2010 DOE Hydrogen Program Merit Review

Development of a Centrifugal Hydrogen Pipeline Gas Compressor

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Concepts NREC (*CN*)

June 9, 2010

Project ID#: PD017

Project Overview

Timeline

- Project Start: June, 1, 2008
- Project End: June 1, 2011
- Percent Complete: 50% (all Ph.I and Ph. II in Progress)

Budget

- Total Project Funding
  - DOE Share: $4,202,562
  - Contractor Share: $850,055
- Funding Received in FY08 (Phase I)
  - $1,055,000
- Funding for FY10 (Phase II)
  - $1,076,924

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 of Hydrogen Piston Compressors

Project Lead

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

Project Partners

- Praxair (Industrial User/Engineering Assistance)
- Texas A&M University (TAMU) (Materials Testing)
- HyGen Industries (Hydrogen Industry Consultant)

Technical Collaboration

- Sandia National Lab, Argonne National Lab, Savannah River National Lab
- ABB, Cotta Transmission, GE, KMC, Flowserve, Tranter, Heatric

Project Objectives - Relevance

- Demonstrate Advanced Centrifugal Compressor System for High-Pressure Hydrogen Pipeline Transport to Support DOE’s Strategic Hydrogen Economy Infrastructure Plan
  - Deliver $1,200^{+}$ psig and 100,000 to 1,000,000 kg/day of pure hydrogen to forecourt station at < $1/GGE
  - Reduce initial installed system equipment cost to less than $5.4 million uninstalled based on DOE’s HDSAM 2.0 Economics Model
  - Reduce Operating & Maintenance Costs via improved reliability
    - DOE’s HDSAM 2.0 Economics Model indicates 4% of installed cost per year or $0.01/kWhr
    - Improved reliability eliminates the need for system redundancies
  - Reduce System Footprint

Three-phase Program Approach

Phase I Initial Design (COMPLETED) (06/2008 to 12/2009)	Phase II Detailed Design (IN PROGRESS) (01/2010 to 09/2011)	Phase III System Validation Testing (10/2010 to 06/2011)
• Initial design criteria and performance specifications • Subsystems Modeling: aerodynamic and structural analysis of compressor • Initial integrated systems analysis • Initial design and cost analysis • Final design specifications • Materials and/or coatings investigated for use in high-pressure hydrogen environment • Revised Phase II Program Plan	• Detailed subsystems modeling • Detailed integrated systems analysis • Critical components design, testing, and development • Detailed integrated design of full-scale and laboratory validation systems • Detailed cost analysis of full-scale system	•Component Procurement •Two-stage centrifugal compressor system assembly •Performance evaluation test plan •Lab testing and system maturation •Final design of full-scale system completed •Field demonstration program plan prepared

Project Milestones

- May, 2008 - START Alternative system designs reviewed and selection made of preferred approach. Materials and components testing will be completed and a material selected for the compressor rotor.

- Dec., 2009 - Go/No-Go Decision – Detailed design and cost analysis of full-scale pipeline system completed. Design of Laboratory Validation System finalized.

- Oct., 2010 - Go/No-Go Decision – Fabrication and testing of two-stage Laboratory Validation System completed. Revised design and updated manufacturing cost analysis completed.

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

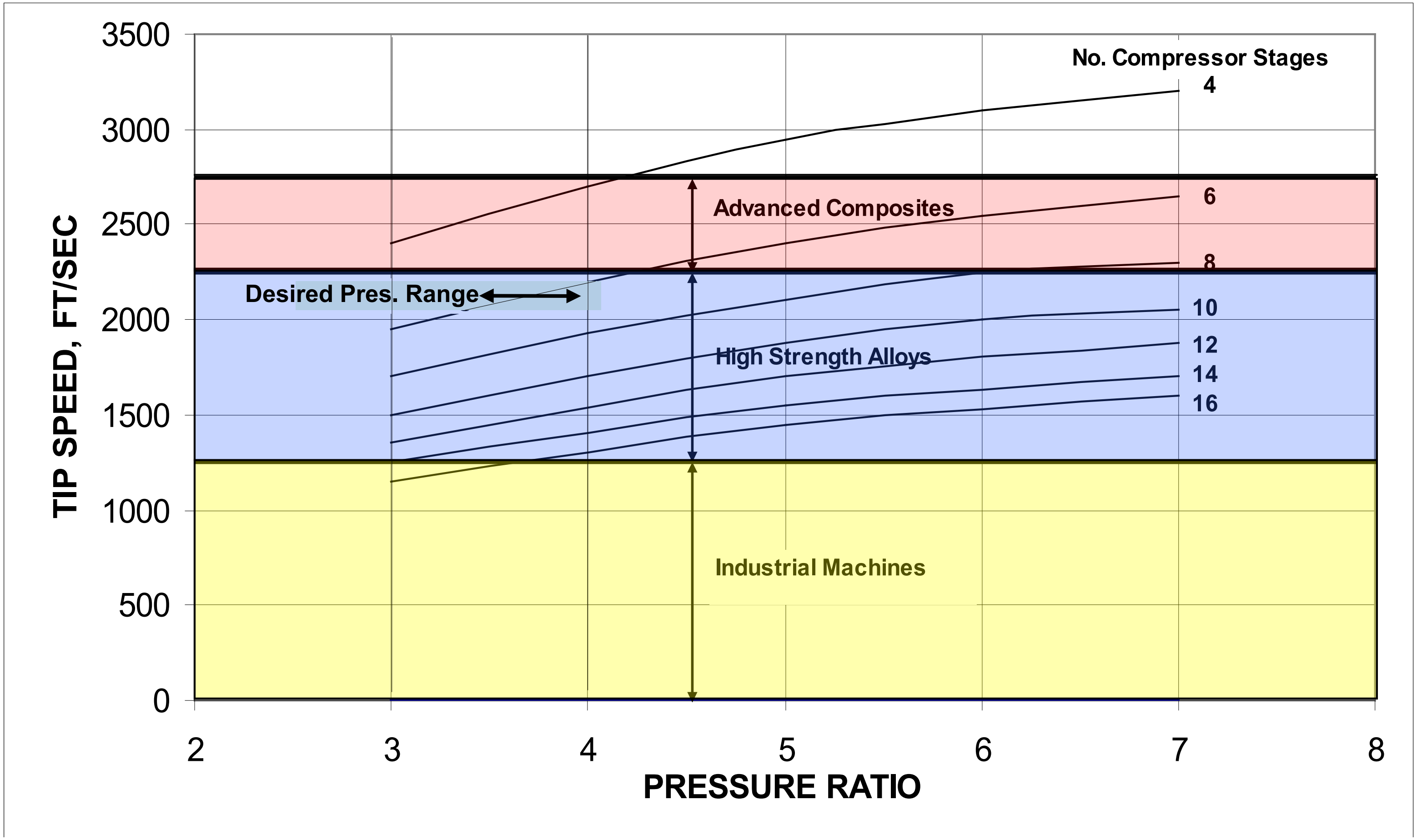
- Maximize centrifugal compressor tip speed within stress limitations of material.
  - Pressure ratio is proportional to $\text{rpm}^2 \times \text{radius}^2$, so small increase in tip speed results in significant increases in pressure.
  - Maximum thermodynamic efficiency is typically achieved at high 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 $\text{rpm}^2 \times \text{radius}^2 \times \text{material density}$. 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.

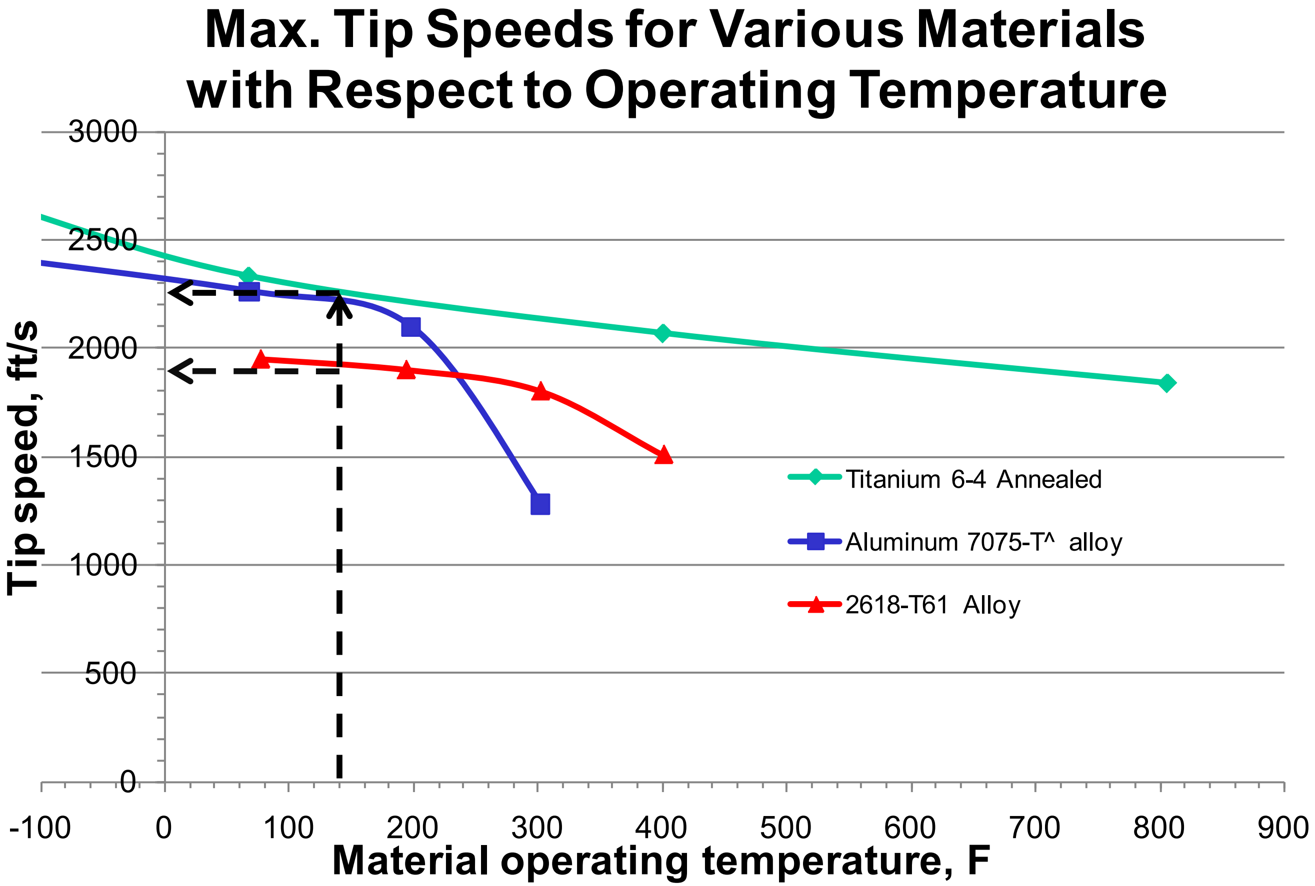
Project Engineering Approach - 3
Operational Design Envelope

Design Options for Alternative Operating Conditions



Design Experience Associating Material Properties with Tip Speed of 2,200 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)



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Project Collaborations:

Strengths & Responsibilities of Partners

- Praxair
  - Provides industrial user experience, gas industry specification data, and “hands-on” experience with compressor systems, including hydrogen compression, for industrial gas industry
  - Future industrial customer
- Texas A&M University
  - Provides 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

Phase I Summary: 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
Hydrogen Efficiency (f)	[btu/btu]	98%	98%
Hyd. Capacity (g)	kg/day	100,000 to 1,000,000	240,000
Hyd. Leakage (d)	%	< 0.5	0.2 (per Flowserve Shaft Seal Spec)
Hyd. Purity (h)	%	99.99	99.99 (per Flowserve Shaft Seal Spec)
Discharge Pressure (g)	psig	>1000	1285
Comp. Package Cost (g)	$M	6.2	4.5
Main. Cost (Table 3.2.2)	$/kWhr	0.007	0.005 (per *CN* Analysis Model)
Package Size (g)	sq. ft.	300 to 350 (per HyGen Study)	175 to 200 (per *CN* Design)
Reliability (e)	# sys.s req'd	Eliminate redundant system	Modular sys.s with 240K kg/day with no redundency req'd

