DESIGN OF A HIGH-SPEED

TURBOMACHINERY TEST CELL AND COMPONENTS

by

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VI. LIST OF SYMBOLS

C	damping
c_p	specific heat of air at constant pressure
d	diameter
f	frequency
g	gravitational constant
G	modulus of rigidity
h	enthalpy
HP	horsepower
Hz	hertz
I	inertia
J	polar moment of inertia
K	torsional stiffness
l	length
$\dot{m}$	mass flow rate
N	speed
η_c	compressor efficiency
p_o	total pressure
rpm	revolutions per minute
T	torque
T_{02}	inlet temperature
γ	ratio of specific heats
θ	angular displacement
ω	circular natural frequency

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VII. INTRODUCTION

Recent research efforts in the Mechanical Engineering Department at Virginia Polytechnic Institute and State University have led to the development of single- and six-channel on-rotor telemetry systems which have been used for measuring surface pressures on the rotating blades of an axial-flow compressor under various operating conditions [1, 2, 3]. This work has been conducted on a low-speed two-stage axial-flow compressor, driven by a direct-coupled 7 1/2 HP cradled dynamometer, capable of speeds to 3000 rpm. This facility has provided the necessary apparatus for testing and verification of the on-rotor telemetry package. However, investigation of high-speed "state-of-the-art" compressor rotors and the application of current and future generation telemetry packages to such high speed systems requires the development of a new facility capable of turning rotors at the desired higher speeds.

The objective of this research was to conduct the analytical and design work necessary to allow development of such a facility. The goals were to design a facility capable of operating test rotors at speeds of 20,000 rpm, with provisions for creating stall and surge conditions within the compressor and with the versatility to allow expansion into other areas of compressor research such as noise and inlet flow distortion studies. A major requirement was development of the facility at minimum initial cost while providing a versatile, easily utilized, relatively economical research tool.

The basic facility design which resulted from the investigation incorporates a drive system, mounting hardware, and monitoring and control systems. The drive system is composed of a prime mover, gearboxes to provide the desired speeds, the test compressor and couplings. The mounting hardware was designed on the basis of strength, safety, stiffness, size, versatility and cost requirements. The monitoring and control systems were selected according to specifications outlined in operating manuals for the various components used and are designed to insure operation within the specified limits. This report discusses the selection and utilization of these major components.

VIII. REVIEW OF LITERATURE

Research Facilities

Many turbomachinery research facilities exist throughout the U.S. today. All engine manufacturers, many governmental agencies, and some private concerns operate test facilities for various purposes. These range from the testing of individual components such as starters and control systems to full scale engine testing under simulated flight conditions.

An example of a modern jet engine development facility is one owned by AiResearch Manufacturing Co. in Torrance, Calif. [4]. This facility includes six test cells, assembly, instrumentation, preparation and data acquisition areas. Performance tests on both production and experimental engines are conducted with principal emphasis on measurement of engine thrust and parameters such as temperatures, pressures, rotational speeds and vibrations. Special tests on individual components such as rotor burst and fatigue tests are performed in a special spin pit. Tests of certain engine accessories (gearboxes, controls, etc.) are also conducted in special areas. A central computer controls test runs, records up to 1000 individual parameters from each test, and evaluates and displays data. This facility, while more lavish than many, indicates the complexity involved in engine development work.

Another facility, designed for testing of component parts rather than the total engine, is exemplified by the Turbopropulsion Laboratory at the Naval Postgraduate School [5]. This laboratory is composed of three separate test areas: a turbine test area for research on high-speed single-stage turbines of up to 9.7 inch diameter; a compressor test area with a drive unit delivering 450 HP at 30,000 rpm for work in both normal and transonic ranges; and a spin pit for evaluating thermal and centrifugal stresses in rotors at speeds to 50,000 rpm. Each test area utilizes an air-turbine drive unit supplied with air from a central compressor. Data is handled by a centralized data acquisition system which records up to 100 parameters and uses a telephone connection to the campus computer for data reduction purposes. The facility also has a calibration tube for use in testing instrumentation prior to actual experimental runs.

Vibration Analysis

In the design of high-speed rotational systems, careful attention must be given to the prevention of vibration problems. The present drive system consisted of a prime mover, gearboxes, the test compressor, and couplings interconnecting these components to form a system. Foster-Pegg [6] has pointed out that 'when driver and driven equipment are supplied by more than one manufacturer, the user is advised to obtain a vibration analysis of the complete system. The analysis should be performed sufficiently early so that design changes may be made to insure that no operating speed falls within ± 5% of any system critical speed.' Rieger [7] has noted the principal vibratory motions in a geared system to be 1) Torsional 2) Bending 3) Axial and 4) Coupled. He also notes that factors promoting vibration problems in power transmission system are high power flow and high speeds.

Pure vibration modes, which include the first three vibratory motions mentioned above, may be analyzed by numerous techniques. Loewy and Piarulli [8] indicate that torsional vibrations of engines and geared systems have usually been considered purely torsional problems because of the lateral constraints used (sufficiently frequent bearing locations and rigidity of the system). With this pure vibration mode assumption, methods such as those of Holzer, Myklestad-Prohl, and impedance matching are readily applied.

The Holzer method [9] is a sequence calculation scheme begun by assuming an initial frequency. Then, starting at one end of the system, a balance of torques and displacements (torsional case) is obtained, step by step. The final external torque required to achieve balance is called the residual torque. If this is zero, the assumed frequency is a natural frequency of the system. A plot of residual torque versus frequency yields all of the natural frequencies of the system.

The Myklestad-Prohl method, more commonly known as the Transfer-Matrix Approach, relates the generalized forces and displacements at one end of a shaft system to those at the other end by successive multiplication of matrices. The matrices account for effects of stiffness and inertia properties of various sections of the system. This technique is very suitable for machine computation and a program utilizing this method has been reported by Wong [10].

Mechanical impedance and mobility methods involve a determination of system natural frequencies based on knowledge of impedances or mobilities of component parts of a system. A program based on this approach which utilizes the method of "node forces" to assemble impedance matrices has been developed by Mitchell [11]. A lumped-mass model is used to represent the system. Input data to the program includes stiffness, inertia, and damping values for components, plus a matrix loading scheme which enables the program to assemble the input data in the proper sequence. Output obtained is a listing of amplitude and phase information for each system element versus frequency, from which the resonant frequencies and mode shapes may be obtained.

Analysis of bending and axial mode vibrations are obtained by methods similar to those outlined above. Coupled modes, which involve interactions between any of the above pure modes, present a more complex problem. In this situation, non-critical vibrations in one mode may excite critical vibrations in another mode. Badgley and Hartman [12] have noted that the analytical capability for performing the complete solution to the coupled vibration problem does not exist. A structural system dynamic approach (e.g. NASTRAN), in which the shaft-bearing system is represented by a series of beams, springs and masses, can be used for analysis, although not for a complete solution.

IX. INVESTIGATION AND DESIGN

Facility Requirements

The basic requirements for the research facility were initially established. The building was to provide:

1) A work area in which drawings, repairs, assembly of components, calculation and evaluation of data could be made.
2) A control room from which test parameters could be monitored and controlled.
3) A test cell, isolated from the remainder of the building, in which operation of components under investigation could be conducted.
4) A storage area in which supplies and materials necessary for facility operation could be stored.

Design of the test cell required establishment of operational capabilities necessary to perform the proposed research, i.e., investigation of "state-of-the-art" turbomachinery. Treager [13] enumerates various machines which operate at speeds to 50,000 rpm, flow rates of 200 lb/sec, pressure ratios of 24:1, and require as much as 40,000 HP to drive. Machines of this magnitude would create numerous problems, requiring power and airflow which could not be provided within the limitations of this facility. Therefore, the design was limited to machines requiring only a few thousand horsepower and speeds below 20,000 rpm. This does not severely restrict facility capabilities for research because the majority of present-day turbomachines operate below 20,000 rpm and the horsepower requirements can be reduced by driving only a few stages rather than complete rotors. This does not hamper the research effort as only a few stages are normally instrumented and the effects of downstream stages are either negligible or easily simulated.

The nature of the intended research required that a variable-area exhaust duct be provided for control of the compressor back pressure so that stall and surge conditions may be induced. To make the data obtained meaningful, complete knowledge of system parameters such as temperatures, pressures, speeds, flow rates, etc. must be available.

The facility was also intended to provide the flexibility needed for research into other related areas without requiring unnecessarily difficult modifications. This was a primary consideration in the selection of components and the design of the mounting system.

Selection of Building

With the basic objectives established, it was possible to proceed with the facility design. The selection of a building to house the facility involved consideration of such goals as convenience, effects of noise on the surrounding community, and safety, in addition to the requirements previously noted. Consideration of the noise and safety aspects eliminated the possible remodeling of the present low-speed facility.

An existing building at the VPI&SU airport (Fig. 1) was available and met the above requirements. Renovation of this building resulted in the layout shown in Fig. 2, which provides sufficient room for

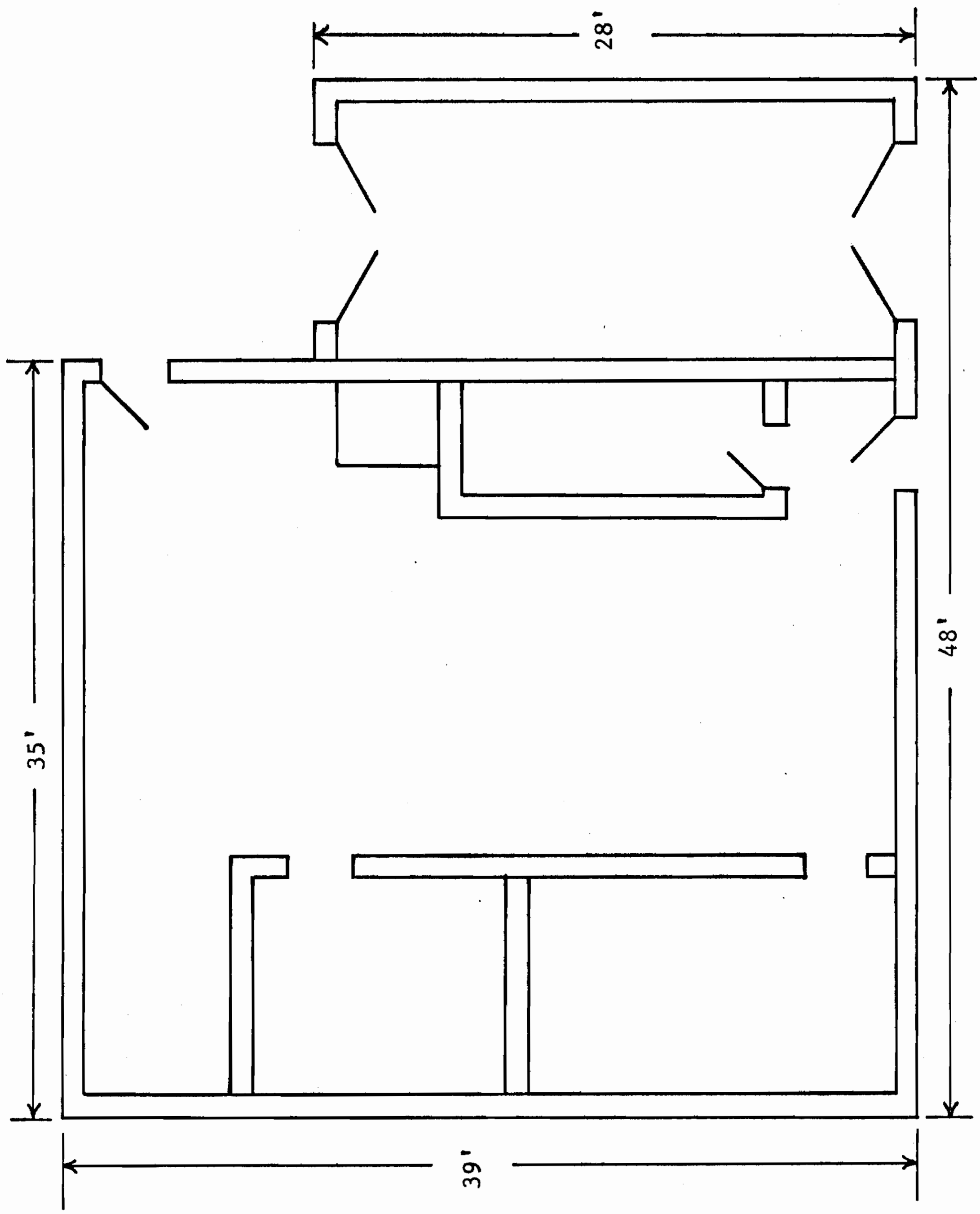

Figure 1. EXISTING BUILDING

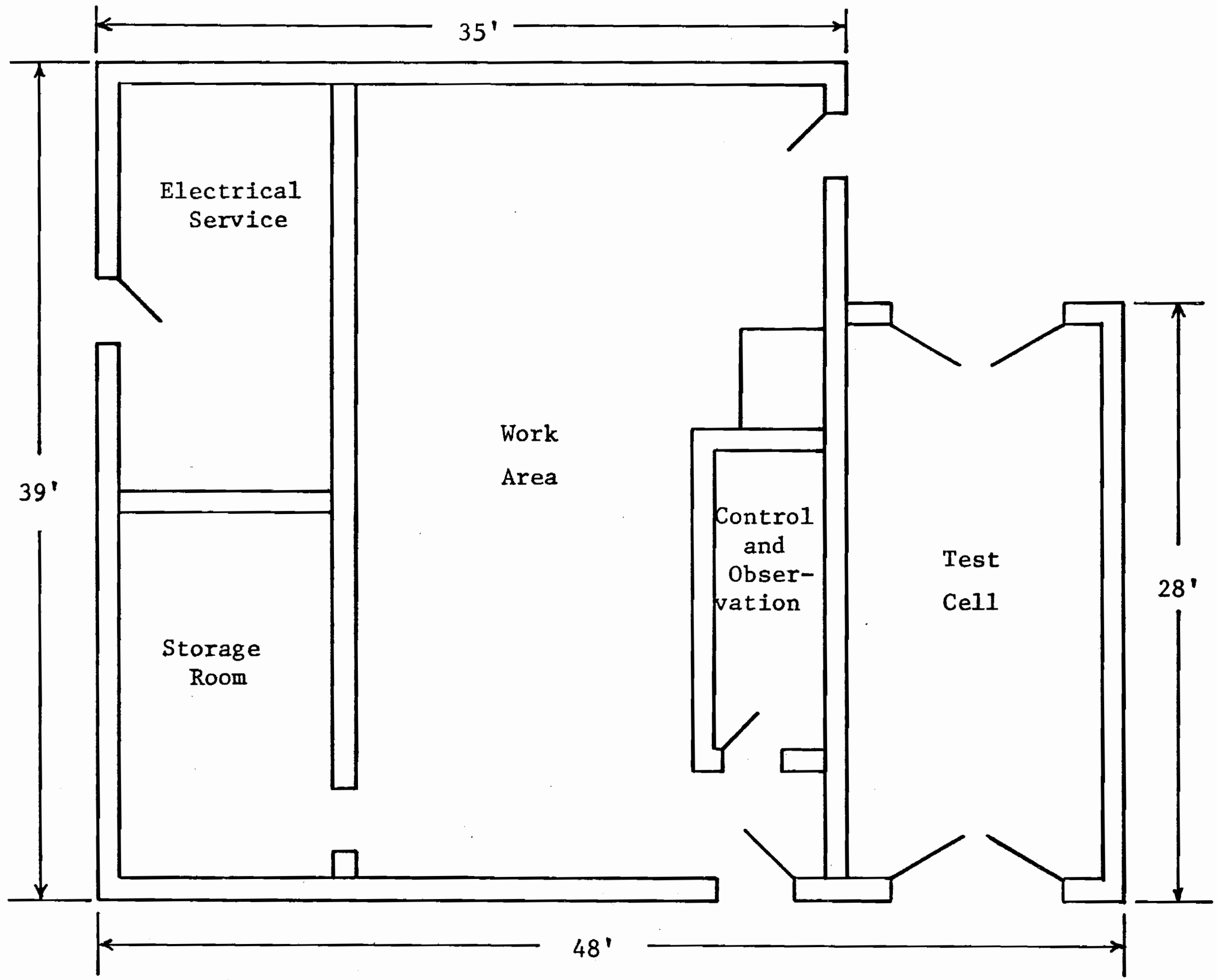

Figure 2. RESEARCH FACILITY

current research activities with some allowance for future expansion. Views of the facility work area and test cell are shown in Fig. 3 and Fig. 4, respectively.

Selection of Equipment

Drive Engine

The initial consideration for the drive engine was to utilize an available diesel engine with a speed-increaser gearbox to provide the desired operating range. This provided a readily available, easily designed and operated means of achieving the established goal. However, the limited horsepower available would allow operation of only one or two stages of typical turbomachines and the speeds desired would require operation of the diesel at high speeds which would reduce engine life. In addition, vibration problems involved with a system of this nature could prove severe.

