FIELD PERFORMANCE OF AN INNOVATIVE AMB-SUPPORTED TURBOEXPANDER-GENERATOR FOR NATURAL GAS PRESSURE LETDOWN

Jeremy Liu  
Sapphire Technologies  
Cerritos, CA

Ovais Najeeb  
Calnetix Technologies  
Cerritos, CA

Larry Hawkins  
Calnetix Technologies  
Cerritos, CA

Rasish Khatri  
Calnetix Technologies  
Cerritos, CA

ABSTRACT

This paper is a follow up to GT2022-82293, which discusses the design validation of an innovative turboexpander-generator (TEG) used for natural gas pressure letdown (PLD). This paper presents the field performance data of the 280kW machine and compare it to predictions. The TEG has a flanged housing that is directly mounted on a pipeline. The high-speed motor is supported by active magnetic bearings (AMB) in the hydrocarbon environment without lubrication. The paper discusses power performance of the impeller and compares it to the power prediction from a mean line aerodynamic analysis presented previously. Additionally, this paper compares magnetic bearing dynamic controls and transfer functions measured in air to those with gas flow in the pipeline; the effect of gas flow on the closed-loop dynamics is discussed. The authors review the thrust balancing design and the axial control current as a function of gas flow. The magnetic current is linearly related to load, and thus gives a good approximation of aerodynamic load vs flow. This paper also presents the results of a non-linear auxiliary back-up bearing high-speed rotor drop simulation, and compares to measured rotor drop data. The drop testing was conducted as part of the design verification of the prototype unit, and the results are discussed in depth in GT2022-82293.

Keywords: Turboexpander Generator, Natural Gas, Hydrogen, Pressure Letdown, Joule-Thomson, High-Speed Generator, Expander, Clean Energy, Power Generation, Active Magnetic Bearing

NOMENCLATURE

AMB  Active Magnetic Bearing
CFD  Computational Fluid Dynamics
HSG  High-Speed Generator
J-T  Joule-Thomson
MCS  Maximum Continuous Speed
PLD  Pressure Letdown
PM  Permanent Magnet
PRV  Pressure Regulating Valve
TEG  Turboexpander-Generator
TF  Transfer Function
VSD  Variable-Speed Drive

1. INTRODUCTION

An innovative turboexpander-generator (TEG) has been developed to harness energy from pressure differentials and generate electricity. Throughout the natural gas distribution network, there are locations where pressure is reduced in pressure letdown (PLD) stations. At these PLD stations, a Joule-Thomson (J-T) valve or pressure regulating valve (PRV) is commonly used to reduce the pressure of natural gas. A TEG can be used in parallel or in place of the PRV to generate electricity with no consumption of the natural gas, as shown in Figure 1. The TEG will expand high pressure gas thru a turbine to create mechanical shaft power. Then that mechanical shaft power is converted to electricity by a generator and then the electricity is conditioned by a variable speed drive (VSD) to connect to the grid.

![FIGURE 1. PROCESS FLOW DIAGRAM FOR TEG PLD APPLICATION](image-url)
2. TURBOEXPANDER-GENERATOR DESIGN

While this section gives an overview of the major design components of the TEG, Khatri et al. [1] describes the design of the TEG in detail. The turboexpander drives a high-speed generator (HSG) supported by permanent magnet (PM)-biased homopolar active magnetic bearings (AMB), as shown in Figure 2. The PM synchronous generator has a rated speed of 25,000 rpm and an overspeed of 30,000 rpm. Figure 1 also shows the gas flow path; one of the key features of the TEG is the ability to utilize the gas flow path to cool the HSG. The pre-heated inlet gas enters through the annulus around the AMBs (from the right in the figure) and enters the turbine wheel radially. The expanded (and cooler) gas exits axially through the wheel, flows through the annulus between the HSG rotor and stator, and exhausts axially (to the left in the figure).

