# 2014 DOE Hydrogen Program Merit Review

## Development of a Centrifugal Hydrogen Pipeline Gas Compressor

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**Concepts NREC (*CN*)**

**June 17, 2014**

Project ID#: PD017

There is a patent pending regarding the subject matter of this presentation.

# Project Overview

## Timeline

- **Project Start: June 1, 2008**
- **Project End: November 2012**
- **Percent Complete: Ph. I and Ph. II - 100%; Ph. III in Progress)**

## Budget

- **Total Project Funding**
  - DOE Share: $3,352,507
  - Contractor Share: $850,055
- **FY14 Funding (Phase III)**
  - A No Cost Extension with Total Project Expenditure to date: $3.19M

## Barriers/Tech. Objectives

- Pipeline delivery of pure (99.99%) hydrogen at <$1/GGE with 98% hydrogen efficiency
- Reduce initial capital equipment and O&M cost
- Reduce compressor module footprint & increase reliability; reduce R&D risk – utilize commercially available, state-of-the-art components

## Project Lead

- Concepts NREC (Chelmsford, MA, and Wilder, VT)

## Project Partners

- Texas A&M University (TAMU) (Materials Testing)
- HyGen Industries (Hydrogen Industry Consultant)

## Technical Collaboration

- Air Products and Chemicals, Inc.
- Sandia National Lab, Argonne National Lab, Savannah River National Lab
- Artec Machine Systems (gearbox), RMT (Bearings), Flowserve (shaft seal), Tranter HX, Hyundai (Motor)

There is a patent pending regarding the subject matter of this presentation.

# Hydrogen Pipeline Compressor Project Objectives – Relevance

- **Demonstrate Advanced Centrifugal Compressor System for High-pressure Hydrogen Pipeline Transport to Support[1]**
  - Delivery of 100,000 to 1,000,000 kg/day of pure hydrogen to forecourt station at less than $1/GGE with less than 0.5% leakage and with pipeline pressures of $1200^{+}$ psig
  - Reduction in initial system equipment cost to less than $6.3 million which is the uninstalled cost for a hydrogen pipeline based on DOE's HDSAM 2.0 Economics Model
  - Reduction in Operating & Maintenance Costs via improved reliability
    - DOE's Model also indicates $O&M cost of 3% of installed cost per year, or $0.01/kWhr by 2017
    - Improved reliability eliminates the need for system redundancies
  - Reduction in system footprint

**1. Reference: Delivery Section (Sec. 3.2) of the *"Hydrogen, Fuel Cells and Infrastructure Technologies Program Multi-year Research, Development, and Demonstration Plan"***

# A Three-Phase Program Approach

| Phase I<br>Initial Design (COMPLETED)<br>(06/2008 to 12/2009) | Phase II Detailed Design<br>(COMPLETED)<br>(01/2010 to 12/2010) | Phase III System Validation Testing<br>(IN PROGRESS) |
|---|---|---|
| • Initial design criteria and performance specifications<br>• Subsystems Modeling: aerodynamic and structural analysis of compressor<br>• Initial integrated systems analysis<br>• Initial design and cost analysis<br>• Final design specifications<br>• Materials and/or coatings investigated for use in high-pressure hydrogen environment<br>• Revised Phase II Program Plan | • Detailed subsystems modeling<br>• Detailed integrated systems analysis<br>• Critical components design, testing, and development<br>• Detailed integrated design of full-scale and laboratory validation systems<br>• Detailed cost analysis of full-scale system | • Component Procurement<br>• One-stage centrifugal compressor system assembly<br>• Performance evaluation test plan<br>• Lab testing and system maturation<br>• Final design of full-scale system completed<br>• Field demonstration program plan prepared |

# Project Engineering Approach – 1
# Innovative Compressor Design

- Technical Approach
  - Utilize state-of-the-art aerodynamic/structural analyses to develop a high-performance centrifugal compressor system able to provide high-pressure ratios under acceptable material stresses.
  - Utilize proven bearings and seal technology to reduce developmental risk and increase system reliability at a competitive cost.
  - Utilize acceptable practice for high-speed gear materials, tip speeds, and loadings.
  - With project and industrial collaborators, prepare an implementation plan that can provide for near-term industrial pipeline applications.
- Methodology
  - Investigate and prioritize alternative system configurations using operating conditions that meet initial capital and operational costs to meet near-term applications.
  - Identify critical engineering constraints of commercially available components and operational limitations of state-of-the-art materials, compatible with hydrogen to increase the range of safe compressor operating speeds.
  - Design and test critical rotor aerodynamics and material components under design conditions, and demonstrate full-scale components in an integrated compressor system.

# Project Engineering Approach – 2
# Primary Engineering Challenge

## The Engineering Challenge

- Design centrifugal compressor with highest acceptable pressure ratio and thermodynamic efficiency per stage to minimize system size, complexity, and cost, and to maximize system performance and reliability.

## Solution

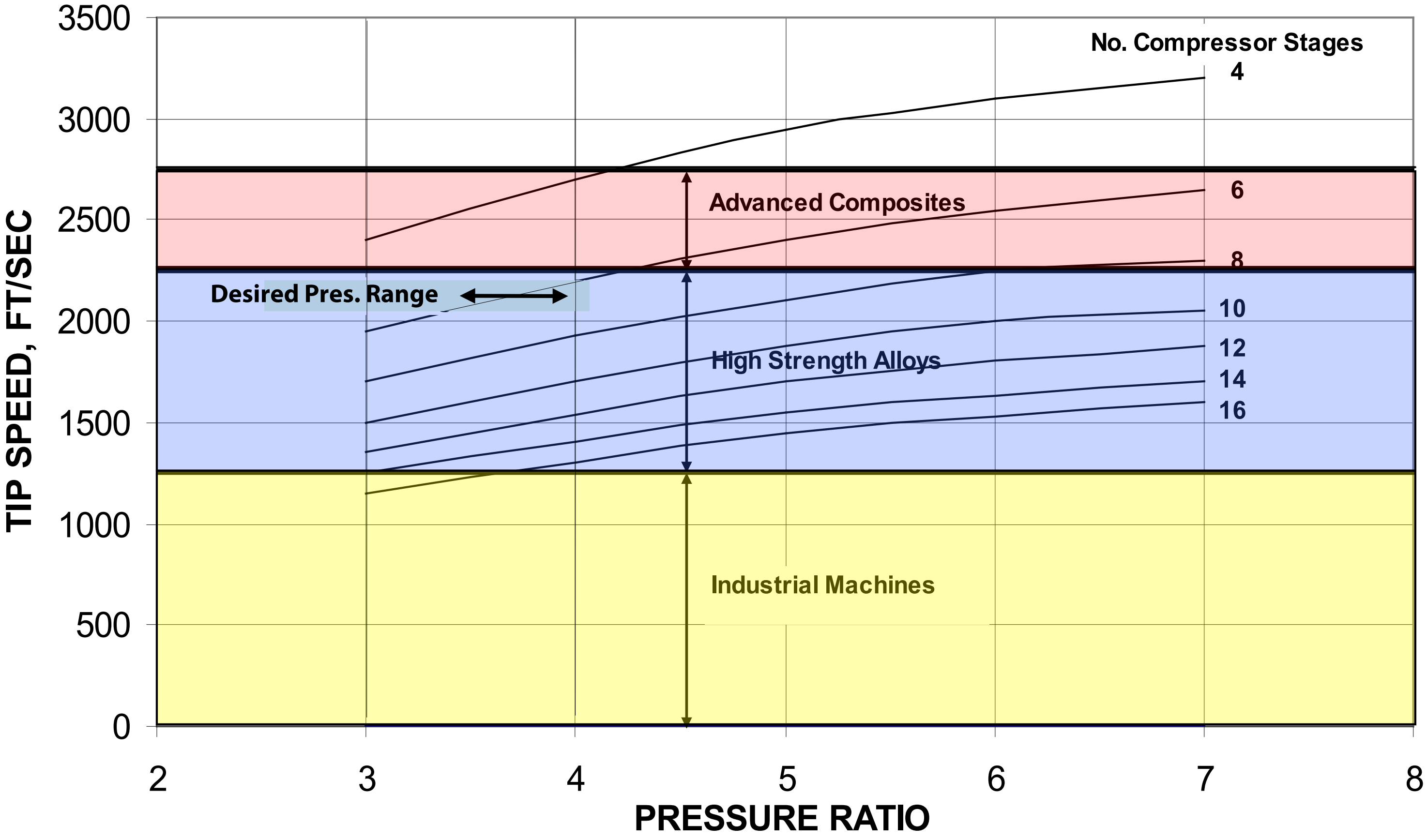
- Maximize centrifugal compressor tip speed within stress limitations of material.
  - Pressure ratio is proportional to $rpm^2$ x $radius^2$, so small increase in tip speed results in significant increases in pressure.
  - Maximum thermodynamic efficiency is typically achieved at high operating tip speeds.
- Utilize advanced diffuser systems to maximize recovery of dynamic head into static pressure.

## Constraints

- High operating speeds increase impeller material stresses.
  - Stress is also proportional to $rpm^2$ x $radius^2$ x material density. Therefore, pressure rise is limited by maximum stress capability of impeller material.
- Need to select materials that are not significantly affected by hydrogen embrittlement.
- Limited number of materials that have high strength to material density ratio and are resistant to hydrogen embrittlement.

# Project Engineering Approach – 3
# Operational Design Envelope

## Design Options for Alternative Operating Conditions



# Summary of DOE Target/Goals and Project Accomplishments

**Progress Towards Meeting Technical Targets for Delivery of Hydrogen via Centrifugal Pipeline Compression**

{Note: Letters correspond to DOE's 2007 Technical Plan-Delivery Sec. 3.2-page 16}

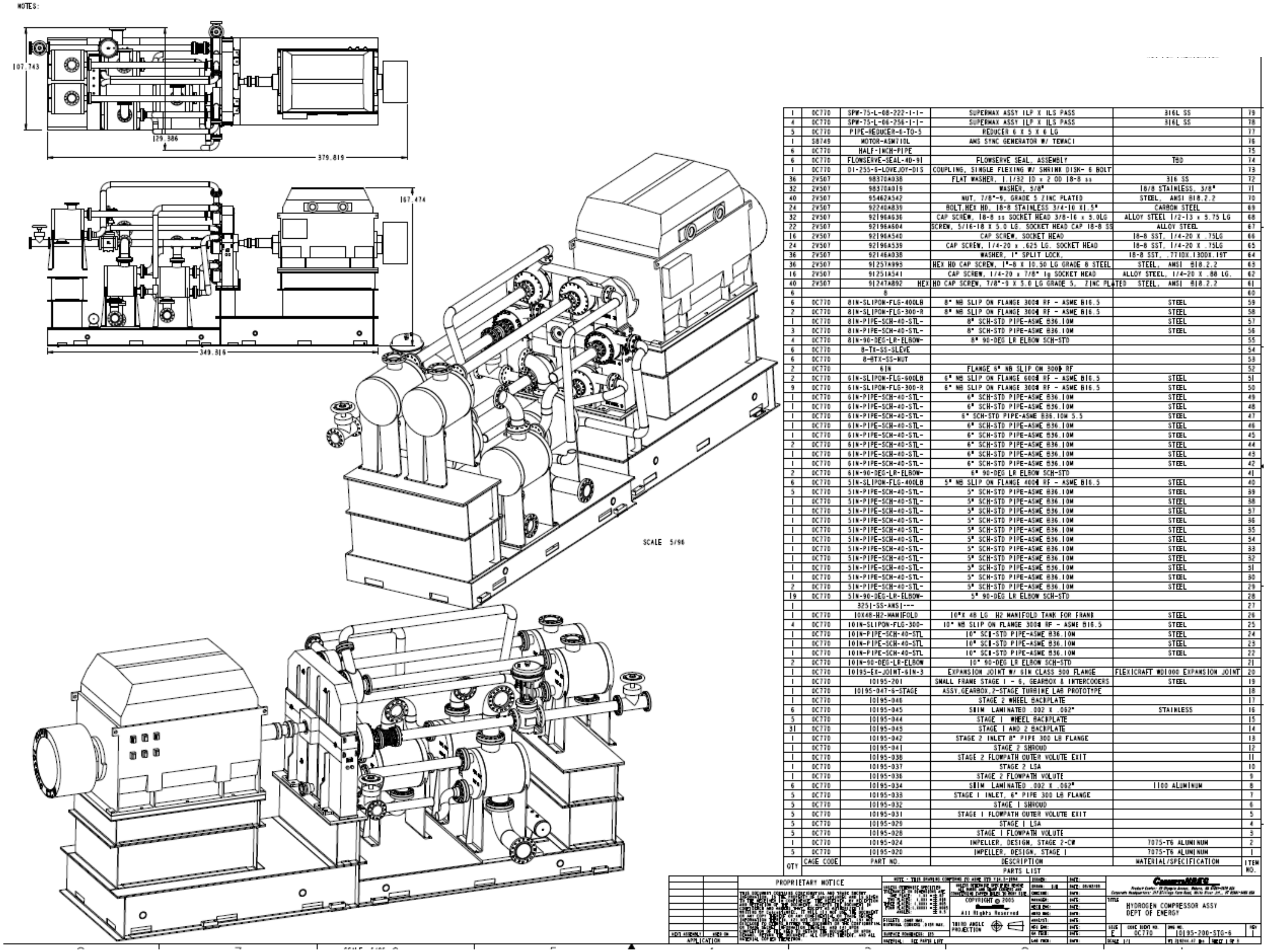
| Characteristic | Units | DOE Target | Project Accomplishment | STATUS |
|---|---|---|---|---|
| Hydrogen Efficiency (f) | [btu/btu] | 98% | 98% | Objective Met |
| Hyd. Capacity (g) | Kg/day | 100,000 to 1,000,000 | 240,000 | Objective Met |
| Hyd. Leakage (d) | % | < .5 | 0.2 (per Flowserve Shaft Seal Spec.) | Objective Met |
| Hyd. Purity (h) | % | 99.99 | 99.99 (per Flowserve Shaft Seal Spec) | Objective Met |
| Discharge Pressure (g) | psig | >1000 | 1285 | Objective Met |
| Comp. Package Cost (g) | $M | 6.0 +/- 1 | 4.5 +/- 0.75 | Objective Met |
| Main. Cost (Table 3.2.2) | $/kWhr | 0.007 | 0.005 (per CN Analysis Model) | Objective Met |
| Package Size (g) | sq. ft. | 350 (per HyGen Study) | 260 (per CN Design) | Objective Met |
| Reliability (e) | # Sys.s Req.d | Eliminate redundent system | Modular sys.s with 240K kg/day with no redundency req.d | Objective Met |

**In Summary: The original DOE proposal requirements were satisfied with the Detailed Design of a Pipeline Hydrogen Compressor that Utilizes all State-of-the-Art AND Commercially Available Components including: High Speed Centrifugal Compressor, Gearbox, Intercooler, Tilt-Pad Bearings, Oil Free Dry Gas Shaft Seal and Controls**

**Result of Research Development: A Pipeline-capacity, Hydrogen Centrifugal Compressor can be made available NOW to meet the Hydrogen Economy needs of the future !**

