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Rotordynamic Feasibility of a Conceptual Variable-Speed Power Turbine Propulsion System for Large Civil Tilt-Rotor Applications

Samuel A. Howard
Glenn Research Center, Cleveland, Ohio

July 2012

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National Aeronautics and
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Glenn Research Center
Cleveland, Ohio 44135

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Samuel A. Howard
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

Abstract

A variable-speed power turbine concept is analyzed for rotordynamic feasibility in a Large Civil Tilt-Rotor (LCTR) class engine. Implementation of a variable-speed power turbine in a rotorcraft engine would enable high efficiency propulsion at the high forward velocities anticipated of large tilt-rotor vehicles. Therefore, rotordynamics is a critical issue for this engine concept. A preliminary feasibility study is presented herein to address this concern and identify if variable-speed is possible in a conceptual engine sized for the LCTR. The analysis considers critical speed placement in the operating speed envelope, stability analysis up to the maximum anticipated operating speed, and potential unbalance response amplitudes to determine that a variable-speed power turbine is likely to be challenging, but not impossible to achieve in a tilt-rotor propulsion engine.

Nomenclature

D	pitch diameter
H	blade height
K_{ij}	cross-coupled stiffness (stiffness in *i*, *j* direction)
M	mass of an entire rotor
m	mass of one rotor stage
T	stage torque
V_{RMS}	root mean squared velocity
$X_{pk\text{-}pk}$	peak-to-peak displacement amplitude
β	efficiency factor
ω	frequency of oscillation

Introduction

Interest in an LCTR vehicle for short take-off and landing capability has sparked research into slowed rotor concepts. Desire for high efficiency, high-speed forward flight requires slower main rotor velocities than that which is required for take-off. At present, there are at least two philosophies on how to accomplish a variable speed main rotor. One concept is the use of a variable or multiple speed gearbox to transmit power from a conventional engine to the main rotor. This concept requires shifting of the gearbox while under power to achieve a reduced main rotor speed. There is much research concerned with successfully implementing a multiple speed gearbox of this type. Lewicki gives a brief history of research efforts in the area (Ref. 1). The second concept is to utilize a conventional, single-speed gearbox coupled with a variable-speed power turbine engine that can simply vary the speed of the output power shaft and thus the main rotor speed (Ref. 2). In general, it is likely that the preferred method will be dictated by the nature of the mission. Larger speed ranges may favor the variable gearbox approach, while smaller speed ranges may favor the variable-speed power turbine approach. Still others may require a combined approach.

Objective

One of the many goals of NASA's Subsonic Rotary Wing Project is efficient high-speed rotorcraft propulsion. An LCTR concept was identified in a recent NASA study (Ref. 3) as having potential for satisfying commercial airspace requirements for the future in the short haul regional market. The concept vehicle includes the following capabilities: 90 passenger capacity, 300 knots cruise velocity, 1000 nautical mile range, tilt-rotor for vertical take-off and landing. The current concept incarnation, designated LCTR2 (Large Civil Tilt-Rotor iteration 2), is shown in Figure 1, and features variable main rotor tip speed for increased efficiency and reduced noise. Tip speed varies from 100 percent, 198 m/s (650 ft/s), at hover to 54 percent, or 107 m/s (350 fps), at cruise (Refs. 4 and 5).

Traditionally, rotorcraft gas turbine engines have a single design speed and are coupled to a fixed ratio gearbox such that the main rotor maintains a more-or-less constant rotation speed. Forward velocity and rate of climb (in the case of traditional rotorcraft), and thrust (in the case of a turbo-prop vehicle) are controlled by adjusting the collective pitch of the rotor blades and the fuel rate to the engine (i.e., power output of the engine). One notable exception to this is the U.S. Military, Bell-Boeing V-22 Osprey tilt-rotor which utilizes an ~85 to 100 percent speed variation in the power turbine.

To achieve the variable rotor speed desired in the LCTR2 concept, a variable speed gearbox or a variable speed engine or both is needed. Each arrangement has technical challenges, which must be overcome to provide a viable propulsion system. This paper is focused on the variable speed engine concept wherein an engine with a VSPT is coupled to a fixed-ratio gearbox to provide the necessary tip speed variation for the LCTR2 concept. Two of the major technical challenges associated with this method include: development of incident tolerant blades and achieving acceptable dynamic response (vibration) throughout the speed range. Incident tolerant blading is needed because as the power turbine speed changes, the angle of attack of the turbine blades changes and efficiency can suffer. Incident tolerant blades would minimize this problem and maintain acceptable efficiency over the desired speed range. Rotordynamics is a concern because a relatively wide operating speed range for the power turbine could bring natural frequencies of the system into play. The flexible power turbine shaft could have natural frequencies within the operating speed range or the running speed could coincide with another engine natural frequency resulting in large vibration amplitudes. This paper is focused on the rotordynamic aspects of the notional LCTR2 engine, specifically to determine if a feasible engine architecture exists that satisfies the design requirements of the vehicle while demonstrating acceptable rotordynamic behavior.

The basic engine design (i.e., turbine and compressor stage sizes and weights, shaft speeds, etc.) is obtained from a previous system study analysis of the mission requirements for LCTR2. Information from this study is used to build a rotordynamic model of a notional LCTR2 VSPT turboshaft engine. The model is analyzed to determine rotordynamic behavior over the desired power turbine speed range.

The system study, conducted using NASA's WATE++ analysis code, resulted in a three-spool engine for optimum efficiency. The three rotors are comprised of an axial four-stage power turbine, an axial seven-stage LP compressor and single-stage LP turbine, and a single-stage axial/single-stage centrifugal HP compressor and single stage HP turbine. The WATE++ code calculated the stage geometry information (stage diameter, disk shape, mass, etc.) that was used to build the rotordynamic model. The resulting full engine model is shown in Figure 2. For more about the mission and system study, see Snyder (Ref. 5).

Figure 1.—Artistic rendering of the LCTR2 Concept Vehicle (Ref. 5).

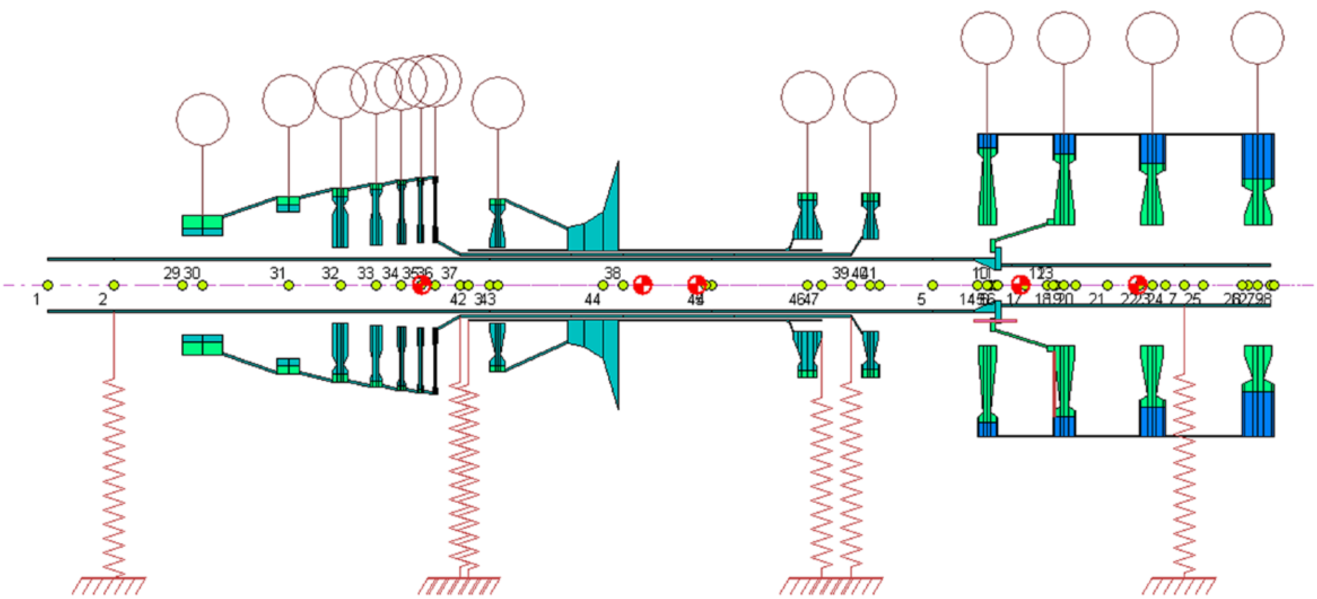

Figure 2.—Full Engine Rotordynamic Model Based on WATE++ Analysis Results for the LCTR2 Concept Vehicle.

The main focus of this analysis is to determine the effect of a VSPT on the rotordynamics of a concept engine, and to identify any issues that would be potential “show-stoppers” for the VSPT concept. The other two rotors are included in the model for completeness, and to identify any potential engine issues associated with the rotors interacting dynamically. As such, the power turbine rotor is separated from the rest of the engine for the initial analyses to make it easier to visualize and interpret results. The power turbine rotor model is shown schematically in Figure 3. The spring icons represent the bearings and the circles represent added mass of the blades.

