OVERVIEW OF NSF-TRC SQUEEZE FILM DAMPER RESEARCH PROJECT (YEAR 1)

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May 1995

TRC-RD-4-95

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Overview of NSF–TRC Squeeze Film Damper Research Project (Year I)

by

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A Research Progress Report
to the
Turbomachinery Research Consortium

May 1995

TRC Project:
Experimental and Analytical Study of the Non–Linear Response of Squeeze Film Damper Supported Rotors.
Principal Investigator: Dr. Luis San Andres, Associate Professor

Abstract

Users of turbomachinery want to push the performance limits of earlier machines. This means machinery with higher power to weight ratios and improved efficiencies requiring more flexible, light weight rotors running at higher speeds. These operating conditions and configurations make for a machine that is prone to damaging vibration affecting machine reliability. A vital part of a modern turbomachine, such as a jet engine, is a squeeze film damper (SFD) to isolate structural components and provide viscous damping forces that dissipate vibrational energy by reducing rotor vibration amplitudes. The National Science Foundation (NSF) and the Turbomachinery Research Consortium (TRC) sponsor a research program to investigate the effects of SFD's on the rotordynamics of a rotor-bearing system. This 3 year program, started in August 1994, will test a rotor supported on SFD's to obtain an understanding of the interactions between the SFD and the rotor-bearing system. The experimentation will be conducted with a pair of conventional design SFD's and then a pair of KMC integral SFD's for a comparison. Presently the program is approaching the end of its first year-long phase. An analysis performed to predict the critical speeds of the test rotor was completed. Two rigid mode critical speeds lie within the selected operating range. The test rig has been designed and the parts requiring machining or fabrication are being worked. The components of the test rig such as the instrumentation, motor, and lubrication system have been bought or are on order for delivery by June 1995. Test rig assembly starts in June 1995 with trouble shooting scheduled for mid-August 1995 (Phase II).

Introduction

Modern high performance turbomachinery, built to improve both power output and efficiency, trends towards lighter, faster, and more flexible designs. Turbomachines then operate under conditions that leave them increasingly sensitive to potentially damaging vibrations, and therefore a reliability risk. An essential tool for controlling these vibrations is the squeeze film damper (SFD). A squeeze film damper, if properly designed, can isolate the structural components and dissipate the vibrational energy without any excessive loads on the machine or affecting useful output. With the vibrations controlled, the turbomachine can push the performance envelope with improved reliability.

SFD's have a complex dynamic behavior and analysis can not predict this with accuracy. Presently, SFD's are designed using over-simplified models that may ignore vital structural and fluid dynamics effects. An example is the π-film cavitation model used with many SFD analytical solutions (Mohan and Hahn, 1974; Li et al, 1987; San Andres et al, 1988). This model has given results that match poorly with some experimental results (Zhang, 93). The result of a theory-based SFD design is erratic behavior that may render the design ineffective in its application. This incomplete understanding may stem from a lack of experimental data and actual experience with SFD's

Squeeze film dampers are used primarily in aircraft jet engines to provide viscous damping to rolling element bearings which otherwise would contribute no appreciable amount of damping to the rotor-bearing system (Vance, 1988). Manufacturers and users of high performance compressors have installed SFD's in series with tilting-pad bearings to provide an extra margin of stability (San Andres, 1994). These experiences demonstrate SFD's to have strongly nonlinear behavior in certain operating regimes. Beyond these experiences, however there has been little research into the complex effects of the SFD's fluid and mechanical interactions coupled with the rotordynamics of an entire rotor-bearing system.

A further step to expand the experience with SFD's is the inclusion of the KMC integral SFD in the experimentation phase of the NSF sponsored research program. The integral SFD is an ingenious design based on the modern machining process of wire Electrical Discharge Machining (EDM). The integral SFD has squeeze film lands in series with pads supported on flexible beams, all cut from a single piece of metal. The separate "cells" (pad, flexible beams, and squeeze film) allow the damper to be designed as a split unit and easily installed in an existing machine without affecting the existing configuration, much as many retrofit tilting-pad bearings. This requirement also conforms with American Petroleum Institute (API) specifications. The integral SFD will be detailed later.

Previous Studies (San Andres, 1994)

In its most simple form a squeeze film damper consists of an inner non-rotating journal and a stationary outer bearing, both of nearly identical radius. Figure 1 shows an idealized schematic view of this type of fluid film bearing. The journal is mounted on the external race of the rolling element bearing and prevented from spinning with loose pins or a squirrel cage which also acts as a centering spring mechanism. The annular gap between the journal and housing is filled with some type of lubricant provided as a splash from the roller bearing lubrication or by a pressurized delivery. In operation, as the journal moves due to dynamic forces acting on the system, the fluid is displaced to accommodate these motions. As a result, generated hydrodynamic pressures exert fluid film forces on the journal surface and provide for a mechanism to attenuate transmitted forces and reduce the rotor amplitude of motion. The amount of damping produced is the critical design consideration. If damping is too large the SFD acts as a rigid constraint to the rotor-bearing system with large forces transmitted to the supporting structure. If damping is too light, the damper is ineffective and likely to permit large amplitude vibratory motion with possible sub-harmonic resonances.

SFD's in practice operate with low levels of external pressurization which generally does not prevent the lubricant in the fluid film lands from liquid vaporization or entrainment of external gaseous media into the film lands. Investigations of lubricant cavitation in squeeze film dampers has been mostly experimental with results showing it to be a phenomena of extreme complexity with a profound impact on the forced performance of the dampers tested (Walton et al., 1987, Sun et al., 1992,1993). Hibner and Bansal (1979) showed first that under the influence of fluid cavitation, the pressure fields and fluid film forces were of a unique character and did not correlate well with predictions based on classical lubrication theory. Walton et al. (1987) found that fluid cavitation onset and extent depend on the damper operating parameters like whirl frequency, journal orbit size and eccentricity position, level of supply pressure and end seal restrictions. Zeidan and Vance (1989a,b,c,1990) have identified five regimes of cavitation in a SFD according to operating conditions. The experiments demonstrated that air entrance produces gaseous cavitation which under certain conditions determines a nonlinear forced response akin to that of a soft spring. On the other hand, vapor cavitation was shown to lead to a characteristic nonlinear hardening effect. Sun et al. (1992, 1993) discuss the major differences between dynamic gaseous and vapor cavitations and the likelihood of their existence in a typical practical application. The significance of these experimental investigations cannot be overlooked since they have already prompted significant changes in the current philosophy of damper design.

In many practical circumstances, dampers are designed with a feeding central groove to insure a continuous flow of lubricant through the squeeze film lands. The groove, usually of large volume, is thought to provide a uniform flow source with constant pressure around the journal surface. The groove also divides the flow region into two separate dampers operating independently. For the groove-SFD configuration, it is well known theoretically that the level of force obtained is one fourth that available for a damper with twice the land length. In practice, however, grooved-dampers have been observed to perform much better than current theoretical predictions (Holmes, 1990). Large dynamic pressures have been measured at the groove regions connecting the two squeeze film regions as reported by San Andres et al.(1987) and also Zeidan et al.(1989b). Roberts et al. (1986,1990), Ramli et al. (1987) and Rouch (1990) have presented experimental force coefficients for a grooved squeeze film damper which are, for all tested eccentricities, an order of magnitude larger than predictions from the short bearing model. It is clear then that central grooves in dampers do not isolate the adjacent film lands, but rather interact with the squeeze film regions and are capable of producing an appreciable dynamic pressure and force response. Current heuristic explanations for the dynamic effect observed at grooved dampers have referred to fluid inertia and turbulence effects, geometric discontinuities, fluid compressibility, or a combination of all these factors, as the primary sources for the occurrence of the phenomena. Recent detailed experiments and analyses carried out by San Andres (1992) and Arauz et al. (1993a,b) have elucidated with rigor the complex flow interactions between feeding groove volumes and the squeeze film lands. The experiments showed measured pressure fields at deep grooves and film lands to be of the same order of magnitude, and where the groove contributed greatly to the overall damping force performance of the damper. For uncavitated films, the experimental measurements correlated favorably with predictions from this new model and brought to question the typical assumptions leading to the conventional theory of squeeze film flows.

The importance of fluid inertia in the performance of squeeze film dampers has been demonstrated by numerous theoretical and experimental investigations. The relevance of fluid inertia is related to the squeeze film Reynolds number ($Re_s=\rho\omega c^2/\mu$) which ranges from 1 to 50 in most practical applications. Theoretical advances on the modeling of high speed dampers have been made by Tichy et al.(1978), and San Andres and Vance (1986). Experimental works relevant to the understanding of SFD performance with fluid inertia effects are given by Vance et. al. (1975), Tichy et al. (1984), San Andres et al. (1987, 1993), Ramli and Roberts (1986), Roberts et al.(1989,1990), Kinsali and Tichy (1989), and Jung et al.(1990). Some of these experimental works have addressed directly the effect of fluid inertia, while others were interested in the identification of the damper dynamic force characteristics by especial methods. Correlation of experimental measurements with analytical predictions including fluid inertia effects have ranged from poor to adequate. In general, measured results seem to be highly dependent on actual operating conditions, test hardware configuration and coupling to the dynamics of the structural system.

