III.4 Oil-Free Centrifugal Hydrogen Compression Technology Demonstration

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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 Hydrogen, Fuel Cells and Infrastructure 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 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 DOE technical targets from the Hydrogen Delivery section of the Hydrogen, Fuel Cells and Infrastructure Technologies Program Multi-Year Research, Development and Demonstration Plan (see Table 1).

TABLE 1. Technical Targets for Hydrogen Compression

Category	2005 Status	FY2012	FY2017
Reliability	Low	Improved	High
Energy Efficiency	98%	98%	>98%
Capital Investment (1M) (based on 200,000 kg of H2/day)	$15	$12	$9
Maintenance (% of Total Capital Investment)	10	7	3
Contamination	Varies by Design		None

Accomplishments

- Reviewed design requirements for compression of hydrogen with a flow of 500,000 to 1,000,000 kg/day with an output pressure of 1,200 psig.
- Completed preliminary compressor design with seven compression stages and established the impeller diameter and the desired rotational speeds.
- Conducted stress analysis to determine the range of stresses developed in the rotor for several selected materials.
- Conducted rotor dynamic analysis to ensure dynamic stability of the rotor design.

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Introduction

One of the key elements in realizing the 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, 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, 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 conducting 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) project management and reporting.

Results

A preliminary compressor design study was performed using commercial aerodynamic performance software. The accuracy of this software's performance predictions was first validated with published data from a commercially available compressor wheel. The design software was able to predict the performance over a wide range of speed and flows with errors of 3%. Results were further validated by using MiTi's own in-house compressor performance code.

The parametric study began with a detailed analysis of a single compression stage that would then form the basis for the multi-stage frames. The proposed compressor design uses a double entry design so that each impeller is only required to compress 250,000 kg/day, or half of the total projected flow, to achieve the required 700 psig pressure rise. In order to achieve the required 700 psig pressure rise with six or seven stages of compression, each stage will need to have a pressure ratio between 1.13 to 1.15. The exact pressure ratio required per stage will be determined through iterative analysis with results from single stage predictions and other details of the system design, including factors such as seal losses and effectiveness of intercooling. Based upon several preliminary single stage design iterations, the baseline nominal impeller stage size was selected for use in the parametric analysis. The single stage parametric study was then performed with six different compressor diameters at seven different operating tip speed conditions while maintaining the hydrogen inlet conditions fixed as noted below:

Fluid: Hydrogen Gas

Inlet Flow: 382 lb/min (250,000 kg/day)

Inlet Temperature: 68°F

Inlet Pressure: 500 psig

Using the fixed hydrogen inlet conditions, the first compressor stage performance was predicted at tip speeds of 1,200-1,800 ft/s. For this tradeoff and preliminary design study, each stage consisted of the impeller, crossover and return channel. Each tradeoff design point evaluated was optimized for the imposed tip speed and geometry conditions. This approach was chosen to permit a direct comparison of stage efficiencies between each design configuration. In Figure 1, the results indicate that for all impeller diameters analyzed, the desired stage pressure ratio of at least 1.13 is obtained at a compressor impeller tip speed of 1,500 ft/s or higher. While the different scaled impeller diameters show very similar predicted pressure ratios at speeds above 1,400 ft/s, the predicted efficiencies and power requirements, as seen in Figures 2 and 3, show a clear benefit of using the full-scale impellers as well as those scaled up as much as 25%. Conversely, the 90% and 80% scaled diameters demonstrate considerably lower efficiency and higher power requirements.

Having confirmed the validity of the full-scale impeller design through the tradeoff study, it was then used in the investigation in the multi-stage performance at an impeller tip speed of 1,500 fps. The selected double entry design was used for the preliminary multi-stage analysis. In earlier analyses, MiTi identified that a limit of three stages of compression (i.e., six single stage impellers) on a single shaft should be imposed in order to obtain suitable rotor-bearing system dynamic performance while also providing for enhanced intercooling options and modular assemblies. Therefore, the double entry 3-stage machine was modeled as two