# Hydrogen Compressor Phase II Detailed Design Accomplishment:

## 240,000 kg/day (6.1 Lbm/s); 350 to 1285 psig; 6300 kWe



| QTY | CAGE CODE | PART NO. | DESCRIPTION | MATERIAL/SPECIFICATION | ITEM NO. |
|---|---|---|---|---|---|
| 1 | 0C770 | SPW-75-L-08-222-1-1- | SUPERMAX ASSY ILP X ILS PASS | 316L SS | 79 |
| 4 | 0C770 | SPW-75-L-06-256-1-1- | SUPERMAX ASSY ILP X ILS PASS | 316L SS | 78 |
| 5 | 0C770 | PIPE-REDUCER-6-TO-5 | REDUCER 6 X 5 X 6 LG | | 77 |
| 1 | 58749 | MOTOR-ASM710L | AMS SYNC GENERATOR W/ TEWAC1 | | 76 |
| 6 | 0C770 | HALF-INCH-PIPE | | | 75 |
| 6 | 0C770 | FLOWSERVE-SEAL-40-91 | FLOWSERVE SEAL, ASSEMBLY | TBD | 74 |
| 1 | 0C770 | DI-255-6-LOVEJOY-DIS | COUPLING, SINGLE FLEXING W/ SHRINK DISK- 6 BOLT | | 73 |
| 36 | 2V507 | 98370A038 | FLAT WASHER, 1.1/32 ID x 2 OD 18-8 ss | 316 SS | 72 |
| 32 | 2V507 | 98370A019 | WASHER, 3/8" | 18/8 STAINLESS, 3/8" | 71 |
| 40 | 2V507 | 95462A542 | NUT, 7/8"-9, GRADE 5 ZINC PLATED | STEEL, ANSI B18.2.2 | 70 |
| 24 | 2V507 | 92240A839 | BOLT.HEX HD. 18-8 STAINLESS 3/4-10 X1.5" | CARBON STEEL | 69 |
| 32 | 2V507 | 92196A636 | CAP SCREW, 18-8 ss SOCKET HEAD 3/8-16 x 5.0LG | ALLOY STEEL 1/2-13 x 5.75 LG | 68 |
| 22 | 2V507 | 92196A604 | SCREW, 5/16-18 X 5.0 LG. SOCKET HEAD CAP 18-8 SS | ALLOY STEEL | 67 |
| 16 | 2V507 | 92196A540 | CAP SCREW, SOCKET HEAD | 18-8 SST, 1/4-20 X .75LG | 66 |
| 24 | 2V507 | 92196A539 | CAP SCREW, 1/4-20 x .625 LG. SOCKET HEAD | 18-8 SST, 1/4-20 X .75LG | 65 |
| 36 | 2V507 | 92146A038 | WASHER, 1" SPLIT LOCK, | 18-8 SST, .771OX.130OX.19T | 64 |
| 36 | 2V507 | 91257A993 | HEX HD CAP SCREW, 1"-8 X 10.50 LG GRADE 8 STEEL | STEEL, ANSI B18.2.2 | 63 |
| 16 | 2V507 | 91251A541 | CAP SCREW, 1/4-20 x 7/8" lg SOCKET HEAD | ALLOY STEEL, 1/4-20 X .88 LG. | 62 |
| 40 | 2V507 | 91247A892 | HEX HD CAP SCREW, 7/8"-9 X 5.0 LG GRADE 5, ZINC PLATED | STEEL, ANSI B18.2.2 | 61 |
| 6 | | 8 | | | 60 |
| 6 | 0C770 | 8IN-SLIPON-FLG-400LB | 8" NB SLIP ON FLANGE 300# RF - ASME B16.5 | STEEL | 59 |
| 2 | 0C770 | 8IN-SLIPON-FLG-300-R | 8" NB SLIP ON FLANGE 300# RF - ASME B16.5 | STEEL | 58 |
| 1 | 0C770 | 8IN-PIPE-SCH-40-STL- | 8" SCH-STD PIPE-ASME B36.10M | STEEL | 57 |
| 3 | 0C770 | 8IN-PIPE-SCH-40-STL- | 8" SCH-STD PIPE-ASME B36.10M | STEEL | 56 |
| 4 | 0C770 | 8IN-90-DEG-LR-ELBOW- | 8" 90-DEG LR ELBOW SCH-STD | | 55 |
| 6 | 0C770 | 8-TX-SS-SLEVE | | | 54 |
| 6 | 0C770 | 8-BTX-SS-NUT | | | 53 |
| 2 | 0C770 | 6IN | FLANGE 6" NB SLIP ON 300# RF | | 52 |
| 2 | 0C770 | 6IN-SLIPON-FLG-600LB | 6" NB SLIP ON FLANGE 600# RF - ASME B16.5 | STEEL | 51 |
| 9 | 0C770 | 6IN-SLIPON-FLG-300-R | 6" NB SLIP ON FLANGE 300# RF - ASME B16.5 | STEEL | 50 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 49 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 48 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M 5.5 | STEEL | 47 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 46 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 45 |
| 2 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 44 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 43 |
| 1 | 0C770 | 6IN-PIPE-SCH-40-STL- | 6" SCH-STD PIPE-ASME B36.10M | STEEL | 42 |
| 2 | 0C770 | 6IN-90-DEG-LR-ELBOW- | 6" 90-DEG LR ELBOW SCH-STD | | 41 |
| 6 | 0C770 | 5IN-SLIPON-FLG-400LB | 5" NB SLIP ON FLANGE 400# RF - ASME B16.5 | STEEL | 40 |
| 5 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 39 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 38 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 37 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 36 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 35 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 34 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 33 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 32 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 31 |
| 1 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 30 |
| 2 | 0C770 | 5IN-PIPE-SCH-40-STL- | 5" SCH-STD PIPE-ASME B36.10M | STEEL | 29 |
| 19 | 0C770 | 5IN-90-DEG-LR-ELBOW- | 5" 90-DEG LR ELBOW SCH-STD | | 28 |
| 1 | | 3251-SS-ANSI--- | | | 27 |
| 1 | 0C770 | 10X48-H2-MANIFOLD | 10"X 48 LG H2 MANIFOLD TANK FOR FRAME | STEEL | 26 |
| 4 | 0C770 | 10IN-SLIPON-FLG-300- | 10" NB SLIP ON FLANGE 300# RF - ASME B16.5 | STEEL | 25 |
| 1 | 0C770 | 10IN-PIPE-SCH-40-STL | 10" SCH-STD PIPE-ASME B36.10M | STEEL | 24 |
| 1 | 0C770 | 10IN-PIPE-SCH-40-STL | 10" SCH-STD PIPE-ASME B36.10M | STEEL | 23 |
| 1 | 0C770 | 10IN-PIPE-SCH-40-STL | 10" SCH-STD PIPE-ASME B36.10M | STEEL | 22 |
| 2 | 0C770 | 10IN-90-DEG-LR-ELBOW | 10" 90-DEG LR ELBOW SCH-STD | | 21 |
| 1 | 0C770 | 10195-EX-JOINT-6IN-3 | EXPANSION JOINT W/ 6IN CLASS 300 FLANGE | FLEXICRAFT WD1000 EXPANSION JOINT | 20 |
| 1 | 0C770 | 10195-201 | SMALL FRAME STAGE 1 - 6, GEARBOX & INTERCOOLERS | STEEL | 19 |
| 1 | 0C770 | 10195-047-6-STAGE | ASSY,GEARBOX,2-STAGE TURBINE LAB PROTOTYPE | | 18 |
| 1 | 0C770 | 10195-046 | STAGE 2 WHEEL BACKPLATE | | 17 |
| 6 | 0C770 | 10195-045 | SHIM LAMINATED .002 X .062" | STAINLESS | 16 |
| 5 | 0C770 | 10195-044 | STAGE 1 WHEEL BACKPLATE | | 15 |
| 31 | 0C770 | 10195-043 | STAGE 1 AND 2 BACKPLATE | | 14 |
| 1 | 0C770 | 10195-042 | STAGE 2 INLET 8" PIPE 300 LB FLANGE | | 13 |
| 1 | 0C770 | 10195-041 | STAGE 2 SHROUD | | 12 |
| 1 | 0C770 | 10195-038 | STAGE 2 FLOWPATH OUTER VOLUTE EXIT | | 11 |
| 1 | 0C770 | 10195-037 | STAGE 2 LSA | | 10 |
| 1 | 0C770 | 10195-036 | STAGE 2 FLOWPATH VOLUTE | | 9 |
| 6 | 0C770 | 10195-034 | SHIM LAMINATED .002 X .062" | 1100 ALUMINUM | 8 |
| 5 | 0C770 | 10195-033 | STAGE 1 INLET, 6" PIPE 300 LB FLANGE | | 7 |
| 5 | 0C770 | 10195-032 | STAGE 1 SHROUD | | 6 |
| 5 | 0C770 | 10195-031 | STAGE 1 FLOWPATH OUTER VOLUTE EXIT | | 5 |
| 5 | 0C770 | 10195-029 | STAGE 1 LSA | | 4 |
| 5 | 0C770 | 10195-028 | STAGE 1 FLOWPATH VOLUTE | | 3 |
| 1 | 0C770 | 10195-024 | IMPELLER, DESIGN, STAGE 2-CW | 7075-T6 ALUMINUM | 2 |
| 5 | 0C770 | 10195-020 | IMPELLER, DESIGN, STAGE 1 | 7075-T6 ALUMINUM | 1 |

PARTS LIST

PROPRIETARY NOTICE

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THIRD ANGLE PROJECTION

TITLE
HYDROGEN COMPRESSOR ASSY
DEPT OF ENERGY

| SIZE | CAGE CODE | DWG NO. | REV |
|---|---|---|---|
| E | 0C770 | 10195-200-STG-6 | 1 |

There is a patent pending regarding the subject matter of this presentation.

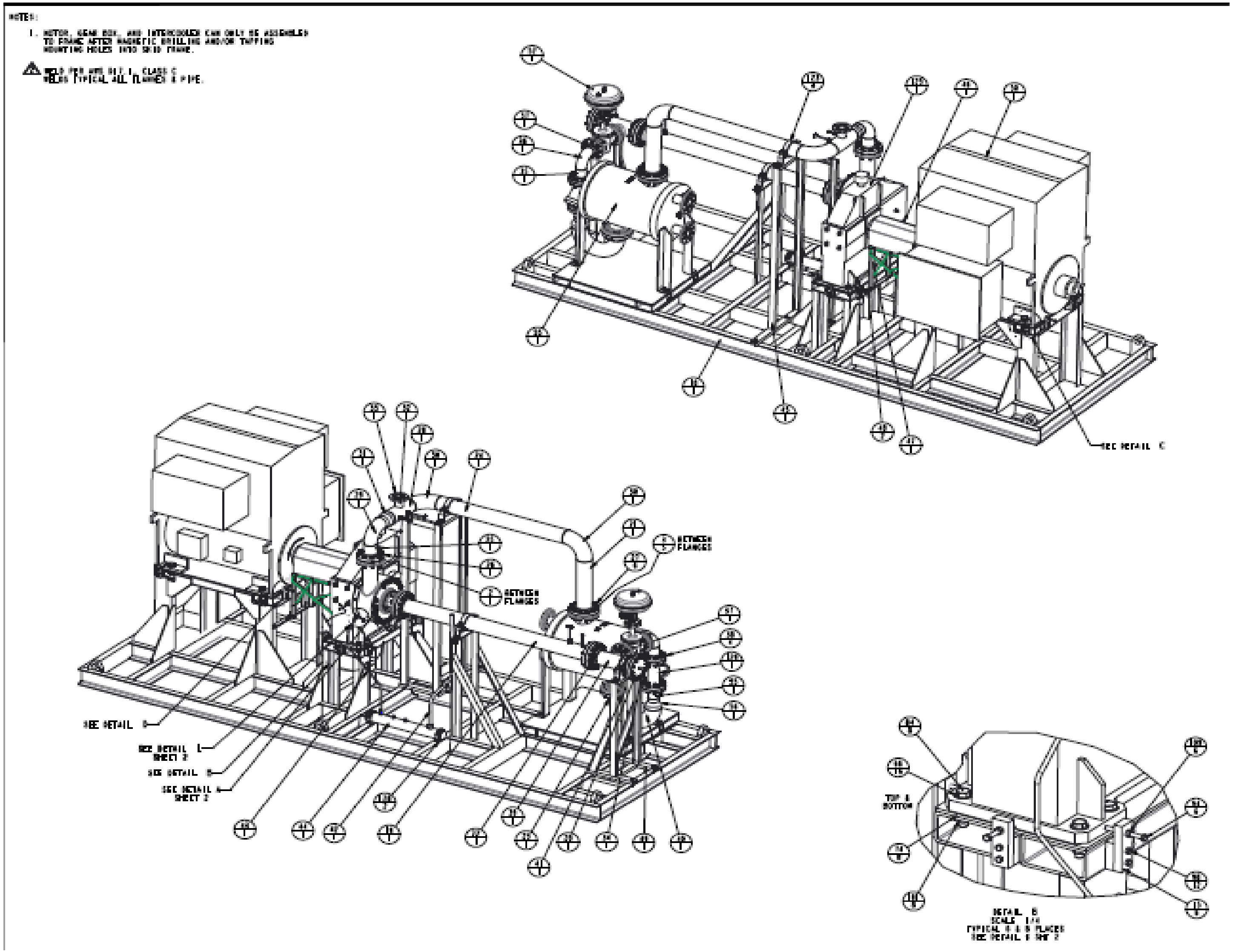
# Compressor Module Design Specifications and Major Components

- **Compressor design specifications for near-term gas industry and DOE infrastructure applications**
  - $P_{comp.}$= 350 psig to 1285 psig; flow rate = 240,000 kg/day
  - Six-stage, 60,000 rpm, 3.56 pressure ratio compressor
  - 7075-T6 aluminum alloy
  - Nitronic-50 pressure enclosure
  - Integral gearbox pinions driving 6 overhung impellers
- **Design of compressor's major mechanical elements completed and manufacturers selected**
  - Artec Machine Systems (Nova Gear, Ltd) gearbox with one-speed step gear operating at acceptable gear tip speeds and loads
  - RMT tilting-pad radial bearing designs confirmed for use
  - Flowserve gas face-seals confirmed to meet necessary specifications for hydrogen applications
- **Tranter Plate-type Heat Exchanger design meets specifications to cool hydrogen gas to 105°F between stages using 85°F water**

**In Summary: All compressor subsystems (from shaft seals to bearings to gearing to aluminum impellers) are available "near-term".**

