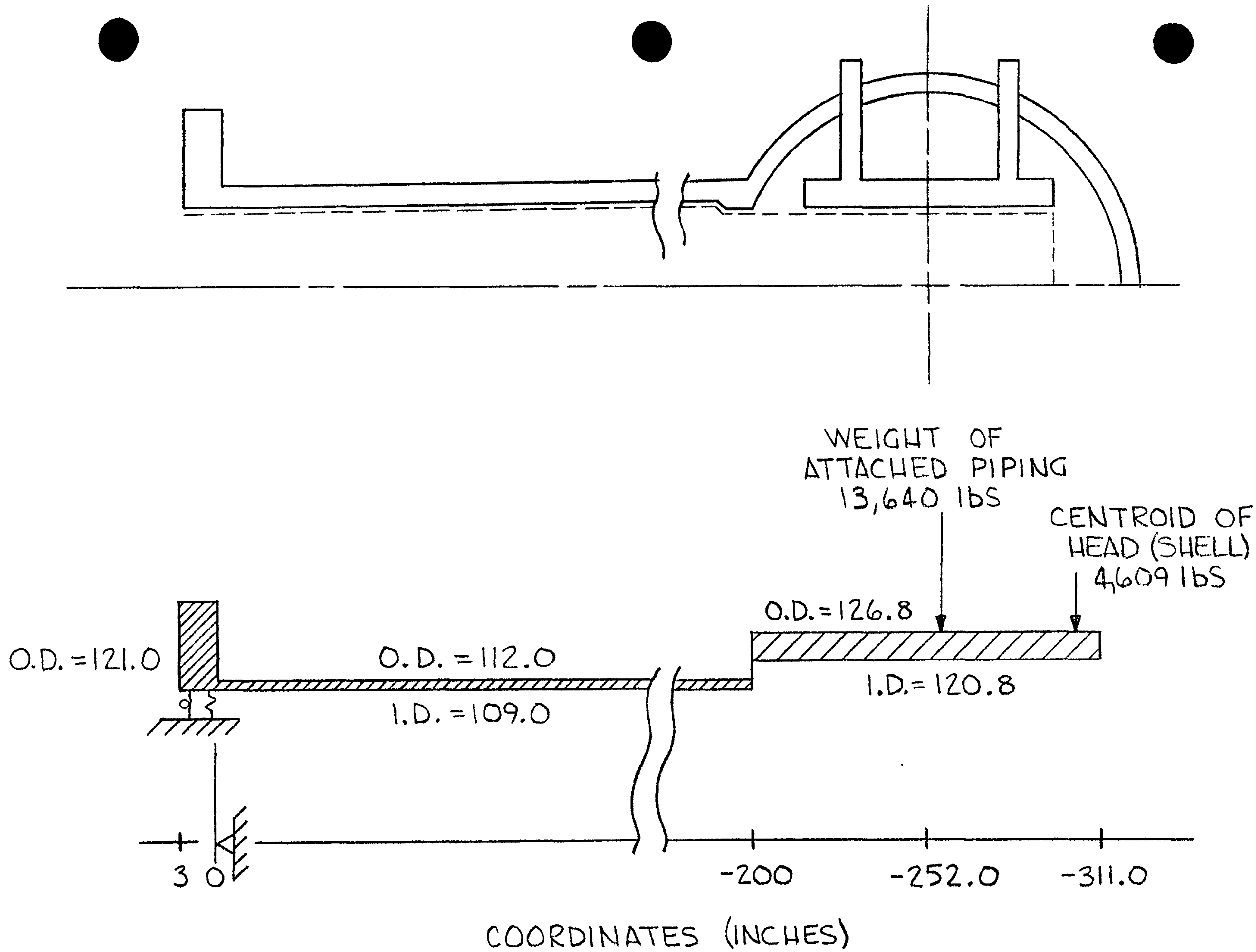
Figure 4.4 shows the equivalent shear beam model used to represent the outer tank. It is the longest of the three components which extend below the foundation. Consequently, it determines the minimum elevation of the collinear beam model.

The spherical portion at the bottom of the tank is represented by a cylindrical section with wall thickness of 3".

The weight of sodium between the internal support structure and the outer tank is assumed to be added to the weight of the outer tank. A similar assumption is made in the hydraulic assembly region.



INNER STRUCTURE FIG.4.3



OUTER TANK FIGURE 4.4

4.2.5 Motor Mount

The motor mount was modeled as a cylindrical section without access windows. The actual structure will thus have a smaller moment of inertia for the same geometry. This representation is considered adequate since the only load acting on the structure is due to the driver weight, and the geometry can be readily modified in the Detailed Design Phase to provide the required stiffness.

The equivalent beam properties for the motor mount are shown in Figure 4.3.

4.3.2 Motor Housing

Detailed engineering drawings of the motor housing were not available at the time the modal analysis was performed. Consequently, a mass-elastic model was developed based on the following criteria:

1. The fundamental lateral mode of the motor housing was to be greater than 30 Hz (on a fixed base).

2. The estimated weight of approximately 97,000 pounds was to be distributed uniformly.

3. The overall geometric dimensions of length and diameter were 12.625 feet and 10 feet, respectively.

It was found that the stiffness of the motor housing could be represented by a shear beam of 120 inches outside diameter and 3.5 inch thickness. These parameters were used in the present analysis.

4.3 Motor Components

The pump driver will be a 10,000 horsepower unit running at 600 RPM. The data used to describe the motor rotor and frame is typical motor data (either estimated or average) from data of other motors listed in Table A (Appendix A).

4.3.1 Motor Shaft

Detail drawings of the motor rotor were not available. Consequently a "typical" motor was constructed for analysis purposes as stated above, based on estimates of significant rotor characteristics. These data are:

Rotor weight	28 kips
Coupling shaft diameter	12.25 inches
Bearing span	10 feet
Natural frequency	38.6 cps

An equivalent diameter was calculated to represent the stiffness of the length of shafting beneath the motor laminations. This was accomplished by factoring the 12.25 inch coupling shaft diameter by the ratio of equivalent shaft diameter to coupling shaft diameter obtained from an 8000 horsepower motor. (See Reference 2.10). The calculated equivalent diameter of 17 inches is shown in Figure 4.2 which illustrates the driver constructed for the analysis.

4.4 Attached Piping

The entire pump structure is connected by piping to the rest of the reactor system at the suction and discharge nozzles. The presence of this piping will tend to dynamically couple the motion of the pump in three dimensions. There may also be significant dynamic coupling between the piping and the pump if piping frequencies are in the range of pump frequencies. For the present analysis, however, only an effective piping mass of 6,820 pounds was included at each nozzle location. Justification of this approach is given in Reference 2.3.

4.5 Boundary Conditions

4.5.1 Shaft

The pump and motor shafts are connected with a rigid coupling into a single rotating element. This "shaft" is supported by two oil lubricated journal bearings in the motor, and two hydrostatic sodium bearings on each side of the impeller.

These bearings were assumed to have the same stiffnesses as provided by General Electric for the CRBRP project motor which was 5000 horsepower and 1126 RPM. The stiffnesses are:

$$K(\text{upper motor}) = 2.0 \times 10^6 \text{ lbs/in}$$

$$K(\text{lower motor}) = 2.0 \times 10^6 \text{ lbs/in}$$

Hydrostatic bearing stiffness parameters were obtained from Reference 2.9 which determines the stiffness range to be $0.74 \times 10^6 \leq K \leq 1.35 \times 10^6$ lbs/in. These calculations were based on a bearing with 24" OD and 17" length and with suction pressure at both ends. Thus these data are directly applicable to the hydrostatic bearing in the conceptual design and a mean value of $K_{(\text{lower hydrostatic})} = 1.00 \times 10^6$ lbs/in could be used. However, for conservatism, two hydrostatic bearings were assumed to be only 50 percent efficient and the stiffness was assumed to be

$$K_{(\text{upper hydrostatic})} = 0.50 \times 10^6 \text{ lbs/in}$$

$$K_{(\text{lower hydrostatic})} = 0.50 \times 10^6 \text{ lbs/in}$$

All bearing stiffnesses were assumed to be independent of journal deflection i.e., the values were held constant throughout the analysis.

4.5.2 Inner Structure

The "inner structure" beam is comprised of the hydraulic assembly, internal support structure, motor mount, and motor housing. It is connected to the outer tank at the foundation by a jack-screw arrangement which permits raising and lowering of the structure to control leakage at the nozzle locations. A lateral stiffness, and a rotational stiffness was assumed at this location between inner structure and outer tank (See Figure 4.5).

This analysis also assumes that a rigid connection exists at the nozzle centerline elevation due to internal loading misalignments.

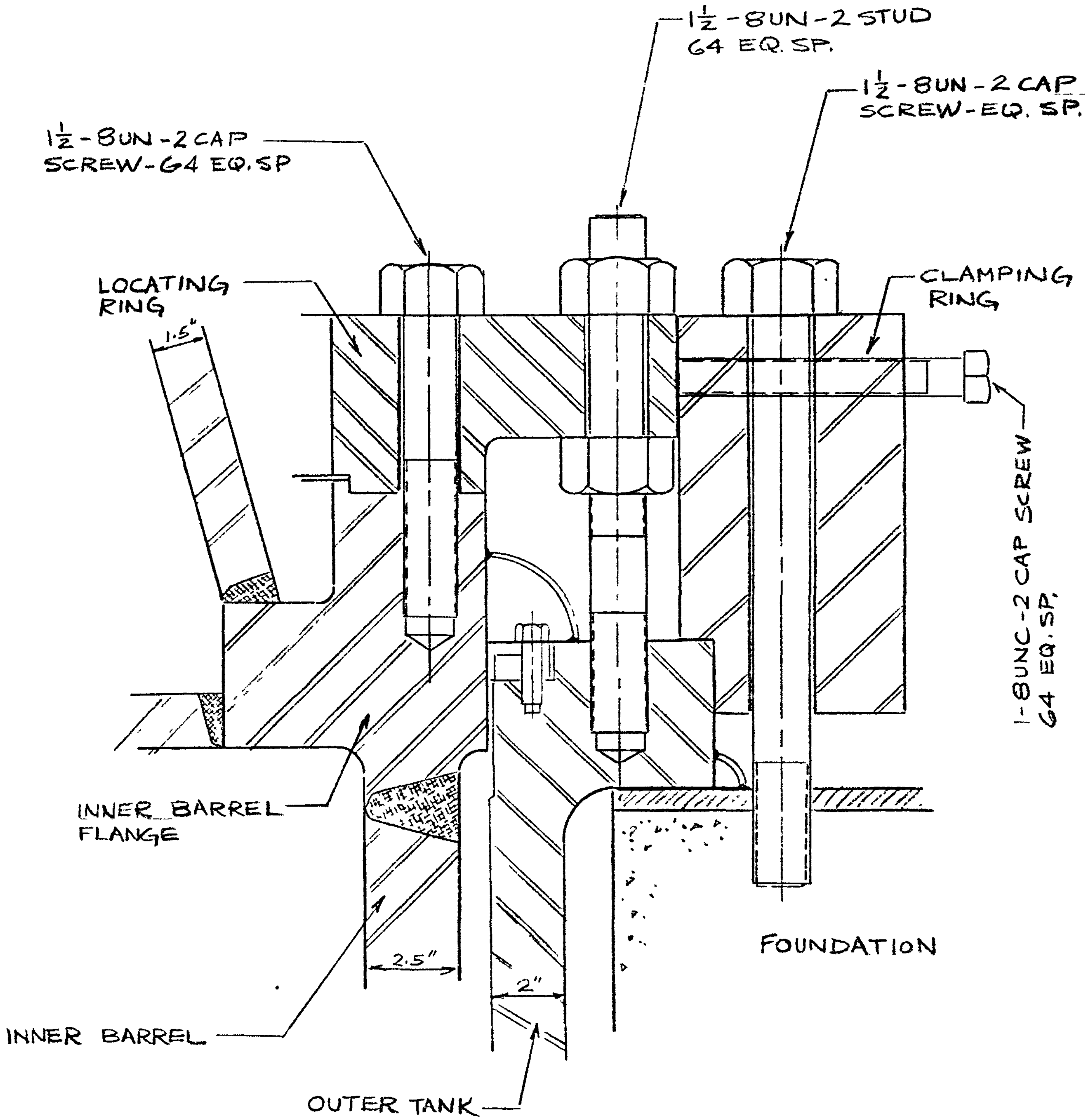


FIG. 4.5 – PUMP SUPPORT CONCEPT-HOT LEG
COMMERCIAL UNIT

Calculated stiffness between the inner structure and outer tank were

K_{rot}(foundation) = 1.35×10^{12} in-lbs/rad

K_{horiz}(foundation) = 1.496×10^{7} lbs/in

K_{horiz}(nozzle) = 1×10^{10} lbs/in (rigid)

A lateral stiffness was assumed between the inner structure and foundation as well. This is due to the fact that the inner structure can be seen to be restrained by the clamp ring and clamp bolt assembly. Although the clamp ring itself is not in direct contact with the locating ring (considered a part of the inner structure), they are indirectly in contact by a cap screw which is screwed into the clamp ring until contact is made with the locating ring.

The calculated lateral stiffness between the inner structure and foundation is:

K_{horiz} (inner to foundation) = 1.98×10^{6} lb/in

4.5.3 Outer Tank

The entire pump assembly is supported by the foundation at elevation 0.0 feet. The analysis assumes the following stiffness between the tank and the foundation:

K_{horiz} = 1.980×10^{6} lbs/in

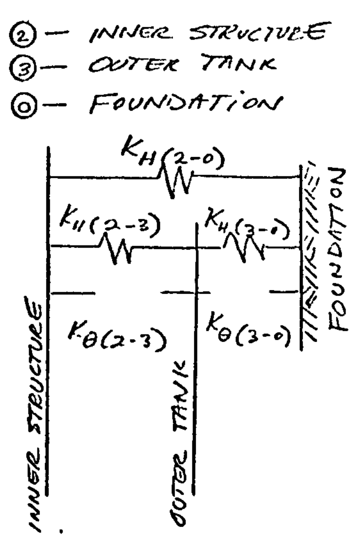
K_{rot} = 1.186×10^{12} in-lb/rad.

These stiffnesses were obtained using the assumption that the possible lateral seismic load would be of such magnitude to overcome the friction force at the outer tank-foundation contact point. Under such conditions, the lateral restraint between outer tank and foundation would be provided largely by the flexural stiffness of the 64 clamp ring bolts. The stiffnesses listed above are 64 times the individual bolt stiffnesses. (See Appendix B for Friction and Lateral Load Calculations)

The structure is also completely restrained against vertical motion at this location.

4.5.4 Variation of Stiffnesses

A run was first made using rigid restraints at the foundation level. Then a run was made using the calculated stiffnesses. From this run it was seen that the lateral stiffness between the inner structure and foundation ($K_{H(2-0)}$ and the lateral stiffness between the outer tank and foundation ($K_{H(3-0)}$) See Fig. 4-6) were the governing factors in the lateral frequency analysis.



STIFFNESSES AT THE FOUNDATION LEVEL

FIGURE 4-6

The effect of varying these stiffnesses, (i.e., $K_{H(2-0)}$, $K_{H(3-0)}$) was investigated by making four more runs substituting the following values for both $K_{H(2-0)}$ and $K_{H(3-0)}$:

$$
\begin{aligned}
K &= 1 \times 10^{6} \text{ lbs/in} \\
&= 1 \times 10^{7} \text{ lbs/in} \\
&= 1 \times 10^{8} \text{ lbs/in} \\
&= 1 \times 10^{10} \text{ lbs/in}
\end{aligned}
\qquad \text{(Run Set 1)}
$$

Then, three more runs were made using the following values for $K_{H(2-0)}$ and $K_{H(3-0)}$ but all other restraints at the foundation level were assumed rigid. The values are:

$$
\begin{aligned}
K &= 1 \times 10^{6} \text{ lbs/in} \\
&= 1 \times 10^{7} \text{ lbs/in} \\
&= 5 \times 10^{7} \text{ lbs/in}
\end{aligned}
\qquad \text{(Run Set 2)}
$$

5.0 MODAL CHARACTERISTICS

5.1 Lateral Characteristics

Six computer runs were made to determine the fundamental frequencies of the individual components and of the entire assembled unit under different boundary condition assumptions. Appendix E contains the mode shape plots of each run. Table 5.1 and Figures 5.1 to 5.15 summarize the results of the analyses.

File #1 is a shaft analysis with pinned connections assumed at all four bearing locations. The first mode natural frequency is 23.24 cps and represents a resonance of the shaft between the lower motor bearing and the upper hydrostatic bearing. The second mode is 42.27 cps and is a resonance of the motor shaft between the two motor bearings.

File #2 is identical to File #1 except the stiffnesses of the four bearings have been included. The first mode natural frequency is 18.29 cps and corresponds to the first mode of the pinned analysis (Figure 5.1). The second mode natural frequency (Figure 5.2) is 28.01 cps and represents the second shaft bending mode.

File #3 is a motor analysis with pinned constraints. The first mode natural frequency is 38.55 cps.

File #4 is a pump analysis with rigid constraints at the foundation level. The first mode (Figure 5.3) is seen to be a structural bending mode of the portion of the pump below the foundation level. The first

(1) = Shaft
(2) = Inner Structure
(3) = Outer Tank
(0) = Foundation

File #	Component	Constraint	Mode and Natural Frequency 1	2	3	4
1	Shaft	Pinned Constraints	23.24	42.27	60.85	108.60
2	Shaft	Spring Constraints	18.29	28.01	28.52	36.39
3	Motor	Pinned Constraints	38.55	100.9	---	---
4	Pump	Rigid Foundation Constraints	15.38	17.90	23.61	26.65
5	Pump	Calculated Spring Constraints	7.18	15.49	17.74	25.55
6	Pump	Calc. Spring Constraints with $K_{(Hor)}$ = 1×10^7 lbs/in. for (2-0)&(3-0)	11.62	17.65	17.80	25.95

TABLE 5-1

Summary of Lateral Modal Characteristics mode natural frequency is 15.38 cps. The second mode (Figure 5.4) is a shaft bending mode with natural frequency of 17.9 cps. The third mode (Figure 5.5) is seen to be a structural bending mode of the portion above the foundation level. It's natural frequency is 23.61 cps. The fourth mode is the second shaft bending mode with natural frequency of 26.65 cps. (Figure 5.6)

File #5 is a pump analysis utilizing the calculated spring constants of Section 4.5.2 and 4.5.3 at the foundation level. The first mode natural frequency is 7.18 cps and represents a rigid body mode due to lateral displacement. The second mode natural frequency is 15.49 cps and is seen to be a rigid body rocking mode. The third and fourth modes, when compared to the first and second modes of File #2 can be seen to be shaft bending modes with very little dynamic coupling with any structural mode. The mode shapes for these vibrating motions are presented in Figure 5.7 thru 5.10.

Thus, the lateral constraints at the foundation level are governing factors, in the lateral frequency analysis.

The effect of the lateral stiffness between the inner structure and foundation and the stiffness between the outer tank and foundation was investigated by varying the spring constraints as discussed in Section 4.5.4. These results are plotted and summarized in Figure 5.11

File #6 presents results from Figure 5.11 for the case where $K = 10^7$. This is the minimum value of $K_{H(2-0)}$ and $K_{H(3-0)}$ which will yield acceptable structural frequencies. The first mode natural frequency of 11.62 cps represents a structural bending mode which is partially effected by the lateral stiffness at the foundation (Figure 5.12).

The second and third modes at 17.65 cps and 17.80 cps, respectively, are combined modes of the first shaft bending mode with the cantilever mode of the pump unit above the foundation. Figure 5.13 shows the lower frequency mode to be an in-phase resonance while Figure 5.14 shows an out-of-phase resonance for the higher frequency.

