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November 2005

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**Abstract.** Vibration test data from an operating 2 kWe closed-Brayton-cycle (CBC) power conversion system (PCS) located at the NASA Glenn Research Center was used for a comparison with a dynamic disturbance model of the same unit. This effort was performed to show that a dynamic disturbance model of a CBC PCS can be developed that can accurately predict the torque and vibration disturbance fields of such class of rotating machinery. The ability to accurately predict these disturbance fields is required before such hardware can be confidently integrated onto a spacecraft mission. Accurate predictions of CBC disturbance fields will be used for spacecraft control/structure interaction analyses and for understanding the vibration disturbances affecting the scientific instrumentation onboard. This paper discusses how test cell data measurements for the 2 kWe CBC PCS were obtained, the development of a dynamic disturbance model used to predict the transient torque and steady state vibration fields of the same unit, and a comparison of the two sets of data.

## INTRODUCTION

The NASA Glenn Research Center conducted a first order analysis during the 2003 calendar year to determine the vibration effects that a simultaneously operating pair of rotor dynamic closed-Brayton-cycle (CBC) engines would have on a conceptual 100 kWe Nuclear Electric Propulsion (NEP) rigid spacecraft (Yu, 2003). This analysis concluded that the CBC engines would not impart significant dynamic loads (torque or vibration) to the overall spacecraft model. However, there is no test data to verify that the assumptions and methods used in the 2003 100 kWe CBC engine model were satisfactory.

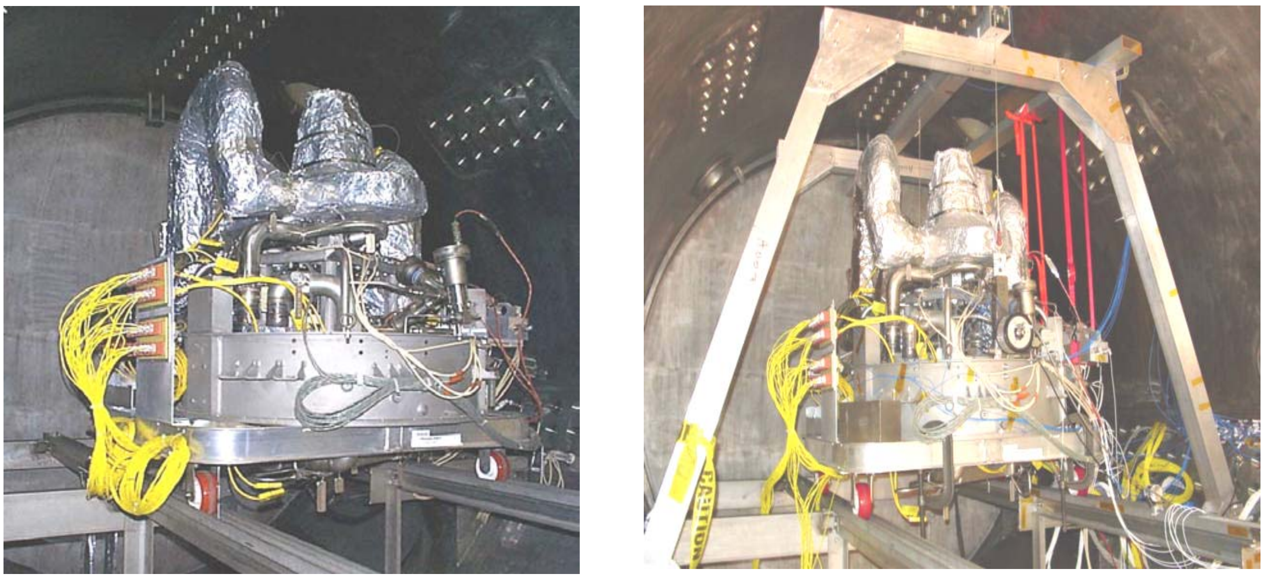
A 100 kWe CBC engine test bed unit does not exist and verifying the variables used in the 2003 analysis cannot be done until such test bed is available. However, an operating 2 kWe CBC engine test bed unit exists at NASA's Glenn Research Center. This test bed was used to validate a dynamic model of the 2 kWe CBC engine and to show that an accurate, test verified model of a 2kW unit can be developed. This correlation effort will then be used to gain confidence in modeling the Brayton engine dynamic disturbances and in the assumptions needed for an accurate 100 kWe model.

Accurate torque and vibration disturbance prediction of a closed-Brayton-cycle (CBC) Power Conversion System (PCS) is important for two reasons. First, flexible on-orbit spacecraft control/structure interaction analyses will need to be performed for an NEP spacecraft such as the Jupiter Icy Moons Orbiter (JIMO) and the CBC torque and vibration disturbances are needed for such an analysis. Second, the vibration disturbance field affecting the scientific instrumentation onboard a spacecraft such as JIMO will also have to be characterized to ensure that accurate scientific measurements can be taken during the mission. A confident understanding of how to model the CBC torque and vibration disturbances is necessary to support the implementation of such a PCS onto a spacecraft mission.

## BRAYTON 2kWe CBC HARDWARE

The Brayton 2 kWe PCU unit that was modeled and was used to acquire the test data is a modified version of a unit built for the Solar Dynamic (SD) Ground Test Demonstration (GTD). In the SD-GTD form, it compiled many hundreds of hours of operation in thermal vacuum with no failures or degradation. This hardware is shown in Figure 1a.

