Proceedings of the Fortieth Turbomachinery Symposium
September 12-15, 2011, Houston, Texas

BUYING / SELLING SERIAL #1

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ABSTRACT

The machinery utilized in the Oil and Gas Industry, (Upstream, Midstream and Downstream) is undergoing very significant changes. In the past years, the use of new designs or prototype machinery was strongly discouraged. Changes were modest and taken slowly. However, rapidly escalating changes in operating environments and requirements are requiring users and manufacturers to pursue "Step Outs" or, as defined in this paper, Serial #1 machinery. The authors detail some of the design challenges being encountered by users and manufacturers. A number of specific design challenges that have been encountered in the centrifugal compressor arena are described. Considerable engineering and design detail for these specific challenges are discussed. It is hoped that this discussion will provide the readers with some level of understanding of both the challenges they are facing and the manner in which they may address them while managing the risk inherent in these designs.

INTRODUCTION

Why would any sane machinery engineer want to buy SERIAL NUMBER ONE turbo machine? And, what sane machinery manufacturer would want to sell SERIAL NUMBER ONE? Interesting questions that most likely were drilled into your heads by your mentors and peers as you grew up in the industry. And, to make sure we are all on the same page, by SERIAL NUMBER ONE, we mean the first operating version of a particular machine design. It may contain new components, may be an entire new design, or may be operating in an environment or operating regime for which this class of machine has no operating experience.

But, our world is changing. In Kazakhstan, there are injection compressor applications and natural gas services that are designed for 1000 psi (690 bar) discharge pressures in the Tenghiz facility. When this project was conceived, there was no operating experience in this compressor service. Figure 1 depicts the wellhead to delivery layout of these compressors in both injection and acid gas injection services. Figure 2 shows the process layout of the three compressor bodies in this service. The reference document (Hopper et al (2008)) describes the manner in which the participants in the project proceeded to their ultimate success. Others in the industry have been built for sour services approaching 40% H_2S. Subsea pumps, pumps under 1800 - 2500 meters (5900 ft to 8200 ft) of water, have been designed and installed in offshore Angola. In Figure 3, there is a cutaway of a typical subsea helicoaxial pump and impeller section. In Figure 4, there is a cutaway of a section of the Serial #1 rotor that was designed for this Angola service. This service and pump design is discussed in the reference3 (Bibet et al (2009). Figure 5 shows an Electro Submersible Pump as designed for service in the Gulf of Mexico in deepwater service. This is discussed in the reference (Gould and Loaiza (2010)). At numerous sites around the world, there are water injection pumps operating at over 5000 PSI discharge (350 bar). One such service is shown in Figure 6 from a Gulf of Mexico installation which is rated at 1458 US GPM (338 m3/hr) and 8575 psi (605 bar). This service is discussed in Waterfield et al (2002). There are other services in the United States and elsewhere where CO_2 dense phase pumps and compressors are in operation at a range of pressures and there are demands for more severe duties, some in combination with sour gas. There are very large LNG Propane Refrigeration Compressors being purchased for services of over 150 MW (200,000 bhp) in Qatar and Australia with more on the horizon. This discussion just shows a few of the examples in the world. There are many, many more engaging just about all machinery manufacturers and many or all of the major and minor users.

These SERIAL NUMBER ONE applications have occurred and will continue to occur.

This tutorial will address techniques and processes that permit the USER and MANUFACTURER to approach these applications in a sound engineering manner that should limit the Risk to manageable levels while optimization the designs.
Why is this happening? Operating needs and requirements are moving rapidly away from HAPPY and/or SAFE ZONES in which there was a lot of experience into the unknown zone. The search and production of hydrocarbons has used up a large portion of the more common, old operating zones and the major companies around the world need to discover and operate in the new zones to keep filling the barrel, the pipeline, and the gas tank. This is a common mode around the world and includes most of the significant hydrocarbon companies around the world, national and international. In order to maintain and/or grow their business position, most of the turbomachinery manufacturers must play in this arena.

The detailed discussion in the tutorial will limit itself to centrifugal compressors. This is driven both due to some of the specific expertise of the presenters and the time available for the tutorial itself.

