DESIGN AND SHOP TESTING OF A 165KW CRYOGENIC EXPANDER/GENERATOR ON MAGNETIC BEARINGS

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ABSTRACT

A low cost cryogenic expander system that incorporates a high performance, high-speed permanent magnet generator and low loss magnetic bearings is described. The expander system will be used in industrial air separation plants that produce gases such as nitrogen, oxygen, and argon using a cryogenic distillation process. Magnetic bearings enable the key advantage of a completely oil-free system that reduces the risk of accidental contamination of the process. A single generator/bearing module was designed to accommodate a wide range of expander aerodynamic stages that operate at different speeds and power levels ranging from 9,000 RPM to 30,000 RPM and 10 kW to 165 kW. Shop testing of the unit to 30,000 rpm was completed in December 2003. The prototype unit will undergo additional loop testing at the system manufacturer's facility and is scheduled for field installation in mid 2004. System design and test results are presented.

INTRODUCTION

Refrigeration in industrial air separation plants is typically provided by expanding a pressurized process gas across a high-speed radial expansion turbine. Outlet temperatures as low as –320 deg F are needed to produce gases such as nitrogen, oxygen, and argon using a cryogenic distillation process. The stored energy in the process gas is converted into shaft energy by the expansion turbine. In the past, economics required shaft energy below 100 kW to be consumed by a dissipative system such as an atmospheric blower stage or oil-loader integral to the expander shaft. Today, this wasted energy can be converted into electricity with a direct-drive generator connected to the plant grid, reducing plant power consumption. With the expander wheel directly mounted onto the generator shaft, costly gearbox, maintenance, and support systems are eliminated. Furthermore, by incorporating magnetic bearings, the system becomes a completely oil-free system that reduces the risk of accidental contamination of the process. A single generator module has been designed to accommodate a wide range of expander aerodynamic stages that operate at different speeds and power levels ranging from 9,000 RPM to 30,000 RPM and 10 kW to 165 kW.

The expander/generator module is shown in Figure 1 with the nominal 114 mm (4.5 inch) expander wheel used for the initial prototype application. Up to 110 kW of refrigeration power will be consumed by the prototype expander which will be placed into service in an air separation plant in Europe. The expander/generator utilizes a high efficiency permanent magnet generator and magnetic bearings contained within a high-pressure-capable housing. The generator has a conventional tooth type stator together with a two pole, high strength steel sleeved rotor assembly. This arrangement is configured for minimum rotor losses and the cold temperatures the rotor may see in the cryogenic expansion process. The expander/generator rotor is supported by two radial permanent magnet bias magnetic bearings and an anisotropic (load capacity in one direction is larger than the other direction) active axial magnetic bearing. The generator output is connected to a Power Conditioning Unit (PCU) to condition and regulate power back to the plant grid. A System Controller provides user interface and communicates with the magnetic bearing control box, the PCU, and the plant DCS system. The System Controller, PCU, and magnetic bearing controller are all contained in the same cabinet.

A primary design objective was to apply the same "plug-in" generator unit across a broad range of applications. The application range includes power output from 10 kW to 165 kW and inlet pressure up to 5.0 MPa (up to 720 psig). Expander wheel sizes range from nominal diameters of 89 mm (3.5 in) to 229 mm (9 in) and 0.18 kg (0.4 lbm) to 2.32 kg (5.1 lbm). As the process conditions vary from application-to-application, there are two significant impacts on the magnetic bearing system design: 1) the size and therefore inertia of the expander wheel changes, significantly altering the lowest shaft bending mode frequencies, and 2) the aerodynamic loads change, particularly static axial load. The expected static aerodynamic load varies from 445 – 5350 N (100 – 1200 lbf) so the thrust bearing was sized for the max static load plus 2453 N (550 lbf)