Brush seals on both sides of the impeller collectively control leakage and control any pressure imbalance on either side of the wheel, see Figure 3. The inboard brush seal at the impeller exit restricts the high-pressure inlet gas from circumventing the impeller. The outboard brush seal on the opposite end of the impeller reduces leakage into the AMB section under the inlet cone shown on the right in Figure 3. The bleed tube vents pressurized gas from AMB section under the inlet cone back to the impeller exit. This balances the back pressure on the outboard side of the wheel with the exit pressure on the inboard side of the wheel. This brush seal configuration significantly reduces the nominal thrust load imbalance experienced by the turboexpander wheel.

Figure 4 shows the major components comprising the AMBs. The configurations for the AMBs are similar to those described by Filatov et al. [2] and Filatov and Hawkins [3] for other applications. The outlet or exhaust side of the machine is supported by a radial bearing, consisting of a radial PM-bias homopolar actuator and radial position sensor. The inlet side of the machine is supported by a side-by-side (SBS) combination PM-bias homopolar radial and axial actuator and a combination radial/axial position sensor. The magnetic bearings are driven by a magnetic bearing controller (MBC).

The machine also includes two back-up bearing assemblies that will support the rotor in the event of a process overload, magnetic bearing system malfunction or deactivation at zero speed. Each back-up bearing assembly (Figure 5) includes a metal tolerance ring, used as a resilient element, between the bearing outer ring and the housing. Additional information about the magnetic bearing, MBC, and back-up bearing assemblies can be found in [1].
3. AERODYNAMIC POWER PERFORMANCE

This paper reviews the larger of the two machines from previous papers [1,4]. The design point of the TEG system was originally 280kW at the grid for a mass flow of up to 20000 Nm$^3$/h with an inlet flow condition of 2.0MPa at 45°C and an outlet pressure of 0.8MPa.

To ensure that the rated power can be delivered to the grid, the impeller aerodynamic design considers losses that occur during operation. To meet power requirements, a net shaft power of 304 kW turbine power is required at the TEG shaft, see TABLE 1. The permanent magnet generator has an electromagnetic efficiency of 98% and the variable speed drive has an efficiency of 94% for a total efficiency from turbine shaft power to grid of 92%.

TABLE 1. DESIGNED NET POWER CALCULATION [1]

<table>
<thead>
<tr>
<th>Power, kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Net Shaft Power</td>
</tr>
<tr>
<td>TEG Electromagnetic Losses [98% Efficiency], kW</td>
</tr>
<tr>
<td>TEG Net Power</td>
</tr>
<tr>
<td>Variable Speed Drive Losses [94% Efficiency], kW</td>
</tr>
<tr>
<td>Grid Power</td>
</tr>
</tbody>
</table>

The TEG was installed in parallel with a pressure regulating J-T valve at the Toho Gas Yokkaichi terminal station. The flow across the TEG was controlled by an upstream flow control valve, FCV, that varies the inlet pressure to the TEG, see Figure 6. The total pressure drop across the PRV is 1.37MPa (2.22 MPa to 0.85 MPa). Flow across the TEG path is regulated as an intermediate pressure at outlet of the FCV was adjusted between 0.85MPa to 1.96MPa to vary flow across the TEG during this set of testing. Across these test points, the inlet temperature decreases with isenthalpic expansion across the FCV. Figure 6 shows the following measured parameters from instruments:

1. Inlet
   a. FT1: Flow Rate
   b. TE1: Temperature
   c. PT1: Pressure
2. Between FCV and TEG
   a. PT2: Pressure
3. Outlet
   a. TE3: Temperature
   b. PT3: Pressure

NIST Refprop database was used to calculate the inlet conditions to the TEG assuming an isenthalpic expansion across the FCV, see TABLE 2.