EVOLVING END USER REQUIREMENTS

Among the changing compression requirements being raised by the end users to the manufacturers are:

Acid Gas

Over the years, fields with high concentrations of acid gas were often bypassed because there were poor economics to producing them in comparison to fields with little or no acid gas. SERIAL NUMBER ONE compressors were needed which would be expensive to develop and build. And, the produced hydrocarbons might have their value debited from the presence of acid gas contamination. Thus, when these fields had been discovered in the past, they were usually bypassed and left in place. Demand and supply conditions have changed so that these fields now need to be produced. H_2S concentrations of as high as 40% are showing up on data sheets. In many of these fields, the compression services are being used to re-inject the gas after liquids are removed. Injection pressures of as high as WWW are being required.

One of the many problems with these services is the required material properties of the compressor construction. These require exotic materials which are often more difficult to obtain, form, and are always expensive. In addition, some of these conditions are new to the designer and determination of true material properties under operating conditions might require difficult and expensive testing.

But ACID GAS is here. A great example is the Shah Field in Abu Dhabi which Occidental recently contracted. The field located about 111 miles (180 KM) southwest of Abu Dhabi in the UAE, holds 200 trillion cubic feet (5.7 trillion cu.m.) of gas some or mush of which is sour. An investment of $10 billion is expected to recover and process the gas.

Multi-Phase Flow

Multi-phase operation requires a compressor to move a combination of gas and liquid. As long as the gas volume fraction is in excess of 90%, this can be done with more conventional equipment. However, that limit is being severely pushed. .In subsea applications, separation vessels can be outrageously expensive, especially in deep water where pressure ratings of vessels may be 10,000 psi or 15,000 psi (700 bar to 1,000 bar). . On platforms, separation vessels cost a penalty in both weight and footprint. In many applications, separation also requires parallel gas and liquid lines which can add greatly to the complexity and cost. In viscous crude applications, multiphase pumping can be more efficient since the gas acting as a significant friction reducer. And, in most applications, separation will lead to additional hardware costs and considerably more complexity to the configuration. In a number of the newer developing fields, the relative amount of gas on oil are not yet well understood and will change (often dramatically) over the life of a field.

Very Large Sizes and Capacities

LNG and Ethylene processes are relatively mature processes that may not have a lot of opportunity for process breakthroughs which increase the margins in these largely commodity businesses. One of the remaining means to increase margins and profitability is through economies of scale. These have led to increasing demands on the capacities of the compression trains in these facilities.

High Pressure

Higher pressure fields are being pursued at an increasing rate in many parts of the world, e.g., Kazakhastan, Indonesia, Oman. Injection services have climbed to 500 bar (7,250 psi) and as much as 800 bar (11,600 psi) requirements. These are considerable step-outs from more conventional pressures.

New Processes

GTL (Gas to Liquid) plants are being proposed and built on a very large scale. These facilities require very large air separation facilities. The GTL processes are exothermic but produce mediocre quality steam which must be consumed. Very large air compressors, greater than 80 MW (100,000 bhp) are being employed driven by steam turbines powered by 175-250 psi (12-17 bar) inlet steam to drive these machines.

Subsea

There is a desire and, in some cases, a requirement for turbomachinery to be deployed on the bottom of the sea in water depths of over 2000 meters (6,500 ft). This requires very high pressure casings, seal-less machines with electric motor drives, product lubrication (or a sophisticated composite system), and a footprint and weight that are manageable in these services. In addition, there is an extraordinarily high cost for intervention (maintenance) which drives the need for very predictable mean time between maintenance of the compressors and all their supporting systems. Figure 7 shows a 3-D view of a subsea compressor designed for the Ormen Lange field (Reference Vannini et al (2011). Figure 8 is a photo of the actual compressor-motor rotor hung for a free-free vibration test.

Seal-Less Compression

Above and below the water, there is a growing desire for seal-less compression. Drivers are weight and footprint limitations and the desire to limit the complexity of the overall system configuration. There is also a strong desire in hazardous services to positively contain the process fluids without the peripheral sealing devices that can add greatly to the likelihood of a loss of containment. Subsea compressors, by their nature are seal-less.

Limited Definition of Process Fluids or Process Requirements

Many of these services are operating well outside the "normal" boundaries of past experience. As such, it may be found that the properties of the fluids being moved are not well defined in all the ranges of operation. Gas property models may not have been established for the extremes of operation being required. In some cases, typical property models may not be sufficient and process simulations may be required. An example is the performance of high pressure CO2 with the presence of some water.

Material properties may also not be well defined in some of the extreme range of operations.