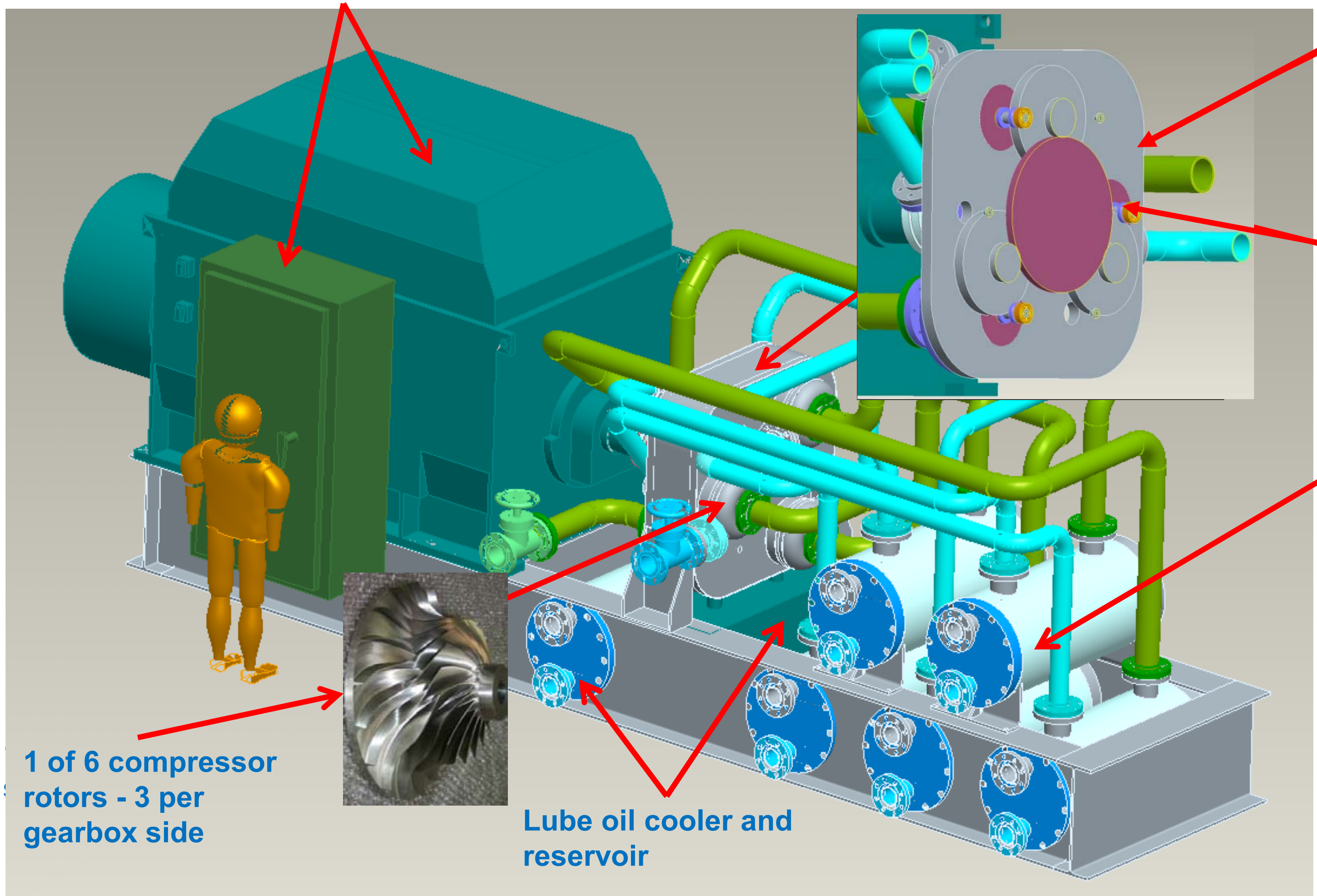
Project Objectives - Relevance to DOE Hydrogen Economy Planning

Completed in Phase I ✓

DOE Stated Objectives :

- ✓ Develop and demonstrate an advanced centrifugal compressor system for high-pressure hydrogen pipeline transport to support DOE's strategic hydrogen economy infrastructure plan
- ✓ Delivering 100,000 to 1,000,000 kg/day of 99.99% hydrogen gas from generation site(s) to forecourt stations
- ✓ Compressing from 350 psig to 1,000 psig or greater
- ✓ Reduce initial installed system equipment cost to less than $9M (Compressor Package of $5.4 M) for 200,000 kg/day system by FY 2017
- ✓ Reduce package footprint and improve packaging design
- ✓ Achieve transport delivery costs below $1/GGE
- ✓ Reduce maintenance cost to below 3% of Total Capital Investment by FY 2017
- ✓ Increase system reliability to thus avoid purchasing redundant systems
- ✓ Maintain hydrogen efficiency (as defined by DOE) to 98% or greater
- ✓ Reduce H2 leakage to less than 0.5% by FY 2017

Hydrogen Compressor Phase I Feasibility Design Results: 240,000 kg/day; 350 to 1,285 psig; 6,300 kWe



1,800 rpm Synchronous Motor and Controls

Compressor-Gearbox (10:1 and 3.33:1, parallel bull gear driving 6 pinions [3 per side])

Compressor drive connection

Intercoolers (cooling hydrogen at compressor inlet to 100°F)

Overall dimensions: 26 ft long x 6-8 ft wide x 10 ft high

1 of 6 compressor rotors - 3 per gearbox side

Lube oil cooler and reservoir

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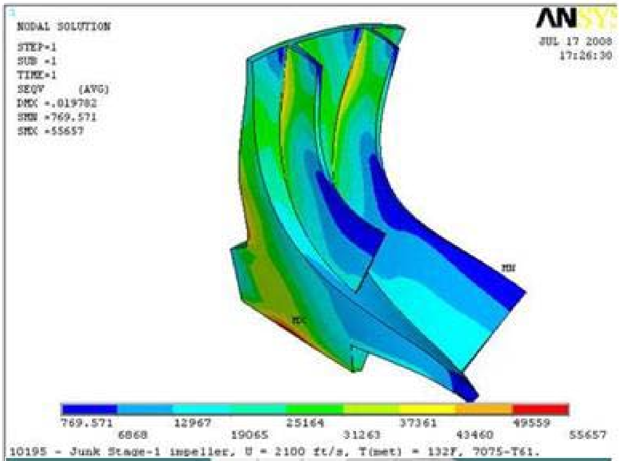
Hydrogen Pipeline Compressor Module Design Specifications for Conservative Choice

- Compressor design conditions confirmed by project collaborator Praxair as necessary for industrial applications
  - P_{inlet}= 350 psig, P_{outlet}= 1,285 psig; Flow rate = 240,000 kg/day
- 6-stage, 60,000 rpm, 3.56 pressure ratio compressor
  - A more advanced, experimental compressor rotor design is still under aero research and may provide the same pressure ratio, but with one less stage
- Integral gearbox pinions driving individual, overhung impellers
  - Cotta Gearbox with two-step gearing operating at state-of-the-art gear tip speeds
- Design of compressor's major mechanical elements completed and satisfied by two manufacturers per component:
  - KMC tilting-pad radial and thrust bearing designs confirmed for use
  - Flowserve gas face-seals have been confirmed to meet necessary specifications for hydrogen applications
- Heat exchanger specifications met by two manufacturers to cool hydrogen gas to 100°F between stages
  - Tranter Plate-type Heat Exchanger Design
  - Heatric Heat Exchanger (compact, plate-fin surface core)

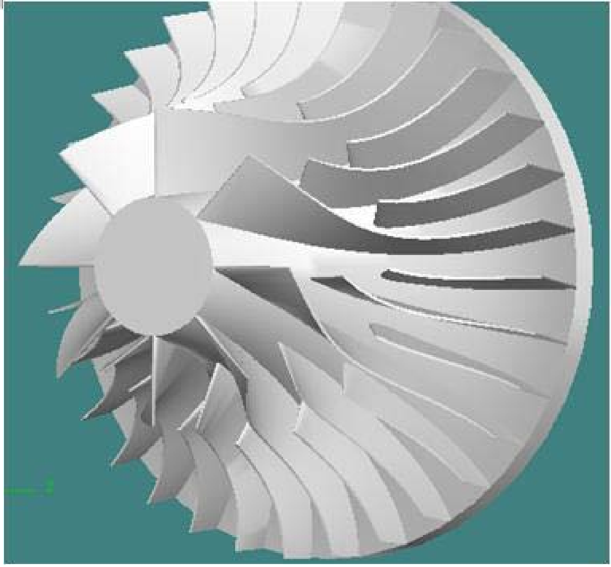
Technical Accomplishments and Progress

Compressor Design Specifications

- **Compressor Design Details:**
  - Centrifugal Compressor overall efficiency = 80.3%
  - A nominal shaft speed of 60,000 rpm
  - 6 stages (aluminum rotor)
  - Tip speed ~ 2,100 ft/s
    (corresponding to a hub stress of less than 60 kpsi)
  - Power of 1,400 hp per wheel,
  - Suction pressure 350 psig, discharge pressure 1,250$^+$ psig for an overall pressure ratio of 3.6
  - 240,000 kg/day hydrogen flow rate
    (ranging from 200,000 to 250,000 kg/day)
- **Geometry Advances**
  - Open passages with two splitter vanes
  - Forward sweep at vanes exit (not shown)
  - IGV causing negative swirl
- **Boreless Hub Design**
  - Decreases rotor hub stress
- **Multiple Patents Pending**



10195 - Junk Stage-1 impeller, U = 2100 ft/s, T(met) = 132F, 7075-T61.



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Phase II – Detailed Engineering Design

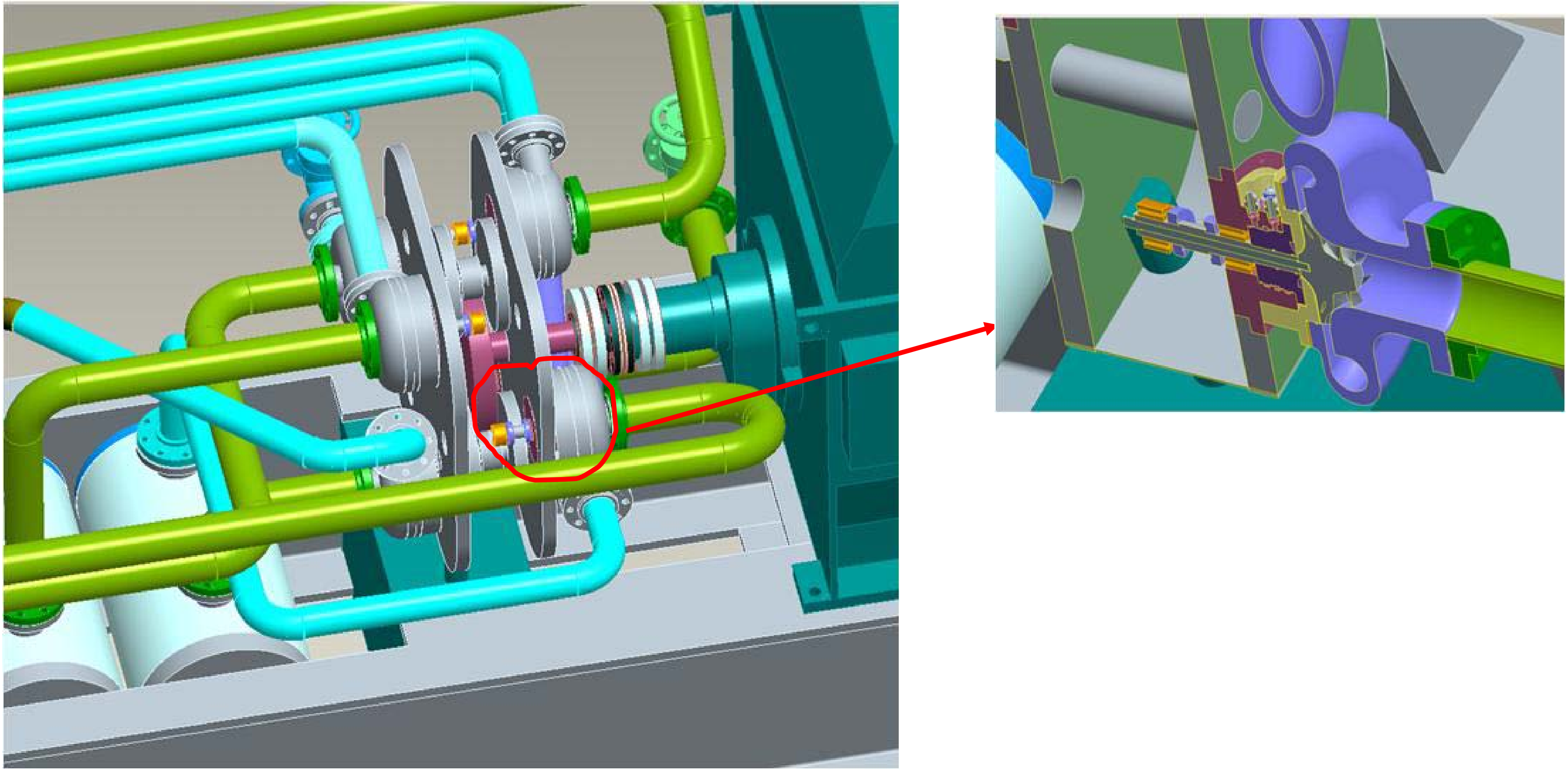
PHASE II OBJECTIVES:

- Critical components development and testing (rotor, shaft seal, bearings)
- Detailed design and cost analysis of a complete pipeline compressor system
- Go/No-Go decision regarding proceeding into Phase III: Fabrication of Complete Two-stage Hydrogen Compressor for Laboratory Testing