TABLE 2. CONDITIONS ACROSS THE FCV

With a fixed geometry of the aerodynamic design of the nozzle and impeller in the TEG, a corrected mass flow changes as a function of pressure drop. For a given pressure ratio, mass flow increased as inlet density increases. To compare to previously presented mean line analysis, flow is normalized to inlet pressure and inlet temperature with a corrected mass flow to characterize flow as a function of pressure where corrected mass flow is defined as the following (“inlet” is the inlet of the expander):

\[ \text{Flow}_{\text{Corrected}} = \text{Mass Flow} \times \frac{\sqrt{\text{Abs Temp}_{\text{inlet}}}}{\text{Abs Pressure}_{\text{inlet}}} \]

Corrected flow for the maximum measured power condition is calculated as follows:
Pressure Ratio of the TEG is calculated as a ratio of absolute pressure. The following is the example of the pressure ratio for the maximum measure power condition.

\[
\text{Pressure Ratio} = \frac{1.96 \text{MPa} + .1013}{0.84 \text{MPa} + .1013} = 2.19
\]

Power was measured directly at the grid. Net shaft power is estimated with a 92% efficiency from net shaft power to grind (estimated 98% electromagnetic efficiency & estimated 94% VSD efficiency). The following is the calculation for maximum measured shaft power.

\[
\text{Power}_{\text{Shaft}} = \frac{\text{Power}_{\text{Grid}}}{\eta_{\text{Shaft to Grid}}} = \frac{292.1 \text{kW}}{92\%} = 317.5 \text{kW}
\]

The aerodynamic efficiency is calculated as a comparison of shaft power to the ideal available power from isentropic expansion.

\[
\eta_{\text{Aerodynamic}} = \frac{\text{Power}_{\text{Grid}}}{\text{Power}_{\text{Isentropic}}}
\]

Where

\[
\text{Power}_{\text{Isentropic}} = \frac{\text{mass}}{s} \times \left( \frac{h_2 - h_3}{\text{kg}} - \frac{810.5}{\text{kg}} \right)
\]

\[
\text{Power}_{\text{Isentropic}} = 385.8 \text{kW}
\]

\[
\eta_{\text{Aerodynamic}} = \frac{317.5 \text{kW}}{385.8 \text{kW}} = 82.3\%
\]

To compare to power analysis to the mean line design analysis, power is normalized to temperature and pressure with corrected power. Power potential at a given pressure ration is increased as either pressure or temperature is increased (“inlet” is the inlet of the expander).

\[
\text{Power}_{\text{Corrected}} = \frac{\text{Power}_{\text{Shaft}}}{\sqrt{\text{Abs Temp}_\text{inlet} \times \text{Abs Pressure}_\text{inlet}}}
\]

Corrected flow for the maximum measured power condition is calculated as follows:

\[
317.5 \text{kW} \quad \frac{\sqrt{(67.4^\circ \text{C} + 273.15) \times (1.96 \text{MPa} + 1000 + 101.3)}}{0.00445 \quad \frac{\text{kW}}{\sqrt{\text{K}} \times \text{kPa}}
\]

The following data in TABLE 3 shows a summary of measured parameters and calculated conditions across several test conditions of the TEG. Measured values are highlighted in bold.

<table>
<thead>
<tr>
<th>After Flow control Valve (Expander inlet)</th>
<th>Pressure Ratio</th>
<th>Mass (kg)</th>
<th>Mass (kg)</th>
<th>Mass (kg)</th>
<th>Mass (kg)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1</td>
<td>1.13</td>
<td>1.21</td>
<td>1.49</td>
<td>1.7</td>
<td>1.92</td>
<td>1.96</td>
</tr>
<tr>
<td>P2</td>
<td>5.23</td>
<td>5.39e3</td>
<td>5.136e3</td>
<td>5.301e3</td>
<td>4.994e3</td>
<td>4.987e3</td>
</tr>
<tr>
<td>T2 (if)</td>
<td>661</td>
<td>640</td>
<td>670</td>
<td>674</td>
<td>670</td>
<td>674</td>
</tr>
<tr>
<td>T2 (if)</td>
<td>kPa</td>
<td>kPa</td>
<td>kPa</td>
<td>kPa</td>
<td>kPa</td>
<td>kPa</td>
</tr>
</tbody>
</table>

