III.2 Oil-Free Centrifugal Hydrogen Compression Technology Demonstration

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Subcontractor:
Mitsubishi Heavy Industries, Ltd, (MHI) Compressor Corporation, Hiroshima, Japan

Project Start Date: September 25, 2008
Project End Date: November 30, 2013

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) 2013 Objectives

- Validate performance of coupled oil-free motors
- Construct a dedicated test cell and install the required closed-loop piping
- Perform validation testing of single-stage compressor system in air and in helium

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 Gaseous 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 following DOE technical targets from the Hydrogen Delivery section of the Fuel Cell Technologies Office Multi-Year Research, Development and Demonstration Plan (see Table 1).

TABLE 1. Technical Targets for Hydrogen Compression

Category	2005 Status	FY 2012	FY 2017
Reliability	Low	Improved	High
Isentropic Efficiency	NA	NA	>88%
Capital Investment ($M) (based on 200,000 kg of H_2/day)	$15	$12	$9
Maintenance (% of Total Capital Investment)	10%	7%	3%
Contamination	Varies by Design		None

NA - not applicable

FY 2013 Accomplishments

- Completed validation testing of two coupled 100-kW oil-free motors.
- Completed installation of a dedicated test cell and required closed-loop piping for testing of the single-stage compressor.
- Performed initial validation testing of single-stage compressor system in air and in helium.

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INTRODUCTION

One of the key elements in realizing a hydrogen economy 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 will meet project 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 conduct 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 include: (1) compressor design analysis; (2) mechanical component detailed design; (3) detailed design and fabrication of a full-scale single-stage centrifugal compressor; (4) compressor performance testing wiith air and helium; (5) system design refinement; and (6) project management and reporting.

RESULTS

The MiTi® hydrogen compressor design consists of three frames operating at the same speed with a rotor tip velocity of 1,600 fps. The system capacity is 500,000 kg/day with a pressure ratio of approximately 2.4. A single-stage compressor system has been developed to verify aerodynamics of the MiTi® oil-free, high-speed, centrifugal compressor system. Fabrication, assembly and installation of the single-stage compressor has been completed (Figure 1) and initial performance verification testing has been conducted. The single-stage compressor system has been placed in a reinforced test cell and all instrumentations and controls have been completed. Key instrumentations and controls are located outside the cell in order to protect personnel and expensive instrumentations. The 200-kW motor drive system is fitted with seven pressure transducers and eight thermocouples positioned along the length of gas cooling path within the oil-free electric motors to monitor the performance of the foil bearings and electric motors. The compressor system is instrumented with nine proximity probes to monitor rotor vibration during operation. A high resolution video camera for remote monitoring and video recording is placed inside the cell. A custom command console has been completed for monitoring of all system data and simultaneous monitoring of the drive motor. A custom graphical user interface has been developed using LabVIEW® software (National Instruments). The new graphical user interface allows for direct command of motor speed control, monitoring of all pressure and temperature data, as well as high frequency spectral analysis of up to four proximity probes for vibration measurement. The command console has remote access capabilities to allow the operator control of all compressor and drive functions from a safe distance away from the cell. A closed-loop liquid cooling system for the motor drives has been installed that consists of a ¾-HP electric motor, solid particulate filter and liquid-to-air radiator. The system is capable of removing greater than 50 kW of heat load if necessary.

FIGURE 1. Dedicated MiTi® test cell, 200-kW single-stage compressor and closed-loop gas system.

The single-stage compressor is driven with two oil-free, high-speed, 100-kW motors coupled together (Figure 2) using the MiTi® Coupling Technology. Using air as the test gas, the performance of each motor was successfully validated from 10,000 rpm to full speed of 60,000 rpm. Foil bearing temperatures were monitored during testing and stable bearing performance was confirmed. Bearing temperatures were less than 150°F at full speed. Rotor vibration was recorded using fiber-optic proximity probes. The maximum rotor motion measured at full speed was 0.0002 inch, which represents extremely low vibration, as it is approximately equivalent to the mechanical run-out of the rotor.

After successfully demonstrating stable, high-speed operation in air, the compressor was coupled to a closed-loop flow system for helium gas testing. The closed-loop, stainless steel flow system includes a large gas accumulator tank, particle filter, remotely actuated throttle valve and flow meter. The flow loop was constructed and instrumented according to the guidelines of American Society of Mechanical Engineers PTC-10.

Preliminary compressor testing in air (open-loop) and in helium (closed-loop) has been performed to verify the aerodynamics of the high-speed, oil-free, single-stage

FIGURE 2. MiTi® Single-stage compressor system with high speed oil-free motor drive.

