Transient Rotordynamic Analysis for the Space-Shuttle Main Engine High-Pressure Oxygen Turbopump

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Theme

A SIMULATION study has been conducted to examine the transient rotordynamics of the space shuttle main engine (SSME) high pressure oxygen turbopump (HPOTP). The study utilizes a general-purpose, transient, flexible-rotor simulation model to anticipate and avoid rotordynamic problems *early* in the development program. By contrast, simulation efforts in the past have resulted from (sometimes catastrophic) hardware failures, and have been hampered by ad hoc models of limited capability.

Simulations were performed for steady-state operations at EPL (emergency power levels) and for critical speed transitions. No problems are indicated in steady-state operation of the HPOTP at EPL; however, a rubbing condition is predicted at the turbine floating-ring seals during critical speed-transition at shutdown.

Contents

The SSME HPOTP is being developed by the Rocketdyne Division of Rockwell International. The study reported here was conducted after the turbopump design was generally fixed, but prior to the completion of all component drawings, and was the result of a cooperative effort by the author, NASA, and Rocketdyne personnel.

The transient rotordynamics of the model were examined via a digital simulation model, which was based on Childs'[1] formulation for flexible rotating equipment. The model accounts for both the rigid body and structural dynamics of the rotor, and was defined in terms of the following Rocketdyne supplied data:

a) "Free-free" natural frequencies and bending modes for the rotor (at zero speed). Rocketdyne personnel developed a lumped-parameter structural-dynamic model, which consists of 13 rigid bodies connected to each other by massless beam elements. To approximate the bending deflections of the rotor, each rigid body has both a displacement and rotation degree of freedom. Torsional and longitudinal deflections of the rotor are neglected. The first four free-free modes of the structural-dynamics model (two rigid bodies and two bending modes) are required for the transient model. The natural frequencies for the first two bending modes are $\lambda_1 = 2{,}677.8$ rad/sec $= 25{,}584$ rpm, $\lambda_2 = 6{,}091.9$ rad/sec $= 58{,}203$ rpm.

b) Speed-dependent radial stiffness coefficients for the ball bearings were calculated, and the following representation resulted.

$$k_f\,(\text{forward}) = a_{f0} + a_{f1}\dot{\phi} + a_{f2}\dot{\phi}^2 + a_{f3}\dot{\phi}^3 \tag{1}$$

where $a_{f0} = 1.799 \times 10^6$ lb/in., $a_{f1} = 105.2$ lb sec/in., $a_{f2} = -0.3124$ lb sec^2/in., $a_{f3} = 5.912 \times 10^{-5}$ lb sec^3/in.

$$k_f\,(\text{rear}) = a_{r0} + a_{r1}\dot{\phi} + a_{r2}\dot{\phi}^2 + a_{r3}\dot{\phi}^3 \tag{2}$$

where $a_{r0} = 1.321 \times 10^6$ lb/in., $a_{r1} = 81.28$ lb sec/in., $a_{r2} = -0.1857$ lb sec^2/in., $a_{r3} = 3.633 \times 10^{-5}$ lb sec^3/in. In the preceding, $\dot{\phi}$ is the rotor spin velocity in rad/sec.

c) Estimates were provided for the speed-dependent side loads developed at the boost (forward) and main stage impellers, and the following representation resulted.

$$\begin{aligned} f\,(\text{boost}) &= -c_1\dot{\phi}^2, \quad c_1 = 1.80 \times 10^{-5} \text{ lb sec}^2 \\ f\,(\text{main}) &= c_2\dot{\phi}^2, \quad c_2 = 1.53 \times 10^{-4} \text{ lb sec}^2 \end{aligned} \tag{3}$$

d) Estimates of the rotor's imbalance distribution employed here were a) 1 g in at the boost stage impeller, b) 5 g in at the main impeller, and c) 5 g in at the turbine wheels. In this study, the imbalance at the main impeller was 90° out of phase with the other two imbalances.

The assumption was made that the modal (internal) damping in the rotor was two percent of critical. The linear damping coefficients used to account for dissipation at the forward and rear bearings are: $c_f = 42$ lb sec/in., $c_r = 31$ lb sec/in. As a result of these two damping sources, the modal damping factor for the model's first natural frequency (located at approximately 13,000 rpm) is 3.5%.

The HPOTP rotating assembly is illustrated in Fig. 1. The minimum, nominal, and emergency operating speeds for this turbopump are 20,890, 29,250, and 31,160 rpm, respectively. Rocketdyne's calculated critical speeds for this turbopump lie at approximately 13,000 and 40,000 rpm. Hence, although the operating speed range of the turbopump is well removed from critical speed locations, a transition through the first (13,000 rpm) critical speed is required at startup and shutdown. The proposed system operation of the SSME is such that the critical speed transition is considerably more rapid at startup than at shutdown; hence, only the shutdown transition was examined.

