MODELING TECHNIQUES FOR EVALUATING THE EFFECTIVENESS OF PARTICLE DAMPING IN TURBOMACHINERY

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ABSTRACT

High power turbopumps are frequently used to supply propellants to the combustion chambers of rocket engines. Due to the high pressures and flowrates required, turbopump components are subjected to harsh environments which include dynamic excitation due to random, sine, and acoustic vibration. Additionally, fluid-induced forces can couple with the dynamics of the structure resulting in flow induced instabilities (flutter). Structural response to these forms of excitation results in reduced fatigue life and increases the likelihood of an operational failure. Particle damping has been used successfully on vibration problems in the past by increasing the damping and therefore reducing the response to acceptable levels. Empirical methods have typically been employed to evaluate the performance of the particles in reducing the structural response. This report explores the use of finite element methods to estimate the effectiveness of particle damping in a typical non-rotating turbopump component. Axisymmetric harmonic models are used to estimate the increase in modal damping produced by the addition of particles in the cavity of an axisymmetric seal. Target modes of vibration are evaluated to quantify how the effective particle damping is altered by geometry changes in the seal design. A new method to predict the performance of particle dampers is developed and shown to provide more reasonable estimates of damping.

INTRODUCTION

High cycle fatigue cracks were found on the inboard and outboard knife edge (KE) seals of the Space Shuttle Main Engine's High-Pressure Oxidizer Turbopump (HPOTP) shown in Figure 1. As a result of the investigation, it is theorized that the outboard seal, addressed herein, is prone to fluid-structure instability (flutter). Contributing to this instability is the cantilevered configuration of the seal which does not provide adequate stiffness, and the lack of structural damping. Redesign of the seal involved modifying the seal geometry and increasing its damping.

The redesigned outboard seal, shown in Figure 2, was developed to address these issues. The damping cavity shown can accommodate either a friction damper or particle damping (or both). Particle dampers dissipate energy by both impact and friction between adjacent particles and with the surrounding enclosure.

This report addresses the development of a finite element axisymmetric harmonic model to evaluate the effectiveness of particle damping as applied to the outboard KE seal.