The operating Brayton Power Conversion Unit (BPCU) is run in a vacuum tank. Since the vacuum tank background vibration was assumed to be very high due to the operating vacuum pump systems, a vibration isolation test stand was developed for this task. The vibration isolation test stand was designed and built to utilize existing mounting pads within the vacuum tank and to suspend the CBC engine from a 2-cable suspension system in order to attenuate the high noise environmental vibration. The as tested hardware with the vibration isolation test stand is shown in Figure 1b.



(a) Before Vibration Isolation (b) After Vibration Isolation

**FIGURE 1**. Brayton Power Conversion Unit Test Hardware

## DEVELOPMENT OF A BRAYTON 2kWe CBC ANALYTICAL MODEL

The initial 2 kWe CBC engine analytical model, known as Model-1, was created with the Simulink software code and represented a generic model of the CBC engine test hardware. This model was created prior to any dynamic test data being taken on the actual test hardware. Model-1 assumed rough estimates for the bearing stiffness, rotor mass properties, rotor flexibility, rotor housing mass, and rotor housing mount stiffness. Imbalance forces (lumped at bearings in the model) due to mass imbalance and alternator electrical forces were assumed to be fixed at a conservative level for all rotor rotation speeds and only the fundamental imbalance (once-around) was modeled. Also included in this model were empirical estimates of test assembly mass, and suspension cable stiffness and tension.

The model had three rigid elements: the CBC rotor shaft, shaft housing, and the support base and a representation is shown in Figure 2. In addition, the first fundamental bending mode of the rotor shaft was included. The support base encompassed all components of the engine external to the CBC shaft housing such as the heater, heat exchanger, cooler, and various plumbing and was the heaviest part of the test assembly. The shaft housing contained the CBC shaft.

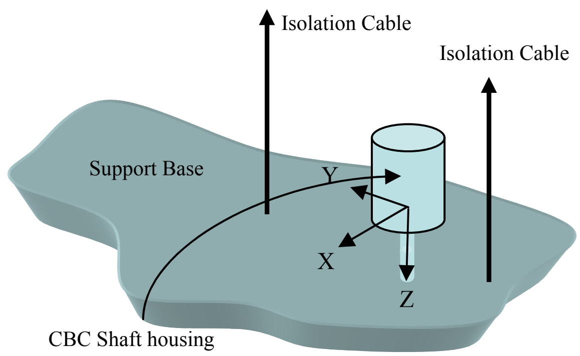



**FIGURE 2.** Brayton Power Conversion Unit, Model-1, Representation

The CBC shaft and housing is assembled from several segments held together by a threaded rod and bolts. These segments are the turbine wheel, thrust bearing, foil bearings, alternator, and compressor wheel as shown in Figure 3. This multi-body shaft is the only moving part of the engine.

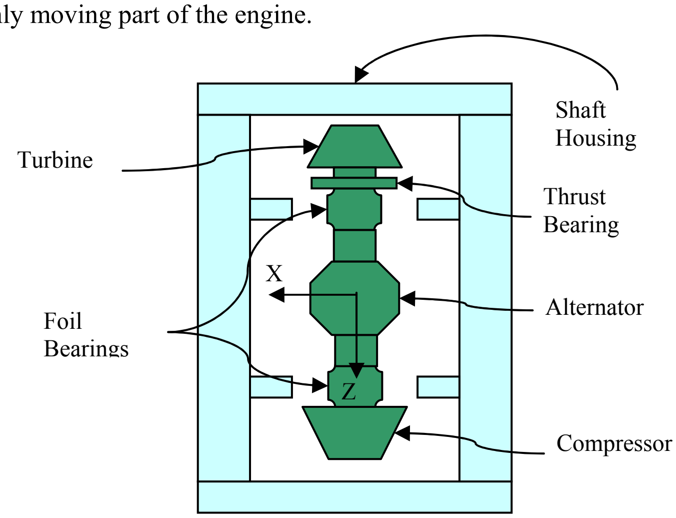



**FIGURE 3**. Brayton Power Conversion Unit, Model-1, Shaft Assembly Representation

In the Simulink model, the support base and housing subsystems contained equations of motion based on a second order damped mass-spring system. To have a reasonable representation of the disturbance created by the CBC shaft, a modified flywheel rotor model was implemented. The shaft had similar rotor dynamics as a flywheel rotor. The CBC shaft subsystem has the following dynamic features: the zero-spin dynamics, non-linear gyroscopic dynamics, mass imbalances, and bearing spring load offsets. The fundamental bending mode of the shaft as well as the speed-dependent gyroscopic couplings between tilts and shaft bending were also included in this model version.

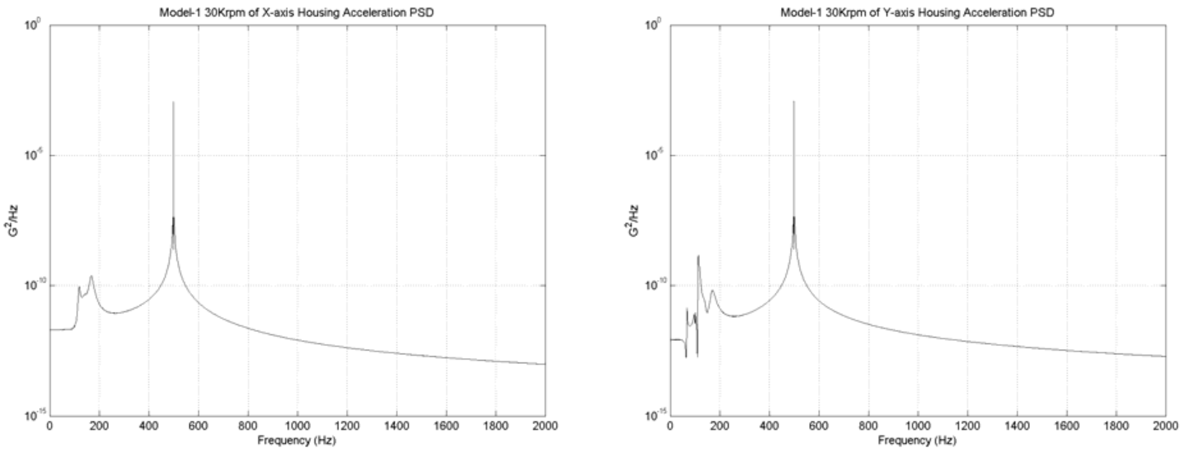
To represent the test configuration, the whole CBC engine system was modeled as a pendulum system using isolation cables as shown in Figure 2. The cable system was very generic and could be quickly modified to reflect the actual test configuration. The mass, inertia, cables tension and linear stiffness used in the Simulink model were measured values of the test assembly. Details of inputs and assumptions for this model are provided in Table 1.