The validity of a transient model is primarily established by the degree to which it correctly identifies the location and form of critical speeds. The transient model used in this study correctly identifies both the critical speed locations and the deflection mode shapes of the rotor during critical speed transitions.

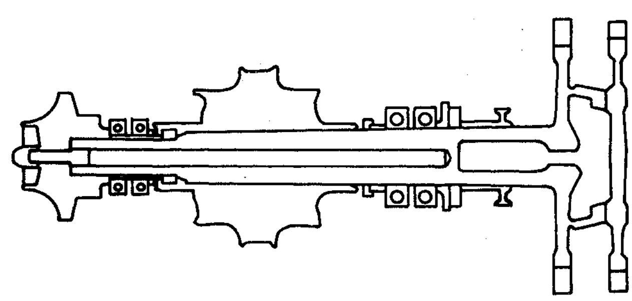

Fig. 1 HPOTP rotating assembly.

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Index categories: Liquid Rocket Engines; Structural Dynamic Analysis.

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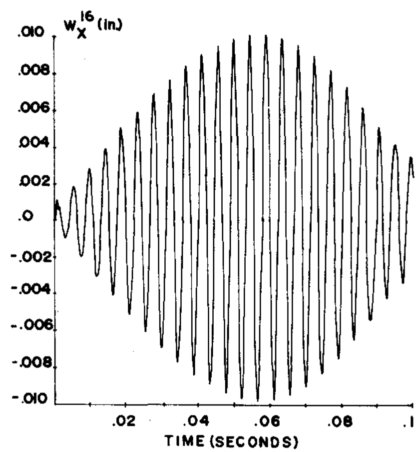

Fig. 2 ***X*-*Z* plane motion at floating ring seals.**

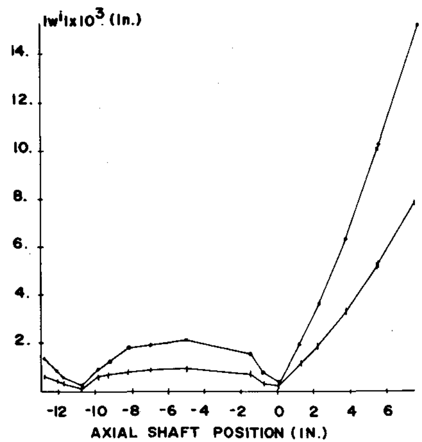

Fig. 3 Shaft deflection amplitudes during critical speed transition.

Figure 2 illustrates some of the results of a transition through the 13,000 rpm critical speed. The rotor is decelerated from 13,380 to 12,500 rpm at a nominal deceleration rate of 940 rad/sec^2. Figure 2a illustrates the transient motion of the shaft at the turbine floating ring seal location. The motion illustrated is in the X-Z plane, where Z coincides with the nominal axis of symmetry of the turbopump rotor.

Figure 2a illustrates the maximum shaft magnitudes experienced during critical-speed transition. A better appreciation of this figure is obtained by noting that the magnitude (absolute value) of deflection is plotted. The two curves represent the deflection amplitudes at two distinct times during the critical-speed transition. The forward bearings on the turbopump are located at -10.7 and -9.8 in., the rear bearings are located at 0.12 and 1.24 in., and the extreme right-hand station corresponds to a location between the turbine wheels. The figure basically illustrates that large deflections are the result of the overhung turbine wheel design, with its consequent mode shape. The deflections in the neighborhood of 5.5 in. are sufficient to cause rubbing on the turbine floating ring seals. The clearance at this location is 0.005 in. and the predicted deflections are seen to be on the order of 0.010 in. A simulation run was executed to examine the motion of the turbopump at EPL. The results indicate that shaft deflection magnitudes are generally less than 0.002 in., which does not constitute a problem.

The rubbing condition indicated for the HPOTP is presently under investigation by both NASA and Rocketdyne personnel. The HPOTP test program has been designed to determine whether or not this rubbing condition will arise, and whether such rubbing would compromise the operational integrity of the turbine floating-ring seals. Specifically, the test and development program will ascertain the operational integrity of the turbine floating-ring seals.

Reference

[1] Childs, D. W., "A Rotor-Fixed Simulation Model for Flexible Rotating Equipment," *Transactions of the ASME, Journal of Engineering for Industry,* Vol. 96, May 1974, pp. 659–669.