formation. Bending effects such as distance from the neutral axis are included.

Investigation of S-N curves of typical metals shows that a relatively modest reduction in vibratory stress gives a large, often indefinitely large, increase in cycles to failure. This is the basic reason why the dry Coulomb damping provided by several sleeves and rings has proved so successful.

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Philosophy, Design, and Evaluation of Soft-Mounted Engine Rotor Systems

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Gas turbine engine rotors are conventionally supported by bearings mounted on relatively stiff supports. The resulting vibratory loads and deflections can be reduced significantly by judiciously soft-mounting the bearings through squirrel cages and/or squeeze films. In addition to minimizing loads and stresses in an engine, it is important that clearances during conditions of maneuvers, thermal bow, and rotor whirl due to unbalance (even under extraordinary conditions such as loss of blades) be controlled. For high-speed rotors, it becomes necessary to support the rotors on resiliently mounted bearings to achieve vibration-free, long-life, close-clearance engines. In this paper, the design philosophy, criteria, and methods of evaluation for soft-mounted turbine engine rotor systems used in General Electric aircraft engine design are described. A major constituent of this method is a computer program for system vibration and static analysis [VAST]. This program is capable of finding natural frequencies, normalized modes, and responses due to any distribution of exciting forces considering gyroscopic and shear-deflection effects. Aircraft mounting and excitations from the helicopter rotor are also included in the computer analysis. General Electric's T700 turboshaft engine, under development for the U.S. Army, serves to illustrate the squeeze film, soft-mounting concept of design. Results from tests of the T700 engine, Advanced Technology Axial Centrifugal Compressor (ATACC), T64 turboshaft, TF34 turbofan, and other engines are summarized verifying the advantages of soft-mounted rotor systems.

Introduction

THE ultimate objective of an aircraft gas turbine engineer is to design an ultra-lightweight, low cost rotor system free from excessive vibrations throughout the operating range, so as to achieve: 1) infinite life for supporting bearings and structural components; 2) a high performance engine with very little clearance-loss taken at critical locations of the engine due to engine vibrations or due to maneuver loads imposed on the engine; 3) a design that has "built-in" vibration fixes to solve vibration problems if encountered during the development of the engine; 4) a simple design with good growth potential and excellent reliability and maintainability features; 5) an engine tolerant to abusive situations where very high unbalances are introduced for a short time.

Since the objectives are contradictory, a difficult task of balancing various requirements lies in the hands of the engineer and hence, different approaches are used to design an optimum rotor compromised to the situation under consideration. Obviously, no single way of designing a rotor for all requirements and constraints can be formulated. However, an attempt is made by the author to highlight the merits of partially or fully soft-supporting the rotors which goes a long way towards achieving the forementioned objectives.[2-5]

The T700 is a unique engine whose gas generator rotor is solely mounted on two resilient squeeze film damped bearings and its power shaft damped by two squeeze film bearings. These design features were selected as a result of extensive analysis and rig and full-scale engine tests. The T700 has excellent vibration characteristics with less than 5 g's acceleration on casings under normal operating conditions. In this engine the designer's ultimate dynamics objective has virtually been achieved. The design concept, details, analyses, and development effort that went into the soft-mounted rotor system are enumerated in detail in this paper. Experience from other aircraft engines at General Electric is also cited verifying the benefits of soft mounting.

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Index category: Structural Dynamic Analysis.

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Hard System

The conventional engine dynamics approach is to design a stiff rotor mounted on stiff supports. If a rotor could be designed with stiff supports, such that all rotor criticals are outside of the operating range with good margin, and with acceptable weight, the design would be excellent. However, the forementioned design is still not necessarily the best arrangement from the point of view of high unbalance situations at top speed where the magnification is greater than one (soft-mounted systems can be designed to have a magnification close to zero). Generally, gas turbine engines on stiff supports have criticals below or around idle, and some just above top speed. These criticals could be troublesome in either going through them or due to the proximity of the criticals to the operating range. Generally, stiff-mounted engines do not run smoothly as top speed is approached, due to the proximity of the critical that is above the operating range.

Soft and Hybrid Systems

The concept of soft mounting is to design a stiff rotor supported on soft bearing supports. By proper arrangement of spring rates at the bearings, the criticals can be placed in the desired speed ranges. Generally, springs are used to center and tune the rotor to the desired criticals. Damping devices are usually added in parallel so as to suppress the rigid body criticals which have to be traversed below idle. They also are designed to act as snubbers during maneuvers or other abusive situations to limit the rotor excursion. Often soft mounting is accomplished by utilizing squeeze films without centering springs. In these situations, care must be exercised to ensure that the clearance is acceptable up to the speed at which the oil film is sufficiently strong to provide a centering action.