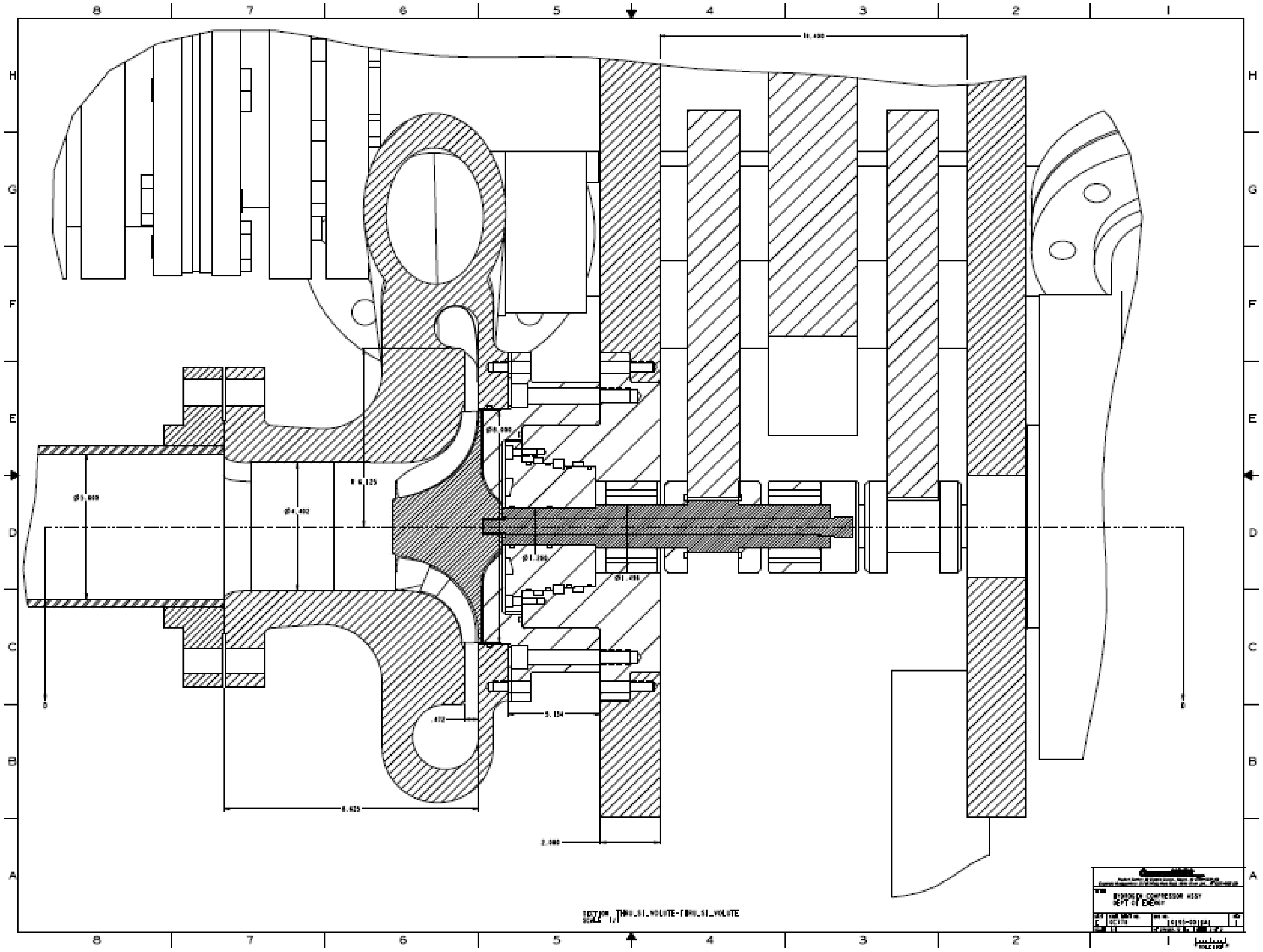
TASK DESCRIPTION:

Task 1 Detailed Subsystems Modeling
Task 2 Detailed Integrated Systems Analysis
Task 3 Components Design
Task 4 Critical Components Testing and Development
Task 5 Integrated System Design
Task 6 Detailed Cost Analysis
Task 7 Revised Phase III Program Plan (Go/No-Go Decision)
Task 8 Program Management and Reporting

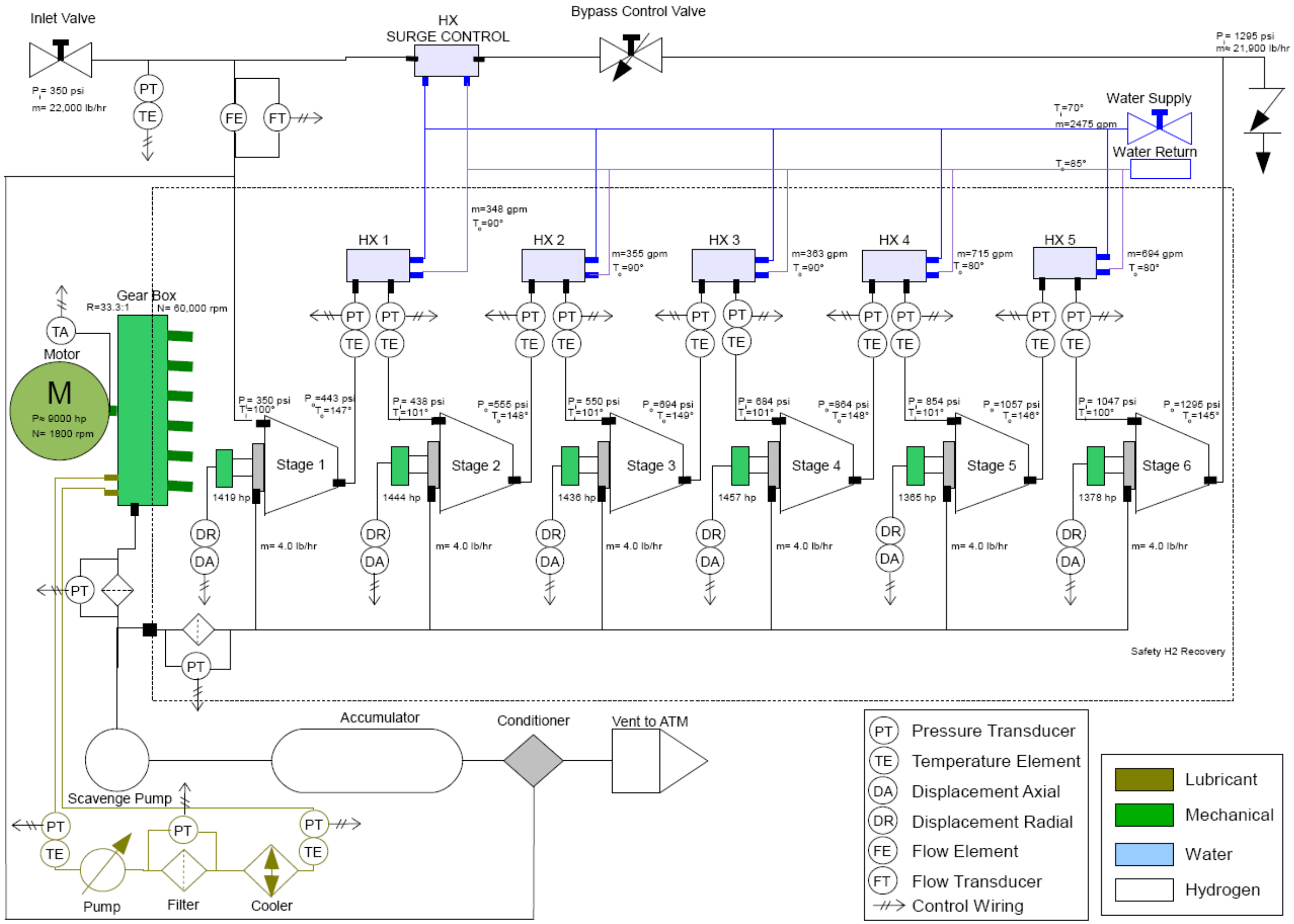
Mechanical Detail of Compressor Stage
All Stages Have the Same Mechanical Design



Mechanical Design Detail of Compressor Stage Coupled to Gearbox



General Piping and Instrumentation Flow Diagram for Hydrogen Compressor System



Phase II Accomplishments In Progress

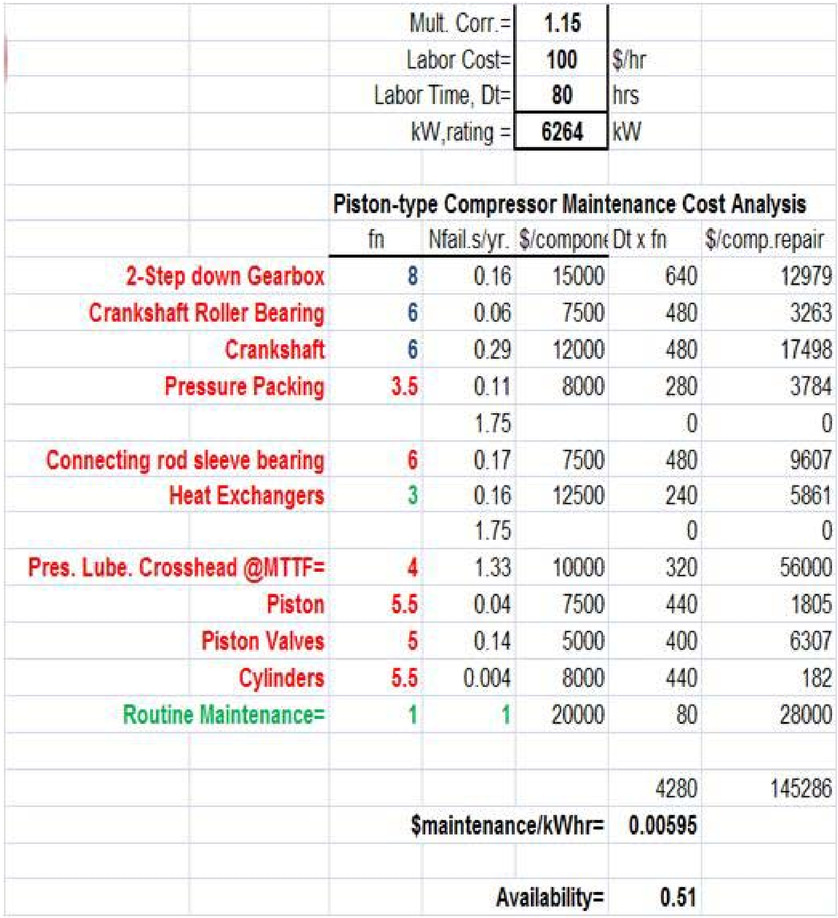
- FMEA Analysis
- Comparative Reliability Assessment
- Comparative O&M Assessment
- Algorithm for Anti-surge Valve Sizing for Emergency Shutdown

FMEA Document Has Been Prepared for Compressor Subsystems Shown

FMEA Working Component List

ID#	Sub-Assembly / Component
1	**Motor Subsystem**
1.1	Motor Shaft
1.2	Motor Bearings
1.3	Motor Windings
1.4	Motor Cooling
2	**Gearbox Subsystem**
2.1	**Low Speed (Input) Stage**
2.1.1	Input Coupling
2.1.2	Input Shaft
2.1.3	Input Shaft Bearings
2.1.4	Input Shaft Seal
2.1.5	Input Gear
2.2	**Intermediate Speed Stage**
2.2.1	Int. Gear (in)
2.2.2	Int. Shaft
2.2.3	Int. Bearings
2.2.4	Int. Gear (out)
2.3	**High Speed (Output) Stage (2X)**
2.3.1	High Speed Gears
2.3.2	High Speed Shaft
2.3.4	High Speed Bearings
2.3.5	Thrust Bearing
2.3.6	High Speed Shaft Seals
2.4	**Lubrication Subsystem**
2.4.1	Lubricant
2.4.2	Pump
2.4.3	Filter
2.4.4	Lubrication Jets
3	**Compressor Stages Subsystems**
3.1	**Stage #1**
3.1.1	Stage #1 Shaft
3.1.2	Stage #1 Impeller
3.1.3	Stage #1 Impeller Attachment
3.1.4	Stage #1 Shaft Seal
3.1.5	Stage #1 Housing
3.2	**Stage #2**
3.3	**Stage #3**
3.4	**Stage #4**
3.5	**Stage #5**
3.6	**Stage #6**
4	**Piping and Intercooling Subsystem**
4.1	**Piping**
4.1.1	Flanges / Seals
4.1.2	Pipe
4.2	**Intercoolers**
4.2.1	Flange / Seal, Working Fluid
4.2.2	Flange / Seal, Coolant
4.2.3	Internal Piping
4.2.4	Coolant
5	**Hydrogen Containment Subsystem**
5.1	**Containment Housing**
5.2	**HP Re-Introduction System**
5.3	**LP Ventilation System**
6	**System Skid**
7	**Controls and Instrumentation**

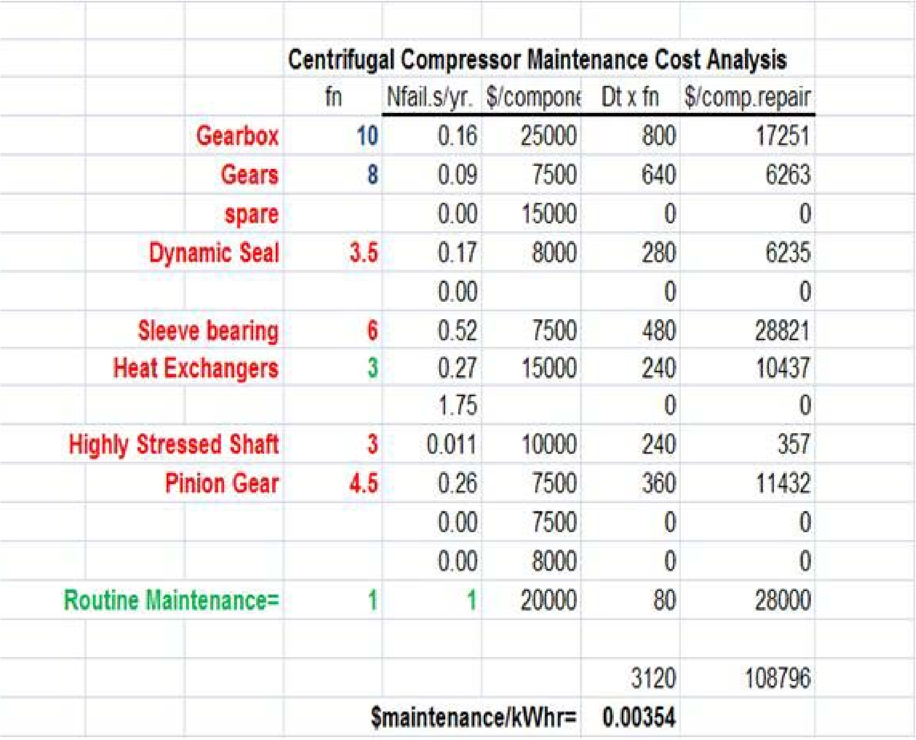
Example of Methodology for Comparing the Relative Maintenance Cost of a Piston and Centrifugal Hydrogen Compressor