Acceleration time history data predicted to occur at the geometric center of the shaft housing assembly was generated by the Model-1 Simulink model. This acceleration was the product of the predicted shaft assembly excitation acting on the test assembly model. Two rpm rates were investigated. The first was at 30,000 rpm and the second was 40,000 rpm. This data was then converted into the frequency domain in the form of power spectra density (PSD) plots. These plots are shown in Figure 4 and Figure 5.

From Figure 4a and Figure 5a, the dynamics of the Simulink model can be seen for the X direction. Below 200 Hz, response due to the shaft backward whirl, bounce mode, and forward whirl are shown. Additionally, the once per revolution excitation frequency is obviously dominant. From Figure 4b and Figure 5b, the dynamics of the Simulink model can be seen for the Y direction. Below 200 Hz, response due to the housing and mount modes, shaft backward whirl, bounce mode, and forward whirl are shown. Additionally, the once per revolution excitation frequency is obviously dominant. These predicted excitations are tabulated in Table 2. It was expected that the PSD plots from actual testing would be somewhat comparable to the predicted results. Adjustments to Model-1 were expected to be required after comparison to the test data and that by adjusting the mass properties and various stiffness properties it would be possible to match the test up to 1000Hz. The model correlation task was also expected to be an iterative effort.

**TABLE 1.** Brayton Power Conversion Unit, Model-1, Properties

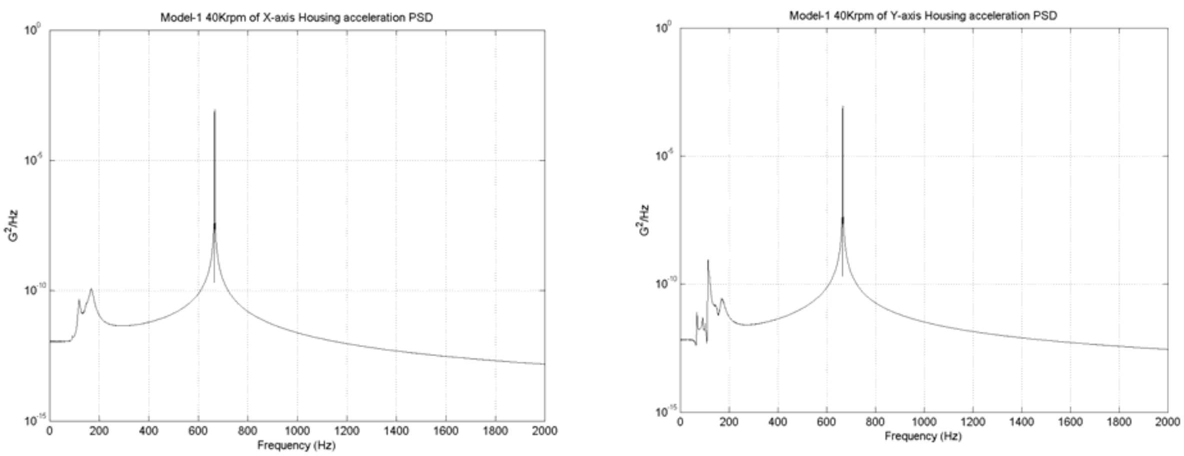
| Property | Model-1 Values |
|---|---|
| Rotor Mass | 2.627 $lb_m$ |
| Rotor Polar Moment of Inertia (Ip) | 0.7044 lbm-in$^2$ |
| Rotor Transverse Moment of Inertia about C.G. (It) | 16.303 lbm-in$^2$ |
| Flexible Shaft Modes Modeled | Yes, Fundamental Mode (1974 Hz) |
| Speed Dependant Gyro-Flex Dynamic Couplings Modeled | Yes (cross-axis rigid body coupling = $I_p/I_t$ = 0.0432) |
| Bearing Model (Thrust and Radial) | Spring Constant 3500 $lb_f$/in per Bearing For All Speeds |
| Bearing Viscous Damping (on shaft bounce and tilts) | 3% of Critical |
| Gyroscopic Destabilizing Rotating Damping (on gyroscopic motions) | 3% of Critical |
| Moment Arm From Rotor c.g. to Upper Bearing Center (towards turbine) | 1.973 in |
| Moment Arm From Rotor c.g to Lower Bearing Center (towards compressor) | 1.680 in |
| Housing and Mount Mass | 30.56 $lb_m$ |
| Housing and Mount Mass Moment of Inertia About c.g. | 11492 lb-in$^2$ for all inertia-principal axes |
| Mount Linear Stiffness | 20000 lb/in |
| Mount Torsional Stiffness | 12000000 lb-in/rad |
| Lumped Mass Imbalance and Electrical Effects (“Imbalance Forces”) | Fundamental (Once-A-Round) |
| Rotating Imbalance Force at Upper Bearing | 0.667 lb (with 135$^0$ phase offset from Lower Bearing) |
| Rotating Imbalance Force at Lower Bearing | 1.334 lb |
| Test Assembly Mass | 1031 lb |
| Test Assembly Mass Moment of Inertia about c.g. | 350000 lb-in$^2$ |
| Cable Tension | Front Cable = 745 lb<br>Rear Cable = 297 lb |
| Cable linear stiffness | 500 lbs/inch for each cable |
| Cable Length | 33.00 in |
| Number of Cables | 2 |