Finally, the definition of the process requirements in some of these applications may not be understood with a high level of accuracy. The performance of many of the "extreme" oil and gas fields must be extrapolated as there is very limited or no prior performance history from those fields.

OEM PRODUCT DEVELOPMENT

There are three primary reasons why OEMs develop new equipment or new components for existing equipment. First, they must respond to client demands. As noted above, there are a variety of reasons for changes in end users requirements. Regardless, if the OEM wishes to be a player in a particular market, they must supply equipment that satisfies the new or modified requirements. In many cases, this can involve upgrades to existing products but might also include the development of new machinery configurations. For example, as different liquefied natural gas (LNG) processes were developed, requirements arose for centrifugal compressors with varying numbers of sidestreams. At first, machines might have had one or possibly two sidestreams but now three sidestream units are commonplace. Still, the change from one to two to three required development on new styles of sidestreams to minimize bearing span while not compromising the aerodynamic design. Similarly, as the plant capacity for LNG increased, it was necessary for OEMs to develop higher capacity, high Mach number impellers to address these more demanding capacity requirements.

The second major reason for OEMs to develop new equipment is to maintain a competitive edge over competing OEMs. If an OEM can offer mechanical, performance, and/or cost advantages over their competition, they are in good position to capture a larger share of the market. Therefore, OEMs monitor the performance levels quoted by competitors and also keep tabs on the latest developments in centrifugal compressor technology to ensure that they are offering competitive products. If not, they must make the necessary investments to upgrade their products to meet the market levels or to develop a novel approach to meeting the client requirements while at the same time differentiating themselves from their competitors. Examples of novel solutions include the introduction of back-to-back compressors for re-injection services, welded versus riveted impellers, fabricated versus cast stationary components, etc.

Of course, other drivers that influence the ability to upgrade products are the advancements of manufacturing and analytical technologies. As described by Sorokes and Kuzdzal Sorokes and Kuzdzal (2010), there considerations have had a dramatic effect on the attainable aerodynamic and mechanical performance of centrifugal compressors. Advanced analytical techniques such as computational fluid dynamics (CFD) and finite element analysis (FEA) have provided OEMs with far more knowledge on the aerodynamic and mechanical behavior of compressor components, allowing designers to optimize such to provide increased performance. Similarly, the advanced manufacturing techniques have made it possible to produce the more complex designs necessary to provide the upgraded performance.

Finally, there are occasions when an OEM, through research and development efforts, discovers a novel technology that provides significant advancement of the state-of-the-art. The OEM will then seek to use the new technology either in existing products or, if the concept is sufficiently far-reaching, in a brand new product that has not previously been used in the industry. Examples of the former include: low solidity vaned diffusers; damper seals; damper bearings; or non-metallic labyrinth seals. The latter might include: the integrated motor/compressor or so-called “compact compression system”; the compander; or the rampressor.

END USER - WHAT IS NEEDED TO VALIDATE THE DECISION FOR BUYING SERIAL #1

The end point for the end user is an acceptable, completed Risk Assessment, i.e., the Cost versus Benefit to employ the new design. In some cases, e.g., subsea compression, there are no existing devices and a new design must be employed if any of the benefit is to be achieved. In many cases, conventional designs will not provide a sufficient profit margin to make a project viable. If a Serial Number 1 is not used, the project would be killed and no benefit would be accrued.

Risk is the likelihood of a negative result times the cost of that result. Cost is not only strictly financial but also includes other factors primarily safety and environmental issues along with reputation, capital exposure, etc. Every user and manufacturer has Risk Assessment protocols they use. They follow the same basic principals but most companies have their own terminology, work process, and "hurdle rates".

New designs are basically machines which are designed for operation that has never been achieved in the past by that class or size of machine. This may involve pressures, capacities, product processed, size and speed, efficiencies required, drivers utilized, etc.

Since the machine has not, as a whole, operated in the service, an evaluation must be performed. The compressor must be broken into critical components, sub-assemblies, or supporting systems.