TABLE 3. MEASURED PROCESS FLOW AND POWER ACROSS TEG

The corrected flow and corrected power were calculated from the data and plotted in Figure 7, Figure 8, and Figure 9. The mass flow is a little lower than expected. However, the overall system efficiency is better than expected. We were able to reach full power with less than the rated flow. Expected corrected flow and corrected power at the design pressure drop of 2.33 is extrapolated from the highest 2 points on Figure 7 and Figure 9.
The maximum measured grid power of 292.1 kW was measured and continuous running was demonstrated with an expander inlet condition of 1.96 MPa, 67.4°C, and a mass flow of 14954 Nm³/h. Based on two highest power test conditions at 276.6 kW and 292.1 kW, an extrapolation of corrected power is made for the rated condition where the inlet pressure is 2.0 MPaG and the outlet pressure is 0.8 MPaG (Pressure ratio of 2.331). The extrapolated corrected flow is $0.0299 \frac{kg}{sec \sqrt{K_{inlet}}/kPa}$ and extrapolated corrected power is $0.00896 \frac{kw}{\sqrt{K} \times kPa}$; this is shown by the triangles in Figure 7 and Figure 9.

Calculated flow and corrected power can be calculated using the rated inlet temperature of 45°C. The calculated power at 2.33 pressure ratio shows that the TEG is capable of 309 kW at the rated pressure ratio and inlet conditions. Therefore, we can up rate the machine to 300 kW.

$Mass \ Flow = 0.0299 \frac{kg}{sec \ kPa} \times \frac{\sqrt{K_{inlet}}/kPa}{(2.0 \ MPa \times 1000 + 101.3 kPa)} \times \frac{1000 + 101.3 kPa}{\sqrt{45°C + 273.15}}$

$Power_{Shaft} = Power_{Corrected} \times \sqrt{Abs \ Temp_{inlet}} \times Abs \ Pressure_{inlet}$

$Power_{Shaft} = 0.00896 \frac{kW}{\sqrt{K} \times kPa} \times \sqrt{45°C + 273.15} \times (2.0 \ MPa \times 1000 + 101.3 kPa)$

$Power_{Shaft} = 309 kW$

4. ROTORDYNAMICS & CONTROL SYSTEM TESTING

The testing of the rotordynamics and controls system was performed in three main stages. The first stage involved the use of a dummy impeller that had the same mass, polar moment of inertia, and transverse moment of inertia as the real impeller. This dummy impeller was installed in the turbomachine without seals. The measured and predicted plant transfer functions were compared, and the MBC compensator was tuned accordingly.

The second stage involved the installation of the actual impeller with brush seals. The plant transfer functions were compared before and after seal installation, and the MBC compensator was tuned again. The controller measures transfer functions per ISO 14839 [6] following the methods described in [5]. Data from the first two stages were presented in the two papers preceding this one [1,4].

The machine was then shipped to Japan for the third stage of testing, which involved testing the TEG under gas flow conditions and making any necessary adjustments. Data from this stage is presented in this paper.

The 280-kW machine was installed in the gas pipeline by cutting a section of the pipe and dropping the TEG in a flange-to-flange configuration. The first step in the commissioning process was to measure the transfer function with no natural gas flowing through the TEG. Figure 10 shows the measured plant transfer function at three speeds (0, 15 krpm, 25 krpm) for radial and axial magnetic bearings with no gas flowing through the machine.

The gain is similar at low frequencies showing little variation with running speed (Figure 10a). At a running speed of 15 and 25 krpm, there is a 80 Hz housing mode excited by the energy of the spinning machine. The peaks at 250 Hz and 417 Hz refer to respective running speeds of the machine at 15 krpm and 25 krpm, not visible at the 0-speed transfer function. At 25 krpm, the bending mode at 583 Hz tends to split as a forward and backward bending mode due to gyroscopic stiffening.

Gain and phase are speed dependent for the bearing 2 radial bearing (Figure 10b). The gain increases while the phase decreases as the TEG is spun up. This effect is due to the presence of brush seals. Bearing 2 is closer to the impeller. Therefore, brush seals have a greater impact on the plant transfer function of Bearing 2. The brush seals provide significant damping and cause almost complete removal of the structural modes seen at 15 and 25 krpm at the bearing 1 end (no peak seen at structural mode of 80 Hz). The seals however are too soft to alter the phase or gain of the bending modes.
Figure 10c shows no gain or phase difference at lower frequencies. The red and peaks shown at higher frequencies are the associated running speeds and their corresponding harmonics.

