III.2 Oil-Free Centrifugal Hydrogen Compression Technology Demonstration

Hooshang Heshmat
Mohawk Innovative Technology, Inc. (MiTi®)
1037 Watervliet Shaker Road
Albany, NY 12205
Phone: (518) 862-4290
Email: HHeshmat@miti.cc

DOE Managers
Erika Sutherland
Phone: (202) 586-3152
Email: Erika.Sutherland@ee.doe.gov

Katie Randolph
Phone: (720) 356-1759
Email: Katie.Randolph@ee.doe.gov

Contract Number: DE-FG36-08GO18060

Subcontractor
Mitsubishi Heavy Industries, Ltd, Compressor Corporation, Hiroshima, Japan

Project Start Date: September 25, 2008
Project End Date: August 29, 2014

Overall Objectives

Design a reliable and cost-effective centrifugal compressor for hydrogen pipeline transport and delivery

- Eliminate sources of oil/lubricant contamination
- Increase efficiency by using high rotational speeds
- Reduce system cost and increase reliability

Fiscal Year (FY) 2014 Objectives

- Perform validation testing of single-stage compressor system in air and in helium per American Society of Mechanical Engineers (ASME) PTC-10
- Conduct system refinement of multi-stage system for pipeline compression

Technical Barriers

This project addresses the following technical barriers from the Hydrogen Delivery section of the Fuel Cell Technologies Office Multi-Year Research, Development, and Demonstration Plan:

(B) Reliability and Costs of Hydrogen Compression

(J) Hydrogen Leakage and Sensors

Technical Targets

This project is directed towards the design, fabrication and demonstration of the oil-free centrifugal compression technology for hydrogen delivery. This project will identify the key technological challenges for development and implementation of a full scale hydrogen/natural gas centrifugal compressor. The project addresses the DOE technical targets from the Hydrogen Delivery section of the Fuel Cell Technologies Office Multi-Year Research, Development, and Demonstration Plan (see Table 1).

ABLE 1. Technical Targets for Hydrogen Compression

Category	2005 Status	FY 2011 Status	FY 2015	FY 2020
Reliability	Low	Low	Improved	Improved
Isentropic Efficiency	88%	88%	>88%	>88%
Losses (% of H_2 throughput)	0.5	0.5	0.5	<0.5
Capital Investment (based on 3,000 kW motor rating)	\$2.7M	\$2.7M	\$2.3M	\$1.9M
Maintenance (% of Total Capital Investment)	4	4	3	2
Contamination	Varies by Design	Varies by Design	Varies by Design	None

NA – not applicable

FY 2014 Accomplishments

- Performed validation testing of single-stage compressor in helium per ASME PTC-10
- System refinement of multi-stage compressor system
- Final report has been written and reviewed for submission to the DOE

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INTRODUCTION

One of the key elements in realizing a mature market for hydrogen vehicles is the deployment of a safe, efficient hydrogen production and delivery infrastructure on a scale that can compete, economically, with current fuels. The challenge, however, is that hydrogen, the lightest and smallest of gases with a lower viscosity than natural gas, readily migrates through small spaces. While efficient and cost-effective compression technology is crucial to effective pipeline delivery of hydrogen, today's positive displacement hydrogen compression technology is very costly, and has poor reliability and durability, especially for components subjected to wear (e.g., valves, rider bands, and piston rings). Even so called "oil-free" machines use oil lubricants that migrate into and contaminate the gas path. Due to the poor reliability of compressors, current hydrogen producers often install duplicate units in order to maintain on-line times of 98–99%. Such machine redundancy adds substantially to system capital costs. Additionally, current hydrogen compression often requires energy well in excess of the DOE goal. As such, low capital cost, reliable, efficient and oil-free advanced compressor technologies are needed.

APPROACH

The MiTi team has met program objectives by conducting compressor, bearing, and seal design studies; selecting components for validation testing; fabricating the selected centrifugal compressor stage and the corresponding oil-free bearings and seals; and testing of the high-speed, full-scale centrifugal compressor stage, and oil-free compliant foil bearings and seals under realistic pressures and flows in air and helium (used as a simulant gas for hydrogen). Specific tasks included (1) compressor design analysis for an oil-free, multi-stage, high-speed centrifugal compressor system; (2) mechanical component detailed design of oil-free bearings, seals, and shaft system; (3) detailed design and fabrication of a full-scale, single-stage centrifugal compressor for aerodynamic design verification and component reliability testing; (4) compressor performance testing with air and helium; (5) system design refinement; and (6) program management and reporting.