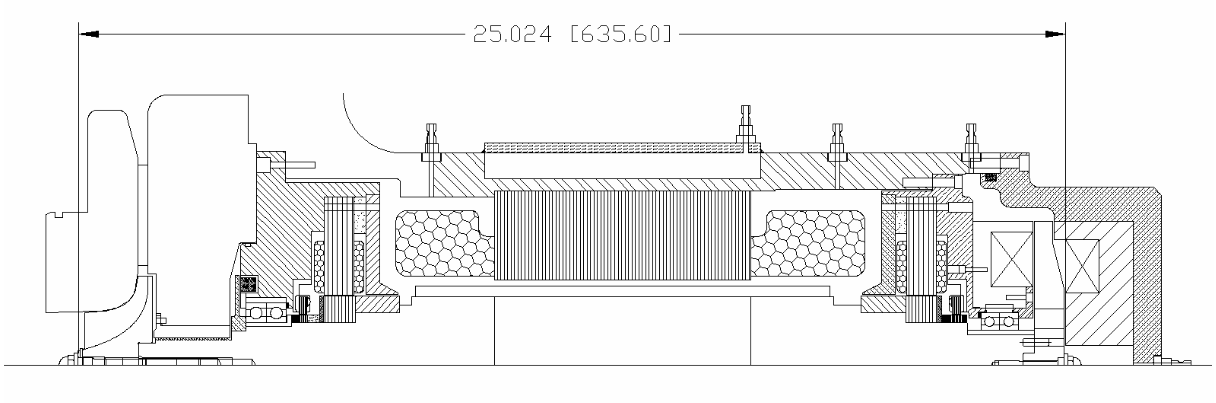

FIGURE 1. Cross-section of expander generator with nominal 114 mm (4.5 in) wheel.

to cover transient loads and design margin. The range in wheel inertia characteristics was more difficult to address with a single solution because the mass of the overhung expander wheel ranges from an insignificant 0.8% of the mass of the rotating assembly to about 7.5% of the rotating assembly mass. Therefore, considerable attention was paid to pushing the first rotor bending mode as high as possible to allow a control solution that could be applied across all units.

MAGNETIC BEARING ACTUATORS

The two radial bearings are homopolar, permanent magnet-bias bearings with design load capacity of 780 N (175 lb). Each bearing has one active control pole and one *dead pole*. The dead pole is used to close the flux path. Filatov [1] discusses the design and testing of these bearings in detail.

The axial bearing is an electromagnetic bias bearing with asymmetric force characteristics. It has design capacity of 7803 N (1750 lb) in the outboard direction and 3338 N (750 lb) in the inboard direction. The outboard load capacity includes the peak expected static pressure load plus a transient load. The load capacity in the inboard direction covers a smaller static load and the same transient capacity. Because the inboard direction load requirement was smaller, a smaller pole area could be used, allowing placement of the backup bearings under the inboard thrust stator. This configuration resulted in a shorter rotating assembly length, increasing the bending frequencies of the machine, simplifying the control design. The design and testing of the axial bearing is also discussed in [1].

BACKUP BEARING DESIGN

The backup bearing system consists of two pairs of duplexed, angular contact ball bearings. The bearing pair on the expander end of the machine provides backup in both thrust and radial directions (Figure 2). The races are conventional SAE 52100 steel and the balls are SiN3. Brass landing sleeves on the rotor provide a non-sparking touchdown surface on the shaft, a requirement for some of the potential applications of the generator/expander unit. The radial backup bearing clearance is 0.09 – 0.11 mm (0.0035-0.0042 in).

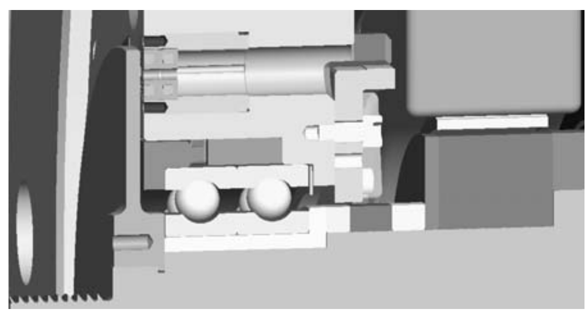

FIGURE 2. Cross-section showing radial/thrust backup bearing.

Both backup bearing pairs are mounted in a resilient mount which reduces the support stiffness to 1.0E7 N/m (57,000 lbf/in). The mount is intended to serve several purposes: 1) reduce the synchronous reaction forces during a critical speed traverse (by lowering the forward natural frequency), 2) reduce the impact force during a drop down event, and 3) reduce the potential backward whirl frequency (by lowering the lowest backward mode natural frequency). The resilient mount also contributes light damping through slipping at the interfaces of the mount and the housing. The lower limit of backup bearing support stiffness is determined by the maximum radial displacement allowed by critical clearances in the machine. This is particularly important in turbomachinery where it is desired to keep design clearances between the impeller and shroud as small as possible. In this machine, the allowable travel on the resilient mount is restricted to only 0.025 – 0.051 mm (0.001–0.002 in) radial. Beyond that the resilient mount is bypassed by contact between the bearing outer race and the housing, resulting in a much stiffer support stiffness due to the duplex bearing pair.