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HPOTP Knife Edge Seals

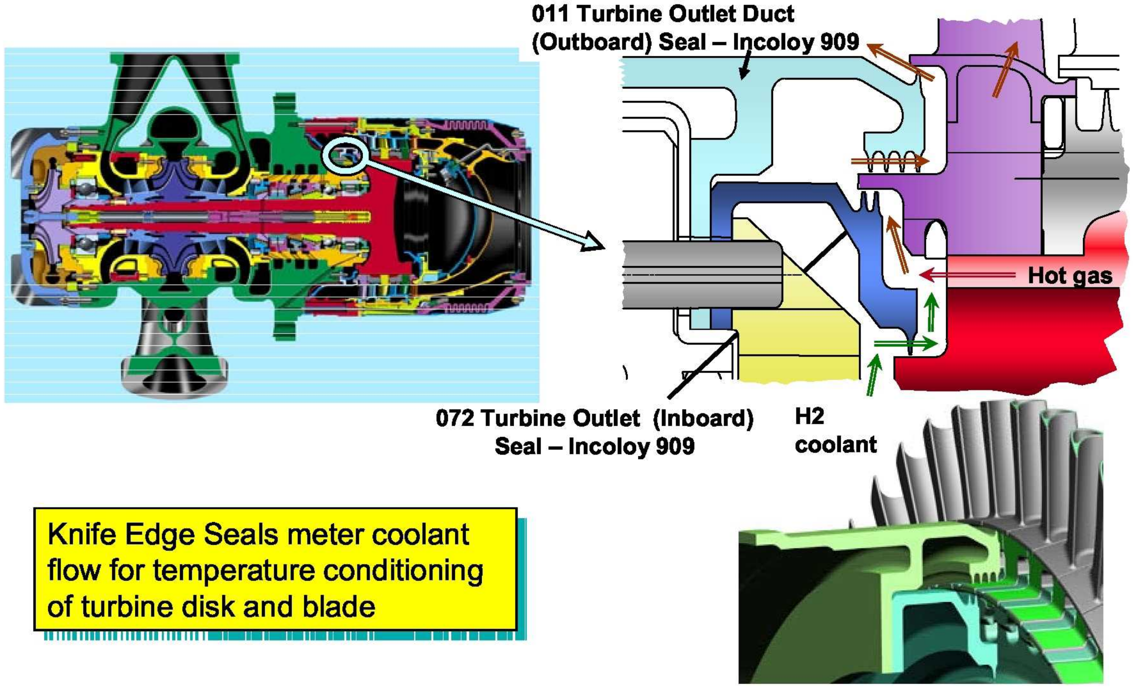


Figure 1. HPOTP Inboard and Outboard Knife Edge Seals Prior to Redesign

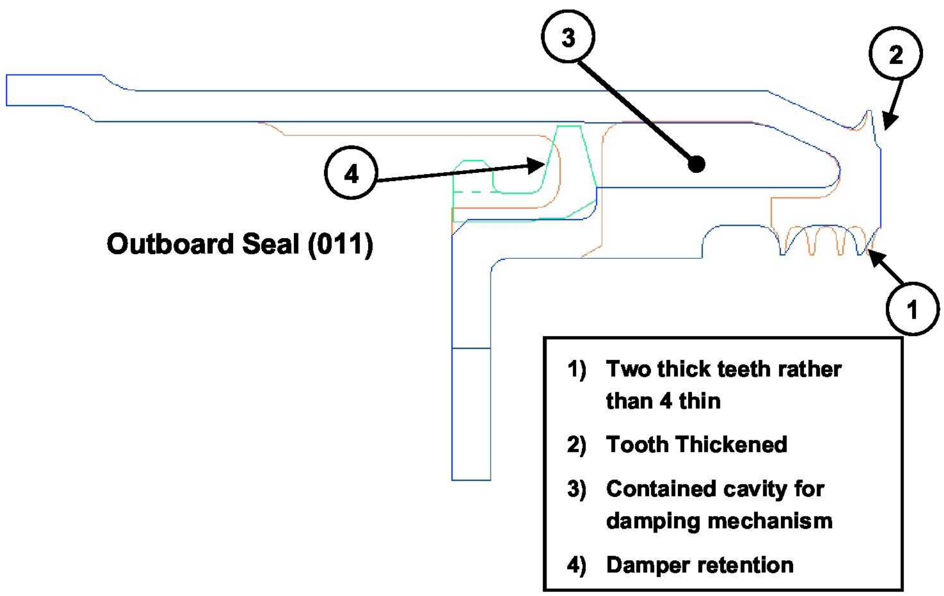


Figure 2. Outboard KE Seal Modifications Which Includes a Cavity for Addition of Particle Damping or a Leaf-spring (Friction) Type Damper.

RESULTS AND DISCUSSION

FINITE ELEMENT MODEL

The outboard KE seal is a circular structure that lends itself to analysis using axisymmetric techniques. Axisymmetric models have the advantages of being easy to build as well