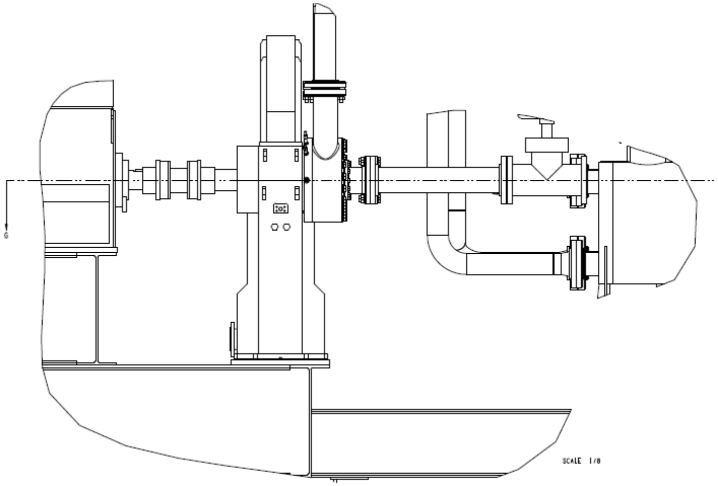
# Hydrogen Compressor Prototype:

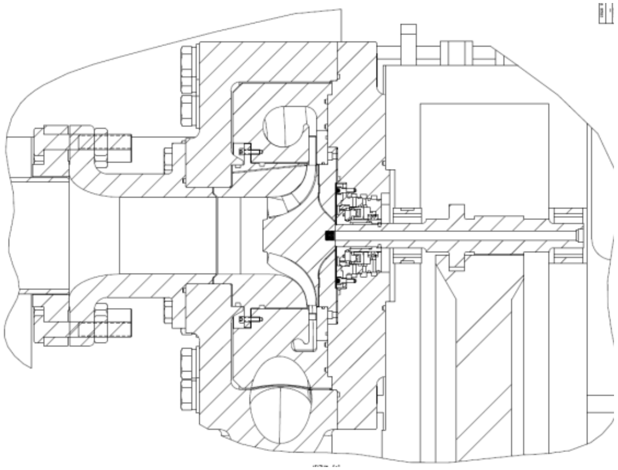
## 240,000 kg/day (6.1 Lbm/s); 350 to 450 psig; 1,100 kWe

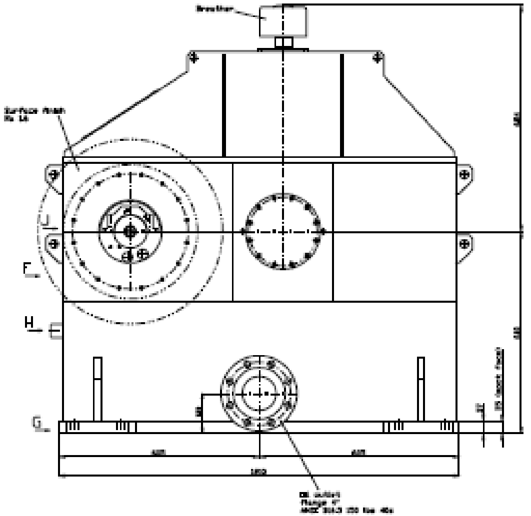


There is a patent pending regarding the subject matter of this presentation.

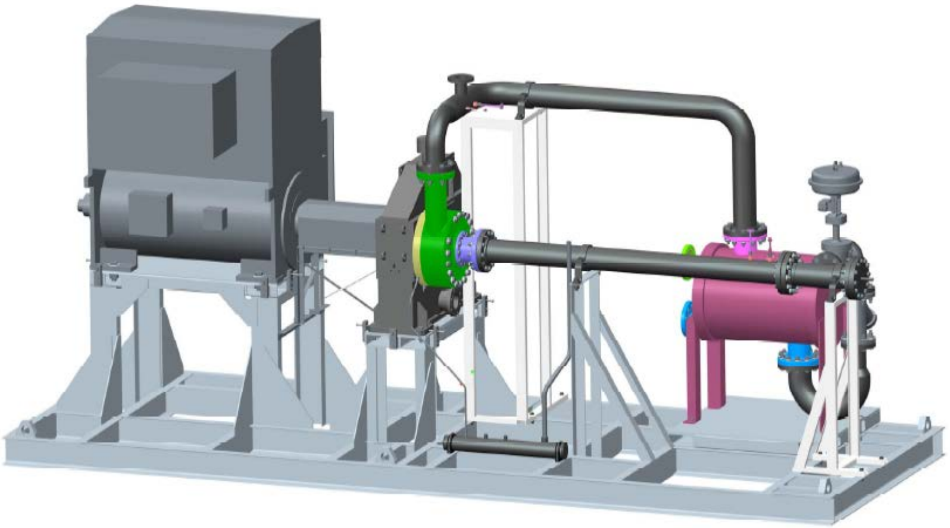
# One-step Gearbox for Prototype:
## 1100 kWe; 60,000 to 3600 rpm (16.67:1)







# Focus of Phase II Was Also the Design of a Laboratory Prototype



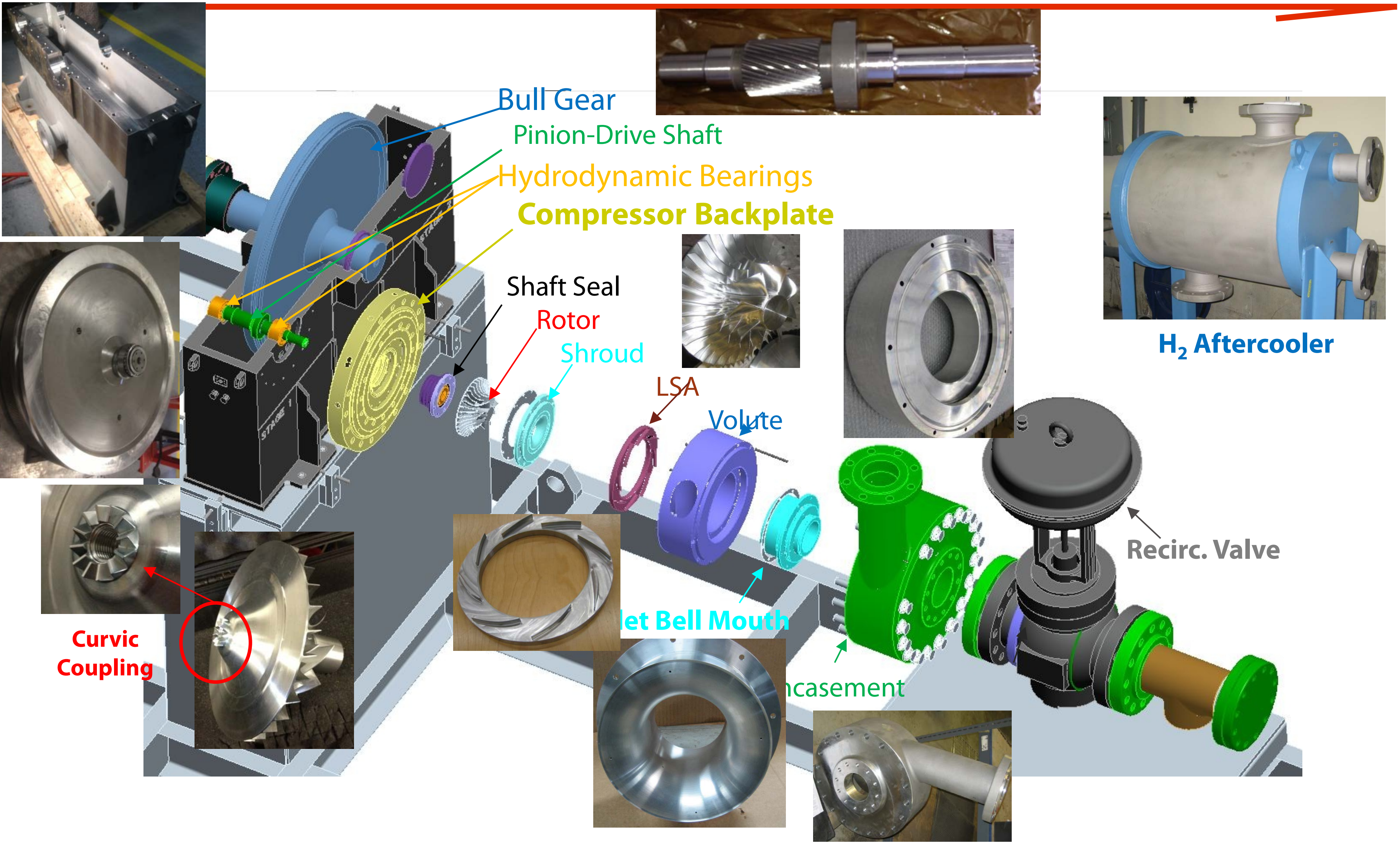
**The 1-Stage Compressor Module is 16 ft long, 8 ft wide, and 9 ft tall. based on the itemized weights shown here:**

| | |
|---|---|
| 1. 4160 Vac, 1500 hp Induction Motor (3600 rpm): | 7500 bf |
| 2. Artec Gearbox (3600 rpm) : | 4500 Lbf |
| 3. One, Compressor : | 2500 Lbf |
| 4. One, Intercooler: | 2500 Lbf |
| 5. 6” comp. out. piping (sch. 40, 20ft): | 500 Lbf |
| 6. 6” comp. in piping (sch. 40, 30ft): | 450 Lbf |
| 7. Fittings: | 700 Lbf |
| 8. Purge Tank (12” d. x 6 ft long): | 700 Lbf |
| 9. Base Frame and Support Pedestals: | 5000 Lbf |
| 10.Shut-Off/Recirc. (PRV) valve & Misc | 2,500 Lbf |

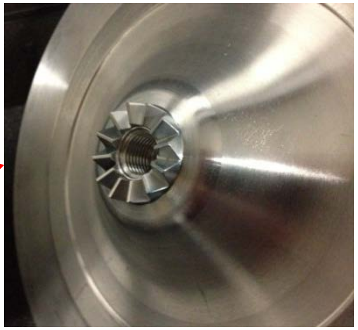
**PHASE III- PROTOTYPE SYSTEM COMPONENT PROCUREMENT, BUILD, & TEST:**

- **COMPLETED – P&I Diagram, Controls Specification, Safety Systems**
- **COMPLETED – All compressor components**
- **IN PROGRESS – Assembly of**
  - **Modified 1-stage Gearbox**
  - **PLC & Controls**
  - **Assembly of Prototype (as shown on left)**

# Detail of Prototype, One-stage Hydrogen Compressor Module

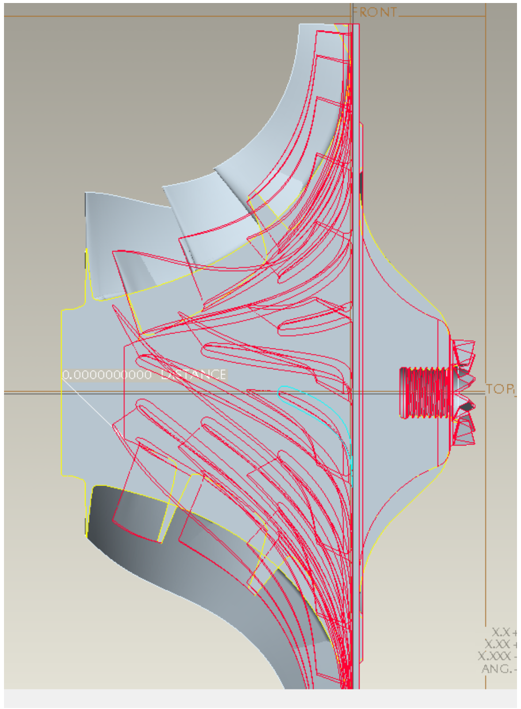


# Detailed Engineering Design for All Six Compressor Rotors Completed and First Stage Manufactured



**Curvic Spline Couples Rotor to Drive Shaft**

**First Stage of 6-Stage Compressor and Drive Shaft with Pinion and Thrust collar**



**Overlay of First and Sixth Stages for Size Comparison**

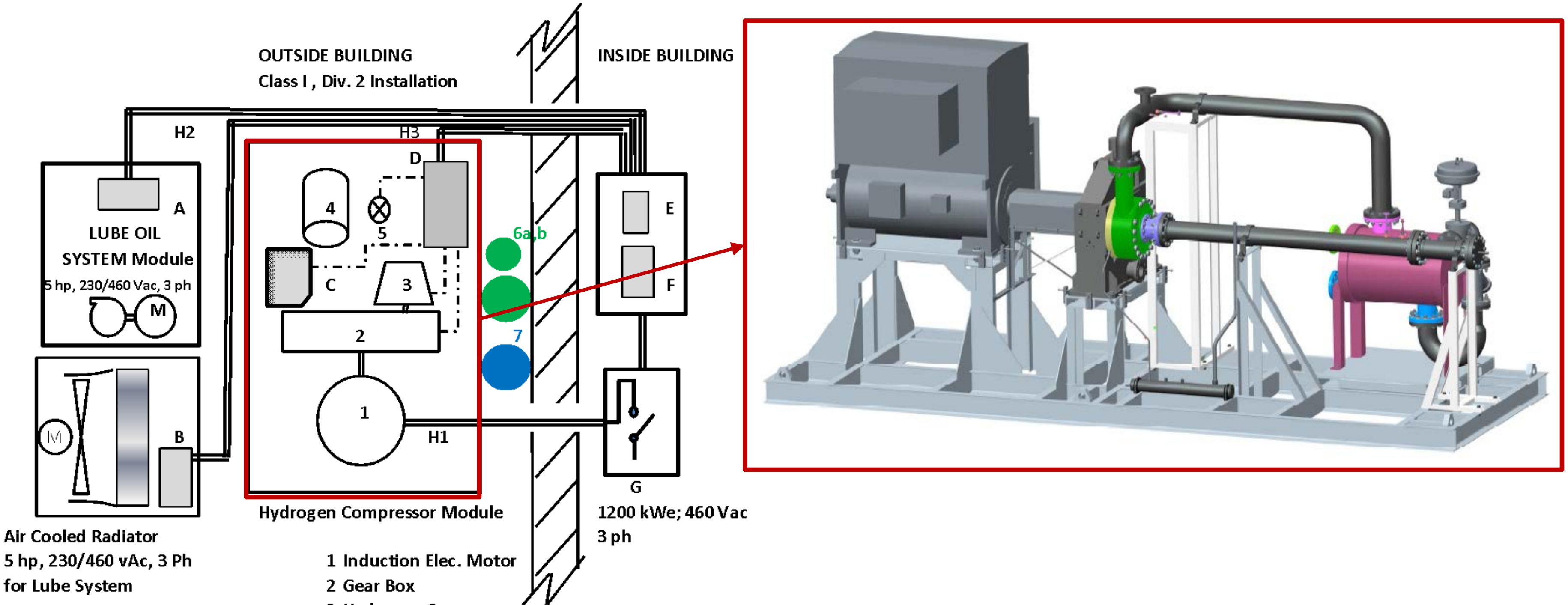
# Accomplishment and Progress :
## Compressor has been successfully spun to 10% overspeed for 15 minutes (66,000 rpm = 2300 ft/s tip speed)

**7075-T6 Aluminum (boreless) rotor shown after 5-axis machining; _CN_ and TAMU testing have confirmed compatibility of alum. alloy with hydrogen**

**Spin Test Successful:**

1. Fluorescence Penetrate Inspection indicated no micro-stress fractures or strain issues after
2. Structural analysis has also determined that there is not any concern for material creep at operating temperature (145 F) vs. 1,200 F melting temperature and stress
3. The low blade frequency and stress and the operating requirement of 24/7 duty for pipeline compressor applications eliminates any concern of material fatigue