Another alternative was to use a gas turbine engine as the driver. Such an engine, a General Electric model T64-6B turboshaft, was available to the project and was extremely well suited for this application. The free power turbine of the engine is capable of delivering 3080 HP @ 13,600 rpm, which is sufficient but not excessive shaft power for the intended research. Controlled engine speeds from 12,000-17,000 rpm are available, thereby allowing either a direct-drive system or one involving only a moderately sized speed-increaser gearbox. Moreover, the gas turbine is inherently vibration free in comparison with a diesel engine. Based on these considerations, the T64-6B engine was

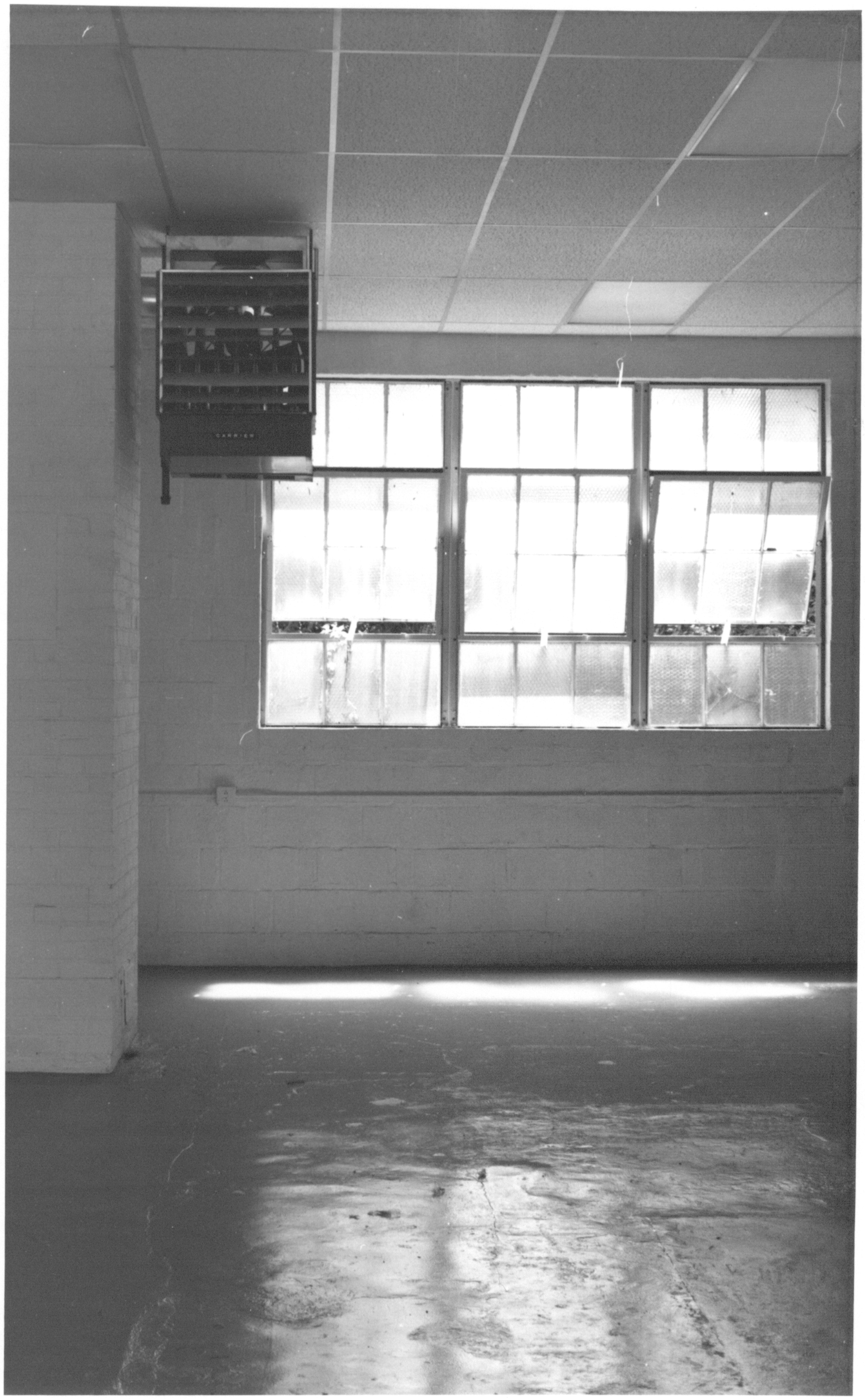

Figure 3. PHOTOGRAPH OF WORK AREA

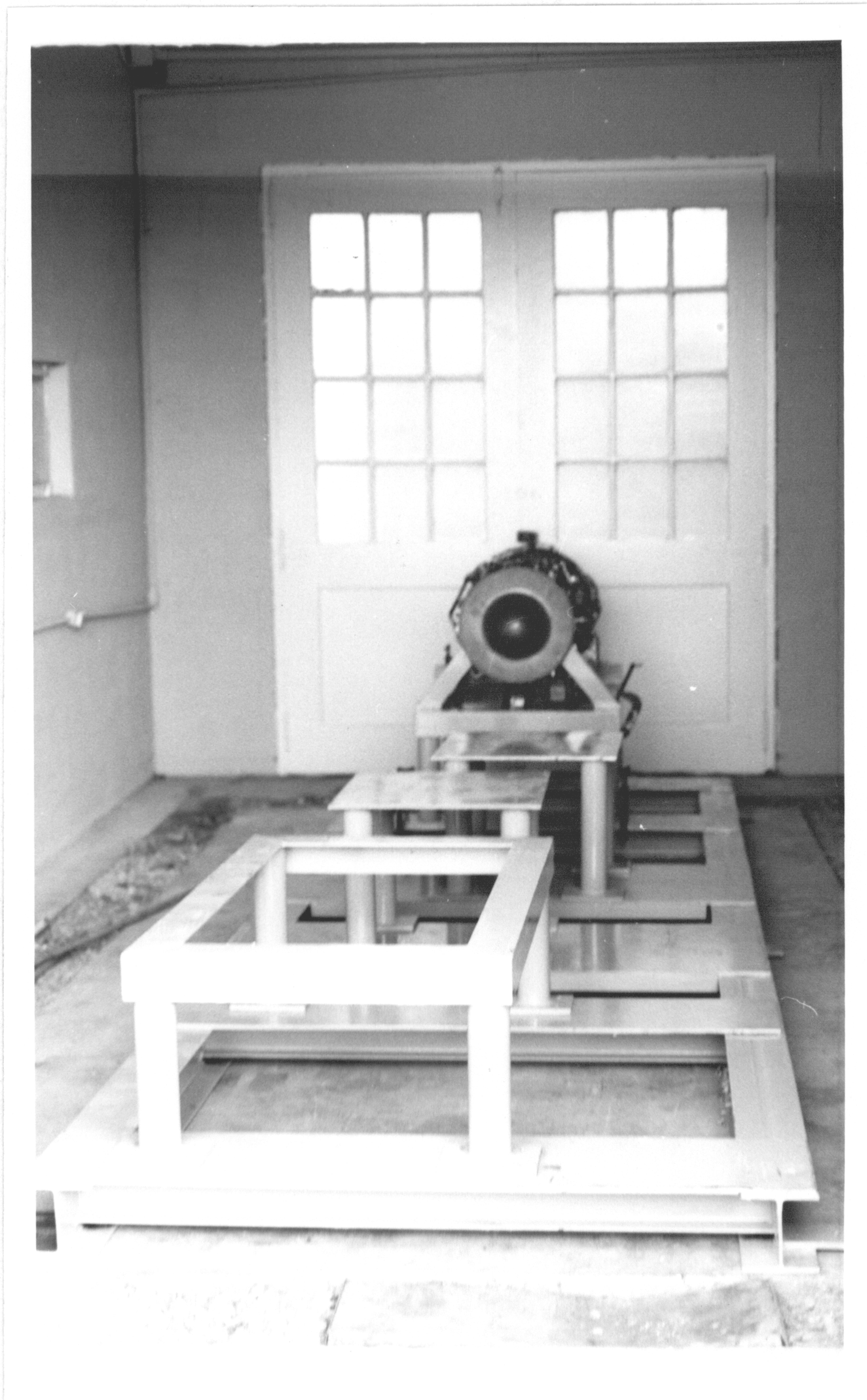

Figure 4. PHOTOGRAPH OF TEST CELL

selected as the drive unit for the facility. The T64-6B engine is shown in Fig. 5 and complete specifications are given in Appendix A.

Test Compressor

The desired test compressor was to be indicative of current "state-of-the-art" turbomachines in regard to speeds, mass flow rates and operational characteristics. Such a compressor was obtained from a T64 engine similar to the drive engine. This compressor is a 14 stage axial-flow model which is designed to deliver 24.5 lbm/sec at an overall pressure ratio of 12.6 to 1 when operated at 17070 rpm. This unit is well matched to the drive engine and thereby greatly simplifies the design of the drive system. The compressor is shown in Fig. 6.

Gearbox

The T64 engine is capable of driving the test compressor at its design speed. The output speed range of the free power turbine ranges to a maximum of 17000 rpm, which is the approximate gas generator speed. However, this requires operation at the engine maximum speed limit, which is an undesirable situation. Additionally, one of the facility goals was the capacity for spinning rotors to 20,000 rpm. These considerations required the inclusion of a gearbox in the drive train.

Because the T64 is rated at 13,600 rpm, this was chosen as the base from which the 20,000 rpm capability would be obtained. This

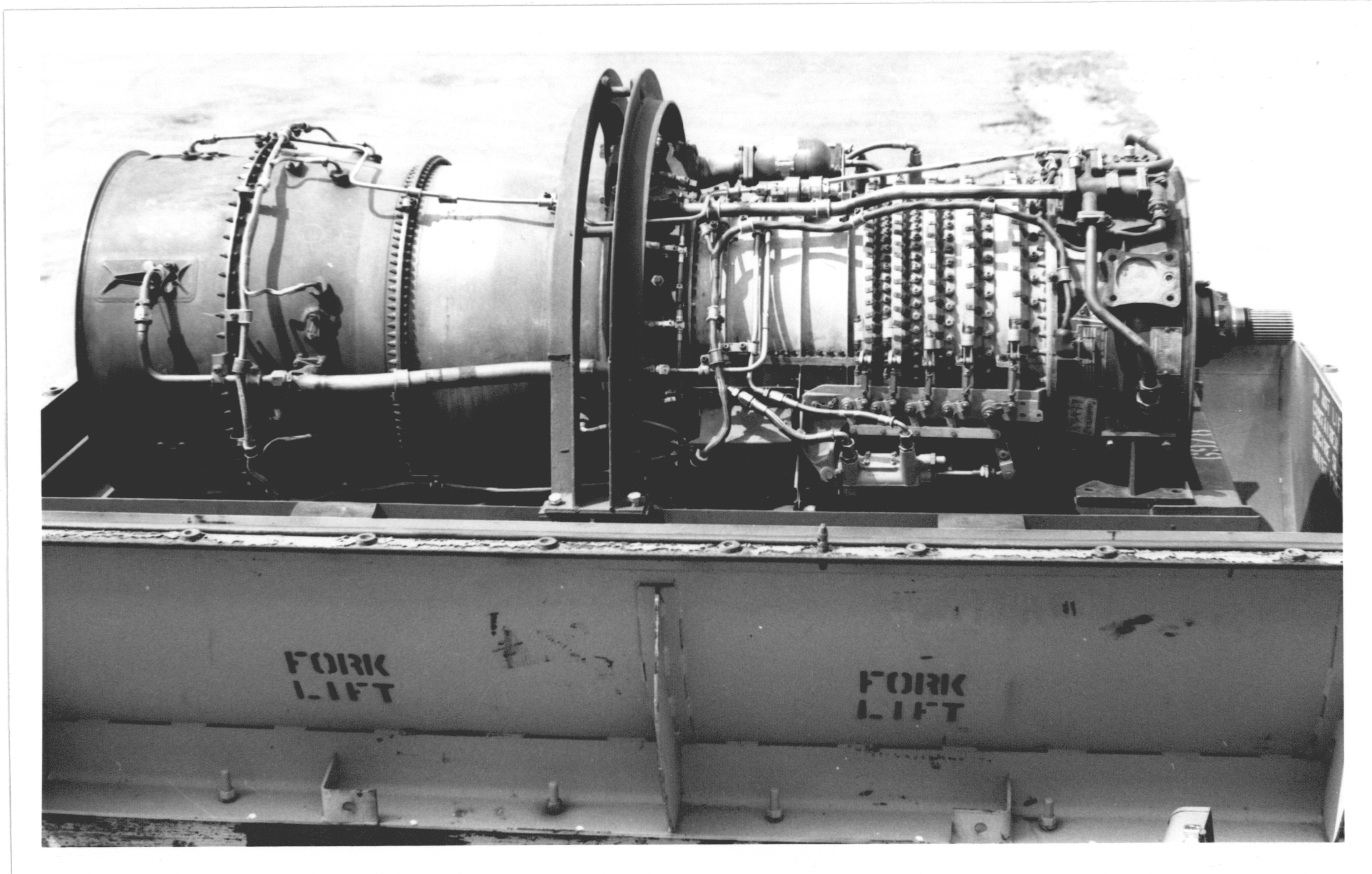

Figure 5. PHOTOGRAPH OF T64 ENGINE

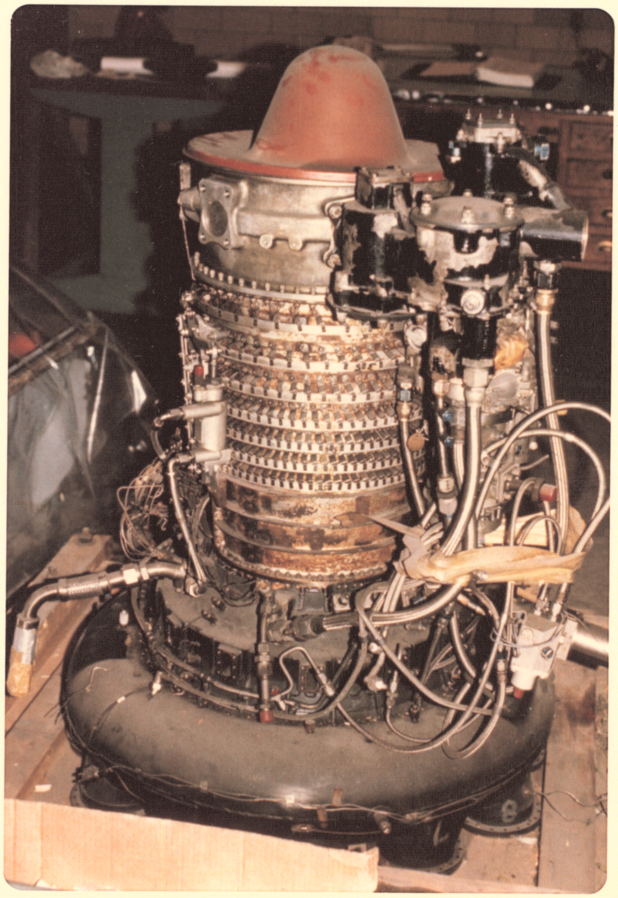

Figure 6. PHOTOGRAPH OF TEST COMPRESSOR

required a 1.47 ratio speed-increaser gearbox and resulted in a system with a widely variable operating range without overspeeding the engine.

Consideration of the total drive system indicated that a two-shaft gearbox would introduce unwanted complications due to the resultant reversal in rotational direction. The test compressor is designed to be driven from the rear in the same direction as the engine output shaft rotates. Driving the compressor from the inlet end, while possible, introduced the following problems:

1) Inlet flow distortion would be created by the shaft connection and proximity of the gearbox;
2) The possibility of the engine and compressor competing against each other for airflow was created;
3) Application of the instrumentation package would be difficult from the compressor exhaust end;
4) Future utilization of the facility for compressor noise studies would be impossible because an open inlet would not be available.

These considerations, which affected facility versatility, required selection of a gearbox arrangement which provides the proper shaft rotational direction.

Two alternatives were available to meet this requirement: 1) a three-shaft gearbox or 2) two two-shaft gearboxes. Consideration of these choices indicated that utilization of two two-shaft gearboxes would result in greater versatility because:

1) The two gearboxes would be individually less complicated than a three-shaft model and, therefore, less prone to failure;

2) Changes in gear ratios of either of the two gearboxes would be easier than a similar change in a three-shaft model;

3) Two gearboxes provided the capability of driving rotors in either rotational direction with equivalent speed ranges available.

The chosen system is thus a two-gearbox drive delivering 20,000 rpm from a 13,600 rpm input, thereby requiring an overall ratio of 1.47. This ratio can be provided by numerous combinations of ratios between the two gearboxes. However, a design using a 1:1 reversing gearbox with a 1:1.47 increasing gearbox provided the most versatile coverage of the intended speed range. Using gearboxes with these ratios, speeds from 12,000 to 25,000 rpm in either rotational direction could be provided. This design was selected as the first candidate system for analysis. Selection of the gearboxes to be used necessitated specification of actual horsepower requirements. These were obtained from a more complete investigation of the test compressor.

Compressor Horsepower Requirements

To aid this investigation a short computer program (Appendix B) was developed which determined the horsepower required to drive successive stages of a compressor designed to operate at 17,070 rpm, a pressure ratio of 1.2 per stage, a mass flow rate of 24.5 lbm/sec, and 80% efficiency, which should adequately simulate the test compressor being used. The procedure follows that outlined in an example problem given by Dusinberre and Lester [14] and does not include bearing losses, transmission losses, or power to drive auxiliary equipment such as the lube pump. The procedure is to calculate the change in enthalpy of the air resulting from the compression process, based on a 60°F inlet temperature. Multiplying this increase in enthalpy by the mass flow rate of air through the compressor yields the total energy put into the air by the compressor. The power required is thus given by:*

$$\text{Power} = \dot{m}\Delta h_o = \frac{\dot{m}c_p T_{02}}{\eta_c}\left[\left(\frac{P_{03}}{P_{02}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

The results of these calculations for various speeds and various numbers of compressor stages are shown in Fig. 7. It is evident that 8 stages of this particular compressor could be driven at design speed by the T64 engine.

Discussions with gearbox manufacturers indicated that both cost and delivery time would exceed program constraints if a 3000 HP gearbox was selected rather than a gearbox capable of only 1500 HP. For this reason, the two gearboxes previously chosen were specified with a 1650 HP capacity. These gearboxes allow operation of five stages of the test compressor, which is more than adequate for all planned research purposes.

Couplings

With selection of the major drive-train components completed, the problem was to choose the proper couplings for interfacing the system.

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* See Appendix B for details of equation application.