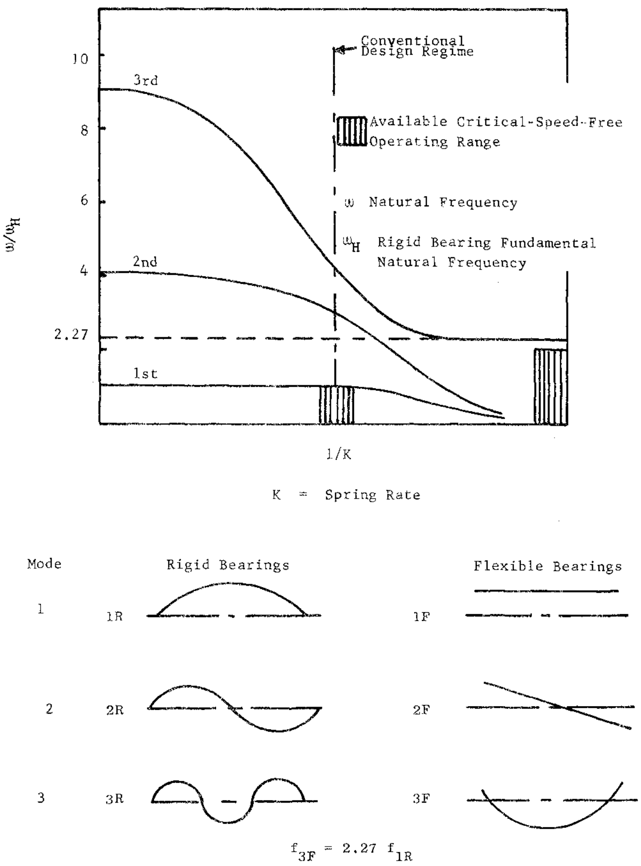


Fig. 1 Natural frequency of a uniform shaft as a function of bearing support stiffness.

Figure 1 shows the critical speeds of a uniform shaft as a function of the bearing support stiffness. It is apparent that soft mounting permits the rotor to run at a much higher speed without encountering a rotor bending critical. By proper design, the bending critical of a nonuniform rotor on soft springs can be designed to be higher than 2.27 times the first bending critical of the shaft on rigid bearings. Soft bearing supports are achieved in practice by a variety of means: 1) a squirrel cage support (Fig. 2a); 2) a squirrel cage and/or squeeze film (Figs. 2b–d); and, 3) an outer race mounted on a flexible ring at discrete points circumferentially (which, in turn, is mounted on the bearing support between the outer race mount locations). High-speed aircraft engine rotors are generally designed on soft supports so as to place the rigid body criticals below idle and the bending critical above the maximum speed with sufficient margin. Since for high-speed rotors the criticals on rigid supports cannot be placed outside the operating range, soft mounting is a must. The first two modes involve no rotor bending (Fig. 1). Hence, it is apparent that the first two translation criticals can be traversed easily, even with a rotor that is balanced as a rigid body on a conventional balancing machine.

It is common to support the aircraft engine rotor on squirrel cages with the desired stiffnesses. These stiffnesses can be calculated easily and the criticals predicted accurately for a soft-mounted rotor with high confidence. Modern aircraft engines at General Electric generally employ a hybrid mounting (soft supports on some bearings and stiff supports on others) on low-speed engines and soft mounting on high-speed engines. Damping devices such as squeeze films or friction dampers can be added easily for a soft support so as to make the engine tolerant to abusive situations. Squeeze films may also be added to a stiff-mounted rotor for solving vibration problems. This converts the stiff system to a hybrid system since the damper is added in series with a stiff support, thereby introducing a soft spring.

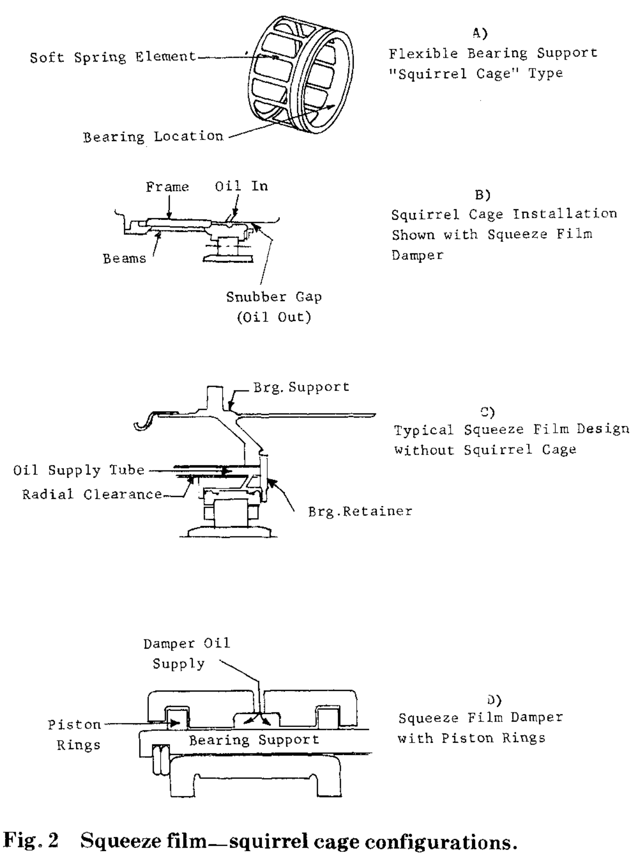


Fig. 2 Squeeze film—squirrel cage configurations.