(a) Predictions in the X Direction

(b) Predictions in the Y Direction

**FIGURE 4.** Brayton Power Conversion Unit, Model-1, 30,000 RPM Vibration Predictions




(a) Predictions in the X Direction

(b) Predictions in the Y Direction

**FIGURE 5.** Brayton Power Conversion Unit, Model-1, 40,000 RPM Vibration Predictions

## STRUCTURAL DYNAMIC TEST MEASUREMENTS

The objective of the structural dynamic test measurements was to obtain vibration measurements for an operating 2 kWe CBC unit for comparison with analytically predicted values. This would help answer the question on what type and how much dynamic disturbances the CBC engine might impart into a spacecraft vehicle. The development of a dynamic test and measurement approach for the 2 kWe Brayton engine required a plan on where to place the measurement sensors and locations, identify and recommend test rig and vacuum chamber modifications necessary to enable accurate vibration, acquire dynamic data for several run conditions and to summarize the test data for use in correlating 2 kWe Brayton dynamic model. Because the analytical model predicted the CBC to be a very quiet machine, the Microgravity Emissions Laboratory at the NASA Glenn Research Center was given the structural dynamics test measurement task. The Microgravity Emissions Laboratory (MEL) performs testing services utilizing a highly sensitive acceleration measurement system to measure structure borne disturbances produced by the operation of electro-mechanical components, subassemblies, or assemblies such as the 2 kWe CBC. The vibration testing was conducted in Bldg 301, Vacuum Facility 6 (VF6) at the NASA Glenn Research Center.

## Instrumentation

Table 2 contains a list of the accelerometer instrumentation and location description that was recorded during the CBC testing. A total of 28 measurement channels were utilized. For this paper, accelerometer channels 13 and 14 are most important in that they were located on the CBC housing and measure accelerations at the same location of the CBC model predictions.

**TABLE 2**. Structural Dynamic Test Acceleration Measurement Channels

| Measurement Channel No. | Type (Model) | Location |
|---|---|---|
| 1 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 2 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 3 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 4 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 5 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 6 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 7 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 8 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 9 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 10 | PCB Seismic 393B05 | Brayton Mounting Ring Base |
| 13 | B&K 4370 | compr. plate 30Y- |
| 14 | B&K 4370 | compr. plate 30X- |
| 15 | PCB 357B33 | compr. bracket 20Y+ |
| 16 | PCB 357B33 | compr. bracket 20X- |
| 17 | PCB 357B33 | compr. bracket corner Z+ |
| 18 | PCB 357B33 | recup. supp. bracket 20X+ |
| 19 | PCB 357B33 | recup. supp. bracket 20Y+ |
| 20 | PCB 357B33 | recup. supp. bracket Z+ |
| 21 | PCB 357B33 | chiller off 20deg X+ |
| 22 | PCB 357B33 | chiller Z+ |
| 23 | PCB 357B33 | cable supp. near Z+ |
| 24 | PCB 357B33 | cable supp. far Z+ |
| 25 | B&K 4370 | heater back bottom X+ |
| 26 | B&K 4370 | heater back bottom Y+ |
| 27 | B&K 4370 | heater back bottom Z- |
| 28 | B&K 4370 | PTS Z- |

Data was acquired, as conditioned normalized voltage signals, using the Structural Dynamics Laboratory (SDL) multi-channel, HP E1432A measurement system. The measurement system is controlled with MTS IDEAS Test software. The test data was acquired as time history files for post processing by IDEAS.

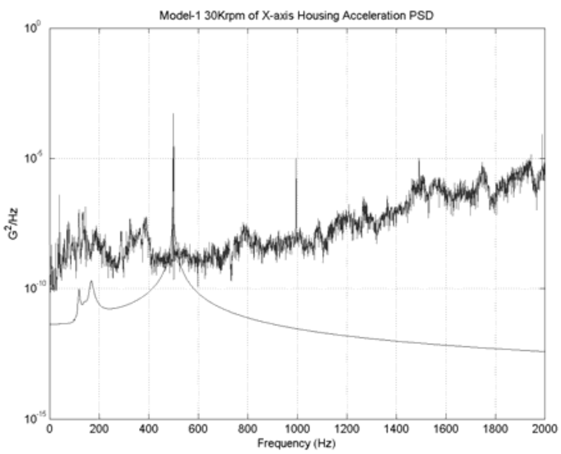
As data was acquired, the motion of the CBC on the isolation cables was as quiescent as 'reasonably' practical before test operations and measurements commenced. The ambient background was measured prior to the CBC being operated to insure that the environmental noise floor was below the expected measurement levels. Table 3 shows two of the operating CBC run conditions for which acceleration vibration data was recorded.