The fourth mode (Figure 5.15) represents the second shaft bending mode with natural frequency of 25.95 cps.

5.2 Vertical Characteristics

The vertical modal characteristics were determined separately from the lateral modes and the results are given below. The calculations are presented in Appendix C.

MODE	NATURAL VERTICAL FREQUENCY
1	37.60 cps
2	214.12 cps
3	374.88 cps

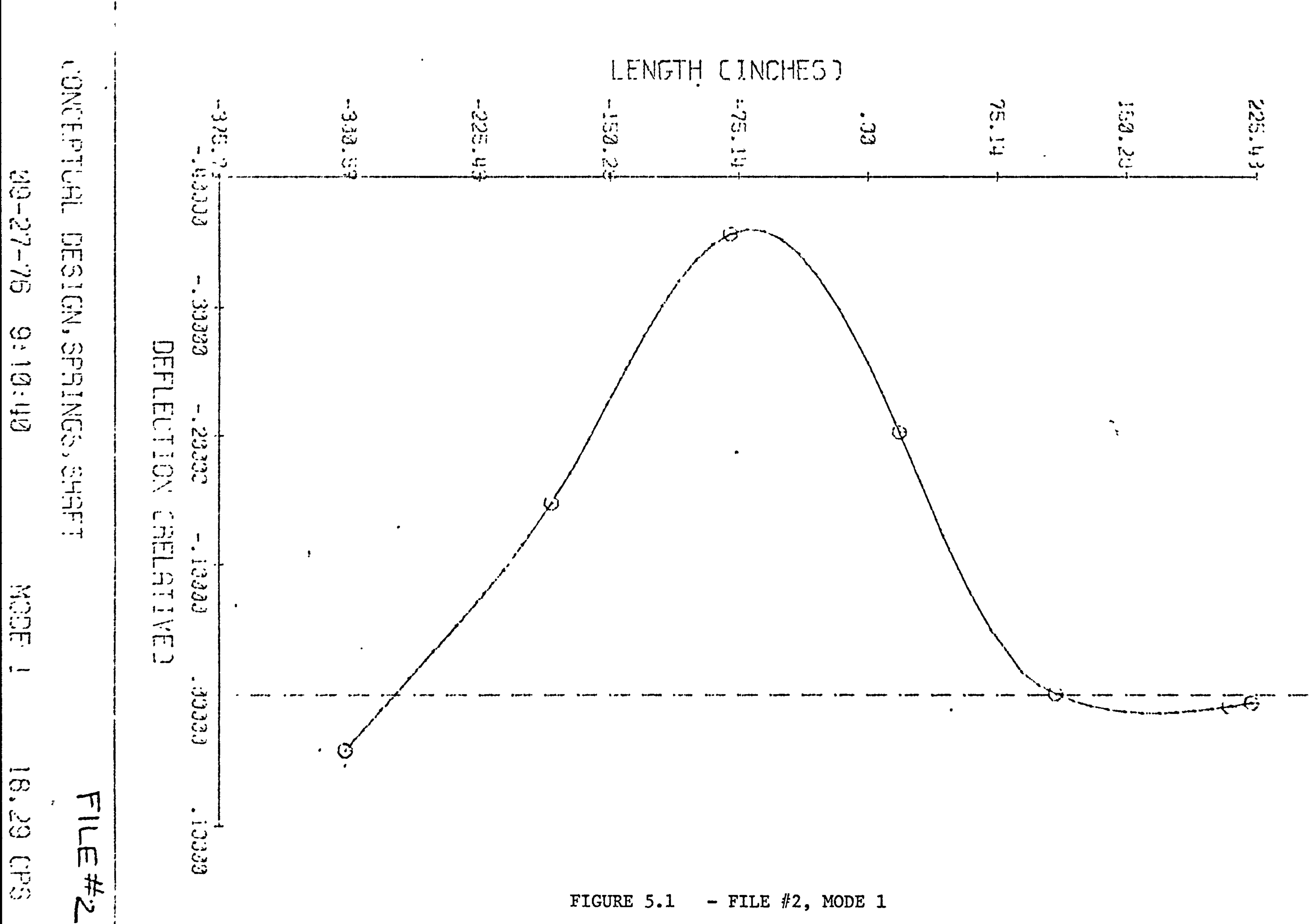


FIGURE 5.1 - FILE #2, MODE 1

CONCEPTUAL DESIGN, SPRINGS, SHAFT

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MODE 2

26.91 CPS

FILE #2

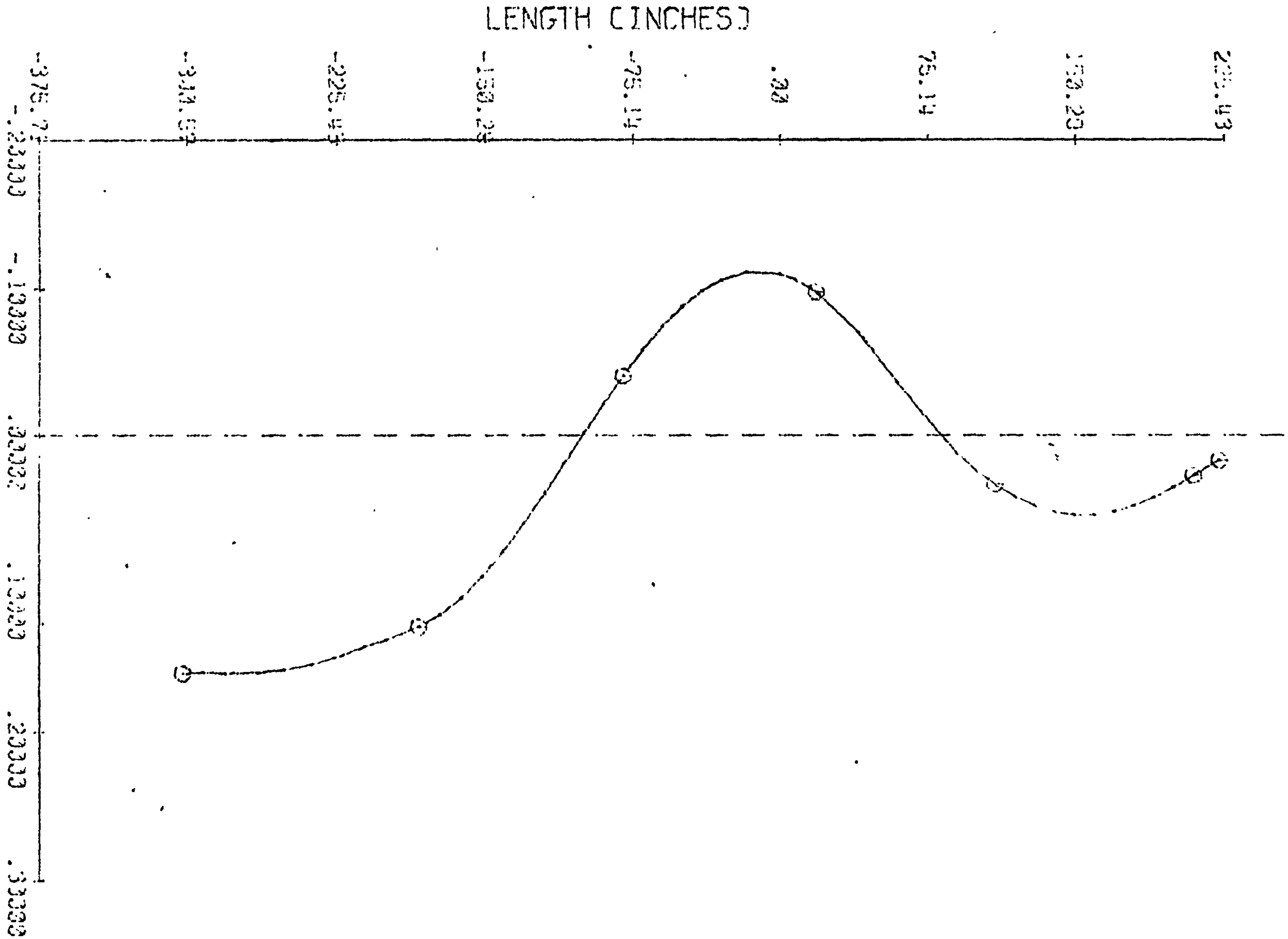