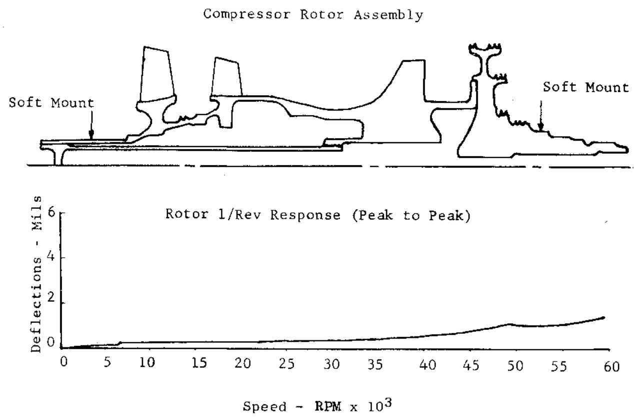
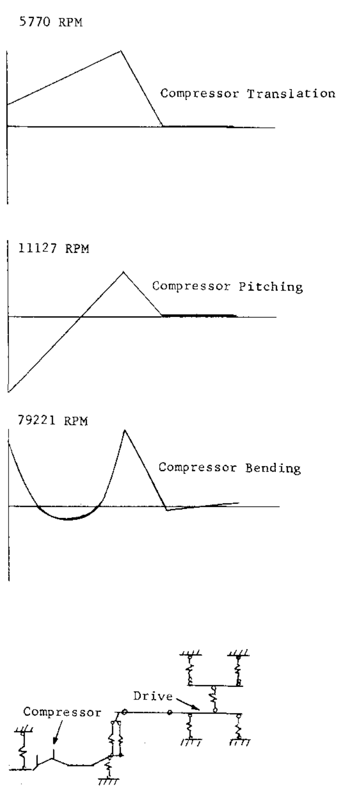


Fig. 3 ATACC vehicle schematic and test data.

Experience at General Electric

Test results from a few aircraft engines whose rotors are soft-mounted are discussed next. The ATACC compressor vehicle, which was a fully soft-mounted high-speed rotor, ran to a maximum design speed of 60,000 rpm with virtually no vibrations under normal operating conditions (Fig. 3). The two rotor bearings were each supported by means of squirrel cages having 10 *k* lbs/in. spring rate. Squeeze films were added in parallel with the squirrel cages as an added assurance to damp the rigid body modes located below idle. The mathematical model used in the analysis (together with the rotor criticals, mode shapes and energy distributions) is given in Fig. 4. As noted, the bending critical of the rotor was placed outside of the operating range with good margin. The ATACC rig proved the extraordinary benefits of soft mounting a high-speed rotor.

The T700 engine, under development for the Army, is another example of unqualified success in the area of soft-mounted rotor systems. Design details, test data and the evolution of the T700 engine are covered in a later part of the paper. Other General Electric engines which successfully use soft mounting on some of their bearings include the CF6, TF39, TF34, T64, J101, and GE4. Figure 5 shows back-to-back tests with and without squeeze films on the hybrid mounted T64, TF34, and development engines.



ENERGY DISTRIBUTION

Component	%KE	%PE
Impeller	34	-
Balance Piston	32	-
Aft Comp Shaft	8	-
Cone & Fwd Cplg.	5	-
Inducer	4	-
Shaft	2	-
Stg. 1 & 2 Comp.	7	-
Aft Support	-	87
Front Support	-	13
Stg. 1 Comp.	46	-
Fwd. Overhang	15	-
Stg. 2 Comp.	12	-
Cone & Fwd. Cplg.	5	-
Aft Comp. Shaft	5	-
Balance Piston	5	-
Inducer	4	-
Front Support	-	86
Aft Support	-	13
Coupling	20	1
Cone & Fwd. Cplg.	17	5
Fwd. Overhang	17	1
Impeller	19	11
Stg. 1 Comp.	12	3
Inducer	2	13
Balance Piston	-	20
Stg. 2 Comp.	-	10
Lower Spline	6	6
Shaft	-	5

VAST Model

Fig. 4 ATACC vehicle VAST schematic, mode shapes and energy summary.