**TABLE 3.** Operating CBC Measurement Conditions

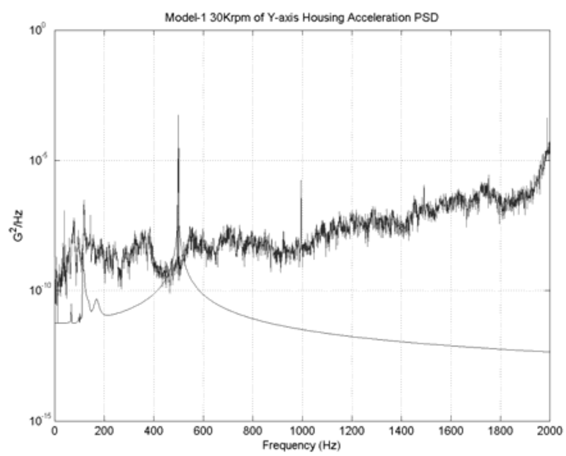
| RPM | Bandwidth (Hz) | Data Sample Rate samples/sec | Measurement Duration (min) |
|---|---|---|---|
| 30000 | 2000 | 5000 | 2 |
| 40000 | 2000 | 5000 | 2 |

## INITIAL CBC MODEL AND TEST MEASUREMENT COMPARISIONS

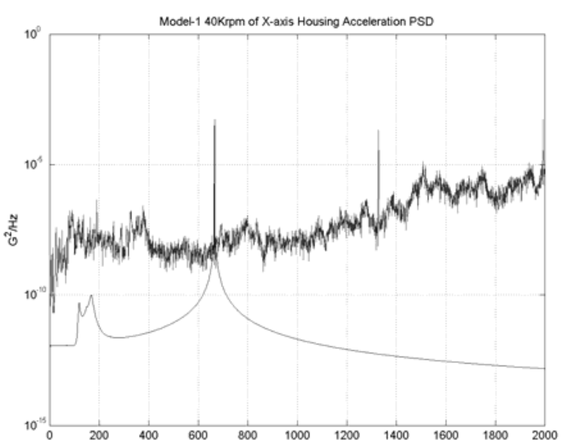
The data from measurement channels 13 (X direction) and 14 (Y direction) were post processed for both the 30000 and 40000 rpm cases into power spectra density (PSD) data. Similarly, PSD data was generated by the CBC model, Model-1, at the same locations. A comparison of the data is shown in Figure 6.



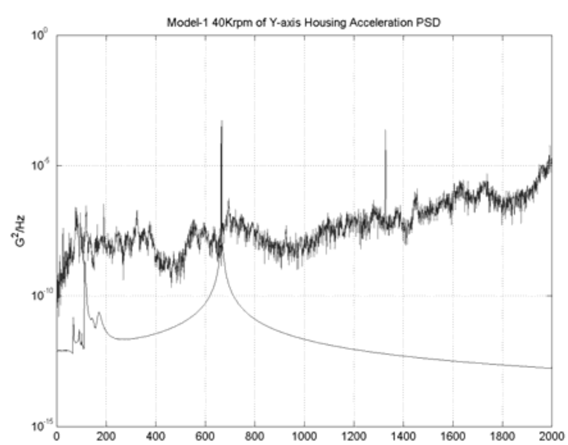

(a) 30000 RPM Comparison in the X Direction
(Black Test Data, Grey Model Prediction)




(b) 30000 RPM Comparison in the Y Direction
(Black Test Data, Grey Model Prediction)




(c) 40000 RPM Comparison in the X Direction
(Black Test Data, Grey Model Prediction)




(d) 40000 RPM Comparison in the Y Direction
(Black Test Data, Grey Model Prediction)

**FIGURE 6.** Brayton Power Conversion Unit, Model-1 and Test Data Comparison

As can be seen in Figure 6 for the X and Y directions and for the 30000 and 40000 rpm cases, Model-1 does a very good job predicting the once per revolution excitation frequency. However, the Model-1 predicted magnitudes ($G^2$/Hz) do not compare favorably with the test data values. The first observance is that the predicted imbalance force at the once per revolution frequency was too high and that the as modeled values are too conservative. Additionally, the second harmonic excitation, most likely due to the alternator hardware interactions, was not predicted by Model-1 for either rpm case. There also appears to be a random noise component across the entire spectrum. While there is yet no quantifiable explanation for this phenomenon, the leading candidate appears to be flow noise

As stated in the introduction, accurate vibration disturbance predictions of a closed-Brayton-cycle (CBC) Power Conversion System (PCS) is important for two reasons. First, flexible on-orbit spacecraft control/structure interaction analyses will need to be performed for an NEP spacecraft such as the Jupiter Icy Moons Orbiter (JIMO) and the CBC torque and vibration disturbances are needed for such an analysis. Second, the vibration disturbance field affecting the scientific instrumentation onboard a spacecraft such as JIMO will also have to be characterized to ensure that accurate scientific measurements can be taken during the mission. For both of these concerns, vibration excitations below 600 Hz are of primary concern. Therefore, the underlying Grms value of both the predicted acceleration from Model-1 and the actual measured value from the test data were calculated. This value give the structural dynamics analyst an idea of the amount of excitation energy that exists which could be of concern for the spacecraft analyses. A comparison of this value for Model-1 and the test data is shown in Table 4.