In the dynamic analysis of rotor-bearing systems, SFD's are regarded as highly non-linear mechanical elements providing forces obtained from relationships based on the instantaneous journal center eccentricity. Current analyses of rotor-disk assemblies supported on SFD's are based on overly simplified analytical expressions for fluid film forces as derived from the short journal bearing model with the so called π-film cavitation assumption (Mohan and Hahn, 1974, Li et al.1987, San Andres et al., 1988). Computational predictions based on direct numerical integrations, and lately analytical solutions based on current non-linear dynamics models with elegant mathematical methods, have shown that the forced response of rotor-SFD's systems is highly non-linear with extreme sensitivity to unbalance levels. Little effort has been placed on finding a combination of operating and design parameters which will avoid the damper undesirable response. On the other hand, the richness of the non-linear (theoretical) behavior has been exploited to prove beyond practical limits the amazing accuracy of the non-linear dynamics models. It is however noted that there is little physical evidence and practical experience attesting to the veracity of the theoretical predictions. In the last five years, 1990-94, over 24 journal and conference publications have presented extensive theoretical non-linear dynamics treatments of the problem, and only one publication has investigated the phenomenon by experimental means. No analytical work has provided a sound set of conditions (of practical value) for the development of an experiment directed to validate (or disprove) the theoretical findings.

The work of Zhang et al. (1993) is praised for providing fundamental experimental evidence that shows current theoretical models fail to predict with any degree of accuracy the performance of actual rotor-damper hardware. Zhang et al. found that the π-film model is unable to correlate with measurements which showed a larger level of damping than expected. In essence, the test rotor-damper system demonstrated a complete absence of jump phenomena and aperiodic responses. However, Zhang et al. also failed to understand the nature of the flow in SFD's and stretched the limits of application of the conventional models to the absurd. As quoted from their work, "the theoretical model need to be used with cavitation pressures as low as -700 KPa (-37 psi) absolute to provide some correlation with the measurements". This assumption allows the lubricant to sustain large levels of tension in a open ends damper configuration. Pressure measurements were performed but regrettably not reported at length. Zeidan et al. (1989a,b,c), and Sun (1993) showed that entrained air is the most likely form of cavitation in open ended dampers, and thus, lubricant pressures below ambient conditions are not likely to occur. It appears then that the preferred non-linear flow model, namely the π-film short length SFD, used in advanced rotordynamic studies has little relationship with reality.

The research program on squeeze film dampers at Texas A&M University has lead to a number of innovations in damper technology and contributed greatly to a better understanding of squeeze film flows. The research effort was funded continuously from 1983 to 1990 with industrial support from the Turbomachinery Research Consortium and the Rotordynamics System Group at General Electric Co. The experimental work concentrated on the measurement of pressure fields and fluid film forces in damper apparatus with constrained journal motions of the circular type. The analytical work focused on developing sound theoretical models and efficient computational tools for the prediction of SFD pressures and dynamic forced responses. State-of-the-art analyses addressed the effects on damper behavior of fluid inertia, fluid compressibility, end seals and inlet feeding devices. Numerous experimental versus theoretical correlations as well as discussions of relevant issues are detailed in over 21 technical reports and 24 journal publications. The continued research in this field has left a number of open issues related to the fundamental mechanics of squeeze film flows. Among these, dynamic cavitation and the coupled interaction between dampers and the rotor-bearing system are the most relevant and yet least understood.

Current Funded Work:

A research program to investigate the effects of SFD's on the dynamics of rotor-bearing systems is currently funded by the NSF as a 3 year project started in August 1994. This program includes both prediction and measurement of the dynamic force of rotors supported on SFD's, and the analysis of SFD's flows and their impact on rotor response. The project goals are:

a) To design and build an instrumented rotor-bearing test rig supported on both conventional and integral SFD's.

b) To measure the dynamic force response of the rotor-SFD system under a variety of dynamic load conditions (imbalance response).

c) To develop a mathematically tractable procedure to study the flow characteristics of SFD's and their force response including the effects of dynamic cavitation from bubbly mixture and two phase flow.

d) To provide a reliable data base to allow direct comparison with current analytical treatments for the dynamic response of a rotor-SFD system.

e) To identify future research needs in rotor-bearing system design, analysis and testing.

The Test Rig

The test rig for the NSF/TRC SFD research program consists of 3 major sub-systems.

* Test Rotor System
* Instrumentation
* Support Equipment

Each of the sub-systems is described in detail below. A description of the present status of the program then follows. The test apparatus is located at the Texas A&M University (TAMU) Rotordynamics Laboratory.

Test Rotor System

Figure 2 presents an overview of the test rig. A DC motor powers the rotor through a flexible coupling. The rotor is supported on high precision ball bearings mounted inside SFD's installed within the bearing support housings. The rotor and supports rest on a base plate, and the motor rests on an isolated support. The base plate attaches to the work table by leveling screws at each corner and rests on a vibration absorbing pad. The motor mount's feet have rubber inserts to isolate the motor vibrations.

The motor in Figure 2 is a 15 HP (11.2 kW) DC unit fed by a 10 HP (7.5 kW) DC power supply in the laboratory. The motor was selected based on its speed range and power. The motor spins the rotor up to 8000 rpm through a direct connection to the rotor by a flexible coupling. The flexible coupling prevents misalignment between the motor and the rotor from putting a load on the rotor. Within the coupling is a drawn cup roller clutch that allows the motor to bring the rotor to speed and then be shut off without applying drag to the rotor. This allows for a better coast down test of the rotor as the motor friction will not affect the rate of rotor deceleration.

The test rotor (Figure 3) was taken from a retired tilting-pad bearing test rig existing in the TAMU Engineering Department. The rotor consists of a 26 in (660.4 mm) shaft of diameter 3 in (76.2 mm) except for the last 7 in (177.8 mm) of the drive end, and 5.5 in (139.7 mm) of the free end which are machined to fit the SFD's and the ball bearings. A detail of the shaft ends depicted in Figure 4 shows the fine surface for the ball bearings and the threads for bearing preloading. The shaft has three wheels shrink fit at evenly spaced intervals of 2.5 in (63.5 mm). Two of the wheels are 11 in (279.4 mm) diameter, and the third is 9 in (228.6 mm) diameter. All three wheels are of 1 in (25.4 mm) width. The shaft and wheels are constructed of 4140 steel. The mass of the rotor is approximately 98 lbm (45 kg). The rotor was chosen due to its accessibility and features that make it suitable for this research program. There are provisions for attaching imbalance masses in threaded circumferential holes in each wheel. The general rotor design is an effective inertia model of a jet engine rotor.

The split bearing support housings are modified Centritech housings, also taken from the retired test rig. Figures 5 and 6 show the key dimensions, and the alterations machined into the existing housings. The modifications include a circumferential groove in the mid-length of the support, and two fine bores for use with alignment bars during assembly. The circumferential groove is the seat for the SFD when installed. Besides locating the SFD, the groove acts as a plenum to distribute lubricant to the damper and ball bearings. Note that the housings are split along the center of the bore. This allows a "drop-in" installation of the rotor ends with the SFD's. When installed on the base plate, the housings will mirror each other about an axis perpendicular to the rotor length. The two faces of the housings facing each other are described as the inside faces, and the opposite faces are the outside faces. There is a circumferential ring of holes on the outside face that are attachment points for the support stiffness hardware (squirrel cage, to be described later) of the conventional SFD. The 4 holes on the inside face are mounting points for holders that will position the proximity probes.

Two types of SFD's will be tested, a conventional SFD's and an integral SFD's. The conventional SFD (Figure 7) consists of a journal connected to the housing through flexible beams and press fitted to the rotating shaft through high precision ball bearings. This configuration restricts the journal from rotation but allows it to orbit with shaft vibration amplitude within a stationary outer bearing. The damping action is provided by an oil film between the journal and outer bearing. The SFD is designed with dimensions similar to those used in jet engines, ie: of small length to diameter ratio. The dimensions of the SFD are a land of 0.920 in (23.37 mm) length and a clearance of 4.65 mils (0.116 mm). The radius from center to land is 1.8 in (46.5 mm). The support stiffness (squirrel cage) is designed to be 20,000 lbf/in (1.7E4 N/m), but provisions are made to use different diameter flexible beams to select a variety of support stiffnesses. More details about the support stiffness will be given when a discussion of the rotordynamic analysis is given. The predicted damping values are 23 lbf-sec/in (4179 N*s/m) for direct damping and 3 lbf-sec/in (535 N*s/m) for cross coupled damping coefficients using the formula in equation [1] (Vance, 1988).

$$C_{tt} = \frac{\pi \mu R L^3}{2c^3(1-\epsilon^2)^{3/2}} \qquad [1a]$$

$$C_{rt} = \frac{2\mu R L^3 \epsilon}{c^3(1-\epsilon^2)^2} \qquad [1b]$$

Where:
μ = viscosity , c = radial clearance, R = radius, L = length , ϵ = eccentricity/c

Table 1 describes damper geometry and oil type used and the corresponding predicted damping coefficients. These equations are for open end SFD's assuming a π-film model. There are provisions designed into the SFD for end plate seals. The effect of seals is to increase damping as the seals simulate a longer SFD by stemming axial flow. A fine alignment feature is included in the squirrel cage mounting bracket. When assembled, the alignment screws can be used to move the squirrel cage mounting point and move the SFD to an off-center position.