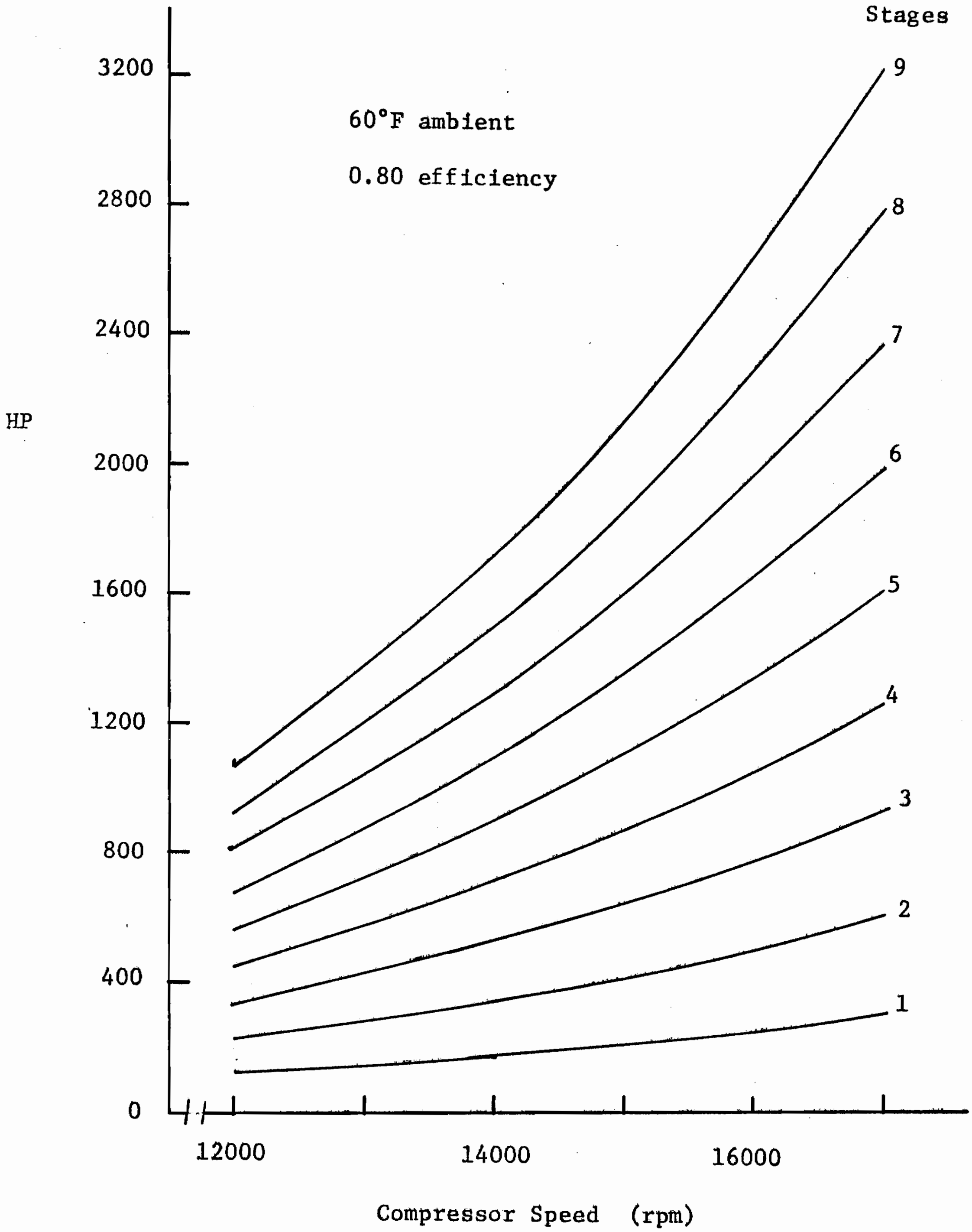


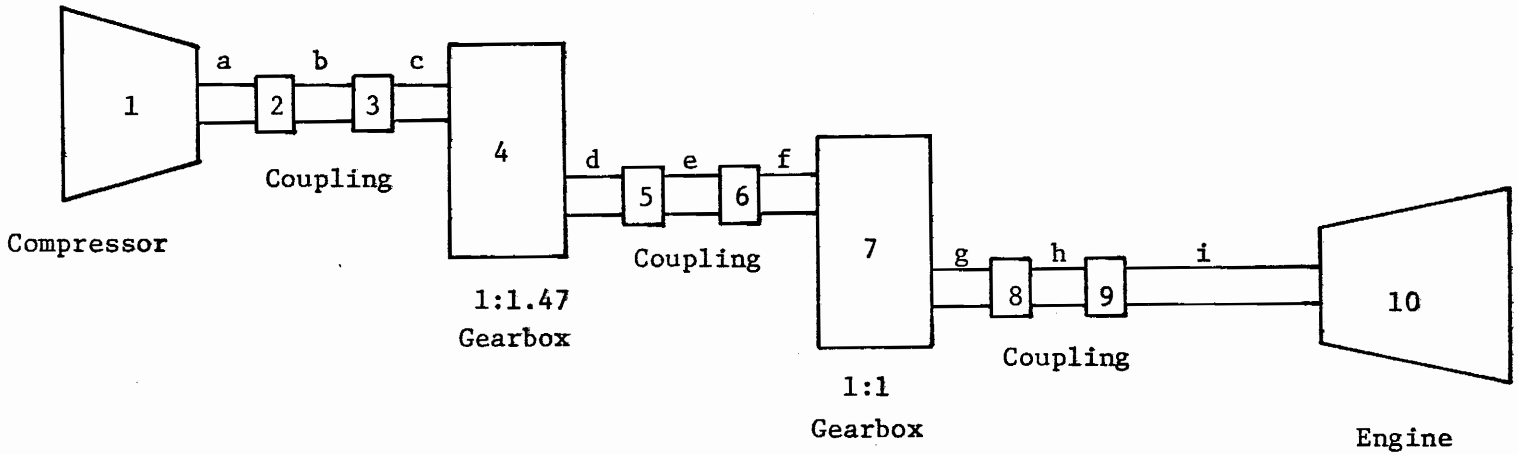
Figure 7. COMPRESSOR HORSEPOWER REQUIREMENTS

The couplings were required to transmit the horsepower and speeds desired, account for angular and parallel shaft misalignment, and provide torsional stiffness as required to reduce torsional vibration problems. The three couplings needed to connect the engine, two gearboxes, and the test compressor were selected from standard models available in the industry which were capable of providing the desired characteristics. Possible choices included gear- and flexible disk-type couplings. Geared couplings are capable of larger misalignments and higher power and speed levels than similarly sized flexible disk couplings. However, geared couplings require a continuous lubrication system which is not necessary with flexible disk models. This fact, plus consideration of the relatively low-power, intermittent operation required of the couplings, led to the selection of Rexnord double-flexing disk couplings. Specifications are included in Appendix A.

Once selection of the couplings was completed, a torsional vibration analysis of the total system was conducted to determine probable resonance frequencies.

Torsional Vibration Analysis

This analysis was based on a ten degree of freedom system as shown schematically in Fig. 8. Values of the polar moments of inertia and torsional shaft stiffnesses indicated in this figure were determined from information supplied by the manufacturers of the various components. The three couplings (b, e, h in the figure) had the same stiffness value, given by the manufacturer as 5.8×10^6 in-lb/rad.



Numbers represent lumped inertias

Letters represent torsional springs and dampers

Figure 8. SYSTEM SCHEMATIC

The compressor shaft stiffness (a) was estimated by using dimensional data supplied in a shaft spline drawing. Using the standard formula for a hollow circular shaft as given by Shigley [15] yields:

$$K_a = \frac{T}{\theta} = \frac{GJ}{\ell} = \frac{(11.5 \times 10^6)\pi(2.95^4 - 2.45^4)}{32(12)}$$

$$= 3.735 \times 10^6 \text{ in-lb/rad}$$

Gearbox shafts (c,g) and (d,f) were of equal size, and were estimated from dimensional data provided by the manufacturer. Using the method indicated above gives:

$$K_{c,g} = \frac{(11.5 \times 10^6)\pi(1.5)^4}{32(8.25)} = 0.693 \times 10^6 \text{ in-lb/rad}$$

$$K_{d,f} = \frac{(11.5 \times 10^6)\pi(2.0)^4}{32(10)} = 1.806 \times 10^6 \text{ in-lb/rad}$$

The gearboxes were oriented as shown in Fig. 8 so that coupling (e) would have the same bore on both ends, and therefore be easier to obtain and balance.

Sufficient data was not available for estimating the turbine power shaft stiffness by the above method. However, the installation manual gave 28 Hz as the torsional natural frequency of the power turbine and shaft suspended as a cantilever. Using this and the value for the polar moment of inertia of the turbine and shaft, which is 10.5 lb-ft^2, the stiffness can be determined as follows:

$$\omega = 2\pi f = \sqrt{K/I}$$

$$2\pi(28) = \sqrt{\frac{K(386)}{10.5(144)}}$$

$$K_i = 0.124 \times 10^6 \text{ in-lb/rad}$$

A lumped inertia modeling technique was used to reduce the distributed inertia of the system into the 10 finite elements shown in Fig. 8. This technique yielded an adequate simulation of low to high frequency phenomena with minimum computation.

As previously noted, the given value for the lumped turbine rotor and power shaft inertia (10) was 10.5 lb-ft^2, or 1512 lb-in^2.

The coupling inertia was determined by lumping the total inertia value supplied by the manufacturer (130 lb-in^2) to the two end pieces, so that each end (2,3,5,6,8,9) was 1/2 the total inertia, or 65.0 lb-in^2.

The 1:1 ratio gearbox inertia (7) was given by the manufacturer as 362.9 lb-in^2 for the total gearbox as a single lumped inertia. The 1:1.47 ratio gearbox inertia (4) was 527.0 lb-in^2 for the total box referred to the low speed shaft. The use of this referred value will be explained subsequently.

The value for the compressor rotor and shaft inertia (1) was given in the installation manual as 1769 lb-in^2. However, as previously explained, horsepower limitations made necessary the removal of some of the compressor stages. In the case analyzed, all but the first three stages were removed (Fig. 9), as planned for initial operation. It was then necessary to determine the reduction in inertia caused by

Figure 9. PHOTOGRAPH OF THREE-STAGE TEST ROTOR

the removal of the blades from the remaining stages. This was estimated by weighing five blades from each of the stages removed and then obtaining an average blade weight. This weight was then multiplied by the square of the radius from the center of rotation to a point 1/4 of the distance from the blade root to its tip (the assumed mass center). Multiplication of this value (the wr^2) by the total number of blades in the stage, resulted in the reduction of inertia due to removal of blades from that stage. These values are compiled in Table 1. The total reduction in inertia is 240 lb-in^2, which leaves an overall inertia of 1529 lb-in^2 for the compressor rotor and shaft.

The amount of damping within the system was not given and is not easily obtained. However, in torsional systems the damping is generally found to be small, and the absolute value used is of little significance [17] (unless a gross error is made in the value employed) and should not affect the determination of resonance points. The damping occurs in the system due to viscous effects at the bearings and from the blades moving through the air. Some structural damping will also be present in the shafts. Due to the lack of knowledge in this area, a value of 12 in-lb-sec/rad was assumed in conjunction with each torsional stiffness value for computational purposes. Knowledge of exact values for damping would allow determination of which, if any, resonance frequencies are actually critical speeds. This was not possible in this analysis.

The torsional vibration analysis was conducted by forming an equivalent model of the system in Fig. 8. All values of inertias and elastic constants were referred to the low speed shaft, which is the

TABLE 1

Inertia Reduction per Compressor Stage

Stage #	Average Wt. (lb)	Radius (in)	wr^2 ($lb\text{-}in^2$)	# of Blades	Total wr^2 $lb\text{-}in^2$
14	0.00337	5.58	0.1049	118	12.38
13	.00358	5.59	0.1118	118	13.20
12	.00361	5.59	0.1128	118	13.31
11	.00406	5.60	0.1273	110	14.00
10	.00407	5.60	0.1276	110	14.04
9	.00589	5.58	0.1834	98	17.97
8	.00642	5.56	0.1985	98	19.45
7	.01093	5.465	0.3264	86	28.07
6	.01322	5.41	0.3869	76	29.41
5	.01932	5.38	0.5592	64	35.79
4	.02734	5.29	0.7651	56	42.84
					240.46

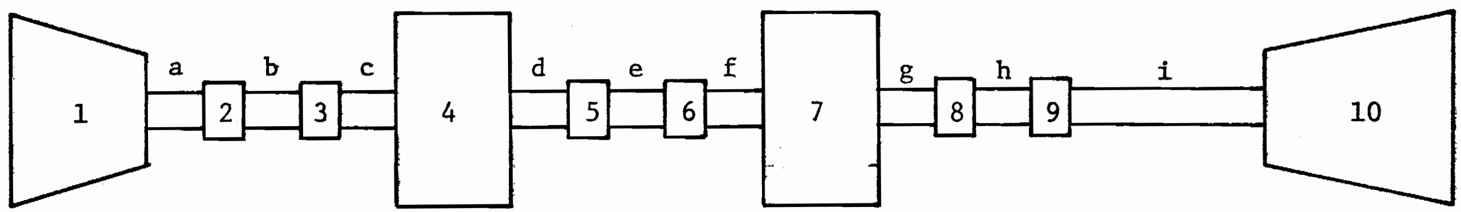
engine side of the system. This was accomplished by multiplying all values on the driven side of the system by $(N_2/N_1)^2$, that is, by the square of the speed ratio between the two sides. The resultant equivalent system is shown in Fig. 10 and the equivalent inertias and elastic constants are summarized in Tables 2 and 3, respectively.

The system, modeled in this manner, was thus in a form suitable for analysis by mechanical impedance methods. The impedance modeling program developed by Mitchell [11] performed this analysis and was available to the author.

With the data in this form and with the matrix loading set-up established, the program was used to determine the mobility amplitude and phase relationships for each system component versus frequency. By locating the points in the program output at which the mobility rapidly rises to a peak or, alternatively, where the phase relationship shifts from "+" to "-", the natural frequencies are obtained.

Using this program and the equivalent data previously developed, the torsional natural frequencies for the system were obtained. These resonance points occurred at speeds of 1655, 6072, 12228, 17233, and 30000+ rpm. Two of these, 12228 and 17233 rpm, fell within the planned operating range and were undesirable. To gain a better understanding of these results, so that corrective action could be attempted, a plot of the mode shapes at the resonance points was useful. These are given in Figs. 11, 12, 13 and 14.[1] Figure 14 indicated that

[1]The mode shapes given in these diagrams should show a phase reversal occurring at each gearbox. However, in the interest of clarity, this phase reversal has been omitted herein to yield a more conventional mode shape. Also, the resonance values given are accurate only to within ±3% for the system as programmed. Therefore, the expected



Numbers represent lumped inertias

Letters represent torsional springs and dampers

Figure 10. EQUIVALENT SYSTEM SCHEMATIC

TABLE 2

Polar Moments of Inertia (Initial Analysis)

#	Identification	Actual ($lb\text{-}in^2$)	Equivalent ($lb\text{-}in^2$)
1	compressor	1529.0	3210.0
2	coupling	65.0	136.7
3	coupling	65.0	136.7
4	gearbox (1:1.47)	527.0	527.0
5	coupling	65.0	65.0
6	coupling	65.0	65.0
7	gearbox (1:1)	363.0	363.0
8	coupling	65.0	65.0
9	coupling	65.0	65.0
10	turbine	1512.0	1512.0

TABLE 3

Shaft Torsional Data (Initial Analysis)

#	Identification	Torsional Stiffness $\left(x\ 10^{-6}\ \frac{\text{in-lb}}{\text{rad}}\right)$		Torsional Damping $\left(\frac{\text{in-lb-sec}}{\text{rad}}\right)$	
		Actual	Equivalent	Actual	Equivalent
a	compressor shaft	3.735	7.854	12.0	24.9
b	coupling	5.8	12.194	12.0	24.9
c	1:1.47 gearbox output	0.693	1.457	12.0	24.9
d	1:1.47 gearbox input	1.806	1.806	12.0	12.0
e	coupling	5.8	5.8	12.0	12.0
f	1:1 gearbox output	1.806	1.806	12.0	12.0
g	1:1 gearbox input	0.693	0.693	12.0	12.0
h	coupling	5.8	5.8	12.0	12.0
i	turbine shaft	0.124	0.124	12.0	12.0