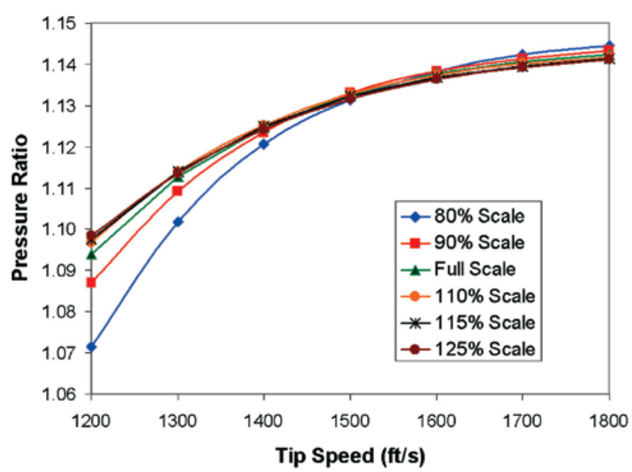

FIGURE 1. Pressure Ratio as a Function of Tip Speed and Impeller Size

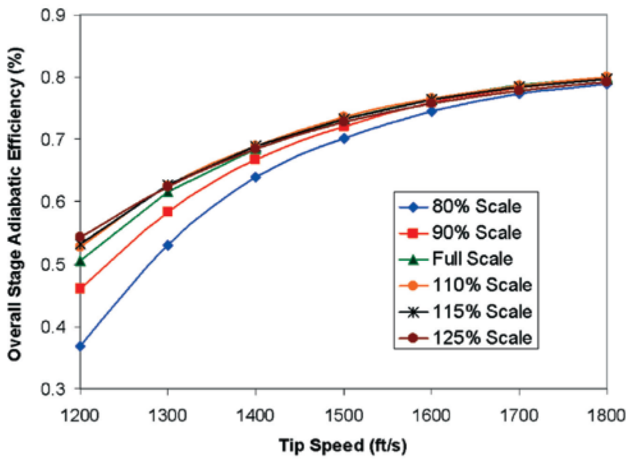

FIGURE 2. Adiabatic Efficiency as a Function of Tip Speed and Impeller Size

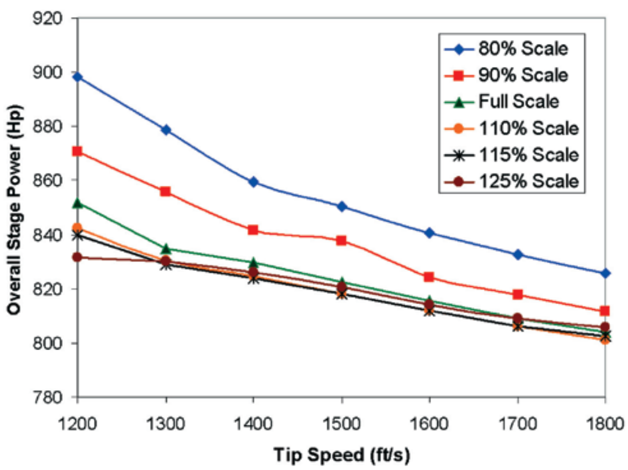

FIGURE 3. Gas Power as a Function of Tip Speed And Impeller Size

separate 3-stage machines, each compressing half of the total hydrogen inlet flow of 250,000 kg/day. The desired pressure rise of 217 psig was achieved at a polytropic efficiency of 74% and would require input power of approximately 2,430 HP for the three stages (or a total of 4,860 HP for the entire frame).

The multi-stage frame was further investigated for the possible benefits of placing intercoolers between stages 1 and 2 and 2 and 3. The addition of intercoolers reduced the power required in stage 2 by 5% and stage 3 by 9%. Despite the small pressure drop experienced with intercoolers, the reduced power demand warrants their inclusion in the system. It was also observed that the use of intercoolers would make it easier to match performance between stages and, thus, contribute to higher overall system performance.

Finite element analysis was used to evaluate the stress distribution of the rotor design selected for the compressor. Four different materials with a wide range of elastic properties and strengths were selected and evaluated in this analysis. In addition, certain aspects of the design were varied in the analysis to evaluate the effects of geometric design on the stress distribution. In the analysis, a rotational speed of 60,000 rpm, which corresponds to a tip speed of nearly 1,800 ft/s, was used.

The stress analysis was performed following mesh generation and assigning appropriate boundary conditions. One end of the shaft was assumed to be fixed and the rotational speed of 60,000 rpm was applied to the entire body as an internal load. Therefore, only centrifugal stress was considered. The analysis was performed for a high strength stainless steel, high strength aluminum alloy, a titanium alloy and high strength steel. The overall von Mises stress distribution, the distribution in the cross section and the distribution on one of the rotor disks are shown in Figure 4 for one of the materials. The analysis confirmed that the stress concentration on the backside of the disk could be greatly improved by modifying the transition radius. These results and similar future analyses will be used to down select the rotor material and the specific geometric design for the rotor.