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	

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**	

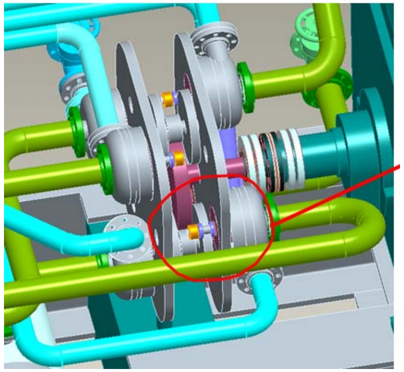
Number of Impellers= 6

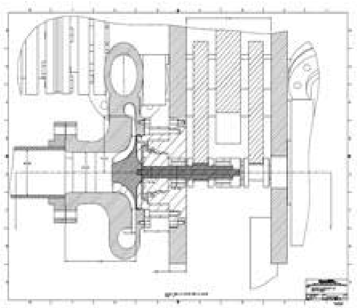
Time Period (yrs)= 3

	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:





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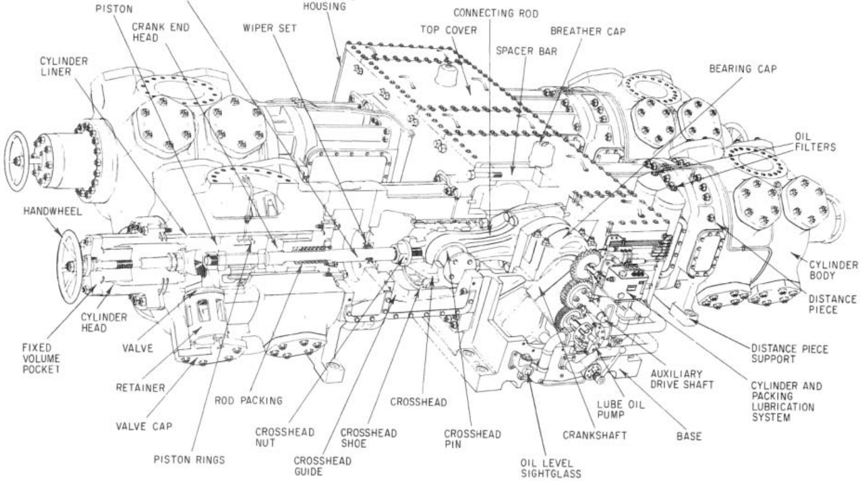
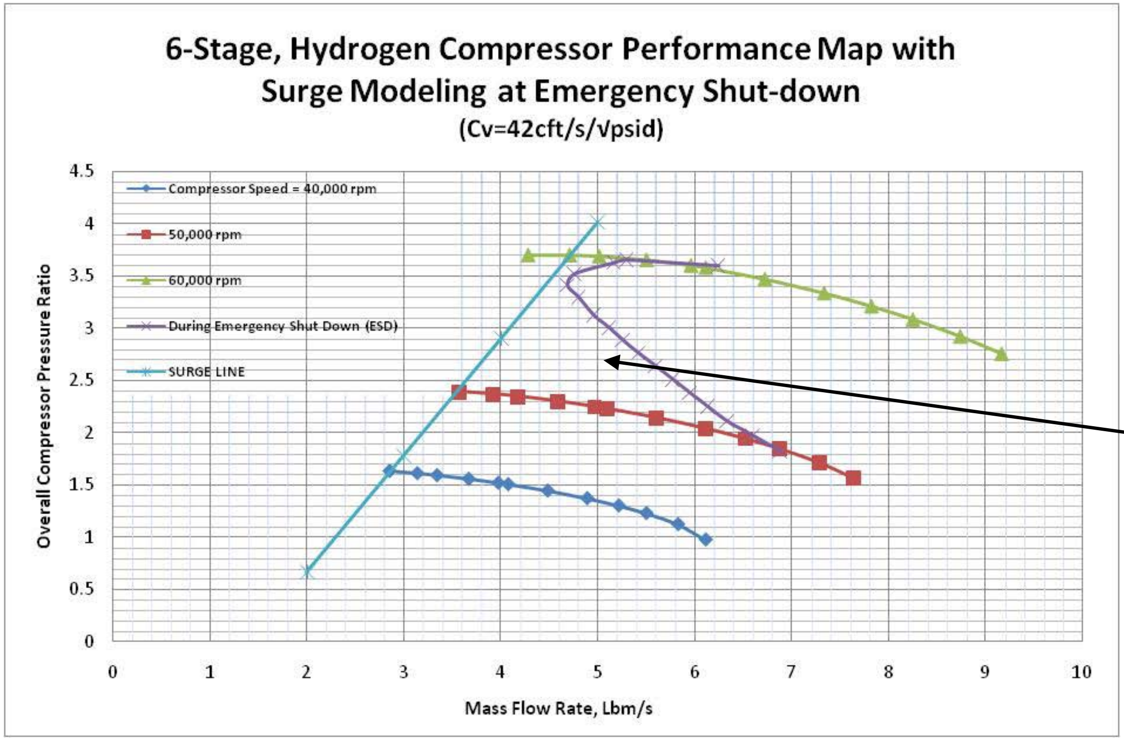
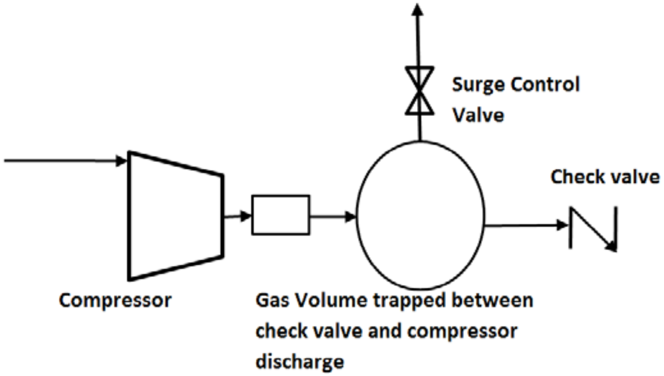


Fig. 6-85 HVC engine—compressor.

Anti-surge Control Model Algorithm for Emergency Shutdown

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





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

Materials Testing Technical Accomplishments in Progress by Texas A&M University

- Paraphrasing comments made during many interviews with researchers, the quick answer is: **"…no known coating materials exist to prevent hydrogen diffusion and hence the embrittlement of the base material"**
- Texas A&M will conduct coating experiments with coatings recommended by *CN* and others
  - Proposed coatings are all aluminum oxide-based (although DLC-Diamond-like Coating, has also been considered, but is discouraged by Texas A&M and others)
    - Accuratus (APS Company)
    - Alodine EC2 ElectroCeramic (Henkel Corp)
    - Sermalon (Sermatech International)
- Some structural concerns:
  - Can the coating be applied without affecting compressor material or vane design?
  - Will it compromise the base material by exposing even a small activation site on the base material if coating is chipped, cracked, or otherwise broken?
  - Will it contaminate the hydrogen during long-term use?

Small Punch Test Apparatus by TAMU to Determine Effects of Hydrogen Exposure

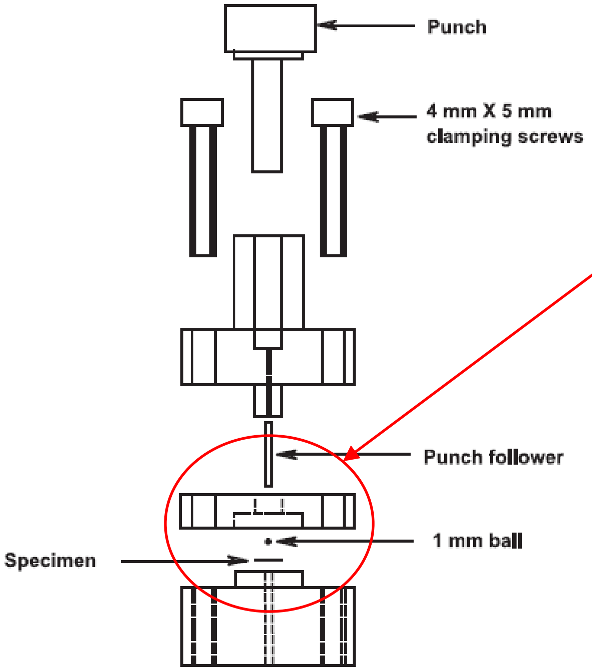
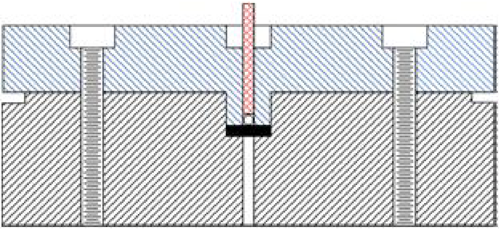


Fig. 1. Schematic diagram showing the test jig to test disc specimens 3 mm in diameter and 0.25 mm in thickness.



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

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

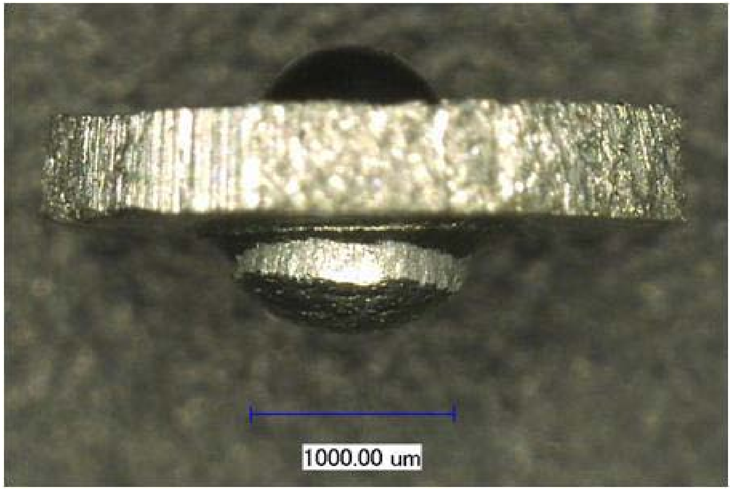
Sources:

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

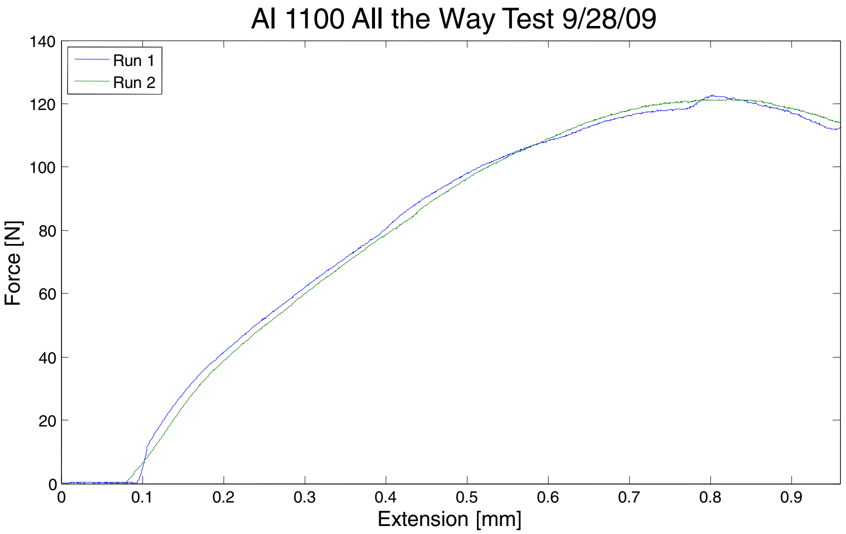
2. Song, S. H. *et al.*,” Small Punch Test Evaluation of Neutron-Irradiation-Induced Embrittlement of a Cr-Mo Low-Alloy Steel,” ELSEVIER, 53: 35-41, 2004.

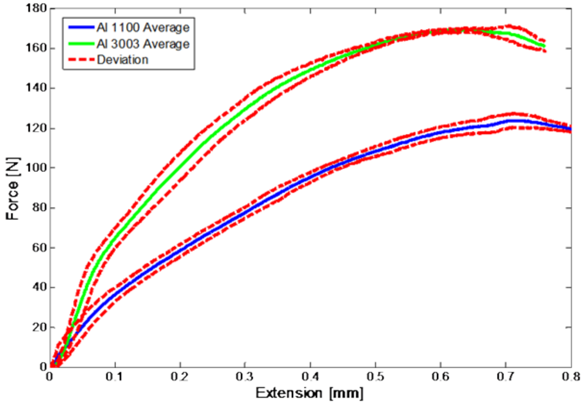
3. Lee, J., *et al.*, “Application of Small Punch Test to Evaluate Sigma-Phase Embrittlement of Pressure Vessel Cladding Material,” Journal of Nuclear Science and Technology, 40(9): 664-671, 2003.