ROTORDYNAMICS AND MAGNETIC BEARING CONTROLS

As mentioned above, the same generator unit will be used with a wide variety of expander wheels. The overhung mass of the largest expander wheel is 5.1 lbm or about 7.7% of the mass of the rotating assembly, whereas the smallest wheel is about 0.4 lbm or less than 0.6% of the rotating assembly mass. This means that there will be a substantial difference in the first rotor bending mode across the application range of the machine. To mitigate this, much attention was paid to keeping the shaft as short and stiff as possible so that the machine could be operated below the first bending mode in all configurations. The predicted first forward bending mode frequencies of a subset of the possible expander wheel sizes are given in Table 1 to illustrate the results. The normal operating speed is 30,000 rpm for the lighter wheels and 25,000 rpm for the heaviest wheel (W5). Thus, the machine can be operated subcritical for all planned applications.

Table 1. Expander Wheel Designations

Designation	**Wheel Mass kg (lbm)**	**Polar Inertia $kg\text{-}m^2$ ($lbm\text{-}in^2$)**	**1st Bending Mode Frequency @ 30,000 rpm (cpm)**
W1	0.32 (0.7)	2.93 E-4 (1.0)	50,300
W2	0.68 (1.5)	1.47 E-3 (5.0)	46,800
W3	1.14 (2.5)	2.64 E-3 (9.0)	41,800
W4	1.59 (3.5)	3.28 E-3 (11.2)	37,700
W5	2.32 (5.1)	9.21 E-3 (31.4)	35,000

Figure 3 compares plant transfer functions (non-expander end bearing) for three different expander wheel sizes at 0 rpm and at 30,000 rpm. These three sizes represent the small, mid, and heavy end of the spectrum of wheel sizes. It is important to note that gain of the bending modes for all three wheels is at least 20 dB down from the DC value throughout the speed range. This makes it easier to gain stabilize the modes, allowing a relatively simple compensator to work for a broad range of wheel sizes.

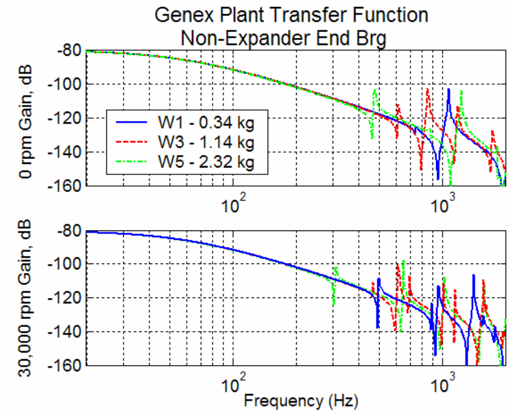

FIGURE 3. Comparison of predicted plant transfer functions for three different expander wheels.

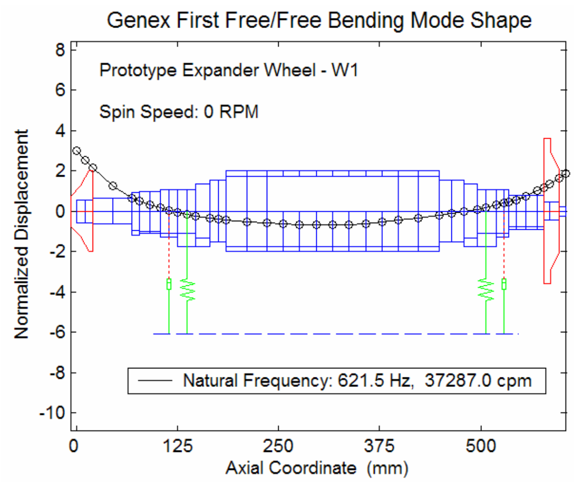

FIGURE 4. Rotor model and first free/free bending mode shape with wheel W1.