as providing accurate results. Figure 3 shows the axisymmetric finite element model of the outboard KE Seal. Mode shapes of this structure are characterized by displacements in the plane of Figure 3 that vary harmonically with the circumferential coordinate. In order to capture this aspect of the seal structural dynamics, axisymmetric harmonic elements were used. As opposed to a pure axisymmetric element, the unique feature of harmonic elements is that variations are allowed in the circumferential direction.

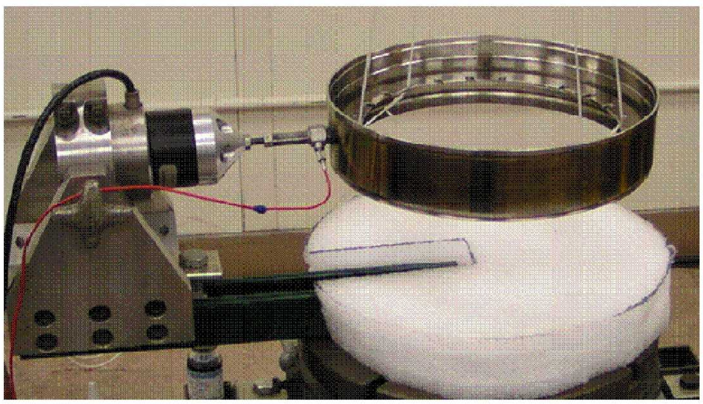
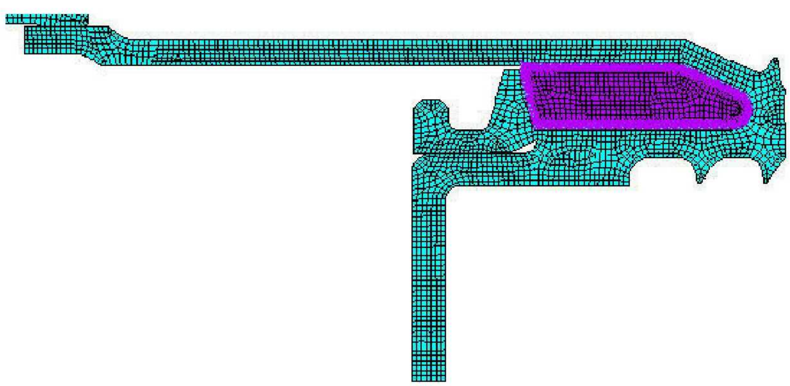


