III.3 Oil-Free Centrifugal Hydrogen Compression Technology Demonstration

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Contract Number: DE-FG36-08GO18060

Subcontractor:
Mitsubishi Heavy Industries, Ltd, Hiroshima, Japan

Project Start Date: September 25, 2008
Project End Date: August 31, 2011

Objectives

Demonstrate key technologies needed to develop reliable and cost-effective centrifugal compressors for hydrogen transport and delivery:

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

Technical Barriers

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

(B) Reliability and Costs of Hydrogen Compression

(I) Hydrogen Leakage and Sensors

Technical Targets

This project is directed towards the design, fabrication and demonstration of the oil-free centrifugal compression technology. This project will identify the key technological challenges for development 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 Program 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
Energy Efficiency	98%	98%	>98%
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

Accomplishments

- Completed compressor design with nine compression stages and established the impeller diameter, blade design and desired rotational speed.
- Conducted stress analysis to determine the range of stresses developed in the rotor for several selected materials and established the safety margin for each material.
- Conducted rotor dynamic analysis to ensure dynamic stability of the rotor at the operating speed.
- Prepared preliminary drawings of the compressor frame and the overall system layout, including the drive system and transmission.

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Introduction

One of the key elements in realizing 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 2% 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 a hydrogen gas environment. Specific tasks include: (1) Preliminary oil-free, multi-stage, high-speed centrifugal compressor system design; (2) Detailed design of a full-scale centrifugal compressor stage; (3) Mechanical component detailed design of the oil-free bearings, seals and shaft system needed to test the compressor stage; (4) Test hardware fabrication; (5) Dynamic test; (6) Compressor performance test; (7) System design refinement; and (8) Program management and reporting.

Results

The goal of the compressor design analysis was to determine the number of stages necessary to reach a discharge pressure of 1,200 psig and 500,000 Kg/day mass flow. The investigation began by using the maximum allowable tip speed (1,600 ft/s), as determined by earlier finite element analysis (FEA) stress analysis. This condition is labeled "Full Speed" in Figure 1. While higher tip speeds are possible in a non-hydrogen environment, the potential reduction in material strength under hydrogen embrittlement dictates that lower stress conditions be used. Two under-speed conditions were also investigated, denoted as "95% of Full" and "90% of Full". The results show that the Full Speed condition would require seven stages to achieve the desired discharge pressure, while the under-speed conditions would require eight and nine stages, respectively. Previous preliminary analysis had indicated that a maximum of three frames (i.e., nine stages) should be used. Therefore, using a lower speed and nine stages would be convenient because it allows a system to be designed with three identical compressor frames. The use of lower speed also provides increased stress and fatigue safety margins.

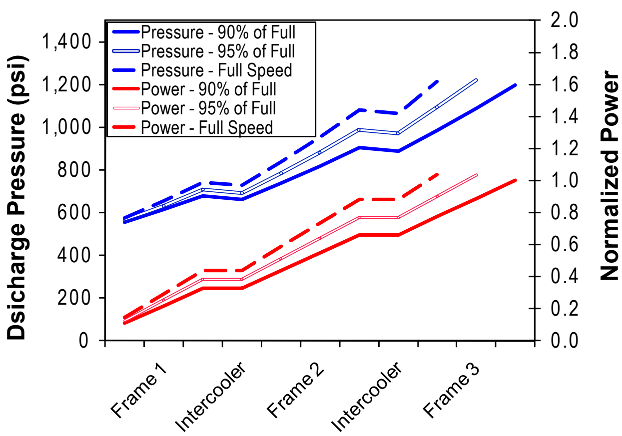

FIGURE 1. Multi-stage performance prediction at three operating speeds.

Further analysis was performed to evaluate if other designs could achieve the desired discharge pressure in two identical frames (i.e., six stages) without exceeding the selected maximum allowable tip speed. The two most common approaches used in compressor design to increase discharge pressure are employing forward-swept blade angle and vaned diffusers. A single compression stage was analyzed to determine the performance gain of forward-sweep and vaned diffusing. As expected, the discharge pressure increased in both cases. The costs of these approaches, however, are increased power requirement, increased manufacturing costs and decreased surge-choke margin. In the multi-stage analysis, both a vaned diffuser and a forward-swept blade resulted in about 8% differential pressure improvements. The power penalty was much greater with the forward-swept blade (15%) compared to only 2% from the vaned diffuser. However, the vaned diffuser and forward-swept blade approaches greatly increase the likelihood of system instability. Since the operating conditions could vary greatly, the most versatile, stable and cost-effective design is a three-frame design (i.e., nine stages) with vaneless diffuser and back-swept impeller blades.