The integral SFD is a KMC, Inc., product donated to the laboratory. Figure 8 displays a schematic view of a typical integral SFD. The integral SFD designed for testing is shown on Figure 9. The dimensions specified for the test integral SFD are similar to those for the film land of the conventional SFD, including the option for running with end seals. The configuration of the donated pair of integral SFD's is 4 pads. As mentioned previously, the integral SFD can be fabricated as a split unit. However, for this research, a split unit is not mandatory. The ball

Viscosity:	65.9 lbf/ft*s (0.007 Pa*s) for oil SAE 10@100°F (38°C)
Radius:	1.8 in (46.5 mm)
Length:	0.92 in(23 mm)
Clearance:	4.65 mils (0.116 mm)
for ϵ = 0.1	
C_{tt} = 23 lbf s/in (4179 N s/m)	
C_{rt} = 3 lbf s/in (535 N s/m)	
for ϵ = 0.5	
C_{tt} = 36 lbf s/in (6338 N s/m)	
C_{rt} = 26 lbf s/in (4659 N s/m)	

Table 1: Predicted Damping Values

bearings are the same as those used for the conventional SFD's (see next paragraph) and are interference fit with the integral SFD's pads. The structural webs of the integral SFD provide centering of the rotor with an offset equal to the predicted static deflection due to rotor weight. Note that the integral SFD's are very compact. They require only a little more length than the ball bearings, allowing it to fit in existing machines. This suits applications in process-liquid machines where the lubricant for the bearings is a low viscosity process fluid or water. Bearings have low damping in these conditions and an integral SFD installed in series with the bearing provides extra damping while fitting in the same area (Zeidan, 1994).

The ball bearings are high precision SKF bearings. The nominal dimensions are 1.6535 in (42 mm) O.D. and 0.9843 in (25 mm) I.D. with length of (9 mm). The tolerances of these bearings have been verified to be 0.04 mils (1 μm) within specifications. The bearings are mounted as spaced back to back pairs in each SFD. The spacers are rings 0.24 in (6 mm) long and correspond to the inner race diameter and the outer race diameter. These rings simulate the back to back configuration, but allow room for the lubricant supply to flow through the bearings.

For this project, the critical speed analysis was done with the RAPP and PUP software. These programs use the Transfer Matrix Method for the linear critical speed analysis of the rotor on bearings (Vance, 1988). The support stiffness for the conventional SFD configuration is the squirrel cage, consisting of four precisely machined bars that connect the SFD journal to an extension of the support housings. This assembly has a design stiffness of 20,000 lb/in (1.7E4 N/m) at each SFD. This stiffness value is based on the results of the rotordynamic analysis and a design criterion of a first critical speed well below 8,000 rpm. The rotordynamic analysis was run with several support stiffness to find the stiffness that has the critical speed to fit the criterion. For this analysis, the rotor is modeled with 22 stations with the wheel mass and inertia added at their respective positions. The wheels are not modeled directly, since they are shrink fit and will not affect the bending stiffness of the rotor. The results of this analysis (Figure 10) show the first critical speed to be at 4200 rpm. This is a cylindrical rigid rotor mode as shown in Figure 11. A second overdamped rigid conical mode at 6,000 rpm is also present. The results shown also include effects of a linear damper with the coefficients calculated for the SFD's in table 1 with ϵ = 0.1.

Instrumentation

The test rig is well instrumented. This instrumentation includes transducers for rotor position, and acceleration, fluid pressure, and temperature, and rotor speed. All electronic transducers' signals are acquired by a lunch box size personal computer (PC) dedicated for use with the ADRE© 3 data acquisition software from Bently Nevada. This is paired with a ADRE© 208 data acquisition interface. This setup allows the use of eight simultaneously sampled channels of data plus 2 independent keyphasors. With the PC, all the data can be analyzed and presented or imported to other computers for analysis. This software also allows analysis in the frequency domain which is critical for this type of testing.

The instrumentation layout relative to the rotor is shown in Figure 12. The position transducers are non-contact eddy current proximity probes. The test rig has these probes in mounts at two positions, X and Y directions, on the inner face of each support housing. An additional pair of proximity probes are located at mid-span of the rotor. A piezo-electric accelerometer with a magnetic base can be attached to any metal surface to check vibration levels. This data is useful for getting a net vibration level of the rotor, if there is severe vibration of the supporting structure. Pressure transducers are installed in taps provided in both the conventional and integral SFD's. These sensors will measure the fluid film pressure in the film lands. Pressure gauges measure oil feed pressure and monitor the lubrication system as well. Thermocouples measure the oil temperatures before and after the SFD. One optical tachometer will be installed to detect the shaft speed and act as keyphasor.

Support Equipment

The support equipment includes the lubrication system and assorted hardware. The lubrication system has a 40 gallon (371 liters) reservoir, and a gear pump delivers the oil to the test rig through a network of lines and valves that will allow control of the SFD oil feed pressure (Figure 13). A smaller gear pump returns the oil from the SFD to the reservoir. There are plans to include an air-to-oil heat exchanger between the secondary pump and the reservoir. However, at this stage the heat exchanger has not been sized.

The entire test rig rests on a 3' x 4' (0.9 m x 1.2 m) concrete and steel table. This table is from the retired test rig mentioned previously. The table's legs rest on a 5' x 5' (1.5 m x 1.5 m) area steel bed. Butyl foam is sandwiched between the steel plate foundation and the laboratory floor to isolate the building and test rig from vibration. A steel base plate locates the SFD support housings. This is a machined surface that allows for precise location of the test rig and accurate alignment. This plate is mounted to the table with vibration isolation material sandwiched between the plate and the table as seen in Figure 2. A separate mount supports the motor and is adjustable in X, Y, and Z directions to align with the test shaft. This mount is also vibration isolated through rubber feet. The mid-span proximity probes are located on a probe stand designed to also catch the shaft should an accident occur that allows the shaft to orbit uncontrollably. Another related safety feature is a steel enclosure fitted to the test table.

Present Project Status

A significantly more modest experimental research program has been sponsored by the TRC since August 1993. This project has recently been completed and a description of the work is given in a TRC report (San Andres et al, 1994b). It is noted that funds from this project were dedicated to the support of a graduate student.

In the first year of the SFD research program, all of the test rig components listed in **Table 2** have been purchased, sent out for fabrication, or ordered for delivery in time for June 1995 assembly.

Shaft: (Figure 3)
- 26" in length
- 3 disks
- shaft diameter 3"

Bearing/SFD supports (Figures 5&6)
- provide location for bearings
- anchor SFD support stiffness

Base Plate
- provides location and anchoring for hardware

Foundation Table
- Rigid table of concrete and steel
- mounted on steel plates
- vibration isolated

Oil Reservoir and Pump (Figure 13)

DC motor (Figure 2)
- 15 HP
- 8000 rpm
- motor mount

Controllable DC power supply

Instrumentation and Data Acquisition System (Figure 12)

SFD's (conventional & integral)

Table 2: List of Components ordered/purchased

Research Schedule

The project has a time budget of 3 years from August 1994. The work has been divided into three phases, each a year long. The schedule for the remainder of the present phase is also shown in Figure 14.

PHASE I (present stage, each step completed unless otherwise noted) :

a) Design and construction of rotor-SFD test apparatus (see hardware list, Table 2) to be fully assembled by June 15, 1995

b) Procurement of drive motor, power supply and lubrication system.

c) Programming data acquisition software for the specific instrumentation of test rig.

d) Procurement of instrumentation for test and measurements.

PHASE II (beginning August 1995):

a) Balancing and initial trouble-shooting of test rig.

b) Fine tuning of software for measurement of dynamic forced response of test rig.

c) Initial benchmark dynamic tests of rotor-bearing supported only on ball bearings and spring supports without the action of squeeze film dampers (SFD's run dry).

d) Coast-up (-down) experiments of test-rig dynamic forced response with squeeze film dampers for increasingly large values of disk unbalance at constant rotor acceleration rates for conventional and integral SFD's. Also to be varied is oil feed pressure and temperatures, sealing condition, static shaft position and support stiffness.

e) Identification of system frequency response and determination of regimes of operation.

f) Based on measurements of dynamic imbalance response, and with the support of advanced theoretical models, develop a simple and tractable model for the flow field and fluid forces in squeeze film dampers.

PHASE III:

a) Correlation between measured test-rig dynamic response and predictions from the novel models developed for relevant cases of general interest.

b) Quantification of the dynamic response in terms of linearity and non-linearity, periodic and aperiodic response, etc.

c) Experimental tests to study the effect of end seal conditions on the dynamic response of the system.