The prototype application wheel is the lightest one in Figure 3 and Table 1 – designated as W1. Although there are lighter wheels planned for the system, their dynamic characteristics are similar to W1. The rotor model and first free/free bending mode shape are shown in Figure 4 for the expander with the W1 wheel. The rotor free/free natural frequency map is shown in Figure 5. Although the bending modes are quite gyroscopic, the sensors and actuators are near the nodes of the mode. This characteristic significantly reduces the gain of the mode, which is reflected in the plant transfer functions of Figure 3. This characteristic is desirable for subcritical operation, as it simplifies the required compensator. Of course, if supercritical operation were planned, this would be an undesirable characteristic. Also, as pointed out above, the gain of the first three bending modes is low regardless of the wheel size.

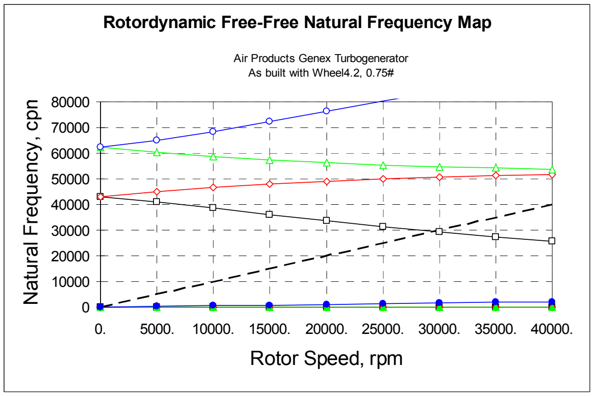

FIGURE 5. Natural frequency map with wheel W1.

The magnetic bearing transfer function used for the expander end radial actuator is in Figure 6. In addition to the control compensator, this transfer function includes models for the position sensor, amplifier, anti-alias filter, and calculation delay. The control compensation is generated in a dedicated DSP (digital signal processor). Because the bending modes were pushed high enough, the compensator can be rolled off quickly enough to gain stabilize all of the bending modes throughout the speed range (gain scheduling not required).

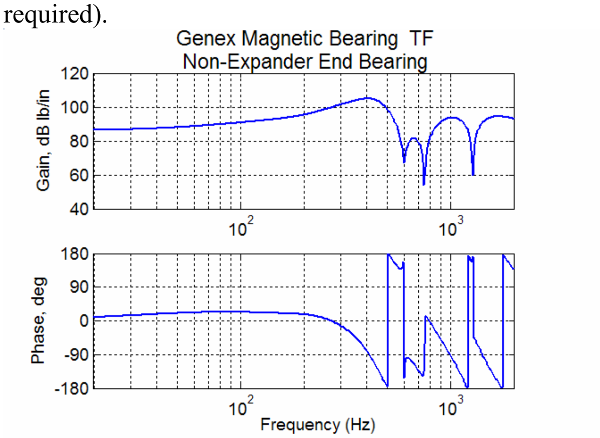

FIGURE 6. Magnetic bearing transfer function for non-expander end bearing.

The predicted sensitivity transfer function for the non-expander end radial bearing is shown in Figure 7. Sensitivity is shown for the expander/generator with three different wheel sizes (W1, W3, W5) from light to heavy. All three curves are calculated with the same compensator transfer function. Sensitivity is the proposed stability criteria in the ISO standard currently under development for magnetic bearings [2]. The CD designation indicates that the standard is in the Committee Draft stage and has been through one cycle of review by standards organizations in member countries. This standard was recently discussed at length by Matsushita & Kanemitsu [3]. The sensitivity function maps the reciprocal of the distance from the system open loop transfer function to the critical point (-1,0) on a Nyquist plot. The proposed limits would recommend sensitivity below 8 dB for new machines (Zone A). Any machine with sensitivity below 12 dB is considered acceptable for unrestricted long-term operation (Zone B). Figure 7 shows that the sensitivity of the generator/expander falls within Zone A for all three wheel sizes. Although not required, it is desirable for practical reasons to be able to use the same magnetic bearing control compensator for all wheel sizes.

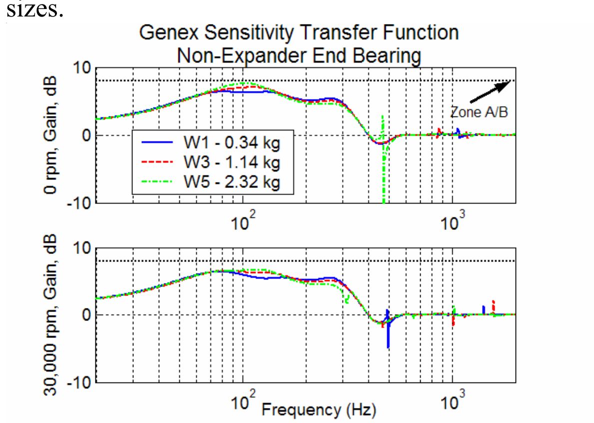

FIGURE 7. Comparison of predicted sensitivity transfer functions for three wheel sizes.