# Prototype “Lab” Test Site Installation



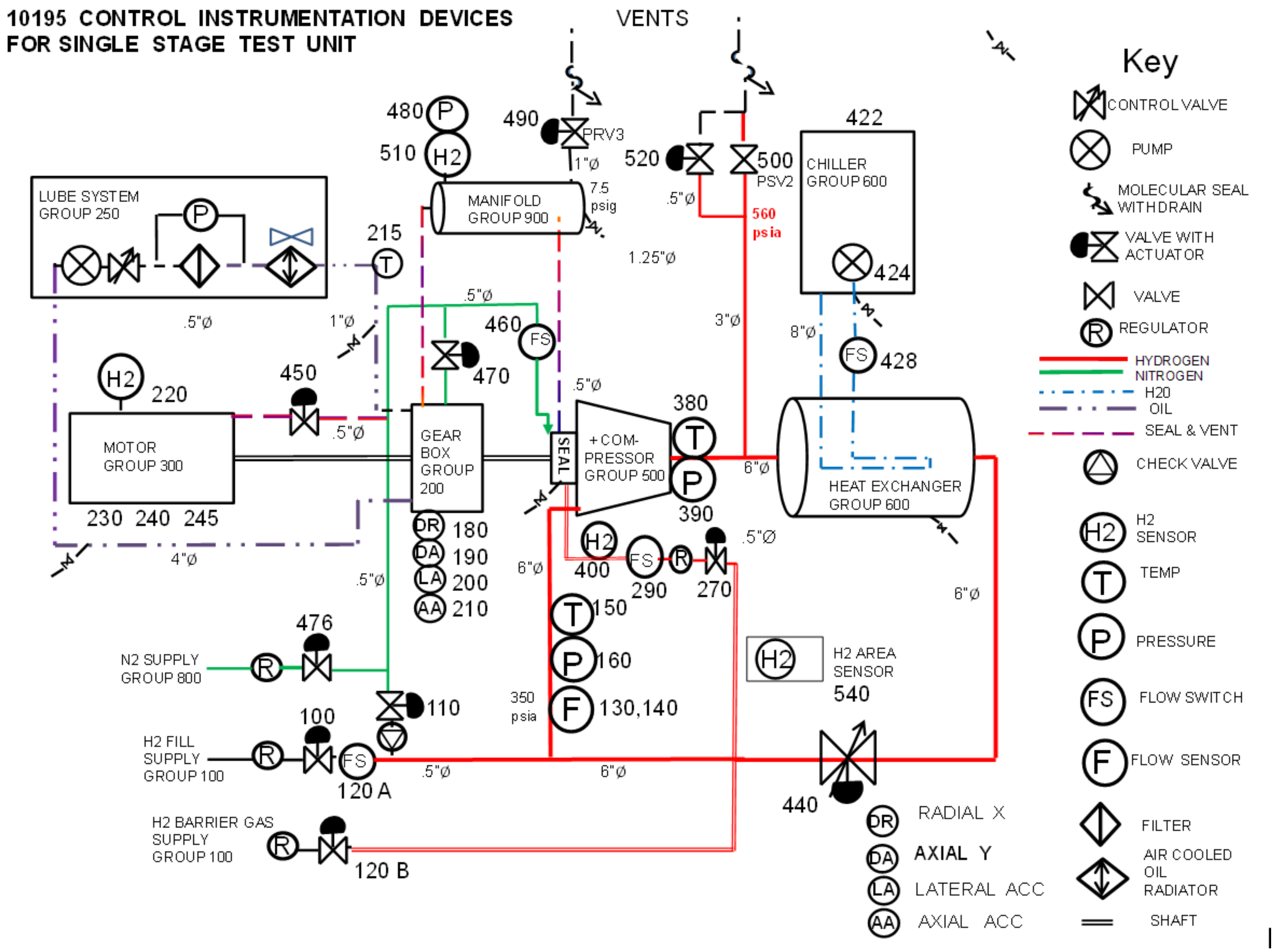
1 Induction Elec. Motor
2 Gear Box
3 Hydrogen Compressor
4 Intercooler
5 Recirc. Valve

6a, 6b Hydrogen Bottle Supply
7 Nitrogen Bottle Supply

A Lube Oil Control Panel with Disconnect from Concepts NREC
B Air Fan Motor Starter from Concepts NREC
C Hydrogen Monitor from Concepts NREC
D I/O Interconnect panel for connecting PT's, TT's and Compressor Vib. Monitoring
E Data Acquisition (combined with PLC ?)
F PLC
G Motor Disconnect and Soft Start
H1,2,3 Electrical Conduit for controls and power wiring

There is a patent pending regarding the subject matter of this presentation.

# Lab Prototype P&I Diagram



# Project Advisors & Collaborators: Strengths & Responsibilities of Partners

- **Air Products and Chemicals, Inc.**
  - Provides industrial gas user technical experience and gas industry specification data on major components: electric motor, hydrogen safety system, intercooler design, selection of materials of construction
- **Texas A&M University**
  - Provided material science expertise and coordination of materials testing with Sandia and Savannah River National labs
- **HyGen Industries**
  - Provides experience in hydrogen fueling infrastructure: pipeline and refueling station systems, has a database of customer-user engineering specifications. Assists in developing implementation plan for pipeline applications for hydrogen compressors

# Future Phase III Project Work in Progress

- **Phase III System Validation Testing**
  - Continue component procurement for the Lab Prototype, Single-stage hydrogen compressor system (Scheduled completion:  April 2014)
  - Assembly of the one-stage centrifugal compressor and closed-loop, lab prototype as a completely functioning compressor system (Scheduled Completion:  May 2014)
  - Install lab prototype system and conduct aerodynamic testing and assessment of mechanical integrity of the compressor system (Scheduled Completion:  July 2014)

There is a patent pending regarding the subject matter of this presentation.

# Project Summary

- **Relevance:** **An advanced pipeline compressor system has been designed that meets DOE's performance goals for:**
  - High reliability with 350 to 1200$^+$ psig compression of 240,000 kg/day at 98% hydrogen efficiency
  - footprint 1/4 to 1/3 the size of existing industrial systems at projected cost of less than 80% of DOE's target
- **Approach:** **Utilize state-of-the-art and acceptable engineering practices to reduce developmental risk and provide a near-term solution for the design of a viable hydrogen pipeline compressor:**
  - Aerodynamic/structural analyses for acceptable stresses in materials (7075-T6 Rotor, Nitride 31 Chrome Moly Shaft, & Nitronic-50 enclosure) compatible with hydrogen
  - Industrially proven bearings, seal technology, gearing, heat exchangers, and lube system
- **Tech. Accomplishments & Progress:** **Aerodynamic analysis and design of a cost-effective, six-stage centrifugal compressor and a one-stage full-power lab prototype have been completed; spin test of aluminum stage verifies its mechanical integrity, all commercially available compressor subsystems purchased. Research has demonstrated that a Hydrogen Pipeline Centrifugal Compressor is available NOW to meet the Hydrogen Economy requirements of the future!**
- **Technology Transfer/Collaboration:** **The collaborative team consisted of Air Products, an industrial technical experienced user of hydrogen compressors; a materials researcher, Texas A&M; a hydrogen refueling industry consultant, HyGen; and the coordinated technical support of several National Labs and major component manufacturers.**
- **Proposed Future Research:** **The laboratory testing of a closed-loop, one-stage prototype hydrogen compressor system.**

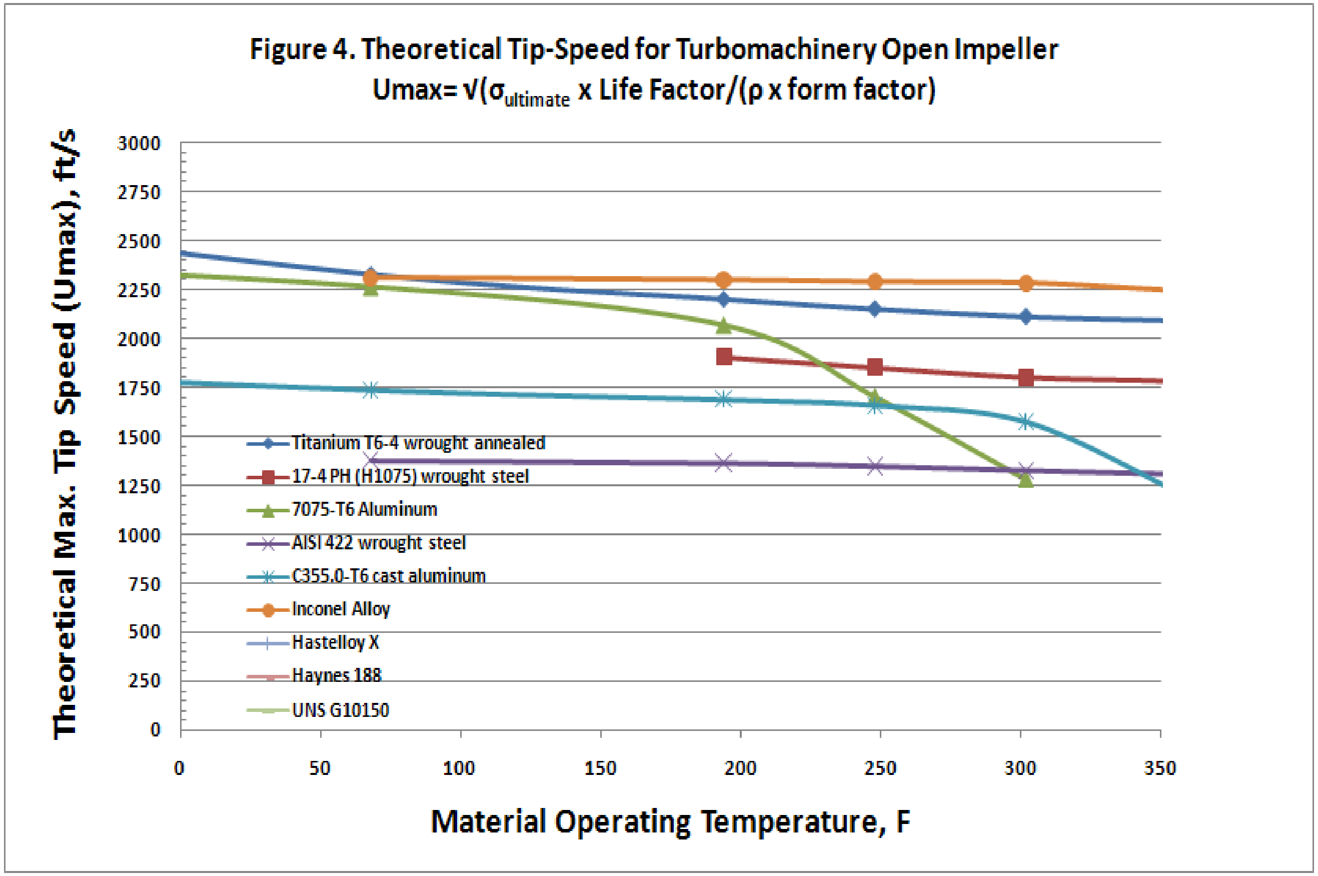
There is a patent pending regarding the subject matter of this presentation.

# Technical Back-Up Slides

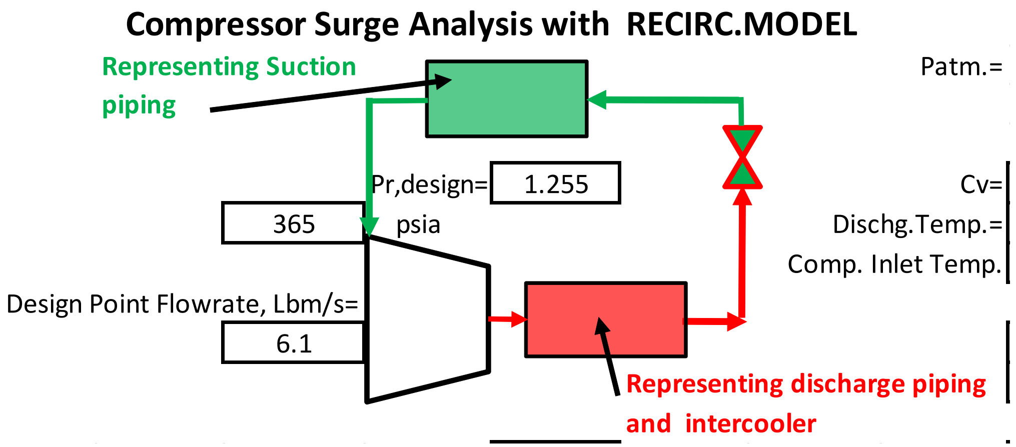
**The following slides are included here to provide additional support during the question and answer period.**

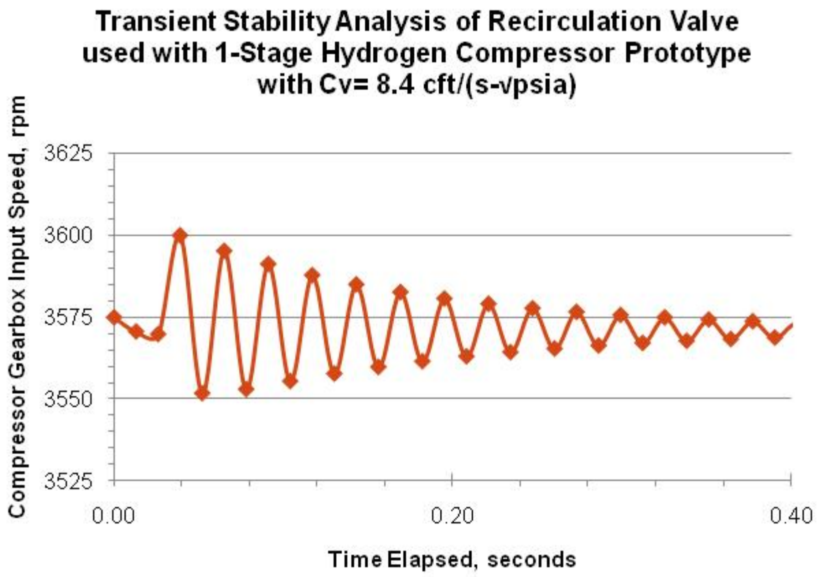
# Design Experience Associating Material Properties with Tip Speed of 2200 ft/s with Aluminum Alloy - 2