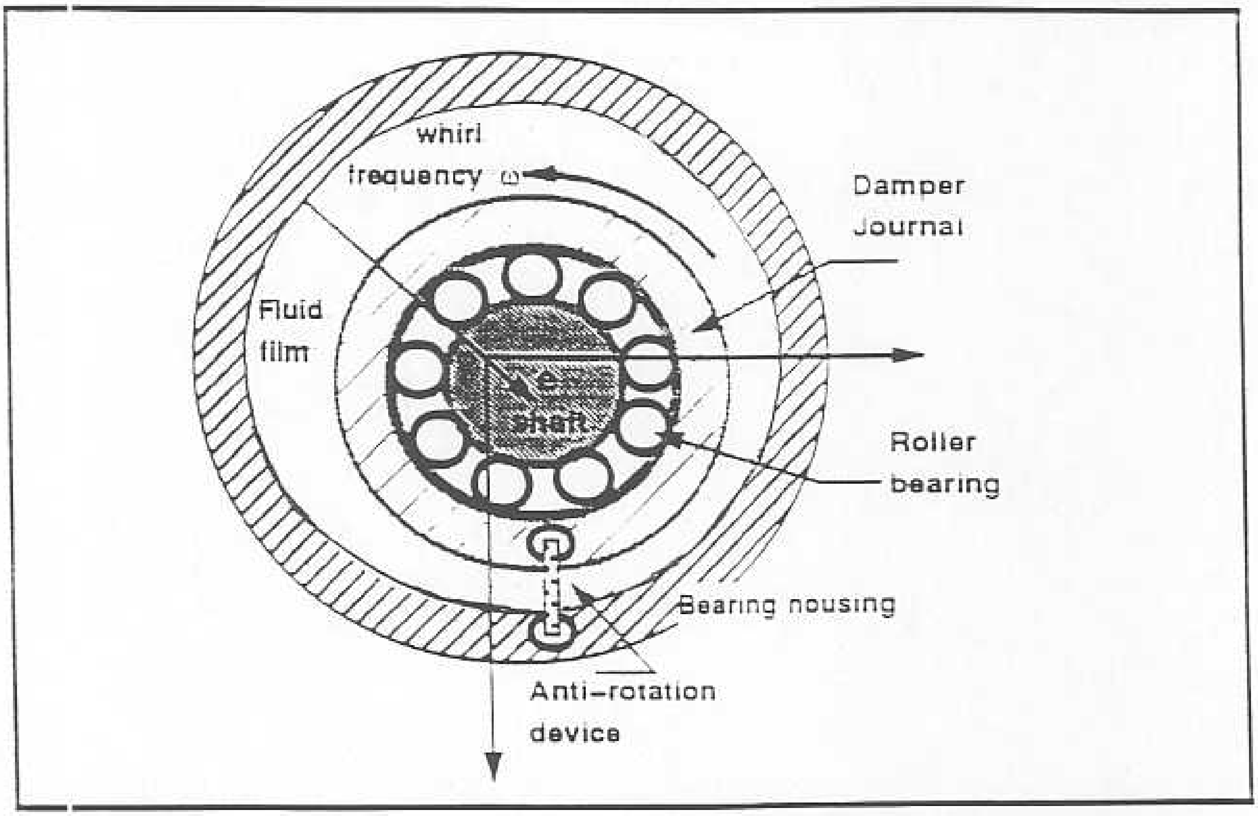


Figure 1. Schematic View of a Typical Squeeze Film Damper

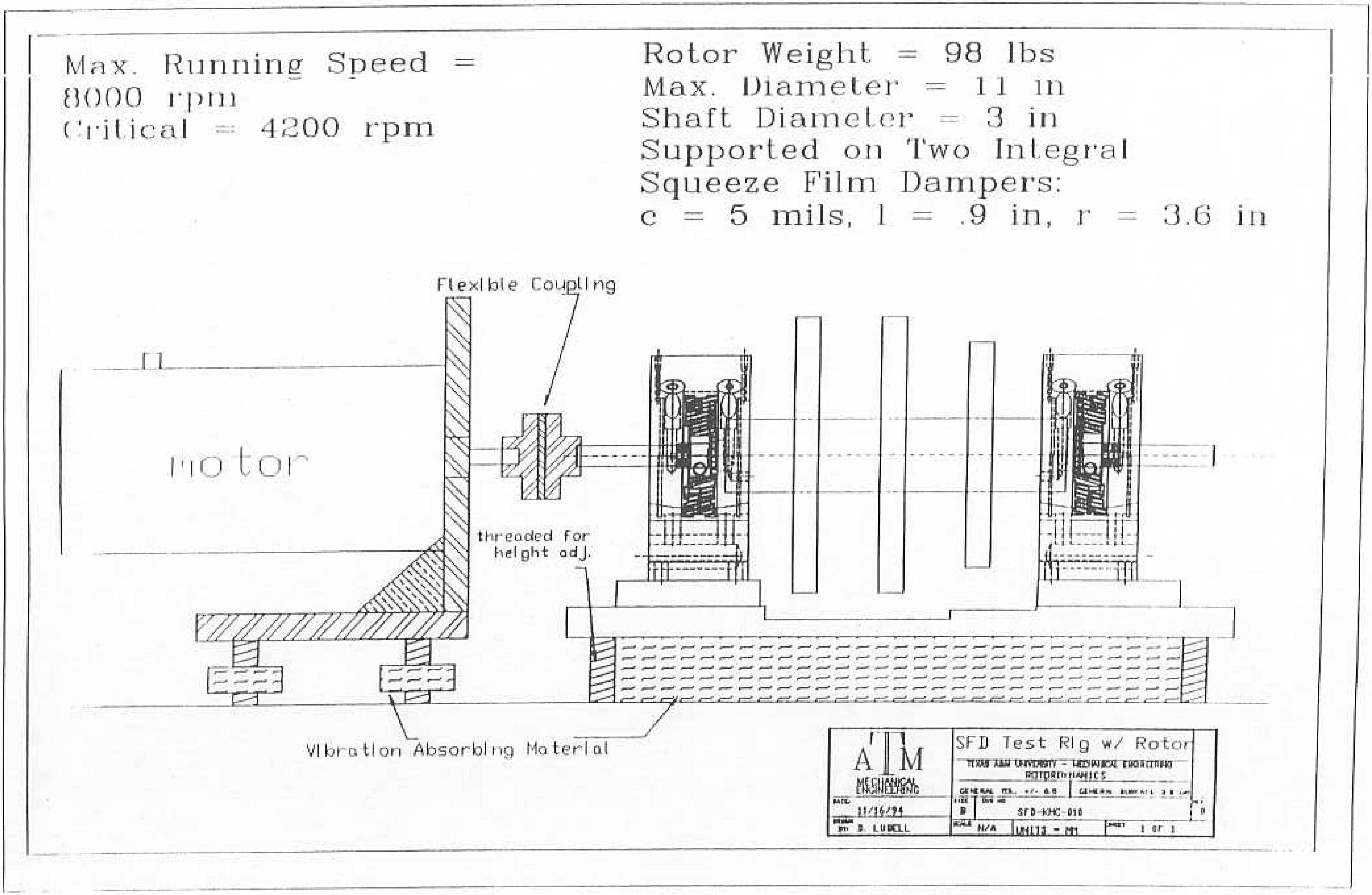


Figure 2: Test Rig Overview

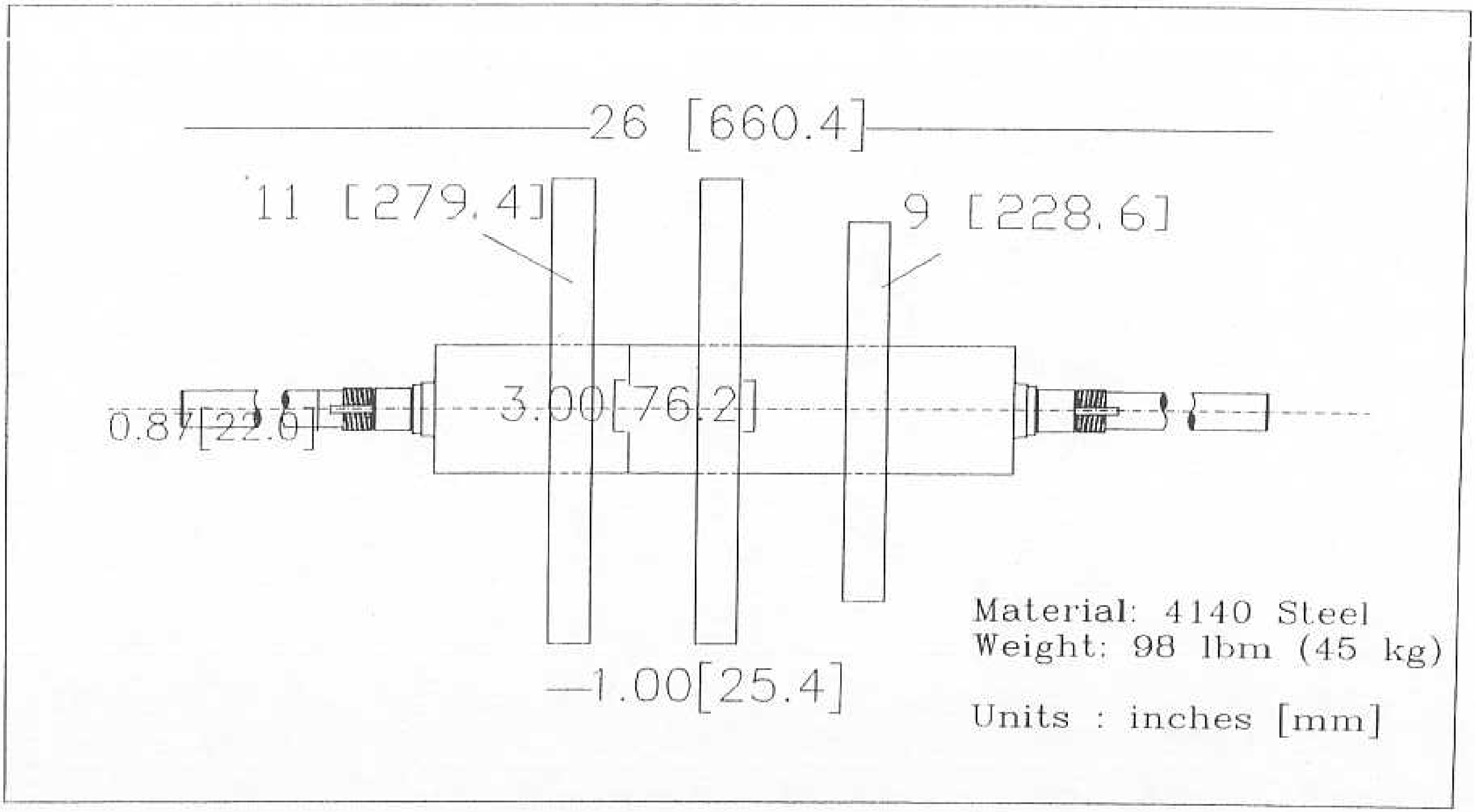


Figure 3: Test Rotor

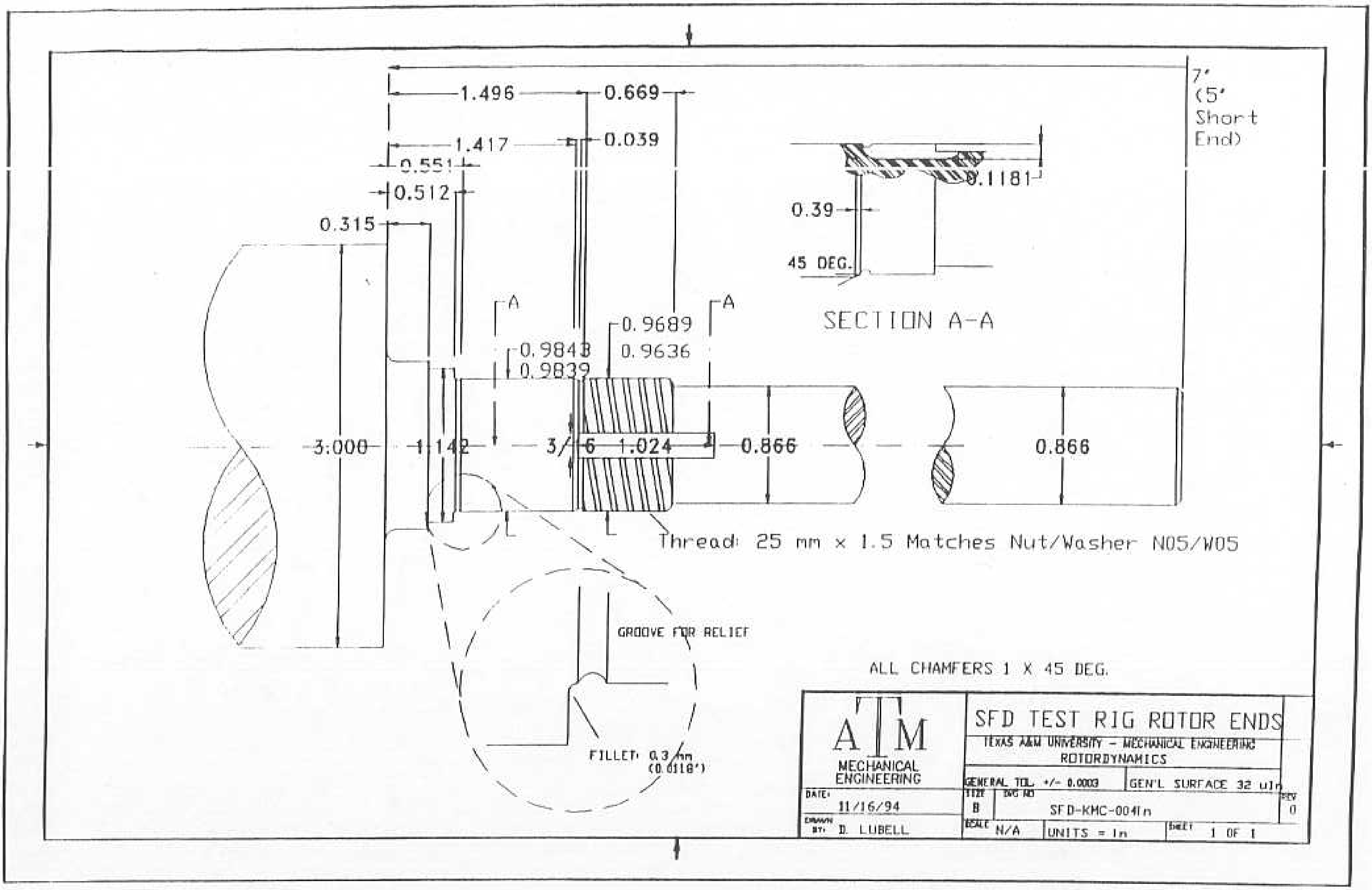


Figure 4: Detail of Rotor End

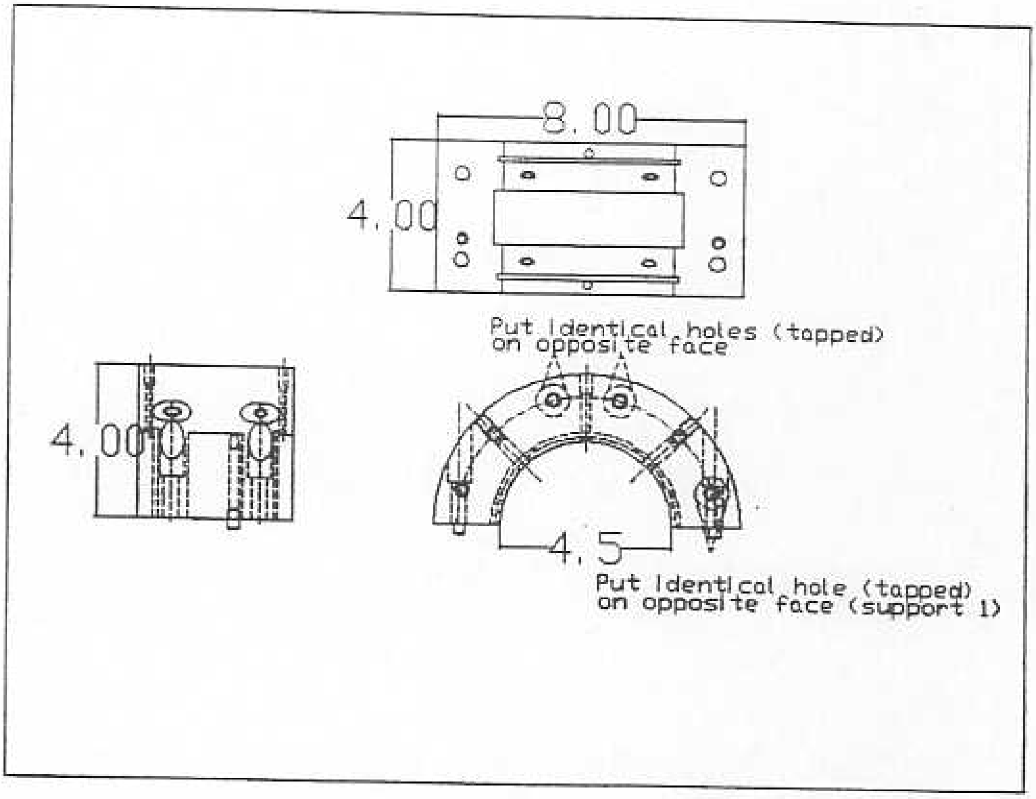


Figure 5: Support Housing, Top

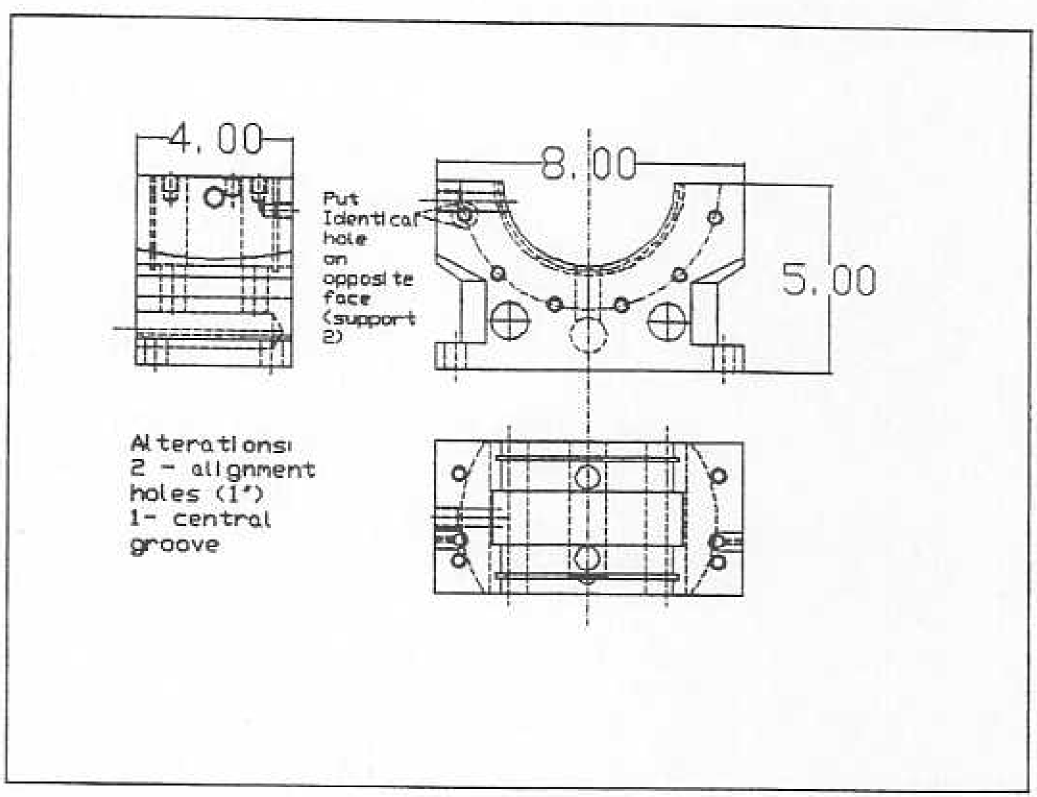
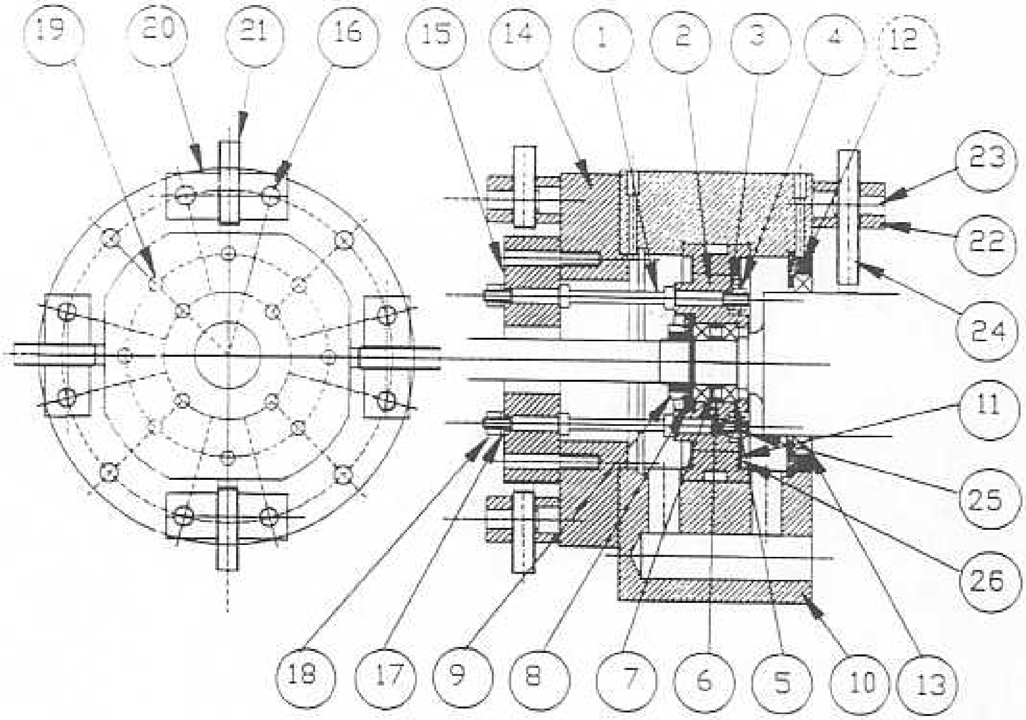


Figure 6: Support Housing, Bottom