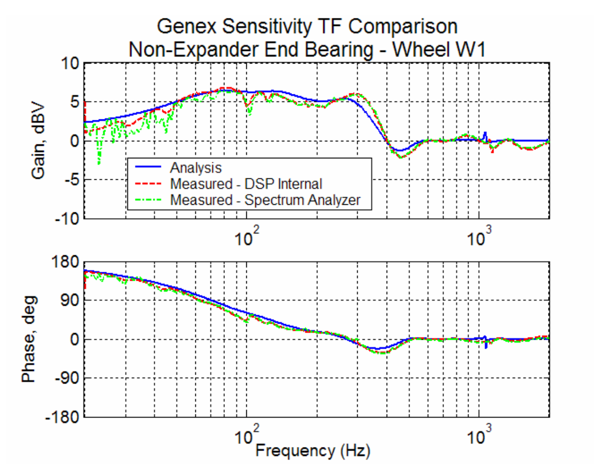

FIGURE 8. Comparison of predicted and measured sensitivity transfer functions for prototype wheel W1.

Figure 8 is a comparison of the measured sensitivity to the predicted sensitivity for the W1 impeller at 0 rpm. Two experimental curves are shown: 1) one using an internal DSP based algorithm, and 2) one using an external measurement from a spectrum analyzer. The two measured curves have very close agreement except at low frequency where the external measurement is somewhat noisy. The agreement of the two measurements with the analysis is quite good although the measurement shows about 1 dB higher sensitivity near the 300 Hz rigid body/compensator mode. As in Figure 7, the sensitivity is well within the ISO Zone A for all modes.

SHOP TESTING

The generator/bearing system was tested to full speed and partial load at the Calnetix facility (shop testing). The unit was then delivered to the customer for complete testing in their flow test facility. The shop testing is reported here.

In the final system, the expander/generator is driven by expansion of a process stream across the expander wheel. A suitable process stream was not available for the shop testing so the unit was spun in two steps. For initial system checkout, the generator was driven as a motor up to 23,000 rpm using an available motor drive. The speed was limited because the generator is wound for 640 VAC at 33,000 rpm and the available motor drive used the nominal 480 VAC line. This intermediate speed allowed completion of much of the required testing: validation of the design control algorithm, setup of the UFRC parameters, and traverse of the rotor rigid body modes. Backup bearing drop testing was also performed in this configuration because the rotor was uncoupled as it would be in the field installation.

Next, an external motor was used to spin the rotor up to full speed (30,000 rpm) through a flexible coupling. The generator was tested to 50 kW (the power capability of the drive motor) by supplying power from the generator to a load bank. The PCU was then grid-tied and the complete system was operated through the PCU and system controller for final shop checkout.

Synchronous response data from a test run up to 30,000 rpm is shown in Figure 9. The peak response was about 0.02 mm (0.0008 in) near 25,000 rpm. Synchronous current (not shown) was never greater than 0.25 Amps on any radial axis between rest and full speed. Synchronous cancellation (UFRC) was used throughout the speed range. Two algorithms were used for different parts of the speed range of the machine. Up to 10,000 rpm, a gain matrix based current minimization algorithm was used. Above 10,000 rpm (above the rigid body modes), an input error minimization algorithm was used.

A series of 10 drop tests onto the backup bearings was performed in the initial testing. In each test, the rotor was dropped on to the backup bearings at 23,000 rpm. The speed was held constant for about 1.0 seconds and then pulled down to zero speed by dumping power

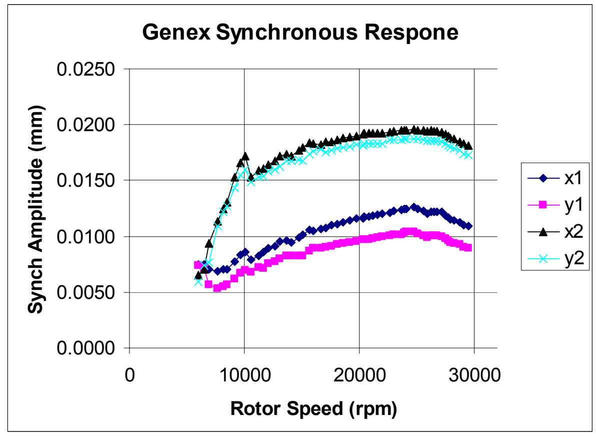

FIGURE 9. Measured synchronous displacement amplitude from full speed run of machine.