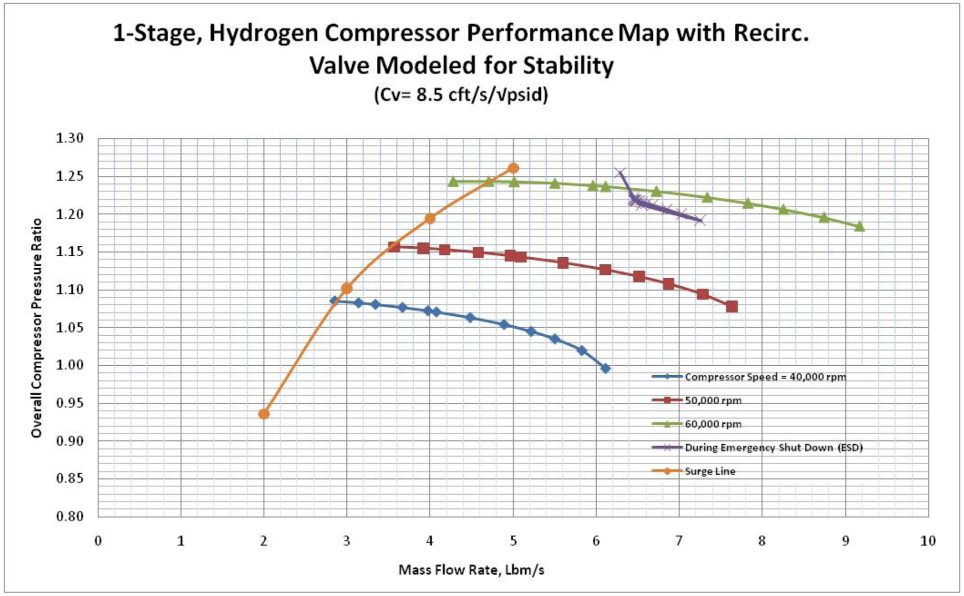
**Literature Survey (Rocketdyne Lab Tests for NASA) and reviews with materials researchers at national labs and private consultants indicate Aluminum Alloy shows no effect from hydrogen …. AND aluminum is an excellent structural material for high-speed impellers based on specific strength (ultimate strength/density)**



# Recirc. Control Valve Model Algorithm for Laboratory Prototype



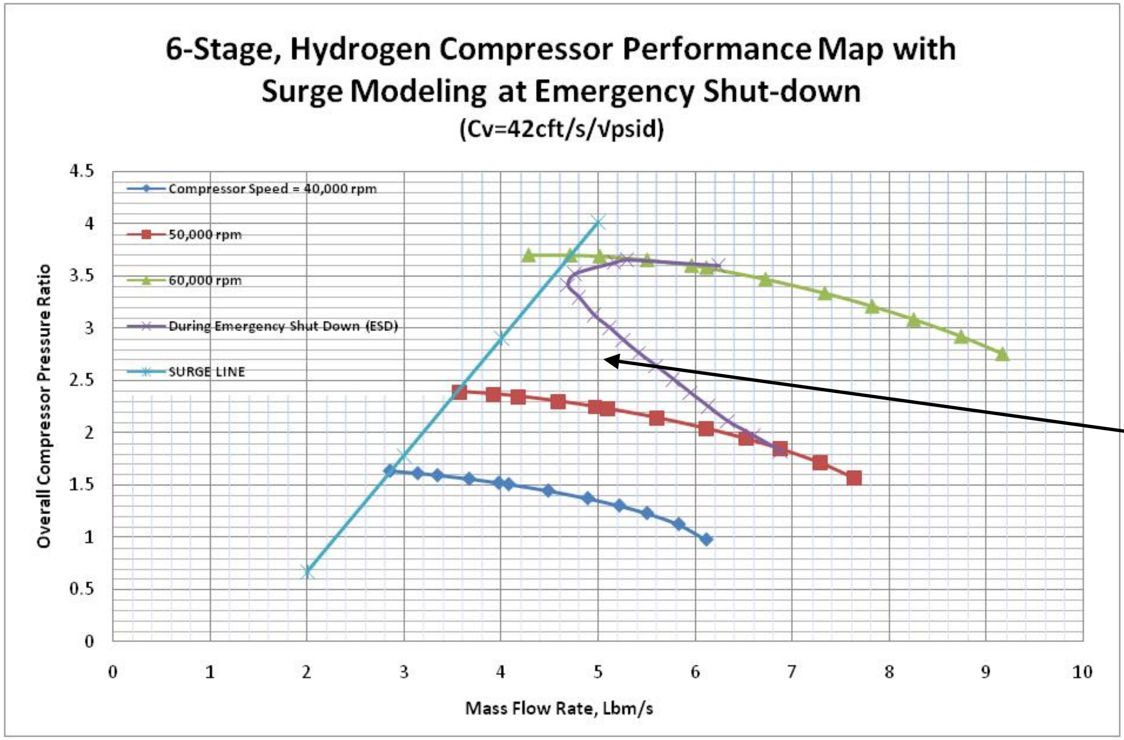




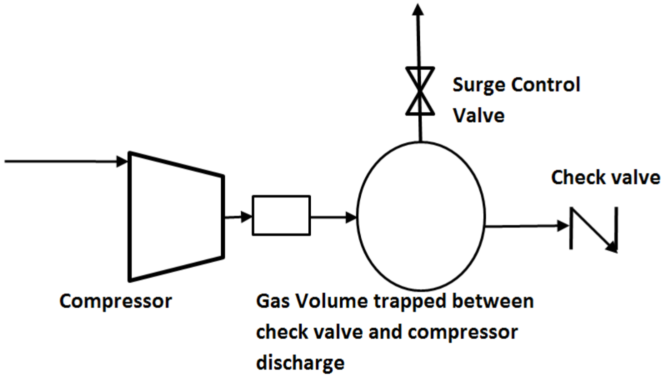
There is a patent pending regarding the subject matter of this presentation.

# Anti-Surge Control Model Algorithm for Emergency Shutdown

- **Enables the sizing of Anti-surge Control Valve and Downstream Piping**

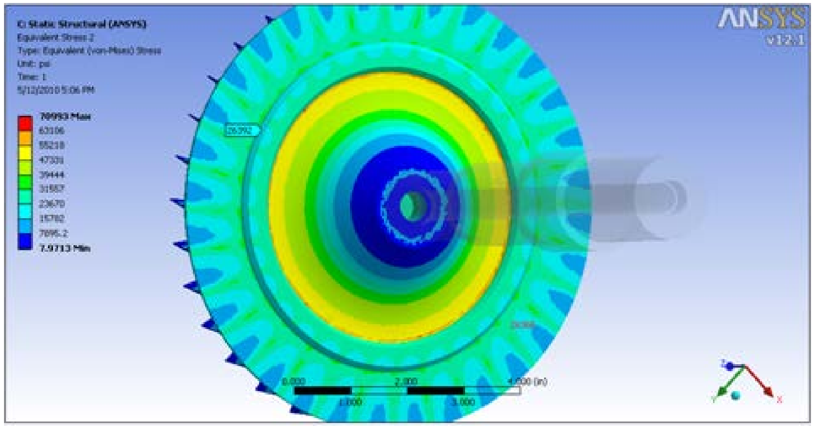

6-Stage, Hydrogen Compressor Performance Map with Surge Modeling at Emergency Shut-down

(Cv=42cft/s/√psid)



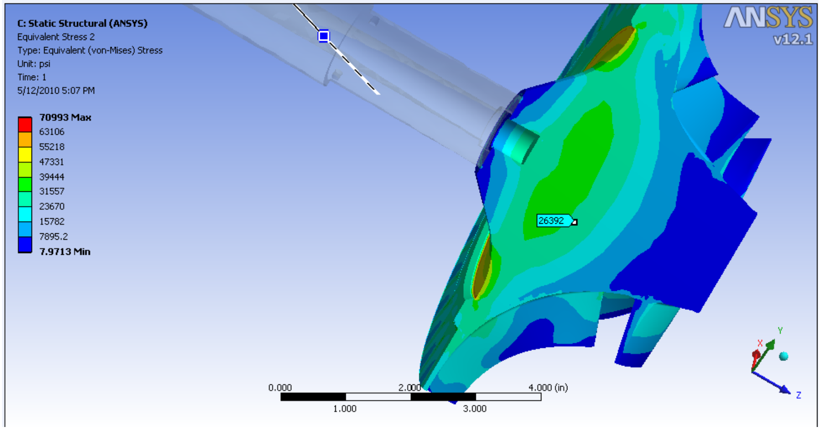
**Pressure ratio & flow rate path of compressor as it almost exceeds surge control with valve Cv=42**

# FEA by Concepts NREC Confirms Acceptable Rotor Stress Levels at 2100 ft/sec and Rotor Stability at 60,000 rpm



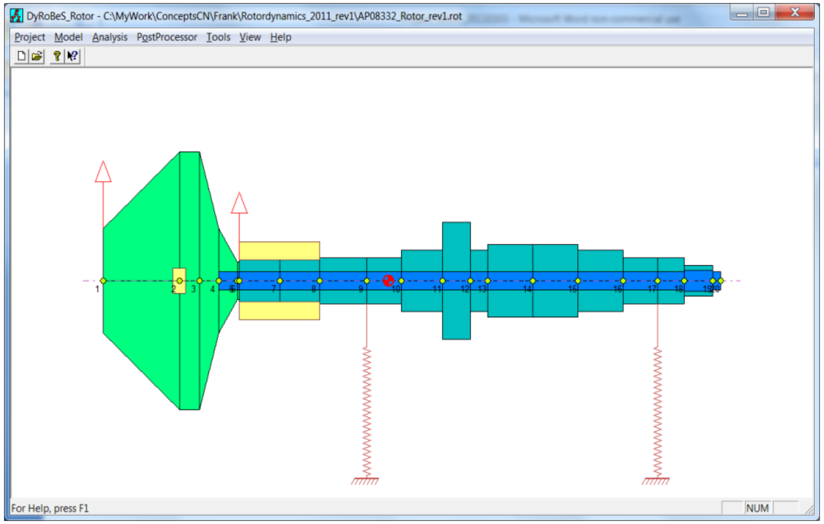
C: Static Structural (ANSYS)
Equivalent Stress 2
Type: Equivalent (von-Mises) Stress
Unit: psi
Time: 1
5/12/2010 5:06 PM

70993 Max
63106
55218
47331
39444
31557
23670
15782
7895.2
7.9713 Min

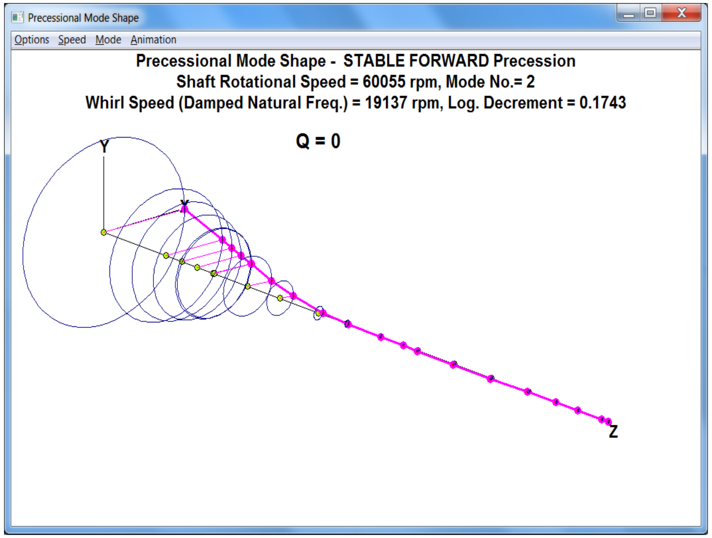


C: Static Structural (ANSYS)
Equivalent Stress 2
Type: Equivalent (von-Mises) Stress
Unit: psi
Time: 1
5/12/2010 5:07 PM

70993 Max
63106
55218
47331
39444
31557
23670
15782
7895.2
7.9713 Min



DyRoBeS_Rotor - C:\MyWork\ConceptsCN\Frank\Rotordynamics_2011_rev1\AP08332_Rotor_rev1.rot



Precessional Mode Shape

Precessional Mode Shape - STABLE FORWARD Precession
Shaft Rotational Speed = 60055 rpm, Mode No.= 2
Whirl Speed (Damped Natural Freq.) = 19137 rpm, Log. Decrement = 0.1743

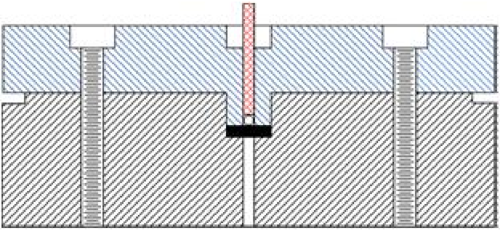
Q = 0

There is a patent pending regarding the subject matter of this presentation.

# In Response to Reviewer Ques. #1: Small Punch Test Apparatus by TAMU to Determine Effects of Hydrogen Exposure

The TAMU testing may have limited use with regards determining absolute yield stress but it has provided convincing evidence that the 7075-T6 aluminum is not affected by its long term exposure to a high pres. and high temp. hydrogen environment and that a more common material such as titanium does have compatibility issues. The inexpensive testing protocol also initiates another documented means of comparing materials, if only on a relative basis.

**Specially machined fixture for small hole punch testing of metal specimens for project tests at speeds of 0.0021 mm/s**



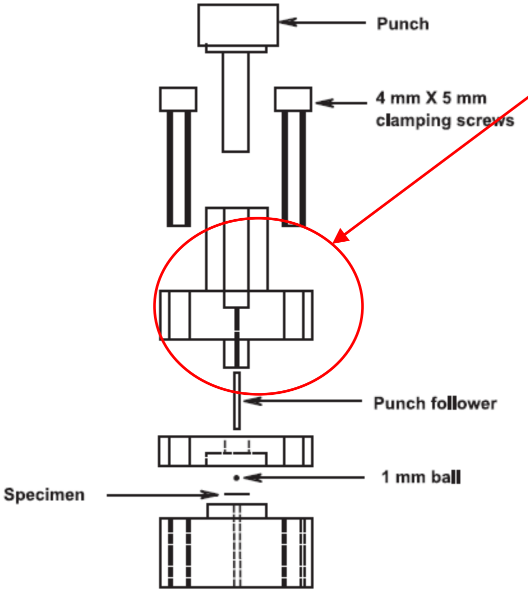


Fig. 1. Schematic diagram showing the test jig to test disc specimens 3 mm in diameter and 0.25 mm in thickness.

atter of this

**The following figures have been reproduced from the three technical papers that have used the technique to test materials.**

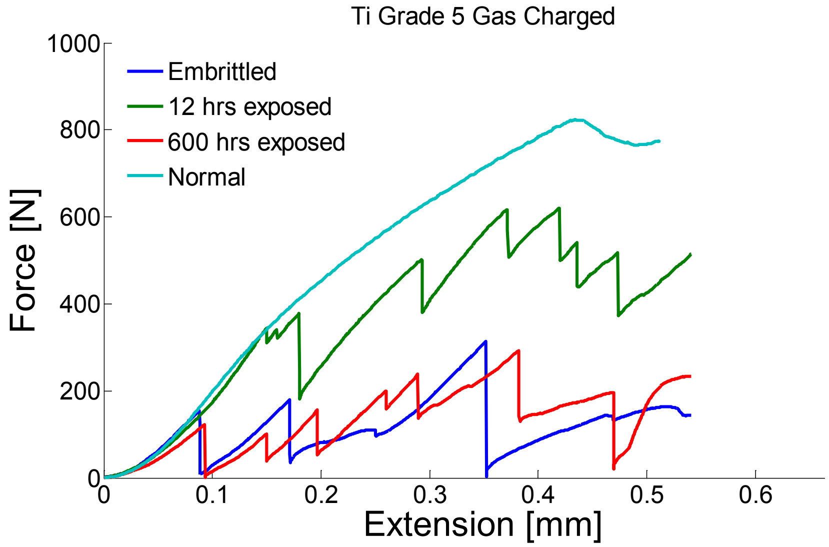
**Sources:**

**1. Klevtsov, I., "Using Small Punch Test for Determination of Tensile Properties of Steel," 6th International DAAAM Baltic Conference, April 2008.**