Critical Speed Analysis

The first analysis conducted with the PT rotor model is a critical speed analysis. A critical speed can be defined as the speed at which the running speed coincides with a natural frequency of the rotor. The critical speeds are functions of the bearing support stiffness, and therefore, it is convenient to plot the critical speeds against bearing stiffness to choose a bearing stiffness for a given design

Figure 4 shows the critical speed map for the PT rotor model for a given shaft diameter. The operating speed range of the power turbine is overlaid on the plot to help visualize the operating range in relation to the critical speeds. The map indicates that bearing stiffness values starting at about 3.5×10^6 N/m (20,000 lb/in.) would yield an operating range clear of critical speeds. Also shown on the plot is a stiffness value of 10.5×10^6 N/m (60,000 lb/in.). This value is chosen throughout this study because it produces critical speeds that lie outside of the operating speed range, and represents a realistic value for bearing stiffness that one might find in an aero engine such as this. In other words, it is an attainable stiffness value that gives the desired results. It is worth mentioning that the model does not contain any bearing support flexibility. Flexible supports can have an effect on the rotordynamics of the system, and it is common or aero engines to have flexible bearings supports (usually in the form of struts and/or squeeze film dampers). However, for this preliminary analysis, it is difficult to include flexibility in the model because it requires knowing a good deal about the engine casing design, secondary flow, lubrication system, etc. These details are not known until later in the detailed design phase of engine development. Given the goal of this study, to determine if this concept is feasible, it is reasonable to only consider the bearing stiffness and damping, and neglect any support flexibility. However, it is worth noting that a combined support/bearing stiffness of 10.5×10^6 N/m (60,000 lb/in.) would likely yield similar results to those outlined here.

Also, the critical speed map depends on the rotor geometry. The map shown represents the stiffness dependence of the rotor critical speeds for one specific geometry. If the rotor geometry changes, the map would change. For example, one might notice that the third critical speed for this rotor occurs at approximately 16,600 rpm when the bearing stiffness is chosen to be 10.5 N/m(60,000 lb/in.). With a maximum speed of 15,000 rpm, that leaves an 11 percent margin for over speed between the maximum speed and the third critical speed. If this margin is deemed to be insufficient, there are two possibilities for increasing it. First, according to this plot,

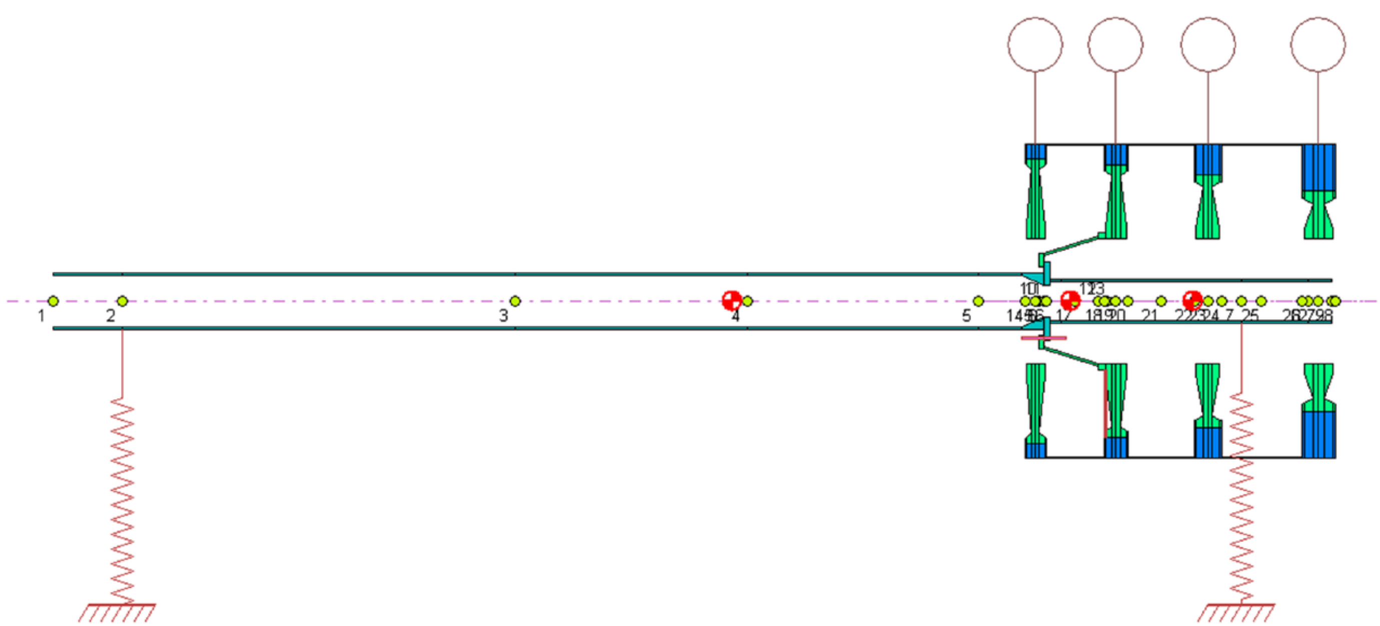

Figure 3.—LCTR2 Power Turbine Rotor Model.

higher bearing stiffness would increase the critical speed slightly and provide a bit more margin. Alternatively, the rotor could be redesigned to try to shift the critical speed curve up. Since the third critical speed is a bending mode, shifting it up would require stiffening the rotor in bending. However, stiffening the rotor might have consequences to the rest of the engine. In fact, an iterative design process such as this was done to arrive at the current rotor geometry. The rotor diameter was smaller in the first iteration, but the bending critical speed did not rise above the operating speed range for any realistic value of bearing stiffness. The rotor diameter was increased forward of the turbine attachment point until the bending mode shape took on the characteristics shown in Figure 4. It may be desirable to increase the critical speed more to gain margin, but making the PT rotor any larger would encroach on the inner diameter of the LP rotor. Thus, it may become necessary during detail design to iterate more on the design to obtain a balance between the desired behavior and all the constraints on the system. For this preliminary feasibility study, the current geometry is found to be acceptable, and the chosen bearing stiffness (10.5 N/m) is carried forward.

A similar ideology is used to choose bearing stiffness values for the LP and HP rotors for the remainder of the analyses. Figure 5 shows the critical speed map for all three rotors.

The LP bearing stiffness is chosen to be 17.5×10^6 N/m (100,000 lb/in.) in order to push the second natural frequency of that rotor up. The HP stiffness is chosen to be 35.0×10^6 N/m (200,000 lb/in.) to get the bounce mode out of the PT operating speed range. It may not be realistic to achieve this high stiffness value depending on the details of the bearing supports for the HP rotor. However, the values chosen at this point are just starting points, and can be modified later in detail design if the need arises. Also, iteration was not done on the LP and HP rotors to optimize the geometry with respect to critical speed placement, which may become necessary later.

The next step in the feasibility study is to look at the stability of all the engine modes and the interaction of the rotors with one another. To study these interactions, the entire engine model must be considered. Thus, in the following discussion, the model shown in Figure 2 is the relevant engine model.

First, it is important to realize the natural frequencies of all three rotors play a role in the vibration characteristics of the engine. Usually, rotor natural frequencies can be excited by their own mass unbalance. Therefore, natural frequencies that are synchronous with the running speed are often excited because the unbalance forcing function matches up with the natural frequency and resonance is observed. The severity of the resonance is dictated by the strength of the forcing function, i.e., the magnitude of the unbalance, and the degree or effectiveness of damping that exists in the system, usually in the form of bearing damping or squeeze film dampers. However, in a multiple rotor system with structural coupling between the rotors, such as in an aero engine, it is possible for the unbalance of one rotor to act as a forcing function to excite the natural frequency of another rotor. Therefore, it is important to take into account all the rotor natural frequencies when analyzing a multiple rotor system. Figure 6 shows a Campbell diagram for the full engine model to enable just such a study.

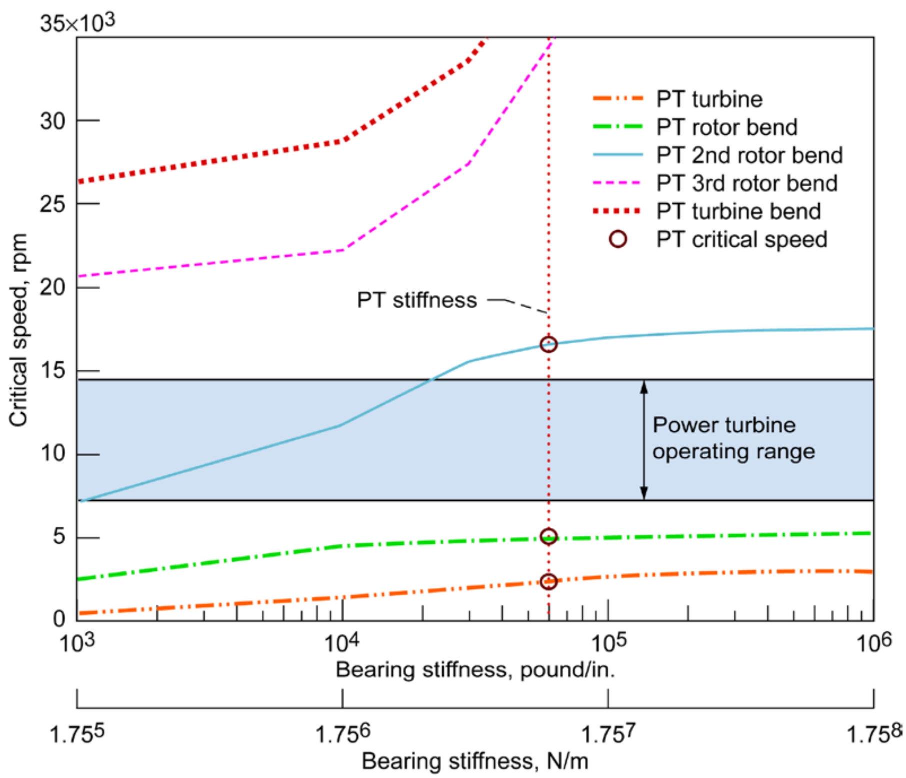

Figure 4.—Critical Speed Map for the LCTR2 Power Turbine Rotor.