The analyses performed for stress distribution in pure rotation was extended by the addition of a rotational torque. In addition to the rotational speed of 60,000 rpm, a torque corresponding to 5,500 HP at 60,000 rpm was applied to the free end of the rotor. In this analysis both the centrifugal and torsional stresses were analyzed. The stress distribution, however, remained about the same as those without the torque.

Rotor dynamic analysis was performed on the selected rotor geometry to determine the speed at which the first critical bend occurs. The model is shown in Figure 5 where the unbalanced forces were included to match the rotor responses at two sensor locations measured from previous experiments. The predicted rotor response at the sensor locations and the bending moments at the bearing locations are shown in Figure 6 for one of the analyses. The analysis predicted that, in this case, the first critical bend would occur at about 35,000 rpm. The finite element model was modified by adding two grounded springs attached to the shaft at each bearing location to constrain the two perpendicular radial directions instead of fixing the shaft at one end. Appropriate bearing stiffness values were assigned as the spring constants and the bending moments at the first bending critical speed from the rotor dynamic analysis were applied to the shaft at the two bearing locations. The analysis was performed for a rotational speed of 60,000 rpm. Therefore, in this analysis both the centrifugal and bending stresses were included. The results indicated that the effect of adding the bending moment was insignificant and the stress values remained practically unchanged. Therefore, it is concluded that

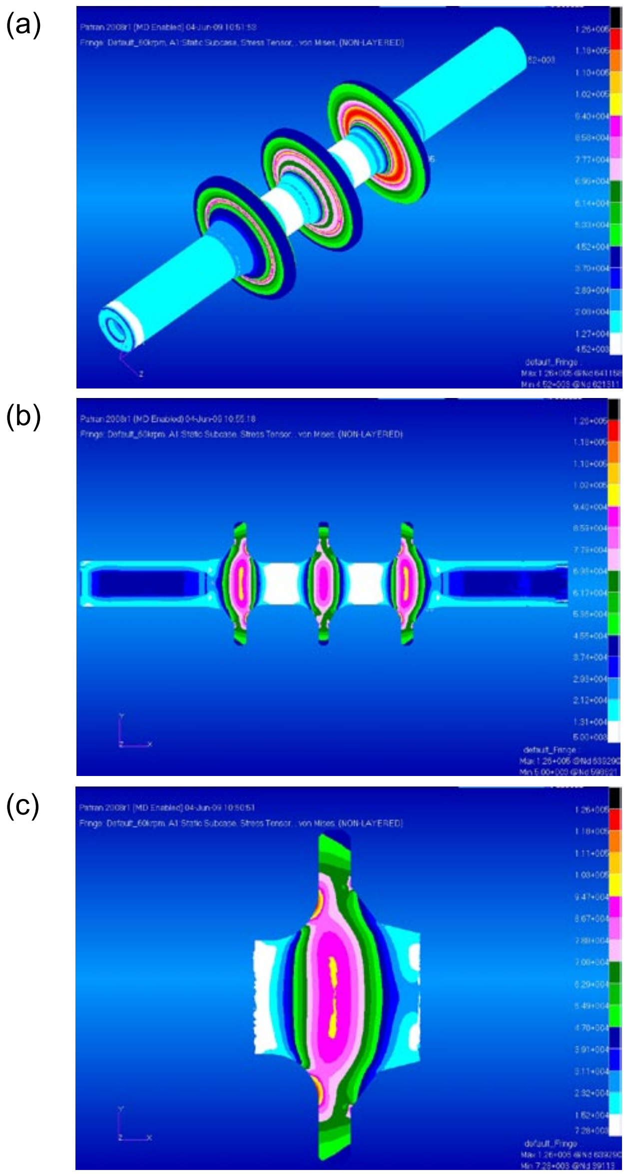

FIGURE 4. Typical Stress Distribution in the Rotor due to Rotation: (a) Surface Stresses, (b) Cross-Section, and (c) Cross-Section in one Disk.