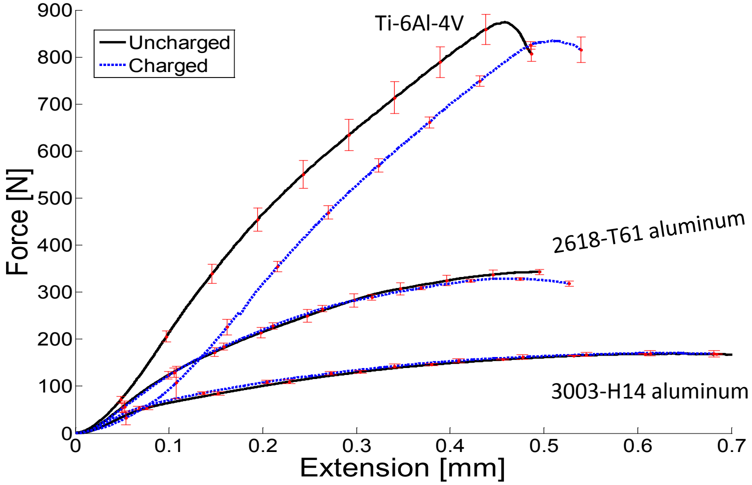
**2. Song, S. H. *et al.*," Small Punch Test Evaluation of Neutron-Irradiation-Induced Embrittlement of a Cr-Mo Low-Alloy Steel," ELSEVIER, 53: 35-41, 2004.**

**3. Lee, J., *et al.*, "Application of Small Punch Test to Evaluate Sigma-Phase Embrittlement of Pressure Vessel Cladding Material," Journal of Nuclear Science and Technology, 40( 9): 664-671, 2003.**

# Summary Details of Small Punch Test by TAMU



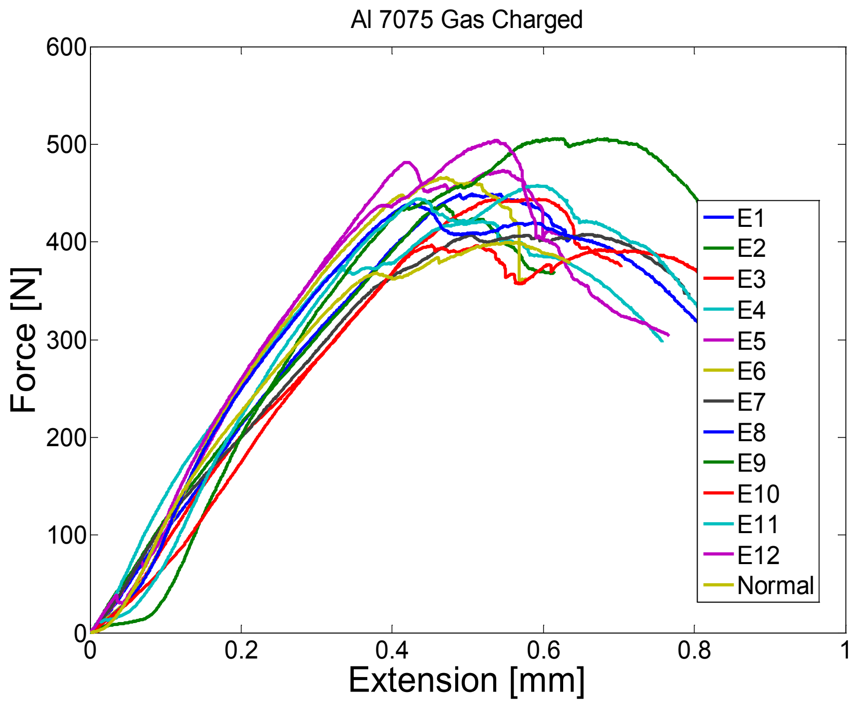
**Actual Test Result with Ti Grade 5 showing degradation of strength in hydrogen over time**

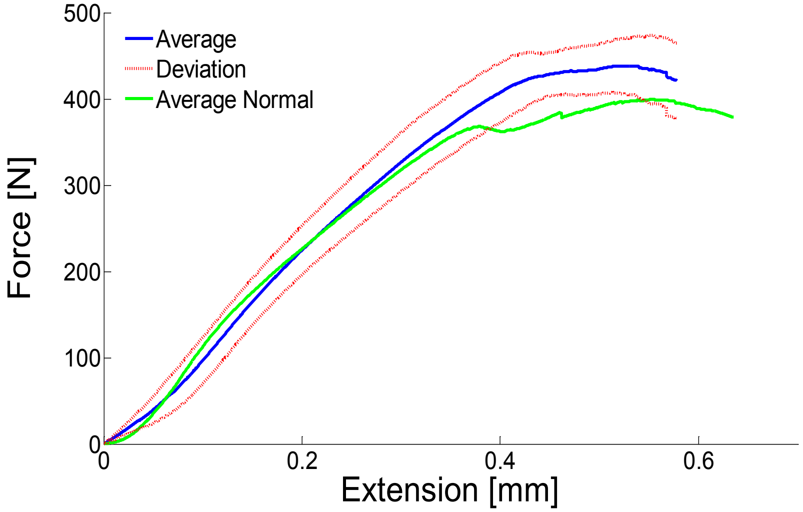


Home-made H charge system, soaking samples in a $H_2$ containing reservoir.

**Force vs. Extension curve showing how the mechanical strength of the Ti-6Al-4V specimens changes over time at room temp. after charging BUT Aluminum specimens are not affected**

# Results of Testing Charged AL 7075 Specimens vs. Normal





**CONCLUSION FROM TESTING:**

1. **Small Punch Test Methodology can discern relative strength of a materials resistance to hydrogen embrittlement**
2. **Results without coating now can serve as a baseline for testing (in progress) specimens with coatings**

# Compressor Capital and O&M Costs as Determined from DOE's HDSAM v.2 Economics Model

**Hydrogen Piston Cost ($) and Operation & Maintenance ($/kWhr) Using DOE's HDSAM v.2 Economics**

| | |
|---|---|
| No. of Piston Stages | 4 |
| kWe rating | 6,226 |
| Kg/day Hydrogen Flowrate | 240,000 |

| | |
|---|---|
| 3% | % Maintenance |
| 2 | Multiple of Capital Equip. Cost |

| | | |
|---|---|---|
| $ compressor= | $ | 6,278,724 |
| $, installation= | $ | 12,557,447 |
| $, maintenance/yr= | $ | 376,723 |
| kW-hr= | | 53,978,993 |
| O&M Cost [$/KwHr]= | | 0.0070 |

**Hydrogen Piston Cost ($) and Operation& Maintenance ($/kWhr)**

| | |
|---|---|
| No. of Piston Stages | 2 |
| kWe rating | 6,226 |
| Kg/day Hydrogen Flowrate | 240,000 |

| | |
|---|---|
| 3% | % Maintenance |
| 2 | Multiple of Captial Equip. Cost |

| | | |
|---|---|---|
| $ compressor= | $ | 4,709,043 |
| $, installation= | $ | 9,418,085 |
| $, maintenance/yr= | $ | 282,543 |
| kW-hr= | | 53,978,993 |
| O&M Cost [$/KwHr]= | | 0.0052 |

The two case Studies shown here have been determined from DOE's HDSAM v.2 Economics model and indicate the cost for a 2 and 4 cylinder Reciprocating compressor. The next slide also provides an independent verification for the cost of commercially available compressors and indicates that the centrifugal hydrogen pipeline compressor that is being developed for DOE is very competitive with the limited systems that are commercially available.

# Concepts NREC commissioned its collaborator: Hygen Industries to assess Market Potential and Competitive System

- **TASK 4: INITIAL DESIGN AND COST ANALYSIS**

**The manufacturing, operating and maintenance costs for various design approaches and operating conditions will be completed to understand the cost impact and levels of risk in meeting the stated performance goals. Generalized equipment costs and scaling factors will be used to assess the costs for various size systems. A cost-benefit-risk analysis assessment of alternative designs will be prepared.**

**HyGen S.O.W.: Analyze and investigate the installation, deployment, project development costs, etc., current systems costs – comparison.**

## HyGen Cost Analysis–Comparison Summary (Nov., 2009 Report):

*"…we have found that there is a clear niche Concepts can exploit. For the technical parameters provided by Concepts, (listed above), there is no significant competition. So far, after contacting more than 30 manufacturers, we have only had 4 responses that would even consider bidding on a competing system. Only one had a single system that could meet the production capacity of 240,000 kg/day, they bid $7.4 million. All others had to combine 2 systems to meet the production demand requested. Those two responders bid $2.5 million for a single system x 2 for a price of $5 million to meet the technical parameters and requirements outlined by CN. The most competitive system was a double system for $2.8 million total. The rest would not attempt to submit because it was outside their technical capabilities. This includes the one other Centrifugal Hydrogen Compressor Manufacturer who stated, "It would take 65 stages to meet 1200 psi". There appears to be several centrifugal air compressor manufacturers, but only 2 (that we found and submitted to) hydrogen compressor manufacturers that makes a centrifugal hydrogen compressor that performs even close to what Concepts is proposing to produce. The other failed to respond at all."*

# “Reliability and Maintenance Cost Algorithms for Piston and Centrifugal Compressor Systems”
## Prepared by: Frank Di Bella, PE (June 15, 2009) (Internal Tech. Report)

***An Internal Technical Report was prepared that provided the derivation of an algorithm that is inteneded to determine the relative Reliability and O&M Cost between a reciprocating compressor and a centrifugal compressor. Two excerpts from this document entitled:***
***“Reliability and Maintenance Cost Algorithms for Piston and Centrifugal Compressor Systems”***
***Prepared by: Frank Di Bella, PE (June 15, 2009) (Internal Tech. Report)*** *is offered as a response to the Reviewers question. The Executive Summary is offered on this slide and a second excerpt (next slide) as well as .ppt slides depicting data from the report are offered on the next 4 slides.*

### Executive Summary

The determination of the reliability (R) of the centrifugal compressor-based hydrogen pipeline compressor system and the cost to maintain the system ($/kwhr) has been estimated for a hydrogen centrifugal compressor project. The algorithms that are presented in this report can be used to provide a determination of an absolute cost using known reliability data for the individual elements that constitute the complete compressor system. If accurate reliability data for each component is not available or not current, the algorithms can still, as a minimum, provide an accurate comparison of the relative reliabilities and maintenance costs between a piston and a centrifugal compressor. That is, if the reliability of a commercially available compressor is accepted via experience to be “x”, then the methodologies presented in this report will enable the ratio of the reliabilities for the piston and the centrifugal compressors (i.e., ratio = x/y) to be determined. While it may not be appropriate to then determine “y” from the equation, it may be sufficient to know only that the ratio x/y is less than or greater than 1; that the cost or reliability of the piston-type compressor is more or less than the centrifugal-type compressor. It is also useful to be able to determine the effect that changes in the number of components may have on the overall reliability of the compressor. For example, changing from six stages to five stages, or using two bearings per rotor and not one per rotor, will increase the reliability and thus may be of value to the design. The methodology used in the Reliability Model can provide a numerical value to this change that hopefully indicates a proportional improvement in reliability with the value of the cost incurred.

The following report provides a summary of the methodology used to enable a comparison of the reliability and maintenance costs for a piston and a centrifugal compressor. However, the examples of the use of the algorithms are provided only to provide an example of the calculations that have been prepared. Values for the reliabilities of the individual components must continue to be updated based on the most current manufacturer’s data.

# Excerpt from Reliability Analysis prepared by F. DiBella

"…Assuming commercially available compressor systems have a reliability of 86.2%, then the calculation of the reliability for a compressor that is composed of the major components as itemized in Figures 1 a&b and using the equations shown above must result in a net reliability of 86.2%. The scaling factor (θ) must first be determined to "fit" this result. The scaling factor that has been calculated for this illustrative Case Study example is 50. The scaling factor is calculated by summing all of the individual hazard rates given in Figures 1a for the piston-type compressor. The result is $281 \times 10^{-6}$ which results in a MTTF = $1 \times 10^{6}/281$ = 3553 hours. However, the MTTF for a system with a reliability of 0.862 is only 528 hrs. Thus a scale factor is calculated to be: 3 years x 8760hrs/year /528= 49.8 .
That this scale factor "fits" the assumed reliability value with the reliability that can be calculated from the individual hazard rates can be confirmed by applying the reliability equation (EQN. 3) using the net hazard rate (λ) of $281 \times 10^{-6}$ for a 3 year MTTF or:
R(t)= exp(-281 x 10-6/**49.8** x 8760 hrs/yr x 3 yrs); then: R= 0.862 ---which checks----

By using the same scaling factor with the given values of the hazard rates for the individual components that constitute a centrifugal-type compressor, a comparative reliability can be fairly determined.

Reliability Engineering Analysis equations such as those given above are very much based on probability distribution functions. For any stochastic-dependent analysis, the lack of accurate data, presumably best determined from actual tests, will skew the analysis and render the results of such an analysis almost meaningless. The values of the MTTF or λ are very much dependent on tests performed (typically) by the manufacturer of the components or system of components. To somewhat reduce the uncertainty associated with the use of the reliability data obtained from the B.S. Dhillon reference, it was decided to provide only a comparison of the reliabilities of a conventional piston-based process gas compressor with the advanced, hydrogen centrifugal compressor. The analysis is considered to provide a fair comparison of the two different types of compressors by using the same MTTF (or λ=1/MMTF) metrics for the compressor components whose engineering function is shared by each compressor. For example, bearings, shaft seals or packing, intercoolers, etc., used in the piston or the centrifugal compressor, are given the same values of MTTF as can be seen in Figures 1 a&b. It is also possible to add a "risk factor" that is associated with one or more of the individual components. The risk factor is a value greater than 1 that attempts to account for any additional risk associated with the individual component as might be affected by its use in the hydrogen centrifugal compressor application. For the Case Study given in the Attachments 1 and 2, the risk factors were assumed to be 1, i.e., no additional risk factor was assumed for the components using hydrogen.
The very preliminary results highlighted in Attachments 1 and 2 would seem to indicate that the reliability of the centrifugal compressor is at least comparable to a piston compressor. **It is interesting to note that reducing the number of stages from six to five, improves the reliability by 3%.**

It must be noted again that all such conclusions are very much dependent upon the values for the individual components and the 3% improvement stated above is likely to be well within the range of uncertainty for each of the individual components[1]. More definitive results of numerical comparisons of the reliability must wait for more accurate values for the individual hazard rates (λ). These values must be provided by the manufacturers with some adjustment for the way that ***CN*** uses the component in the final hydrogen compressor design and then the values must still be confirmed by our prototype and field testing of the complete system. However, the development of the methodology for comparing a piston and centrifugal compressor is substantially complete and remains the major product of the research conducted todate.