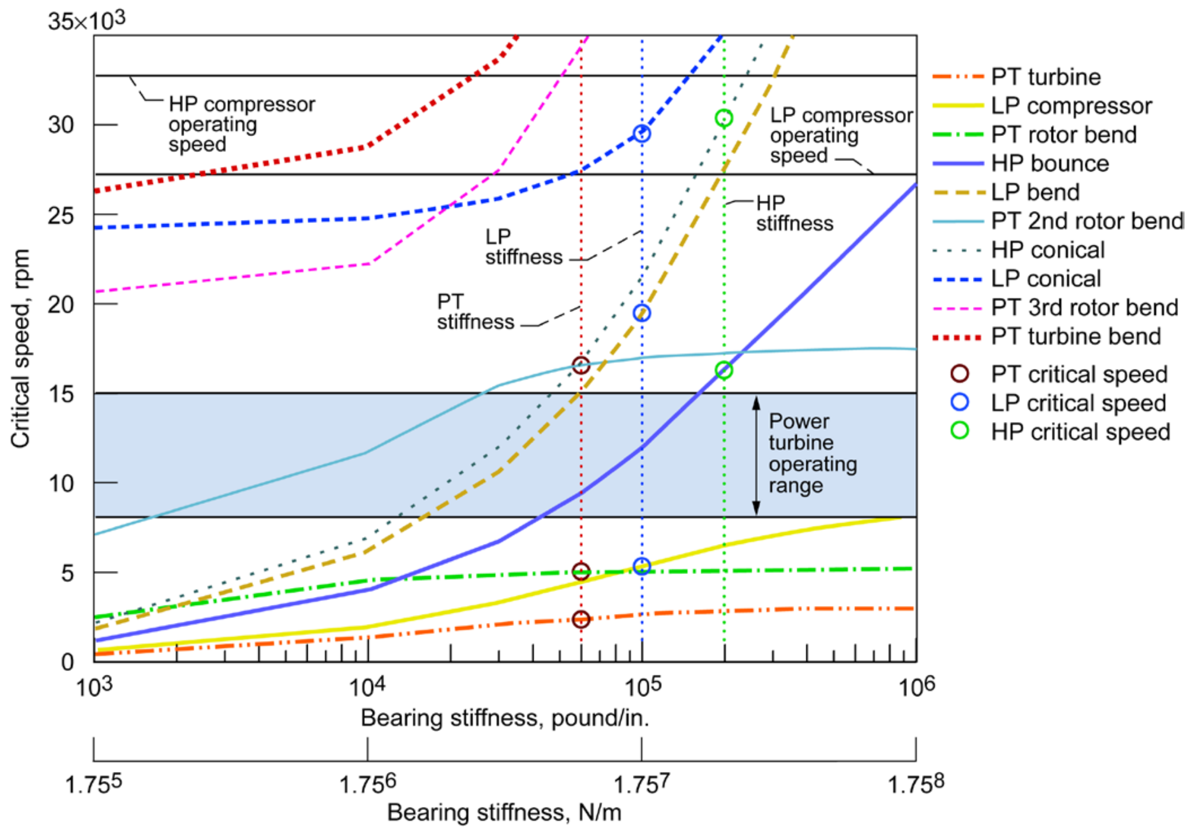

Figure 5.—Critical Speed Map for the Full Engine: All Three Rotors—Power Turbine, Low Pressure, and High Pressure.

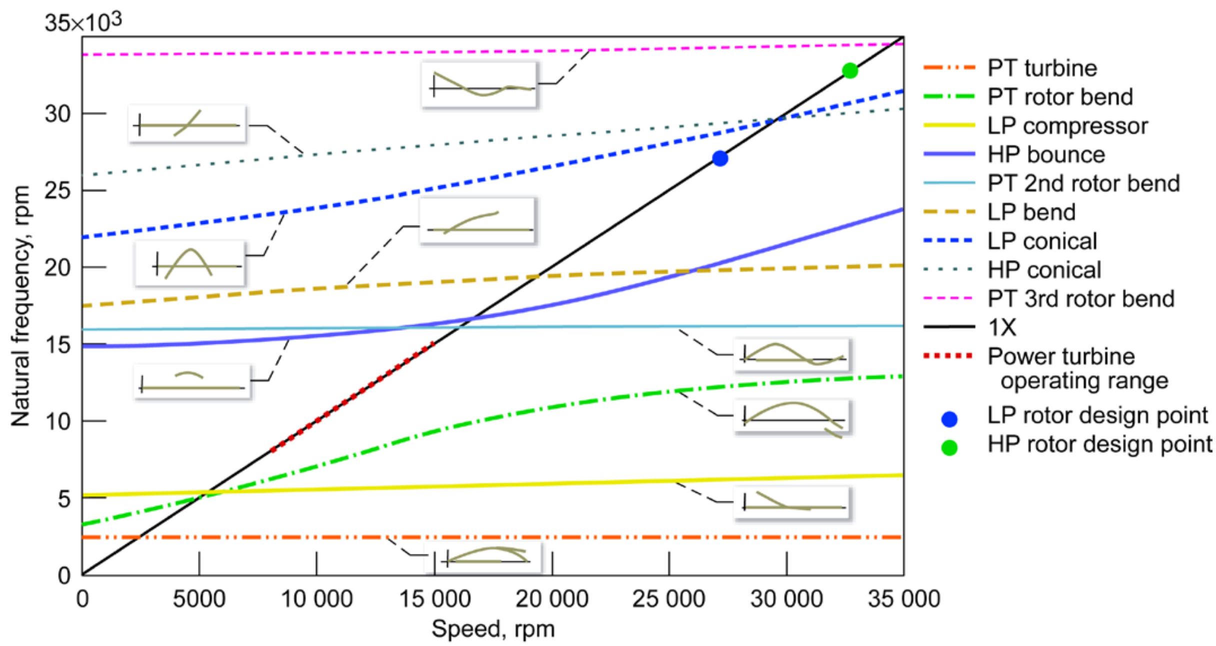

Figure 6.—Full Engine Campbell Diagram up to 35,000 rpm.

The Campbell diagram is a plot of natural frequencies versus speed. The 1X line denotes the synchronous excitation line. Therefore, the speeds at which the 1X line crosses the natural frequencies are defined as critical speeds. Also indicated in the figure are the mode shapes that correspond to each natural frequency. The power turbine operating speed range (8,100 to 15,000 rpm) is shown as a dashed line overlaying the 1X line, and the LP and HP design points are shown as circles overlaying the 1X line at 27,500 and 33,000 rpm, respectively (dots because they are fixed speed rotors).

Several important observations can be made from this plot. First, there are no critical speeds for the power turbine in the operating speed range. This finding agrees well with the above critical speed map discussion. Second, the LP and HP design points are not near critical speeds for those respective rotors. Lastly, there are no natural frequencies that are likely to be excited by the unbalance of another rotor. This is somewhat difficult to see, but if one considers one rotor at a time, it is possible. For example, looking at the LP rotor, its natural frequencies are indicated by the line labeled, “LP Compressor,” the line labeled, “LP Bend,” and the line labeled, “LP Conical.” To see if any of these modes could be excited by the power turbine, one would check to see if any of those natural frequencies occur between the speed of 8,100 and 15,000 rpm. Since they do not, they would not be excited by the power turbine unbalance while running throughout its operating speed range. Similarly, none of the LP natural frequencies coincide with the HP running speed. This can be seen visually on the chart by drawing a vertical line through the LP running speed. Then, project the power turbine operating speed range to the right. Since the projection of the PT operating speed does not intersect any of the natural frequencies of the LP at its running speed, there is no opportunity for excitation. Projecting the HP operating speed to the left, there is no LP natural frequency at the intersection of the HP projection and the LP speed line. Therefore, the HP unbalance will not excite an LP resonance. See Figure 7 for this graphical representation. Likewise, one can do this for each rotor, and see that none of the rotor natural frequencies line up with any of the other rotor’s excitation frequencies. Therefore, this design is likely to be free of unbalance induced resonances at the design speeds. This discussion does not consider any other sources of excitation, such as blade pass frequencies, gear tooth frequencies, or others. Those types of analyses are left to detailed design, but can be considered in a similar fashion.