Some Texas A&M Material Testing Results



Cross section of test sample (0.5-mm x 3-mm dia.) with puncture made by metal 1-mm diameter ball bearing





This figure compares the average force vs. extension curve for the aluminum 1100 and aluminum 3003 samples, showing the standard deviation for each

Future Project Work

- Phase II Detailed Design (12/2009 to 09/2010)
  - Detailed subsystems modeling
  - Detailed integrated systems analysis
  - Critical components design, testing, and development
  - Detailed design of two-stage hydrogen compressor prototype
  - Continue materials testing at Texas A&M University with hydrogen-charged specimens and coatings
- Phase III System Validation Testing (10/2010 to 06/2011)
  - Component procurement for two-stage functional hydrogen compressor system
  - Two-stage centrifugal compressor system assembly
  - Testing at Praxair or National Laboratory hydrogen facility

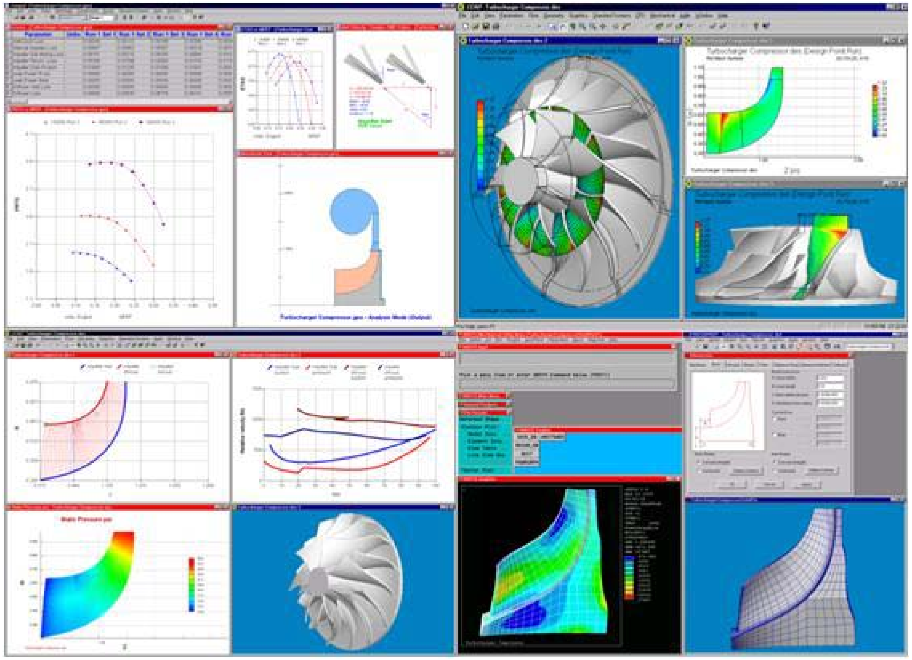
Project Summary

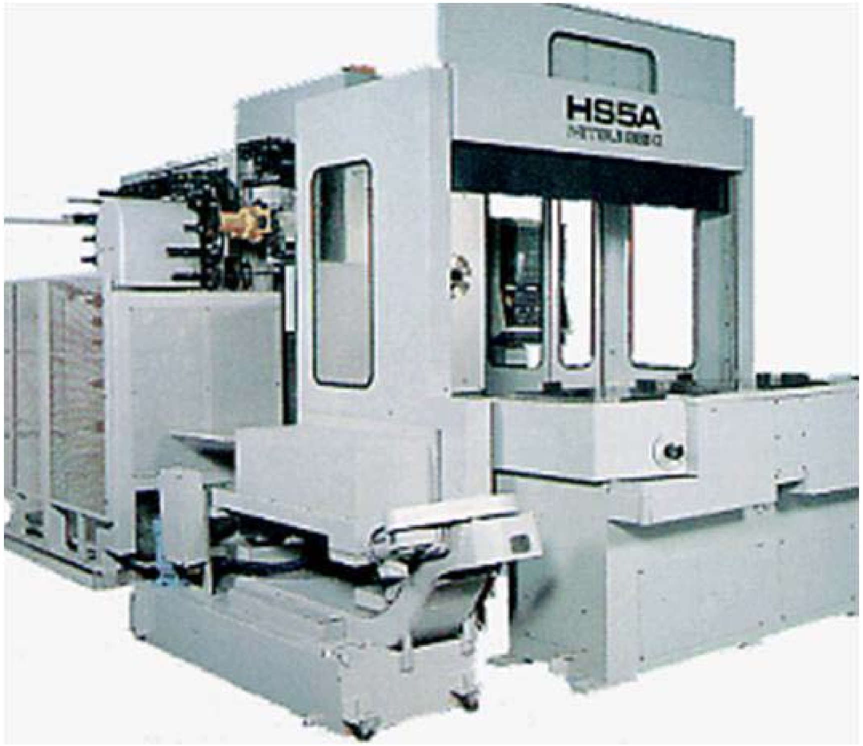
- **Relevance:** An advanced pipeline compressor system has been designed that meets DOE's performance goals for:
  - High reliability with 350 to 1,200$^+$ psig compression of 240,000 kg/day at 98% hydrogen efficiency
  - footprint one-third the size of existing industrial systems at projected cost of less than 75% of DOE's target.
- **Approach:** Utilize state-of-the-art and acceptable engineering practices to reduce developmental risk for:
  - aerodynamic/structural analyses for acceptable material stresses,
  - Industrially proven bearings, seal technology, gearing, heat exchangers, lube system
- **Tech. Accomplishments & Progress:** Aerodynamic analysis and design of a cost-effective, six-stage centrifugal compressor has been completed - the largest hydrogen centrifugal compressor available for pipeline-grade service. Detailed design is underway.
- **Technology Transfer/Collaboration:** The collaborative team consists of an industrial user, Praxair, with engineering experience in pipeline compressors; a materials researcher, Texas A&M; a hydrogen refueling industry consultant, HyGen; and the coordinated technical support of several National Labs.
- **Proposed Future Research:** Complete materials testing with TAMU of specimens and actual rotor forensics after high-speed testing and coatings; continue with detailed design of compressor in Phase II, culminating in the fabrication and laboratory testing of prototype compressor-gearbox in Phase III; update cost of system.

Additional Supportive Data

- The following slides are included here to provide additional support during the question and answer period for the salient summary that has been offered during the formal presentation describing the extensive work that has been performed during the last 10 months.

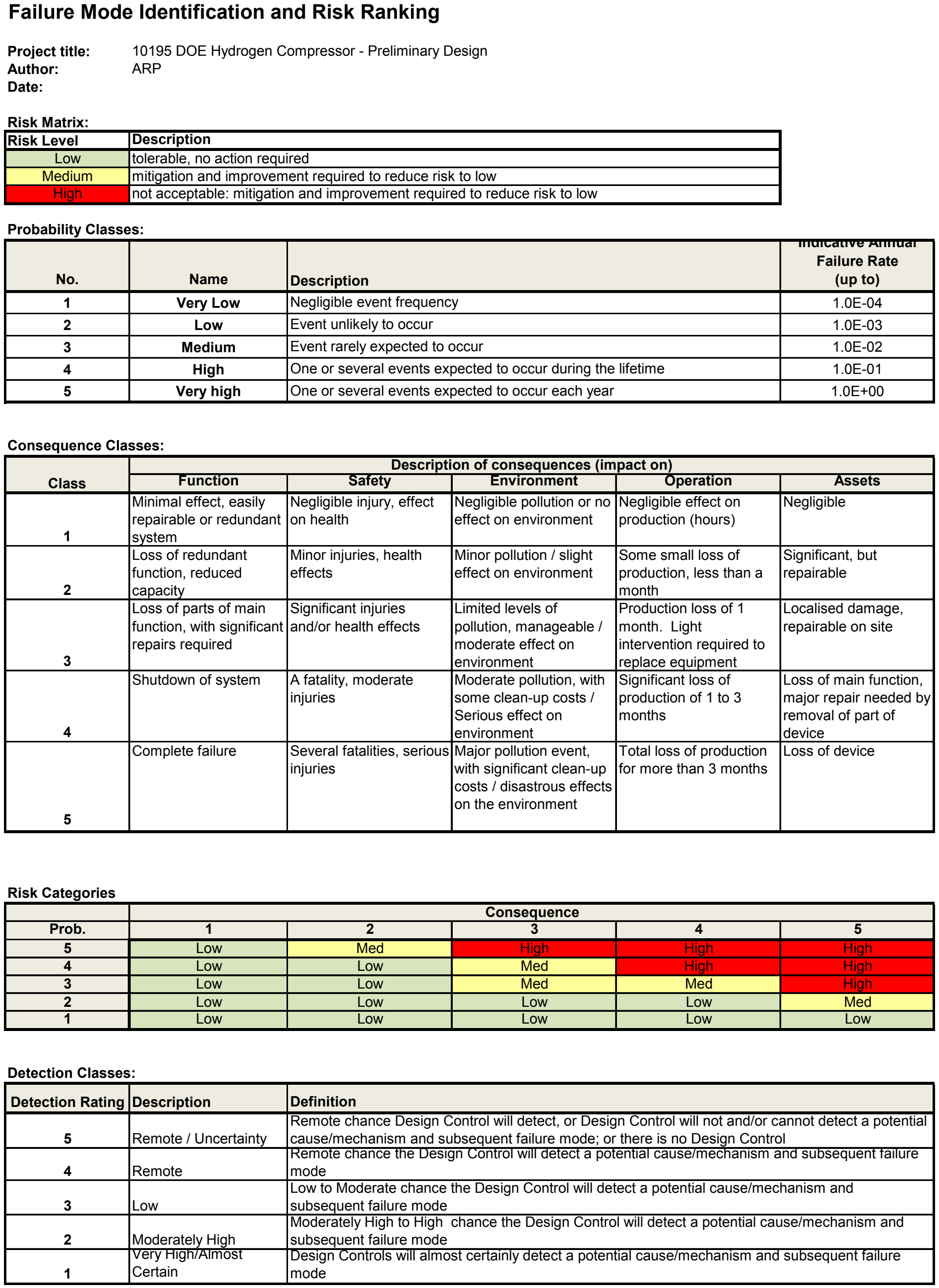
Project Collaborations-
Principal Investigator - Concepts NREC:
Capabilities from Aero Design to Manufacturing





CONCEPTS NREC

FMEA Document Risk Ranking Used with Compressor Subsystems Shown



Failure Mode Identification and Risk Ranking

Project title: 10195 DOE Hydrogen Compressor - Preliminary Design
Author: ARP
Date:

Risk Matrix:

Risk Level	Description
Low	tolerable, no action required
Medium	mitigation and improvement required to reduce risk to low
High	not acceptable: mitigation and improvement required to reduce risk to low

Probability Classes:

No.	Name	Description	Indicative Annual Failure Rate (up to)
1	**Very Low**	Negligible event frequency	1.0E-04
2	**Low**	Event unlikely to occur	1.0E-03
3	**Medium**	Event rarely expected to occur	1.0E-02
4	**High**	One or several events expected to occur during the lifetime	1.0E-01
5	**Very high**	One or several events expected to occur each year	1.0E+00

Consequence Classes:

Class	Description of consequences (impact on)				
	Function	**Safety**	**Environment**	**Operation**	**Assets**
1	Minimal effect, easily repairable or redundant system	Negligible injury, effect on health	Negligible pollution or no effect on environment	Negligible effect on production (hours)	Negligible
2	Loss of redundant function, reduced capacity	Minor injuries, health effects	Minor pollution / slight effect on environment	Some small loss of production, less than a month	Significant, but repairable
3	Loss of parts of main function, with significant repairs required	Significant injuries and/or health effects	Limited levels of pollution, manageable / moderate effect on environment	Production loss of 1 month. Light intervention required to replace equipment	Localised damage, repairable on site
4	Shutdown of system	A fatality, moderate injuries	Moderate pollution, with some clean-up costs / Serious effect on environment	Significant loss of production of 1 to 3 months	Loss of main function, major repair needed by removal of part of device
5	Complete failure	Several fatalities, serious injuries	Major pollution event, with significant clean-up costs / disastrous effects on the environment	Total loss of production for more than 3 months	Loss of device

Risk Categories

	Consequence				
Prob.	**1**	**2**	**3**	**4**	**5**
5	Low	Med	High	High	High
4	Low	Low	Med	High	High
3	Low	Low	Med	Med	High
2	Low	Low	Low	Low	Med
1	Low	Low	Low	Low	Low

Detection Classes:

Detection Rating	Description	Definition
5	Remote / Uncertainty	Remote chance Design Control will detect, or Design Control will not and/or cannot detect a potential cause/mechanism and subsequent failure mode; or there is no Design Control
4	Remote	Remote chance the Design Control will detect a potential cause/mechanism and subsequent failure mode
3	Low	Low to Moderate chance the Design Control will detect a potential cause/mechanism and subsequent failure mode
2	Moderately High	Moderately High to High chance the Design Control will detect a potential cause/mechanism and subsequent failure mode
1	Very High/Almost Certain	Design Controls will almost certainly detect a potential cause/mechanism and subsequent failure mode

Concepts NREC

Technical Accomplishments and Progress
Material Coatings to Inhibit Hydrogen Embrittlement

- Paraphrasing comments made during many interviews with researchers, the quick answer is: **"…no known coating materials exist to prevent hydrogen diffusion and hence the embrittlement of the base material"**
- Texas A&M conducts coating experiments with coatings recommended by *CN* and others
  - Proposed coatings are all aluminum oxide-based (although DLC, Diamond-like Coating, has also been considered, but discouraged by Texas A&M and others)
    - Metallic hydride, tungsten and tungsten carbide, TiO_2, CrO_3
    - Accuratus (APS Company)
    - Alodine EC^2 ElectroCeramic (Henkel Corp)
    - SermaLon (Sermatech International)
- Some structural concerns:
  - Can the coating be applied without affecting compressor material or vane design?
  - Will it compromise the base material by exposing even a small activation site on the base material if coating is chipped, cracked, or otherwise broken?
  - Will it contaminate the hydrogen during long-term use?