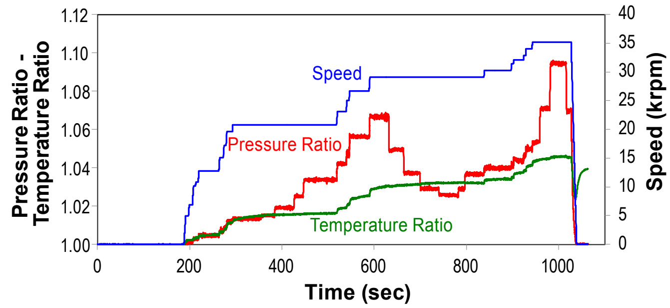

FIGURE 3. Experimental data of the single-stage compressor operating at speeds up to 35 krpm.

compressor. A systematic testing plan was developed to approach the design point of the single-stage compressor with respect to the rotational speed. The test procedure began with pressurization of the flow loop with pure helium and purging of air until less than 1.0 % oxygen was detected. Proper safety measures were taken, including placement of "bomb-proof" Kevlar blankets over the test cell doors and evacuation of all personnel from the surrounding area. The compressor was then started with the speed increasing by 1-2 krpm increments, while rotordynamic behavior and foil bearing temperatures were monitored. The compressor speed was held constant at each speed increment for several minutes to observe the steady-state behavior at that speed. The compressor was then stopped for data analysis and reviewed with project management and the principle investigator.

Results from recent closed-loop compressor testing with helium are shown in Figure 3. The compressor flow was adjusted by position of the throttle valve. The figure shows a rise in pressure ratio and temperature ratio with increasing speed and flow restriction, indicating that significant compression work was being done of the test gas. Pressure ratio and temperature ratio are defined by the following relationships:

$$PR = \frac{P_{OUT-ABS}}{P_{IN-ABS}} \quad \text{and} \quad TR = \frac{T_{OUT-ABS}}{T_{IN-ABS}}$$

A maximum pressure ratio of approximately 1.10 was achieved in this test at 35 krpm with the throttle valve restriction set to 50%. Even though the compressor was operating at 35 krpm with helium gas, bearing temperatures did not exceed 130°F, which is more than 10°F lower than seen in air at 10 krpm. Because the helium gas was in a closed-loop system, the gas was reheated each time it passed through the compressor and, thus, much of the rise in temperature through the compressor system was due to reheating of the same gas volume.

During helium testing, compressor flow was varied by partial closure of the remotely operated throttle valve, as discussed earlier. This procedure was performed at constant speeds of 20 krpm and 29 krpm in order to map compressor performance as a function of flow. The position of the throttle valve was used to approximate flow as a percentage of the maximum flow achieved when the valve was fully open. The compressor performance maps are shown in Figure 4 (a) for the 29 krpm speed condition. The results show that a peak experimental pressure ratio of 1.067 was achieved, which is only slightly lower than the predicted peak theoretical pressure ratio of 1.071 at this speed. These results show a good correlation between the theoretical estimates and measured performance. It is possible that with further flow restriction, an even higher experimental pressure ratio could be achieved. A high-accuracy inline flow meter has been ordered and will be installed next month. Compressor performance mapping will be repeated once accurate flow measurement is added to the flow loop.

Based on the preliminary compressor maps obtained along with temperature measurements of compressor inlet and discharge, compressor efficiency was calculated and experimental results were compared to the theoretical estimates based on mean-line analysis. The experimental adiabatic efficiency (total-to-static) was calculated using the following equation:

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

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. The use of total-to-static adiabatic efficiency is the most appropriate for evaluation of a single stage compressor. A comparison of the theoretical and experimentally measured adiabatic compressor efficiencies is shown in Figure 4 (b). The experimentally measured efficiencies (the red curve) are higher than