Figure 3. KE Seal Vibration Test Setup (Above) and the Axisymmetric FEM with Particle Elements Shown in Purple (Below)



In addition to the seal structure, finite elements were also used to represent the particles in the damping cavity (Figure 3). Material properties used for the damping particles were selected to avoid altering the natural frequencies and mode shapes of the seal. This required selecting a modulus of elasticity for the damping material which was low enough to avoid artificially stiffening the seal. A modulus of 5000 psi was found to work well as higher values (8000, 10000 psi) showed unacceptable increases in the natural frequencies. The density of the elements representing the particles was set to zero to avoid spurious modes of the damping material contributing to the strain energy. The mass of the particles was instead distributed around the perimeter of the damping cavity using lumped mass elements.

Experimental vibration testing of the seal involved a test fixture which was incorporated into the finite analysis. The seal model along with the test fixture model, used for this study, is presented in Figure 4. The material of the fixture is aluminum and the seal material is Incoloy 909.

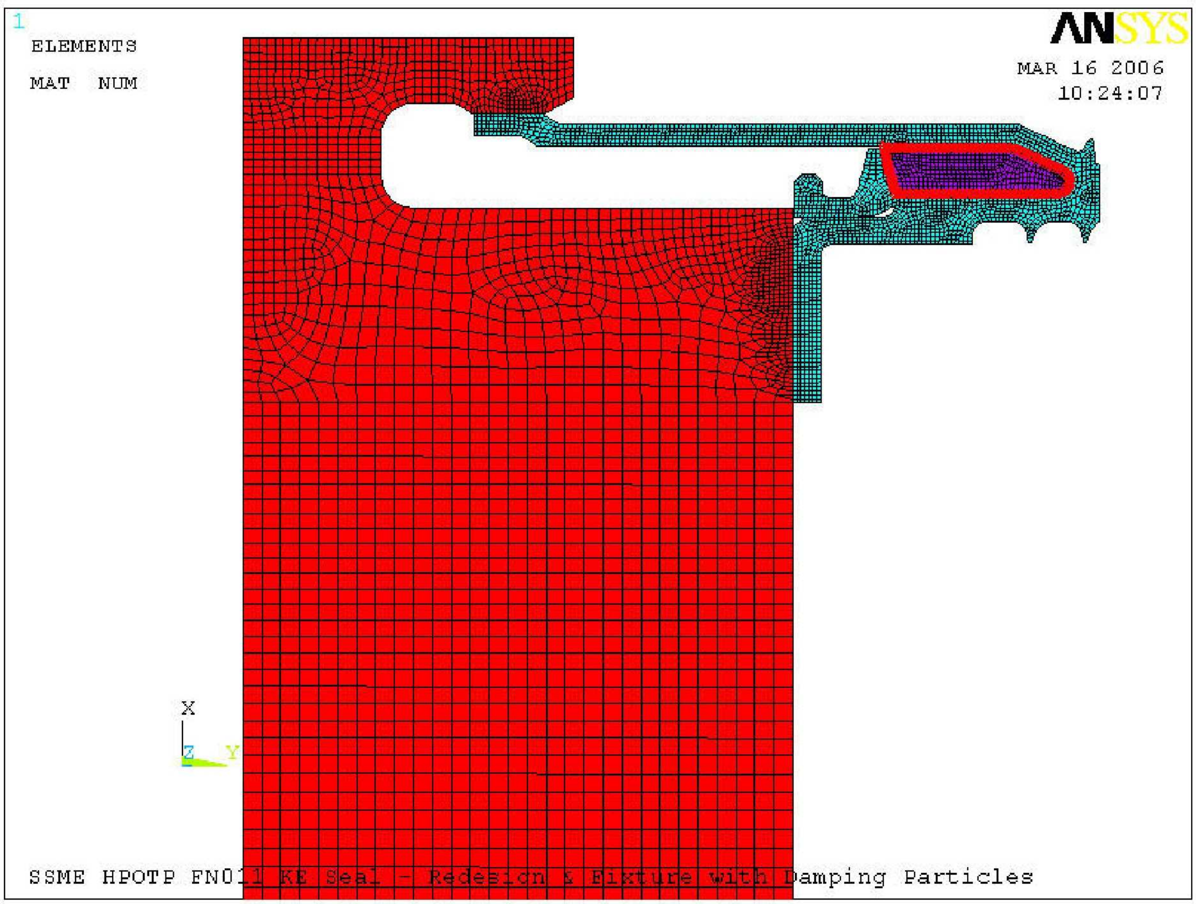


Figure 4. FEM of KE Seal and Test Fixture.

STRAIN ENERGY APPROACH FOR MODELING PARTICLE DAMPING

Previous work modeling particle damping has involved using the strain energy in the mock particle elements. For a given mode of vibration, the strain energy in the KE seal material can be easily calculated from the finite element model (U_1). The strain energy in the particle elements can also be determined (U_2) and will be a function of the assumed modulus of the damping material as well as the seal deflection. Knowing the damping ratio of the seal material and particles (ζ_1 and ζ_2 respectively), an estimate of the total or overall damping can be calculated from,

$$\zeta_T = \zeta_1 \frac{U_1}{U_T} + \zeta_2 \frac{U_2}{U_T},$$

where U_T is the sum of U_1 and U_2.

Note that this method is independent of the form or level of excitation. Since for a given mode, we are using the ratio of strain energy for a linear system, scaling due to the excitation level is not an issue for a given mode of vibration.

Figure 5 shows a plot of the total damping ratio ζ_T (Zeta Total) as a function of the damping ratio of the damping material in the cavity. Analysis of two different seal modes is shown as well as the effect of varying the modulus of the particle damping material.