ITEM	#	DESCRIPTION	OBSERVATION
1	8	RIB	MACHINE SEE DRAWING 1
2	2	JOURNAL	MACHINE SEE DRAWING 2
3	8	WASHER	0.375 *
4	8	NUT	0.375 24- UNF
5	4	BALL BEARING	71905 SKF
6	2	OUTER SPACER	MACHINE SEE DRAWING 6
7	2	INNER SPACER	MACHINE SEE DRAWING 7
8	2	RETAINER	MACHINE SEE DRAWING 8
9	2	LOCK NUT	N 05
10	2	BASE SUPPORT	
11	2	DAMPER	MACHINE SEE DRAWING 11
12	2	LIP SEAL SUPPORT	MACHINE SEE DRAWING 12
13	2	LIP SEAL	NATIONAL 3'
14	2	BASE PLATE	MACHINE SEE DRAWING 14
15	2	CENTERING PLATE	MACHINE SEE DRAWING 15
16	24	SCREWS	0.375 16- UNC
17	8	WASHER	
18	8	NUT	0.375 24-UNF
19	8	SCREWS	0.375 16- UNC
20	8	BASE	MACHINE SEE DRAWING 20
21	8	CENTERING SCREW	0.500 13- UNC
22	4	PROXIMITOR'S BASE	MACHINE SEE DRAWING 22
23	8	SCREWS	0.375 16- UNC
24	4	PROXIMITY PROBE	0.375 24- UNF
25	8	END SEALS	MACHINE SEE DRAWING 25
26	24	SCREWS	0.250 20 UNC

DRAWING : SUBASSEMBLY OF THE SUPPORT
SCALE : 4 : 1
BY : HECTOR LAOS
DATE : 02 / 20 / 95

Figure 7: Conventional SFD Design

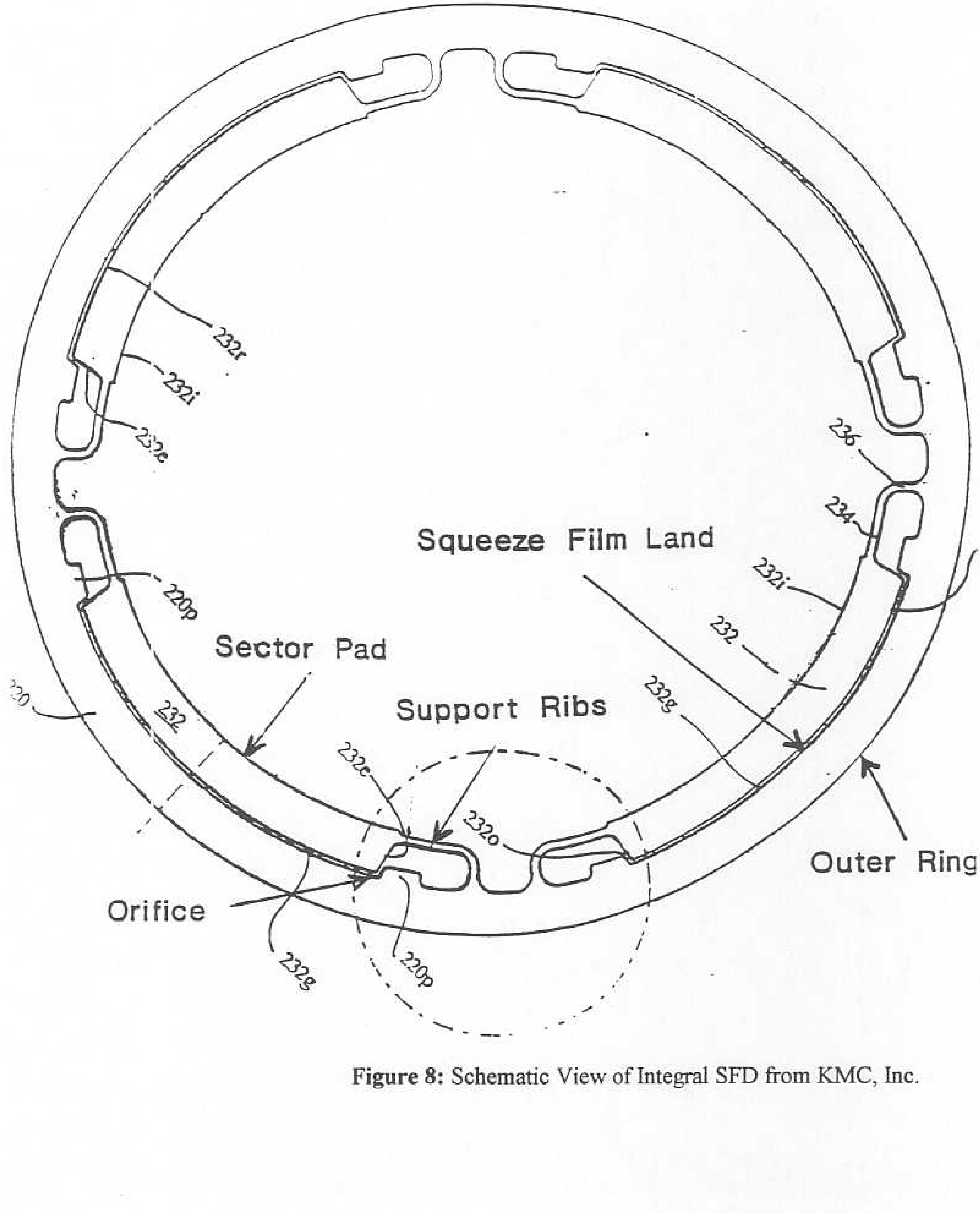


Figure 8: Schematic View of Integral SFD from KMC, Inc.