Every manufacturer and user can produce one or more lists of components and systems. They are probably formatted and ordered based on individual preferences and experiences, both good and bad. A "typical" list prepared for this tutorial consists of:

Table of Evaluation Items

1. Aerodynamic Performance
   a. Swallowing Capacity
   b. Head Capability
   c. Efficiency
   d. Turndown Performance
   e. Gas Composition Flexibility
   f. Prediction Accuracy
   g. Phase Map Assessment
   h. Stationary Element Design
   i. Aerodynamic Flowpath Loading
2. Casing Design Study
   a. Pressure Casing
   b. Head Design
   c. Attachments - Instruments, inlet taps, outlet taps.
   d. Seal Housing
   e. Bearing Housing
   f. End Caps (barrel compressors)
   g. Weight and Footprint
   h. Manufacturability
   i. Assembly methods
3. Material Selection Study
   a. Material Corrosion Resistance
   b. Material Strength
   c. Wet Material Selections
   d. Dry Material Selections
   e. Non-metallic material selection
   f. Dew Point Control
   g. Welding/Brazing/etc
   h. Rotating Element Materials
   i. Shafting
   j. Impellers
4. Seal Design Study
   a. Shaft Sealing Arrangement
   b. Shaft Sealing Support System
   c. Materials of Construction
   d. Casing Sealing Design

5. Rotor Dynamic Study
    a. Analytical Tools Utilized
    b. Lateral Critical Speed Study
    c. Unbalance Response and Critical Speeds
    d. Stability Analysis
    e. Bearing Selection and Design
    f. Torsional Analysis
    g. Transient Analysis
6. Impeller Mechanics and Dynamics
    a. Stresses
    b. Modal Analysis
    c. SAFE Diagram
    d. Fatigue analysis
    e. Interference fits
7. Testing
    a. Aero Testing
    b. Mechanical Testing
    c. Full Load Mechanical Testing
    d. Hydrocarbon Testing
    e. Field Testing
    f. Component and model testing
8. Driver and Driver System
    a. Motor
    b. Steam Turbine
    c. Gas Turbine
    d. Synchronous Motor
    e. Supersynchronous Motor
    f. Gears
    g. Couplings
9. Process Requirements
    a. Operating Point and Range
    b. Guarantee Point and Range
    c. Design Point and Range
    d. Turndown Requirements
    e. Process Fluid Uncertainties
    f. Process Fluid Phase Map
    g. Trace Elements
    h. Contaminants

While the entire machine may not have been employed in the service specified, the individual components may be characterized as:

- Well proven components that are not utilized outside their experience range.
- Components whose range is being "stretched".
- Components which are completely new designs, e.g., new compressor wheel

The evaluation and qualification process will be different for each class of component.

In initiating and evaluation leading to qualification, the machine components must be broken down into these classes. The components must also be rated as to their criticality to overall Risk to successful operation. This requires cooperation and agreement between the user and manufacturer. Of significant criticality, this cooperation and agreement must extend beyond the qualification to management on both sides who have the authority to approve the applications.

Along with the assessment of the components, a very clear and rigorous scorecard needs to be developed which will be used to measure the success of the qualification process. Concrete measures must be defined and agreed to prior to the evaluations proceeding.

Qualification of Well Proven Components

Experience maps and operating histories are often used to define the Well Proven Components. These often will constitute a considerable part of a new design machine and lessen both the Risk and overall workload of the qualification team.

Qualification of Extended Range Components

Experiential data will be used to identify the limits of experience with these components. Analytical work is often used to identify the uncertainties that are added to the design by extending the operating ranges. In some cases, experimental or component test data may be required to validate the acceptability of these components.

Qualification of New Design Components

These are the most challenging components to evaluate in the qualification process. Often, insight into their capabilities may be obtained by discussion and evaluation of their precursors, their family tree. Heavy use is made of analytical assessments of the components. Often, the analytical tools used by the manufacturer must first be evaluated and accuracies defined. This may require test data. Once the analytical tools have been assessed, their results may be used to evaluate the designs.

In order to properly set the Risk, experimental or test work is often required. There are a variety of techniques that may be employed including scale model testing, simplified component testing, e.g., single wheel tests, or a complete prototype machine test. In some cases, field testing may be required to identify the Risk in sufficient detail. For example, subsea equipment might be tested on land or in shallow water. Machinery might be tested in less severe conditions for significant periods of time. Or, machines might be operated in non-critical services where downtime is not as costly.

Completion of Qualification Process

Of critical importance is the establishment of clear targets for qualification that are measurable. This has to be done prior to the qualification process. It should be clear whether a design is acceptable or not once the criteria are applied.

Risk Assessment

Performing a risk assessment or a series of risk assessments can be very beneficial in quantifying the risk involved and in determining the level of scrutiny that must be applied in any application. There are numerous tools and approaches (i.e. so-called risk matrices) that can be applied to perform such an analysis. Consideration must be given to technical as well as financial issues that would put a project at risk. The financial risk can be directly related to the technical risk because the new technology might not deliver the necessary performance or, if the product is late due because of excessive time required to resolve technical issues, production levels will be compromised and result in lost income.