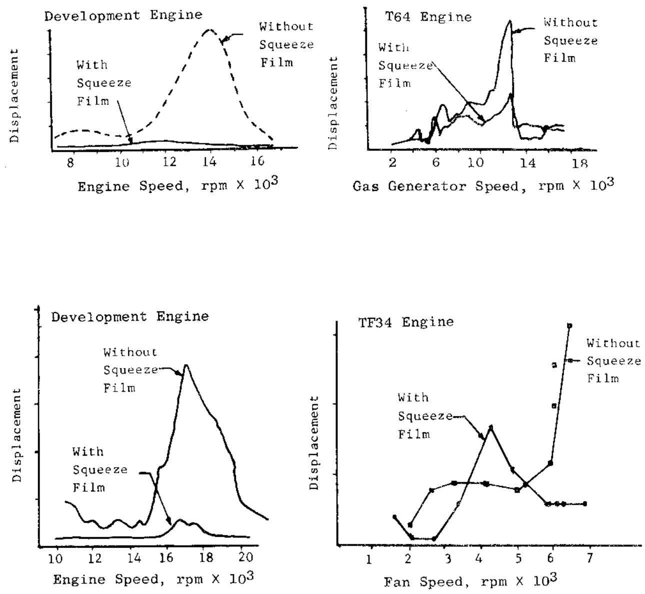


Fig. 5 Response of engines with and without squeeze films.

Design of Squeeze Film Damper/Soft Mount

Vibration problems can be alleviated by means of damping devices. The amplitude at resonance is always bounded for a squeeze film (viscous) damper, whereas, it is unbounded for a dry friction damper beyond a certain level of excitation. The principle involved in the squeeze film dampers is to introduce oil between two nonrotating parts, allowing the oil to squeeze out circumferentially and/or axially during orbiting of the rotor. Soft mounting can be achieved additionally by means of squeeze films since they provide both damping and flexibility. The squeeze film must be designed with care to provide optimum damping so as to benefit the design at the critical without causing harm above the critical.

Computer programs that provide the dynamic response of a system with squeeze film supports have been developed at General Electric similar to the one discussed in Ref. 3. Hydrodynamic forces developed on the support are given by the Reynolds equation. By integrating the equation with proper boundary conditions, the dynamic pressure distribution is obtained [from which forces in phase with the displacement T_K (spring force) and forces 90° away from the displacement T_C (damping force) are evaluated]. Hydrodynamic forces developed in a squeeze film with a rotating load are twice the forces developed in a conventional journal-bearing with a unidirectional load. Nondimensional forces, as a function of eccentricity ratio for a short axial flow squeeze film bearing, are given in Fig. 6.

For a particular application, the preliminary design of a squeeze film damper is made based on analysis and experimental data. The final design satisfying the constraints is developed by extensive rig and full-scale engine tests. The following parameters have to be optimized so as to have a desired response for typical unbalance, occasional high unbalance, and abusive unbalance situations such as loss of blades: axial flow vs circumferential flow, squirrel cage spring rate, land length, radial clearance, groove size, oil

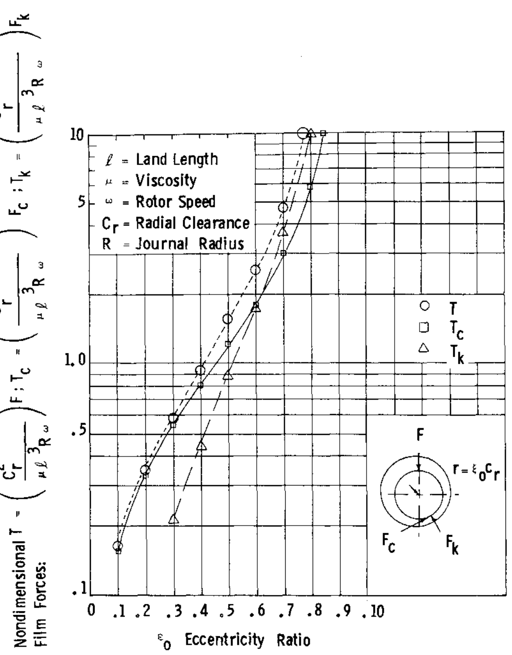


Fig. 6 Nondimensional oil film forces vs eccentricity ratio.

inlet and outlet geometry, and oil flow and pressure. A squeeze film should not be used to damp a mode where the relative motion across the film is small. Every squeeze film design is limited to an upper bound of unbalance beyond which it could be worse than a hard support (Fig. 7a). Response to unbalance can be further reduced by using piston rings[2] which have to be designed so that the system is not overdamped. Circumferential squeeze films where piston rings are used at the ends are generally more sensitive to oil inlet pressure than axial flow films. Below a certain oil inlet pressure level, which is a function of system unbalance, squeeze films become ineffective. The threshold pressure below which a squeeze film is ineffective is lower for axial flow films than for circumferential flow films.

Generally, squeeze films have the following characteristics: a) Radial gap—0.5 to 3.0 mils per inch of film diameter, b) Land length—length to diameter 0.5 to 0.05, c) Groove—central or end grooves, d) End restriction—with or without piston rings, e) Flow—very small to very high flows, f) Oil feed—one or two holes, g) Pressure—ambient to high pressure, h) Antirotation provision—yes and no.

If a squeeze film is used in parallel with a squirrel cage, centrality of the clearance must be maintained, especially when stiff springs are used. For squeeze films with piston rings, the clearance at the ends beyond the squeeze film may be smaller than the oil film gap. If "O" rings are used to seal the oil, provisions should be made such that the load path will not be short circuited at these seals.