**TABLE 4.** Grms Comparisons of Model-1 and Test Data

| Operating Condition | X Direction | | Y Direction | |
|---|---|---|---|---|
| | **Model-1 Grms Value** | **Test Grms Value** | **Model-1 Grms Value** | **Test Grms Value** |
| 0-600 Hz (30000 RMP Case) | 0.026 | 0.0036 | 0.0267 | 0.0037 |
| 0-600 Hz (40000 RPM Case) | 0.00007 | 0.0033 | 0.0001 | 0.0032 |

Table 4 shows that Model-1 over predicts the excitation energy for the 30000 rpm case in both the X and Y directions due to the imbalance force being too conservative in addition to its frequency of excitation occurring below 600 Hz. Additionally, Model-1 under predicts the excitation energy for the 40000 rpm case in both the X and Y directions due to the imbalance force frequency occurring above 600 Hz. It was concluded that Model-1 needed to be adjusted.

## ADJUSTMENTS and ADDENDA to THE BRAYTON 2kWe CBC ANALYTICAL MODEL

As previously discussed, Model-1 was created prior to any dynamic test data being taken on the actual test hardware. Model-1 assumed rough estimates for the bearing stiffness, rotor mass properties, rotor flexibility, rotor housing mass, and rotor housing mount stiffness. An imbalance force due to lumped mass and alternator electrical force was assumed to be fixed at a conservative value for all rotor rotation speeds and only the fundamental imbalance (once-around) was modeled. Also included in this model were the best estimates of test assembly mass and suspension cable stiffness.

Based on the comparisons with the test data PSDs, some of the Model-1 properties were adjusted to develop Model-2. Certain items were determined to have sufficient engineering justification to modify the Model-1 values. Additionally, it was determined that the bearing clock-spring (torsional) stiffness should be included in the model in addition to a random background noise to represent the CBC engine working fluid. A comparison of the property differences between Model-1 and Model-2 is presented in Table 5.

As Table 5 shows, a majority of the changes made in Model-1 to create Model-2 were in the stiffness and damping properties. Additionally, a change was made in the amount of predicted imbalance force with the addition of the second harmonic excitation frequency. A comparison of the Model-2 predicted PSD values and the actual test data PSD values are shown in Figure 8

**TABLE 5.** Comparison of Model-1 and Model-2 Properties

| Property | Model-1 | Model-2 |
|---|---|---|
| Rotor Mass | 2.627 lbm | 2.627 lbm |
| Rotor Polar Moment of Inertia (Ip) | 0.7044 lbm-in$^2$ | 0.7044 lbm-in$^2$ |
| Rotor Transverse Moment of Inertia about C.G. (It) | 16.303 lbm-in$^2$ | 16.303 lbm-in$^2$ |
| Flexible Shaft Modes Modeled | Fundamental mode (1974 Hz) | Fundamental Mode (1974 Hz) |
| Speed Dependant Gyro-Flex Dynamic Couplings Modeled | Yes (cross-axis rigid body coupling = $I_p/I_t$ = 0.0432) | Yes (cross-axis rigid body coupling = $I_p/I_t$ = 0.0648) |
| Bearing Clock-Spring (torsional) Stiffness | Not Included | 2900 lb-in/rad |
| Bearing Model (Thrust and Radial) | As Spring Constant of 3500 $lb_f$/in per bearing for all speeds | Spring Mass System Representing Airfoil/Bumpfoil/Bumpfoil Base/Base Support and Changes with Rotor Speed* |
| Bearing Viscous Damping (on shaft bounce and tilts) | 3% of critical | 3% of critical |
| Gyroscopic Destabilizing Rotating Damping | 3% of critical | 3% of critical |
| Moment Arm from Rotor C.G.. to Upper Bearing Center (towards turbine) | 1.973 in | 1.973 in |
| Moment Arm from Rotor C.G. to Lower Bearing Center (towards compressor) | 1.680 in | 1.680 in |
| Housing and Mount Mass | 30.56 $lb_m$ | 30.56 $lb_m$ |
| Housing and Mount Mass Moment of Inertia about c.g. | 11492 lb-in$^2$ | 11492 lb-in$^2$ |
| Mount Linear Stiffness | 20000 lb/in | 10000 lb/in |
| Mount Torsional Stiffness | 12000000 lb-in/rad | 6000000 lb-in/rad |
| Lumped Mass Imbalance and Electrical Effects ("Imbalance Forces") | Fundamental (Once-A-Round) | Once -A-Round and Second Harmonic |
| Rotating "Imbalance Force" at Upper Bearing | 0.667 lb (with 135$^0$ phase offset from Lower Bearing) | 0.200 lb (with 135$^0$ phase offset from Lower Bearing) |
| Rotating "Imbalance Force" at Lower Bearing | 1.334 lb | 0.355 lb |
| Test Assembly Mass | 1031 lb | 1031 lb |
| Test Assembly Mass Moment of Inertia about c.g. | 350000 lb-in$^2$ | 350000 lb-in$^2$ |
| Cable Tension | Front Cable = 745 lb<br>Rear Cable 297 lb | Front Cable = 745 lb<br>Rear Cable 297 lb |
| Cable Linear Stiffness | 500 lbs/inch for each cable | 500 lbs/inch for each cable |
| Cable Length | 33.00 in | 33.00 in |
| Number of Cables | 2 | 2 |
| Random Background (internal) Disturbances | None | Yes-white noise force spectrum acting internally at each bearing (Frms = 0.016 lb for 0 to 2500 Hz) |