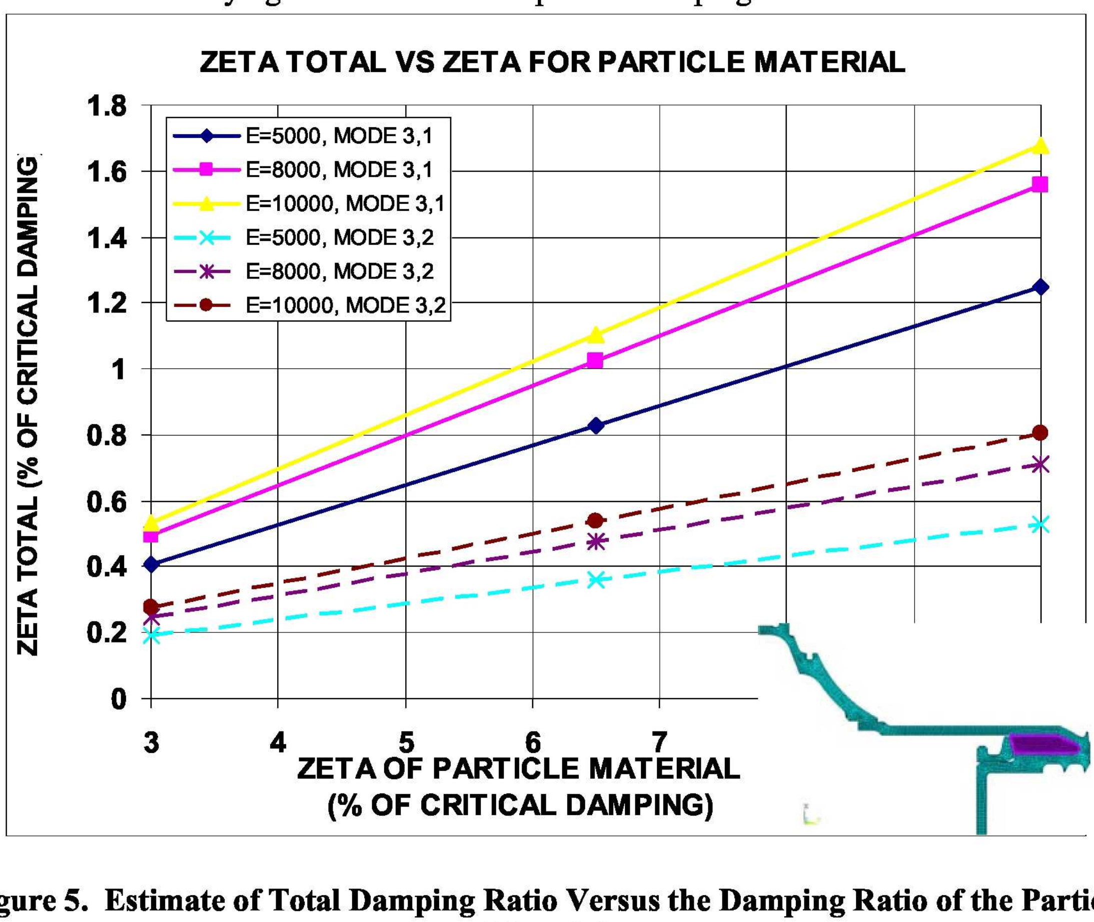


Figure 5. Estimate of Total Damping Ratio Versus the Damping Ratio of the Particle Material

Damper performance analysis using the strain energy approach outlined above was also completed using the seal/fixture model of Figure 4. The damping ratio for the material of the seal and aluminum fixture was assumed to be $\zeta_1 = 0.1$ % and the particle damping material was given a value of $\zeta_2 = 10$ %.

Results of the damper performance analysis are presented in Figure 6 for two seal modes that appear to be similar and, therefore, would be expected to have comparable total damping ratios. The 6068 Hz mode, shown in the right image of the figure, has small deformations of the damping cavity; hence the amount of strain energy stored in the damping material will be minimal. Using the strain energy approach, a calculated damping ratio of 0.149 % is found for the 6068 Hz mode compared to 1.5% for the 4517 Hz mode. This illustrates the limitation of relying only on strain energy which underestimates of the total damping for the 6068 Hz mode. Theoretical modeling of particle damping, although complex and nonlinear, indicates additional energy is dissipation as a function of the velocity.