RESULTS

The MiTi® hydrogen compressor design consists of three identical frames, each with three compression stages operating in series at the same rotation speed of 56,414 rpm (1,600 fps). The system capacity is 500,000 kg/day with a total pressure ratio of approximately 2.4 to achieve the desired 1,200 psi discharge pressure. A full-scale, three-dimensional solid model of a single frame has been fabricated for visual reference as a design aid, shown in Figure 1A. A single compressor stage, identical to those in the multi-stage system, has been fabricated and tested to validate aerodynamics of the MiTi® oil-free, high-speed compressor system. The single-stage compressor with volute is shown in Figure 1B. Fabrication and installation of the single-stage compressor was previously conducted and the final compressor test facility is shown in Figure 1C. The single-stage compressor system is located in a reinforced test cell and features remote access so that the rest rig can be fully operated from a safe distance (Figure 1D). A high-resolution video camera is located in the cell, and used for monitoring and video recording. A custom graphical user interface using LabVIEW software (National Instruments) allows for direct command of motor speed control, monitoring of all pressure and temperature data, as well as high-frequency

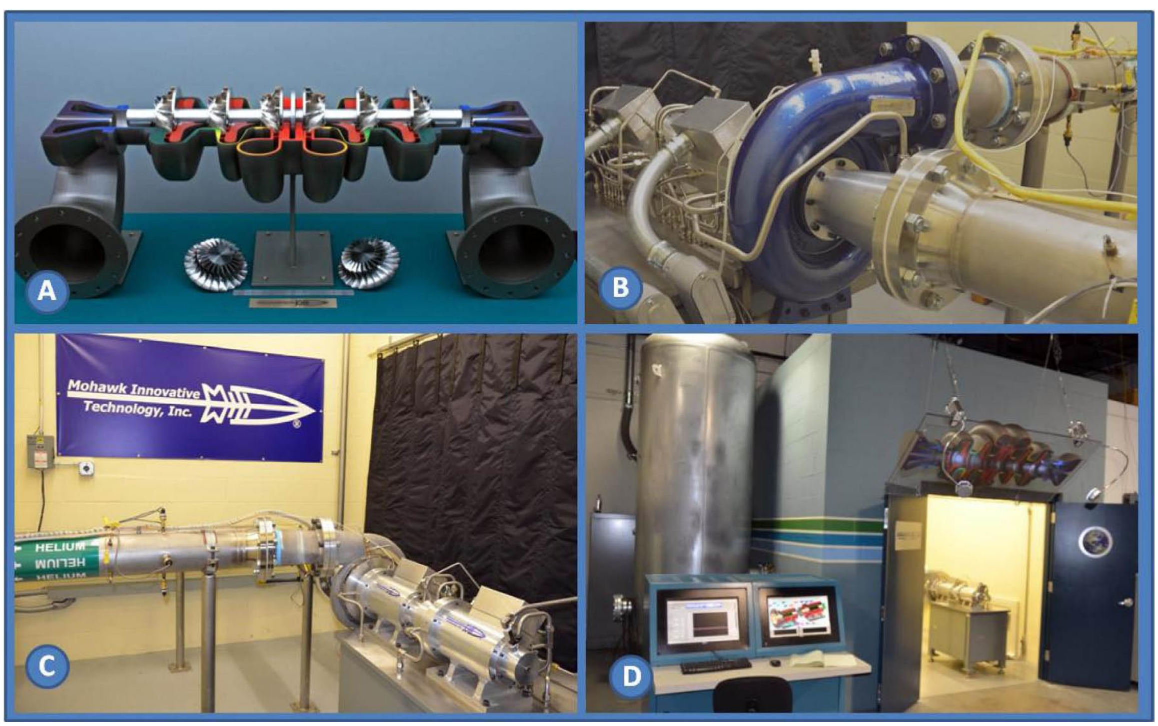

FIGURE 1. MiTi® Single-Stage Compressor System with Oil-Free, 60,000 RPM Motor Drive and Closed-Loop Helium Test Facility

spectral analysis of up to four proximity probes for vibration measurement.

In the past year, extensive aerodynamic performance testing was conducted in accordance with ASME PTC-10. The key function of a Type 2 test is to prove aerodymanic similutude of a compressor design when the exact design conditions are unable to be met. In this project it was not feasible to test in a hydrogen environment. Helium was therefore selected because it most closely matches the physical properties of hydrogen. In order to validate aerodynamic performance of a compressor for hydrogen, it is critical that four non-dimensional quantities be preserved in the test conditions. In Table 2, the list of required non-dimensional variables that must be held constant are shown in Column 1. The values of each variable in the first stage of the full-scale, multi-stage hydrogen compressor are listed in the column labeled "H_2 Design Point." In order to satisfy the requirments of the ASME test code, experimental results must fall within a predefined range, as described in PTC-10. The acceptable variability, as prescribed by PTC-10, is also listed in Table 2. Finally, the experimental results obtained by MiTi® for the single-stage testing conducted in helium are presented in the final column. It can be seen that all experimental results fall within the acceptable range prescribed by ASME PTC-10 and, therefore, validate the aerodynamic performance of the MiTi® hydrogen centrifugal compressor.