Technical Accomplishments and Progress
Materials Selection and Testing Summary

- A wide-ranging literature search and personal discussions with materials researchers (Sandia, Savannah River, Argonne National Laboratories, Failure Analysis Associates) have been conducted (and are continuing)
  - Most hydrogen embrittlement material studies have focused on steels and titanium alloys
  - There is agreement that aluminum alloy is protected from hydrogen embrittlement by its quickly formed oxide layer and the extremely slow diffusion of hydrogen into the metal
- From a turbomachinery design focus:
  - Aluminum (alloys: 1100, 2024, 3003, 2618-T6, 2918-T81 and 7075 - Matl. design choice) is light but strong (as evidenced by its relatively high specific strength), comparable to titanium and thus very suitable for centrifugal compressor applications
  - However, titanium is recognized by most researchers as affected by hydrogen embrittlement, but alloy Ti Grade 2 will be tested to check coating efficacy
- Collaboration with Texas A&M (Dr. Hong Liang) and coordinating their tests with two National Labs (using a small diameter punch test apparatus) is in progress to conduct relevant tests with aluminum per a Test Protocol derived from discussions with researchers, including:
    - Sandia National Labs (fracture mechanics testing; Dr. Chris San Marchi)
    - Savannah River National Labs (specimen “charging” with hydrogen plus tensile testing with H_2; Dr. Andrew Duncan)
    - Argonne National Labs (Dr. George Fenske)

Accomplishment Details (1)

- **Developed Computer Models to Aid in Analysis of Hydrogen Compressor**
  - Compressor Module Performance and Cost Model
    - Suitable as a macro for DOE “HDSAM v2.0” Economics Model
    - Provides a single point summary of each of the components within the package:
      1. Compressor rotor aerodynamics (pressure ratio, power, speed vs. flow rate, and intercooler pressure drop)
      2. Intercooler size vs. effectiveness (i.e., desired outlet temperature)
      3. Electric motor power
      4. Overall hydrogen efficiency based on compressor power, component efficiencies
      5. Compressor shaft diameter sizing based on fatigue loading
      6. Impeller radial and axial loadings calculated

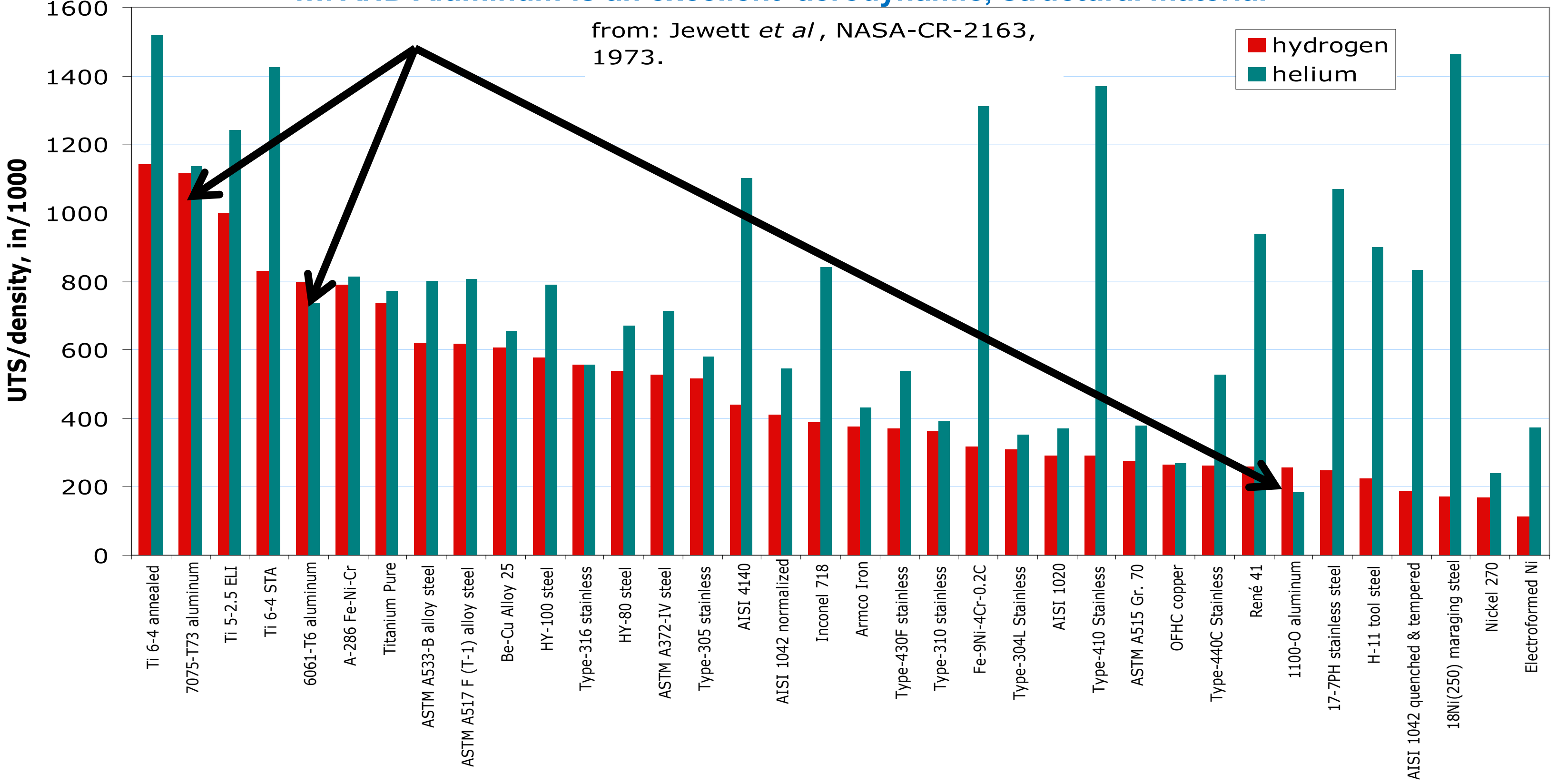
Accomplishment Details (2)

System Reliability and Maintenance Cost Model

- Engineering Reliability and Maintenance Cost Model that uses a consistent methodology and algorithms to determine the relative reliability and maintenance cost for a piston and centrifugal compressor pipeline package
- Uses either manufacturer's reliability of individual components or subsystems that constitute a compressor system (preferred) or textbook values
- Uses FERC operation and maintenance database as the basis for determining the maintenance costs for a centrifugal compressor.
- Uses Failure Mode Effects Analysis as developed by Concepts NREC for this project.

Technical Accomplishments and Progress

Aluminum Material Selection based on Prior Research combined with Compressor Aerodynamic Design Experience Applied - 1



Material's Specific Strength
under 10,000-psi Gas
-room temperature, notched specimens-

Aluminum Alloy shows no effect from hydrogen in Rocketdyne Lab Tests

…. AND Aluminum is an excellent aerodynamic, structural material

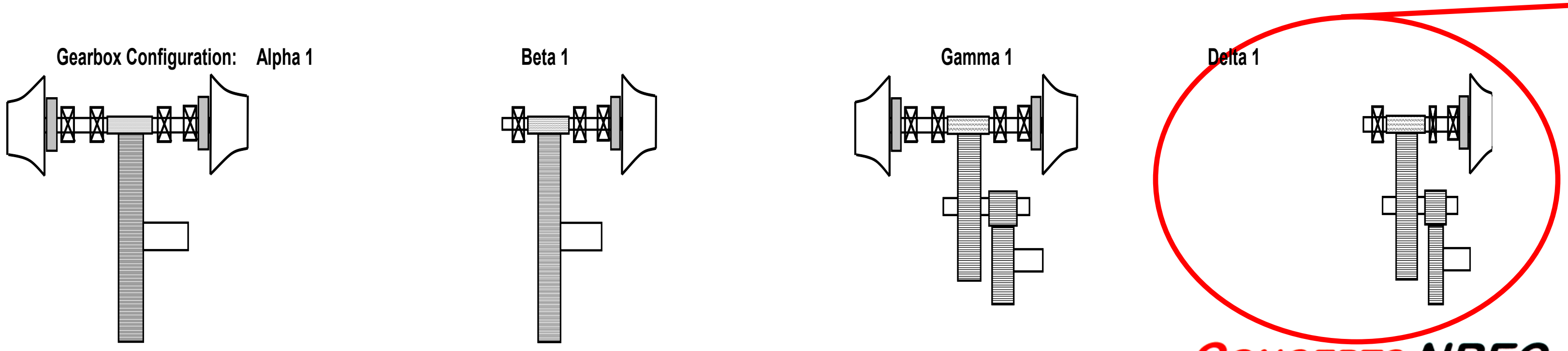
from: Jewett *et al*, NASA-CR-2163, 1973.

Technical Accomplishments and Progress
Compressor-Gearbox “Best Case” Selection Based on Relative O&M and Mechanical Risk Analysis

- Driver speed (1,800, 3,600, and 10,000 rpm)
- Number of stages (4, 6, and 7)
- Number of intercoolers (3 or 5) for impeller temp. < 140°F
- Pressure loss in intercooler and interconnect piping
- Number of drive shafts and number of impellers per shaft drive (1 or 2 impellers per drive shaft)
- Compressor aerodynamics and geometry
  - Hydrogen flow rate
  - Compressor impeller speed (50k to 90krpm)
  - Stage pressure ratio
  - Effect of forward sweep to reduce tip speed for same stage pressure ratio
  - Inlet guide vanes
  - Use of compressor inlet swirl to increase pressure ratio
- Over 30 alternative compressor-gearbox configurations, materials, and compressor drive options (including gas turbine drives with heat recovery for intercooler cooling) studied and evaluated using a Relative Risk and Relative Cost Optimization Program Developed for the project culminating in a “Best” choice for the compressor package

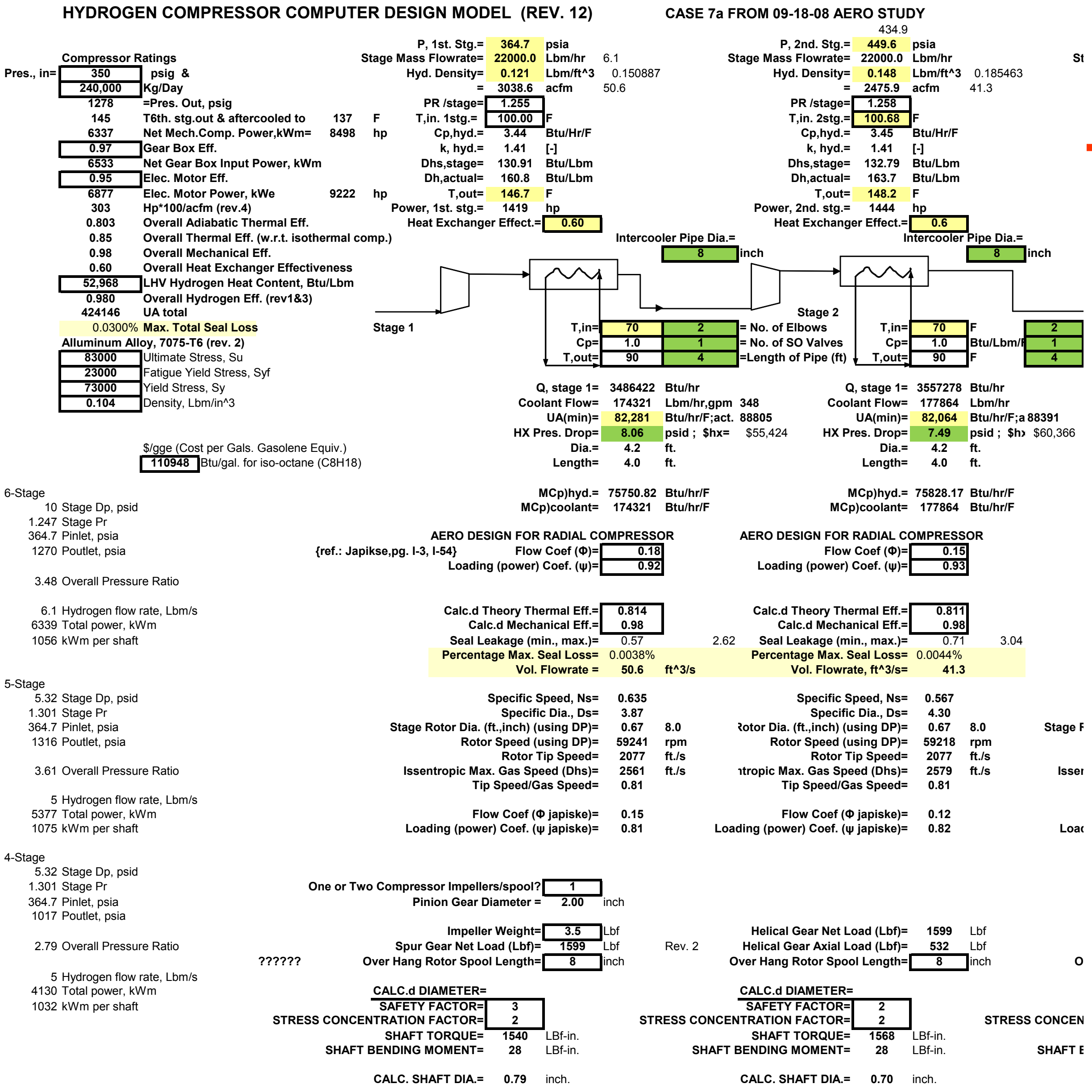
	Compressor Configurations "A" through "F"					
Configuration Designation:	A	B	C	D	E	F
Impeller Speeds (rpm):						
1st and 2nd	70,000	50,000	50,000	50,000	50,000	60,000
3rd and 4th	80,000	50,000	60,000	50,000	50,000	60,000
5th and 6th	90,000	50,000	70,000	50,000	50,000	60,000
Total Power, hp=	8,360	8,354	8,450	8,610	8,543	8,349
Max. Tip Speeds,ft/s=	2,236	2,178	2,194	2,194	2,101	2,094
Avg. Pres. ratio=	1.243	1.243	1.243	1.243	1.2541	1.247