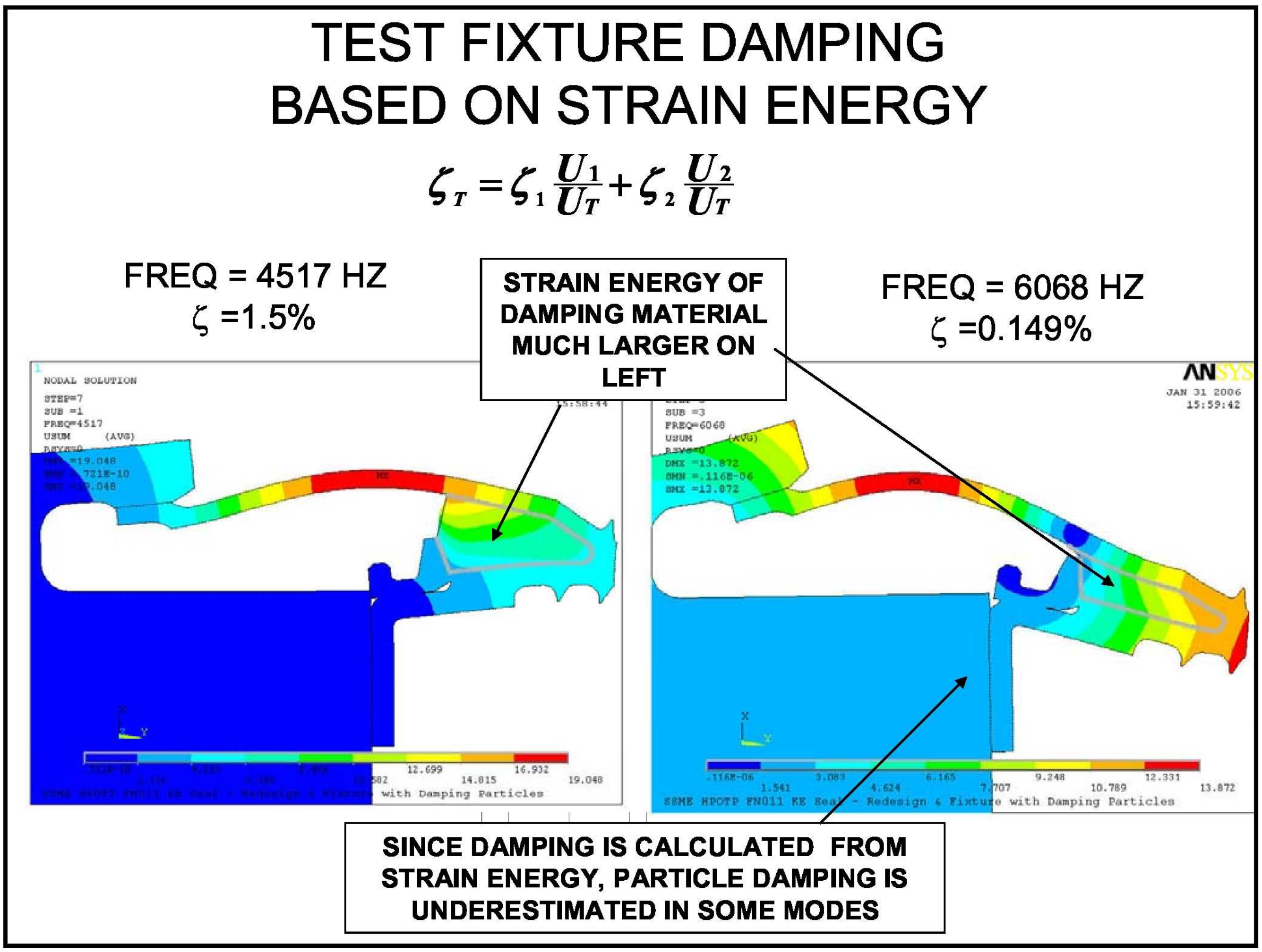


Figure 6. Estimates of Particle Damping for Two Vibration Modes of the KE Seal

STRAIN AND KINETIC ENERGY PARTICLE DAMPING FORMULATION

To address the apparent limitation of the strain energy approach, the kinetic energy of the damping material was also considered. The reasoning behind the use of kinetic energy is that during rigid body motion of the cavity, particles will rub against each other and provide damping. If the strain energy formulation is used for a cavity experiencing rigid body motion then no strain will be produced and the damping will be computed as zero. Therefore the motion of the cavity as a whole must be considered as well as the deformation of the cavity walls.

Introducing an additional constant α and forming the ratio of the kinetic energy in the damping material to the kinetic energy of the entire structure yields a new expression for the total damping of the structure.

$$\zeta_T = \zeta_1 \frac{U_1}{U_T} + \alpha \zeta_2 \frac{U_2}{U_T} + (1-\alpha)\frac{KE_2}{KE_T}$$

Here the damping in the particles can be proportioned between the strain energy and the kinetic energy by choice of the parameter α. For example, setting α to 0.5 would produce an equal contribution of strain energy and kinetic energy to the total damping. Note that the total kinetic energy in a mode, KE_T is numerically equal to the total strain energy, U_T.

Damping estimates using the strain and kinetic energy approach are shown in Figure 7 for the same two modes discussed previously. Here it is seen that the total damping ratios are more reasonable, although verification by comparison to test data has not been done.

TEST FIXTURE DAMPING BASED ON STRAIN ENERGY + KINETIC ENERGY

$$\zeta_T = \zeta_1 \frac{U_1}{U_T} + \alpha\zeta_2 \frac{U_2}{U_T} + (1-\alpha)\zeta_2 \frac{KE_2}{KE_T}$$