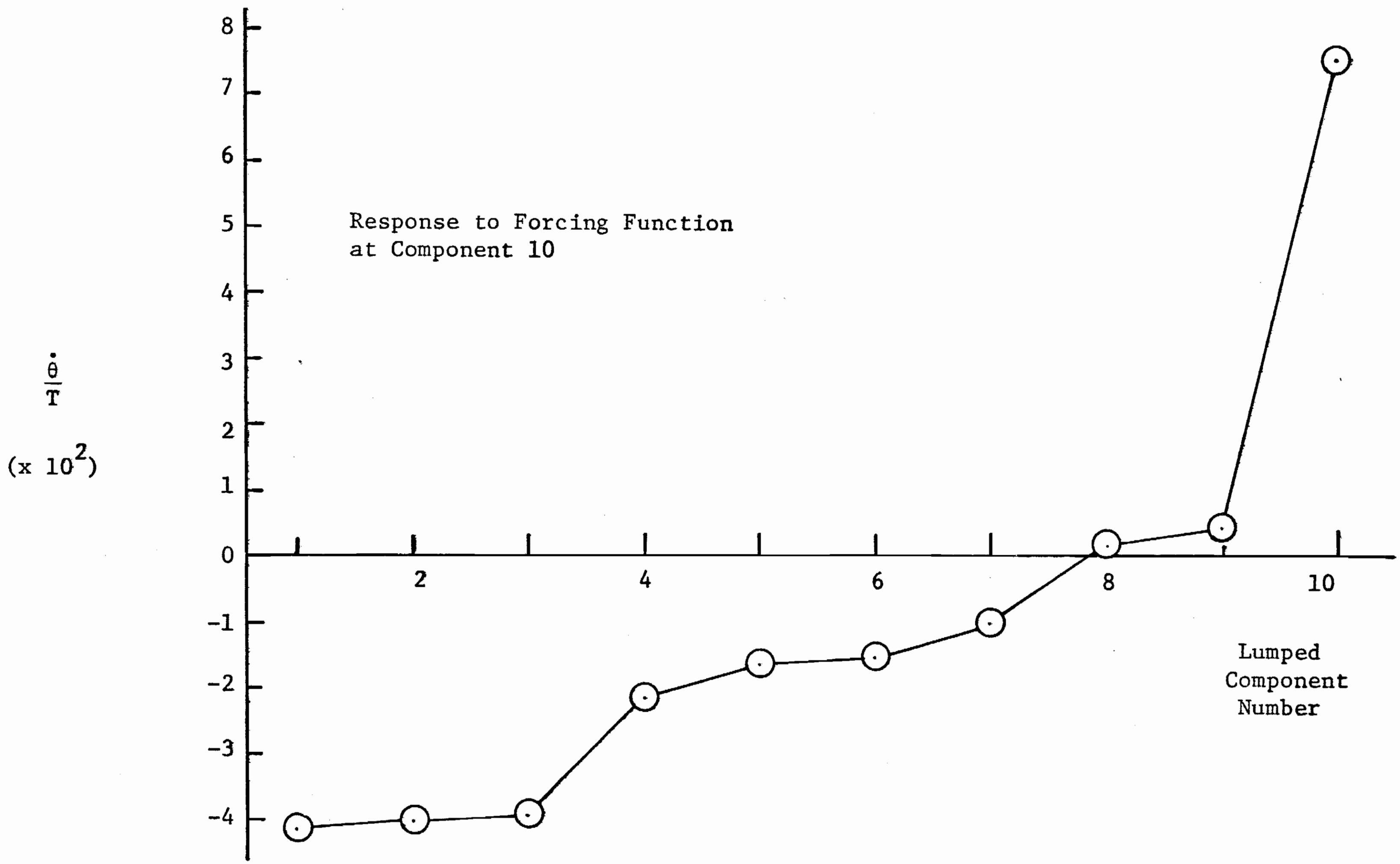


Figure 11. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM - 1655 RPM

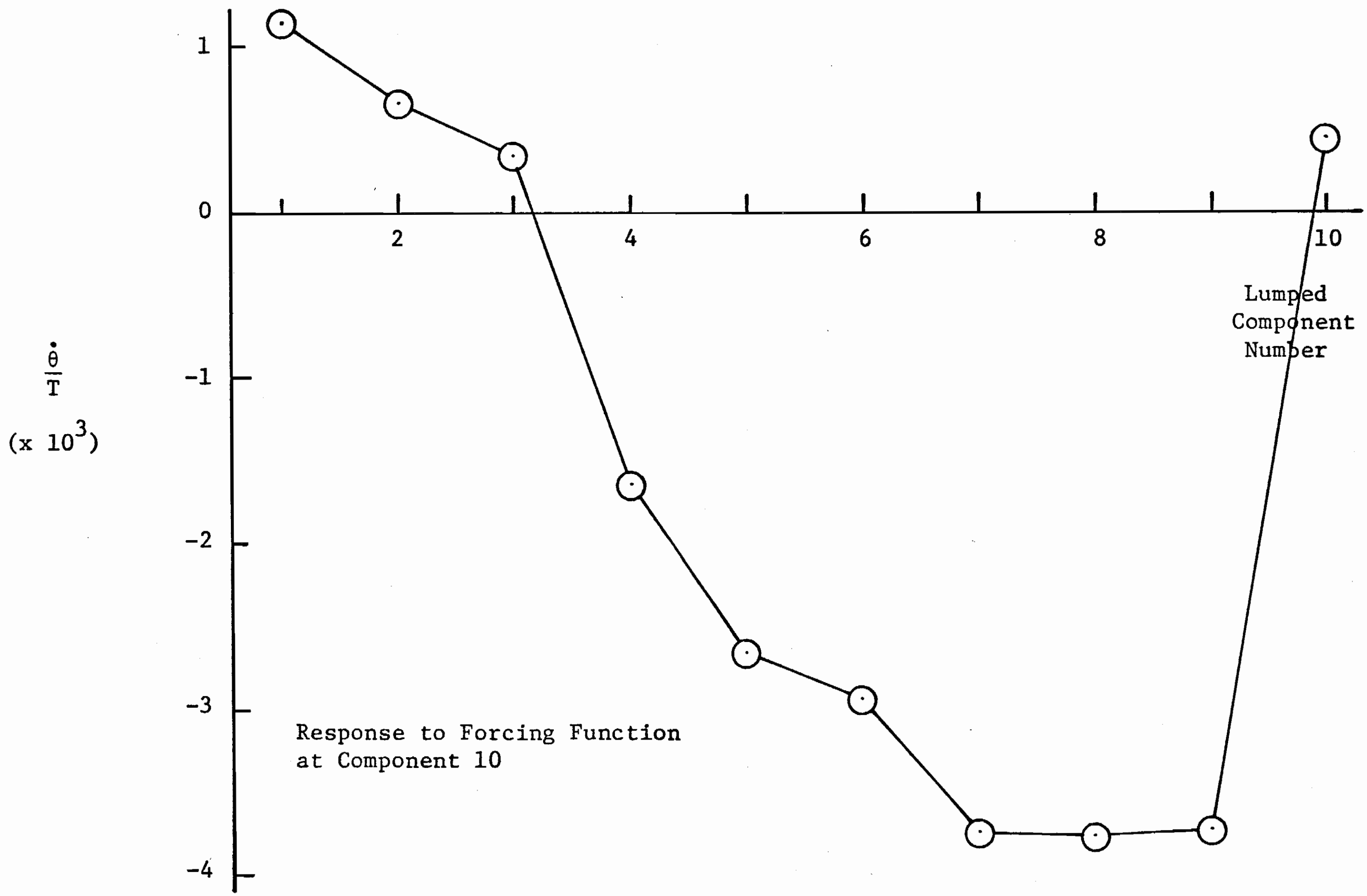


FIGURE 12. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM -- 6072 RPM

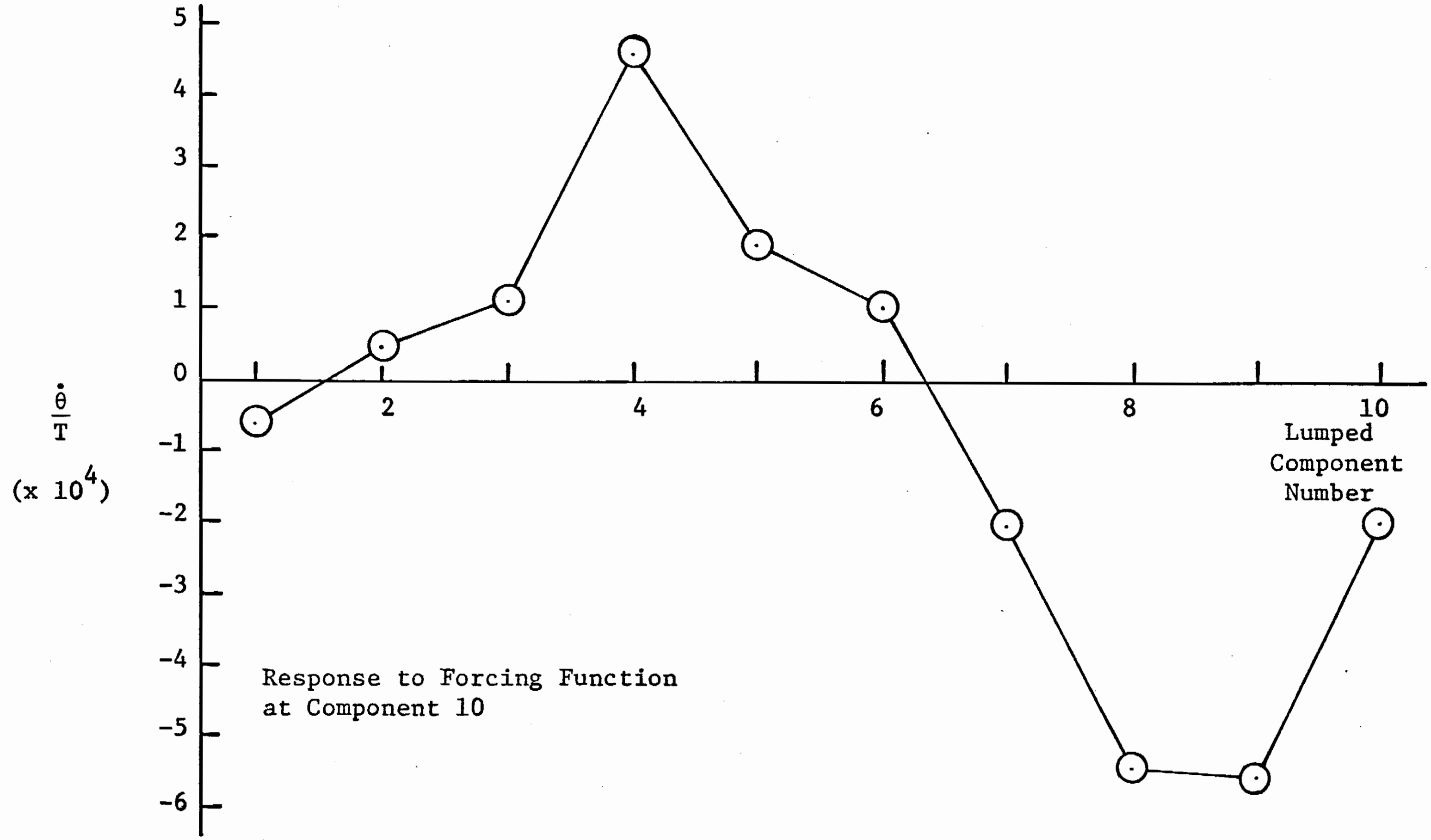


Figure 13. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM - 12228 RPM

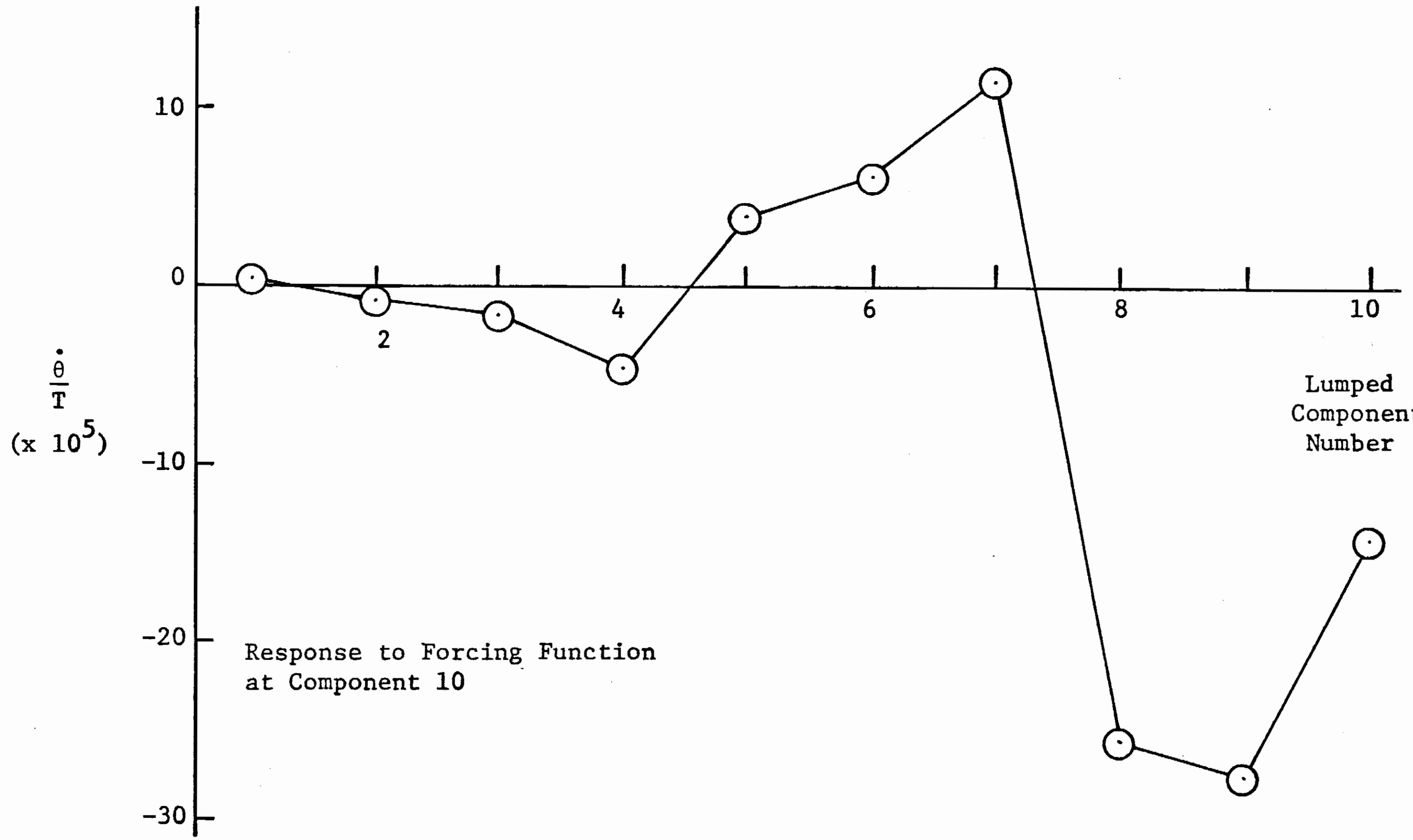


Figure 14. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM - 17233 RPM

shaft (g), the 1:1 gearbox input shaft, was twisted excessively in relation to the other shafts in the system, and thus appeared to be the controlling element. Stiffening of this shaft was expected to reduce the magnitude of this problem and, since this was a nodal point, the change should produce a maximum effect in shifting the resonance point. If this shaft could be stiffened without adding to the inertia, the frequency would be shifted upwards, possibly out of the operating range.

The problem then was to increase the shaft stiffness. Since this was part of the gearbox, changes in shaft size would result in higher costs and increased delays, as a standard model could no longer be used. However, referring to Table 3, it was noted that the output shaft (f) was nearly three times as stiff as shaft (g) and, since this was the 1:1 ratio gearbox, it was a simple matter to reverse the gearbox and to drive shaft (f) instead of shaft (g). This produced an additional benefit in that shaft (f) was a nodal point at the 12228 rpm resonance (Fig. 13) and, by reducing stiffness at this point in the system, the resonance frequency would be lowered. Thus, two desirable stiffness changes were achieved without any increase in the inertia value.

Conducting the vibration analysis for this new system configuration indicated that torsional resonant frequencies would now occur at speeds of 1655, 5659, 10792 and 25171 rpm. The expected results have

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number of axis crossings for the given mode shape may not occur, depending on the error involved. Finally, all analysis was conducted as a response to a forcing function at element 10 and assumes an input force of unity.

occurred and have produced a system which is predicted to be virtually trouble-free in the desired operating range. Examination of the corresponding mode shapes in Fig. 15 and Fig. 16 indicate that a significant reduction in amplitude has also been achieved.

System Modifications

Consideration of the total system design indicated that the standard coupling length would not provide sufficient room for the compressor rear bearing housing. To accomodate this support, a coupling with a 24 in. spacer was selected. This coupling replaces coupling (b) and results in a reduction of shaft stiffness to 1.4×10^6 in-lb/rad and an increase in inertia of (2) and (3) to 94.5 lb-in^2.

The gearbox manufacturer indicated that the 1:1 ratio gearbox would require a redesigned housing, which would impose additional costs and delays. For this reason, a new investigation of the numerous combinations which would provide the desired overall ratio of 1.47 was begun. After consideration of these alternatives, selection of two 1:1.2 ratio gearboxes appeared to offer the most advantageous combination. This selection allowed a forward mode operation output shaft turning in the direction of engine shaft-rotation with either a direct coupled or a 1.44 speed-increaser system. The reverse mode can operate as either a 1.2 increaser or 1.2 decreaser system. These combinations provided coverage of all speeds in the range 12000-20400 rpm in either rotational direction with some overrange as indicated in

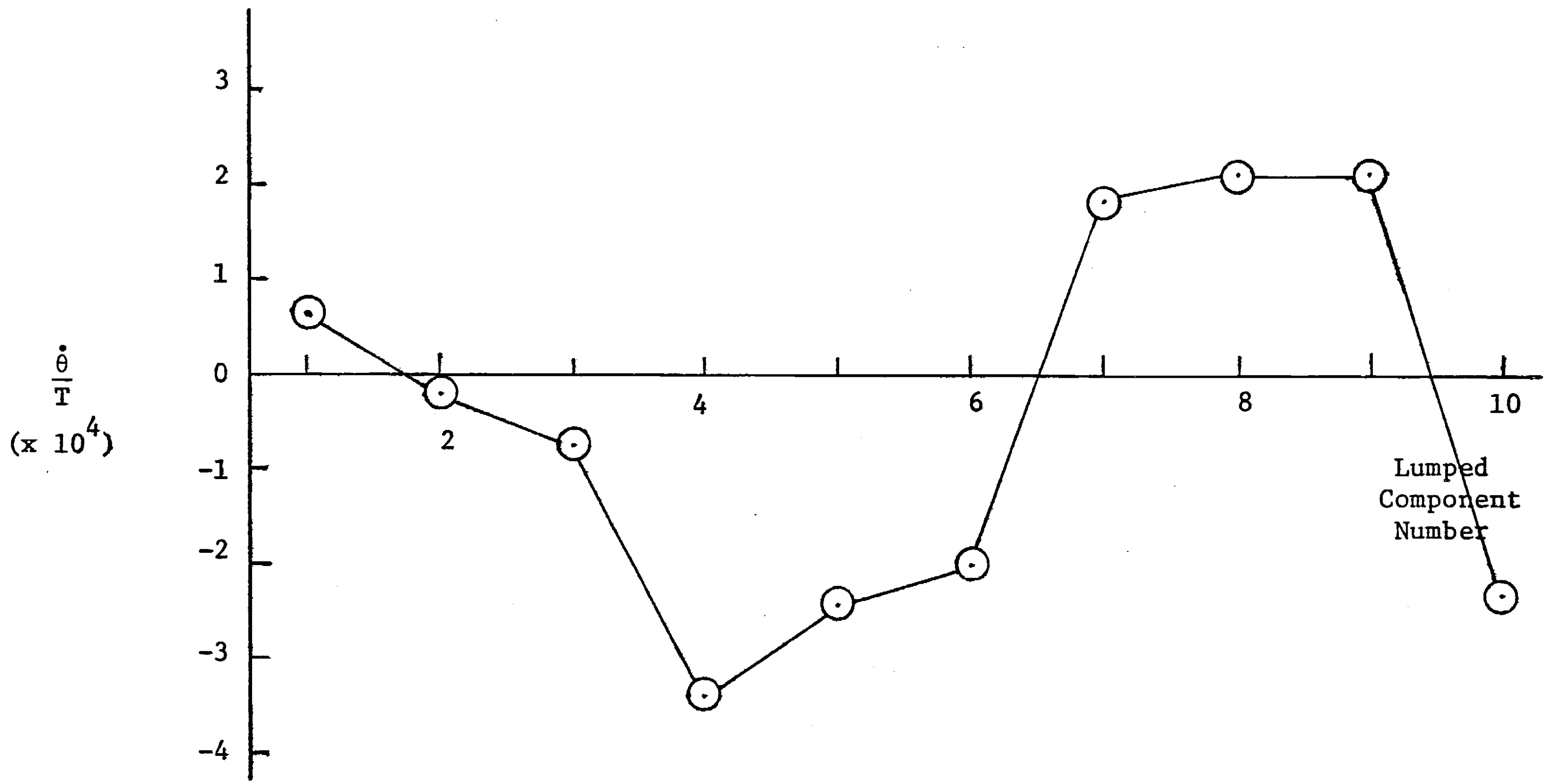

Figure 15. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM WITH GEARBOX REVERSED - 10792 RPM