For well-balanced engines good agreement of vehicle response is obtained by linear analysis. For high unbalances, however, the response should be computed with a nonlinear theory. Such high unbalance situations might be encountered with abnormal or abusive operating conditions or possibly on a restart where thermal gradients may cause significant, transient rotor bow (Fig. 7b).

Advantages and Disadvantages of Hard, Hybrid, and Soft Mountings

Conventional hard-mounted rotor designs carry with them a weight penalty. Special care must be taken for these systems to analyze and predict the response characteristics due to the necessity for accurately calculating the stiffness of the rotor and the complex supporting structure. It is very difficult for such systems to dynamically tune the design and add damping devices. On the other hand, a hard system is less prone to instability problems.[6] For low-speed rotors under some design constraints, a conventional hard system may represent the best design.

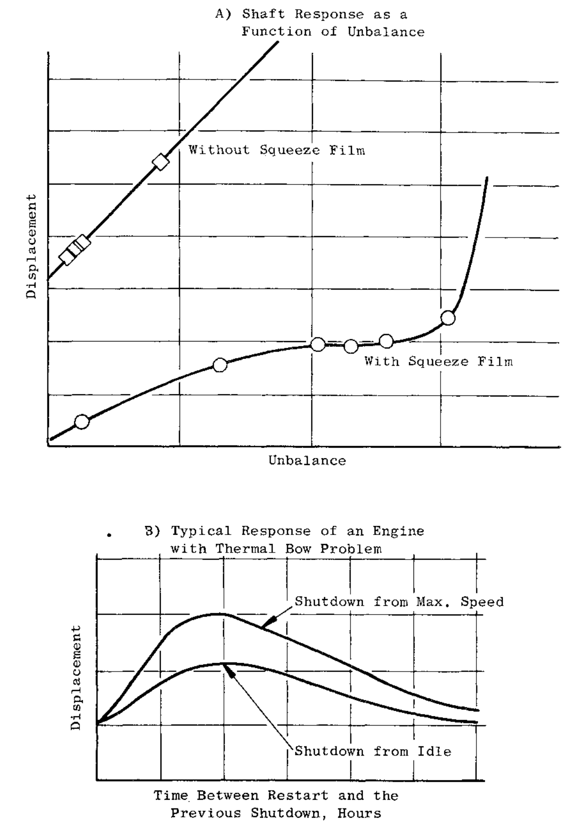


Fig. 7 Test data from development engines.

Hybrid mounting is used in most of the low-speed modern engines. By this type of mounting, tolerance to unbalance is generally enhanced. If required, damping devices can be added easily, and tuning the rotor to obtain the required criticals is facilitated. These systems are less prone to instability problems than the soft-mounted systems. For some design constraints a hybrid system could be the best, resulting in a moderate-weight design.

High-speed rotors necessitate soft-mounting. Response to unbalance at top speed is very small since the rotor runs supercritical. Problem fixes are accomplished very easily by proper tuning and by adding damping. However, soft-mounted systems without dampers are more prone to instability problems. In addition, an engine with a poorly designed soft mounting could be worse than one with a stiff mounting. In all cases a soft-mounted rotor would result in the lightest engine. Again, depending upon the specific requirements, a soft-mounted system might be best for some low-speed engine designs as well as for high-speed designs.

Method of Analysis

The analytic evaluation of aircraft engine vibration characteristics is accomplished primarily by use of a comprehensive system vibration and static analysis computer program [VAST[1]] supplemented by a three-dimensional structural analysis program [MASS[7]]. In addition, extensive component and full-scale engine tests thoroughly evaluate the designs. The analytic procedure utilizing VAST has been applied to all the current engines with excellent success in correlating with tests. VAST is an inplane vibration analysis program considering the following: mass and rotatory inertia in rotors, frames, casings, and mounts; flexibility distributions in all components due to flexure and to transverse shear; allowance for the insertion of influence coefficients to describe the behavior of nonbeam-type flexibility; gyroscopic effects; bearing clearances, offsets and misalignments; unbalance distribution; and, different types of damping, including the handling of squeeze films. Shock loading due to loss of a blade or through the mounts, nonlinear springs or stiffness variation with speed, and variation of stiffness in the horizontal and vertical planes, also can be handled by means of the VAST program. Modified versions of VAST can handle noncircular orbits and a variety of other complexities.

Installation dynamics analysis allowing six degrees of freedom for the engine is conducted by means of the MASS program. This is required to determine the dynamic load input to the airframe from the engine due to unbalance as well as the response of the engine due to airframe vibrations. If the mounting is such that pitch, yaw, and roll are uncoupled, installation dynamics can be accomplished with VAST by conducting the analysis separately in the horizontal and vertical planes.