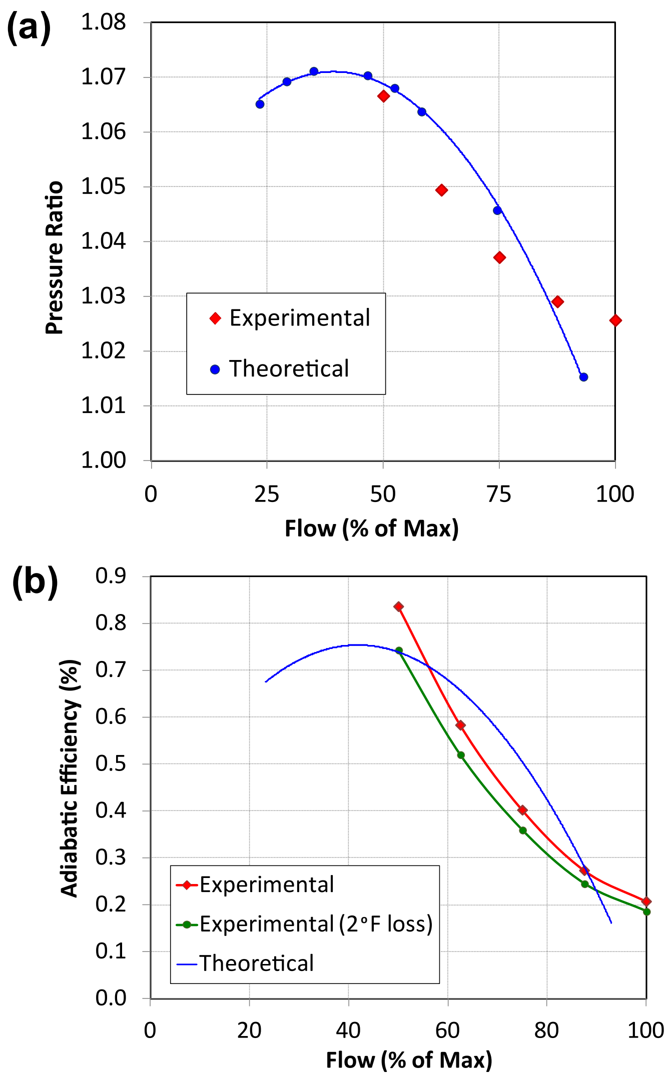

FIGURE 4. Comparison of theoretical and experimentally measured values for the single-stage compressor: (a) pressure ratio and (b) adiabatic efficiencies.

the theoretical prediction. This scenario is impossible and is due to the fact that experimentally measured discharge temperatures are lower than predicted. The lower than predicted compressor discharge temperature is due to the fact that the single-stage compressor is not thermally insulated from the environment and some heat of compression is lost to the surroundings through the piping and the motor housing. In order to account for this thermal loss, a constant value of 2.0°F was added to the compressor discharge temperature in the expression for adiabatic efficiency. This constitutes 11% of the total temperature rise due to compression and is a rather conservative assumption. The resulting calculated efficiency, plotted as the green curve, is closer to the theoretical prediction at the maximum adiabatic efficiency.

At the initiation of the project, it was planned to pursue two design approaches for the hydrogen compressor: a double-entry design by MiTi® and a single-entry design by MHI. During the design process, both companies held frequent conference calls and several site visits to exchange design information and coordinate the design process. The single-stage compressor based on MiTi®'s double-entry compressor design was designed and assembled for testing, as described above. During the design process, an aerodynamic compressor design software, computational flow dynamics, finite element analysis and rotordynamic analysis were performed to assure that the design met all the performance and safety requirements. The single-entry compressor, design by MHI, also followed the same comprehensive analyses but used different software packages available at MHI. During the design process, both MiTi® and MHI checked each other's analyses to ensure accuracy of the different software. Both designs meet the performance criteria for the hydrogen pipeline compressor; however, comparing the two designs with respect to performance, safety and economics, indicated that the double-entry design is superior for the hydrogen pipeline compressor. The safety margin for the single-entry design is less due to larger tip speeds required. Also, the analyses indicated that the single-entry compressor could be more costly due to the need for balance piston and thrust bearings. Therefore, it was decided to pursue the MiTi® double-entry design instead of the MHI single-entry compressor for performance verification testing.

CONCLUSIONS AND FUTURE DIRECTIONS

Installation of the single-stage centrifugal hydrogen compressor, dedicated test cell and closed-loop helium facility were completed and preliminary validation testing was performed. The single-stage compressor system includes two 100-kW, oil-free motors designed and fabricated at MiTi®. The two motors are coupled using MiTi®'s proprietary mechanical coupling technology. Initial validation testing of the motors indicated that each motor was capable of operating at the design speed of 60,000 rpm. Both motors independently evaluated, were thermally and dynamically stable. With proper safety precautions, the compressor system was evaluated in a reinforced test cell up to 35,000 rpm with air and helium used as a simulant gas for hydrogen. The following tasks are planned for the remainder of FY 2013:

- Single-stage performance testing in helium (as a simulant gas for hydrogen)
- Design refinements
- Final report

FY 2013 PUBLICATIONS/PRESENTATIONS

1. J.F. Walton, "Design of a Multi-Megawatt Oil-Free Centrifugal Compressor for Hydrogen Gas Transportation and Delivery," World Hydrogen Energy Conference, Toronto, Canada, June 6, 2012.

2. H. Heshmat, A. Hunsberger, Z. Ren, S. Jahanmir, and J.F. Walton, "Oil-Free Foil-Bearings for Centrifugal Hydrogen Compressor," Tribology Online, January 2013.

3. S. Jahanmir, "Oil-Free Compression for Hydrogen Delivery and Transportation," Hydrogen Delivery Technology Team Meeting, March 27, 2013, Golden, CO.

4. H. Heshmat, "Oil-Free Centrifugal Hydrogen Compression Technology Demonstration," DOE Hydrogen Program Annual Review and Peer Evaluation Meeting, May 15, 2013, Arlington, VA.