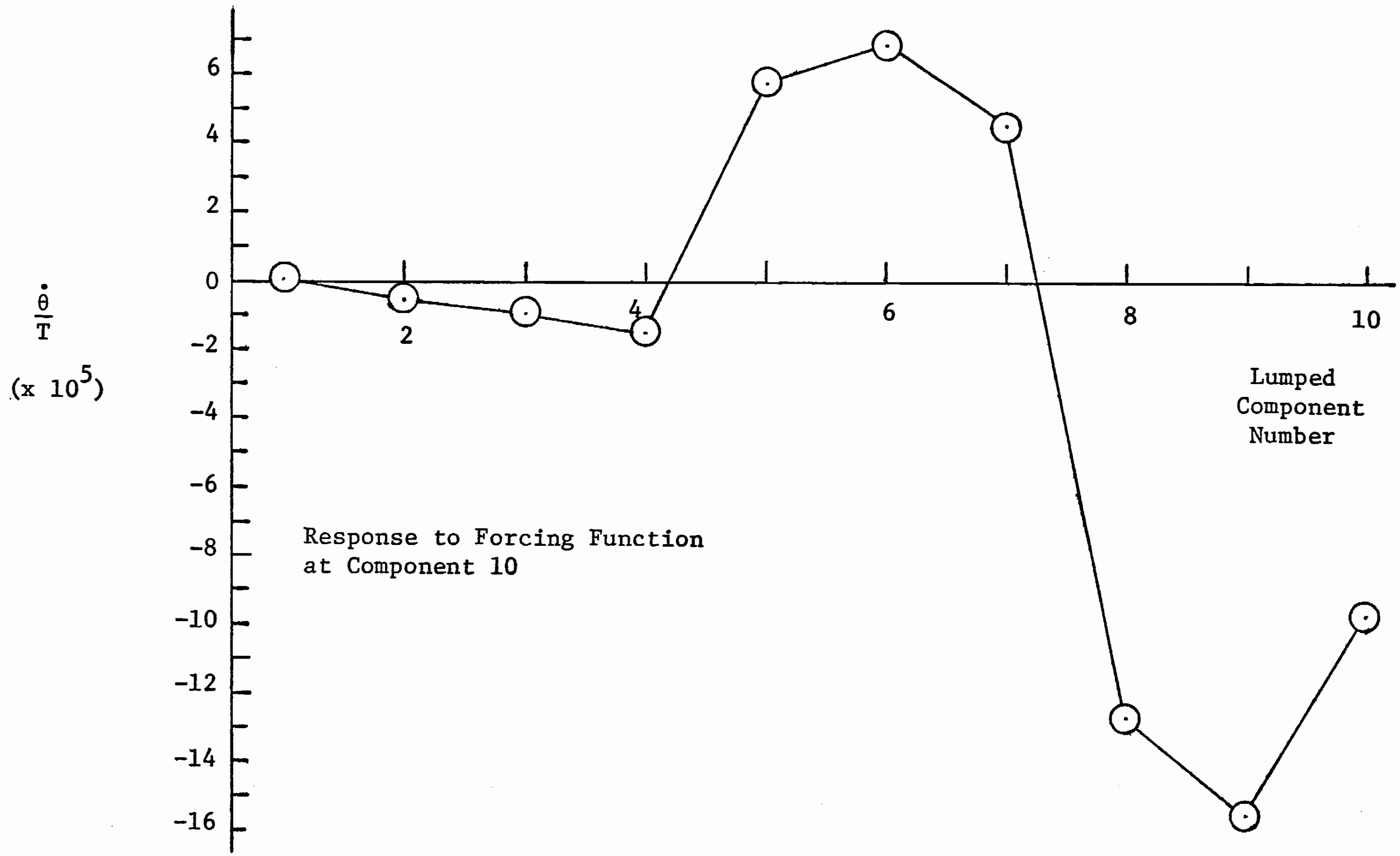

Figure 16. MODE SHAPE FOR CANDIDATE COMPRESSOR DRIVE SYSTEM WITH GEARBOX REVERSED - 25171 RPM

Table 4. This is highly desirable from the standpoint of test cell versatility.

The new gearboxes also resulted in new inertia values which were 576 lb-in^2 referred to the low speed shaft. The new system with the changes in gearboxes, the coupling, and shaft configuration resulted in the torsional data given in Tables 5 and 6. Calculation procedures followed those originally developed for the first candidate system design previously discussed. The vibration analysis based on this data indicated that torsional natural frequencies would occur at 1703, 4833, 8979, 23874 and 28100 rpm. Thus, the longer coupling produced an additional benefit of an even wider operating range between torsional resonance points. The mode shapes for any desired frequencies can be obtained from the program output. For example, the resonant frequencies bounding the operating range are shown in Fig. 17 and Fig. 18. In addition, a frequency response plot of component (5), the end of the middle coupling which undergoes the largest amplitude response within the operating range, is included in Fig. 19 as an indication of the relative amplitude versus frequency and of the typical output from the computer program. Note that the lack of knowledge about damping makes absolute determinations of amplitudes impossible.

Further modifications to the system such as changes in coupling stiffness were studied but did not result in improvements sufficient to warrant the necessary changes, or produced more severe vibration situations. Therefore, the above system was selected as the optimum choice.

TABLE 4

System Operating Modes and Ranges

Forward Mode	Speed Range (rpm)	Resonance Speeds lower	Resonance Speeds upper
Direct Coupled	12,000 - 17,000	2182	19,353
Two Gearboxes	17,280 - 24,480	8979	23,874
Reverse Mode			
Increasing Box	14,400 - 20,400	6599	23,273
Decreasing Box	10,000 - 14,166	6399	16,094

TABLE 5

Polar Moments of Inertia (Final Analysis)

#	Identification	Actual ($lb\text{-}in^2$)	Equivalent ($lb\text{-}in^2$)
1	compressor	1529.0	3166.4
2	coupling	94.5	196.0
3	coupling	94.5	196.0
4	gearbox (1:1.2)	576.0	829.0
5	coupling	65.0	93.6
6	coupling	65.0	93.6
7	gearbox (1:1.2)	576.0	576.0
8	coupling	65.0	65.0
9	coupling	65.0	65.0
10	turbine	1512.0	1512.0

TABLE 6

Shaft Torsional Data (Final Analysis)

#	Identification	Torsional Stiffness $\left(\times 10^{-6} \frac{\text{in-lb}}{\text{rad}}\right)$		Torsional Damping $\left(\frac{\text{in-lb-sec}}{\text{rad}}\right)$	
		Actual	Equivalent	Actual	Equivalent
a	compressor shaft	3.735	7.746	12.0	17.28
b	coupling	1.40	2.903	12.0	17.28
c	1:1.2 gearbox output	0.693	1.437	12.0	17.28
d	1:1.2 gearbox input	1.806	2.601	12.0	14.4
e	coupling	5.80	8.352	12.0	14.4
f	1:1.2 gearbox output	0.693	0.998	12.0	14.4
g	1:1.2 gearbox input	1.806	1.806	12.0	12.0
h	coupling	5.80	5.80	12.0	12.0
i	turbine shaft	0.124	0.124	12.0	12.0

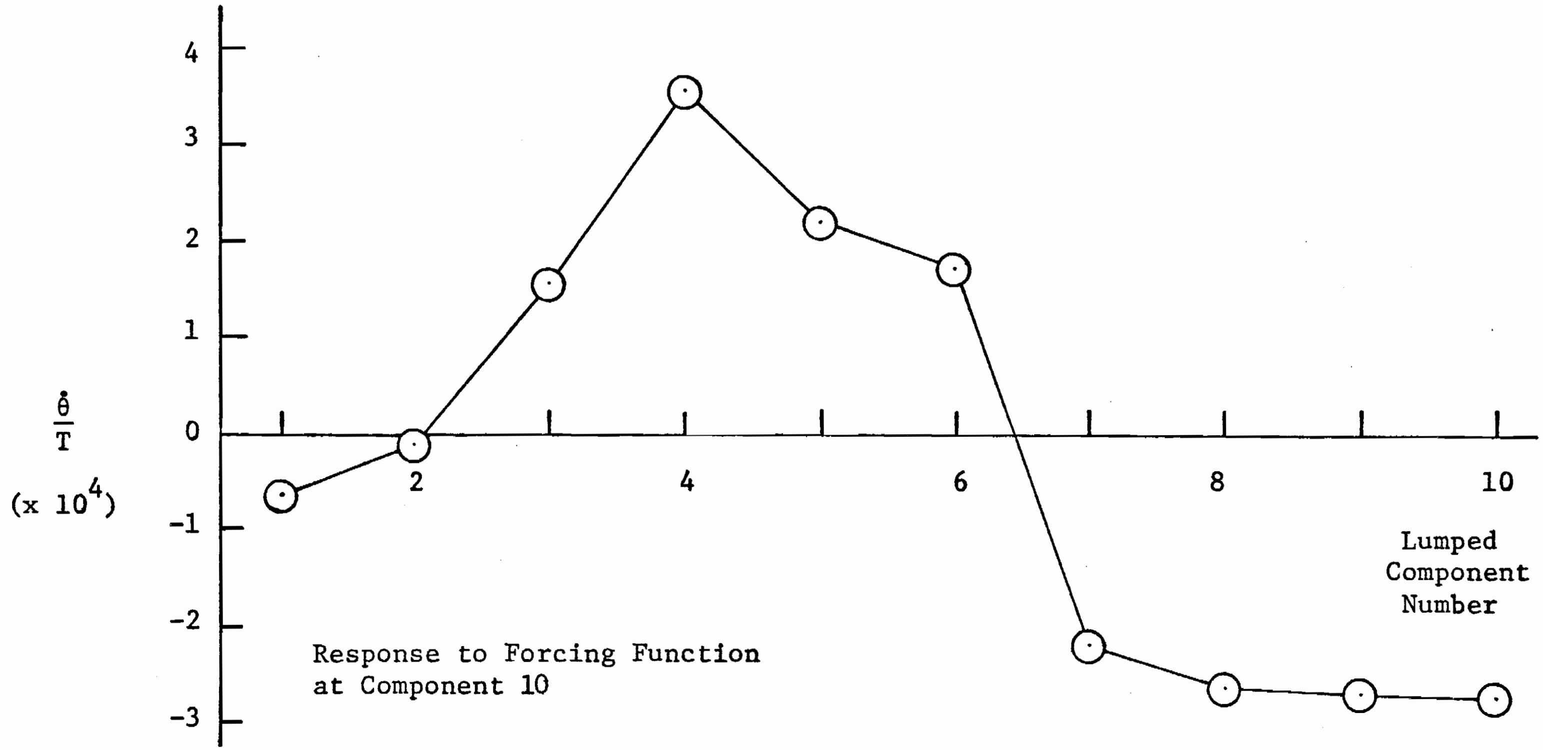

Figure 17. MODE SHAPE FOR FINAL SYSTEM WITH 1.2 RATIO GEARBOXES - 8979 RPM

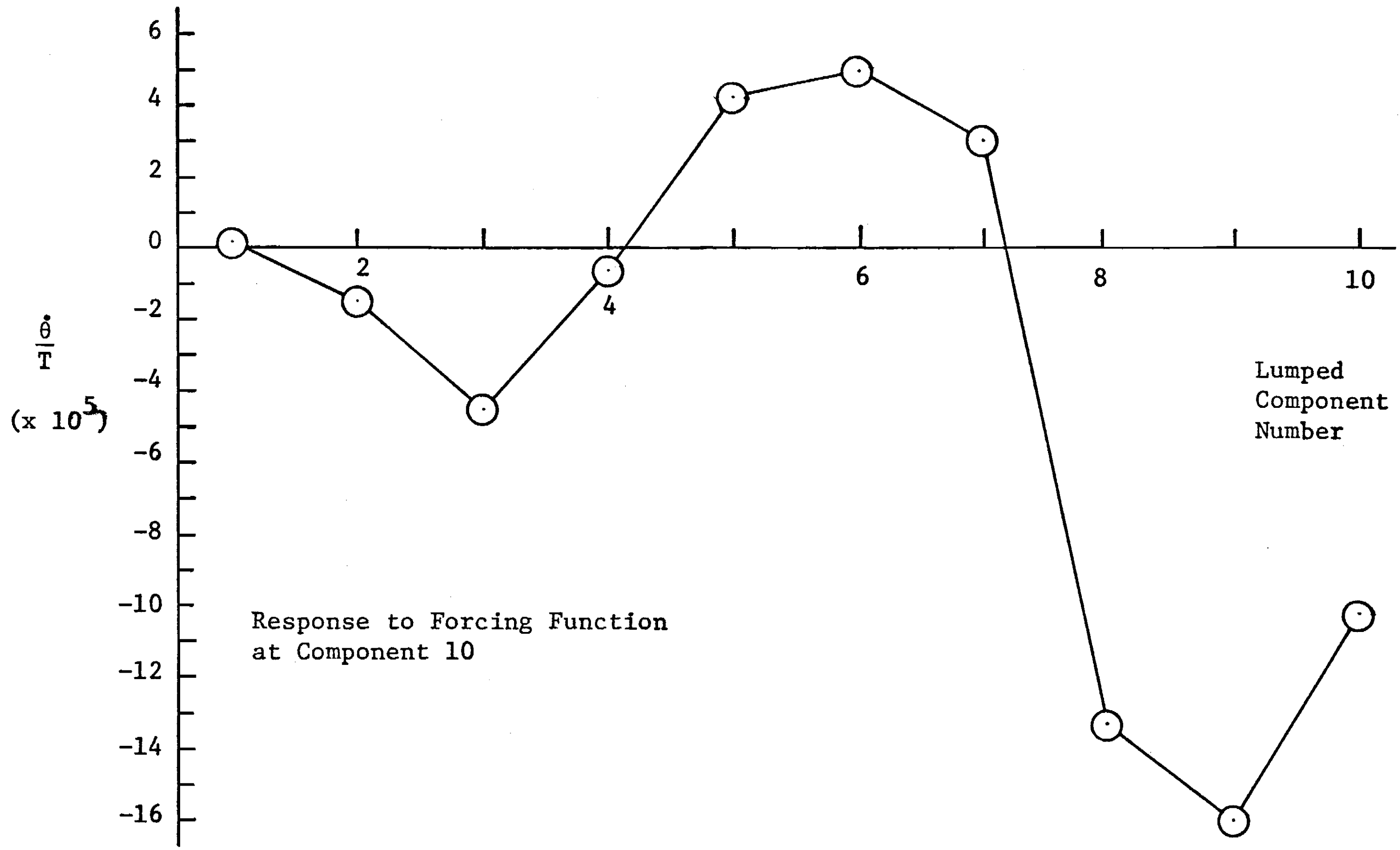

Figure 18. MODE SHAPE FOR FINAL SYSTEM WITH 1.2 RATIO GEARBOXES - 23874 RPM

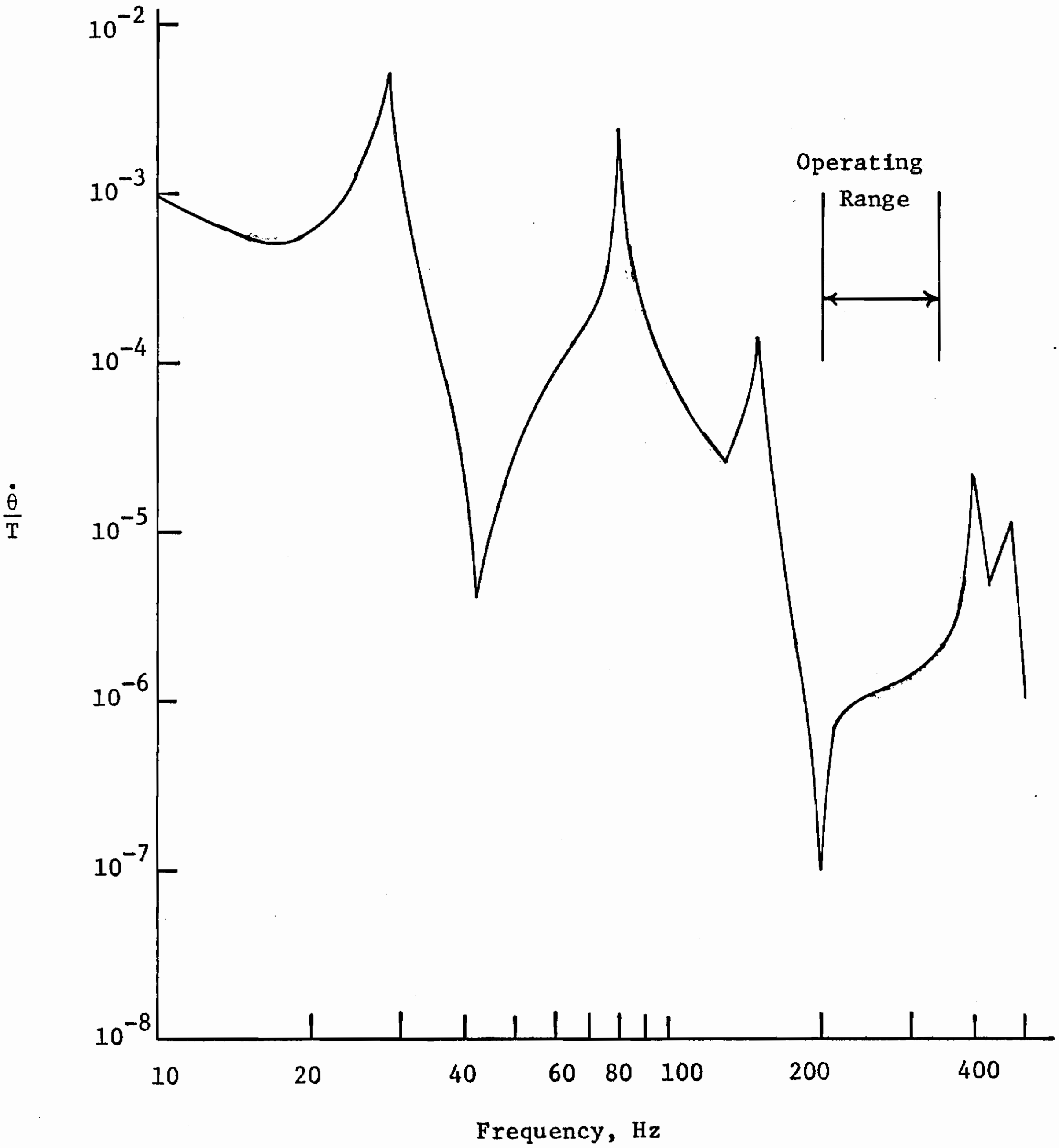

Figure 19. FINAL SYSTEM FREQUENCY RESPONSE FOR LUMPED COMPONENT #5

Because the system may also be operated in direct coupled, 1.2 increaser, and 1.2 decreaser modes, torsional vibrations under these conditions were also analyzed. The direct coupled system results in torsional resonances at speeds of 2182 and 19353 rpm, which are outside of the intended operating range in this mode. The single 1.2 ratio increasing system results in torsional resonances at speeds of 1871, 6599 and 23273 rpm, which are also outside the intended operating range. The 1.2 ratio decreasing system results in torsional resonances at speeds of 1990, 6399 and 16094 rpm, which also fall outside the intended operating range. Thus, all system modes operate in ranges free from torsional resonances. The bounding resonance frequencies are summarized in Table 4.