The next step of the commissioning process was to flow gas through the TEG and assess its performance. Figure 11 shows the measured plant transfer function at different power generation levels (different flow levels) at a running speed of 25 krpm for radial and axial magnetic bearings. Varying Gas flow doesn’t seem to have an impact on the radial bearing 1 transfer functions (Figure 11a). The gain and phase remain the same at all power levels.

Radial bearing 2 shows a unique reaction to varying gas flow. The phase does not vary but the gain increases as gas flow increases at lower frequencies (Figure 11b). This effect is speculated to be due to the presence of brush seals, but the exact phenomenon is unexplained. The authors expected the phase to change along with the gain as seen on transfer functions at 0 speed and pre- and post-seal conditions [4].

Figure 11c shows the axial plant transfer function at different flow/power levels running at 25 krpm. The low-frequency gain drops between no flow and the 55kW power level and then stays constant. This is due to the negative stiffness that is associated with gas flow. The misalignment between the outlet of the gas flow path and the inlet of the impeller, and the slight wobble in the impeller’s rotating axis give rise to negative stiffness in the axial direction causing a drop in gain. The results show that the negative stiffness remains constant at all flow/power levels above 0. All the plots converge after 70 Hz to similar gain and phase levels throughout the frequency spectrum showing no sensitivity to gas flow.

**FIGURE 10.** (a) RADIAL BEARING 1 PLANT TRANSFER FUNCTION AT 3 SPEEDS (b) RADIAL 2 PLANT TRANSFER FUNCTION AT 3 SPEEDS (c) AXIAL BEARING PLANT TRANSFER FUNCTION AT 3 SPEEDS.
5. AXIAL LOAD VS POWER

Khatri [1] reported a predicted axial load at full power of 240 N (55 lbf), and a AMB thrust capacity of 1200 N (270 lbf). The excess capacity is intended to allow the machine to cover a wide range of flow and gas conditions in various applications. Additionally, some extra margin is always included for process upset conditions.

Figure 13 shows a plot of measured axial load versus generated power. This load is calculated – using a force constant of 220 N/A (49 lbf/A) – from the AMB axial coil currents measured via the AMB GUI at the end of commissioning.

Initial deployment of the TEG showed axial currents and loads at full power at around 3.5 A and 770 N (172 lbf). The high steady-state axial load and TEG’s reduced remaining load capacity to react to process upsets resulted in an overload of the axial bearing and a delevitation (rotor drop) during the initial stage of commissioning. Since the axial load was significantly higher than the initial predicted value in the design [1], the TEG was inspected and reworked to ensure proper axial alignment and improved flow in the bleed tube.

Previously predicted axial load at full power predicted a 240 N (55 lbf) in the upstream direction as a reacting force to gas momentum thru the wheel. However, the pressure drop across the bleed tube was not considered in the previous analysis. A CFD analysis was conducted to review the pressure drop along the bleed tube. This pressure drop acts as an axial force in the downstream direction across the area underneath the brush seal diameter, 113.5mm (4.467 in). This CFD shows a 75.8kPa (11.0 psi) pressure drop that would result in a 767N (172 lbf) axial force in the downstream direction. The net predicted force in the downstream direction is 527N (118lbf).

Following the rework, the TEG showed a significantly lower axial load and stable operation through all flow conditions. The final measured axial load at full power was measured at 400N (90 lbf). This is close to the predicted value of 527N (118lbf) and implies that the assumes brush seal leakage in the analysis is conservative. At the measured axial load, there is still a significant margin to the axial load capacity. No additional active control or balance piston was needed for stable operation.

FIGURE 11. (a) RADIAL BEARING 1 PLANT TRANSFER FUNCTION AT DIFFERENT POWER LEVELS AND 25 KRPM (b) RADIAL 2 PLANT TRANSFER FUNCTION AT DIFFERENT POWER LEVELS AND 25 KRPM (c) AXIAL BEARING PLANT TRANSFER FUNCTION AT DIFFERENT POWER LEVELS AND 25 KRPM.