* See Figure 7

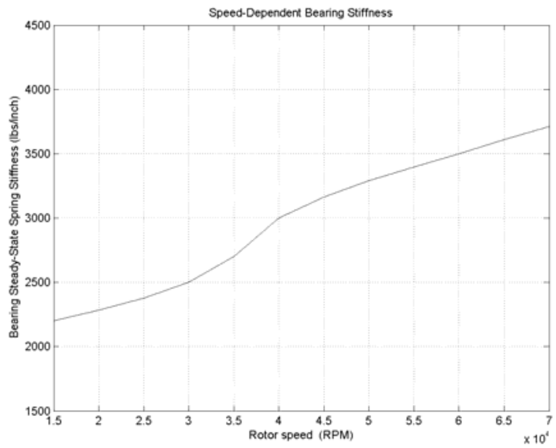
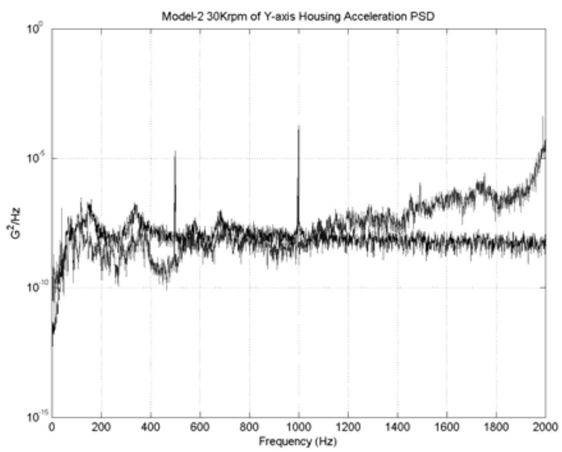
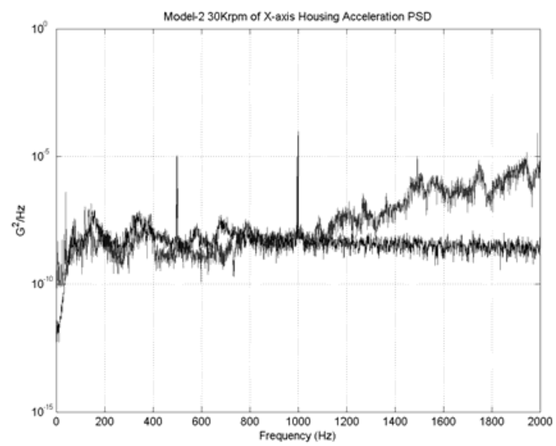


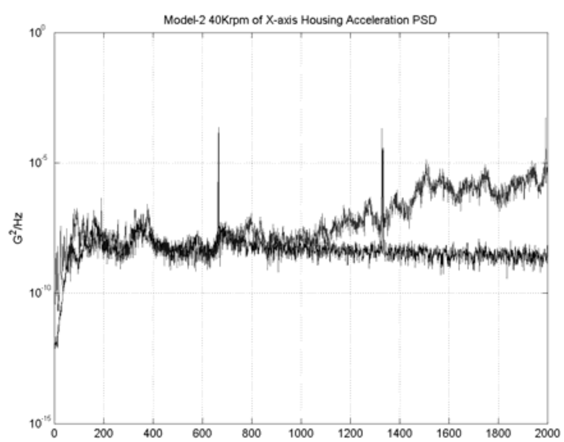

**FIGURE 7.** Brayton Power Conversion Unit Model-2 Bearing Stiffness Model



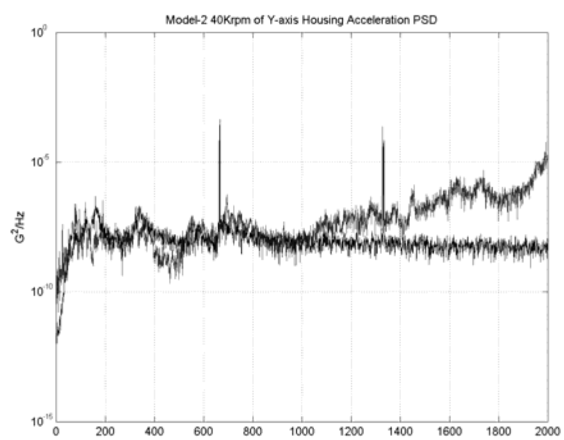

(a) 30000 RPM Comparison in the X Direction
(Black Test Data, Grey Model Prediction)




(b) 30000 RPM Comparison in the Y Direction
(Black Test Data, Grey Model Prediction)




(c) 40000 RPM Comparison in the X Direction
(Black Test Data, Grey Model Prediction)




(d) 40000 RPM Comparison in the Y Direction
(Black Test Data, Grey Model Prediction)

**FIGURE 8.** Brayton Power Conversion Unit, Model-2 and Test Data Comparison

As can be seen in Figure 8 for the X and Y directions and for the 30000 and 40000 rpm cases, Model-2 does a very good job predicting the once per revolution excitation frequency in addition to the second harmonic. However, the Model-2 predicted magnitudes ($G^2$/Hz) do not compare favorably with the test data values for the 30000 rpm case at these frequencies. This is because the imbalance force is fixed in the model (i.e. is not speed dependent) and is more representative of the force that would be present at the 40000 rpm condition. It was also observed that the predicted imbalance force at the once per revolution frequency was still slightly conservative for the 40000 rpm X direction PSD value. The other frequency bands show a better comparison to the random noise background noise suspected to be the result of fluid dynamic excitation of the CBC engine working internal fluids. It is also believed that the excitation energy in the 300-600 Hz band was a result of CBC airfoil-bearings dynamics and should be included in the model. The airfoil-bearings dynamics (as a spring-mass system) were added to the model to capture the effects of fluid dynamics disturbance at least up to 600 Hz. All other frequency bands for which there is a magnitude difference is expected to be the result of structural modes of the vibration test stand structure and the CBC working fluid random excitation at higher frequencies.