Due to the nature of the estimations involved in the data used for the vibration analysis, particularly values associated with the compressor, additional analyses were made in which values for the compressor inertia were decreased by 20% and for the compressor shaft stiffness were increased by 20%. This was expected to shift the resonance frequencies to higher values. The resultant resonance points of 1703, 4993, 9276, 23874, 25482 and 28100 rpm showed a very slight upward shift in the first three natural frequencies of about 3.5%. This result indicated that the final design is stable with regard to resonant frequencies within the operating range such that large inaccuracies in estimating system parameters should not adversely affect expected system performance. Thus the predicted natural frequencies may prove inaccurate, but the system characteristics are expected to be acceptable.

Instrumentation and Controls

The instrumentation and controls provided in the facility were those necessary for monitoring engine parameters and regulating engine functions to meet test requirements. Instrumentation necessary for obtaining on-rotor test data dealing with stall and surge investigations was not considered but has been discussed by Carter [2] and DeFelice [16].

Instrumentation provided includes pressure, temperature, fuel flow, power, speed and vibration monitoring equipment. The instruments are used to insure safe operation of all system components within their respective design limits and as indications of performance parameters required to establish a data base. All monitored parameters are displayed on panel meters within the control room for operational convenience (Fig. 20). A listing of the instrumentation provided is given in Table 7.

Pushbutton switches are provided for control of all test cell electrical power. 28vDC is provided for operation of the ignition unit, overspeed switch, compressor starting valve, and the torque sensor. 110vAC is provided for operation of auxiliary fuel and lube pumps, solenoid valves, and a test cell warning light. The solenoid valves are located in the fuel and lube oil supply lines just after the respective reservoirs and just prior to the engine pump inlets. These solenoids are normally closed so that electrical power must be provided to maintain open supply lines. In the event of loss of power to the test cell, these solenoids will close and terminate system

TABLE 7

Test Cell Instrumentation

Pressure Gages	Range (psig)
P1 Starting Bleed Valve (E)*	0 - 60
P2 Compressor Discharge Static (E,C)	(E) 0 - 300, (C) 0 - 100
P3 Lube-In (E,C)	-30 Hg - 15 psig
P4 Lube Pump Discharge (E,C)	0 - 100
P5 Scavenge Discharge (E,C)	0 - 60
P6 Sump Vent (E,C)	-30 Hg - 15 psig
P7 Fuel In (E)	0 - 60
P8 Fuel Control Discharge (E)	0 - 1000
Temperature Gages	**Range (°F)**
T1 Compressor Inlet (E,C)	0 - 300
T2 Compressor Discharge (E,C)	(E) 0 - 1000, (C) 0 - 300
T3 Power Turbine Inlet (E)	0 - 2000
T4 Exhaust Gas (E)	0 - 2000
T5 Lube-In (E,C)	0 - 300
T6 Scavenge Oil (E,C)	0 - 500
FF Fuel Flow Rate (E)	0 - 5 gpm
N Rotational Speed (E,C)	0 - 18,250 rpm
Q Torquemeter (E)	0 - 1500 lb-ft
V Vibration (E,C)	0 - 20 mils
t Run Time (system)	0 - 9999 hrs
ϕ Power Lever Position (E)	0 - 360°

*Letters in parentheses indicate measurement location, E-engine, C-test compressor

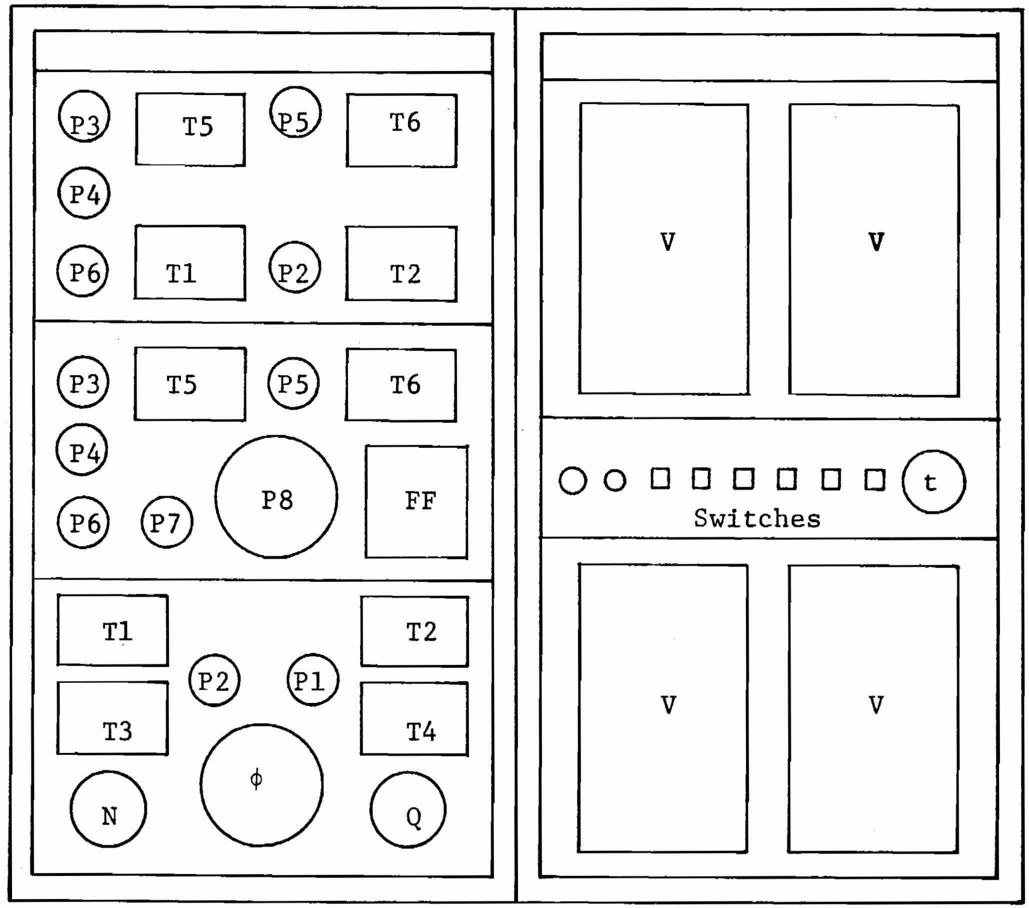

Figure 20. CONTROL PANEL LAYOUT

operation. 220vAC is provided for operation of the hydraulic starting system.

Power and load control levers are used to regulate operating conditions. The power lever setting determines the output shaft speed within the limits of the fuel control. The load lever setting establishes the power available at that shaft speed. Based on these two settings, the fuel control adjusts the variable guide vane position to regulate airflow and meters fuel flow to the combustors to maintain engine parameters within acceptable limits.

A compressor loading duct control allows adjustment of the test compressor exhaust flow area, thereby providing remote control of the compressor back pressure during tests.

Mount Design

The mounting system is designed to perform certain primary functions. Among these are:

1) Accomodation of radial and axial thermal expansion of the engine.
2) Location and alignment of components under conditions of torque and thrust loadings.
3) Integration of the total system so that the resulting dynamic system is free from any undesirable vibratory responses.
4) Provision for relocating components for various test configurations.

The design of the mounting system required utilization of basic principles for stress and strain and determination of deflections due to loads so that support members could be properly sized. The application required a unique structure which was intended for safety so that an overdesign, rather than optimization, resulted. This should insure long life and extra safety margins as required by the intended research application where unusual load situations may occur.

The system concept involved a rigid base which provided a simple means for locating and supporting all components for various tests as indicated in Table 4. Each component was then supported on a stand which provided the necessary vertical height, strength to maintain position under loads, and a means for individually positioning or removing a component from the base structure. Bulkheads were provided for those components requiring instrumentation and monitoring equipment so that all connectors could be brought to a common point. From this point, runs could be made to displays in the control room.

With this concept in mind, structural members were selected and sized based on expected loads at the engine mount points, since these were the maximum expected within the system. Other members for the remainder of the system were then sized on the basis of the results of the engine mount design. Maximum allowable mount reactions for the engine were specified in the Installation Manual. However, these reactions are based on flight maneuver conditions involving "g" loads and are not applicable here. Therefore, the design was based on loads due to engine weight and torque developed with a factor of safety of 4 included. Members were chosen to provide maximum rigidity in the direction of principal loading without requiring usage of structural pieces which would cause fabrication difficulties. Thus, equal leg angle was employed to resist bending loads, steel pipe to resist column buckling, and I-beams for maximum base support strength and rigidity. Steel plates were also used to provide large surface areas to accomodate any necessary lateral and horizontal relocations.

Calculations of loads and resultant deflections of members for the engine mount are given in Appendix C. No deflections larger than 0.0025 inches occurred in any members as sized and all stress levels were such that the factor of safety exceeded 13 under conditions of rated engine torque.

The possibility of a bearing failure resulting in sudden engine seizure was also considered. The Installation Manual indicates that mount reactions due to the torque generated by the sudden seizure will be approximately equal to the mount reactions due to 4 times the "normal" engine torque. Under this condition the stress levels reduce the factor of safety to approximately 4, which is the factor of safety established as a goal of this mount design.

The design features of the resultant system support structure may be summarized as follows. The base is a welded fabrication of wide-flange and I-beam steel sections to form a solid foundation (Fig. 21). The I-beam cross members are located so as to provide a support for the vertical columns. Consideration of the possible use of the facility for compressor noise studies makes movement of the total assembly to the front of the test cell area desirable so that reflections from side walls may be avoided and wider angle sound fields

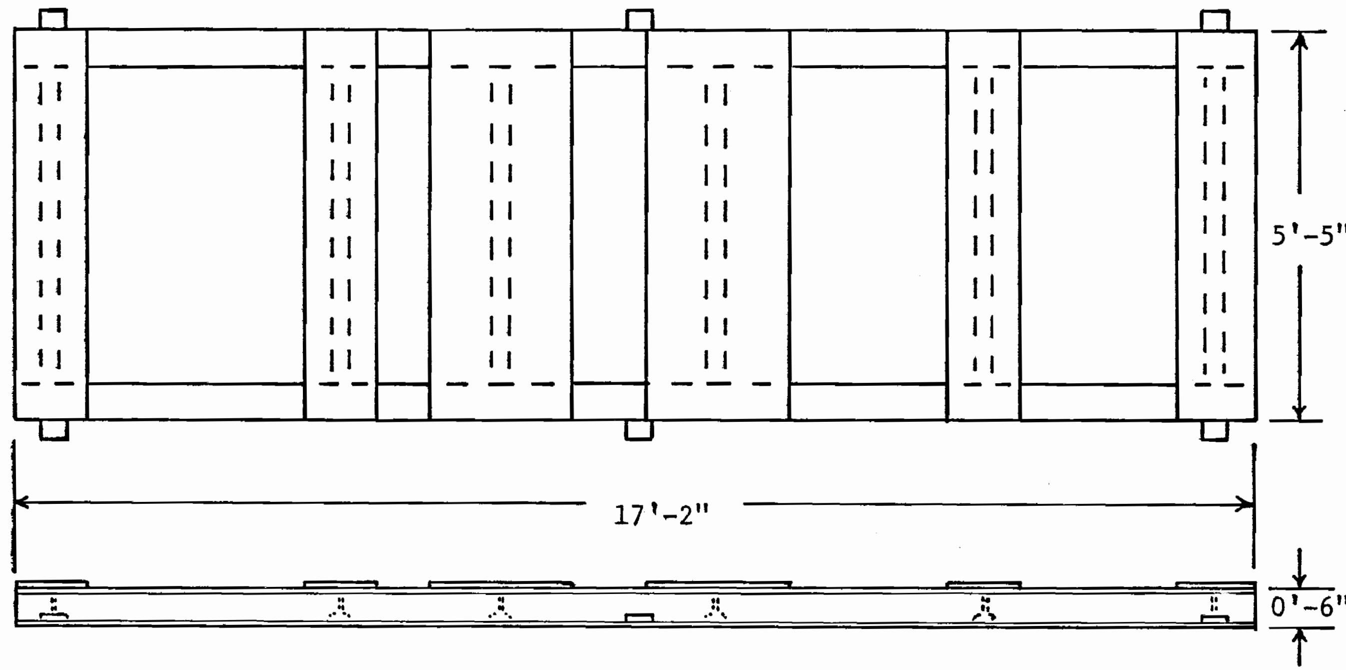

Figure 21. SYSTEM MOUNTING BASE

investigated. To provide for this movement, the structure has been designed so that grooved wheel casters may be added to run on an inverted angle-iron track and thus form a "rail system" on which to move the structure. Additional rigidity has been included in the base frame for this reason. Also, flanges extending from the sides of the base can be used to tie the system to the floor either with or without the rail system.

Base plates positioned over each I-beam cross member provide the surface area necessary for convenient lateral and horizontal positioning adjustments of the component supports. These base plates are bolted through the wide-flange side members to allow movement when needed with various configurations.

Each vertical support comprises either a rectangular angle-iron frame (for engine and compressor supports) or a base plate (for gearbox supports) welded to four steel pipe columns at the corners as shown in Figs. 22 and 23. These supports provide the necessary rigidity and vertical alignment needed by the system. Base plate "feet" welded to each column provide a convenient anchor system and allow repositioning. Adjustments in vertical heights are obtained by adding shims. The compressor and engine as mounted, plus the three configurations of the system support structure are shown in Figs. 24, 25, 26, 27, and 28.