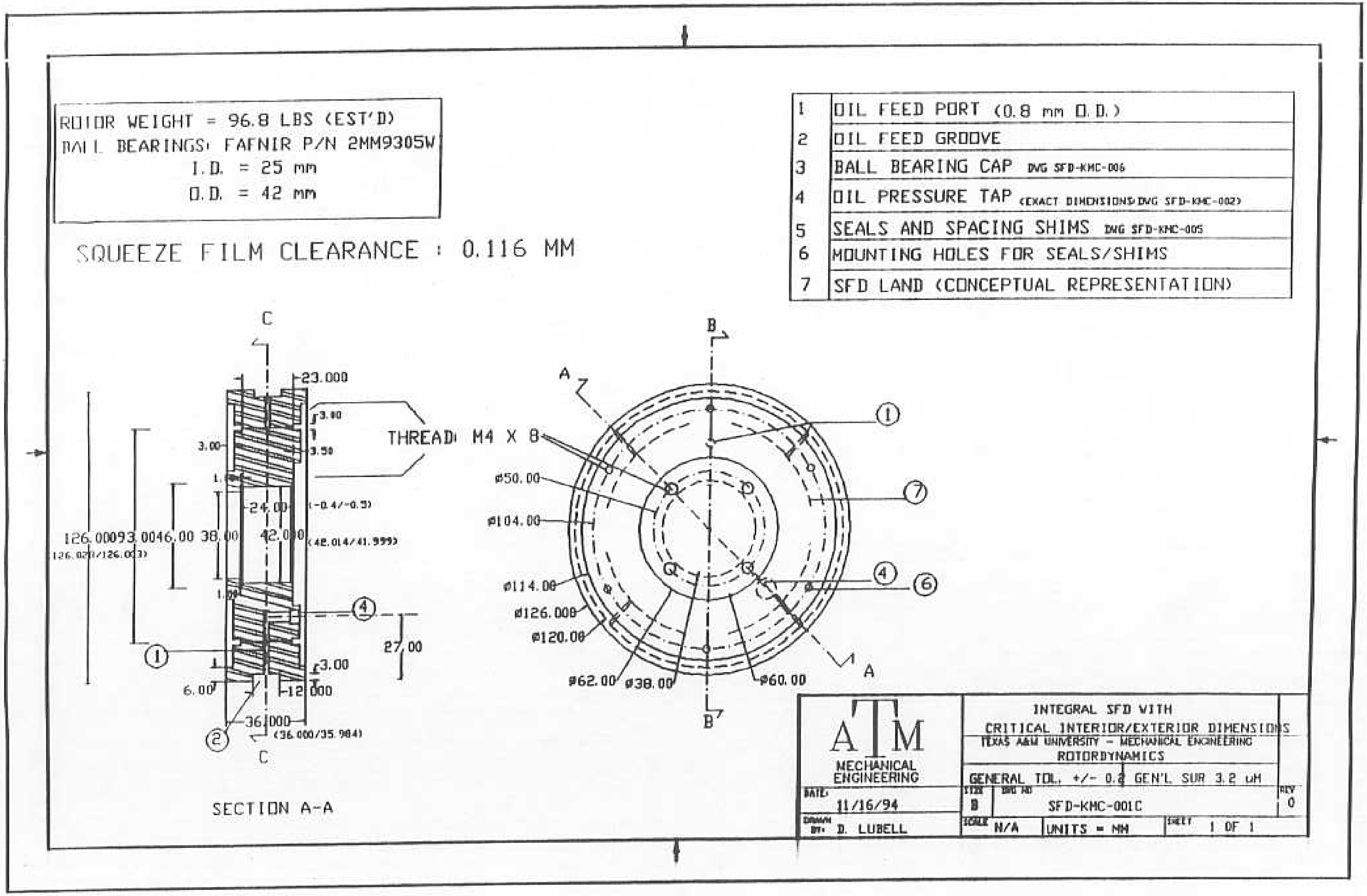


Figure 9: Detail of Integral SFD

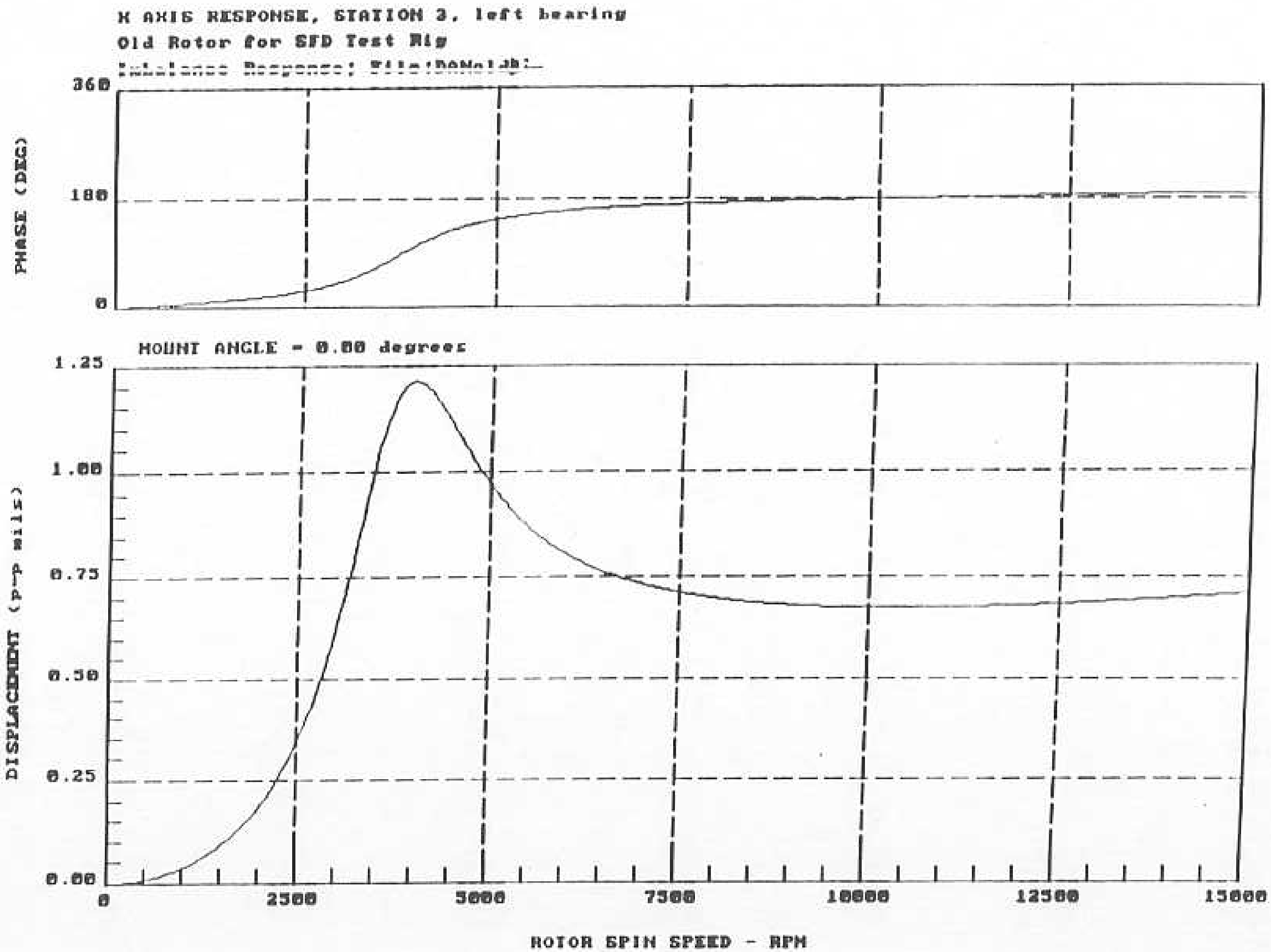


Figure 10: Critical Speed Analysis [PUP]