TABLE 2. Summary of Operating Conditions Met to Successfully Achieve ASME PTC-10 Type 2 Test Requirements

Quantity	H_2 Design Point	ASME Acceptable Test Variation	Experimental Results
Specific Volume Ratio	1.072	1.018–1.126	1.095
Flow Coefficient	0.125	0.120–0.130	0.126
Machine Mach No.	0.327	0.141–0.532	0.322
Machine Reynolds No.	$6.6e^5$	$6.6e^4$–$6.6e^6$	$8.0e^4$

In addition to the successful demonstration of the compressor at a single operating point, the machine was also operated across a wide range of flows and speeds in order to obtain a complete compressor map. The results of the compressor discharge head are shown in Figure 2. Experimental data points are plotted as individual points. Results were obtained at 20, 30, 40, and 45 krpm. For each operating speed condition, the predicted head is shown as a solid line. For all speed conditions, it can be seen that the experimental results exceed theoretical prediction by 5-10%. The compressor adiabatic efficiency ($\eta_{T\text{-}S}$) was also determined, and these results are shown in Figure 3. Again, theoretical prediction is shown as solid lines. At both 40 krpm and 45 krpm, experimentally measured adiabatic efficiency exceeds prediction, as well as DOE efficiency

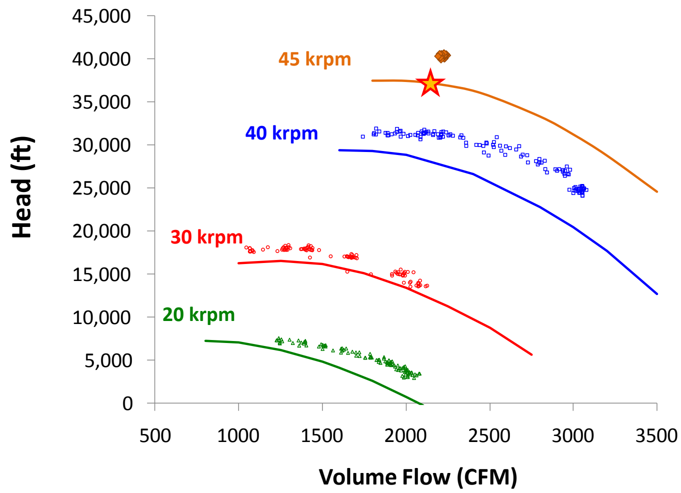

FIGURE 2. Comparison of Theoretical and Experimentally Measured Head for the Single-Stage Compressor (Design point condition indicated with a yellow star)

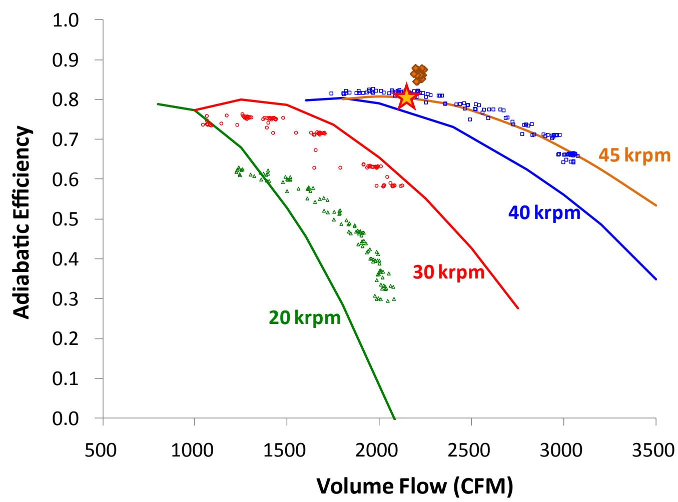

FIGURE 3. Comparison of Theoretical and Experimentally Measured Adiabatic Efficiency for the Single-Stage Compressor (Design point condition indicated with a yellow star)

goals. The experimental adiabatic efficiency (total-to-static) was calculated using the following equation:

$$\eta_{T-S} = \frac{\left(\frac{P_{2s}}{P_{1t}}\right)^{(k-1)/k} - 1}{\left(\frac{T_{2t}}{T_{1t}}\right) - 1}$$

where subscripts "1" and "2" refer to the compressor inlet and discharge, respectively. Subscripts "s" and "t" refer to the static or total thermodynamic condition of the test fluid. While the total-to-total adiabatic efficiency is slightly higher, the use of total-to-static adiabatic efficiency is the most appropriate for evaluation of a single-stage compressor.