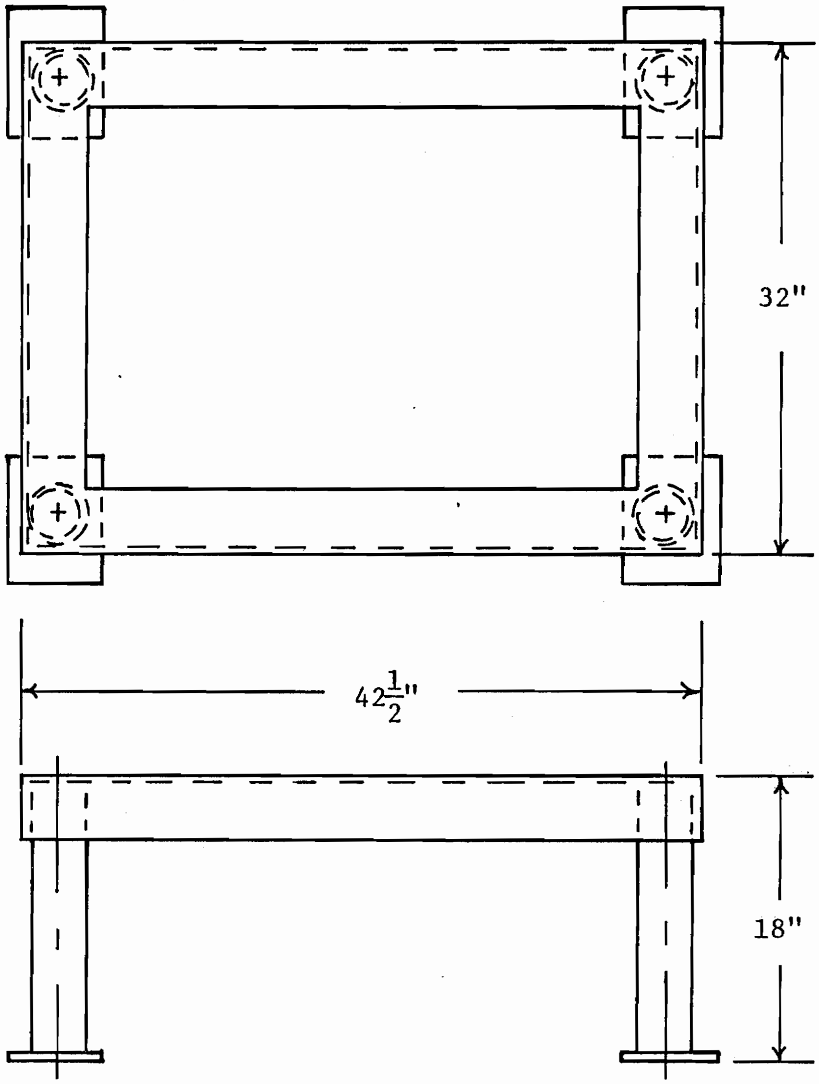

Figure 22. ENGINE AND COMPRESSOR SUPPORT

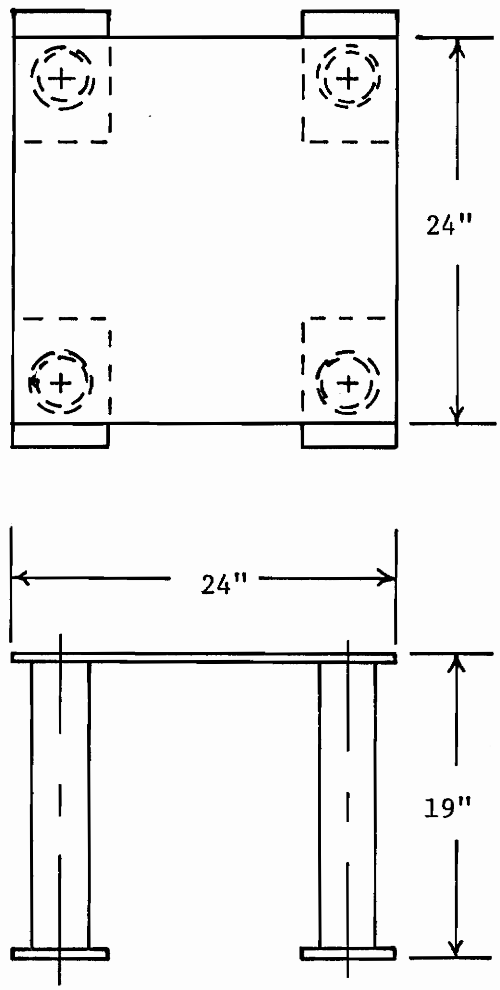

Figure 23. GEARBOX SUPPORT

Figure 24. PHOTOGRAPH OF MOUNTED COMPRESSOR

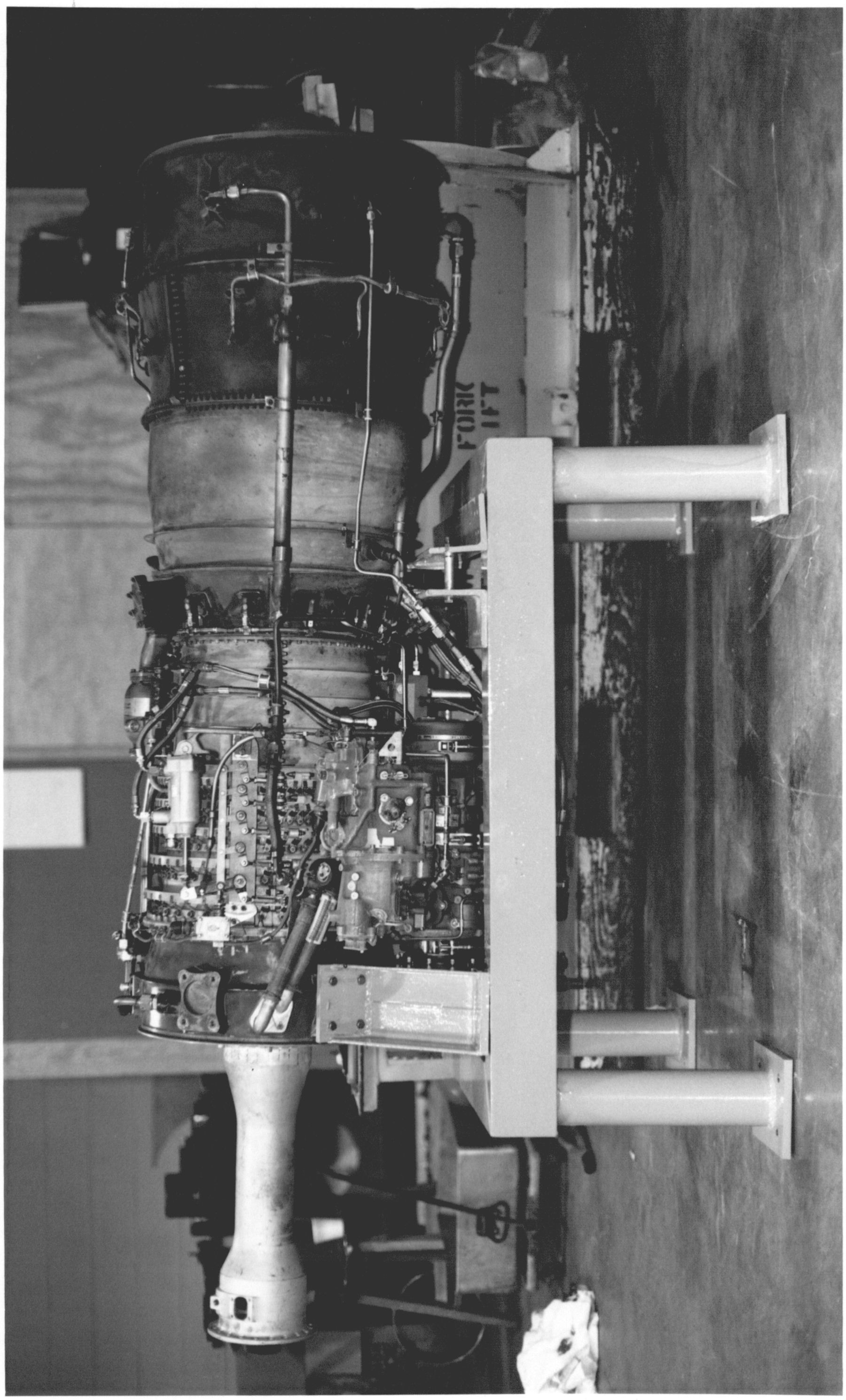

Figure 25. PHOTOGRAPH OF MOUNTED ENGINE

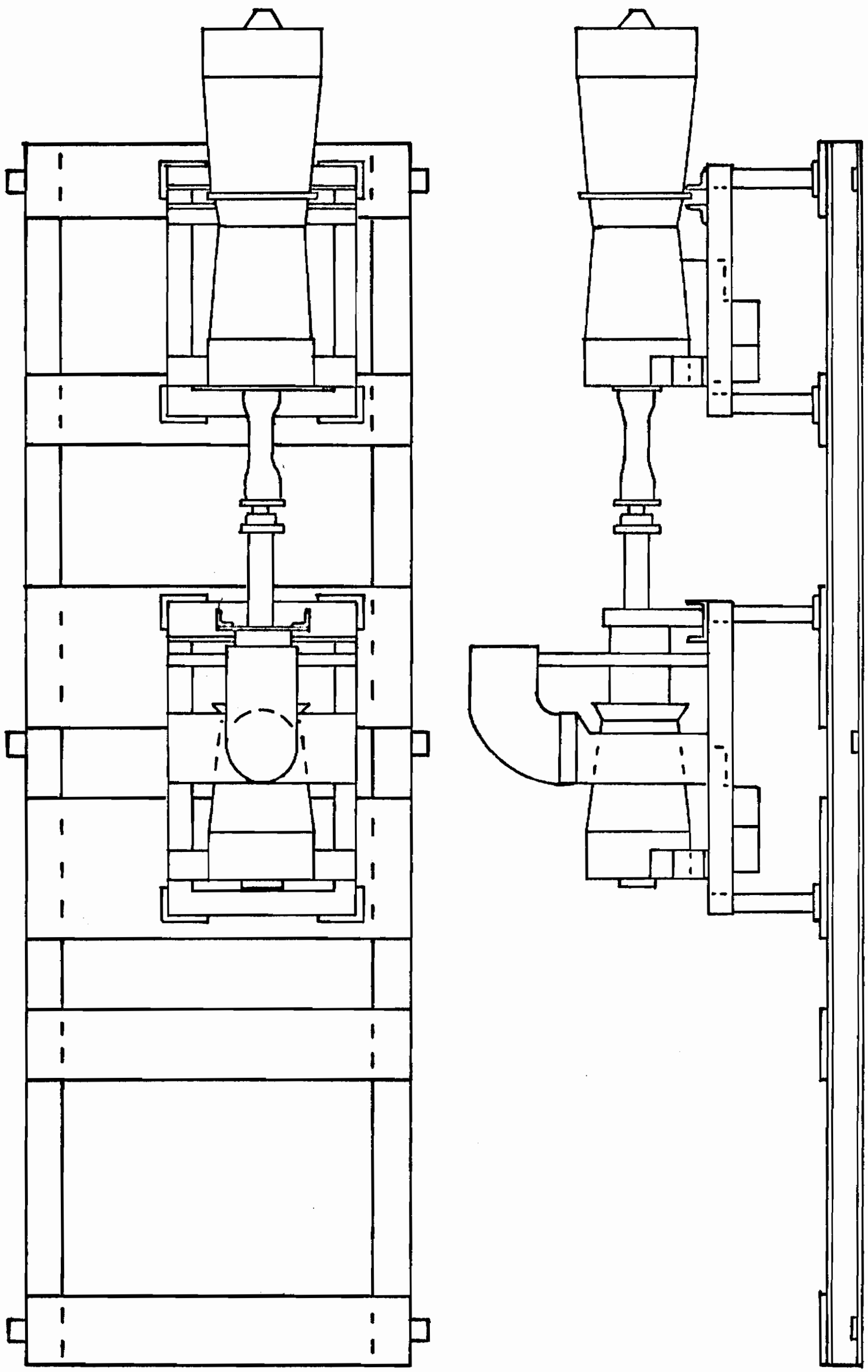

Figure 26. DIRECT COUPLED SYSTEM

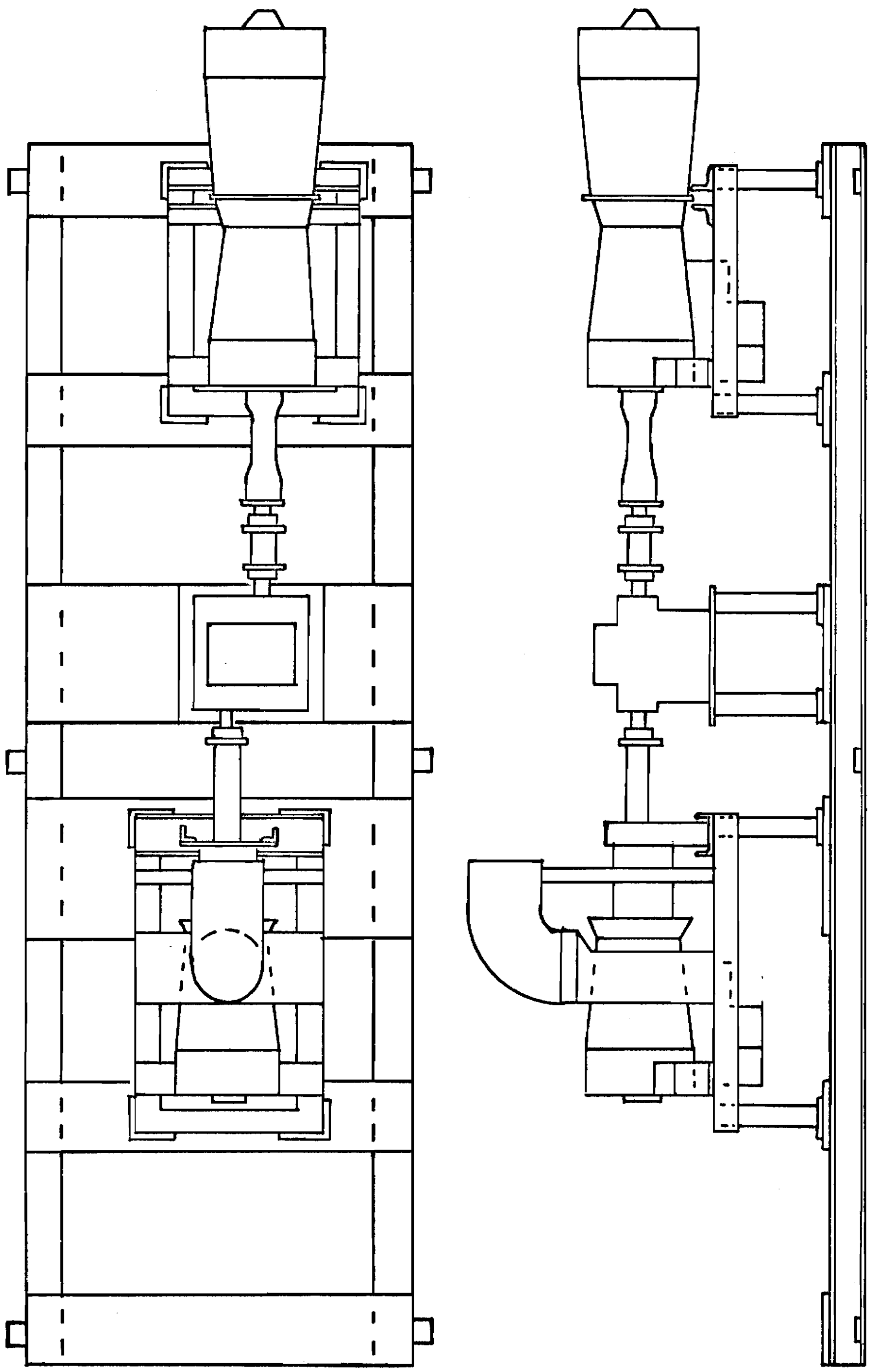

Figure 27. SINGLE GEARBOX SYSTEM

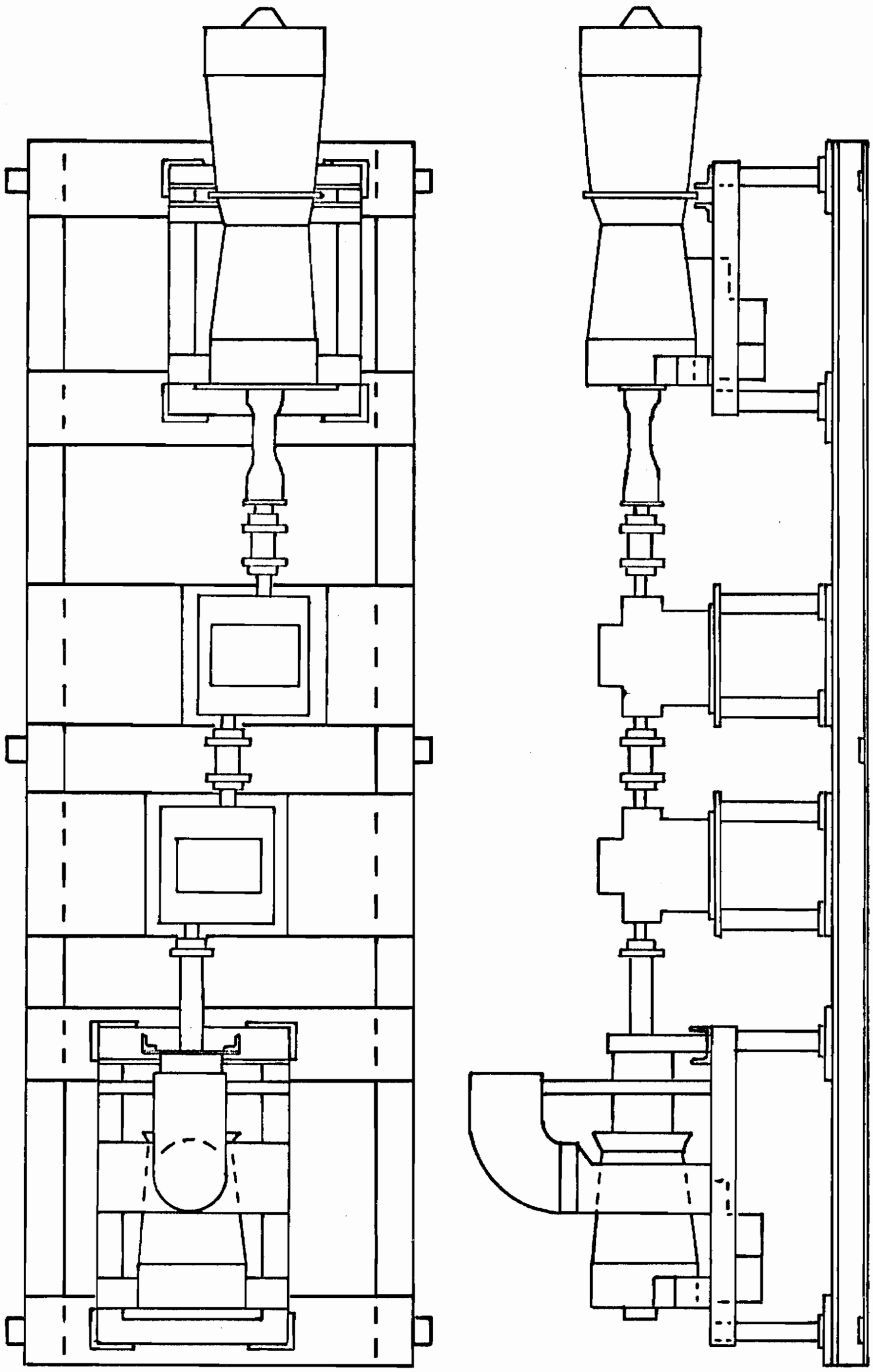

Figure 28. TWO GEARBOX SYSTEM

X. RESULTS

A high-speed turbomachinery test cell has been successfully designed, and is under construction at this writing. The facility provides separate storage, assembly, control, and test cell areas and, although originally intended for investigation of stall and surge phenomena, provides ample room for expansion into related research areas.

Test rotors have been designed for drive by a General Electric T64-6B turboshaft engine which, when used with one or two 1:1.2 ratio gearboxes, is capable of delivering 1650 HP at speeds from 12000-20400 rpm in either rotational direction with some overrange available. Speeds up to 17000 rpm at 3000 HP are possible with the rotor directly coupled to the engine.

The selected test rotor originally had 14 stages of axial-flow blading and was designed to operate at 17070 rpm. Individual stages of blading can be removed as desired. Available power is sufficient to drive up to 5 stages of this rotor at design speeds.

Instrumentation provided allows monitoring of all parameters necessary for control, operation of the system within specified limits, and evaluation of engine and compressor performance.

The mounting system has been designed to perform the required functions of accommodating thermal expansion, locating and aligning components, eliminating undesirable vibratory responses, and providing easily altered configurations. In addition, the total system provides a desirable high factor of safety and complete freedom from any torsional resonances within the specified operating range.

XI. DISCUSSION OF RESULTS

The engine selected was well suited to the intended application. It provided ample speed and power for driving the initial test rotor and other rotors -- larger or smaller -- which may become available in the future, but is not oversized to the extent that operating economy suffers. The major limitation was caused by the fuel control, which prohibits operation below 12000 rpm. Modifications to the fuel control or selection of different gearboxes could provide these lower speeds if desired.

The gearboxes obtained were not the ones originally selected and did not provide as wide a speed range as had been anticipated. However, they did meet the established facility requirements without causing the lengthy delay necessary to obtain the original choices. As mentioned above, different ratios may prove more desirable for other areas of investigation.

The couplings chosen were a flexible disk type and were selected for their simplicity of design, convenience for use, and availability. Research activities involve short, intermittent operation of the system. For applications where lengthy or continuous duty is desirable, gear type couplings may prove more acceptable.

The mount system provided a simple, easily constructed means of obtaining the desired results. Thermal expansion within the system was accommodated by mounting all components from the underside only, which allowed freedom in a radial direction. The couplings allowed expansion in the axial direction. The structure is sufficiently rigid to maintain alignment under all foreseeable conditions and can support larger rotors than involved herein. Location and movement of components for various configurations was provided by the individual support and anchor systems used. The total dynamic system should be acceptably free from vibrations but could require further investigation.

As discussed, the selection of components was based on providing a vibration-free torsional system. This was accomplished to the extent that no torsional resonances occur within the operating range of the system. Although avoiding resonances does not insure a troublefree system, it definitely enhances that possibility.

Any axial vibrations within the system will be accommodated by the couplings, which will allow relative motion between components in the axial direction.

Bending vibrations are generally associated with unbalance within the system. In this case, each component has been designed and balanced for operation at the speeds involved and should create little, if any, problem in this area. Additionally, sufficient bearing locations and a rigid support structure will aid in constraining the system from these motions. However, the possibility does exist that coupling effects may allow non-critical vibrations in one component to excite critical vibrations in an adjacent component. Unfortunately, as noted by Badgley and Hartman [12], the analytical capability for performing the complete solution to the coupled vibration problem does not exist. For this reason, and as a general safeguard, vibration transducers were mounted throughout the system for monitoring of dynamic levels to insure operation within acceptable limits. Thus, conformance with all system requirements has been achieved.

XII. RECOMMENDATIONS

The facility described includes many desirable features which provide for versatility and safety of operation. However, certain improvements could become desirable depending upon operating requirements. Among these are:

1) Filtration of inlet air to the drive unit to reduce the possibility of unwanted accumulation of deposits which may cause maintenance problems.
2) Treatment of the inlet and exhaust to reduce the noise levels associated with test runs.
3) Investigation of the engine control system to determine possible means of achieving lower operational speeds for future research applications.
4) Investigation of the fatigue life of the compressor blades to determine if the surge conditions involved in the test work may produce premature failures.
5) Design of a brake system for use in emergency situations and as an aid in decelerating through lower system critical speeds.
6) The installation of an automatic alarm system which monitors critical system parameters.
7) A tie-in with the campus computer system to aid in data reduction and analysis.