FIGURE 5.2 - FILE #2, MODE 2

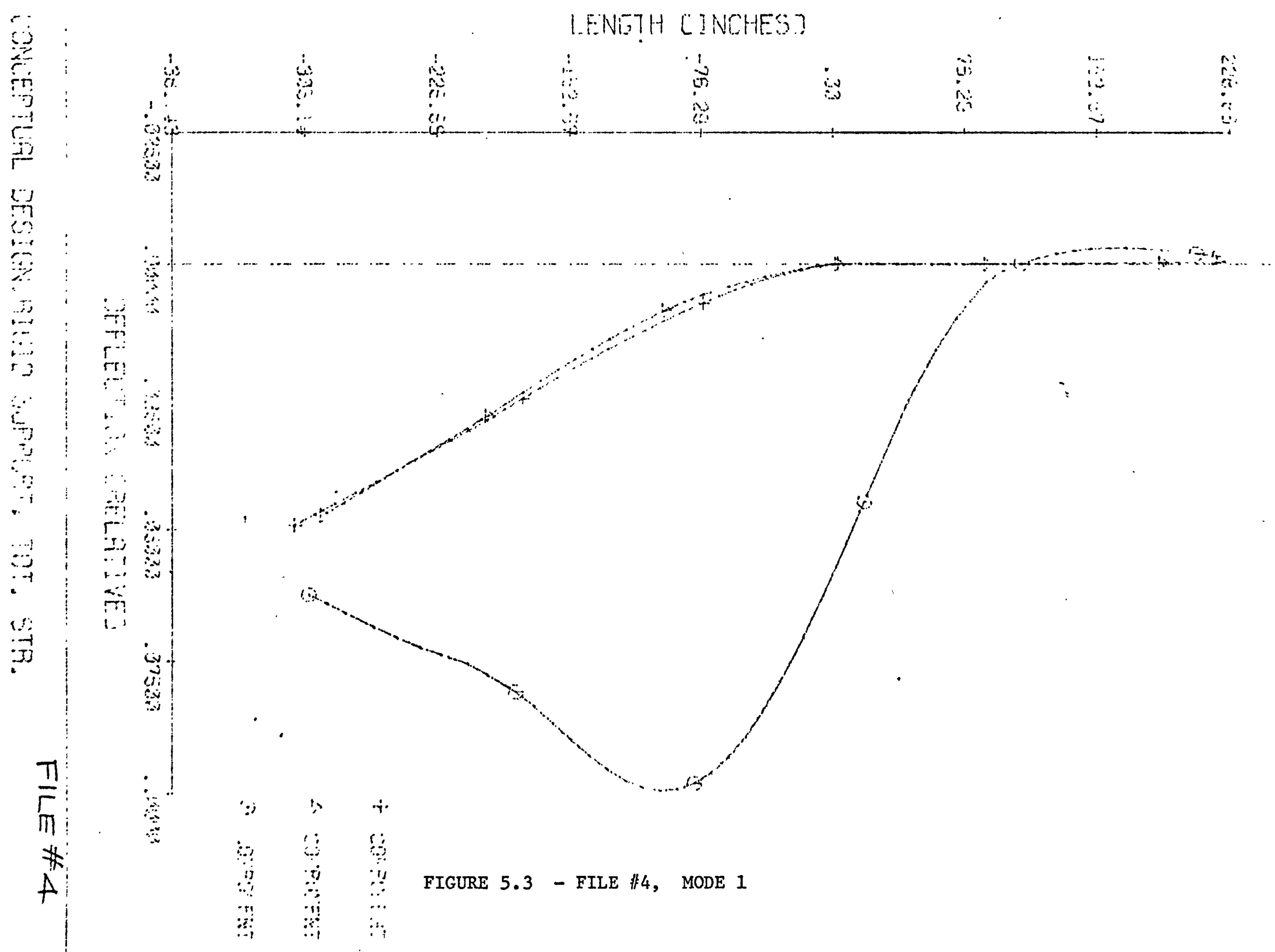


FIGURE 5.3 - FILE #4, MODE 1

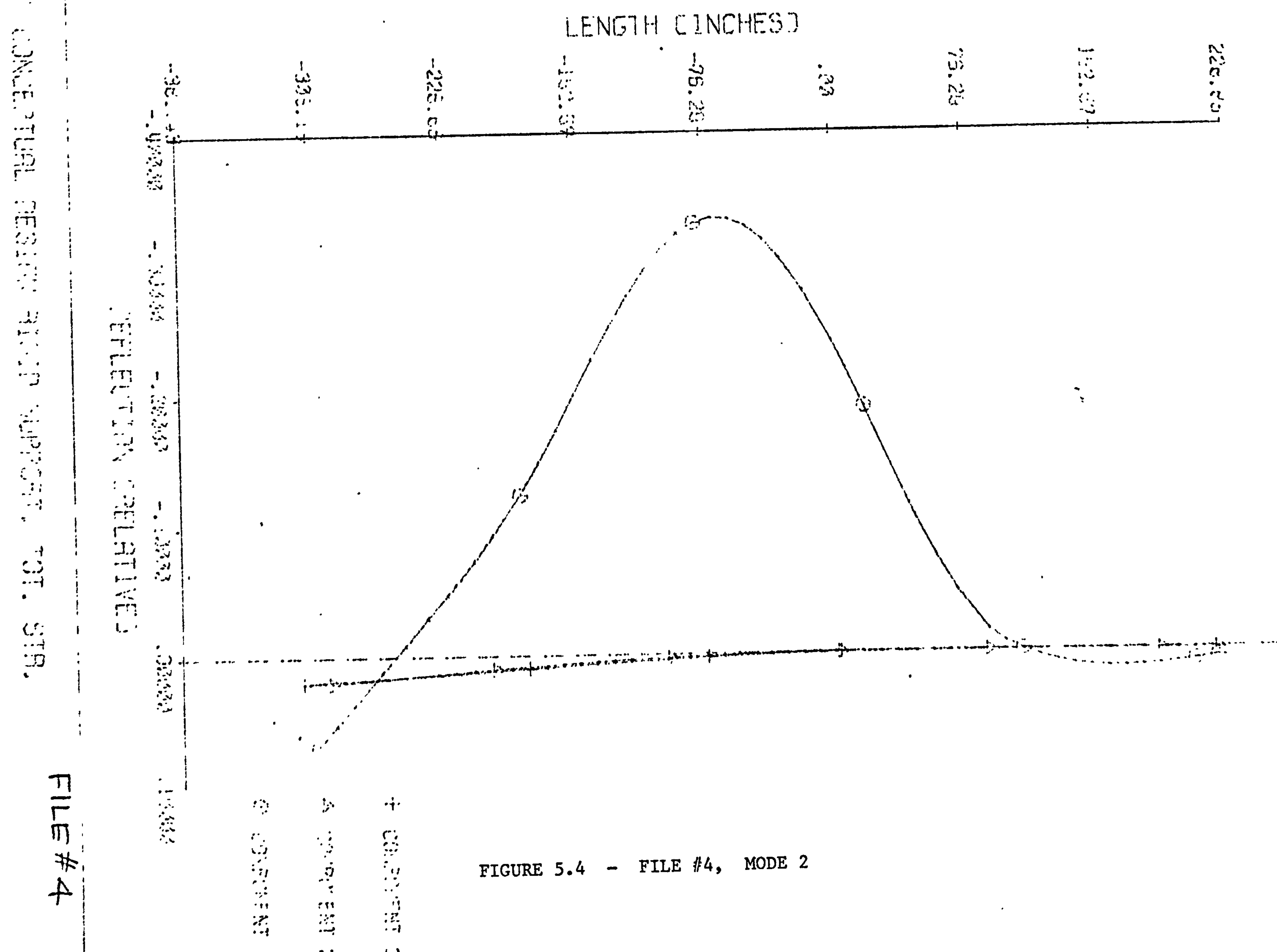


FIGURE 5.4 - FILE #4, MODE 2

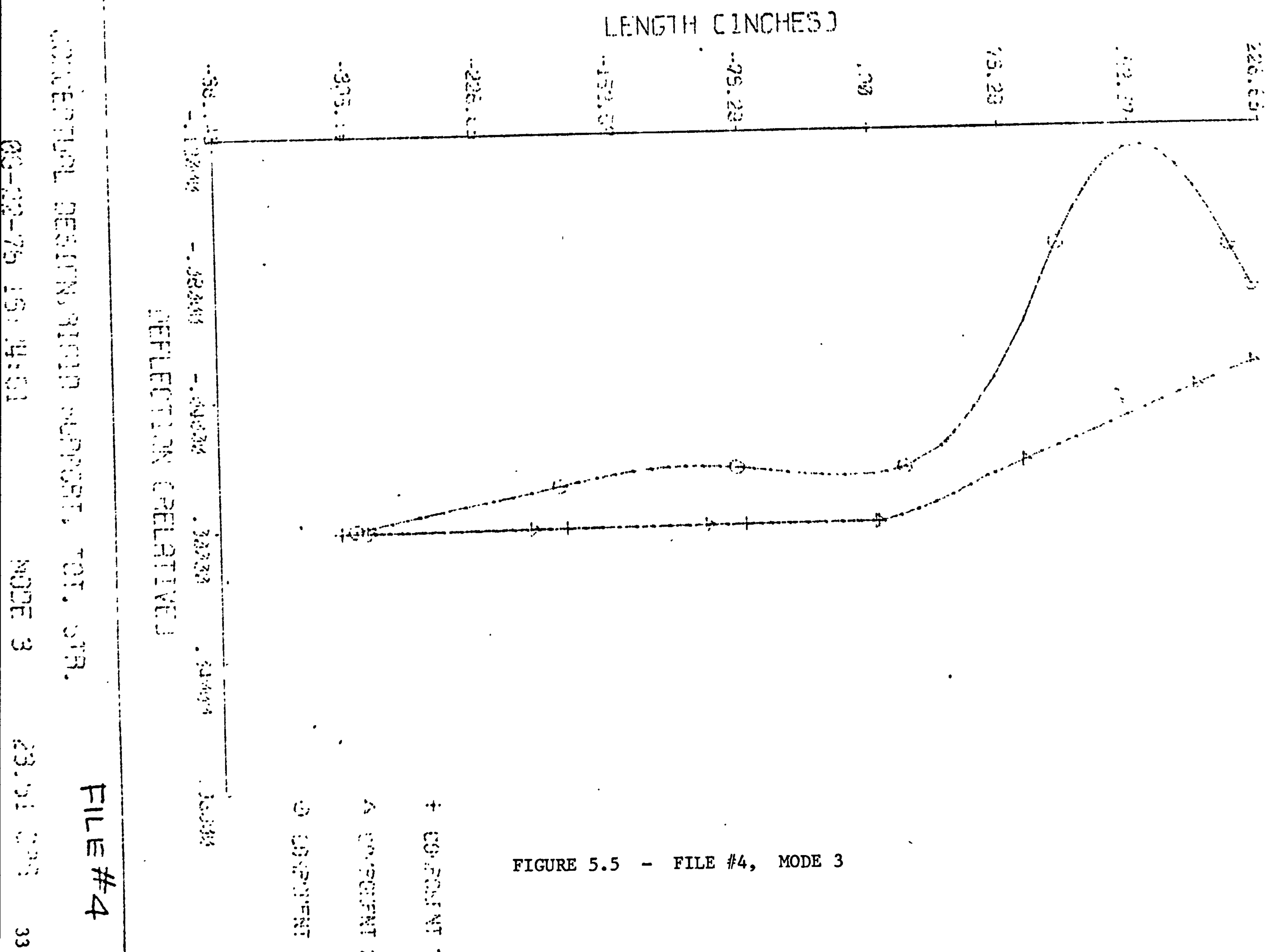


FIGURE 5.5 - FILE #4, MODE 3

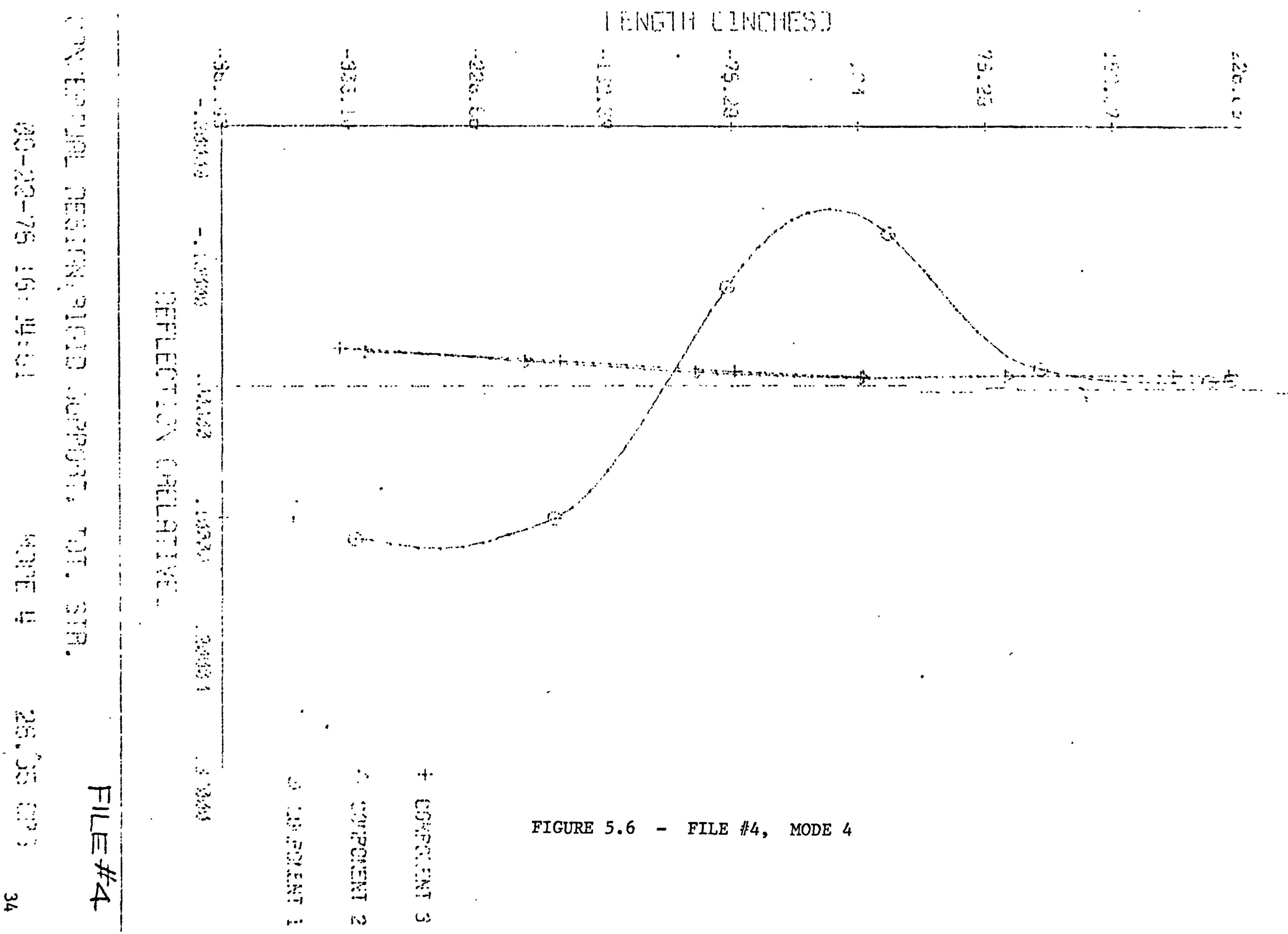


FIGURE 5.6 - FILE #4, MODE 4

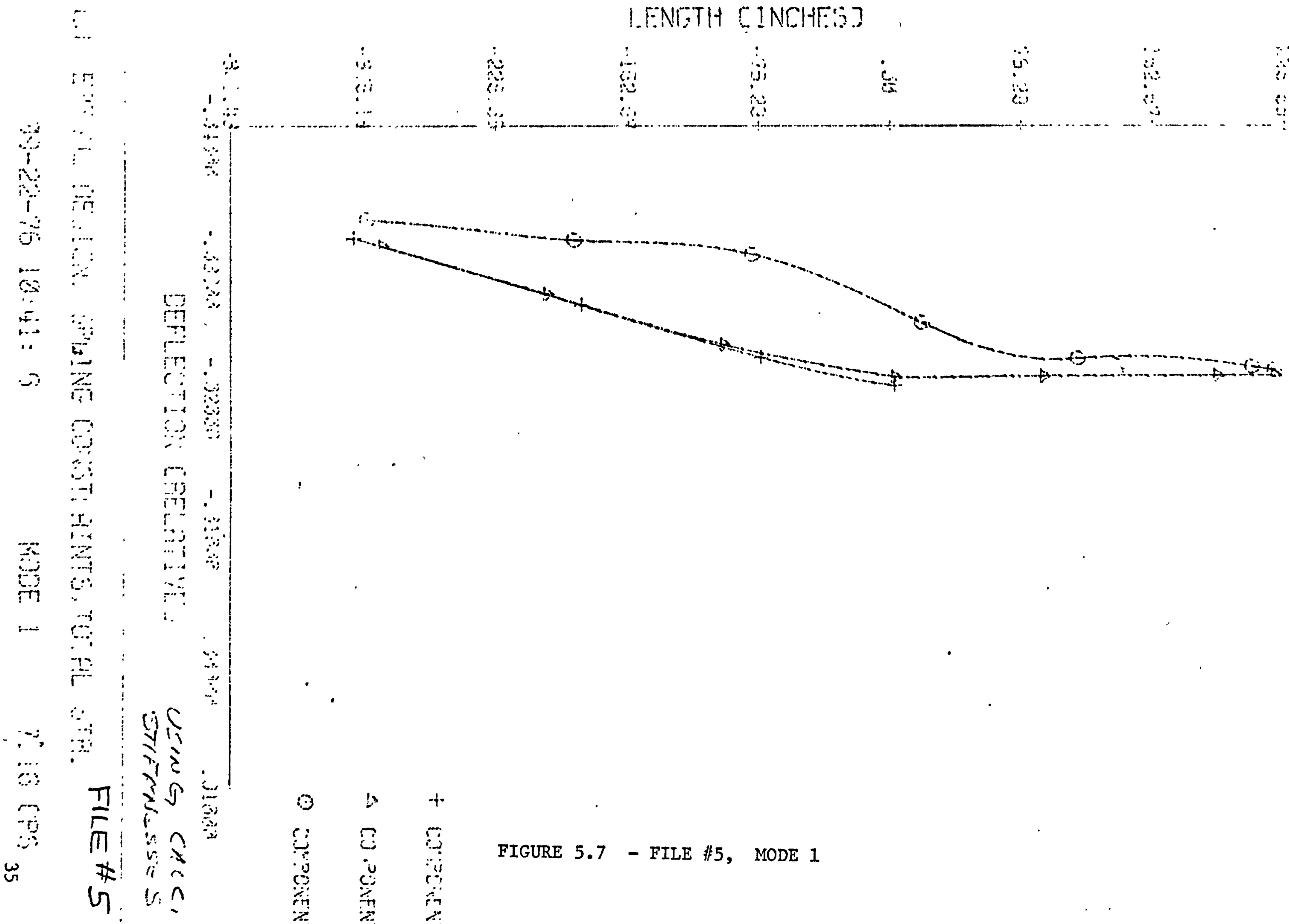


FIGURE 5.7 - FILE #5, MODE 1

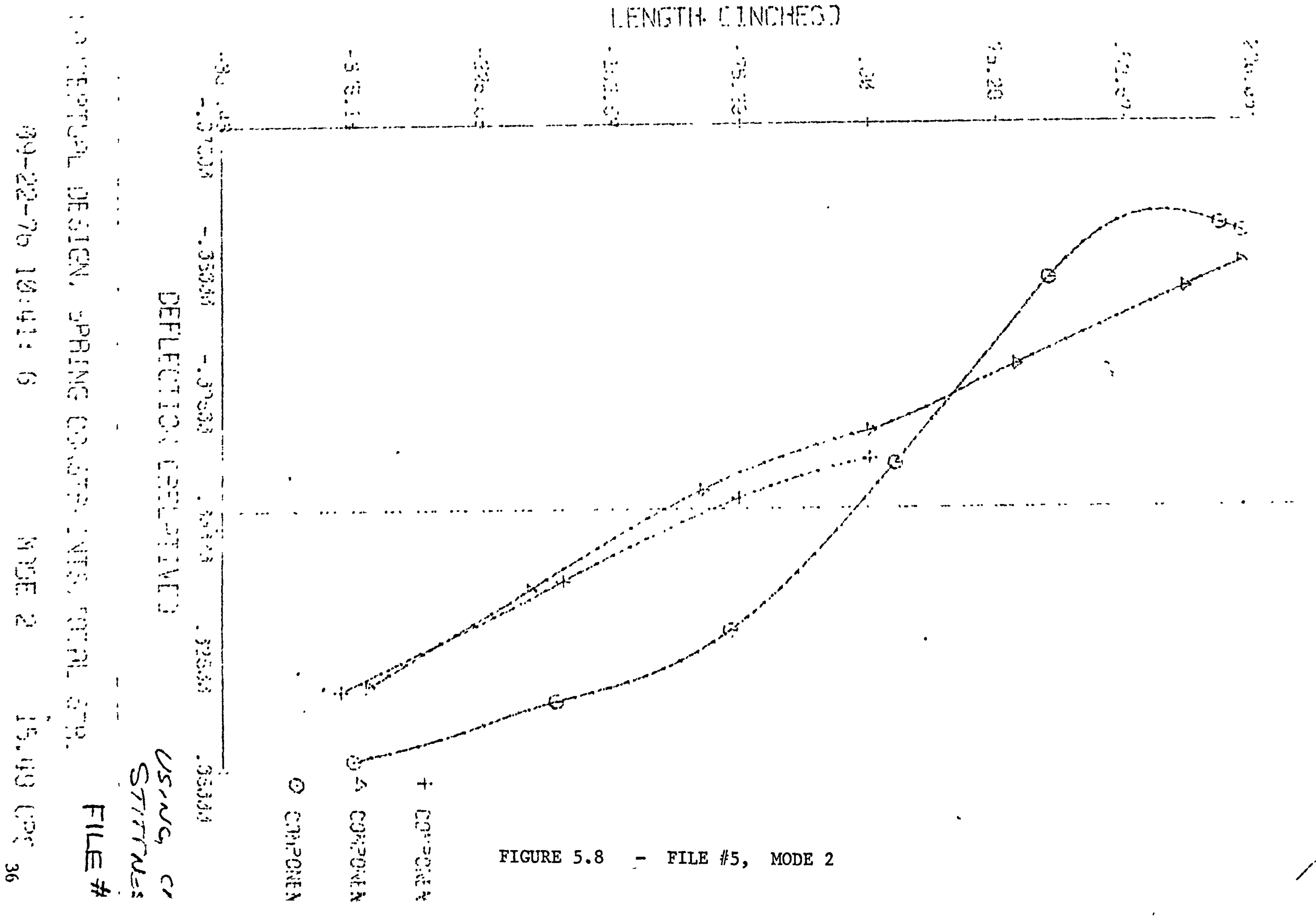


FIGURE 5.8 - FILE #5, MODE 2

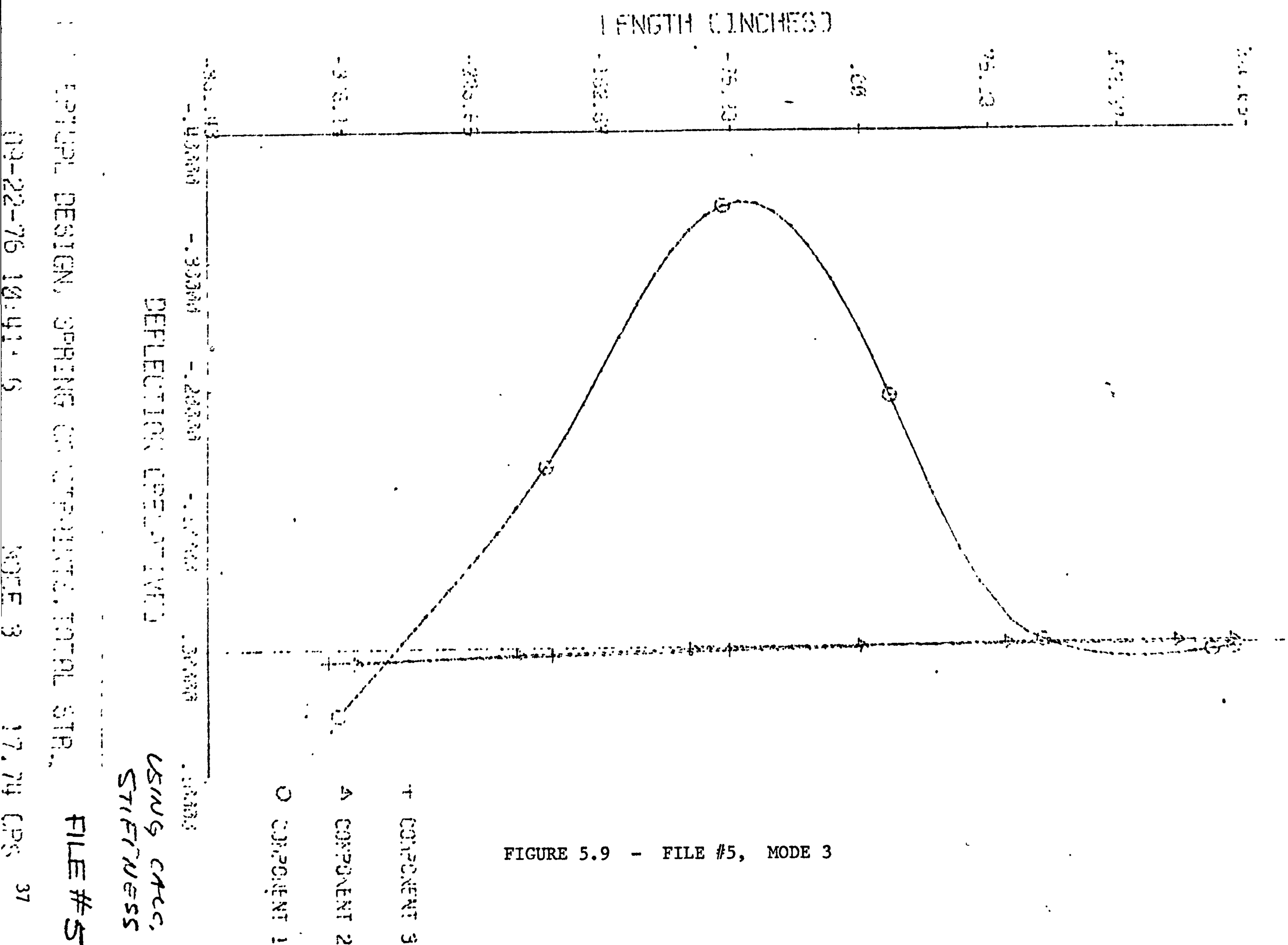


FIGURE 5.9 - FILE #5, MODE 3

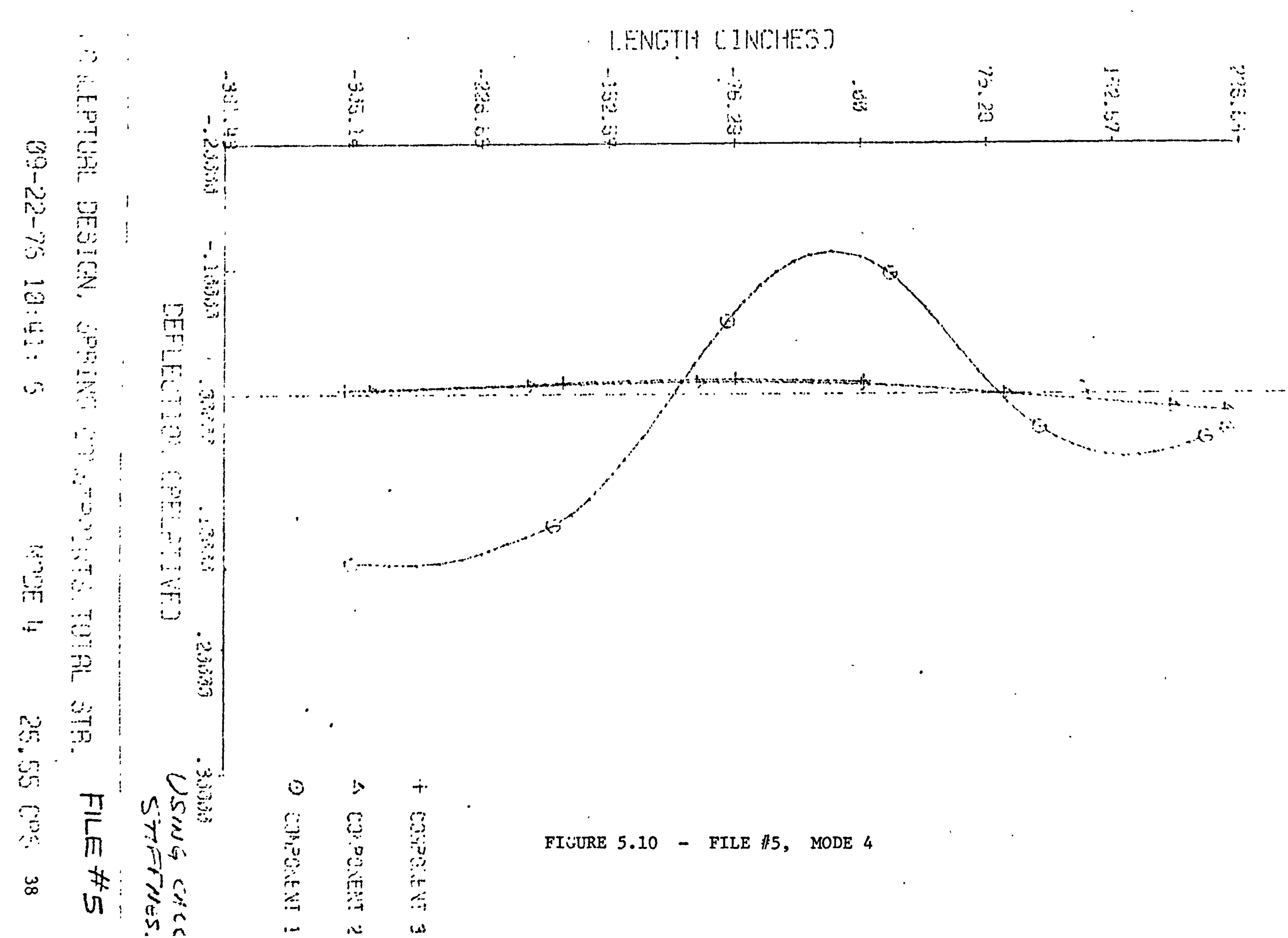
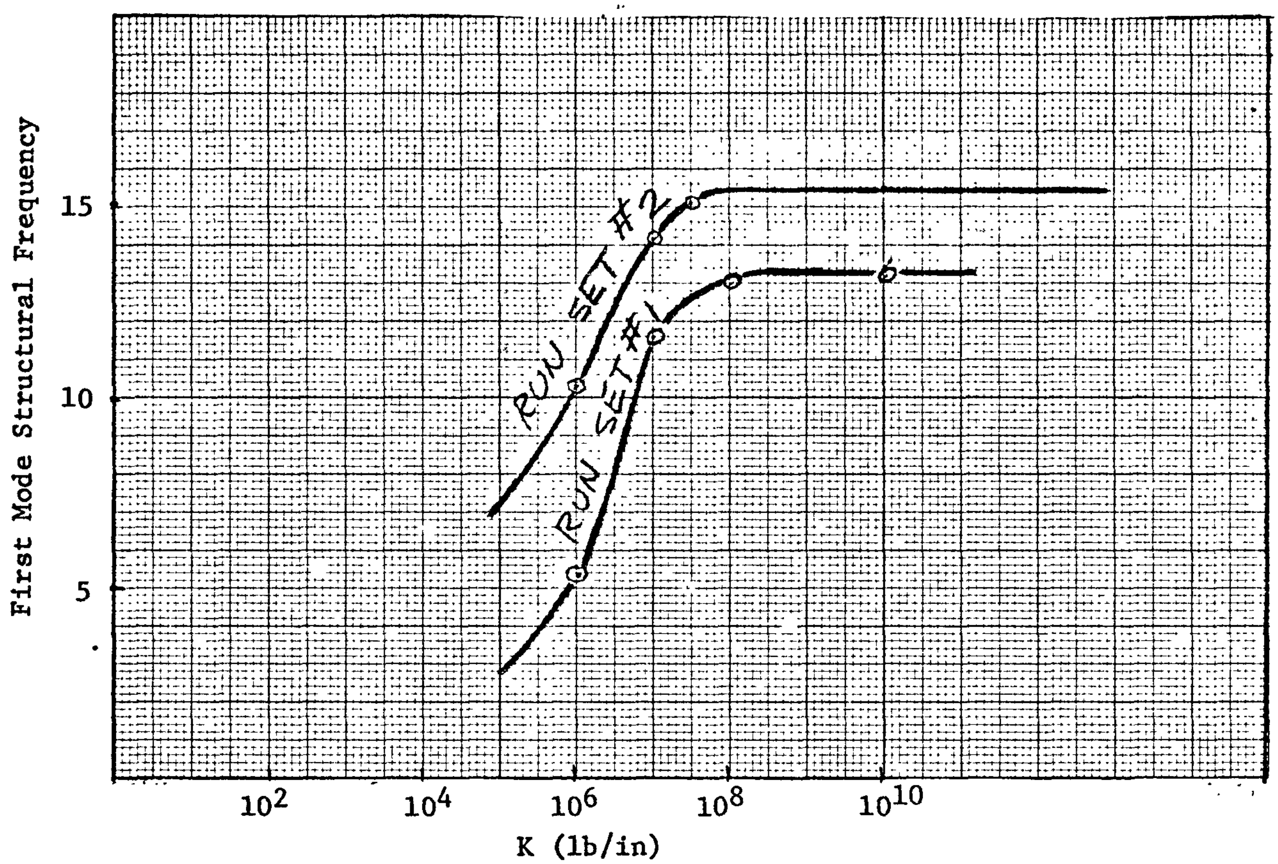