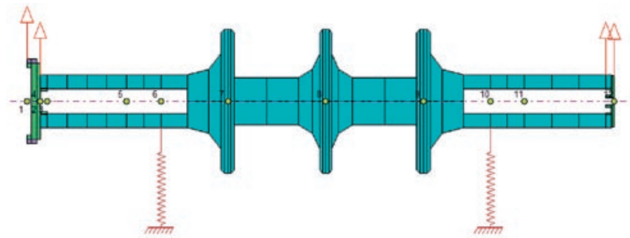

FIGURE 5. Model of the Rotor for Rotor Dynamic Analysis

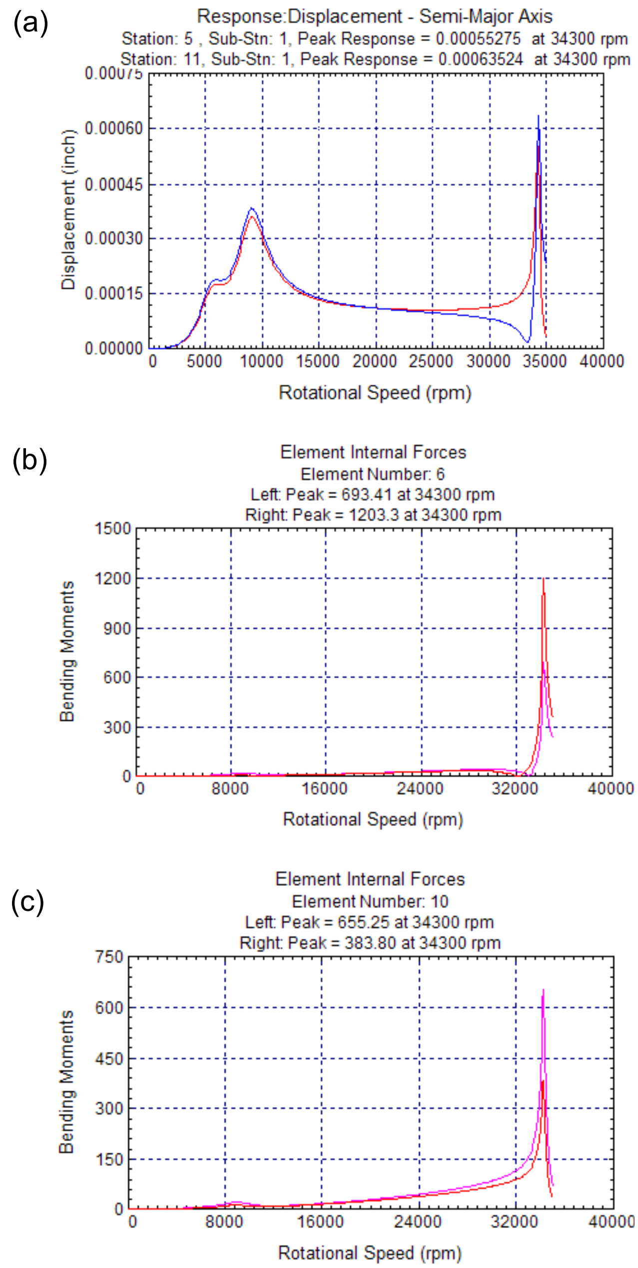

FIGURE 6. Predicted Results from the Rotor Dynamic Analysis: (a) Displacements, (b) Moments in one Bearing Location, and (c) Moments in the other Bearing Location.

the torque and bending moments are carried by the core shaft only and have no effect on the disks, which mainly carry the centrifugal stress.

Conclusions and Future Directions

The design requirements for compression of hydrogen with a flow of 500,000 to 1,000,000 kg/day with an output pressure of 1,200 psig were reviewed. A preliminary compressor design with seven compression stages was completed. The impeller diameter and the desired rotational speeds were established. Stress analysis was conducted to determine the range of stresses developed in the rotor for several selected materials. It was concluded that the addition of torque and bending moments due to passing through the first bending mode are carried by the core shaft only and have no effect on the disks, which mainly carry the centrifugal stress. The following tasks are planned for the remainder of Fiscal Years 2009 and 2010:

- Refine multi-stage/multi-frame compressor concept
- Select one stage for detailed design and test
- Conduct preliminary design review with DOE
- Conduct detailed design

FY 2009 Publications/Presentations

1. "Oil-Free Compression for Hydrogen Delivery and Transportation," Hydrogen Delivery Technology Team Meeting, August 2008, Columbia, MD.

2. "Oil-Free Centrifugal Hydrogen Compression Technology Demonstration," DOE Hydrogen Program Annual Review and Peer Evaluation Meeting, May 2009, Arlington, VA.