Stability Analysis

Stability is also a concern, but to get an accurate prediction of stability, one needs to know more about the bearings and supports than is known at the preliminary design stage. For example, what type of bearings are used, if any squeeze film dampers and centering springs will be used, what the bearing supports look like, what the destabilizing forces are, etc. However, to get an idea of the design’s robustness, a simplified stability analysis is conducted with an assumed level of damping to determine if this design is potentially viable. For the LCTR2 engine, the most likely sources of destabilizing forces are the aerodynamic components, the compressors and turbines. In order to assess the stability characteristics of this engine design, a model of the destabilizing forces is needed. The aerodynamic destabilizing forces are typically called Alford’s forces after the first researcher to publish a mathematical treatment of the destabilizing effect of eccentric axial

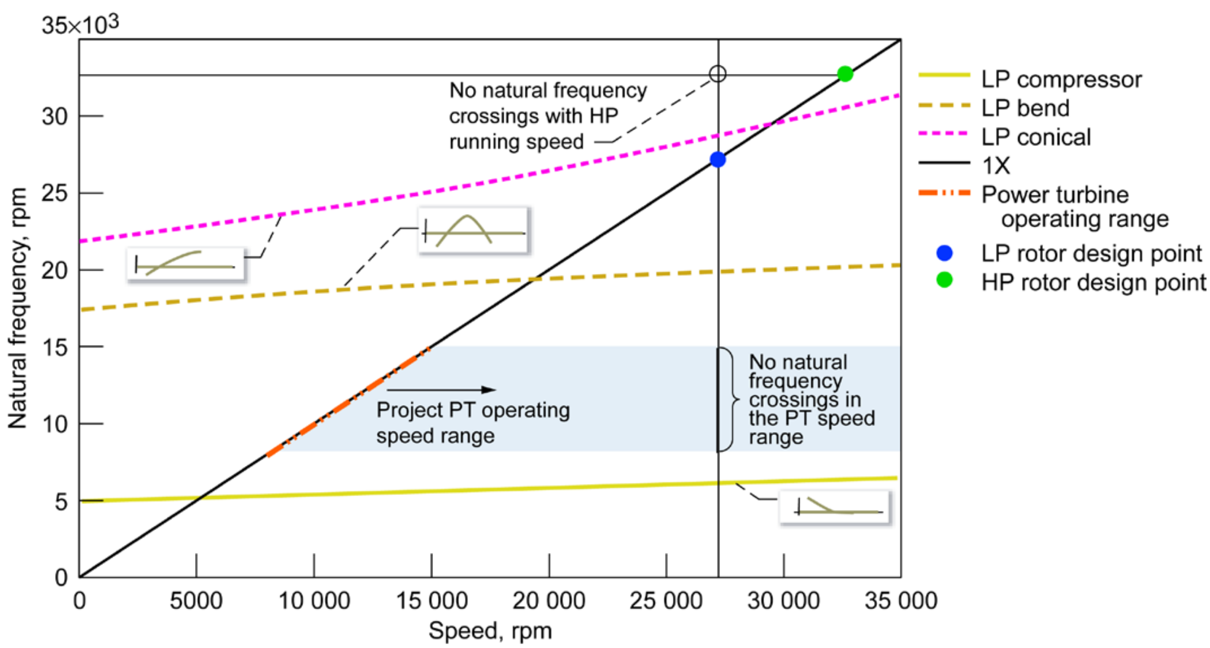

Figure 7.—Campbell Diagram Showing LP Natural Frequencies and Projections of PT and HP Excitation Frequencies.

compressors and turbines (Ref. 6). The theory is based on the assertion that the tangential aerodynamic forces in a compressor or turbine will not be axisymmetric if the compressor or turbine is not concentric with the housing. The unbalanced forces will generate a net tangential force in the direction of rotation that is proportional to the magnitude of eccentricity. Thus, as the blade tips move radially toward the housing, a force builds up that tends to push the wheel to a larger eccentricity where the force grows and pushes the wheel even further. Clearly, this force is not conservative and can drive the system unstable. Mathematically, these forces are modeled by added cross-coupled stiffness in the stiffness matrix of the rotordynamics equations of motion. The destabilizing force generated in each stage is proportional to the power and inversely proportional to the blade height and mean blade pitch diameter. The Alford's force model is applied to each stage of all three rotors to determine the amount of damping needed for stability.

Fluid film bearings (plain sleeve bearings, tilt-pad bearings, etc.) are advantageous because they can supply significant amounts of damping to a rotating system that aids in dynamic stability. They also can be a source of instability with so-called oil-whip self-excited vibrations. Fluid film bearings are typically not used in aero propulsion engines because they are susceptible to sudden and catastrophic failure if the lubricant supply is compromised. On the other hand, rolling element bearings (ball and roller bearings) have the ability to continue running for short periods of time with little or no lubrication. Although starved operation severely shortens rolling element bearing lifetimes, the ability to survive short duration events may give enough time to safely land an aircraft. Thus, rolling element bearings are used almost exclusively in aero-propulsion engines. Rolling element bearings are also attractive from a rotordynamics standpoint in that they do not generate cross-coupled stiffness or destabilizing forces like fluid film bearings. However, they also do not provide significant damping. Thus, rolling element bearings generally do not destabilize a rotating system but they also do not help damp out other sources of instability. One such source of instability is Alford forces discussed previously. Therefore, in the case of aero engines that typically have high power turbines and compressors (large Alford forces) on rotors supported by rolling element bearings (little to no damping), there is a high potential for rotordynamic instability. In turbo-shaft engines, such as the LCTR2 concept engine, there is an added complication. Usually, the power turbine rotor is a relatively long rotor with a large bearing span. This is due to the fact that the rotor must extend beyond the core (LP, HP, and combustor in the LCTR2 case) both in the front of the engine and the rear of the engine. The PT rotor is typically limited in diameter because it must pass through the inner diameter of the core rotor/s. This combination often leads to supercritical operation where the PT runs at speeds above its first (and sometimes second) critical speed (Ref. 7). Since instabilities often manifest in large amplitude (or unbounded) vibration at a sub-synchronous natural frequency, supercritical operation makes a rotor bearing system susceptible to instability because there exists a sub-synchronous natural frequency to excite. Often, these characteristics—large destabilizing forces, low damping, and supercritical operation—result in the need for additional damping to reduce synchronous vibration amplitudes when passing through critical speeds, and to forestall rotordynamic instability.

The most common method to add damping to a rotor-bearing system is through the addition of squeeze film dampers (SFD). An SFD is like a fluid film bearing in that it is comprised of a fluid-filled gap between two concentric cylinders. However, in an SFD, the cylinders do not rotate with respect to one another, rather they allow relative radial motion only. The movement of the inner cylinder with respect to the outer cylinder results in a squeezing effect on the fluid in the gap and an associated damping force, thus the name squeeze-film damper.

For the reasons just discussed, a rotordynamic stability analysis is conducted on the LCTR2 conceptual engine design to determine how much additional damping is required. If the amount of damping required for stability is less than that which could be provided by a typical squeeze-film damper configuration, then the design is deemed feasible from a stability standpoint.

The following Alford force model from Vance (Ref. 7) is used to approximate destabilizing forces in the LCTR2 conceptual engine:

$$K_{xy} = -K_{yx} = \frac{\beta T}{DH} \tag{1}$$

Where T is the stage torque, D is the pitch diameter or mean passage diameter, and H is the blade height. β is an empirical efficiency factor for the stage that ranges roughly from 0.5 to 10.0 for various types of components from shrouded axial compressors to extreme cases of overhung radial impellers. This model is used to calculate the aerodynamic destabilizing forces at each stage. It is important to keep in mind that this is an approximation of the expected magnitude of destabilizing forces, not an exact representation, and the application of this model to axial turbines is not well vetted.

The stability analysis is conducted by inserting the above Alford forces into the equations of motion for the rotor system and solving for the complex eigenvalues. The imaginary part of each eigenvalue represents the frequency of the associated eigenvector (or mode shape), and the real part represents the exponential growth or decay. Thus, if the real part of a given eigenvalue is positive, the amplitude of vibration for that particular mode shape grows in time, and is therefore unstable. If the real part is negative, the amplitude decays with time and the mode is stable. So, in linear stability analysis, one solves the eigenvalue problem for a given rotor/bearing system, and looks for positive real parts to the eigenvalues. Generally, one is only concerned with the stability of eigenvectors (mode shapes) that are forward whirling (as opposed to backward whirling) in nature and are below the maximum speed of operation. Backward whirling mode shapes exist, but are difficult to excite in general and therefore are usually of little concern (they can be excited, for example, by impact rubs). For every rotor system, there exists a speed above which one of the eigenvalues becomes unstable. This speed is called the instability threshold speed (Ref. 8). A goal of robust rotordynamic design is to ensure that the instability threshold occurs above (hopefully well above) the maximum operating speed of the system.