FIGURE 5.10 - FILE #5, MODE 4



Effect of $K_{H(2-0)}$ and $K_{H(3-0)}$

on First Mode Structural Frequencies

FIGURE 5.11

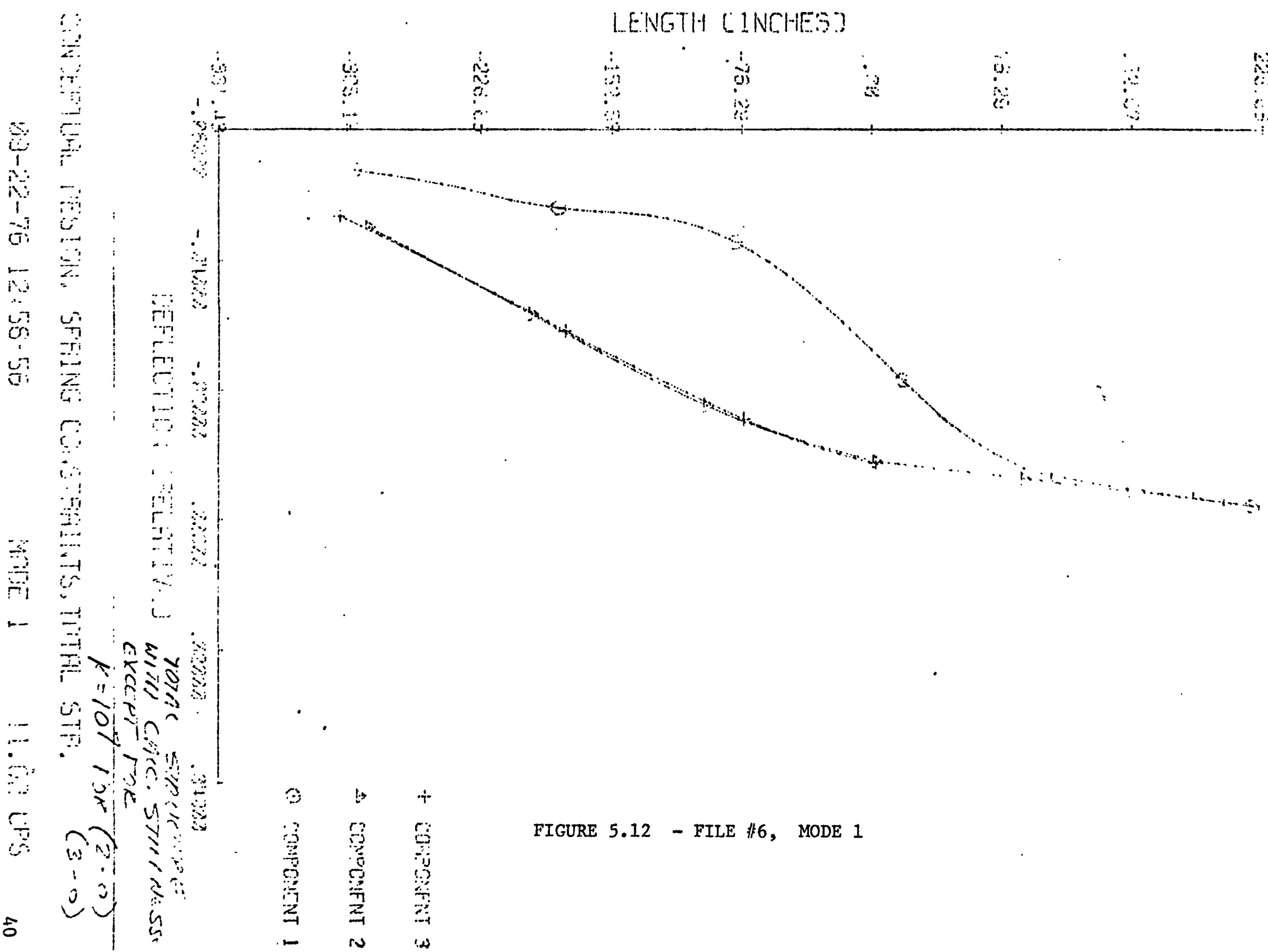


FIGURE 5.12 - FILE #6, MODE 1

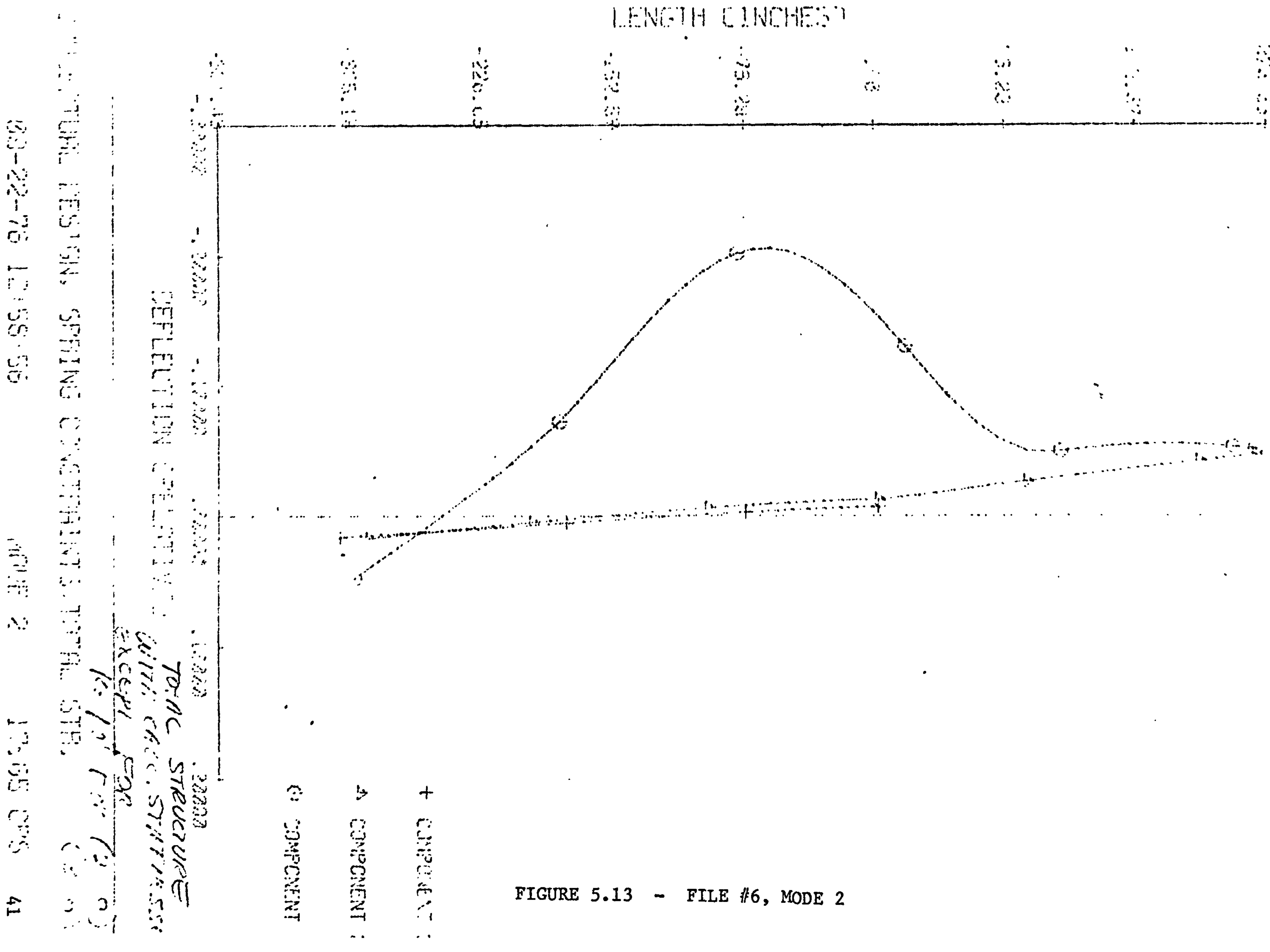


FIGURE 5.13 - FILE #6, MODE 2

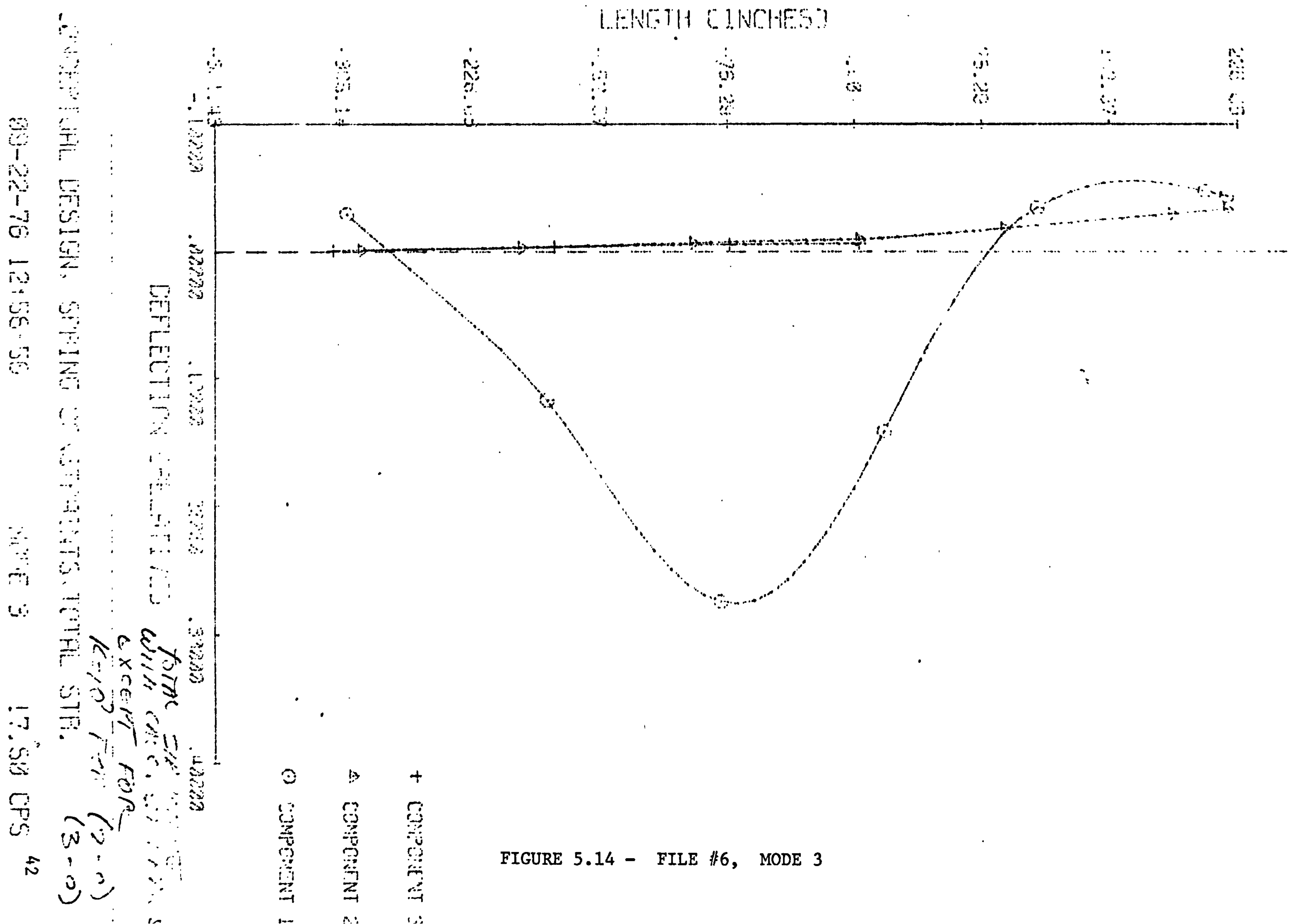


FIGURE 5.14 - FILE #6, MODE 3

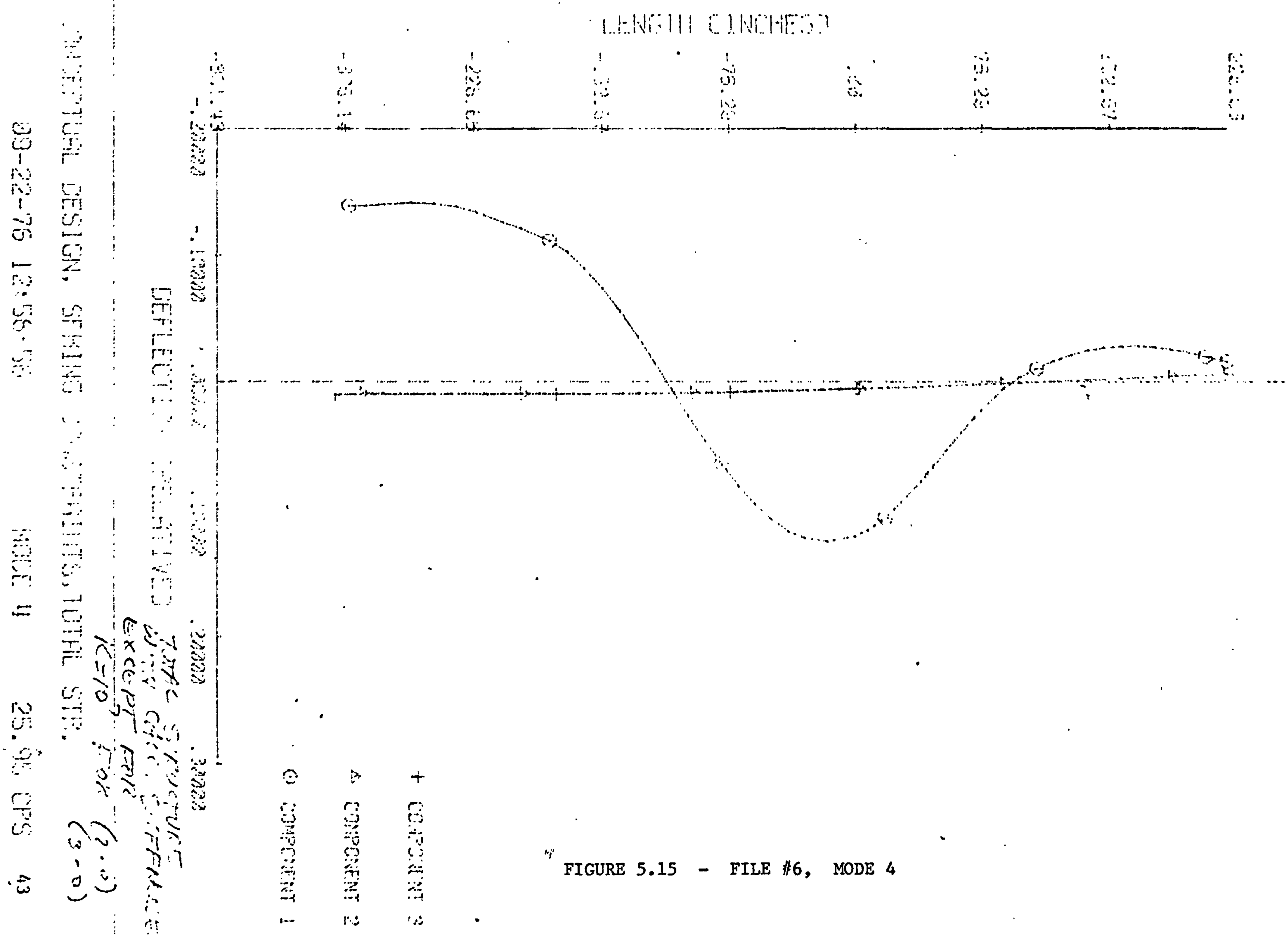


FIGURE 5.15 - FILE #6, MODE 4

6.0 CONCLUSIONS

6.1 Shaft Frequency

It can be seen from File #1 that the shaft design is adequate with a natural first mode frequency of 23.24 cps. From File #2 it is apparent that the rotor bearings provide adequate stiffness since the first mode natural frequency is 18.29, well above the required 12.5 cps (125% of operating speed). The shaft and its constraints appear to be very adequate in lateral vibration.

6.2 Structural Frequency

From File #4 it is seen that the overall design of the pump with a rigid foundation mounting is acceptable since the lowest frequency obtained is 15.38 cps, well above the required 11.5 cps (115% of operating). However, from File #5 we observe that the lateral restraints at the foundation level do not provide the required stiffness since rigid body lateral motion with a natural frequency of 7.18 cps is observed. These results are based on the assumption that friction does not provide adequate lateral constraint to hold the pump assembly stationary. This assumption is further discussed in Section 7.2.

From Figure 5.11 we see that there is a critical sector of both curves where a change in magnitude of the stiffness results in a corresponding change in frequency. At the end of this section there is an asymptotic section of the curve where large increases of stiffness do not result in very large increases of frequency.

The upper curve of Figure 5.11 becomes asymptotic at approximately 15.4 cps. This value corresponds closely to the 15.38 cps obtained from our run File #4 which is our analysis of the pump with rigid foundation constraints.