FIGURE 12. CFD ANALYSIS OF BLEEDTUBE PRESSURE DROP

FIGURE 13. AXIAL LOAD AS A FUNCTION OF GENERATED POWER.
6. BACK-UP BEARING DROP DATA AND PERFORMANCE

During initial stages of commissioning the TEG in the field, an axial process overload caused sufficient backup bearing contact to trigger a delevitation (rotor drop) onto the backup bearings and trip request to the system control. As discussed in a previous section, this delevitation was related to diminished axial load margin that was corrected with a rework to alignment and the bleed tube. Position and current data from the magnetic bearing sensors during the event was collected by the AMB GUI and is shown here in Figures 12 – 15 for discussion.

Figure 12 shows data from the axial bearing position sensor and axial coil current prior to and during the overload event. The system was operating at the 25,000 rpm full speed and near maximum power level (280 kW). Prior to 116.7 seconds, the average axial coil current was about 3.5 amps indicating an axial load of about 770 N (172 lbf). At that point the rotor is driven toward the negative z (axial) direction by an unexpected additional external force. The magnetic bearing control responds with a steadily increasing positive current to pull the rotor back toward the center position. However, the external force exceeds the magnetic bearing load capacity, causing the rotor to impact the backup bearings at about 116.8 seconds. At 117.6 seconds, the magnetic bearing control de-levitates the rotor and asserts a trip condition, triggering a shut down. The trip delay of approximately 1.0 second is used to prevent spurious trips from temporary but recoverable conditions.

Figure 13 shows x1 axis position and current data from a 31 second time slice starting just prior to the overload down to 4,000 rpm. Immediately following the trip, the brake resistor engages and slows the rotor to 23,700 rpm in equilibrium with the still active gas torque. During early commissioning the gas shutoff trigger was in manual mode, so five seconds elapsed before the gas shutoff valve was activated to close at 125 sec. This starts a very fast spin down as the brake resistor is pulling power from the generator with the driving torque removed. As noted on the plot, the rotor is executing a full-forward (cylindrical) whirl from shutdown to about 8,000 rpm. This behavior, explained by Wilkes [7], is understood to be driven by axial load in conjunction with the relatively large radial clearances needed for backup bearings. Further, the whirl frequency tends to lock onto the lowest natural frequency of the rotor/bearing system. This behavior is widely observed in vertical machines [8,9] and horizontal machines with light rotors and high axial load [10].

Figure 14 shows the observed relationship between spin frequency and whirl frequency during the spin down. The whirl frequency is 130 Hz when the rotor speed is ~23,600 rpm (394 Hz) and falls to 100 Hz during the rapid portion of the spin down curve. The drop in whirl frequency with speed may be attributable to the lower whirl amplitude combined with the nonlinear measured resilient mount stiffness noted by Najeeb [4]. At lower speeds, below 12,000 rpm (200 Hz), the whirl frequency begins to track at about 0.45 to 0.50 times the spin speed. Below 8,000 rpm the rotor begins rocking at the backup bearing bottom in a pendulum mode along with some occasional bumping. This behavior is typical for horizontal rotors with low axial load.

As an example of the forward whirl orbit observed, Figure 15 shows a 0.1 sec time slice from 124 sec, 23,600 rpm. The whirl is dominated by a single frequency – 130 Hz at this rotor speed. The approximate expected whirl frequency, based on the nominal resilient mount stiffness and rotor mass is 117 Hz (the lowest rotor/support natural frequency). The primary reason for using a resilient mount is to reduce the whirl frequency and thus minimize the backup bearing loads due to whirling.

![Figure 12: Axial Overload During Commissioning.](image1)

![Figure 13: X1 Axis Sensor Position, Coil Current and Speed/2500 During Spin Down.](image2)
7. CONCLUSION

The TEG has demonstrated very good performance in the field. Power data showed better than expected system efficiency and has allowed the system to be uprated from 280kW to 300kW. The control of the magnetic bearing system was stable over the range of gas flow and power production. Thrust balancing was improved by rework and functions very well with no required active control. Finally, analysis of a field back-up bearing drop shows the successful function of the back-up bearings and resilient mount for the system.

REFERENCES