In conducting a technical risk assessment, one can also evaluate the technology readiness level or TRL of the new product as well as the analytical tools being used to develop or predict the performance of the new product or component. A TRL assessment can provide guidance as to the level of uncertainty associated with any new technology or product that is being applied. In general, a lower TRL indicates that there is limited experience and/or experience with a given product or tool. Conversely, a high TRL means that one is dealing with "tried and true" technology and there is little, if any, risk in its use. More information on risk assessment and TRL can be found in Mankin (1995), Moses et al (2005), O'Neill et al (2007) and many other resources that can be easily located via the internet.

SOME OF THE TYPICAL ISSUES THAT MAY BE FOUND IN SERIAL NUMBER ONE COMPRESSORS

The due diligence normally employed in reviewing and validating a compressor design should be employed in all cases. However, when reviewing a SERIAL NUMBER ONE design, there are typically "potential problem areas" that may require additional care. These can be different depending upon the type of compressor and service being employed.

For example:

- For Very Large Compressors like LNG Propane Refrigeration or Ethylene plant Cracked Gas, typical issues may be:
  - High flow coefficient wheels that are necessary for the flows required. These may stretch the experience range of the manufacturer and, in some cases, require new designs. These will require and extraordinary assessment.
  - Large wheels have the potential for impeller resonances and overstress issues. An extraordinary impeller resonance assessment and loading assessment is often required.
  - Small errors in the performance calculations may result in significant horsepower requirement errors. Unless carefully evaluated, these may limit the compressor throughput.
- For Very High Discharge Pressure Compressors like a variety of injection compressors, some of the issues could be:
  - Shaft seal may be extended beyond their normal experience operational range. Extensive design reviews and/or testing may be required to "prove" the reliability of the seal design.
  - Case and head seals may be operating well beyond previous design experience. Considerable evaluation and potential testing may be required.
  - Material considerations with seals must evaluate "explosive decompression" issues very carefully.
  - Many of these designs require significant head per wheel to be developed. If the analyses are not sufficiently accurate or, if the gas in operation varies significantly from that shown on the design data, the compressor may not be able to perform.
- CO2 Compression
  - Some of the CO2 compressor services have the very high discharge pressure issues.
  - CO2 compression is operated in a portion of the phase diagram for which there is somewhat limited experience and testing. This is compounded by the fact that the gas properties can vary significantly greatly affecting the compressor performance.
  - Properties of CO2 can be quite sensitive to contaminants including H2O. Improper consideration of these contaminants can cause performance predictions to have significant errors.
- Acid Gas Compression
  - This may occur in conjunction with High Pressures and/or CO2 involving those issues.
  - Acid gas can drive corrosion and strength of material issues dramatically. Careful stress assessments will be required and most materials will have to be "de-rated" in these services to eliminate the possibility of wheel failures.
- Seal-less and Subsea Compression
  - Seal-less compressor designs must be carefully evaluated for maintainability issues. By their nature, they add complexity.
  - Subsea compression has extremely high intervention costs.
  - Seal-less compressors incorporate magnetic bearings. These must be addressed in the rotor dynamics assessments in a manner with significant differences from hydrodynamic bearings.
  - Gas density in a seal-less compressor can add significant viscous drag and loss of efficiency to the motor driver. It must be carefully evaluated.
  - Subsea compressors require very high pressure casings. These require careful stress analysis and deflection analysis. In addition, extreme care must be used in designing case penetrations for instruments and controls.

MECHANICAL FACTORS

From a mechanical perspective, the key to successfully buying a serial # 1 of any piece of equipment is to perform a very thorough review of the equipment's design and manufacture, including a thorough review of all testing including component NDT. Safety is of paramount concern, followed by functional performance, and ultimately reliability and life.

There is a substantial framework of industry standards which may be utilized to validate the equipment design and assess its worthiness and fit for service.

For example, if the new piece of equipment has to contain pressure, then the pressure containing envelope maybe validate against the requirements of the ASME Pressure Vessel Code. If the equipment has any penetrations through the pressure containing envelope such as for instrumentation, electric power, a rotating shaft such as in a centrifugal compressor, and any connections such as for process gas inlet and discharge, lube oil supply and drain, seal gas or seal oil supply, drain, and vent connections, then the sealing mechanism utilized to prevent leakage at each location needs to be reviewed for integrity and redundancy.