# Developed a System Reliability and Maintenance Cost Analysis Methodology

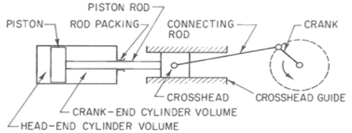


**Fig. 6-82** Double-acting design.

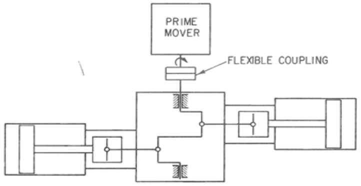


**Fig. 6-83** Balanced-opposed compressor

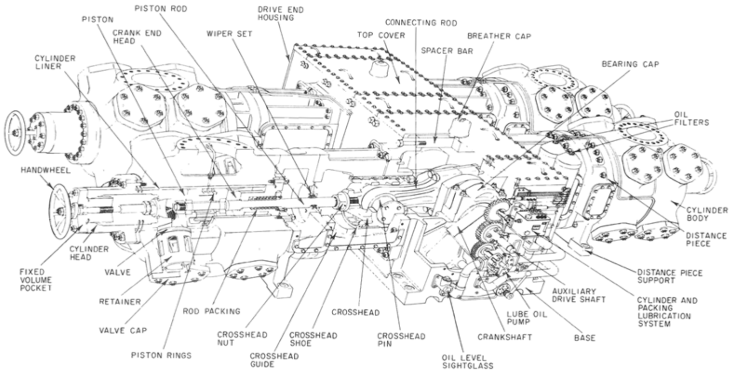
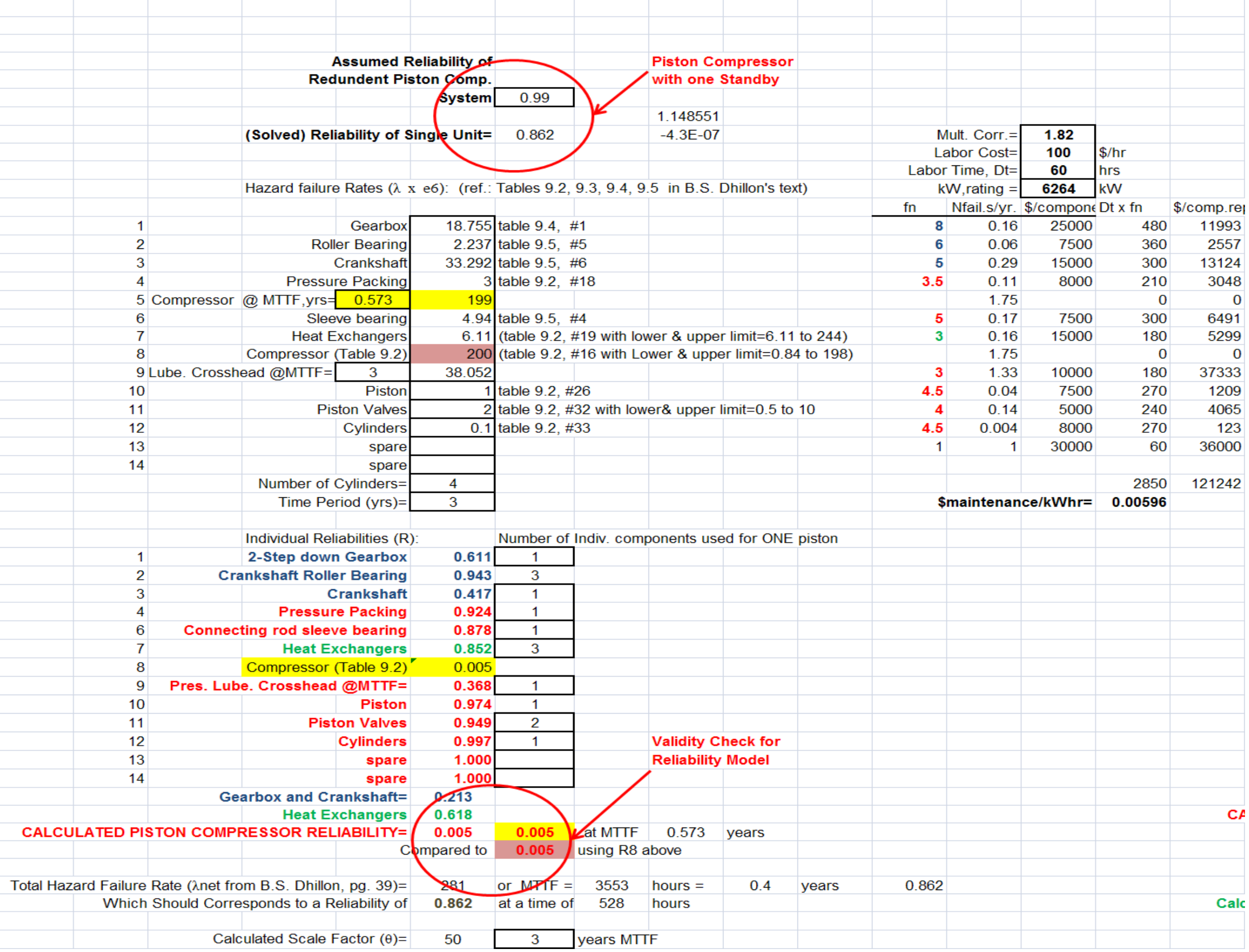


**Fig. 6-85** HVC engine—compressor.



**Assumed Reliability of Redundent Piston Comp. System** 0.99

**Piston Compressor with one Standby**

1.148551

**(Solved) Reliability of Single Unit=** 0.862

-4.3E-07

Mult. Corr.= **1.82**
Labor Cost= **100** $/hr
Labor Time, Dt= **60** hrs
kW,rating = **6264** kW

Hazard failure Rates (λ x e6): (ref.: Tables 9.2, 9.3, 9.4, 9.5 in B.S. Dhillon's text)

| | | | | | fn | Nfail.s/yr. | $/compone | Dt x fn | $/comp.re |
|---|---|---|---|---|---|---|---|---|---|
| 1 | Gearbox | | 18.755 | table 9.4, #1 | 8 | 0.16 | 25000 | 480 | 11993 |
| 2 | Roller Bearing | | 2.237 | table 9.5, #5 | 6 | 0.06 | 7500 | 360 | 2557 |
| 3 | Crankshaft | | 33.292 | table 9.5, #6 | 5 | 0.29 | 15000 | 300 | 13124 |
| 4 | Pressure Packing | | 3 | table 9.2, #18 | 3.5 | 0.11 | 8000 | 210 | 3048 |
| 5 | Compressor @ MTTF,yrs= | 0.573 | 199 | | | 1.75 | | 0 | 0 |
| 6 | Sleeve bearing | | 4.94 | table 9.5, #4 | 5 | 0.17 | 7500 | 300 | 6491 |
| 7 | Heat Exchangers | | 6.11 | (table 9.2, #19 with lower & upper limit=6.11 to 244) | 3 | 0.16 | 15000 | 180 | 5299 |
| 8 | Compressor (Table 9.2) | | 200 | (table 9.2, #16 with Lower & upper limit=0.84 to 198) | | 1.75 | | 0 | 0 |
| 9 | Lube. Crosshead @MTTF= | 3 | 38.052 | | 3 | 1.33 | 10000 | 180 | 37333 |
| 10 | Piston | | 1 | table 9.2, #26 | 4.5 | 0.04 | 7500 | 270 | 1209 |
| 11 | Piston Valves | | 2 | table 9.2, #32 with lower& upper limit=0.5 to 10 | 4 | 0.14 | 5000 | 240 | 4065 |
| 12 | Cylinders | | 0.1 | table 9.2, #33 | 4.5 | 0.004 | 8000 | 270 | 123 |
| 13 | spare | | | | 1 | 1 | 30000 | 60 | 36000 |
| 14 | spare | | | | | | | | |
| | Number of Cylinders= | | 4 | | | | | 2850 | 121242 |
| | Time Period (yrs)= | | 3 | | | | **$maintenance/kWhr=** | **0.00596** | |

| | Individual Reliabilities (R): | | Number of Indiv. components used for ONE piston |
|---|---|---|---|
| 1 | **2-Step down Gearbox** | **0.611** | 1 |
| 2 | **Crankshaft Roller Bearing** | **0.943** | 3 |
| 3 | **Crankshaft** | **0.417** | 1 |
| 4 | **Pressure Packing** | **0.924** | 1 |
| 6 | **Connecting rod sleeve bearing** | **0.878** | 1 |
| 7 | **Heat Exchangers** | **0.852** | 3 |
| 8 | Compressor (Table 9.2) | 0.005 | |
| 9 | **Pres. Lube. Crosshead @MTTF=** | **0.368** | 1 |
| 10 | **Piston** | **0.974** | 1 |
| 11 | **Piston Valves** | **0.949** | 2 |
| 12 | **Cylinders** | **0.997** | 1 |
| 13 | **spare** | **1.000** | |
| 14 | **spare** | **1.000** | |
| | **Gearbox and Crankshaft=** | **0.213** | |
| | **Heat Exchangers** | **0.618** | |

**Validity Check for Reliability Model**

**CALCULATED PISTON COMPRESSOR RELIABILITY=** **0.005** **0.005** at MTTF 0.573 years

Compared to **0.005** using R8 above

**CA**

Total Hazard Failure Rate (λnet from B.S. Dhillon, pg. 39)= 281 or MTTF = 3553 hours = 0.4 years 0.862

Which Should Corresponds to a Reliability of **0.862** at a time of 528 hours

**Calc**

Calculated Scale Factor (θ)= 50 3 years MTTF

## A consistent methodology has been prepared to eventually use MTBF test data and maintenance experience to compare piston and centrifugal reliability and maintenance performance for hydrogen compression

Analysis uses FERC data as reported in several studies by Dr. Anthony Smalley, *et al.* in a paper entitled: “Evaluation and Application of Data Sources for Assessing Operating Costs for Mechanical Drive Gas Turbines in Pipeline Service (Vol. 122, July 2000, Transactions of ASME) and “Benchmarking the Industry: Factors Affecting Compressor Station Maintenance Costs” by John Harrell, Jr. and A. Smalley of Southwest Research Institute (a presentation at the GMRC Gas Machinery Conference, October 2000).

**CONCEPTS NREC**

# Example of Relative Comparison of Centrifugal vs. Piston Compressor Reliability

Hazard failure Rates (λ x e6): (ref.: Tables 9.2, 9.3, 9.4, 9.5 in B.S. Dhillon's text)

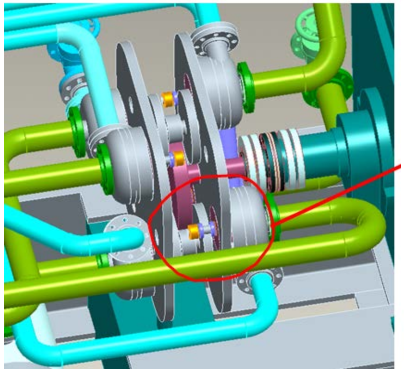
| | | |
|---|---|---|
| A | **Gearbox** | 18.755 |
| B | **Gears** | 5 |
| C | **spare** | |
| D | **Dynamic Seal** | 3.295 |
| E | **spare** | |
| F | **Sleeve bearing** | 4.94 |
| G | **Heat Exchangers** | 6.11 |
| H | Generic Compressor | 200 |
| I | **Highly Stressed Shaft** | 0.2 |
| J | **Pinion Gear** | 5 |
| K | **spare** | |
| L | **spare** | |
| M | **spare** | |

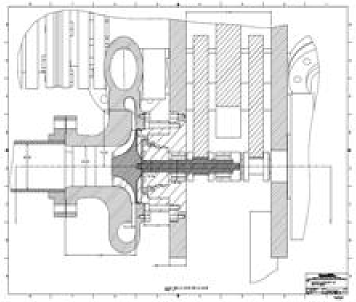
Number of Impellers= 6
Time Period (yrs)= 3

| | Individual Reliabilities (R): | | Increased Risk Multipl Factor |
|---|---|---|---|
| A | **Gearbox** | 0.990 | 1 |
| B | **Gears** | 0.997 | 1 |
| C | **spare** | 1.000 | 1 |
| D | **Dynamic Seal** | 0.998 | 1 |
| E | **spare** | 1.000 | 1 |
| F | **Sleeve bearing** | 0.997 | 1 |
| G | **Heat Exchangers** | **0.997** | 1 |
| H | Generic Compressor | 0.900 | 1 |
| I | **Highly Stressed Shaft** | 1.000 | 1 |
| J | **Pinion Gear** | 0.997 | 1 |
| K | **spare** | 1.000 | 1 |
| L | **spare** | 1.000 | 1 |
| M | **spare** | 1.000 | 1 |

| | | |
|---|---|---|
| **CALC.D SINGLE-STG CENTRIFUGAL COMPRESSOR RELIABILITY=** | **0.990** | **6** |
| **Calculated Gear Box Reliability=** | **0.985** | **1** |
| **Calculated Heat Exchanger Reliability=** | **0.984** | **5** |
| **BASIC COMPRESSOR V** | **0.943** | |