BEST CHOICE



CONCEPTS NREC

Typical output from compressor station performance model - showing first of six stages



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HYDROGEN COMPRESSOR COMPUTER DESIGN MODEL  (REV. 12)          CASE 7a FROM 09-18-08 AERO STUDY
                                                                                         434.9
                                          P, 1st. Stg.=  364.7   psia                        P, 2nd. Stg.=  449.6   psia
         Compressor Ratings        Stage Mass Flowrate= 22000.0  Lbm/hr   6.1         Stage Mass Flowrate= 22000.0  Lbm/hr               St
Pres., in=   350      psig  &            Hyd. Density=  0.121    Lbm/ft^3  0.150887         Hyd. Density=  0.148    Lbm/ft^3  0.185463
          240,000  Kg/Day                           =  3038.6   acfm      50.6                         =  2475.9   acfm      41.3
             1278  =Pres. Out, psig                PR /stage=  1.255                              PR /stage=  1.258
              145  T6th. stg.out & aftercooled to   137   F    T,in. 1stg.=  100.00  F             T,in. 2stg.=  100.68  F
             6337  Net Mech.Comp. Power,kWm=  8498  hp         Cp,hyd.=  3.44   Btu/Hr/F             Cp,hyd.=  3.45   Btu/Hr/F
             0.97  Gear Box Eff.                                k, hyd.=  1.41   [-]                  k, hyd.=  1.41   [-]
             6533  Net Gear Box Input Power, kWm              Dhs,stage=  130.91  Btu/Lbm           Dhs,stage=  132.79  Btu/Lbm
             0.95  Elec. Motor Eff.                           Dh,actual=  160.8   Btu/Lbm           Dh,actual=  163.7   Btu/Lbm
             6877  Elec. Motor Power, kWe      9222   hp          T,out=  146.7   F                     T,out=  148.2   F
              303  Hp*100/acfm (rev.4)               Power, 1st. stg.=  1419   hp          Power, 2nd. stg.=  1444   hp
            0.803  Overall Adiabatic Thermal Eff.   Heat Exchanger Effect.=  0.60         Heat Exchanger Effect.=  0.6
             0.85  Overall Thermal Eff. (w.r.t. isothermal comp.)   Intercooler Pipe Dia.=          Intercooler Pipe Dia.=
             0.98  Overall Mechanical Eff.                                 8   inch                            8   inch
             0.60  Overall Heat Exchanger Effectiveness
           52,968  LHV Hydrogen Heat Content, Btu/Lbm
            0.980  Overall Hydrogen Eff. (rev1&3)
           424146  UA total                                                          Stage 2
      0.0300%  Max. Total Seal Loss        Stage 1        T,in=  70    2  = No. of Elbows       T,in=  70   F          2
         Alluminum Alloy, 7075-T6 (rev. 2)                  Cp=  1.0   1  = No. of SO Valves      Cp=  1.0  Btu/Lbm/   1
            83000  Ultimate Stress, Su                   T,out=  90    4  =Length of Pipe (ft)  T,out=  90   F          4
            23000  Fatigue Yield Stress, Syf
            73000  Yield Stress, Sy                 Q, stage 1=  3486422  Btu/hr           Q, stage 1=  3557278  Btu/hr
            0.104  Density, Lbm/in^3              Coolant Flow=  174321   Lbm/hr,gpm  348   Coolant Flow=  177864   Lbm/hr
                                                      UA(min)=  82,281   Btu/hr/F;act.  88805   UA(min)=  82,064   Btu/hr/F;a  88391
                                                HX Pres. Drop=  8.06     psid ;  $hx=   $55,424   HX Pres. Drop=  7.49  psid ;  $hx  $60,366
                   $/gge (Cost per Gals. Gasolene Equiv.)       Dia.=  4.2   ft.                      Dia.=  4.2   ft.
                   110948  Btu/gal. for iso-octane (C8H18)    Length=  4.0   ft.                    Length=  4.0   ft.

6-Stage                                                  MCp)hyd.=  75750.82  Btu/hr/F          MCp)hyd.=  75828.17  Btu/hr/F
         10  Stage Dp, psid                          MCp)coolant=  174321    Btu/hr/F       MCp)coolant=  177864    Btu/hr/F
      1.247  Stage Pr
      364.7  Pinlet, psia              AERO DESIGN FOR RADIAL COMPRESSOR           AERO DESIGN FOR RADIAL COMPRESSOR
       1270  Poutlet, psia     {ref.: Japikse,pg. I-3, I-54}   Flow Coef (Φ)=  0.18              Flow Coef (Φ)=  0.15
                                             Loading (power) Coef. (ψ)=  0.92    Loading (power) Coef. (ψ)=  0.93
       3.48  Overall Pressure Ratio

        6.1  Hydrogen flow rate, Lbm/s       Calc.d Theory Thermal Eff.=  0.814       Calc.d Theory Thermal Eff.=  0.811
       6339  Total power, kWm                   Calc.d Mechanical Eff.=  0.98           Calc.d Mechanical Eff.=  0.98
       1056  kWm per shaft                   Seal Leakage (min., max.)=  0.57    2.62   Seal Leakage (min., max.)=  0.71    3.04
                                         Percentage Max. Seal Loss=  0.0038%        Percentage Max. Seal Loss=  0.0044%
                                                    Vol. Flowrate =  50.6   ft^3/s        Vol. Flowrate, ft^3/s=  41.3
5-Stage
       5.32  Stage Dp, psid                       Specific Speed, Ns=  0.635                Specific Speed, Ns=  0.567
      1.301  Stage Pr                              Specific Dia., Ds=  3.87                  Specific Dia., Ds=  4.30
      364.7  Pinlet, psia    Stage Rotor Dia. (ft.,inch) (using DP)=  0.67   8.0   Rotor Dia. (ft.,inch) (using DP)=  0.67   8.0        Stage F
       1316  Poutlet, psia               Rotor Speed (using DP)=  59241  rpm         Rotor Speed (using DP)=  59218  rpm
                                               Rotor Tip Speed=  2077   ft./s              Rotor Tip Speed=  2077   ft./s
       3.61  Overall Pressure Ratio  Issentropic Max. Gas Speed (Dhs)=  2561  ft./s   ntropic Max. Gas Speed (Dhs)=  2579  ft./s   Isser
                                           Tip Speed/Gas Speed=  0.81               Tip Speed/Gas Speed=  0.81
          5  Hydrogen flow rate, Lbm/s
       5377  Total power, kWm               Flow Coef (Φ japiske)=  0.15              Flow Coef (Φ japiske)=  0.12
       1075  kWm per shaft       Loading (power) Coef. (ψ japiske)=  0.81   Loading (power) Coef. (ψ japiske)=  0.82              Loac

4-Stage
       5.32  Stage Dp, psid
      1.301  Stage Pr          One or Two Compressor Impellers/spool?   1
      364.7  Pinlet, psia                   Pinion Gear Diameter =  2.00   inch
       1017  Poutlet, psia
                                                  Impeller Weight=  3.5   Lbf          Helical Gear Net Load (Lbf)=  1599   Lbf
       2.79  Overall Pressure Ratio     Spur Gear Net Load (Lbf)=  1599  Lbf   Rev. 2  Helical Gear Axial Load (Lbf)=  532   Lbf
                       ??????   Over Hang Rotor Spool Length=  8     inch     Over Hang Rotor Spool Length=  8     inch           O
          5  Hydrogen flow rate, Lbm/s
       4130  Total power, kWm             CALC.d DIAMETER=                            CALC.d DIAMETER=
       1032  kWm per shaft                   SAFETY FACTOR=  3                           SAFETY FACTOR=  2
                          STRESS CONCENTRATION FACTOR=  2       STRESS CONCENTRATION FACTOR=  2                 STRESS CONCEN
                                       SHAFT TORQUE=  1540   LBf-in.                SHAFT TORQUE=  1568   LBf-in.
                               SHAFT BENDING MOMENT=  28     LBf-in.        SHAFT BENDING MOMENT=  28     LBf-in.           SHAFT E