The lower curve of Figure 5.11 shows that if the values for $K_{(HOR)(2-0)}$ and $K_{(HOR)(3-0)}$ are increased above 1×10^7 lbs/in, the frequency increases to greater than 11.62 cps. This meets the requirements of Reference 2.2 and is not in the range of the seismic response spectrum peak.

The second and third modes were seen from Figures 5.13 and 5.14 to be dynamically coupled modes of the shaft and motor-motor support stand. This is due to the assumption of a relatively rigid motor (Section 4.3.2). When the motor stiffness is decreased to more realistic values, these two frequencies will be separated by approximately 2 to 3 cps and will not be dynamically coupled. In either case the frequency of these modes is sufficiently above the 11.5 cps requirement.

Thus the design of the pump presents no insurmountable dynamic problems. However, the design of the pump support assembly is a critical factor in meeting specified frequency limits. The present support assembly should be altered to provide greater lateral stiffness between the outer tank and foundation.

7.0 RECOMMENDATION

7.1 Shaft

No alterations to the shaft design appear to be necessary since the frequencies obtained were well above the 125% of operating speed limitation.

7.2 Structure

Since rigid body lateral deflections were obtained in the first structural mode, it appears evident that increasing the horizontal stiffnesses at the foundation level is necessary to substantially increase the frequency.

This may be accomplished by adding a shoulder on the outer tank mounting flange to provide a rigid lateral connection between outer tank and the foundation. An alternate to this would be to provide a shim around the entire perimeter of the outer tank mounting flange, which would also create a rigid lateral constraint at this point. A computer run was made, assuming $K_{H(3-0)}$ (Figure 4.6) to be infinitely rigid, and the results show a first mode natural frequency of 13.08 cps. This value should be compared with the results of File #5 (Table 5.1), and varifies the validity of increasing the lateral stiffness at the foundation level to increase the natural frequency. Appendix E has mode shape plots of this run.

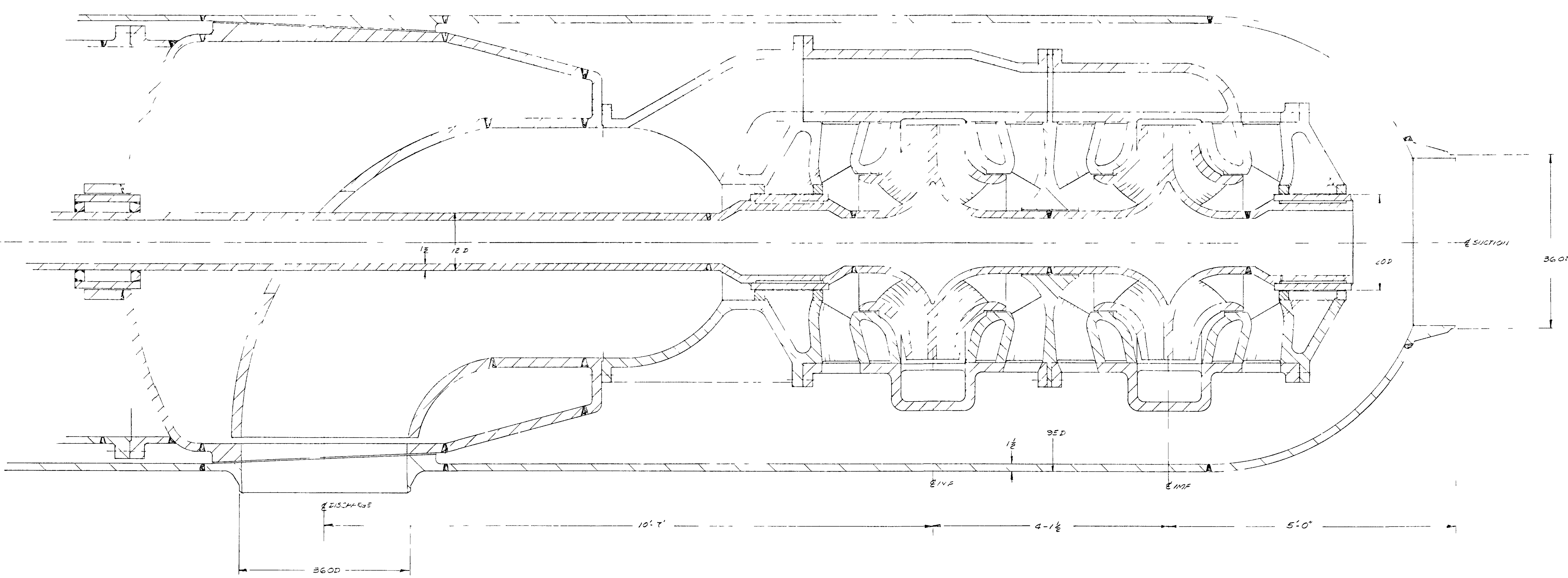
7.3 Pump Support

The results of a concurrnet study to accurately determine the equivalent stiffness of the pump to foundation support assembly are listed in the table below:

PARAMETER	VALUES USED IN COMPUTER INPUT	CONCURRENT STUDY VALUES
$K_{H(2-0)}$	1.98×10^6	1.50×10^5
$K_{H(2-3)}$	1.50×10^7	8.41×10^7
$K_{H(3-0)}$	1.98×10^6	2.92×10^9
$K_{\theta(2-3)}$	1.35×10^{12}	9.50×10^{11}
$K_{\theta(3-0)}$	1.19×10^{12}	3.0×10^{12}

As can be seen from the table, the concurrent study assumes a relatively rigid $K_{H(3-0)}$ (i.e., essentially pinned) consistent with the recommendation of Section 7.2.

The concurrent study value for $K_{H(2-0)}$ is smaller than the estimated value used for this analysis. However, all other spring values used in the dynamic analysis have relatively similar magnitudes. To determine the effect of the pump support stiffnesses determined by this analysis, an additional computer run was made and a first mode natural frequency of 14.69 cps was obtained. This is acceptable and is further removed from the range of the seismic response spectrum peak.



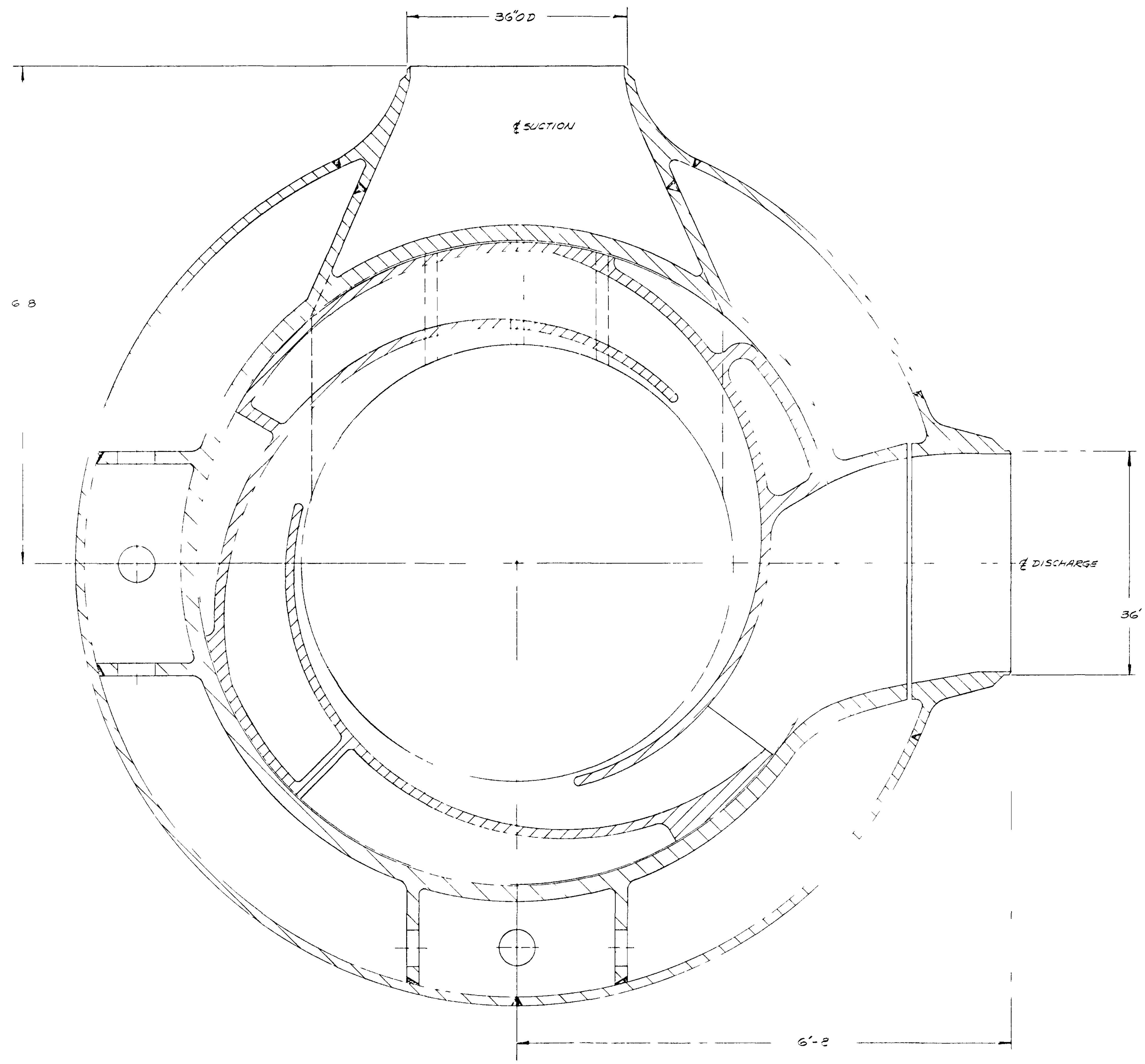
T61-G-0002

LW 10 SEP 76

LARGE SCALE SODIUM PUMP
(HOT LEG)
ALTERNATE DESIGN CONCEPT

DRAWING NO SK-0002-101 0

71 1/2 SPHERICAL RADIUS
761-6-0002
BYRON JACKSON
LARGE SCALE SODIUM PUMP
(HOT LEG)
DRAWING NO
1F-8084
REV
0
SCALE 1/8" = 1"
SHEET 1 OF 3
AEF



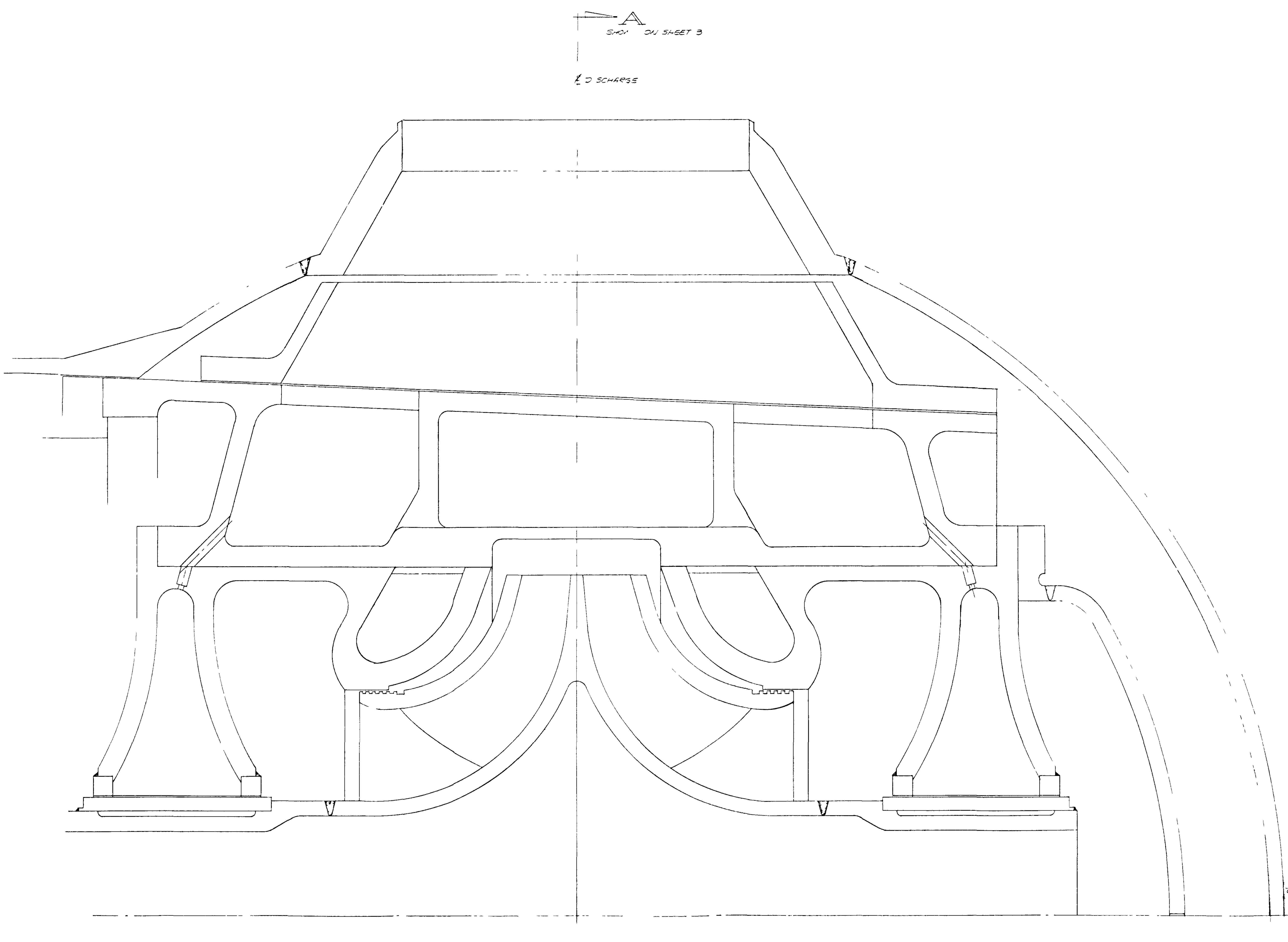
SECTION A-A (SEE SHEET 1 OR 2)

TG1-G-0002
LW 31 AUG 76

LARGE SCALE SODIUM PUMP
(HOT LEG)

	1F-8084	Q
1/8"=1"		SHEET 3 OF 3

AEF



761-G-0002
LW 18 AUG 76

BYRON JACKSON

LARGE SCALE SODIUM PUMP
(HOT LEG)

CODE IDENT NO
DRAWING NO 1F-8084
REV

SCALE 1/4" = 1"
SHEET 2 OF 3

Ø	INITIAL ISSUE				

RUNNING CLEARANCE IN INCHES ON DIAMETER	
A	.060
B	.120
C	.125
D	.125

NOTE.
1 FOR PARTS NUMBER IDENTIFICATION SEE BYRON JACKSON DWG NO
2 SEE FOR SEAL AND HEAT EXCHANGER

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON DECIMALS / ANGLES
SURFACE ROUGHNESS PER MIL STD 10
DRILLED HOLES AS DRILLED
TAPPED HOLES AS TAPPED
DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES

BJ ORDER NO.	7GI-G-0002
DRAWN BY	S. GLISSON
DATE	25 AUG. 76
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BYRON JACKSON
BORG-WARNER CORPORATION

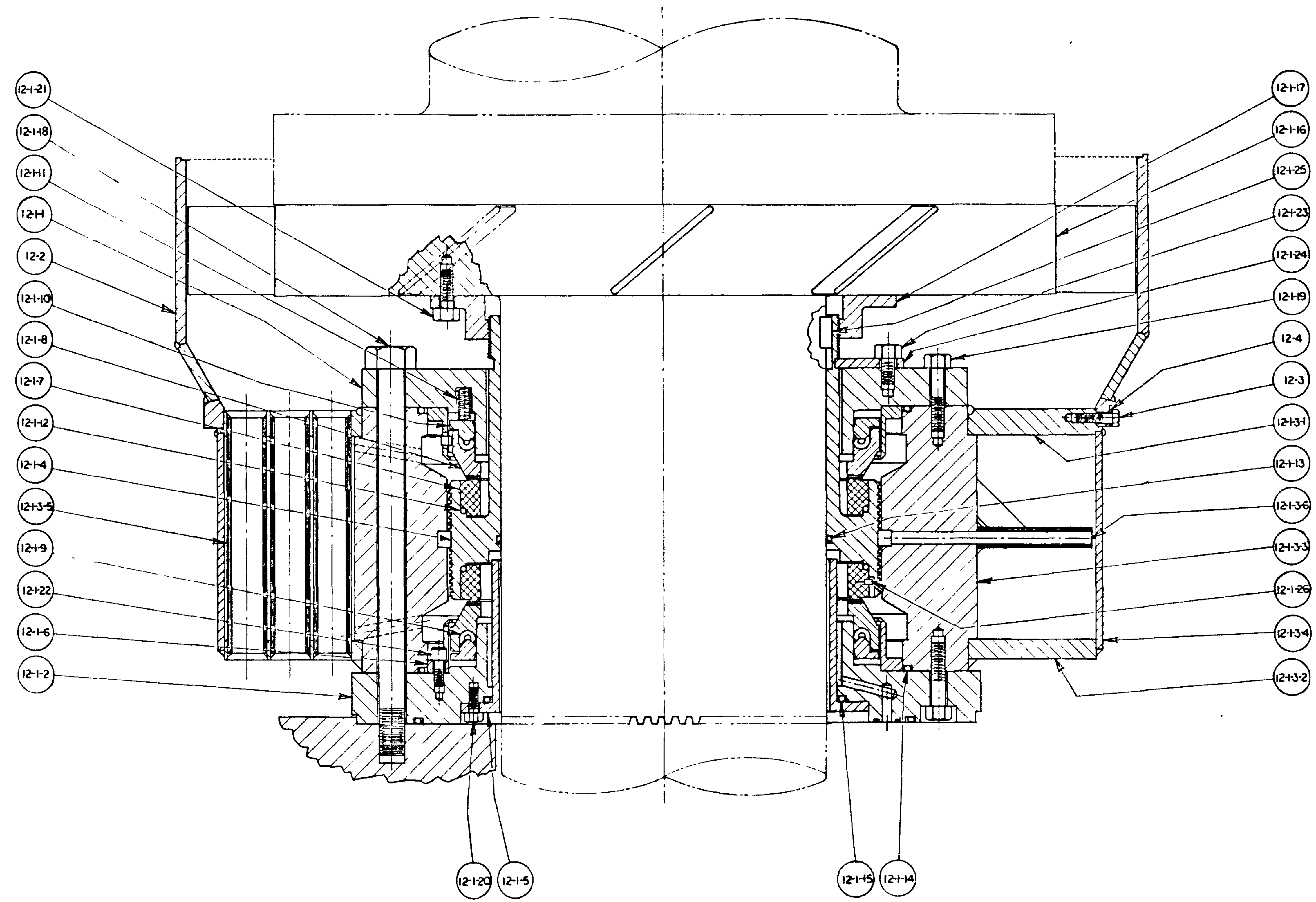
DRAWING TITLE
LARGE SCALE SODIUM PUMP
(HOT LEG)

CODE IDENT. NO.	DRAWING NO.	REV
	IF-8085	Ø

SCALE 1/8" = 1"

AEF

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
⊖	ORIGINAL		



REFERENCE:
FULL SCALE ASSEMBLY IE-3996
CONCEPTUAL DESIGN IC-3572

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION 88.