To that end, the LCTR2 model was analyzed with increasing amounts of damping at the bearing locations until all the mode shapes were stable up to the maximum operating speed of the respective rotors. Rather than looking directly at the real part of the complex eigenvalues, the software used for this analysis reports the logarithmic decrement instead. The logarithmic decrement (log dec for short) is defined as the natural logarithm of the amplitude of one peak in the vibration response curve divided by the amplitude of the following peak for the rotor system of interest. Figure 8 gives a graphical representation of the definition. With this definition, if the log dec is positive, the vibration response decays with time and if the log dec is negative, the response grows in time. Thus, instability is indicated by a negative log decrement. In addition, the value of the log decrement is proportional to how quickly the vibration decays, with a larger positive value indicating faster decay. Mathematically, a log decrement value above zero is stable, but in practice, values above 0.1 or 0.2 are desired to ensure quick decay of excitations. In this analysis, the damping was increased until the log decrement values were mathematically stable. During detailed design, larger damping may be desired to increase the stability margin. Figure 9 shows the stability map (log decrement vs. speed) for the LCTR2 engine. Only the forward modes below the maximum operating speed are shown. As one can see,

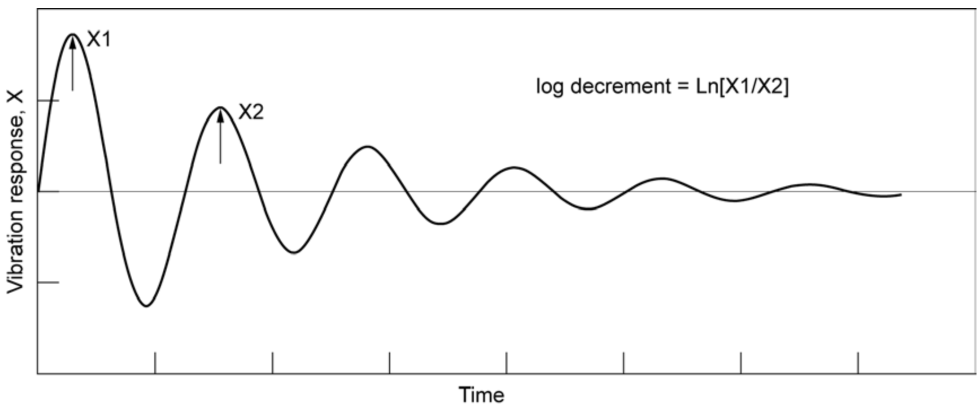

Figure 8.—Definition of Logarithmic Decrement.

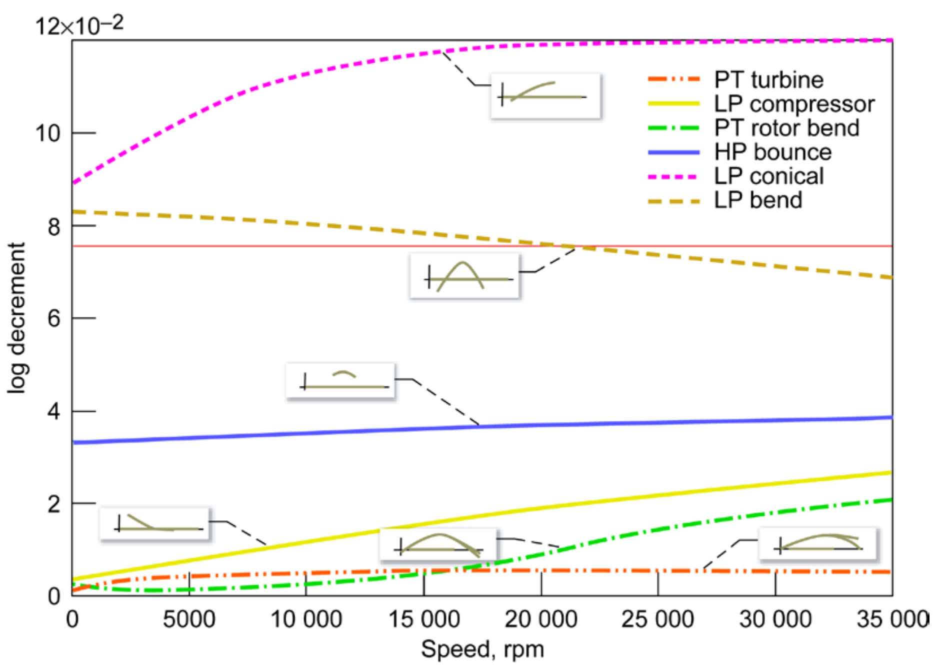

Figure 9.—Stability Map for the LCTR2 Concept Engine.

none of the modes are unstable for the speed range of interest. Table I lists the amount of damping required at each bearing location to result in stable operation. Since all of the damping values are relatively low compared to what one might expect to achieve with an SFD (on the order of 17.5×10^3 Ns/m (100 lb sec/in.) is typically attainable), stability is considered to be possible for this system. This analysis does not guarantee stability because designing an SFD to supply the proper amount of damping is a matter for detailed design. Too much damping and/or damping in the wrong location does not help stability, and can actually be detrimental. However, the analysis indicates that the design is likely to be feasible from the stability standpoint without requiring unreasonable amounts of damping.

TABLE I.—MINIMUM DAMPING FOR STABILITY.

Rotor	Damping at front bearing location, N sec/m	Damping at rear bearing location, N sec/m
PT rotor	525	525
LP rotor	350	350
HP rotor	350	350

Steady State Unbalance Response

The final consideration in this feasibility study is the steady state response to unbalance. This is important to consider because it dictates how well the rotor systems would need to be balanced to result in acceptable amplitudes of vibration both at the operating speeds and passing through resonances.

A common unbalance magnitude for a new aircraft gas turbine engine is on the order of 0.0127 mm * M, where M is the mass of the rotor in grams (this is approximately equivalent to shifting the center of mass of the rotor 0.0127 mm (0.0005 in.) away from the rotational axis). For the current unbalance response analysis, this rule-of-thumb is applied to the model and the predicted response amplitudes are evaluated. This rule-of-thumb unbalance magnitude is for the entire rotor, but in the model the unbalance is distributed along the rotor at various nodes. As an approximation to the expected actual unbalance distribution, an unbalance equal to 0.0127 mm * m is used at each stage, where m is the stage mass in grams. While there are several ways the unbalance could be distributed in the model, this is thought to be a reasonable first approach since most of the rotor mass is concentrated at the stage disks. Figure 10 is a representation of the power turbine rotor with the triangular icons indicating the unbalance locations. The LP and HP rotors are modeled similarly, with unbalance locations at each compressor stage and each turbine stage.

The unbalance response can be visualized in several ways. One convenient visual is called a Bode plot, which is a graph of synchronous vibration amplitude as a function of speed. In this manner, one can visualize the unbalance response not only at the operating speed, but also at all other speeds, including passing through resonances. Figure 11 is a Bode plot for the PT rotor very near the center of gravity for the turbine. The resonances at about 2600 and 5000 rpm are clearly visible as peaks in the response amplitude. These two resonant frequencies agree well with the first two critical speeds shown on the critical speed map (Figure 4) for the PT rotor. The amplitude of the response at 2600 rpm is ~0.178 mm (0.007 in.) radial or ~0.356 mm (0.014 in.) peak-to-peak. The allowable amplitude is a function of the tip clearances in the disks and the resulting load on the bearings.

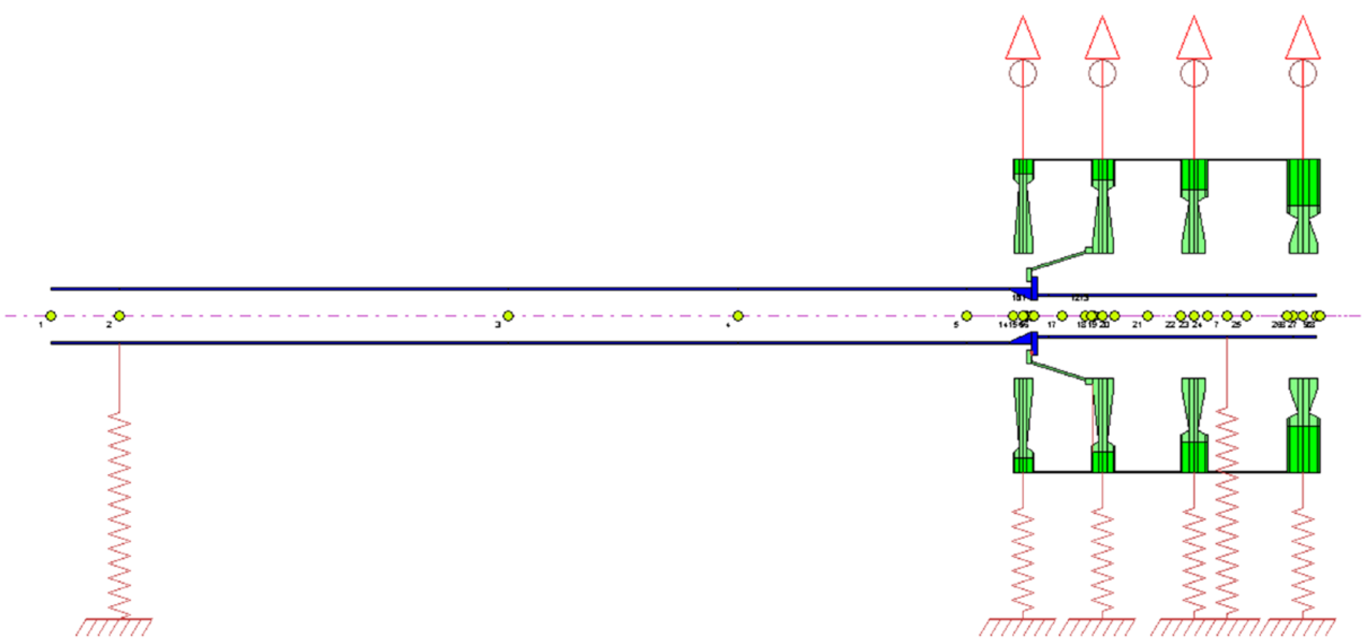

Figure 10.—Power Turbine Rotor Showing Residual Unbalance Planes at Each Stage. Unbalance Denoted by Red Triangles.