                                  CALC. SHAFT DIA.=  0.79   inch.              CALC. SHAFT DIA.=  0.70   inch.
```

CONCEPTS NREC

Developed a System Reliability and Maintenance Cost Analysis Methodology

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

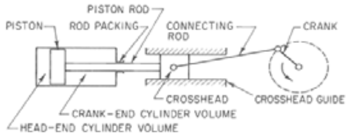


Fig. 6-82 Double-acting design.

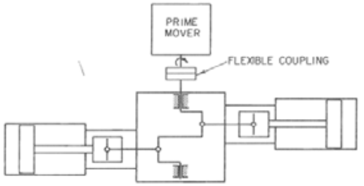


Fig. 6-83 Balanced-opposed compressor

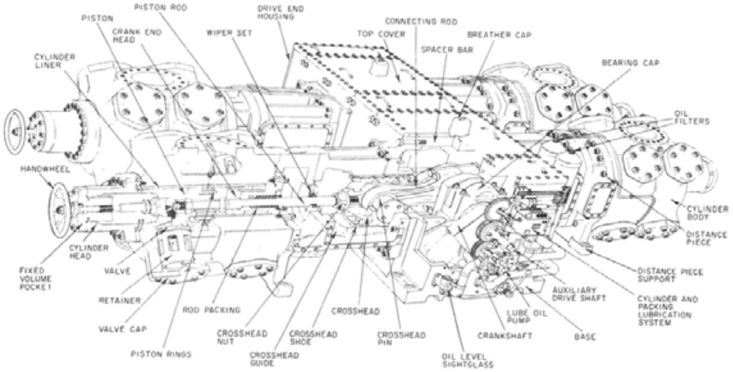
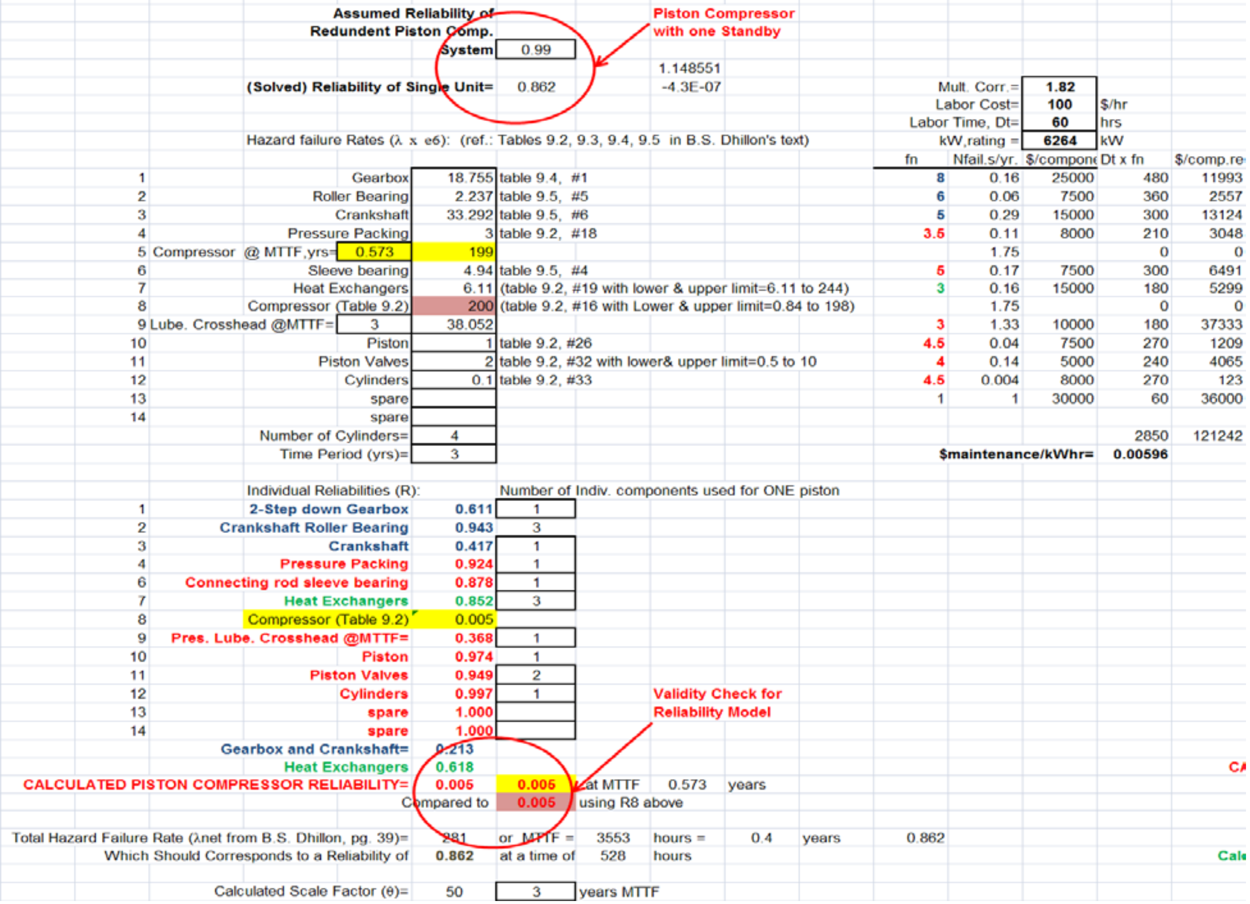


Fig. 6-85 HVC engine—compressor.



Assumed Reliability of Redundent Piston Comp. System 0.99

(Solved) Reliability of Single Unit= 0.862

1.148551
-4.3E-07

Piston Compressor with one Standby

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

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).

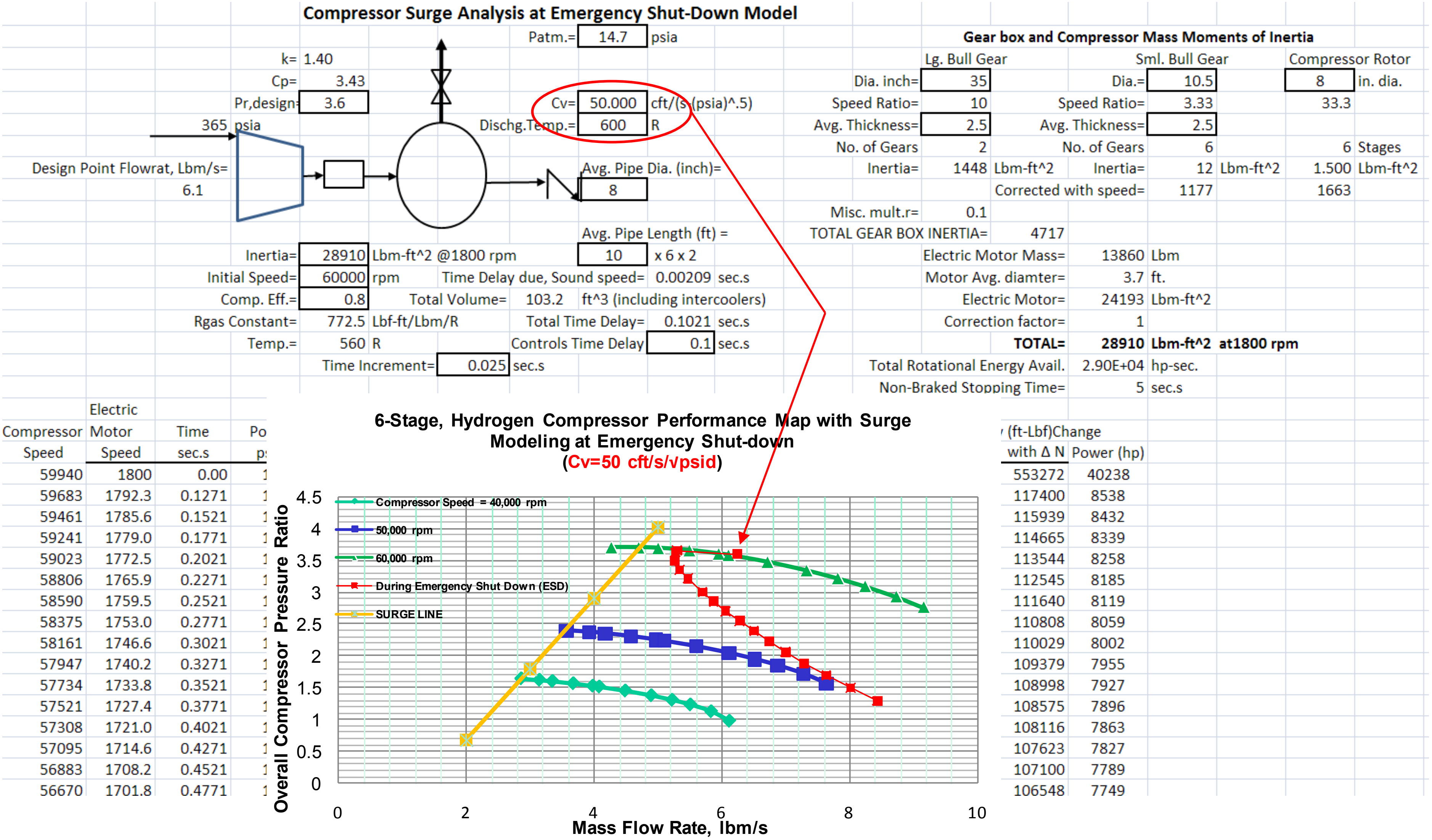
Hydrogen Piston Cost ($) and Operation & Maintenance ($/kWhr) Using DOE's HDSAM v.2 Economics

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

Anti-surge Analysis: Emergency Shutdown Model



Compressor Surge Analysis at Emergency Shut-Down Model

Patm.= 14.7 psia

k= 1.40

Cp= 3.43

Pr,design= 3.6

365 psia

Cv= 50.000 cft/(s-(psia)^.5)

Dischg.Temp.= 600 R

Design Point Flowrat, Lbm/s= 6.1

Avg. Pipe Dia. (inch)= 8

Avg. Pipe Length (ft) = 10 x 6 x 2

Inertia= 28910 Lbm-ft^2 @1800 rpm

Initial Speed= 60000 rpm

Time Delay due, Sound speed= 0.00209 sec.s

Comp. Eff.= 0.8

Total Volume= 103.2 ft^3 (including intercoolers)

Rgas Constant= 772.5 Lbf-ft/Lbm/R

Total Time Delay= 0.1021 sec.s

Temp.= 560 R

Controls Time Delay 0.1 sec.s

Time Increment= 0.025 sec.s

Gear box and Compressor Mass Moments of Inertia

Lg. Bull Gear			Sml. Bull Gear			Compressor Rotor	
Dia. inch=	35		Dia.=	10.5		8	in. dia.
Speed Ratio=	10		Speed Ratio=	3.33		33.3	
Avg. Thickness=	2.5		Avg. Thickness=	2.5			
No. of Gears	2		No. of Gears	6		6	Stages
Inertia=	1448	Lbm-ft^2	Inertia=	12	Lbm-ft^2	1.500	Lbm-ft^2
			Corrected with speed=	1177		1663	

Misc. mult.r= 0.1

TOTAL GEAR BOX INERTIA= 4717

Electric Motor Mass= 13860 Lbm

Motor Avg. diamter= 3.7 ft.

Electric Motor= 24193 Lbm-ft^2

Correction factor= 1

TOTAL= 28910 Lbm-ft^2 at1800 rpm

Total Rotational Energy Avail. 2.90E+04 hp-sec.

Non-Braked Stopping Time= 5 sec.s

Compressor Speed	Electric Motor Speed	Time sec.s	Po p
59940	1800	0.00	1
59683	1792.3	0.1271	1
59461	1785.6	0.1521	1
59241	1779.0	0.1771	1
59023	1772.5	0.2021	1
58806	1765.9	0.2271	1
58590	1759.5	0.2521	1
58375	1753.0	0.2771	1
58161	1746.6	0.3021	1
57947	1740.2	0.3271	1
57734	1733.8	0.3521	1
57521	1727.4	0.3771	1
57308	1721.0	0.4021	1
57095	1714.6	0.4271	1
56883	1708.2	0.4521	1
56670	1701.8	0.4771	1

(ft-Lbf)Change with Δ N	Power (hp)
553272	40238
117400	8538
115939	8432
114665	8339
113544	8258
112545	8185
111640	8119
110808	8059
110029	8002
109379	7955
108998	7927
108575	7896
108116	7863
107623	7827
107100	7789
106548	7749

6-Stage, Hydrogen Compressor Performance Map with Surge Modeling at Emergency Shut-down
(Cv=50 cft/s/√psid)

CONCEPTS NREC

Project Objectives – Relevance to DOE Hydrogen Economy Planning

DOE Stated Objectives :

- Develop and demonstrate an advanced centrifugal compressor system for high-pressure hydrogen pipeline transport to support DOE's strategic hydrogen economy infrastructure plan
- Deliver 100,000 to 1,000,000 kg/day of 99.99% hydrogen gas from generation site(s) to forecourt stations
- Compress from 350 psig to 1,000 psig or greater
- Reduce initial installed system equipment cost to less than $9M (Compressor Package of $5.4M) for 200,000 kg/day system by FY 2017
- Reduce package footprint and improve packaging design
- Achieve transport delivery costs below $1/GGE
- Reduce maintenance cost to below 3% of Total Capital Investment by FY 2017
- Increase system reliability to thus avoid purchasing redundant systems
- Maintain hydrogen efficiency (as defined by DOE) to 98% or greater
- Reduce H2 Leakage to less than 0.5% by FY 2017

DOE Stated Technical Barriers to Establishing Hydrogen as Viable Alt. Fuel:

- This project addresses the following technical barriers from the Delivery (Section 3) of the Hydrogen, Fuel Cells and Infrastructure Technologies Program Multi-year Research, Development and Demonstration Plan:
  - (B) Reliability and Costs of Hydrogen Compression

PHASE II – Detailed Engineering Design

OBJECTIVE:

The overall objective of Phase II is to undertake critical components testing and development, and based on the results, prepare a detailed design and cost analysis of a complete pipeline compressor system. This design will incorporate all the necessary subsystems for stand-alone testing in an actual pipeline system environment. In particular, fabrication and laboratory testing will be performed to verify design parameters for bearings, seals, impellers, and materials in a hydrogen environment. In addition, a laboratory validation test unit will be designed to enable the testing of a partial integrated assembly to take place in Phase III. At the conclusion of this task, a Go/No-Go decision will be made with regard to proceeding into Phase III.

2.1 Detailed Subsystems Modeling

The objective of this task is to prepare detailed analytical models of the centrifugal compressors, gearbox, intercoolers, and prime mover to establish the specific design parameters from which to prepare detailed designs. Analytical modeling will be conducted with regard to various aerodynamic design tradeoffs that affect compressor performance, impeller stress, and dynamic stability. This work will also include the design of the high-speed gearbox (bearing loads, seals, lubrication, etc.), prime mover, and control system. Current design practices, as well as advanced concepts, will be factored into the model to identify critical areas of concern, design approaches, and if necessary, future mitigation design strategies.

2.2 Detailed Integrated Systems Analysis

In parallel with Task 3 2.1, Subsystems Modeling, a detailed integrated system analysis will be performed that defines the predicted performance of the system under alternative operating conditions consistent with the design criteria and specifications defined in Task 3.1.5. This work will include process flow and instrumentation diagrams, mass flow and energy balances, and control strategies.

2.4 Critical Components Testing and Development

The objective of this task is to design, fabricate, and test critical components under simulated operating conditions to validate predicted design. Worst-case operating conditions of the impellers, seals, and bearings will be defined, and high-speed dynamic testing under controlled laboratory conditions will be undertaken. High-speed spin tests will be conducted to validate predicted stresses at various speeds, including operation to failure to define the ultimate stress limit of the impeller. Dynamic stability limits will also be verified.

PHASE II – Detailed Engineering Design (continued)

2.5 Integrated System Design

In this task, two designs, the first for a complete multistage system, and a second for a limited overall pressure ratio two-stage compressor system will be prepared in sufficient detail to estimate the cost of each system. The two-stage compressor system will include all the subsystems, but will operate at a reduced overall pressure ratio and power input to facilitate laboratory testing and development. This will include the compressors, intercoolers, gearbox, motor, lubrication system, skid, and controls. Quotations will be requested for the two-stage compressor equipment to be built and tested in Phase III.

2.6 Detailed Cost Analysis

A detailed manufacturing, operating, and maintenance costs analysis of the proposed system will be prepared. Using established scaling laws, the capital costs of various size systems up to 1 million kg/day will be estimated.

2.7 Revised Phase III Program Plan (Go/No-Go Decision)

This task is to revise the original Phase II Plan to reflect the current program development status. This task reflects the second Go/No-Go decision point in the program. Given the decision to move ahead, a revised program plan will be prepared reflecting the present level of development and critical technology hurdles that must be overcome to achieve the design goals. This plan will include a revised task, schedule, and cost plan with recommendations regarding accelerating, eliminating or redirecting certain activities. This plan will be submitted to the DOE Program for review and approval before proceeding into the next phase of the program.

2.8 Program Management and Reporting

The Program Manager will set goals, plan their accomplishment, maintain effective personnel on the project, negotiate and administer agreements between all participants including subcontractors, and deliver all contract commitments. Periodic status and other report obligations will be submitted to document and summarize the program. A DOE Phase II Final Report, including Topical reports for Tasks 3.2.5, 3.2.6, and 3.2.7, will be prepared.