FREQ = 4517 HZ
α = 0.5, ζ =1.88

FREQ = 6068 HZ
α = 0.5, ζ =1.36%

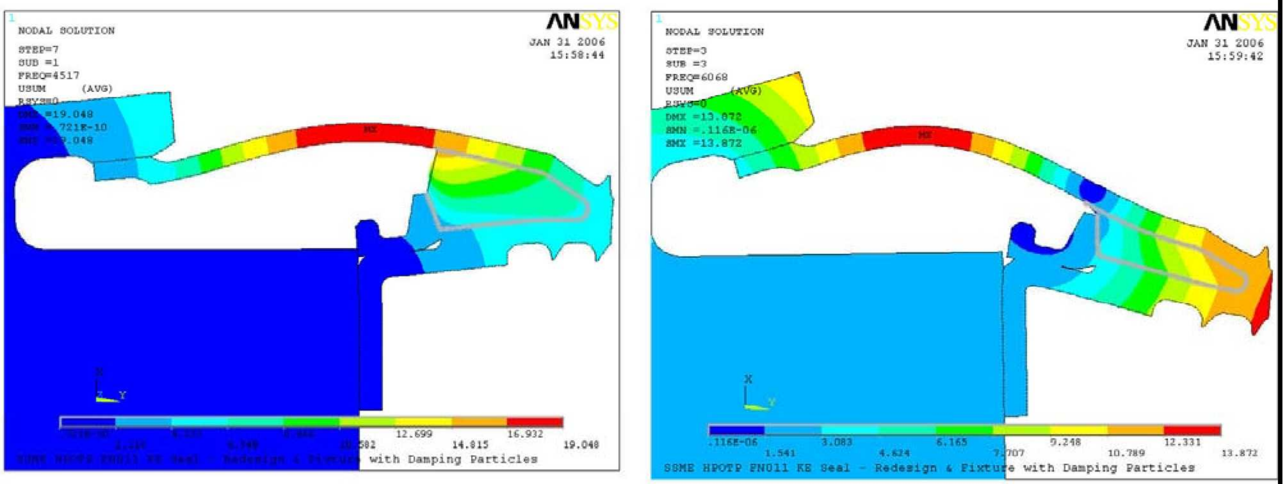


Figure 7. KE Seal Damping Estimate with Kinetic and Strain Energy of the Particles.

Damping estimates were computed for all the modes of the seal using the strain/kinetic energy approach. As a first step natural frequencies for the redesigned seal and test fixture were calculated and are plotted in Figure 8. A small shift in the natural frequencies was noted (not shown) due to the contribution of the particle damping mass.

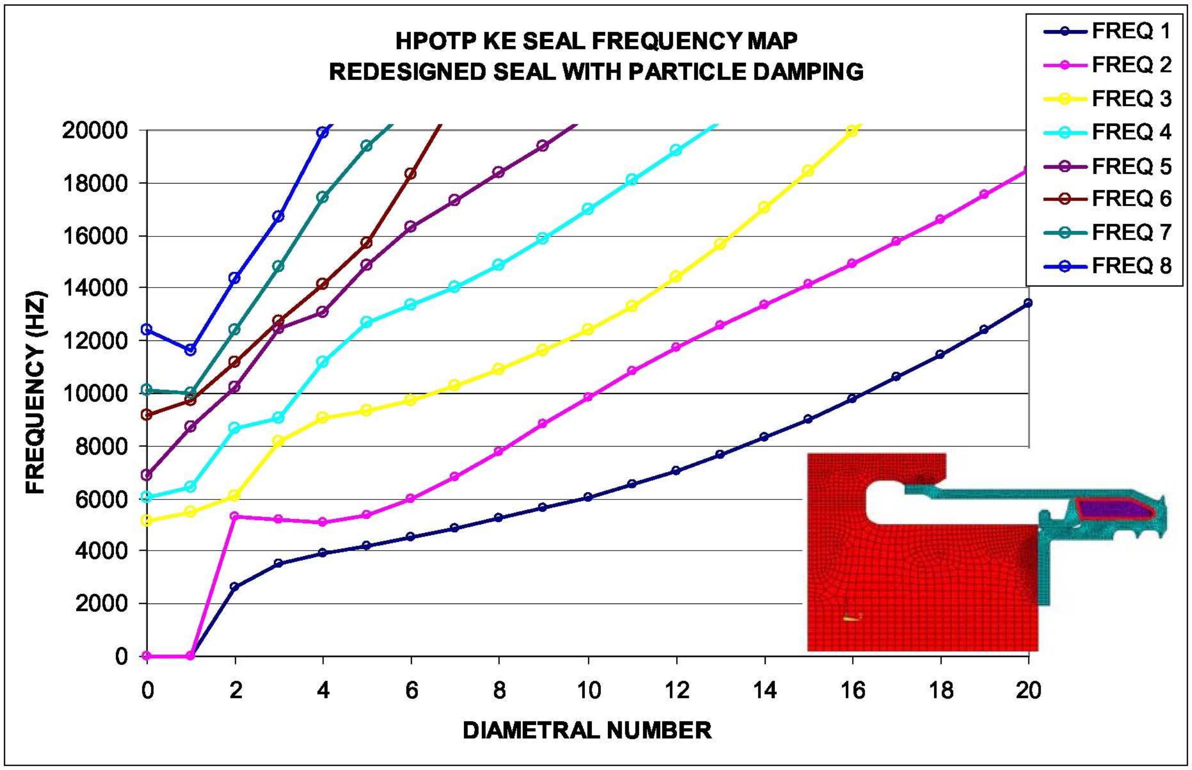


Figure 8. Natural Frequency Map of the KE Seal and Test Fixture.