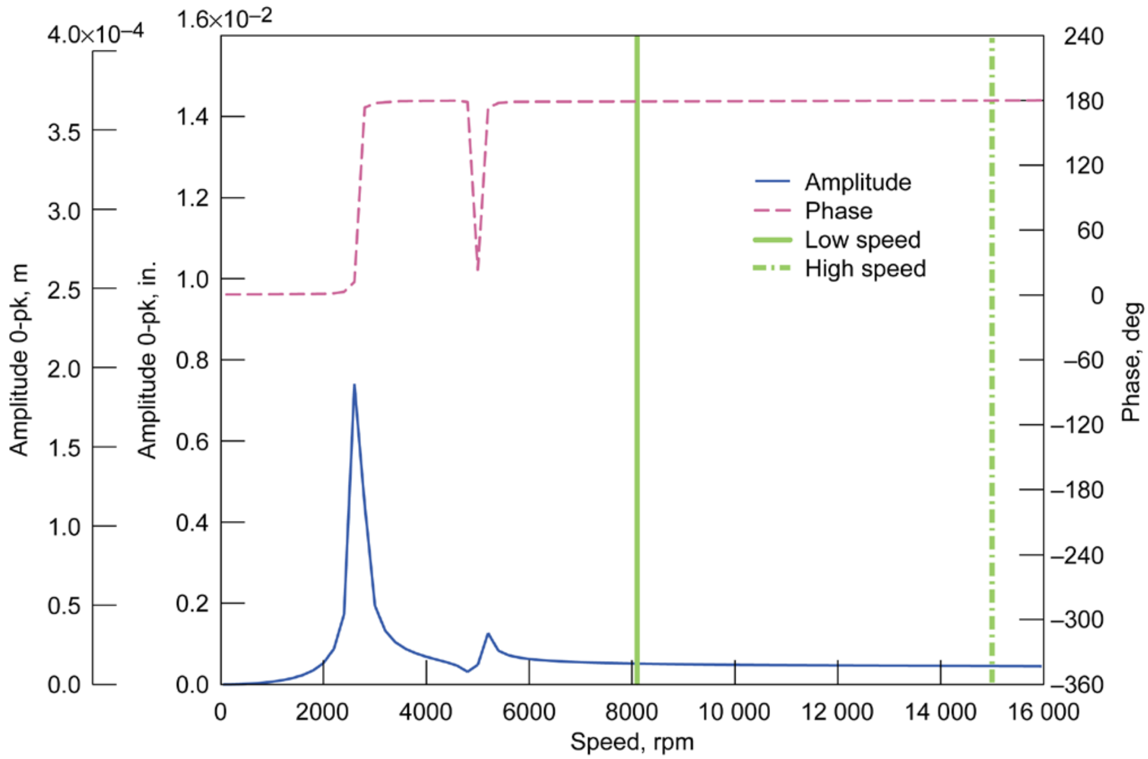

Figure 11.—Bode Plot for the LCTR2 Power Turbine Rotor (Axial Position Near the Second Stage of the Turbine).

It is not known in this preliminary analysis what the tip clearances will be, but 0.178 mm is not thought to be unreasonably large for passing through the resonance for an engine of this size. In reality, other constraints can be as important or even more so than the tip clearances, for example the radial clearance in any squeeze film dampers that may exist, but that is beyond the scope of a feasibility study. More important than the resonance amplitude, assuming the blades do not rub, and the rotor can safely pass through, is the amplitude of steady state vibration at operating speed. Thus, for the power turbine, one would be concerned with the amplitude of vibration from 8,100 to 15,000 rpm. From Figure 11, it is apparent that the amplitude of vibration in that speed range is fairly flat, ranging from 0.127 to 0.010 mm (5.0×10^{-4} to 4.0×10^{-4} in.).

According to an ISO International Standard (Ref. 9), the recommended balance quality for aircraft gas turbines is G6.3. To meet this standard, a machine must exhibit a maximum vibration velocity of 6.3 mm/s root mean squared (RMS) at the bearing locations. The RMS velocity limit can be converted to a peak-to-peak displacement limit if one assumes the vibration response is a single frequency sinusoidal oscillation using the relation:

$$X_{pk-pk} = \frac{\sqrt{2}V_{\mathrm{RMS}}}{\pi\omega} \qquad (2)$$

Where $X_{pk\text{-}pk}$ is the peak-to-peak displacement amplitude in mm, V_{RMS} is the RMS velocity limit in mm/s, and ω is the frequency of oscillation in Hz (Ref. 10).

The response for the power turbine rotor at the bearing locations is listed in Table II. As shown in the table, the response amplitudes at both bearing locations falls below the limit for a smooth-running aircraft gas turbine. Thus, as long as the power turbine can pass through the two critical speeds at 2600 and 5000 rpm, it should provide sufficiently low vibration amplitude throughout its operating speed range.

TABLE II.—VIBRATION AMPLITUDE FOR POWER TURBINE

Speed, rpm	$X_{pk\text{-}pk}$ at front bearing location, mm	$X_{pk\text{-}pk}$ at rear bearing location, mm	Allowable *pk-pk* amp mm
8100	5.08×10^{-4}	8.89×10^{-3}	2.11×10^{-2}
15000	3.81×10^{-3}	8.64×10^{-3}	1.13×10^{-2}

One final consideration in accessing the severity of the unbalance response is the bearing loads. Figure 12 shows the reaction force in the bearings as a function of speed. The maximum load for the front bearing occurs when passing through the second critical speed, and the maximum load on the rear bearing occurs when passing through the first critical speed. The worst-case is the rear bearing while passing the first critical speed, and is on the order of 890 N (200 lb). The dynamic load at operating speed is on the order of 89 N (20 lb). For the size bearing typical of a rotor in the size class of the power turbine, the maximum steady-state and dynamic loads are likely to be on the order of 10 kN. Due to the fact that the balance condition achieved in the real engine could be improved from what was assumed in the model, and the bearing loads are small compared to their capabilities, the design is considered feasible from the standpoint of critical speed margins, unbalance response amplitudes as compared to clearances, and bearing loads.

Similar analyses for the LP and HP rotor reveals that they have larger responses relative to their allowable standards, and larger bearing loads as seen in Figure 13 to Figure 16. However, this is not a serious concern because there was no iteration on the LP and HP rotor designs to achieve maximum margin on critical speeds. Even without the optimum design, the bearing loads are well within acceptable limits for dynamic load capacity of bearings sized for this application, even though the amplitudes are larger than desired. With a bit of rotor optimization, better fidelity on bearing parameters, and tighter balance specifications, it is expected that the amplitudes and bearing loads could be reduced to allowable levels. For example, without doing a complete parametric study, recall that the damping in the model is merely the minimum required for stability. If the damping is increased from 350 Ns/m (2.0 lb sec/in.) for the LP rotor to 3500 Ns/m (20 lb sec/in.), the response becomes much more manageable, and the bearing loads are reduced. Figure 17 and Figure 18 can be compared to Figure 13 and Figure 14 to see the improvement this one change makes. Other changes, such as better balancing, bearing stiffness, and shaft geometry can also help. Likewise, the HP rotor can benefit from a parametric optimization to achieve satisfactory response.

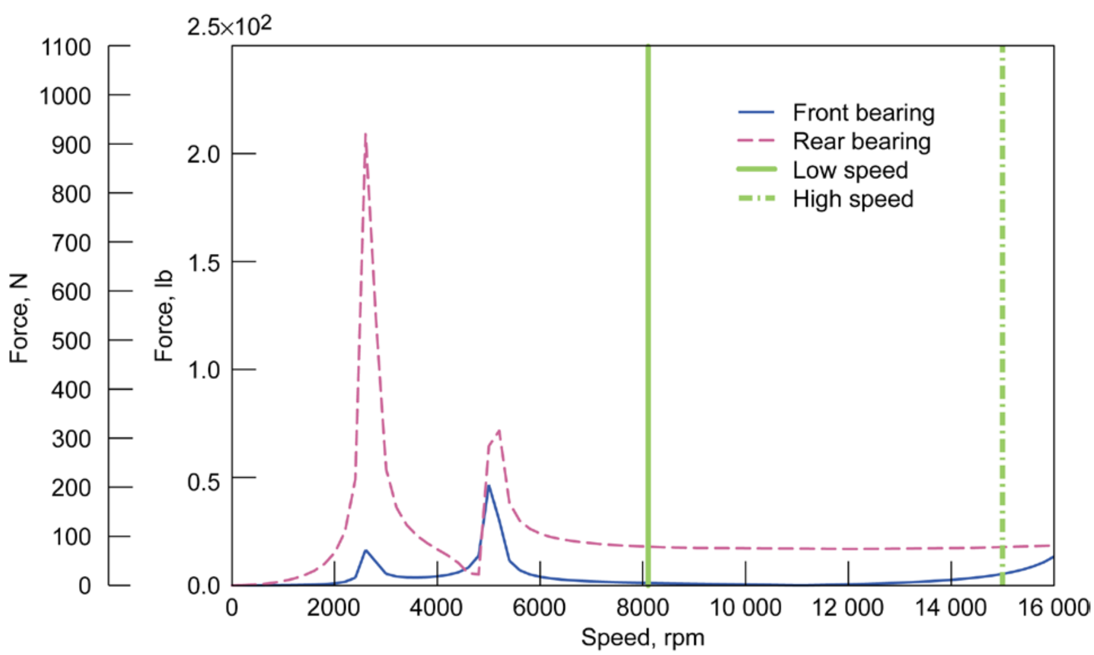

Figure 12.—Plot of Transmitted Forces at the Front and Rear Bearing Locations for the LCTR2 Power Turbine Rotor.