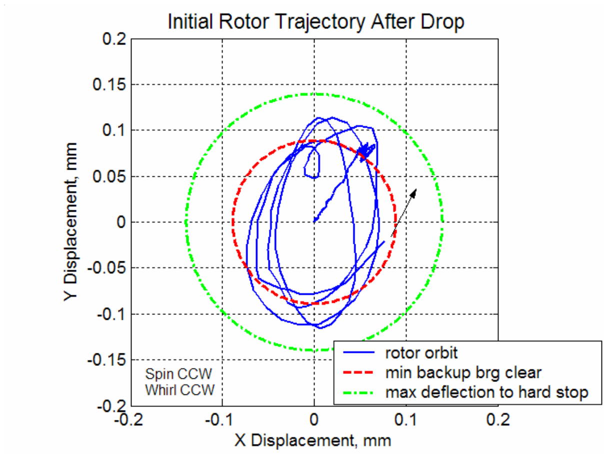

FIGURE 10. Measured rotor/housing displacement orbit after rotor drop at 23,000 rpm.

from the generator into a load resistor. The time delay matches the design trip delay for the shutdown signal in the event of a problem. Position data was recorded internally by the magnetic bearing controller and later uploaded to a PC. When levitated, position and current command data are continuously stored into a circular buffer in RAM. A fault condition, such as excess rotor displacement, triggers the controller to save a preset amount of pre-trigger and post trigger data. Figure 10 shows rotor displacement orbit data measured during one of the drops on the backup bearings. After landing on the backup bearing, the rotor position dwelled for about 100 ms and then started a forward whirl around the bearing clearance. The whirl frequency was about 85 Hz, close to the frequency of the vertical support mode. The characteristics of the response are similar to those observed by Schmied and Pradetto [4]. The elliptical orbit is due to the support asymmetry in the resilient backup bearing mount. The system was later disassembled to install a new set of backup bearings before delivery of the unit. There was very little material loss from the bronze landing sleeve (Figure 11). Change in diameter was about 2.5 μm (0.0001 in) on the expander end and not measurable on the non-expander end.

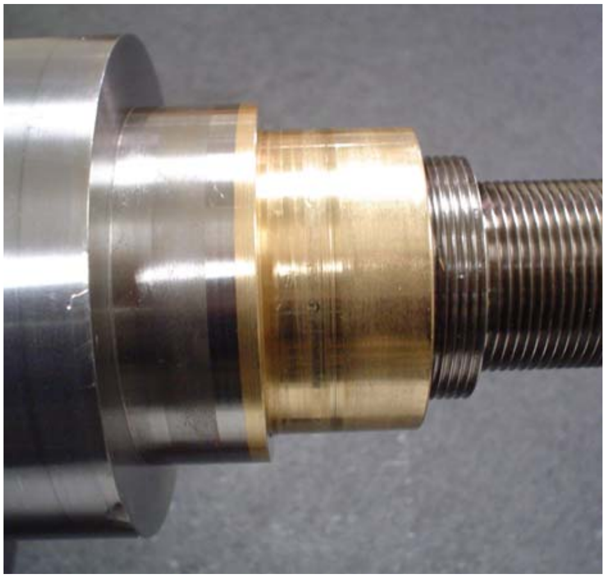

FIGURE 11. Expander end backup bearing landing sleeve after 10 drops and spin downs from 23,000 rpm.

SUMMARY

A direct drive, 30,000 rpm, generator/expander on magnetic bearings has been developed and shop tested. The design allows for a broad range of expander wheels to be used with the same generator module. Shop validation tests included magnetic bearing load testing, stability margin measurement, full-speed operation, partial load testing of the generator, and testing of the backup bearings at 75% of design speed.

The generator/expander is currently undergoing flow loop testing at the system designer's facility.

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