However, as stated previously, accurate vibration disturbance predictions at least up to 600 Hz for a closed-Brayton-cycle (CBC) Power Conversion System (PCS) is important for flexible on-orbit spacecraft control/structure interaction analyses and for understanding the vibration disturbance field affecting the scientific instrumentation onboard a spacecraft such as JIMO. Therefore, the underlying Grms value of both the predicted acceleration from Model-2 and the actual measured value from the test data were calculated. This value give the structural dynamics analyst an idea of the amount of excitation energy that exists which could be of concern for the spacecraft analyses. A comparison of this value for Model-2 and the test data is shown in Table 6.

**TABLE 6.** Grms Comparisons of Model-2 and Test Data

| **Operating Condition** | **X Direction** | | **Y Direction** | |
|---|---|---|---|---|
| | **Model-2 Grms Value** | **Test Grms Value** | **Model-2 Grms Value** | **Test Grms Value** |
| 0-1500 Hz (30000 RMP Case) | 0.012 | 0.010 | 0.016 | 0.007 |
| 0-1500 Hz (40000 RPM Case) | 0.018 | 0.019 | 0.025 | 0.020 |

Table 6 shows that Model-2 still over predicts the excitation energy for the 30000 rpm case in both the X and Y directions due to the imbalance force being too conservative (i.e. is not speed dependent) in addition to its frequency of excitation occurring below 1500 Hz. Additionally, Model-2 shows very good comparison for the excitation energy for the 40000 rpm case in both the X and Y directions. The results from Model-2 show that if the model predictions were used in spacecraft analyses, an overly conservative response would be expected for the 30000 rpm run condition, but a fairly accurate spacecraft response would be expected for the 40000 operating condition. Further improvements such as speed dependent imbalance force and improved CBC engine working fluid random vibration excitation could be made to Model-2 in the future.

It is also important to recognize the low vibration magnitudes reported in Table 4 and Table 6. The operating 2 kWe CBC engine was predicted to be a quiet machine, and this was proven with test data. It is expected that in any future spacecraft analyses, the predicted response of the spacecraft system to the disturbance caused by an operating CBC engine would be negligible regarding spacecraft control authority and effects on scientific instrumentation.

## CONCLUSION

Analytical models of an operating 2kWe CBC engine were created and refined based on operational test data. Through a single refinement stage, the analytical model showed very good comparison to the test data vibration energy for the specific imbalance rpm case modeled. There are still several areas in which the existing model could be improved. These are speed dependent imbalance force, high-frequency airfoil-bearings dynamics, and improved CBC engine working fluid random vibration excitation. These additional improvements were not implemented for this paper but will be pursued in the future in order to gain possible additional understanding of how to model a working CBC engine for spaceflight application.

Note that, the effects of rotating machinery and airfoil-bearing dynamics were conservatively modeled (by necessity), because these internal disturbances alone already manifested the vibration level of the actual system. Even on the frequency band of 0 to 1500Hz, the actual system vibration must have been the result of many more complex disturbances

Furthermore, the final model developed predicted very minimal excitation energy in the frequency band of interest, 0-600Hz, which was validated by test data. This is important information in that the understanding of the on-orbit spacecraft control/structure interactions for a large, flexible spacecraft such as the Jupiter Icy Moons Orbiter (JIMO) will be required and the CBC disturbances will be of primary concern. With such expected low disturbances coming from an operating 2 kWe CBC engine, issues with spacecraft control/structure interactions are not expected. However, this still requires validation by adapting this model to a model of the full-scale system (50-kWe unit). Additionally the vibration disturbance field from an operating CBC engine is not expected to negatively affect the scientific instrumentation onboard a spacecraft.

The performance of the CBC engine models at other operating condition will also need to be investigated. Of particular interest will be transient torque conditions and emergency shutdown transients.

## NOMENCLATURE

| | |
|---|---|
| CBC | Closed Brayton Cycle |
| GTD | Ground Test Demonstrator |
| JIMO | Jupiter Icy Moons Orbiter |
| kWe | kilo Watt electric |
| MEL | Microgravity Emissions Laboratory (at NASA Glenn Research Center) |
| NEP | Nuclear Electric Propulsion |
| PCS | Power Conversion System |
| PSD | Power Spectral Density |
| RPM | Rotations Per Minute |
| SD | Solar Dynamic |
| SDL | Structural Dynamics Laboratory (at NASA Glenn Research Center) |

## REFERENCES

Yu, A., “Preliminary Torque Analysis of Closed Brayton Cycle Rotor Dynamics on a Conceptual Nuclear Electric Propulsion Vehicle”, NASA GRC EDAD Internal Task Report #69037; January 21, 2003.

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Vibration test data from an operating 2 kWe closed-Brayton-cycle (CBC) power conversion system (PCS) located at the NASA Glenn Research Center was used for a comparison with a dynamic disturbance model of the same unit. This effort was performed to show that a dynamic disturbance model of a CBC PCS can be developed that can accurately predict the torque and vibration disturbance fields of such class of rotating machinery. The ability to accurately predict these disturbance fields is required before such hardware can be confidently integrated onto a spacecraft mission. Accurate predictions of CBC disturbance fields will be used for spacecraft control/structure interaction analyses and for understanding the vibration disturbances affecting the scientific instrumentation onboard. This paper discusses how test cell data measurements for the 2 kWe CBC PCS were obtained, the development of a dynamic disturbance model used to predict the transient torque and steady state vibration fields of the same unit, and a comparison of the two sets of data.

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