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Appendix A

Manufacturer Specifications for Components

Item 1 - Engine Specifications

Item 2 - Coupling Specifications

Item 1 - Engine Specifications

Intermediate Maintenance Manual NAVAIR 02B-105AJA-6-1

Model	T64-GE-6B
Type of Engine	Turboshaft
Type of Compressor	Axial-Flow
Output Power	3080 shaft horsepower
Number of Compressor Rotor Stages	14
Variable Geometry	Inlet guide vanes, stator vane stages 1 through 4
Type of Combustion Chamber	Single, annular, through-flow
Gas Generator Turbine Stages	2
Power Turbine Stages	2
Direction of Engine Rotation	Counterclockwise (aft looking forward) (both turbines)
Basic Engine Length (includes torque sensor shaft and housing)	83.36 in.
Basic Gas Generator Diameter (outside diameter across actuators)	21 in. (max)
Engine Weight (dry)	717 lbs (max)
Center of Gravity	1.38 in. below engine horizontal centerline; 27.35 in. aft of front frame forward flange; 0.52 in. left of vertical centerline (aft looking forward)
Engine Mounting	4 main pads, on the front frame at the 2, 4, 8, 10 o'clock positions 2 mounts on the combustion chamber flange (adjacent to forward flange) at the 12 and 6 o'clock positions.
Fuel	MIL-T-5624G, Grade JP-4 or Grade JP-5
Lubricating Oil	MIL-L-23699

Power Turbine Output Shaft Ratings at Standard Sea Level Static Conditions--T64-GE-6B Engine

Rating	Minimum SHP	Max Gas Generator Turbine RPM*	Output Shaft RPM**	Max SFC (lb/shp/hr)	Measured Power Turbine Inlet Temp
Maximum (5 min)	3080	17,800	13,600	0.485	1180°F
Military (30 min)	2910	17,680	13,600	0.490	1160°F
Normal	2450	17,500	13,600	0.511	1110°F
90% Normal	2205	-	13,600	0.525	-
75% Normal	1840	-	13,600	0.556	-

100 percent gas generator turbine speed equals 17,070 rpm.
*Max allowable continuous gas generator turbine speed is 17,800 rpm (104.3 percent).
Max allowable transient gas generator turbine overspeed limit shall be 18,250 rpm (107 percent) for a period not in excess of 10 sec.
**Max allowable continuous output speed is 17,000 rpm.

Item 2 - Coupling Specifications

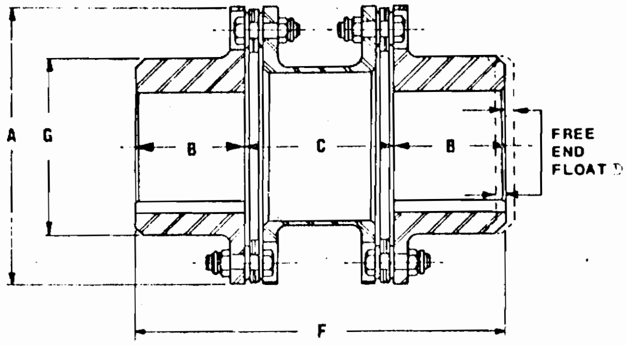
Rexnord Double-Flexing Disc Coupling

Thomas flexible couplings engineering catalog 861

Series 52 couplings are high-speed, high-torque-capacity precision couplings designed for drives requiring a minimum shaft-supported weight. Superior performance is maintained for life by utilizing modern machining and balancing techniques and combining them with the features of: No Lubrication--No Backlash--No Wearing Parts.

These couplings are all steel and have a center-member design that is variable to suit application requirements of shaft spacing. All parts are solidly bolted, permitting high-tensile steel discs to do the work of accepting misalignment while transmitting 100% of the torque.

SERIES 52



Dimension G varies with bore specified.

Size No.	262
Max. Bore	2 5/8 in.
Max. RPM	20,400
Max. Horsepower per 100 RPM	14.9
Dim. A	6 5/8 in.
Dim. B	2 5/8 in.
Dim. C (std.)	5 in.
Dim. F	10 1/4 in.
Dim. G	4 7/32 in.
Weight	26 lb
WR^2	130 lb-in.^{2}
Increase in WR^2 per inch of Dim. C	3.1 lb-in.^{2}

Appendix B

Computer Program for Determining Compressor Horsepower Requirements

The computer program used to produce Fig. 7 in the text was developed as follows. Values for inlet temperature and pressure are assumed along with a value of compressor efficiency. Based on the design stage pressure ratio (1.2) the off-design pressure ratio is calculated as follows:

$$\text{Pressure Ratio} = \left[1.0 + 0.2 * \left(\frac{\text{RPM}}{\text{Design Speed}}\right)^2\right]^N$$

where N is the number of stages in use. This pressure ratio is then used to determine the pressure rise through the compressor.

The mass flow rate is directly related to the ratio of operating to design speeds as follows:

$$\frac{\dot{M}\ \text{operating}}{\dot{M}\ \text{design}} = \frac{\text{rpm operating}}{\text{rpm design}}$$

With these parameters determined, the power required is calculated from the equation given in the text:

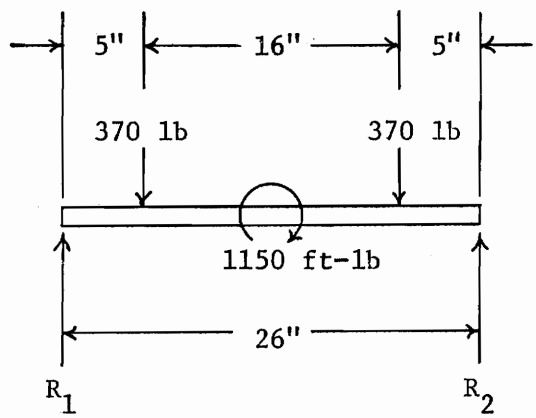
$$\text{Power} = \frac{\dot{m} c_p T_{02}}{\eta_c}\left[\left(\frac{P_{03}}{P_{02}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

```
C   PROGRAM TO CALCULATE HORSEPOWER REQUIRED TO DRIVE COMPRESSOR STAGES
C   AT VARIOUS SPEEDS GIVEN N, P1, RPM, T1, EFF
      REAL M
      N=1
      P1=14.696
      RPM=1000.00
      T1=520.0
      EFF=0.80
    1 WRITE(6,5)N,EFF
    5 FORMAT(//5X,'NUMBER OF STAGES =',I3  ,5X,'EFFICIENCY =',F5.3)
      WRITE(6,8)
    8 FORMAT(/12X,'RPM',12X,'H2',12X,'T2',13X,'M',13X,'HP')
    2 RP=(1.00+(0.20*(RPM/17070.0)**2))**N
      P2=RP*P1
      Y=RP**0.286
      T2I=Y*T1
      WCI=0.24*(T2I-T1)
      WC=WCI/EFF
      H2=0.24*T1+WC
      T2=H2/0.2391
      DT=T2-T1
      M=24.5*RPM/17070.0
      HP=0.2391*DT*M*778.0/550.0
      WRITE(6,40)RPM,H2,T2,M,HP
   40 FORMAT(5F15.4)
      RPM=RPM+1000.00
      IF(RPM-17500.0)2,2,10
   10 N=N+1
      RPM=1000.00
      IF(N-14)1,1,100
  100 STOP
      END
```

Appendix C

Mounting System Design Calculations

Loads produced at the engine front frame:



$$\Sigma M_1 = (5)(370) + 1150(12) + (21)(370) - 26R_2 = 0$$

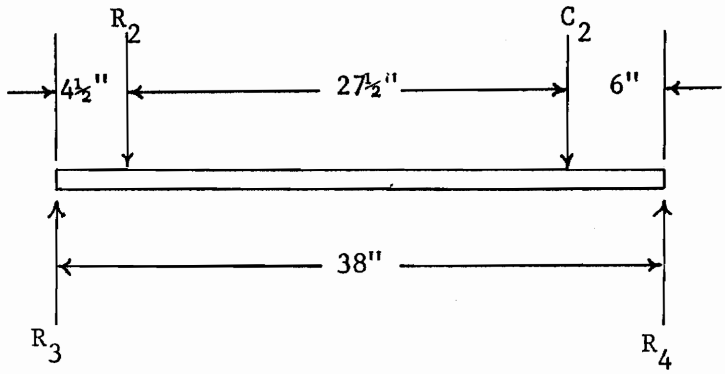
$$26R_2 = 23420$$

$$R_2 = 900.7 \text{ lb}$$

$$R_1 = -160.7 \text{ lb}$$

Loads on support mount side member using a 4 x 4 x 3/8" angle for which

$$I = 4.4 \text{ in}^4 \quad \text{and} \quad S = 1.5 \text{ in}^3$$



Assuming load C_2 to be 740/4 = 185 lb,

$$\Sigma M_3 = 38R_4 - 32C_2 - 4.5R_2 = 0$$

$$38R_4 = 9970$$

$$R_4 = 262.5$$

$$R_3 = 823.3$$

$$\sigma = \frac{Mc}{I} = \frac{M}{S} = \frac{R_3 x}{S} = \frac{(823.3)(4.5)}{1.5} = 2469 \text{ psi}$$

$$\tau = \frac{V}{A} = \frac{R_3}{d\omega} = \frac{823.3}{4(3/8)} = 548 \text{ psi}$$

Tensile strength of steel $S_{UT} = 60{,}000$ psi

Shearing yield strength $S_{SY} = 0.577\ (S_{UT})$

$= 34620$

$$\text{Factor of safety} = \frac{34620}{2469} - 1 = 13.02$$

$$y = \frac{Fba}{6EI\ell}\,(2\ell b - 2b^2) \quad \text{at } R_2 \text{ due to } R_2$$

$$= \frac{900(4.5)(33.5)}{6(29)(10^6)(4.4)(38)} \left[2(38)(33.5) - 2(33.5)^2\right]$$

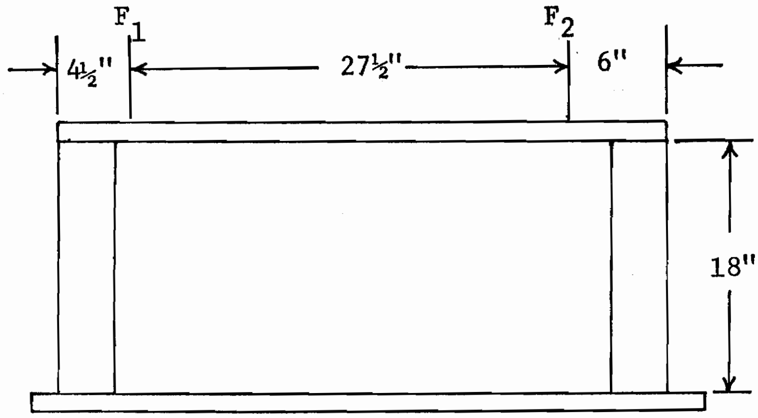
$$= 1.407 \times 10^{-3} \text{ in. due to } R_2$$

$$y = \frac{Fbx}{6EI\ell} \quad 2\ell(\ell-x) - b^2 - (\ell-x)^2 \qquad \text{at } R_2 \text{ due to } C_2$$

$$= \frac{185(6)(32)}{6(29)(10^6)(4.4)(38)} \left[2(38)(6) - 2(6)^2\right]$$

$$= 0.468 \times 10^{-3} \text{ in.}$$

$$y_{total} = 1.875 \times 10^{-3} \text{ in.}$$

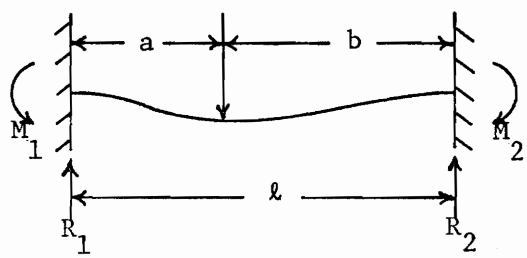


Reaction to load at columns

$F_1 = 900.77$ lb

$F_2 = 185$ lb

Assuming fixed supports with intermediate load, we find the reaction at the columns to be:



$$R_1 = \frac{Pb^2}{\ell^3}(3a + b)$$

$$R_2 = \frac{Pa^2}{\ell^3}(3b + a)$$

$$M_1 = Pab^2/\ell^2$$

$$M_2 = Pa^2b/\ell^2$$

<u>Due to P_1</u>

$$R_1 = \frac{900(33.5)^2}{(38)^2}[3(4.5) + 33.5] = 865.87 \text{ lb.}$$

$$R_2 = \frac{900(4.5)^2}{(38)^3}[3(33.5) + 4.5] = 34.87 \text{ lb.}$$

$$M_1 = \frac{900(4.5)(33.5)^2}{(38)^2} = 3147.58 \text{ in-lb}$$

$$M_2 = 900\ (4.5)^2(33.5)/(38)^2 = 422.81 \text{ in-lb}$$

<u>Due to P_2</u>

$$R_1 = \frac{185(6)^2}{(38)^3}[3(32) + 6] = 12.38 \text{ lb.}$$

$$R_2 = \frac{185(32)^2}{(38)^3}[3(6) + 32] = 172.62 \text{ lb.}$$

$$M_1 = \frac{185(32)(6)^2}{(38)^2} = 147.59 \text{ in-lb}$$

$$M_2 = \frac{185(32)^2(6)}{(38)^2} = 787.15 \text{ in-lb}$$

Summing the reactions:

$R_1 = 878.25$ lb

$R_2 = 207.49$ lb

$M_1 = 3295.17$ in-lb

$M_2 = 1209.96$ in-lb

Eccentricity at column 1,

$$e = M/R = 3295.17/878.25 = 3.75 \text{ in.}$$

at column 2, $\quad e = M/R = 1209.96/207.49 = 5.83$ in.

Using the secant formula we have:

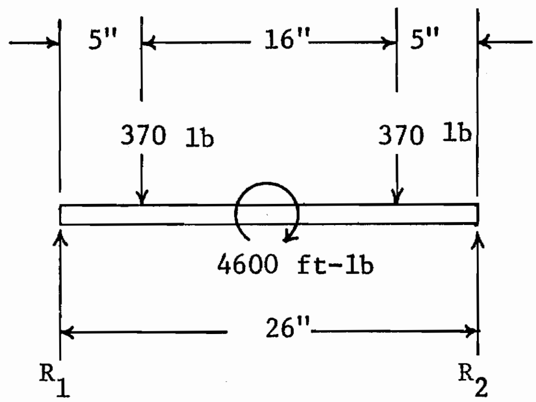
$$|\sigma|_{max} = \frac{P}{A}\left(1 + \frac{ec}{r^2}\sec\frac{L}{r}\sqrt{\frac{P}{4EA}}\right)$$

$$|\sigma|_{max} = \frac{878.25}{2.228}\left(1 + \frac{3.75(1.75)}{(1.16)^2}\sec\frac{18}{1.16}\sqrt{\frac{878.25}{4(2.228)(29)(10^6)}}\right)$$

$$|\sigma|_{max} = 2317.3 \text{ psi}$$

Each column support rests on a 5" I-beam which has a moment of inertia of 12.1 in^4. This large value of inertia results in low levels of stress for the above loads and calculations were not necessary.

Loads due to sudden bearing seizure:



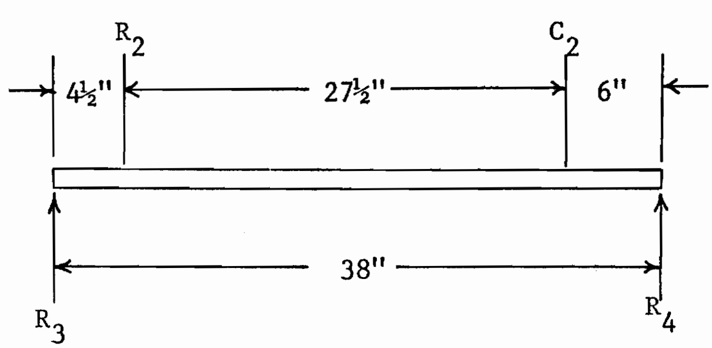
$$\Sigma M_1 = (5)(370) + 4600(12) + 21(370) - 26R_2 = 0$$

$$26R_2 = 64820$$

$$R_2 = 2493.08$$

$$R_1 = - 1753.08$$

Loads on support mount side member using a 4 x 4 x 3/8" angle



$$C_2 = \frac{740}{4} = 185 \text{ lb.}$$

$$\Sigma M_3 = 38R_4 - 32C_2 - 4.5\ R_2 = 0$$

$$38R_4 = 17138.86$$

$$R_4 = 451.02$$

$$R_3 = 2227.06$$

$$\sigma = \frac{Mc}{I} = \frac{M}{S} = \frac{R_3 x}{S} = \frac{(2227)(4.5)}{1.5} = 6681.18 \text{ psi}$$

$$\tau = \frac{V}{A} = \frac{R_3}{d\omega} = \frac{2227}{4(3/8)} = 1484 \text{ psi}$$

$$\text{factor of safety} = \frac{34620}{6681} - 1 = 4.18$$

This stress was shown to be the largest obtained in the previous calculations and thus continuation is not necessary.

XV. VITA

The author, James Emory Thomas, was born August 12, 1950, in Harrisonburg, Virginia.

He entered Virginia Polytechnic Institute in the Fall of 1968 and began his undergraduate education in Mechanical Engineering. At V.P.I. he was a member of Pi Tau Sigma and Tau Beta Pi honor societies. He also participated in the Cooperative Education program with American Safety Razor Co. and Southern Railway Co. In June, 1973, he graduated and was awarded the Bachelor of Science Degree with distinction in Mechanical Engineering.

In September, 1973, he received a research assistantship sponsored by the Office of Naval Research and began his graduate studies in Mechanical Engineering at V.P.I. & S.U.

James E. Thomas

DESIGN OF A HIGH-SPEED TURBOMACHINERY

TEST CELL AND COMPONENTS

by

James Emory Thomas

(ABSTRACT)

The design and construction of a high-speed turbomachinery research facility is reported. This facility provides a drive system utilizing a gas turbine engine, two speed-increasing gearboxes, and interconnecting couplings to deliver 1650 HP to test rotors at speeds from 12000 to 20400 rpm in either rotational direction. A multistage axial-flow compressor was assumed as the load for initial design calculations. A vibration analysis was conducted to insure that no torsional resonances occurred within the operating range of the system.

The mounting system was designed to provide a readily changeable means for interconnecting components so that numerous research activities were possible.

Instrumentation, monitoring and control equipment were provided for continuous regulation of all necessary system and test parameters.