## This:





## Compared to this:

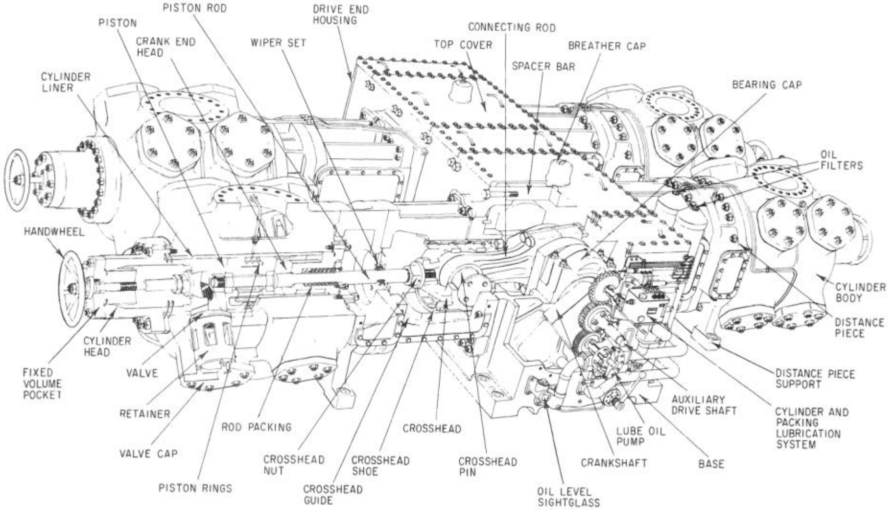
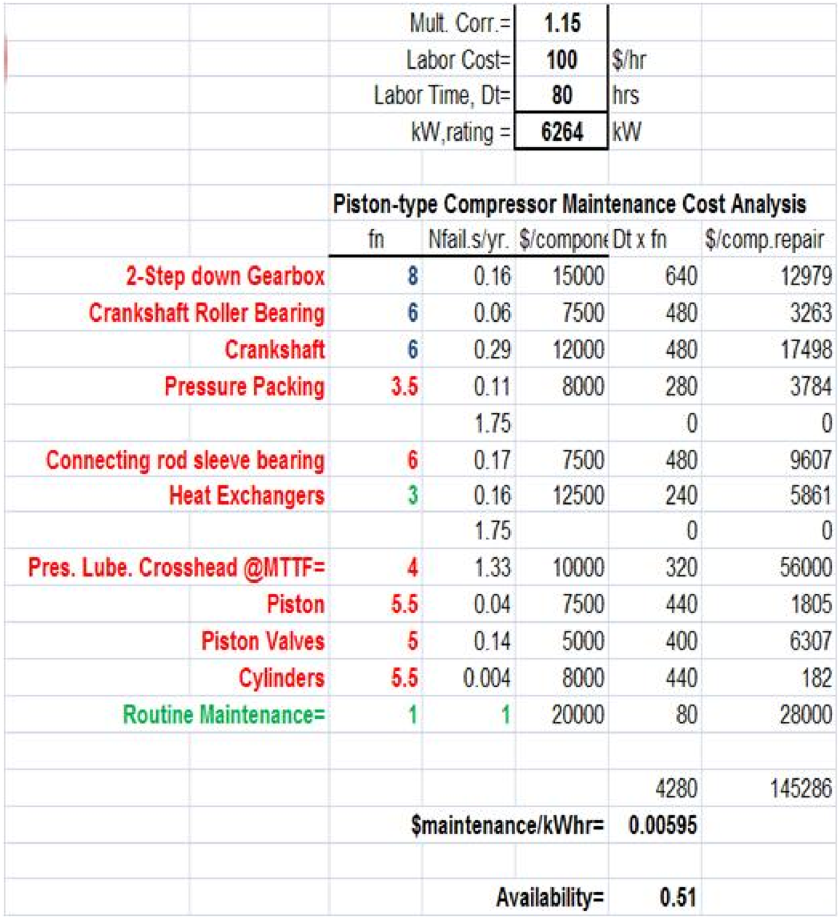


**Fig. 6-85** HVC engine—compressor.

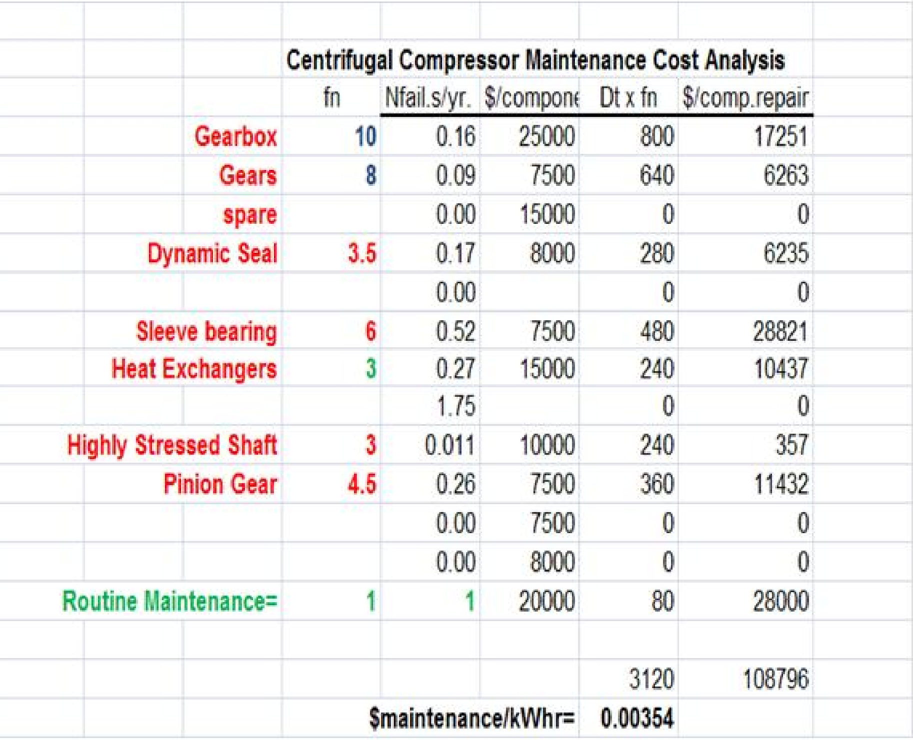
# Example of Methodology for Comparing the Relative Maintenance Cost of a Piston and Centrifugal Hydrogen Compressor



Mult. Corr.= 1.15
Labor Cost= 100 $/hr
Labor Time, Dt= 80 hrs
kW,rating = 6264 kW

**Piston-type Compressor Maintenance Cost Analysis**

| | fn | Nfail.s/yr. | $/compone | Dt x fn | $/comp.repair |
|---|---|---|---|---|---|
| 2-Step down Gearbox | 8 | 0.16 | 15000 | 640 | 12979 |
| Crankshaft Roller Bearing | 6 | 0.06 | 7500 | 480 | 3263 |
| Crankshaft | 6 | 0.29 | 12000 | 480 | 17498 |
| Pressure Packing | 3.5 | 0.11 | 8000 | 280 | 3784 |
| | | 1.75 | | 0 | 0 |
| Connecting rod sleeve bearing | 6 | 0.17 | 7500 | 480 | 9607 |
| Heat Exchangers | 3 | 0.16 | 12500 | 240 | 5861 |
| | | 1.75 | | 0 | 0 |
| Pres. Lube. Crosshead @MTTF= | 4 | 1.33 | 10000 | 320 | 56000 |
| Piston | 5.5 | 0.04 | 7500 | 440 | 1805 |
| Piston Valves | 5 | 0.14 | 5000 | 400 | 6307 |
| Cylinders | 5.5 | 0.004 | 8000 | 440 | 182 |
| Routine Maintenance= | 1 | 1 | 20000 | 80 | 28000 |
| | | | | | |
| | | | | 4280 | 145286 |
| | | | $maintenance/kWhr= | 0.00595 | |
| | | | | | |
| | | | Availability= | 0.51 | |

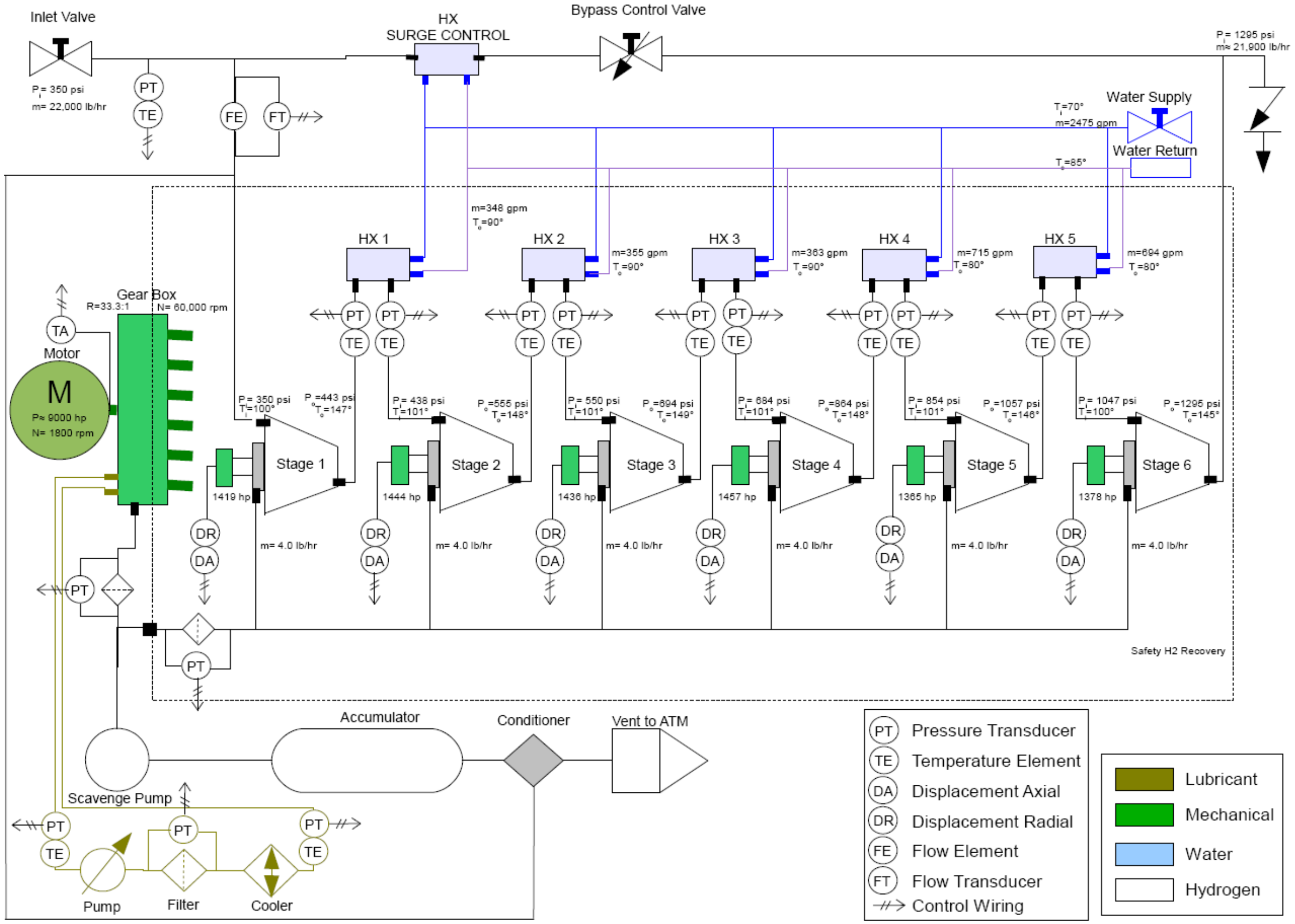


**Centrifugal Compressor Maintenance Cost Analysis**

| | fn | Nfail.s/yr. | $/compone | Dt x fn | $/comp.repair |
|---|---|---|---|---|---|
| Gearbox | 10 | 0.16 | 25000 | 800 | 17251 |
| Gears | 8 | 0.09 | 7500 | 640 | 6263 |
| spare | | 0.00 | 15000 | 0 | 0 |
| Dynamic Seal | 3.5 | 0.17 | 8000 | 280 | 6235 |
| | | 0.00 | | 0 | 0 |
| Sleeve bearing | 6 | 0.52 | 7500 | 480 | 28821 |
| Heat Exchangers | 3 | 0.27 | 15000 | 240 | 10437 |
| | | 1.75 | | 0 | 0 |
| Highly Stressed Shaft | 3 | 0.011 | 10000 | 240 | 357 |
| Pinion Gear | 4.5 | 0.26 | 7500 | 360 | 11432 |
| | | 0.00 | 7500 | 0 | 0 |
| | | 0.00 | 8000 | 0 | 0 |
| Routine Maintenance= | 1 | 1 | 20000 | 80 | 28000 |
| | | | | | |
| | | | | 3120 | 108796 |
| | | | $maintenance/kWhr= | 0.00354 | |

There is a patent pending regarding the subject matter of this presentation.

# General Piping and Instrumentation Flow Diagram for Hydrogen Compressor System



# Operating Conditions Applied for Stage Six

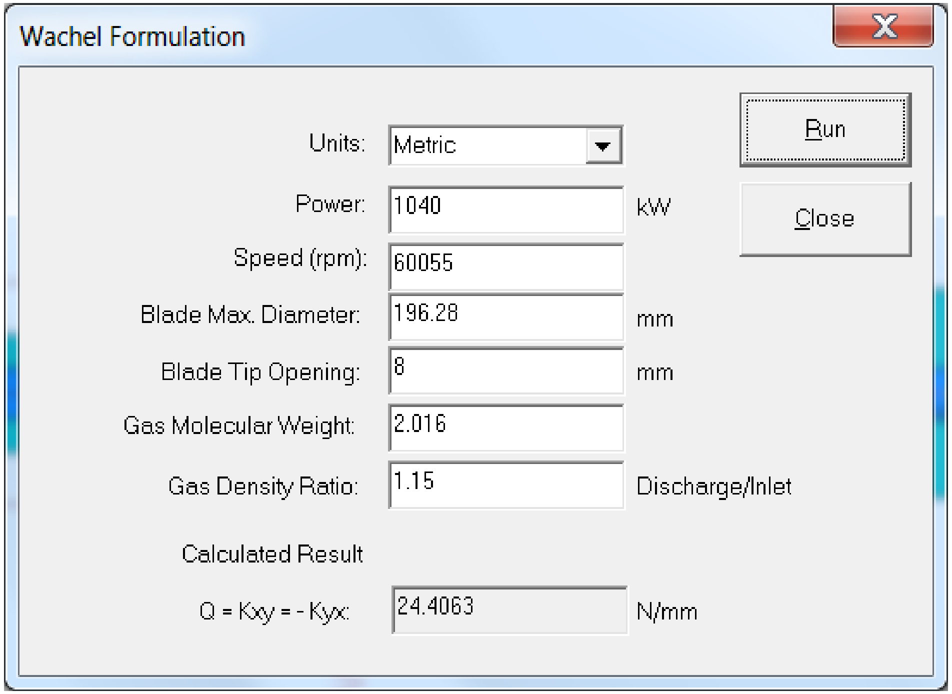
- **Material properties: Nitronic 50 (Volute Casing and Backplate)**
  - Elastic Modulus = 2.8 E7 PSI
  - Poisson’s Ratio = 0.30
  - Density = 0.285 lb/in$^3$
  - Yield Strength (Fty) = 57 KSI
  - Operating Pressure = 1280 PSI
  - HydroTest Pressure = 1920 PSI

- **Material properties: Aluminum 7075 (Volute)**
  - Elastic Modulus = 1.03 E7 PSI
  - Poisson’s Ratio = 0.33
  - Density = 0.1000lb/in$^3$
  - Yield Strength (Fty) = 66.5 KSI
- **Geometry:**
  - Volute Assembly = from Pro/ENGINEER®

# Publication & Presentations

- **Publications and Presentations**
  - Presentations have only been made to DOE’s Hydrogen Pipeline Delivery Technical Committee in August 2010 and December 2010 and to DOE Program Managers (Mr. Paul Bakke, Dr. Monterey Gardiner, Dr. Scott Weil, and Ms. S. Dillich) during several site visits by Concepts NREC. Abstract has been accepted for technical paper to the 2012 ASME International Congress & Exhibition (Houston, TX )
  - ASME Technical Paper for 2012 International Mechanical Engineering Congress & Exposition
- **Patents Pending (filed March, 2010) on system design and individual components**

# A Detailed Mass Model Was Created for Compressor Rotor-Drive Shaft Rotordynamics That Included Cross-Coupling Aero Effects



Units: Metric

Power: 1040 kW

Speed (rpm): 60055

Blade Max. Diameter: 196.28 mm

Blade Tip Opening: 8 mm

Gas Molecular Weight: 2.016

Gas Density Ratio: 1.15 Discharge/Inlet

Calculated Result

Q = Kxy = - Kyx: 24.4063 N/mm

There is a patent pending regarding the subject matter of this presentation.