Two strategies to utilize in the above review process are to assemble a capable group of experienced personnel to function as a "Design Review Team", and secondly, to review the new design against the experience envelope of what has been done in this regard. For example, if the item under consideration is a new compressor case design, then comparing items such as hoop stress, longitudinal stress, contact stress between components, stress concentrations, and deflections, especially in seal areas, for the new design against the range of existing design experience. Plots of where the new case design resides against the prior experience may reveal areas for further study when points reside near of beyond the edge of the prior experience.

The "Design Review Team" is invaluable in the process of new design validation. Such teams are best composed of experienced people from within and without the designer's organization, who can be objective, and provide constructive feedback to the designers. People from the end user community, such as operators, maintenance personnel, and lead rotating equipment engineers, can provide a wealth of input.

Another strategy to employ is to look for what is different about a new design. Also, asking questions like how does this new design do its job better that the existing design is also a clever tactic to highlight items for further investigation or validation.

Benchmarking what is done in unrelated industries may also provide some valuable insight and guidance as to whether a serial # 1 is safe to buy. For example, the automobile and aircraft industries provide a multitude of examples to check and compare against.

Some of the tools which are needed for a thorough mechanical design review are;

1. Finite element analysis of every component and of various assemblies of components. This may include 3-D FEA to determine stresses, deflections, and component natural frequencies. Elasto-plastic FEAs of components that are highly stressed and dynamic FEAs to look at fatigue loading are important for rotating components such as impellers. The dynamic pressures to be utilized in the dynamic FEA are usually generated by a transient 3-D CFD analysis, or taken from dynamic pressure transducers during a test. Thermal analysis may also be warranted to determine the effect of operating temperatures on components fits and clearances.
2. Rotordynamic critical speed and stability analysis. This is critical to the proper functional operation of the compressor.
3. Dynamic System Response Analysis. No compressor is operated merely by itself. It is a part of a system that includes a driver, controls, various auxiliary support systems, and the end user's process plant. Simulating the start-up and shut-down of the compressor under normal, emergency, and off design operation is very important to meeting the reliability, safety, and functional performance aspects of the unit.
4. Experience Plots
   - Pressure vs. Flow
   - Delta-P per stage
   - Case delta-P
   - Impeller tip speed
   - Sealing pressure
   - Seal delta-P
   - Power per stage
   - Power per case
   - Head per casing
   - Etc.

Case Sizing

The general sizing of compressor casings is affected by a variety of factors including:

1. the dynamics of the rotor system
2. the radial space necessary to accommodate the aerodynamic flow path
3. the structural requirements to contain the compressor operating pressure, etc.

Obviously, once a rotordynamically L/D is established and once the aero flow path is determined, a compressor casing must be designed to encapsulate the design. The diameter of the casing must be large enough to accommodate the aerodynamic components. A critical issue here is the diffuser radius ratio required to achieve optimal static pressure recovery. The diffuser is largely responsible for converting the kinetic energy (or velocity pressure) added by the impeller into potential energy in the form of static pressure. Recall that it is static pressure that controls or drives the end user's process.

In very general terms, a longer diffuser results in higher static pressure recovery and higher static pressure recovery results in higher overall stage efficiency. This is especially true for the high flow coefficient stages normally found in LNG liquefaction compressors. Therefore, the casing inside diameter is a direct function of the radial space required to maintain the necessary diffuser length. If insufficient radial space is provided, the diffuser length will be inadequate and the compressor performance will not be optimal.

Once the casing length and inside diameter requirements are established, the designer will determine the details of the casing configuration necessary to withstand the operating pressures. Considerations include:

1. Case wall thickness
2. The need for support ribbing or others structures to minimize distortion at load
3. The horizontal split flange sizing, bolt size, and bolt spacing (for axially-split casings)
4. The head sizing, shear ring sizing, bolt sizing, etc. (for radially-split casings)
5. The material requirements (i.e., low temperature, high temperature, etc.), etc.