Rather than using three identical stages, frames 2 and 3 were modified with alternate shroud contours through a technique known as reprofiling, which dramatically improves stability and surge margin without the added cost of designing and manufacturing unique compressor stages. The additional pressure margin achieved in the reprofiled design is due, primarily, to the addition of the vaned diffuser. The use of a vaned diffuser in the identical stage design was not practical because it would only add instability to the compressor system.

A detailed blade design was conducted using a quasi-three dimensional inviscid internal flow analysis with well-known, commercially available compressor design software. The flow field was analyzed for areas of excessive diffusion, sudden velocity gradients and flow separation. The design was optimized through minor changes in impeller blade geometry. Once a design was found to have satisfactory flow field characteristics, structural and rotordynamic evaluation was performed. Four key design iterations were conducted between aerodynamics, structural and rotordynamics in order to arrive at the final compressor concept. This was followed by computational fluid dynamics (CFD) analysis of a single stage to validate the aerodynamic performance predictions.

Compressor designs created from aerodynamic analysis were verified for structural integrity using FEA. Structural analysis was focused on the rotor, impeller hub and impeller blades. Three high strength materials were evaluated: an aluminum alloy; a titanium alloy and a high-strength steel. The results confirmed low stresses in the impeller blades. Peak stress in the impeller was on the backface of the hub; however, stress magnitudes were well below the yield strength of the material. The results indicated significant safety margin with respect to strength for all three selected materials. However, the low fatigue limit of the aluminum alloy is of major concern. In fact, the blade stress is three times larger than the fatigue limit of this material. Since the steel and titanium alloy provide sufficient safety margins based on strength and low cycle fatigue, and aluminum does not, aluminum was dropped from further consideration.

The MiTi® 9-stage hydrogen compressor design consists of three double entry systems, each with three back-to-back compression stages (Figure 2). Two MiTi® foil journal bearings are located towards either end of the shaft. Two foil thrust bearings are located at the center of the shaft. Coupled finite difference and finite element methods were used to analyze the foil thrust bearing. The pressure profile was predicted through elastohydrodynamic analysis using the finite difference method and applied to the structural finite element model of the foil thrust bearing. Four MiTi® compliant radial foil seals, which are being characterized under a parallel effort, are used for inter-stage sealing. The predicted elastohydrodynamic plus hydrostatic pressure was applied to the finite element model of the radial foil seal.

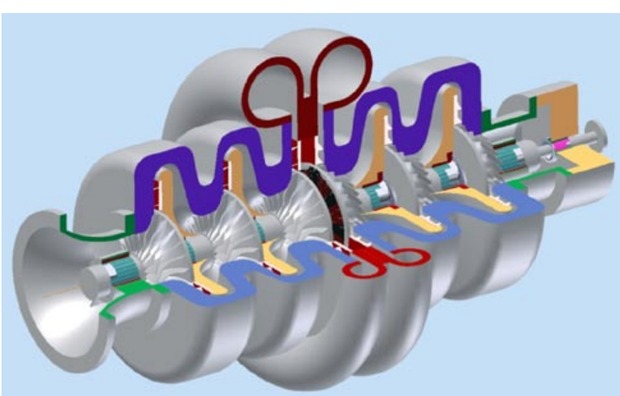

FIGURE 2. Cut-away view of a single frame of the MiTi® hydrogen compressor design.

Whirl speed and stability analysis were conducted using dynamic FEA. The speed-dependent stiffness coefficients were calculated and damping properties were estimated for the foil bearings and seals. Rotor dynamic analysis was conducted for one frame. Based on preliminary aerodynamic analysis, an optimized shaft diameter between the compressor wheels was selected. Rotordynamic analysis was performed with the three selected materials. It was determined that the material type had little impact on the rotordynamic behavior. The whirl speed map in Figure 3 shows the first four damped natural frequencies, including the cylindrical and conical rigid body modes, as well as the 1st and 2nd bending modes as a function of rotational speed. The logarithmic decrement values for these four modes indicated excellent stability for all four modes.