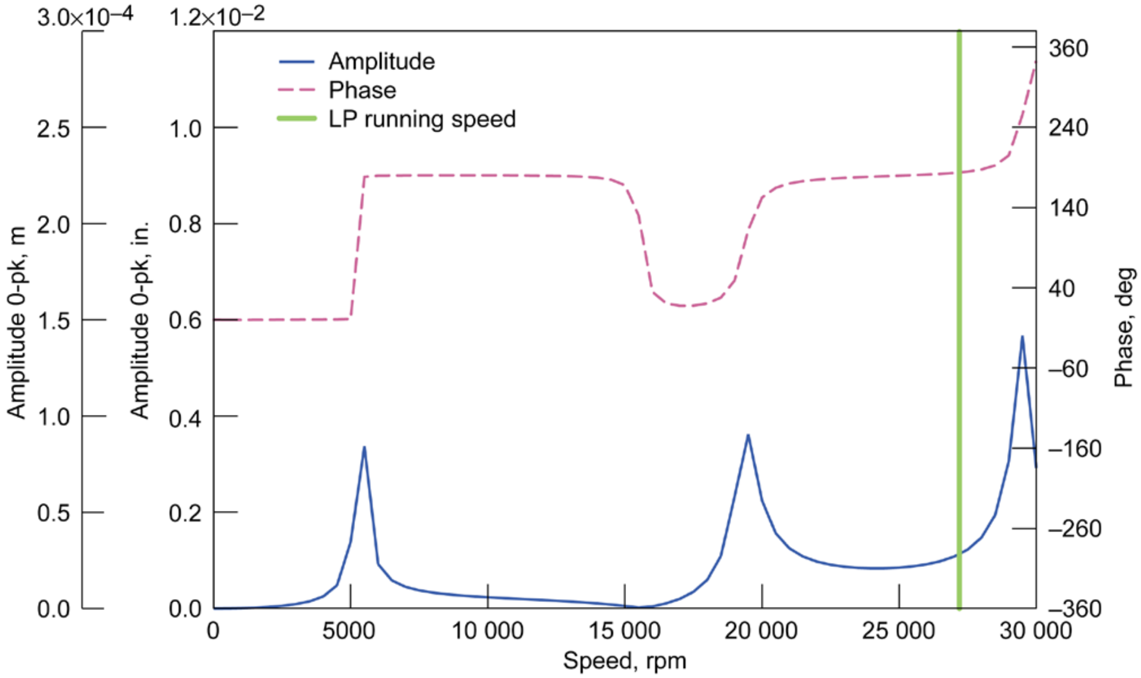

Figure 13.—Bode Plot for the LCTR2 LP Rotor (Axial Position at the Front Bearing Location).

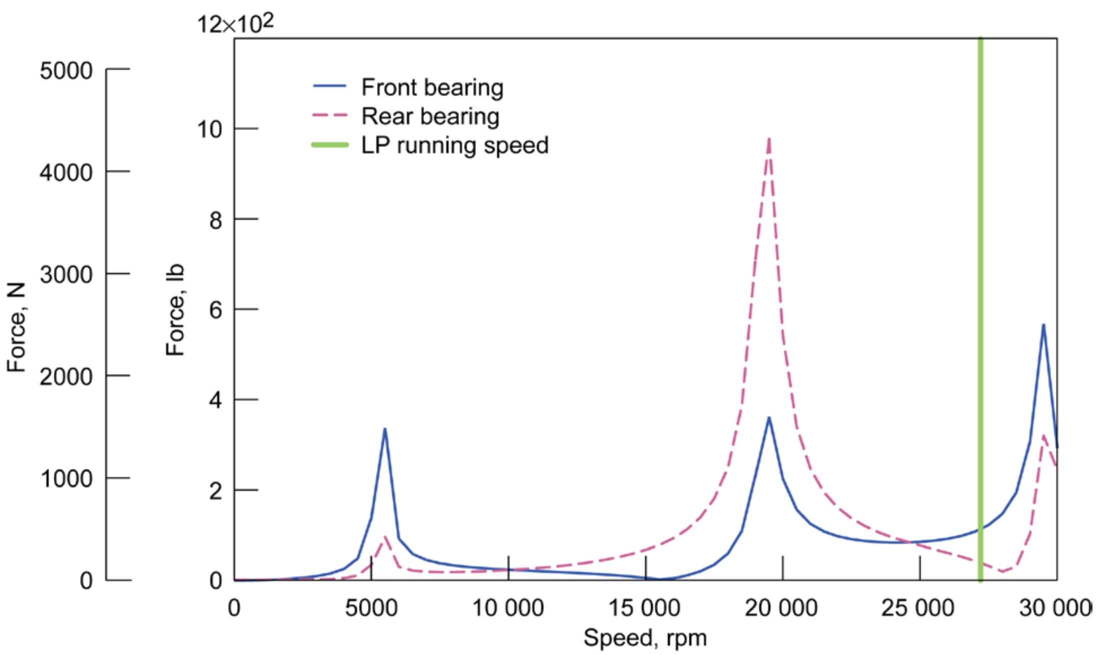

Figure 14.—Plot of Transmitted Forces at the Front and Rear Bearing Locations for the LCTR2 LP Rotor.

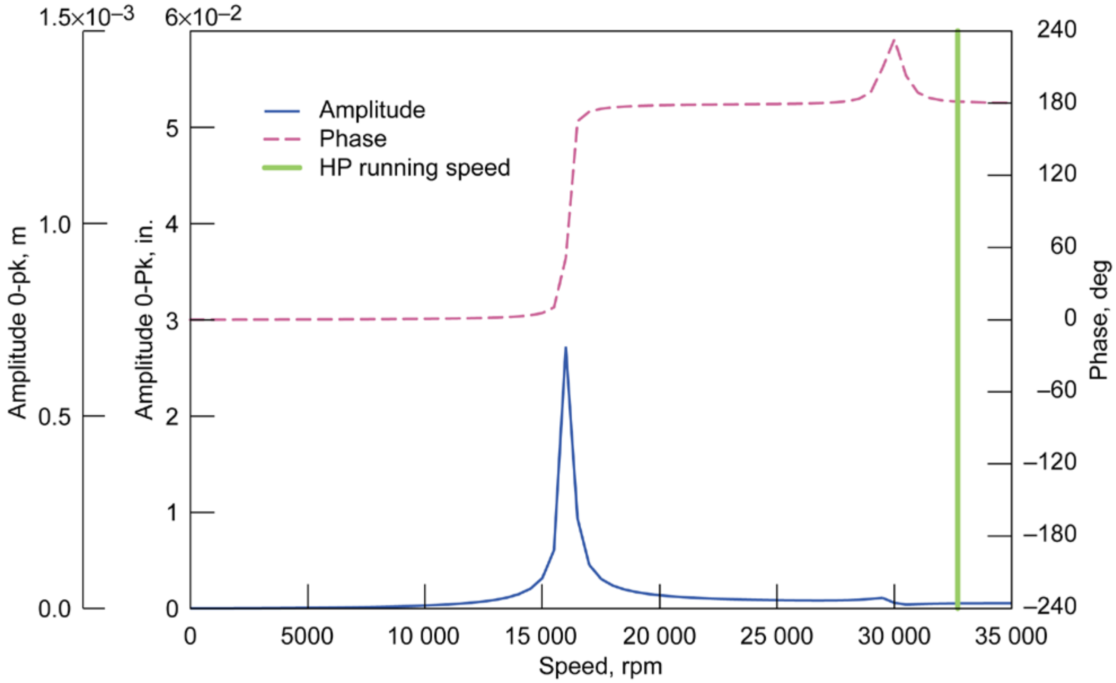

Figure 15.—Bode Plot for the LCTR2 HP Rotor (Axial Position Near the CG of Radial Stage).