The dynamic performance of the compliant foil bearings in a hermetically sealed helium environment was also demonstrated in these tests. For all test conditions, no external cooling gas was passed through the compliant foil bearings which support the entire rotating shaft both radially and axially. A recirculation path exists in the compressor, whereby, a small fraction of compressed helium gas intentionally passes behind the compressor wheel, through the bearing cavities, along the entire length of the shaft and is then plumbed back to the compressor inlet. Compressor performance measurements, including discharge head and efficiency, were measured downstream of the bearing recirculation flow and its effects were already included in their calculation. In order to demonstrate the thermal stability of the compliant foil bearing, the differential temperature within the bearings is shown in Figure 4. Data from only the first thrust and first journal bearing are presented, as these bearings demonstrated the highest operating temperatures of all bearings. Differential bearing temperature is defined as the temperature of the bearing (thermocouple welded to bearing foils) minus the temperature of the compressor discharge (T_{OUT}), which is fed directly to the bearing. A bearing differential temperature greater than zero indicates that some heat from bearing losses is being added to the system. However, a bearing differential temperature less than zero assumes that bearing losses are minimal and more heat within the recirculation flow is lost to housing than is added from the bearing. Because the helium test facility is a closed-loop system and no heat exchanger is present, gas temperature within the system continues to rise as it is compressed. For this reason, it is necessary to discuss bearing differential temperature rather than absolute bearing temperature. From the results in Figure 4 we can see that, for all test points, the temperature of the complaint foil bearings do not exceed the compressor discharge by more than 15°F. The highest differential temperature occurs immediately following a change in compressor speed. Some heat from speed change is a result of power losses in the motor during acceleration or braking of the rotor and has nothing to do with bearing performance. Other causes of heat during change in speed are due to mechanical and thermal transients in the shaft. For each speed change, it can be seen that bearing differential temperatures stabilize and begin to fall after equilibrium is reached. These results indicate very low power loss and a high degree of thermal stability of the complaint foil bearing in a hermetically sealed helium environment. The heat capacity of hydrogen is nearly three times greater than helium. This implies that achieving thermal stability of the compliant foil bearing in the hydrogen environment will be less challenging.

CONCLUSIONS AND FUTURE DIRECTIONS

MiTi® has successfully completed performance testing of a single-stage, centrifugal hydrogen compressor per ASME PTC-10. The Type 2 test conducted in a similitude gas (helium) has shown that the centrifugal compressor stage is capable of exceeding the performance goals predicted by aerodynamic design software and computational fluid dynamics analysis. Experimental adiabatic efficiency was measured to be 86% and discharge head exceeded theoretical prediction by 10%. MiTi®'s compliant foil bearings demonstrated stable and reliable performance with extremely lower power loss. Thermal stability of the compliant foil bearings in a hermetically sealed helium environment was verified. There results provide high confidence in the

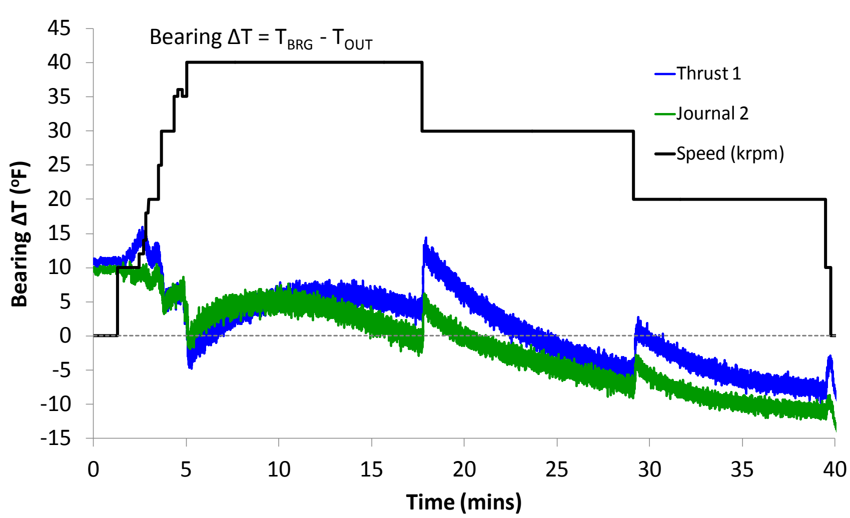

FIGURE 4. Experimental Thermal Performance of the Foil Thrust Bearings during Operation up to 40 krpm

feasibility of the proposed multi-stage centrifugal pipeline compressor concept to meet the DOE's need for 500,000 kg/day of hydrogen with a total pressure ratio of 2.4. The project has been successfully completed.

FY 2014 PUBLICATIONS/PRESENTATIONS

1. H. Heshmat, "Oil-Free Centrifugal Hydrogen Compression Technology Demonstration," DOE Hydrogen and Fuel Cells Program Annual Review and Peer Evaluation Meeting, June 17, 2014, Washington, D.C.