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT — 1/64
TURN — 1/64	2 TO 6 FT — 1/32
	6 FT AND OVER — 1/16

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS: .X ± .XX ± .XXX ±
ANGLES ±
✓ SURFACE ROUGHNESS PER MIL STD 10
DRILLED HOLES AS DRILLED
TAPPED HOLES ✓ AS TAPPED
DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .004 / .015
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CUSTOMER APP'D	DATE

Byron Jackson Pump Division
BORG-WARNER CORPORATION
LOS ANGELES DIVISION

DRAWING TITLE
SEAL CARTRIDGE SECTIONAL

CODE IDENT NO.
DRAWING NO. IC-3571

SCALE | WEIGHT | SHEET

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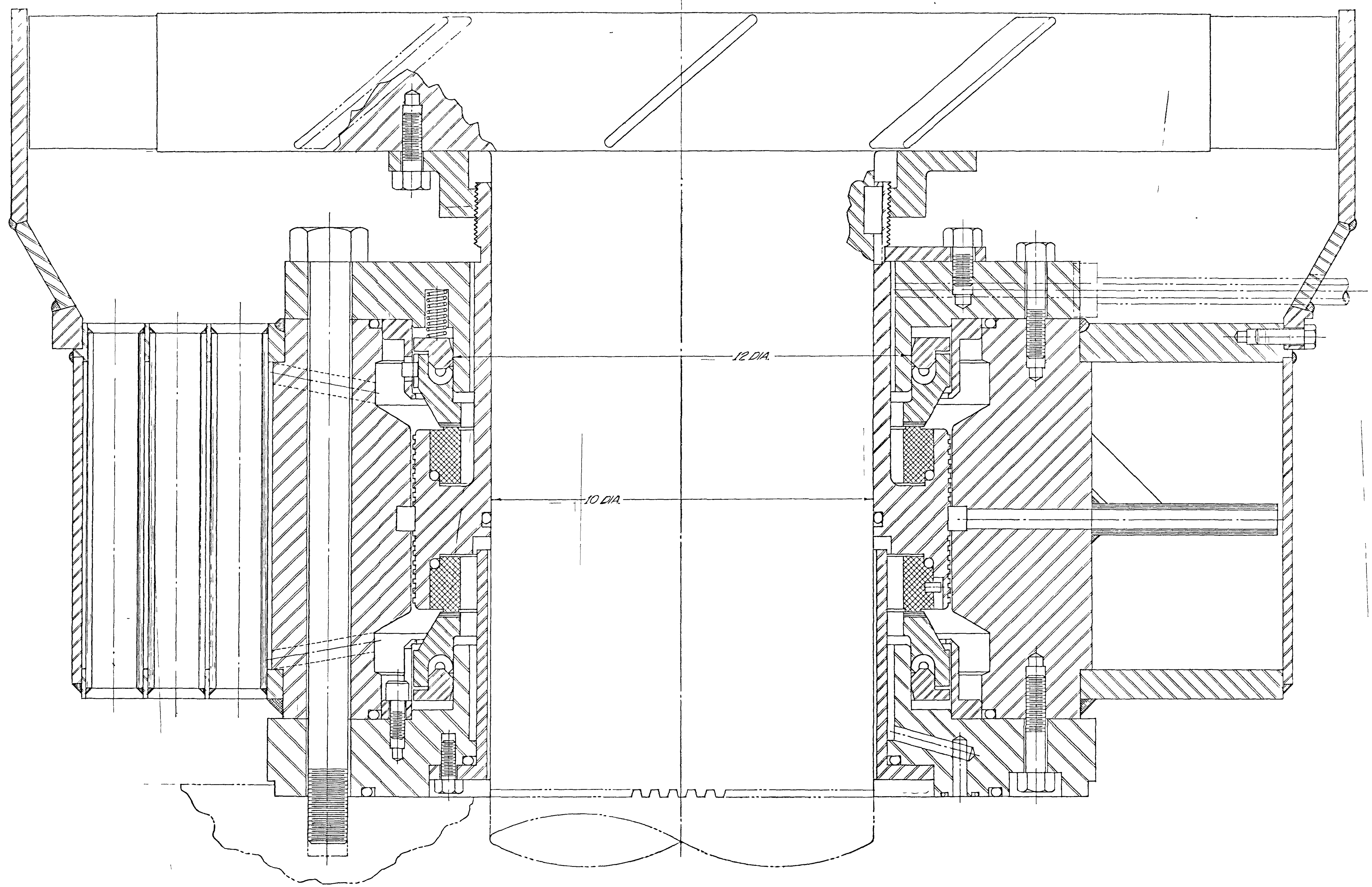

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ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION D2 1

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT — 1/64
TURN — 1/64	2 TO 6 FT — 1/32
	6 FT AND OVER — 1/16

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DECIMALS — ANGLES ±
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XXXX ±

✓ SURFACE ROUGHNESS PER MIL STD 10

DRILLED HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

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BORG WARNER BYRON JACKSON®
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DRAWING TITLE 12 C00 SEAL CARTRIDGE LMFBR PUMP

CONTRACT NO.

CODE IDENT NO. — DRAWING NO. 1E-3996 — REV 0

SCALE FULL — WEIGHT — SHEET

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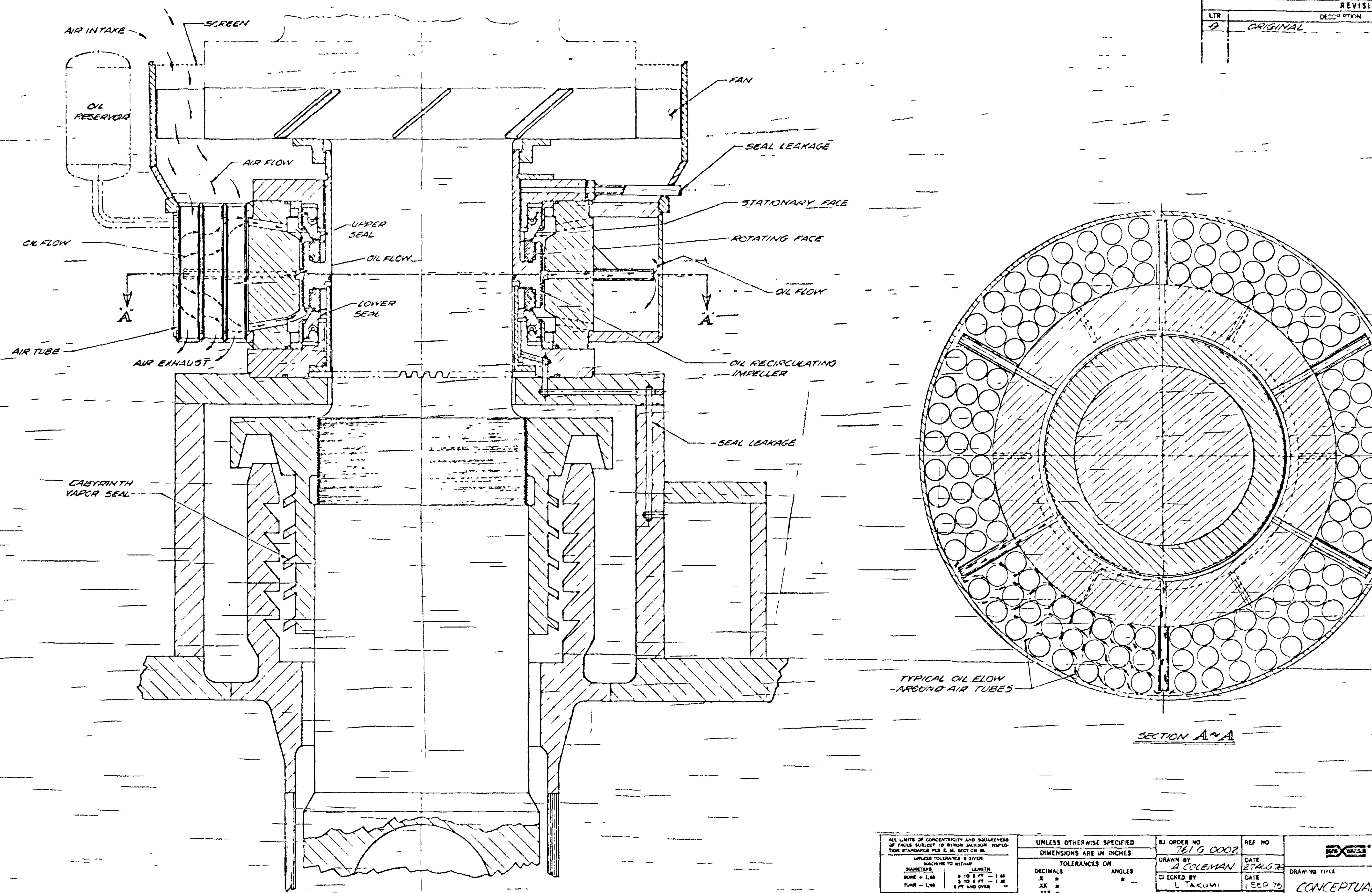

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LTR	DESCRIPTION	DATE	APPROVED
Ø	ORIGINAL		

REFERENCE:
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SEAL CARTRIDGE SECTIONAL IC-3571

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION 86.

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 3 FT — 1/64
TURN — 1/64	3 TO 6 FT — 1/32
	6 FT AND OVER 1/16

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS .X ± .XX ± .XXX ±
ANGLES ±

√ SURFACE ROUGHNESS PER MIL STD 10

DRILLED HOLES √ AS DRILLED
TAPPED HOLES √ AS TAPPED

DO NOT SCALE DRAWING.
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005 / .015
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CONTRACT NO			
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CUSTOMER APP'D	DATE		

Byron Jackson Pump Division
BORG-WARNER CORPORATION
LOS ANGELES DIVISION

DRAWING TITLE
CONCEPTUAL SEAL CARTRIDGE DESIGN LMFBR PUMP

CODE IDENT NO.	DRAWING NO.	
	IC-3572	
SCALE	WEIGHT	SHEET

AEF

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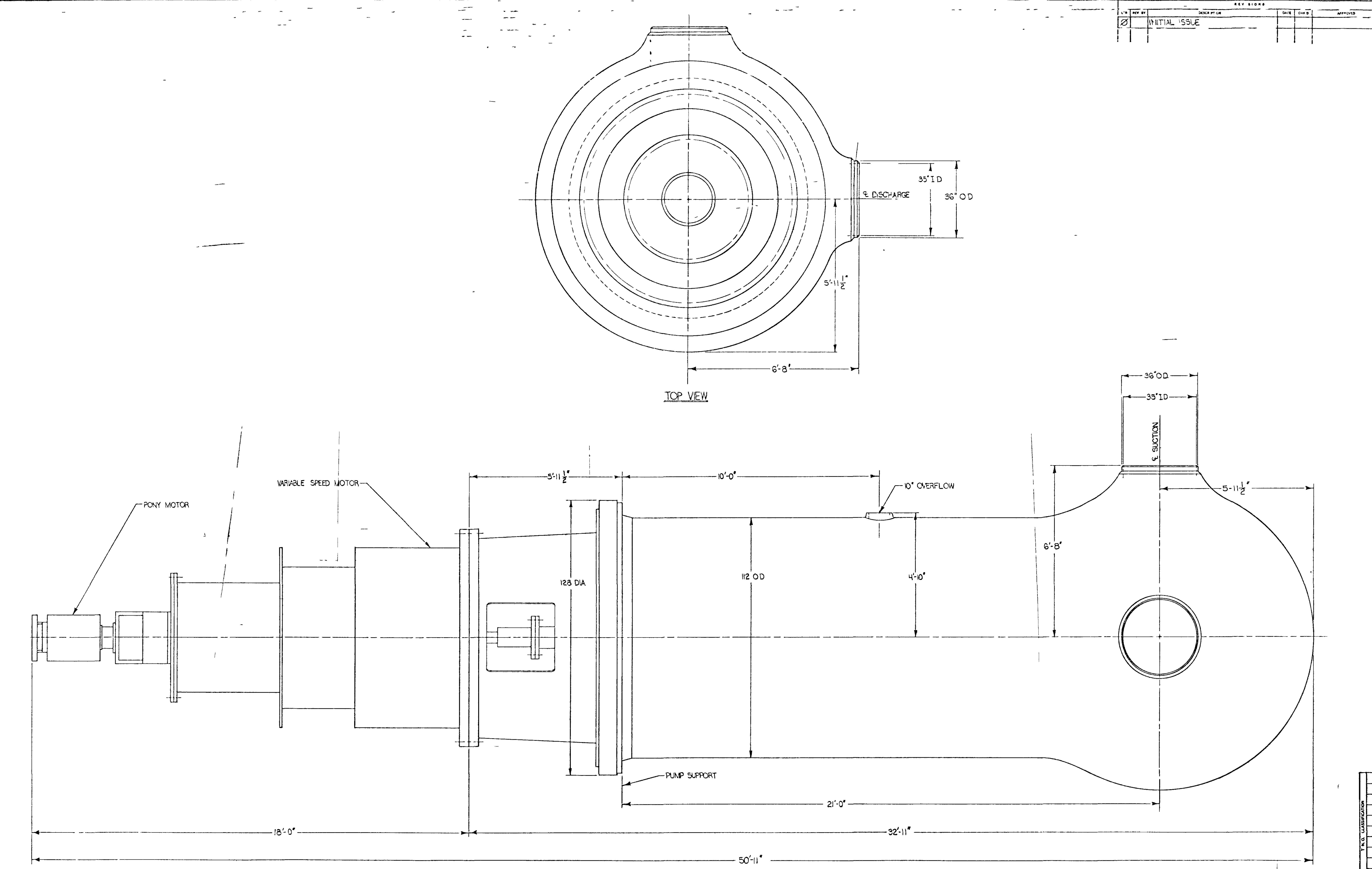

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ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION D2 1

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN
DIAMETERS: BORE + 1/64, TURN — 1/64
LENGTH: 0 TO 2 FT — 1/64; 2 TO 6 FT — 1/32; 6 FT AND OVER — 1/16

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DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS: XX ±, XXX ±, XXXX ±; ANGLES ±
✓ SURFACE ROUGHNESS PER MIL STD 10
DRILLED HOLES V AS DRILLED
TAPPED HOLES V AS TAPPED
DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO 005 / 015
ALL MACHINED CORNERS OR FILLETS 005 / 020

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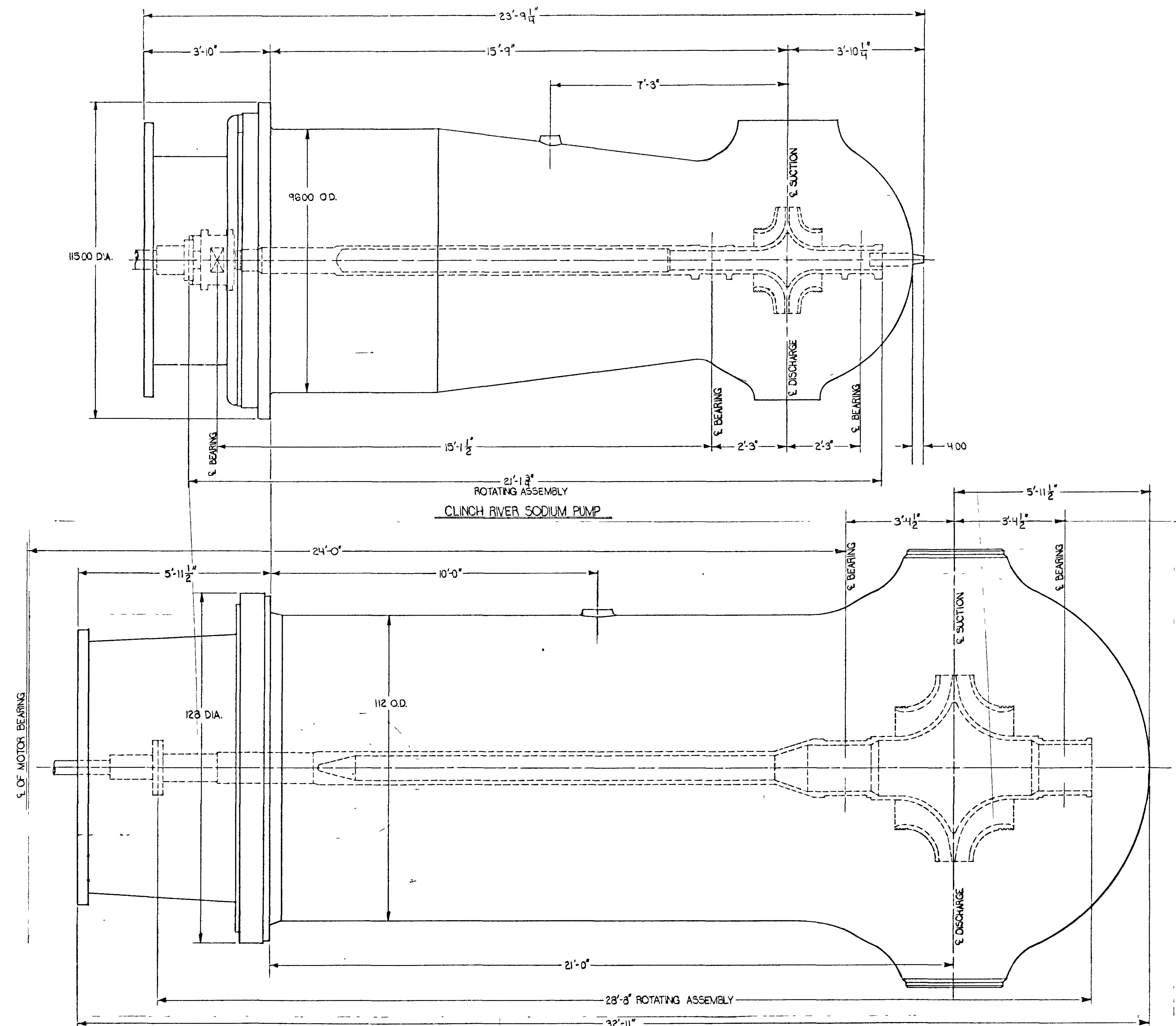
DRAWING TITLE
OUTLINE DRAWING
LARGE SCALE SODIUM PUMP-HOT LEG

CONTRACT NO.
CODE IDENT NO. | DRAWING NO 2F-1757 | REV Ø
SCALE 3/4" = 1'-0" | WEIGHT | SHEET

T.N.O. CLASSIFICATION

AEF



CLINCH RIVER SODIUM PUMP

LARGE SCALE SODIUM PUMP - HOT LEG

REVISIONS					
LTR	REV BY	DESCRIPTION	DATE	CHK'D	APPROVED
Ø		INITIAL ISSUE			

DISTRIBUTION	ARG	AUST	CAN	HOL	JAPAN	L.A.	MEX	TULSA	ORIGINATOR
DATE									

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION D2 11

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT. — 1/64
TURN — 1/64	2 TO 6 FT. — 1/32
	6 FT. AND OVER — 1/16

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS XX ± XXX ± XXXX ± ANGLES ±

✓ SURFACE ROUGHNESS PER MIL STD 10

DRILLED HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005 / .015
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BORG WARNER® BYRON JACKSON®
BORG WARNER CORPORATION

DRAWING TITLE SIZE COMPARISON
LSBRP TO CRBRP
LARGE SCALE SODIUM PUMP - HOT LEG

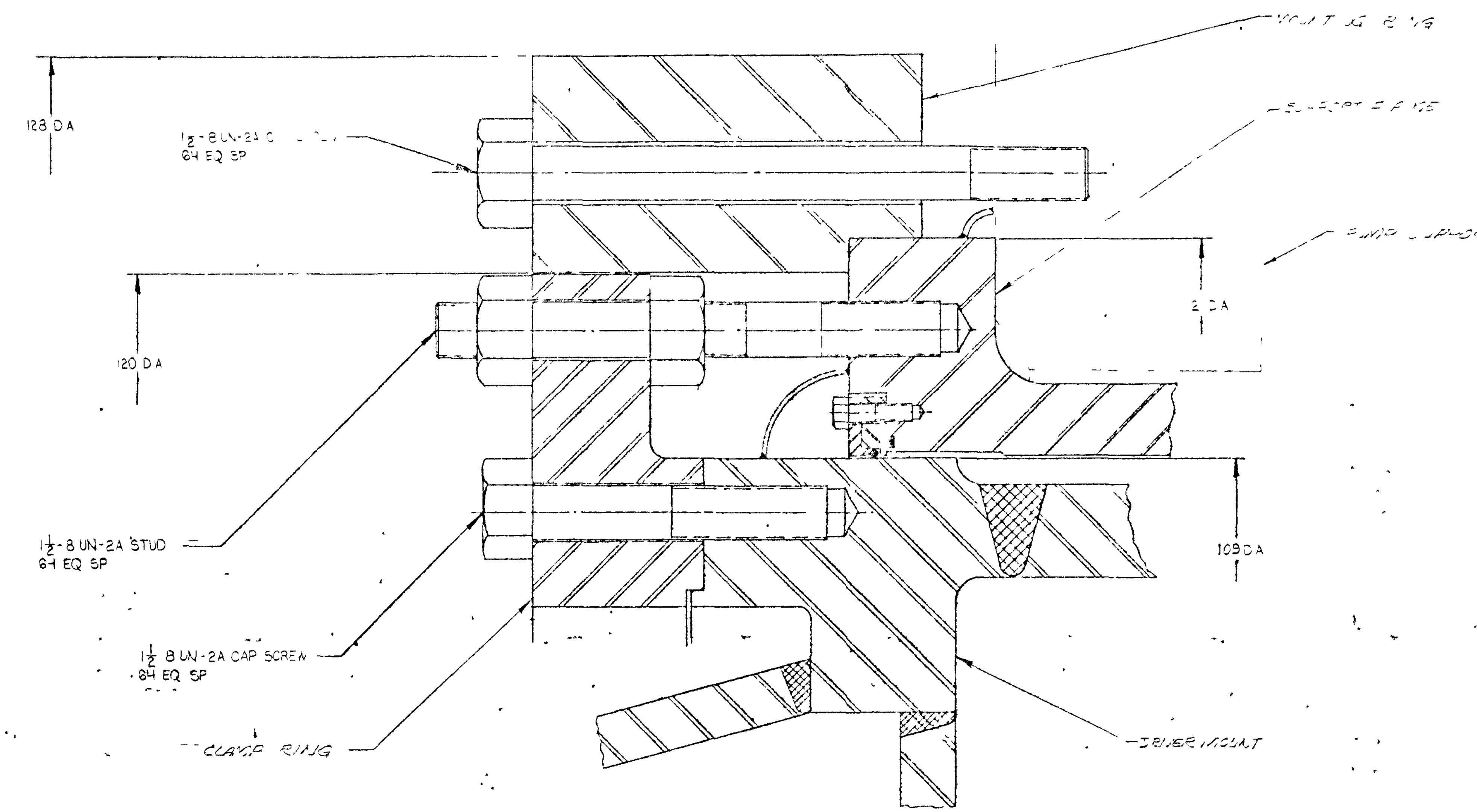
CONTRACT NO.