The first step in the VAST analysis is definition of a mathematical model of the engine as a composite of many spans and translational and angular springs. In this case, strong judgement is needed to achieve a balance between the sophistication of the model and the required accuracy. Extreme care is needed to identify all nonbeam types of flexibilities and to understand the structural function of each component. Between two locations where elements other than beams are involved, the MASS program is used to obtain the stiffness matrix between the two stations. Since calculated stiffnesses may vary considerably depending upon the assumptions, methods for handling special element connections have been standardized as supported by extensive test data. For example, standard methods have been established to specify in detail the appropriate modeling of different types of couplings, splines, bolted joints, rabbets, curvics, frames, cones, squirrel cages and squeeze films, hydrodynamic bearings, rolling element bearings, various types of shaft-disk connections, split casings, etc. Stiffnesses which vary with frequency due to the participation of the mass of the spring element are so identified in the analysis. That is, the dynamic stiffness of a part in the frequency range of interest is used.

The following effects should be considered during the analysis: a) Rabbets which are tight at room temperature may open during transients or at high speeds. b) Hydrodynamic film stiffnesses and damping vary with speed. c) Clearances in the bearing, and thus stiffness change with speed and temperature. d) Stiffnesses of rolling element bearings are nonlinear functions of axial and radial loads. e) Close clearance parts, such as seals, may act as bearings once the clearance is used up. f) Frame stiffnesses may be different in the horizontal and vertical directions and also as a function of the vibration mode. g) The rotor stiffnesses may vary with speed. h) Working splines may lock at high torques while splines with lands may not transmit moments under certain tolerance and operating conditions.

T700 Engine Dynamic Design

Design Objective

The following major requirements controlled the design of the T700 engine from the point of view of engine dynamics: a) The number of bearings and frames required to support the rotors should be minimized to have a low cost, lightweight engine. b) The design should be balanced to achieve high engine reliability and maintainability, field replacement of modules, clearance control during maneuvers and heavy unbalance situations, a minimum number of sumps, high life requirement of bearings, ruggedness of design, minimum thermal bow effects, high engine integrity under abusive situations due to loss of blades, etc. c) Rotor criticals sensitive to unbalance in the low pressure or high pressure rotors should be outside the steady-state operating range with good margin.

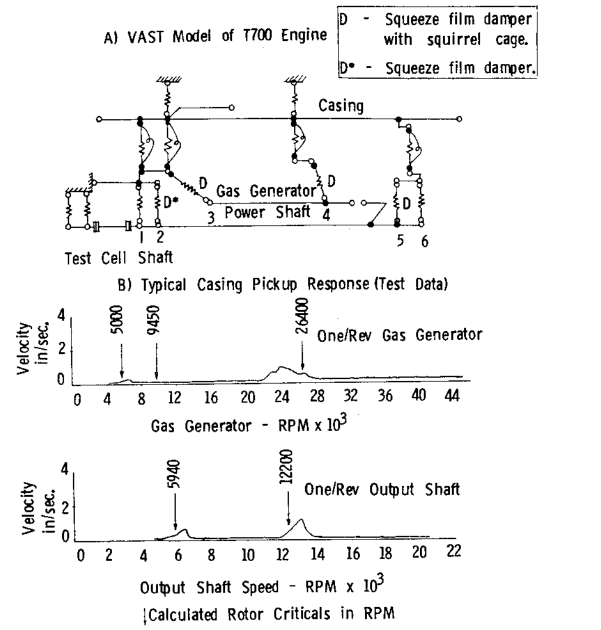


Fig. 8 VAST schematic and typical engine test data of T700 engine.

Design Details and Test Data

A typical mathematical model of the T700 engine dynamic system in a test cell configuration is shown in Fig. 8a. By making use of the VAST program, studies were

Table 1 Summary of system frequencies, T700 test cell engine

	Gas generator rotor criticals
4,960 rpm	
9,450 rpm	
26,400 rpm	
	(30,000 to 44,700 rpm normal operating range)
>70,000 rpm	
	Power turbine rotor criticals
5,940 rpm	
12,200 rpm	
	(17,000 to 21,000 rpm normal operating range)
33,900 rpm	
37,700 rpm	
	Mount criticals
1,940 rpm (32 cps)	
2,570 rpm (43 cps)	
	Engine bending criticals (stator criticals)
3,900 rpm (65 cps)	
13,000 rpm (216 cps)	
43,000 rpm (716 cps)	
56,300 rpm (938 cps)	
	Localized criticals
36,100 rpm	(quill shaft/output shaft)
6,380 rpm (106 cps)	(accessory gearbox)
19,100 rpm (319 cps)	("A" sump/front frame)
20,900 rpm (348 cps)	("C" sump/exhaust frame)
26,900 rpm (448 cps)	("B" sump/mid frame)
44,200 rpm (737 cps)	("B" sump/turbine CSG/exhaust nozzle)