The configuration of the 9-stage hydrogen compressor has been finalized. The system consists of three frames operating at 56,000 rpm. Each frame is a double entry design consisting of three back-to-back compression stages. The design includes vaned diffusers and a return channel. The final stage of each frame discharges to a volute. A complete system layout was drawn (Figure 4). The total system power required is

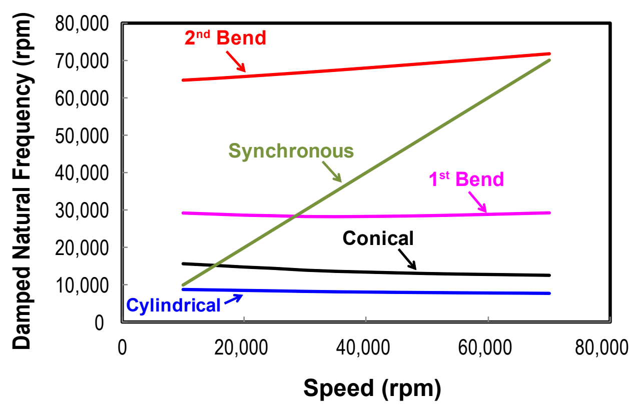

FIGURE 3. Whirl speed map of three stage double entry centrifugal compressor rotor.

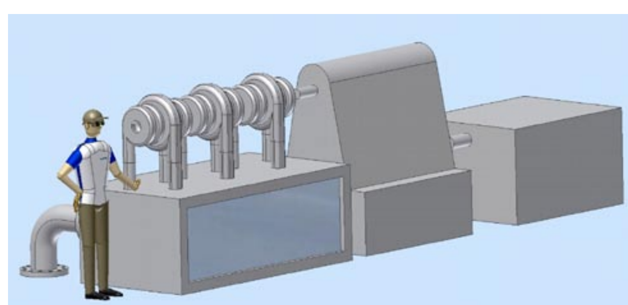

FIGURE 4. Full system layout shown with three-frame compressor, intercoolers, and drive system.

15,128 hp for a specific energy metric of 0.439 kW-hr/kg of compressed H_2, which is less than the DOE's target of 0.6 kW-hr/kg. The system contains three frames coupled to a single rotating shaft. The overall system footprint is estimated to be 160 ft^2, which includes intercoolers located beneath each compressor frame, the drive system and the transmission.

Conclusions and Future Directions

During this reporting period, the design of a 9-stage centrifugal hydrogen compressor was completed. The design process included detailed analysis of a single stage as well as overall system layout, bearing analysis, rotordynamics, and system costs and economics. Analysis methods employed included FEA modeling of stress and deflection of the impeller hub and blades, multi-stage system stability, rotordynamic analysis of the compressor rotor, CFD analysis, and full system layout, including drive components. The results of this analysis indicate that an oil-free centrifugal hydrogen compressor is capable of achieving the performance goals requested by the DOE.

In addition to further analysis of the design concept presented in this report, several other design concepts will be evaluated during the next reporting period. A high-performance and high-speed concept is being analyzed in which the maximum allowable impeller tip speed has been increased to 2,000 fps. While this design will allow for a reduction in the total number of stages, reduced input power and overall system size, such high tip speeds present several technical challenges that are being addressed. A trade-off study will be performed in which the costs and benefits of increased tip speed on system performance is assessed. The following tasks are planned for the remainder of Fiscal Years 2010 and 2011:

- System cost analysis
- Increased speed designs and trade-offs
- Development of a single stage simulation test rig
- Fabrication of components for testing
- Single stage performance testing

FY 2010 Publications/Presentations

1. "Material Needs for Centrifugal Hydrogen Compressor" Hydrogen Pipeline Working Group, August 2009, Boulder, CO.

2. "Oil-Free Compression for Hydrogen Delivery and Transportation," Hydrogen Delivery Technology Team Meeting, December 2009, Washington, D.C.

3. "Oil-Free Centrifugal Hydrogen Compression Technology Demonstration," DOE Hydrogen Program Annual Review and Peer Evaluation Meeting, June 2010, Washington, D.C.