CODE IDENT NO.	DRAWING NO.	REV.
	SK-0002-103	Ø

SCALE 3/4"=1'-0"	WEIGHT	SHEET

AEF

T.R.G. CLASSIFICATION



BJ ORDER NO. 78 3 00 2	REF NO.	BYRON JACKSON			
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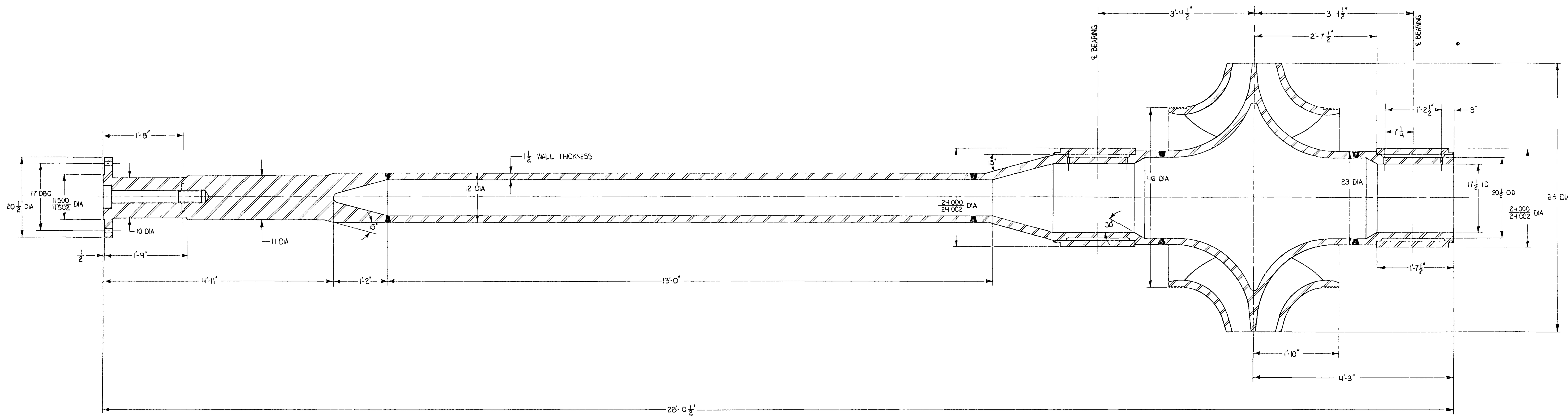
CHECKED BY	DATE				
DESIGN APP'D BY	DATE				
TITLE		CONTRACT NO			
		CODE IDENT NO	DRAWING NO		
		SCALE	WEIGHT	SHEET	

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS
XX ±
XXX ±
XXXX ±

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M. SECTION D2
UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN
DIAMETERS: BORE + 1/64, TURN − 1/64
LENGTH: 0 TO 2 FT — 1/64, 2 TO 6 FT — 1/32, 6 FT AND OVER — 1/16

DISTRIBUTION	ARG	AUST	CAN	HOL	JAPAN	LA	MEX	USA	ORIGINATOR
DATE									

REV	DESCRIPTION
Ø	INITIAL ISSUE



UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS / ANGLES
SURFACE ROUGHNESS PER MIL STD 10
DRILLED HOLES AS DRILLED
TAPPED HOLES AS TAPPED
DO NOT SCALE DRAWING.
REMOVE ALL BURRS AND SHARP EDGES
ALL MACHINED CORNERS OR FILLETS

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DATE	30 AUG 76
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TITLE	
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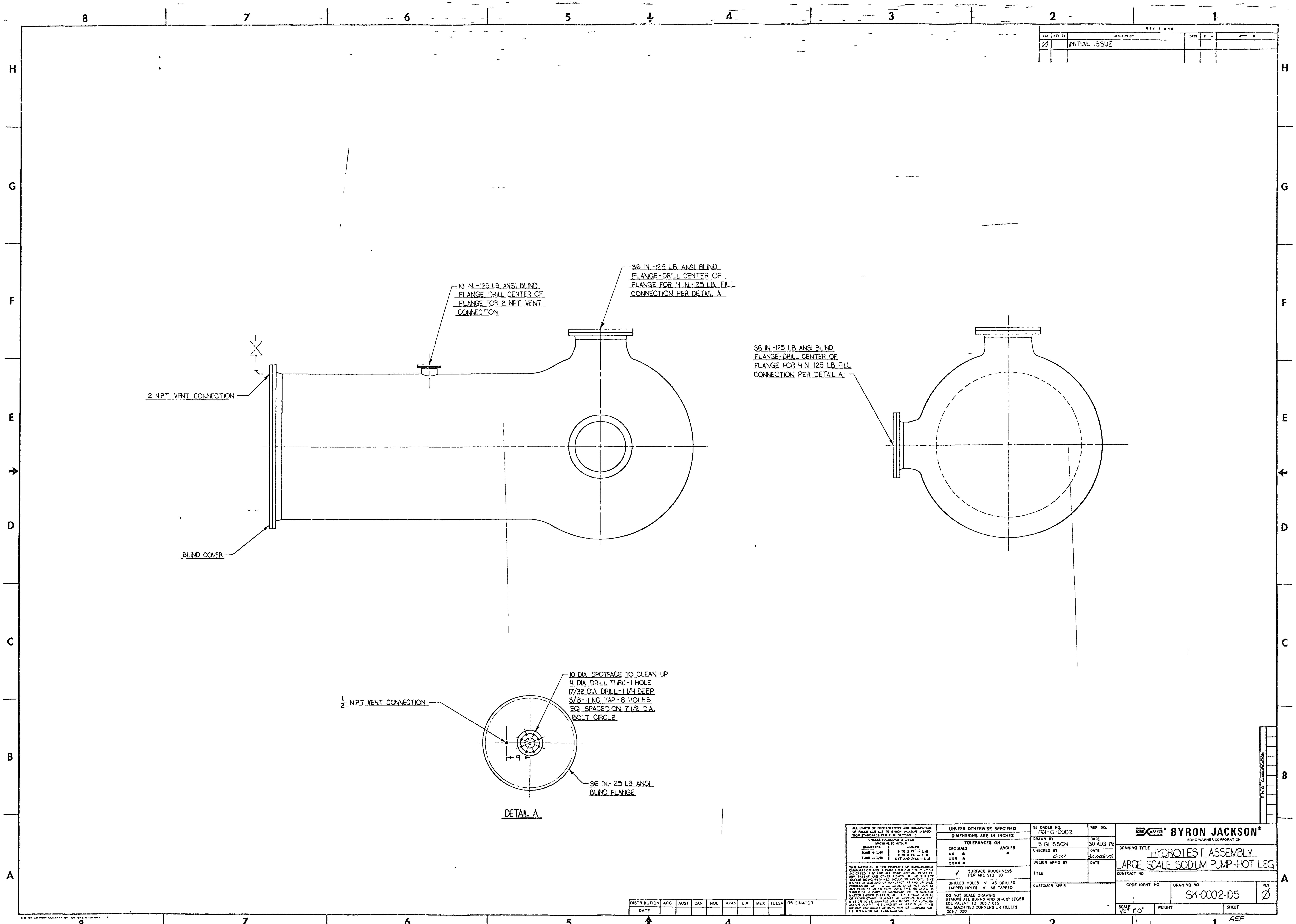
BYRON JACKSON®
BORG WARNER CORPORATION

DRAWING TITLE
PUMP ROTATING ELEMENT
LARGE SCALE SODIUM PUMP-HOT LEG

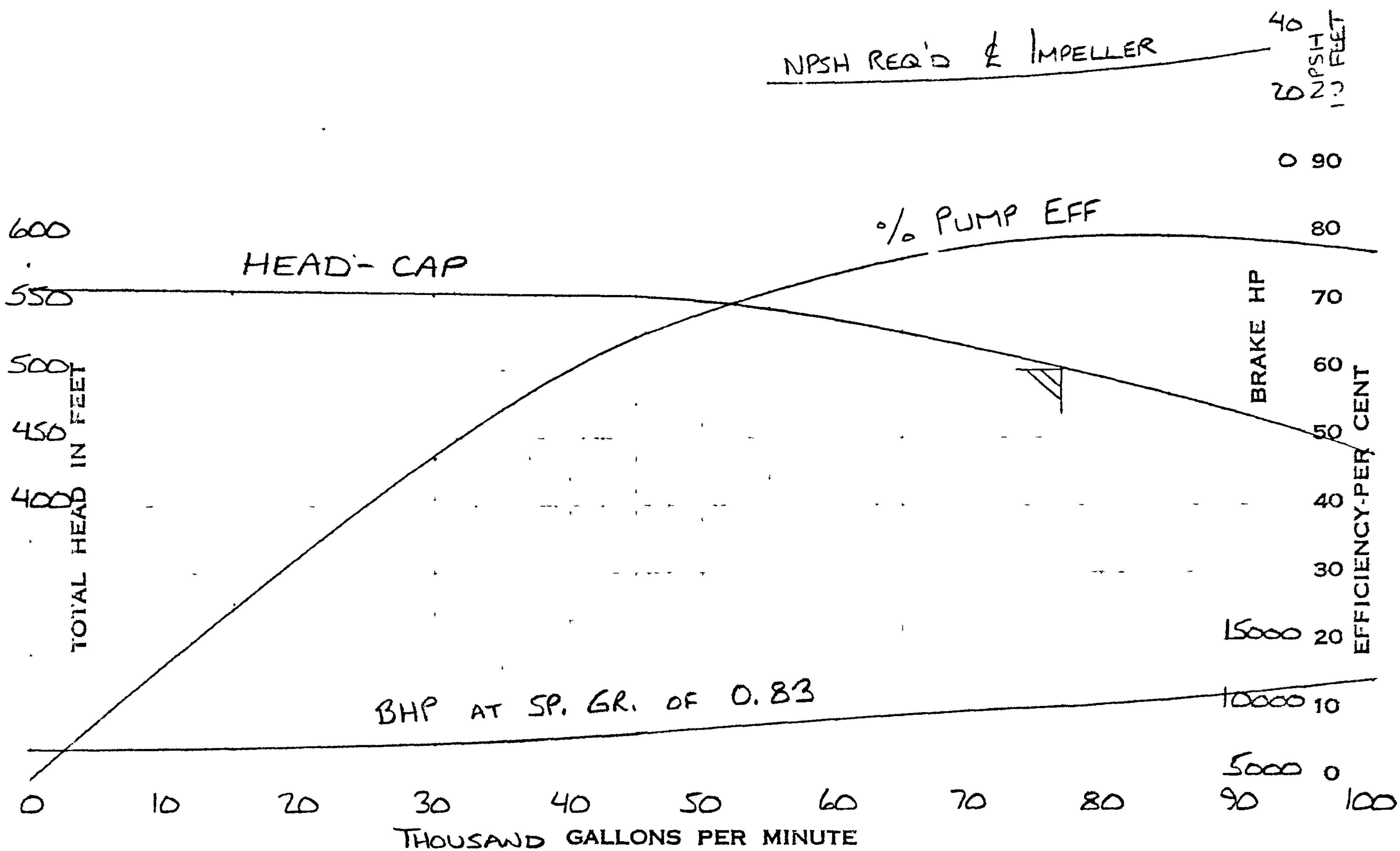
DRAWING NO. SK-0002-104
REV Ø



Ø INITIAL ISSUE
10 IN.-125 LB. ANSI BLIND FLANGE DRILL CENTER OF FLANGE FOR 2 NPT VENT CONNECTION
36 IN.-125 LB. ANSI BLIND FLANGE-DRILL CENTER OF FLANGE FOR 4 IN.-125 LB. FILL CONNECTION PER DETAIL A
2 NPT VENT CONNECTION
BLIND COVER
36 IN.-125 LB ANSI BLIND FLANGE-DRILL CENTER OF FLANGE FOR 4 IN. 125 LB FILL CONNECTION PER DETAIL A
1/2 NPT VENT CONNECTION
10 DIA. SPOTFACE TO CLEAN-UP
4 DIA. DRILL THRU-1 HOLE
17/32 DIA. DRILL-1 1/4 DEEP
5/8-11 NC TAP-8 HOLES
EQ. SPACED ON 7 1/2 DIA. BOLT CIRCLE
36 IN.-125 LB ANSI BLIND FLANGE
DETAIL A
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS
ANGLES
SURFACE ROUGHNESS PER MIL STD 10
DRILLED HOLES V AS DRILLED
TAPPED HOLES V AS TAPPED
DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005 / .015
ALL MACHINED CORNERS OR FILLETS .005 / .020
BJ ORDER NO. 761-G-0002
DRAWN BY S GLISSON
DATE 30 AUG 76
CHECKED BY LW
DATE 30 AUG 76
DESIGN APP'D BY
TITLE
CUSTOMER APP'R
BORG WARNER BYRON JACKSON®
BORG WARNER CORPORATION
DRAWING TITLE HYDROTEST ASSEMBLY LARGE SCALE SODIUM PUMP-HOT LEG
CONTRACT NO
CODE IDENT NO
DRAWING NO SK-0002-105
REV Ø
SCALE 1/2" 1'0"
WEIGHT
SHEET
DISTRIBUTION ARG AUST CAN HOL JAPAN L.A. MEX TULSA ORIGINATOR
DATE

P200R

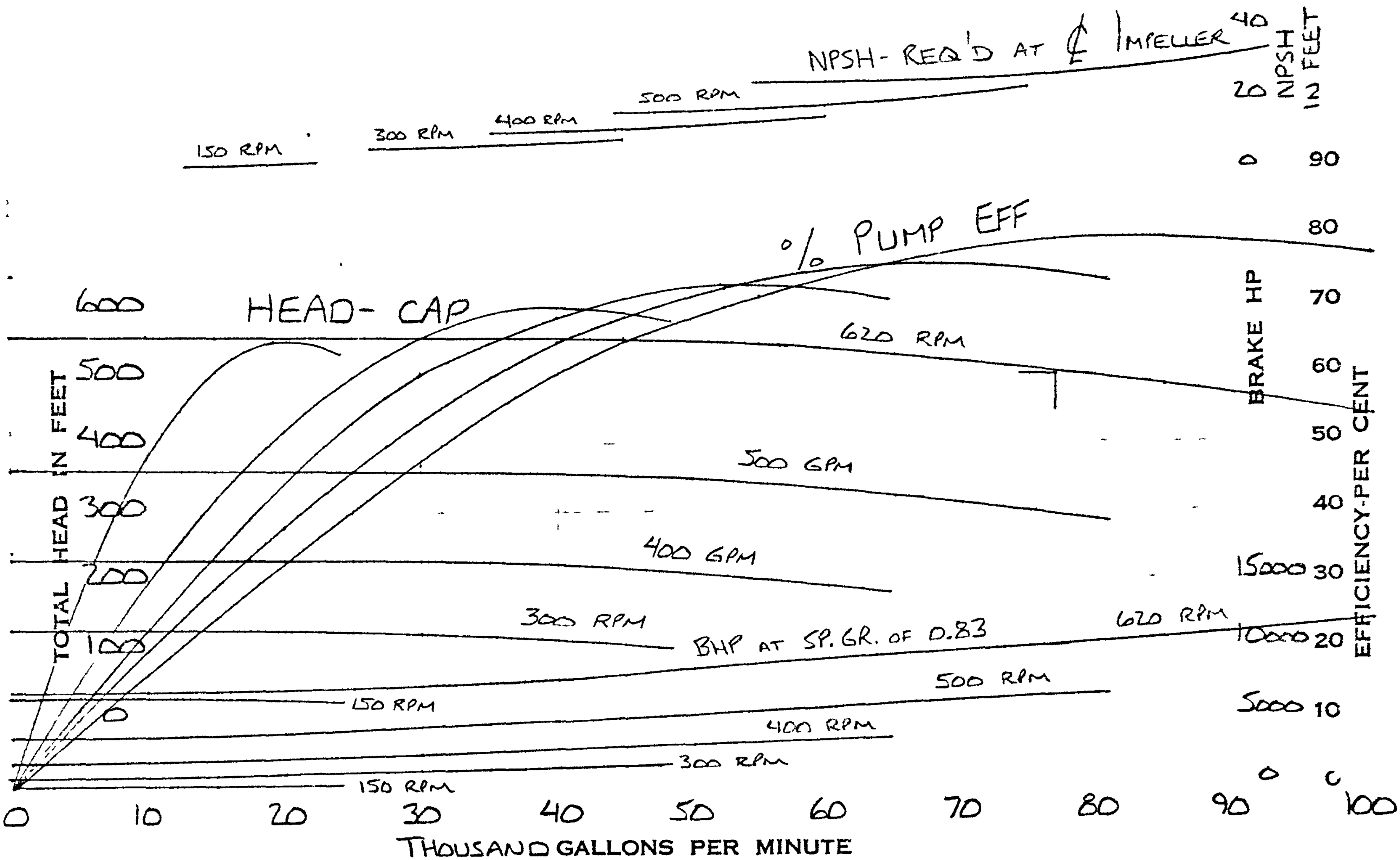
BYRON JACKSON



PUMP SIZE AND TYPE	RPM	CUSTOMER NO	IMPELLER NO	BASED ON	DATE	BYRON JACKSON NUMBER
LARGE SCALE SODIUM PUMP	620	G. S. 761-6-0002	℄ = 3.88	T-30778-1	30 AUG 76	PC 33004
			BRANCH NO	DATA BY MA	DRAWN BY MA	