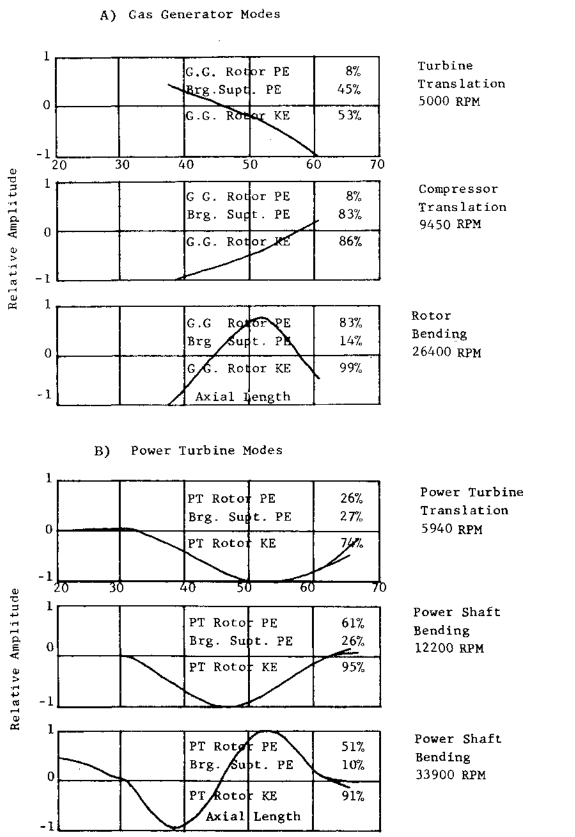


Fig. 9 T700 engine rotor criticals and mode shapes (analysis).

conducted varying the pertinent design parameters, including the bearing support spring rates. Design iterations were then carried out to arrive at the objectives. The system frequencies obtained from the VAST program for the configuration in Fig. 8a, were interpreted based on mode shapes, energy distributions and response to unbalance. They are summarized, in a form so as to evaluate the vibration characteristics of the engine, in Table 1.

As seen in the schematic representation, the T700 gas generator has two bearings mounted on squirrel cages with squeeze films in parallel. The gas generator has three criticals (Table 1) below the idle of approximately 30,000 rpm. No rotor criticals exist in the operating range, which has a maximum speed of 44,700 rpm. The squeeze film dampers provide the necessary damping for passing through the criticals (Fig. 8b) and also act as snubbers under abusive situations. Details of mode shapes and energy distributions from the VAST program for the three subidle criticals are given in Fig. 9a.

The power shaft is hard-mounted on the No. 1 and No. 6 bearings and soft-mounted on damped No. 2 and No. 5 squeeze film bearings (Fig. 8a). No rotor criticals exist in the operating range of 17,000 to 21,000 rpm. Two criticals (Table 1) occur below the operating range whose mode shapes and energy distributions are given in Figure 9b.

Table 3 Bearing loads, in lb

a) Normalized for 1 mil max deflection at gas generator rotor criticals

Critical speed, rpm	No. 3 Brg.	No. 4 Brg.
4,960	2.4	14.4
9,450	21.1	5.2
26,400	25.2	9.1

b) Normalized for 1 mil max deflection at power turbine rotor criticals

Critical speed, rpm	No. 1 Brg.	No. 2 Brg.	No. 5 Brg.	No. 6 Brg.
5,940	2.5	4.4	10.0	2.8
12,200	3.4	8.2	0.8	6.2
33,900	2.5	17.3	36.0	55.5
37,700	4.5	18.9	7.9	4.1

c) With 1 gm-in unbalance at 44,700 rpm gas generator speed

Location	No. 3 Brg.	No. 4 Brg.
Stg. 1 compressor	8.0	13.9
Stg. 3 compressor	4.0	52.8
impeller	3.6	17.3
Stg. 1 GG turbine	0.2	1.2
Stg. 2 GG turbine	2.2	9.5

d) With 1 gm-in unbalance at 21,000 rpm power turbine speed

Location	No. 1 Brg.	No. 2 Brg.	No. 5 Brg.	No. 6 Brg.
Output shaft	4.7	3.3	8.7	11.9
Mid power shaft	17.2	26.6	10.9	28.0
Mid power turbine	0.3	0.9	0.3	3.0

Since the gas generator and the power shaft operate at supercritical speeds, they have excellent tolerance to unbalance. Clearances taken at design speed for 1 gm-in. unbalance at critical locations are given in Table 2 showing the very low amplitudes resulting from soft mounting of the rotors. "Normalized bearing loads" at criticals for 1 mil deflection at the antinode and actual bearing loads at top speed (for 1 gm-in. unbalance at indicated locations) are summarized in Table 3. Actual bearing loads at criticals are obtained by multiplying the normalized bearing loads by scale factors (deflection at the antinode for a given unbalance) that are obtained based on analyses and tests. In addition to the dynamics analysis, deflections of the rotor

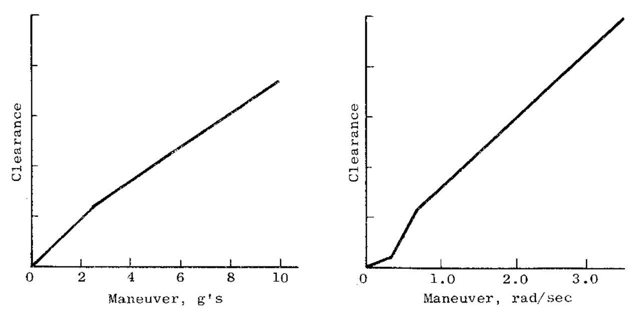


Fig. 10 Impeller radial clearance loss during maneuvers.