Estimates of the total damping (seal damping + particle damping) were calculated for the modes shown in Figure 8, the results of which are presented in Figure 9. Material damping used for the aluminum fixture and seal material was ζ_1=0.1 % and the damping used for the particles was ζ_2 = 10 %. The strain-kinetic energy parameter, α was set equal to 0.5 for this analysis. As can be seen from the figure, total damping values ranged from a maximum of 2.38% to a minimum of 0.32%.

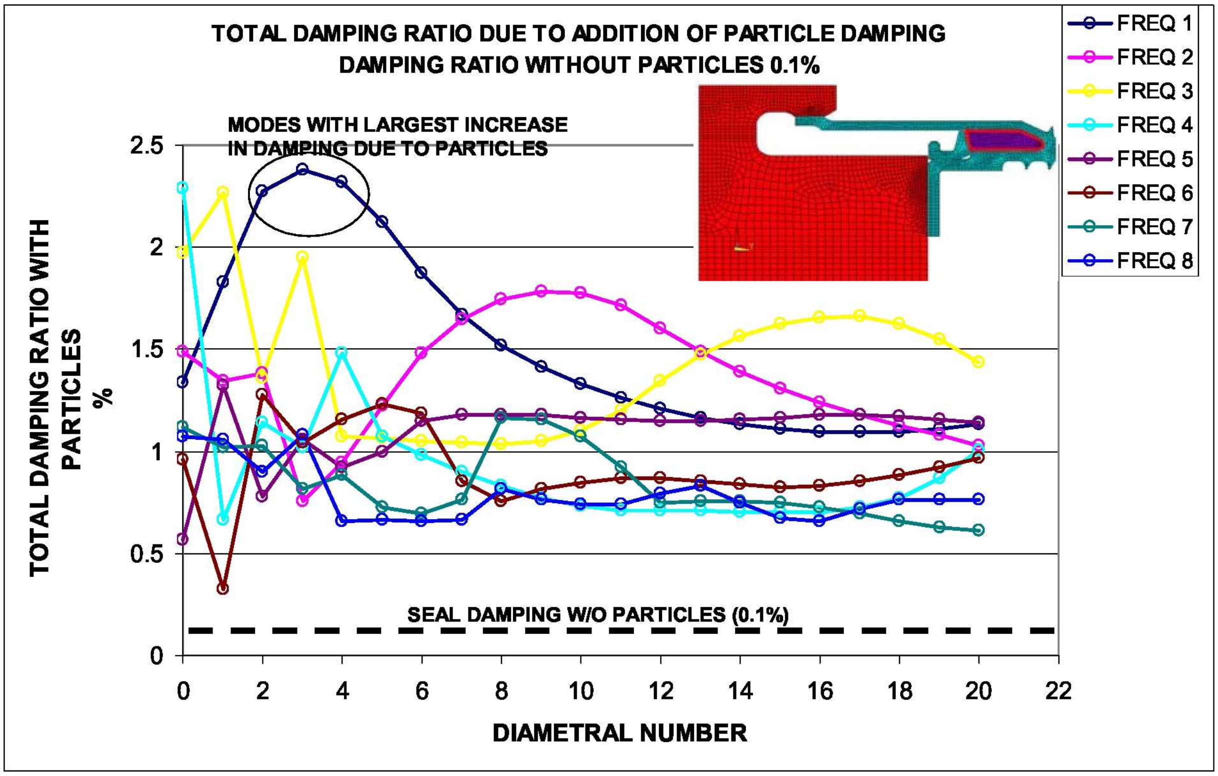


Figure 9. Calculated Total Damping Ratio Using Both the Strain and Kinetic Energy of the Particles.

SUMMARY AND CONCLUSIONS

Axisymmetric harmonic finite element models were used to evaluate the effectiveness of particle damping in the HPOTP knife edge seal. A new method for estimating the total damping in a given mode of vibration, when incorporating particle damping, has been developed that appears to be a significant improvement over previously used methods. The new method will give insight into which modes can be suppressed by the addition of particle damping and with additional test data can possibly be used to accurately predict the performance of particle dampers. The method can be applied to any axisymmetric component considered for particle damping.

FUTURE WORK

The results and conclusions of this report are based on analytical formulations that appear to give reasonable results. In order to verify the strain-kinetic energy approach it is necessary to obtain experimental data to compare with analytical predictions. It is recommended that an experimental program be formulated to measure the damping on a representative structure and compare the results with analytical formulations.

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