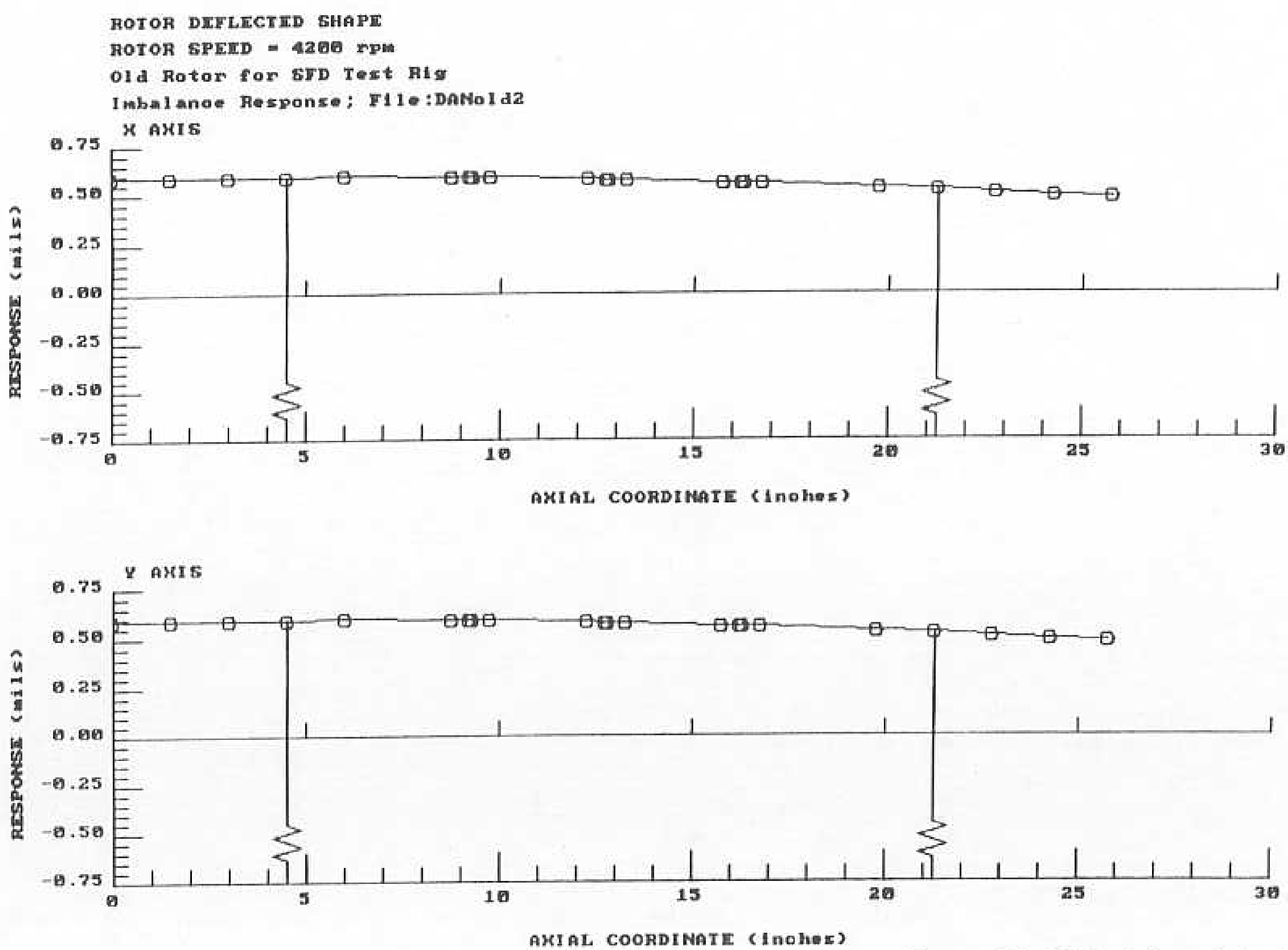


Figure 11: Critical Speed Analysis, Mode Shape

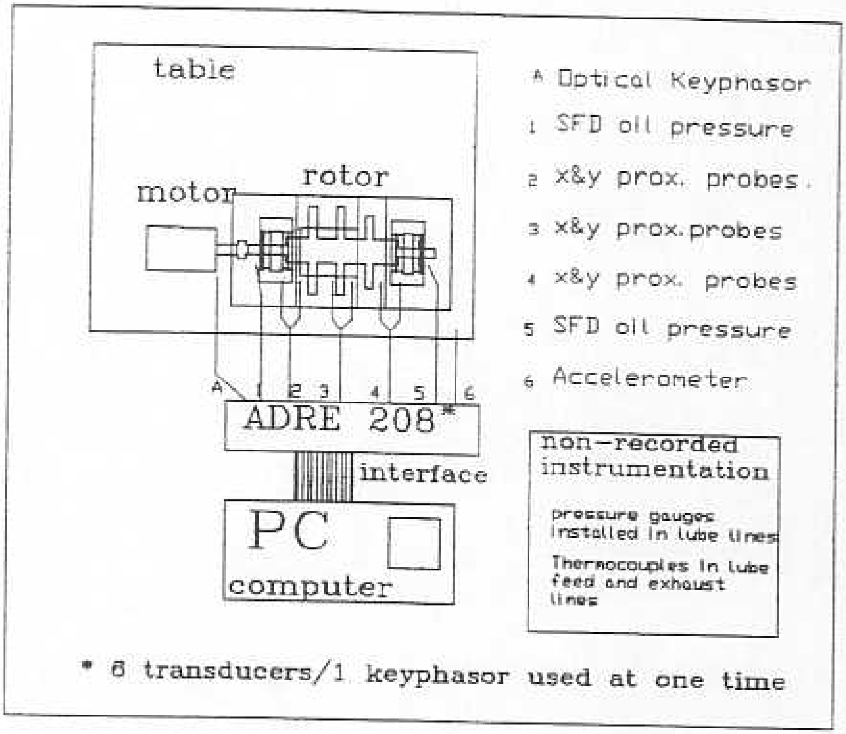


Figure 12: Schematic View of Instrumentation

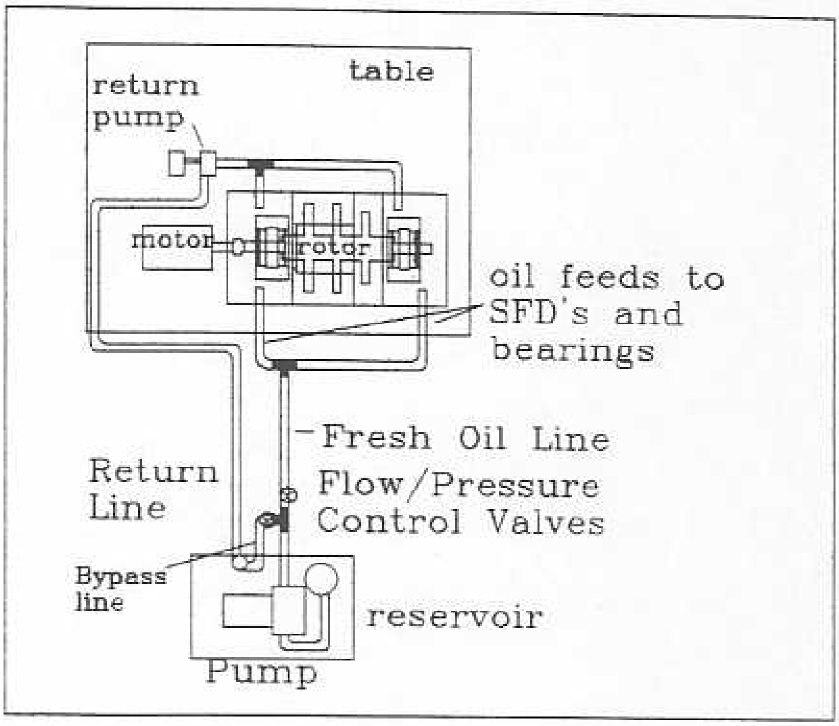


Figure 13: Schematic View of Lubrication System

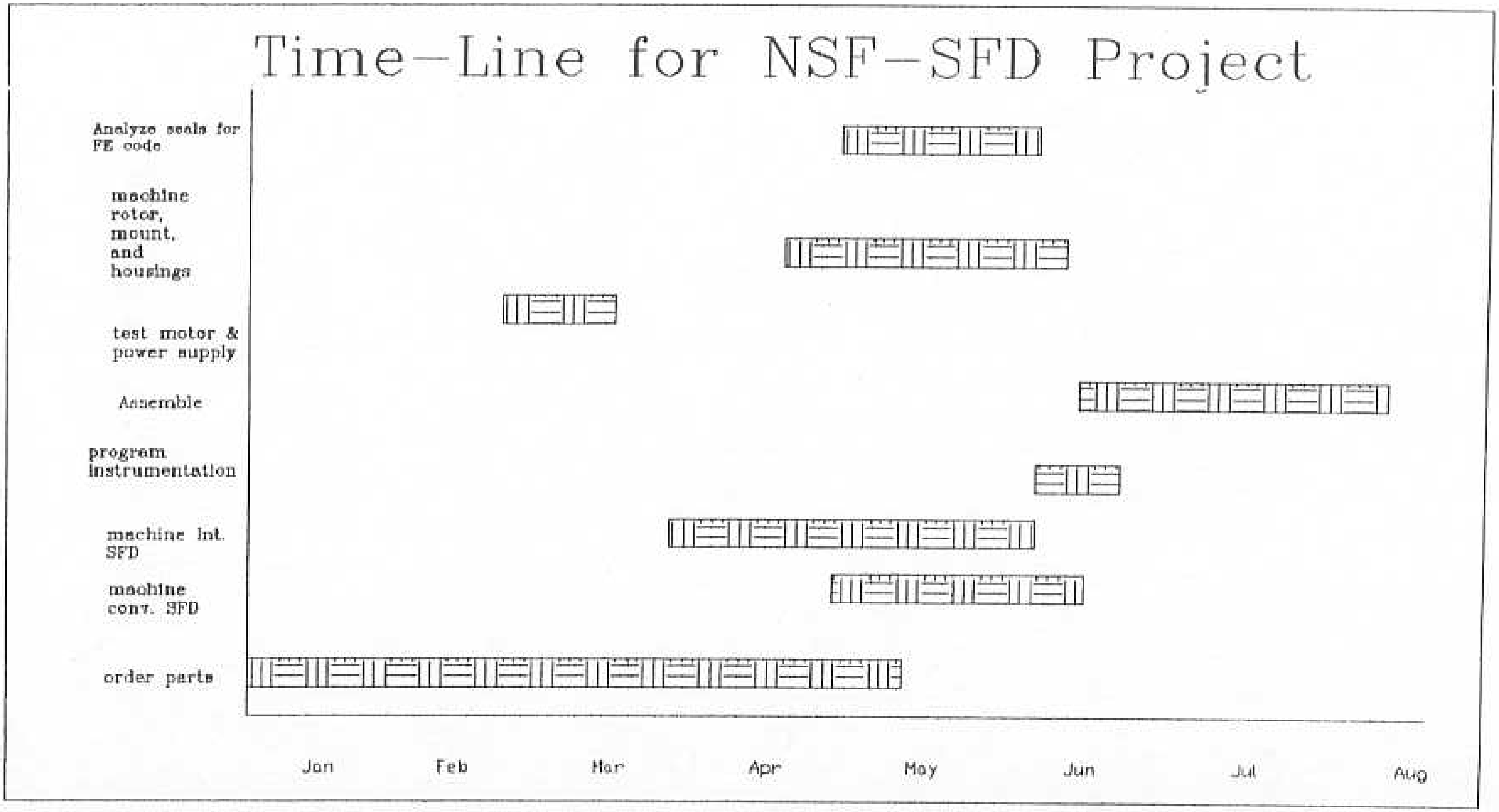


Figure 14: Time Line

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