Table 2 Clearance loss due to unbalance in mils at design speed

1 gm-in. unbalance at listed location	Gas generator @ 44,700 rpm, power shaft @ 20,000 rpm						
	Stage 3 comp.	Impeller Radial	Impeller Axial	GG turbine Radial	GG turbine Axial	Stage 2 power turbine Radial	Stage 2 power turbine Axial
Stage 1 compressor	0.156	0.034	0.157	0.038	0.153	...	...
Stage 3 compressor	0.736	0.041	0.036	0.0004	0.003	...	...
Impeller	0.035	0.160	0.087	0.043	0.185	...	...
Stage 2 GG turbine	0.009	0.065	0.028	0.068	0.007	...	...
Mid power turbine	...	...	...	...	...	0.147	0.012

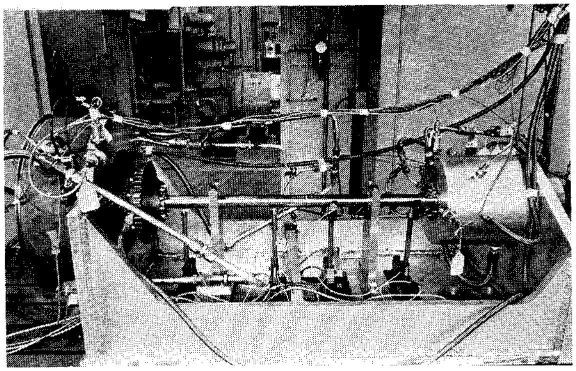


Fig. 11 T700 power shaft rig.

during maneuvers were computed. A typical engine maneuver deflection study at the impeller location is shown in Fig. 10.

To supplement analysis in optimizing the squeeze film parameters, studying the tolerance to unbalance and other practical considerations, a development rig dynamically similar to the power shaft rotor was run (Fig. 11). A typical data plot from the rig test showing the benefit of the forward No. 2 bearing squeeze film damper is shown in Fig. 12 (a and b). Tests were also conducted on the rig to study balancing techniques. Two planes were chosen and the shaft was balanced at both criticals by experimentally obtaining the response to unbalance in the balancing planes. The response of a balanced rotor is shown in Fig. 12c, where the rotor double amplitude is less than 2 mils throughout the speed range. It is impressive that the response of the rotor can be controlled to such low levels by proper balancing and squeeze film application. During the development stage, the rig was used to "high-speed balance" the power shaft rotor. However, since the design philosophy is aimed at having a balance-tolerant field replaceable rotor, modal balancing[8,9] is not done for the normal T700 engines; and, balancing of the shaft is made on conventional balancing machines (rigid rotor balance).

As a result of the detailed analytic, rig and engine development testing, the T700 engine has excellent dynamic characteristics. A typical response of the T700 engine is given in Fig. 8b. Normal characteristics have been less than 1-½ mil single amplitude in the gas generator rotor between idle and 100% speed, with frames showing less than 5 g's of acceleration. The power turbine rotor system typically shows 1 mil single amplitude in the operating speed range, with associated frame vibrations less than 3 g's.

Analysis was also conducted to determine the vibration characteristics of the engine for various mounting configurations by means of MASS, a three-dimensional dynamics program. Mount criticals were placed at appropriate locations minimizing the response of the engine due to the excitation at the mounts and at the engine interface locations. For the mount criticals, the major excitations are

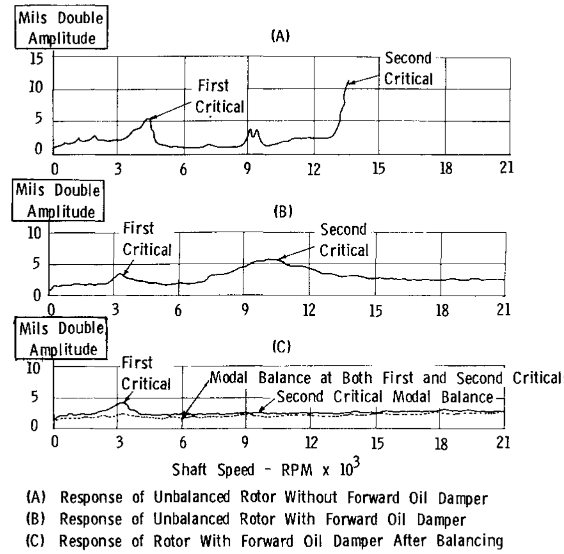


Fig. 12 T700 power shaft rig test data.

from the helicopter rotor. Engine shake tests have been conducted with simulated helicopter input vibration levels for different types of mounting which verified the structural integrity of the installed engine.

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