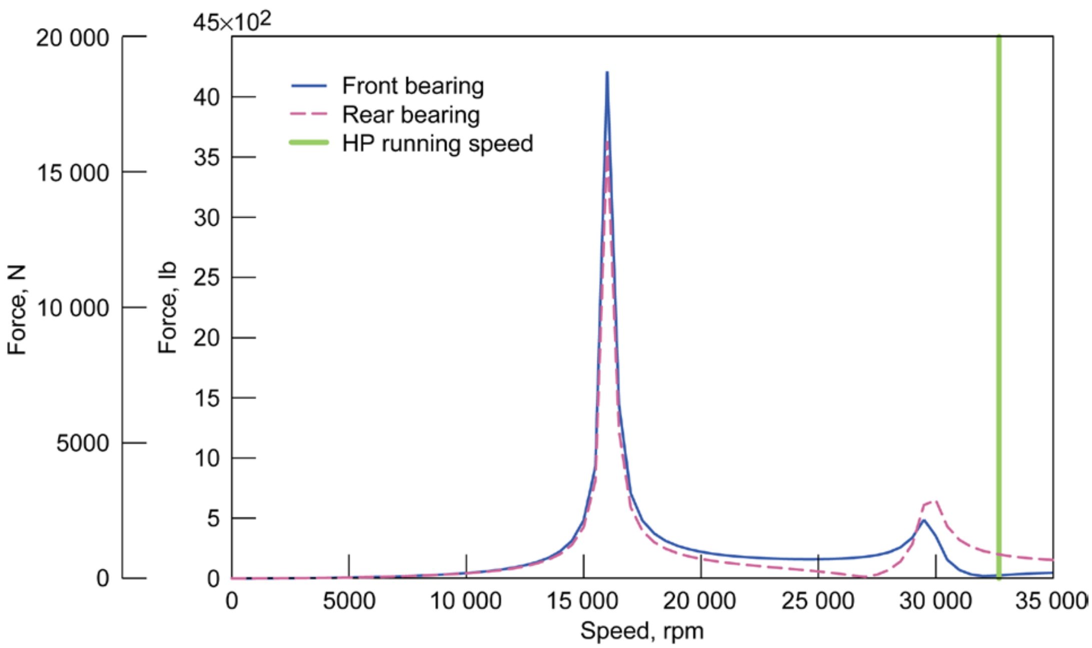

Figure 16.—Plot of Transmitted Forces at the Front and Rear Bearing Locations for the LCTR2 HP Rotor.

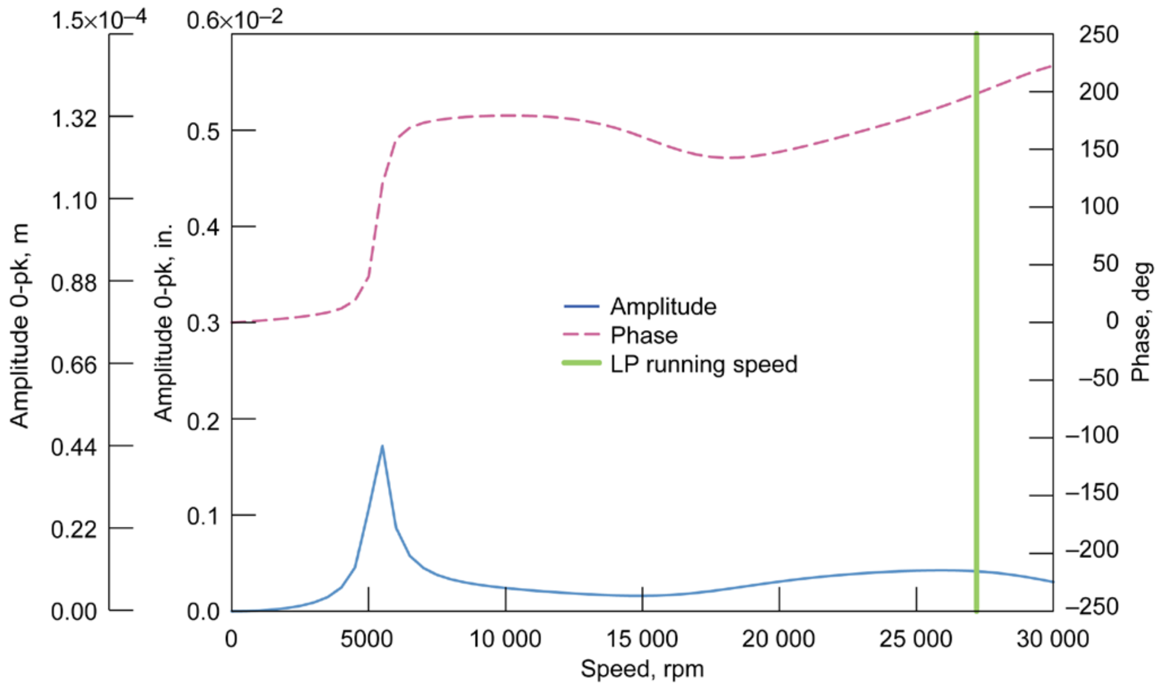

Figure 17.—Bode Plot for the LCTR2 LP Rotor With 20 lb sec/in. Damping at Each Bearing Location (Axial Position at the Front Bearing Location).

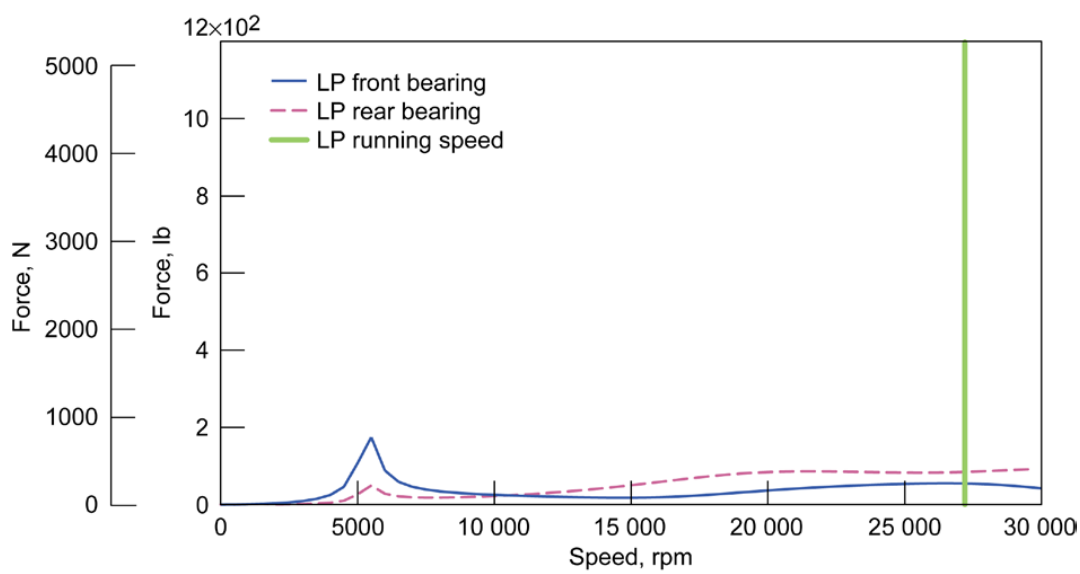

Figure 18.—Plot of Transmitted Forces at the Front and Rear Bearing Locations for the LCTR2 LP Rotor With 20 lb sec/in. Damping.

Conclusions

A rotordynamic model was created and analyzed for a conceptual engine architecture sized to meet the mission requirements of a large civil tilt rotor vehicle. A previous system study resulted in size and mass information for the various compressor and turbine stages, as well as a cartoon engine layout. This data was used in the creation of the rotordynamic model, and some iteration was performed on the power turbine shafting to achieve an operating speed range (54 to 100 percent) free of critical speeds.

The natural frequencies of the three rotors were calculated and it was observed that none of the frequencies exist at the run speed of any of the rotors, thus minimizing the potential for cross-talk between the rotors, i.e., excitation of one rotor's natural frequency by a different rotor. Stability was also analyzed, and the damping required to suppress instability was found to be well within the expected damping range provided by typical squeeze film dampers. Therefore, it is likely that the engine could achieve stable operation at all speeds. Lastly, unbalance response was found to be acceptable in the power turbine rotor, but higher than desired in the LP and HP rotors. The large amplitude unbalance response and somewhat large bearing forces in the LP and HP are not thought to be major concerns because their designs were not optimized and a modification to the damping was shown to significantly improve the response for the LP rotor. Similar improvements are expected with the HP rotor and it is likely that other parameters can have a positive effect as well.

The results of this analysis indicate that a variable speed power turbine in an engine sized for the LCTR is feasible, but does present some challenges. It may prove difficult to optimize the shaft diameters in such a way to eliminate critical speeds from the running range if they need to increase in size from the current iteration. It may also be difficult to design the geometry and bearing support structure (i.e., appropriate stiffness and damping) to ensure acceptable vibration amplitudes for all the rotors at steady state and while passing through the resonances during speed-up and shut-down depending on the level of unbalance attainable in practice.

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14. ABSTRACT
A variable-speed power turbine concept is analyzed for rotordynamic feasibility in a Large Civil Tilt-Rotor (LCTR) class engine. Implementation of a variable-speed power turbine in a rotorcraft engine would enable high efficiency propulsion at the high forward velocities anticipated of large tilt-rotor vehicles. Therefore, rotordynamics is a critical issue for this engine concept. A preliminary feasibility study is presented herein to address this concern and identify if variable-speed is possible in a conceptual engine sized for the LCTR. The analysis considers critical speed placement in the operating speed envelope, stability analysis up to the maximum anticipated operating speed, and potential unbalance response amplitudes to determine that a variable-speed power turbine is likely to be challenging, but not impossible to achieve in a tilt-rotor propulsion engine.

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