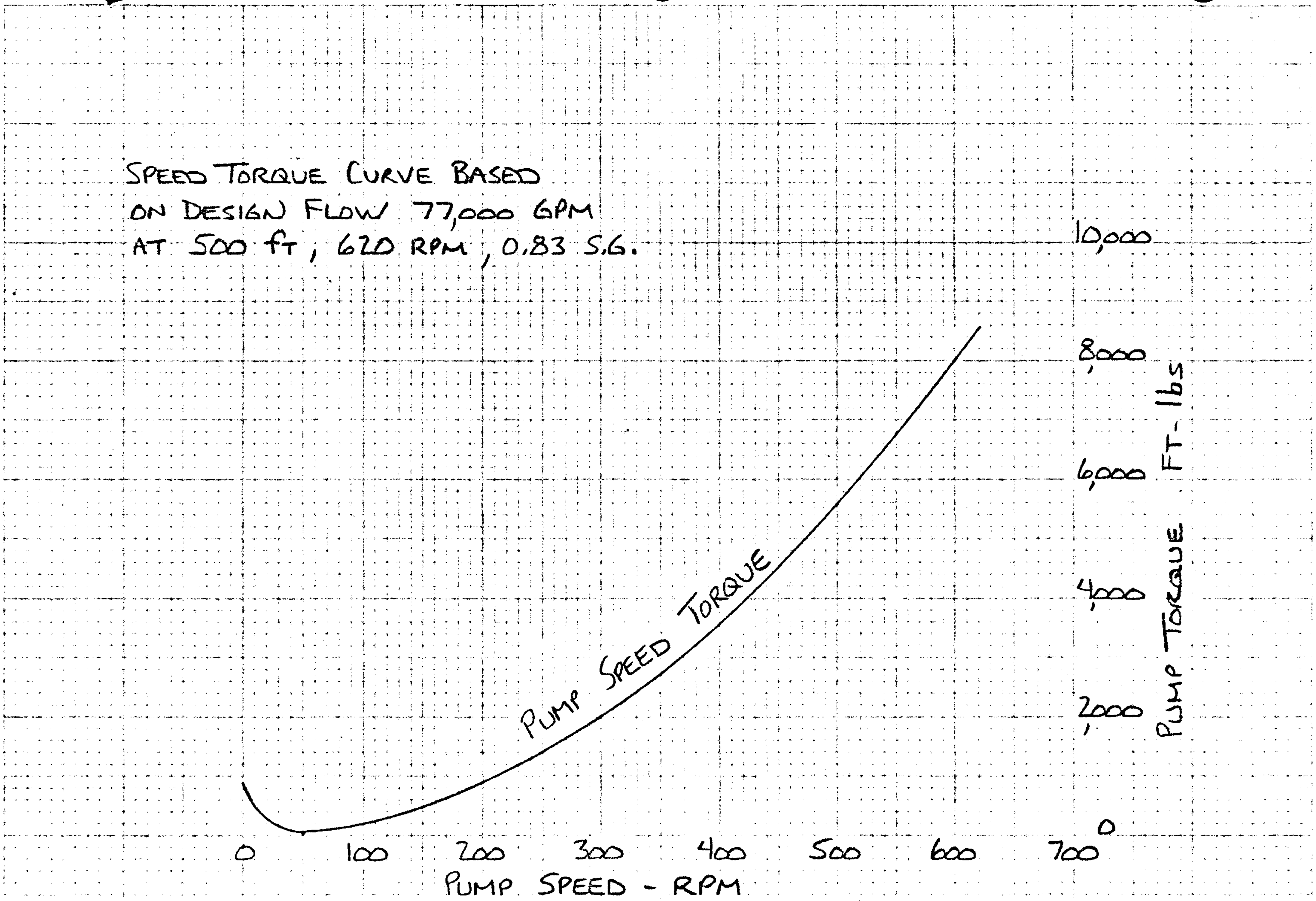
P200R

BYRON JACKSON



PUMP SIZE AND TYPE	RPM	CUSTOMER NO	IMPELLER NO	BASED ON	DATE	BYRON JACKSON NUMBER
LARGE SCALE SODIUM PUMP	VARIOUS		f=3.88	T-30778-1	30 AUG 76	PC 33004-1
		G.S.761-6-0002	BRANCH NO	DATA BY: MA	DRAWN BY: MA	

Byron Jackson Pump Division



LARGE SCALE SODIUM PUMP	PUMP SPEED TORQUE CURVE	DATA BY: MA	JOB NO. 761-6-0002	PC-33004-2
		DRAWN BY: MA	DATE: 31 AUG 1976	

Weights of Pump Sub-Assemblies

		WT. (lbs)
1.	Outer Tank	101,700
2.	Inner Barrel	20,500
3.	Suction Piece (2)	26,000
4.	Volute Assembly	47,000
5.	Rotating Assembly	17,700
6.	Reflector and Insulation Assembly	26,000
7.	Seal Assembly	4,000
8.	Driver Mount Assembly	40,000
	Total	282,900 lbs.

Sodium Inventory (Sodium filled to Overflow Nozzle) 51,800 lbs.

Axial Thrust Requirements imposed by Pump:

Pump Speed RPM	Axial Thrust * Pounds	Axial Thrust Direction
620	36,700	Down
150	18,700	Down
60	17,700	Down

* All thrusts include weight of pump rotating element, 17,700 lbs.

PARTS LIST AND MATERIALS OF CONSTRUCTION

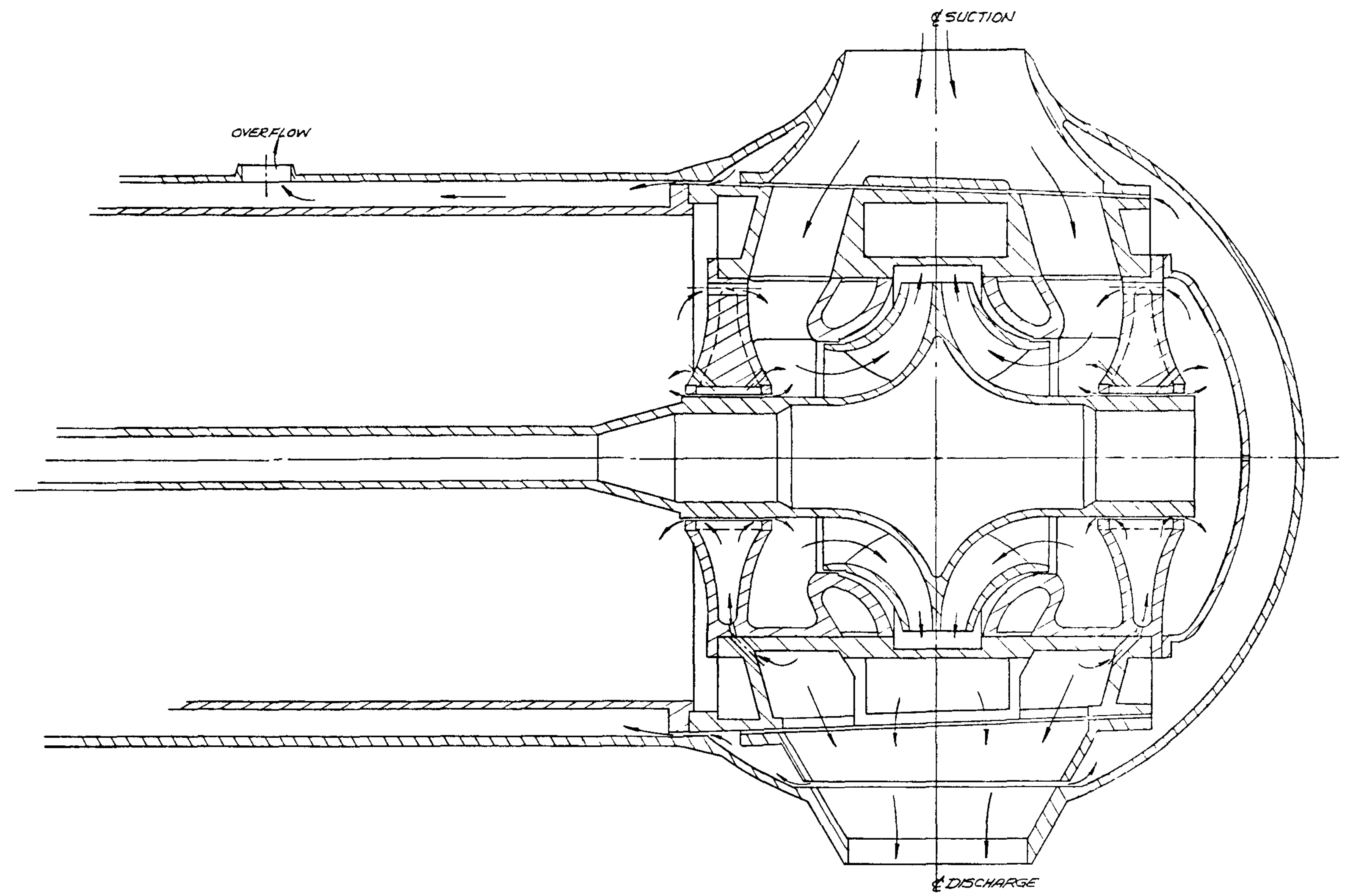
LARGE SCALE BREEDER REACTOR PLANT HOT LEG MECHANICAL PUMP
B.J. ORDER NO. 761-G-0002
REFERENCE DRAWINGS: 1F-8085, 1C-3571

PART NO.	QTY	NAME	MATERIAL SPECIFICATION
1		Pump Tank Assembly	
1-1	1	Tank Mounting Flange	ASME SA-182 Tp. F304
1-2	1	Tank Cylinder	ASME SA-240 Tp. 304
1-3	1	Nozzle - Overflow	ASME SA-182 GR. F304
1-4	1	Transition Ring	ASME SA-182 GR. F304
1-5	1	Suction Nozzle	ASME SA-182 GR. F304
1-6	1	Discharge Nozzle	ASME SA-182 GR. F304
1-7	2	Hemisphere	ASME SA-182 GR. F304
1-8	1	Tank Sealing Cone	ASME SA-351 GR. CF8
2		Inner Tank Assembly	
2-1	1	Inner Tank Cylinder	ASME SA-240 Tp. 304
2-2	1	Lower Flange	ASME SA-182 GR. F304
2-3	1	Upper Flange	ASME SA-182 GR. F304
3		Suction and Bearing Assembly	
3-1	2	Suction and Bearing Piece	ASME SA-351 GR. CF8
3-2	4	Bearing End Ring	ASME SA-182 GR. F304 with #6 Colmonoy Overlay
3-3	1	Bearing Cover	ASTM A-240, Tp. 304
3-3-1	1	Flange - Bearing Cover	ASTM A-182 GR. F304
4		Volute Assembly	
4-1	1	Volute	ASME SA-351 GR. CF8
4-2		Hex Hd. Cap Screw	ASME SA-193 GR. B8
4-3		Hex Hd. Cap Screw	ASME SA-193 GR. B8

PART NO.	QTY	NAME	MATERIAL SPECIFICATION
5		Rotating Element Assembly	
5-1	1	Impeller	ASTM A-351, GR CF8
5-2	1	Bottom Stub Shaft	ASTM A-182, GR. F304
5-3	2	Journal	ASTM A-213, Tp. 304 with #5 Colmonoy Overlay
5-4	1	Upper Stub Shaft	ASTM A-182, GR. F304
5-5	1	Shaft Tube	ASTM A-213, Tp. 304
5-6	1	Top End - Shaft	ASTM A-182, GR. F304
5-7	1	Slinger	ASTM A-182, GR. F304
5-8	1	Bolt	ASTM A-540, GR. B23 Cl. 4
6		Reflector and Insulation Assembly	
6-1	1	Outer Cylinder	ASTM A-240, Tp. 304
6-2	1	Inner Cylinder	ASTM A-240, Tp. 304
6-3	1	Top Plate	ASTM A-240, Tp. 304
6-4	1	Bottom Plate	ASTM A-240, Tp. 304
6-5		Thermal Shield	
6-6		Insulation	
7		Seal Assembly	
7-1		Seal Cartridge	
7-1-1	1	Seal Flange - Upper	ASME SA-182, GR. F304
7-1-2	1	Seal Flange - Lower	ASME SA-182, GR. F304
7-1-3		Seal Housing Assembly	
7-1-3-1	1	Top Flange	ASTM A-240, Tp. 304
7-1-3-2	1	Bottom Flange	ASTM A-240, Tp. 304
7-1-3-3	1	Inner Cylinder	ASTM A-240, Tp. 304
7-1-3-4	1	Outer Cylinder	ASTM A-240, Tp. 304
7-1-3-5	144	Vertical Tube	ASTM A-213, Tp. 304
7-1-3-6	6	Horizontal Tube	ASTM A-213, Tp. 304
7-1-4	1	Pumping Ring	ASTM A-182, GR. F304
7-1-5	1	Stand Pipe	ASTM A-182, GR. F304
7-1-6	2	Retainer	ASTM A-182, GR. F304

PART NO.	QTY	NAME	MATERIAL SPECIFICATION
7-1-7	2	Rotating Face	Carbon
7-1-8	2	Stationary	316 SST with #1 Stellite Overlay
7-1-9	2	U-Cup	Nitrile
7-1-10	2	U-Cup Follower	304 SST
7-1-11	72	Spring	Inconel X-750
7-1-12	2	Seat Gasket	Nitrile
7-1-13	1	O-Ring	Nitrile
7-1-14	2	O-Ring	Nitrile
7-1-15	1	O-Ring	Nitrile
7-1-16	1	Stub Shaft	ASTM A-182, GR. F304
7-1-17	1	Adjusting Cap	ASTM A-182, GR. F304
7-1-18	12	Cap Screw - Hex. Hd.	ASTM A-193, GR. B8
7-1-19	12	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
7-1-20	8	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
7-1-21	8	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
7-1-22	8	Cap Screw - Soc. Hex. Hd.	ASTM A-193, GR. B8
7-1-23	2	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
7-1-24	2	Lock Plate	ASTM A-240, Tp. 304
7-1-25	1	Drive Key	ASTM A-479, Tp. 304
7-1-26	2	Lock Pin	ASTM A-479, Tp. 304
7-2	2	Fan Cover Half	ASTM A-240, Tp. 304
7-3	8	Cap Screw	ASTM A-193, GR. B8
7-5	8	Lock Washer	Commercial S. Steel
8		Coupling Assembly	
8-1	1	Coupling - Motor Half	ASTM A-351, GR. CA-15 HT R/C 23-30
8-2	1	Thrust Ring	ASTM A-479, Tp. 410 HT R/C 26-34
8-3	1	Split Ring	ASTM A-479, Tp. 410 HT R/C 26-34
8-4	16	Stud	ASTM A-193, GR. B6
8-5	32	Nut	ASTM A-194, GR. 6
8-6	32	Lock Washer	Commercial Steel

PART NO.	QTY	NAME	MATERIAL SPECIFICATION
9		Driver Mount Assembly	
9-1	1	Lower Flange	ASME SA-182, GR. F304
9-2	1	Outer Cylinder	ASME SA-240, Tp. 304
9-3	1	Intermediate Plate	ASME SA-240, Tp. 304
9-4	1	Intermediate Flange	ASME SA-182, GR. F304
9-5	1	Cylindrical Cone	ASTM A-516, GR. 70
9-6	1	Top Flange	ASTM A-105
9-7	1	Inner Cylinder	ASME SA-240, Tp. 304
9-8	1	Stand Pipe	ASME SA-182, GR. F304
9-9	1	Outer Cylinder - Oil Drain	ASME SA-240, Tp. 304
9-10	1	Inner Cylinder - Oil Drain	ASME SA-240, Tp. 304
9-11	1	Middle Plate - Oil Drain	ASTM A-240, Tp. 304
9-12	1	Top Plate - Oil Drain	ASTM A-240, Tp. 304
9-13	64	Hex Hd. Cap Screw	ASTM A-193, GR. B7
10		Mounting Support	
10-1	1	Clamp Ring	ASTM A-105
10-2	1	Mounting Ring	ASTM A-105
10-3	1	O-Ring Holder	ASTM A-182, GR. F304
10-4	1	O-Ring Retainer	ASTM A-182, GR. F304
10-5	1	O-Ring	Nitrile
10-6	1	O-Ring	Nitrile
10-7		Hex Hd. Cap Screw	ASTM A-193 GR. B8
10-8	64	Hex Hd. Cap Screw	ASTM A-193 GR. B7
10-9	64	Stud	ASTM A-193 GR. B7
10-10	128	Hex Hd Nut	ASTM A-194 GR. 7
10-11	64	Hex Hd. Cap Screw	ASTM A-193 GR. B8
10-12	1	Canopy Seal, Pump	ASTM A-240 Tp. 304
10-13	1	Canopy Seal, Tank	ASTM A-240 Tp. 304



REVISIONS					
LTR	REV BY	DESCRIPTION	DATE	CHK'D	APPROVED

T N O CLASSIFICATION

DISTRIBUTION	ARG	AUST	CAN	HOL	JAPAN	L A	MEX	TULSA	ORIGINATOR
DATE									

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER E. M SECTION 02 11

UNLESS TOLERANCE IS GIVEN MACHINE TO WITHIN

DIAMETERS	LENGTH
BORE + 1/64	0 TO 2 FT. — 1/64
TURN — 1/64	2 TO 6 FT. — 1/32
	6 FT AND OVER — 1/16

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON
DECIMALS ANGLES
XX ±
XXX ±
XXXX ±

✓ SURFACE ROUGHNESS PER MIL STD - 10

DRILLED HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

DO NOT SCALE DRAWING
REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO 005 015
ALL MACHINED CORNERS OR FILLETS 005 / 020

BJ ORDER NO. 761-G-0002	REF NO
DRAWN BY LW	DATE 15 SEP 76
CHECKED BY	DATE
DESIGN APP'D BY	DATE
TITLE	
CUSTOMER APP R	

BORG WARNER® BYRON JACKSON®
BORG WARNER CORPORATION

DRAWING TITLE SCHEMATIC DIAGRAM
LARGE SCALE SODIUM HOT LEG PUMP

CONTRACT NO

CODE IDENT. NO.	DRAWING NO.	REV.
	SK-0002-107	0
SCALE	WEIGHT	SHEET

AEF

23. PROGRAM PLANS, SCHEDULES AND COST

23.1 Schedules

The conceptual design has been detailed to establish engineering, procurement, and manufacturing time spans. The program overview shows all major milestones of design, development, procurement of materials, manufacture of components, assembly, water test and shipment of one prototype pump and three production plant pumps.

Detail schedules of facility modification, tool and fixture design and fabrication have not been included. The major effort has been directed into the technical overview. With long lead times until start of fabrication there is adequate time for facility improvement or construction.

23.2 Cost

The order of magnitude cost estimates of the prototype pump is $27 Million requiring eight years from award to delivery.

Additional units are approximately $9 Million each requiring approximately three and one-half years from release of material to delivery.

Prototype cost includes development engineering, manufacturing and test facilities and fixtures.

Costs are based on 1976 dollar estimates and material deliveries.

BORG WARNER BYRON JACKSON
BORG-WARNER CORPORATION

LARGE SCALE BREEDER REACTOR PLANT - HOT LEG
PROGRAM PLANS AND SCHEDULES