Finite element analysis (FEA) is commonly used in the development of new casings and designers rely heavily on the ASME Boiler and Pressure Vessel Code or the API-617 guidelines when establishing or assessing case configurations. Casings are also subjected to very stringent hydro-tests to ensure that no leaks occur during in-house testing and/or in field operation. In practice, FEA analyses are run at both the field operating conditions and hydro-test conditions to assess such factors as the horizontal split surface contact pressure, the case stress levels, and the loading on the bolts for axially-split compressor casings (see Figures 8, 9, and 10). If the stress analyses indicate that a leak will occur or that the stresses exceed the allowable levels (specified by the code or the

OEM's design guidelines), adjustments are made to the case geometry until all predicted leaks and/or high stress levels are eliminated.

Rotor Length / Diameter

One of the most critical mechanical issues is related to rotordynamics. To obtain acceptable rotordynamic performance, one must limit the ratio of the shaft length between the bearings to the shaft diameter. In general, compressors with length over width or L/D of 12 or greater could benefit from the use of technologies such as damper bearings, damper seals and the like.

Once an L/D limit is set, the maximum number of stages that can fit within a given compressor then becomes a function of the flow coefficients of the impellers required. Higher flow coefficient stages require wider flow paths and longer, sweeping turns in the impellers and stationary components to attain optimal performance. As such, higher flow coefficient stages require larger bearing span, thereby limiting the number of high flow coefficient stages one can put in a machine and still maintain an acceptable L/D. Conversely, low flow coefficient stages are short axially and many more stages can be installed within a given L/D limit.

Sidestream compressors provide a different challenge to the effective use of axial length or L/D. Because axial space is required to get the incoming sidestream flow introduced into the compressor, sidestreams occupy some of the space normally available for impellers or stages (see Figure 11). As such, the number of impellers or stages one can put into a sidestream machine of a given L/D is reduced over that of a conventional straight-through machine. While it might be possible to put 7 or 8 stages in a conventional machine of a given frame size, it may only be possible to install 5 or 6 stages in a multiple sidestream compressor. Therefore, it is difficult to establish a maximum number of stages per casing because it depends heavily on the application.

AERODYNAMIC CONSIDERATIONS

When faced with the need to justify and/or defend new aerodynamic components used in a SERIAL #1 application, an OEM will resort to three basic methods: (1) prior related experience, (2) advanced analytical methods, and (3) prototype or model testing. The approach taken will depend heavily on how far the new designs deviate from past practice and on how receptive the end user is to an analytical versus experiential validation of the new concept.

In an effort to improve the "read-ability" of this tutorial, some of the aerodynamic reference material is included in an Aero Appendix. Those wishing more detail certain topics will be directed to the appendix in the following discussions.

In considering the various examples cited above (i.e., very large / high capacity compressors, very high pressure compressors, CO2 compression, acid gas compressors, and seal-less or sub-sea compressors), there need not be any new aerodynamic components in most, if not all, of these applications. The flow path components used in very high pressure applications, acid gas compression, and sub-sea compressors can be the aerodynamic design as used by the OEM for other applications. That is, the critical aerodynamic dimension in those parts and the operating conditions under which they operate (i.e., Mach numbers, velocities, pressure ratios, etc.) fall well within the OEM's prior experience. Therefore, aerodynamically, the design can be justified and/or validated based on prior experience (See Experience Data in the AERO APPENDIX). In these cases, it will be the mechanical considerations that must be justified / validated via non-experiential means. For example, an impeller from the OEM's standard product offerings might provide the aerodynamic performance (i.e., efficiency, head coefficient, flow range) to satisfy a very high pressure or acid gas application requirement but in said compressor must operate a much higher pressure than previously experienced or in an environment that contains a much more corrosive gas. Therefore, additional stress work must be done to ensure the impeller can withstand the higher pressure forces or alternate construction materials / methods must be used to ensure the impeller can survive in the more corrosive environment. Still, neither of these considerations need impact the aerodynamic design of the components and so long as the critical aerodynamic parameters (passage widths, blade/vane angles, blade/vane leading and trailing edge radii, radii of curvature, etc.) remain unchanged, the OEM can use prior operating experience to justify / qualify the design. It must be noted that the validation of the aerodynamic design could well be done in compressors for other services (e.g., other gases, alternate inlet conditions) or in different machine sizes (e.g. in a large steel mill blower or a small ammonia machine) so long as the conditions are aerodynamically similar to those use in the new application.

When a new compressor component or combination of components (i.e., a new stage) represents a departure from past practice / experience, the OEM must resort to other means to justify / validate the new design to an end user. It might still be possible to validate the new design via past experience if: (a) it can be shown that the new stage was derived by interpolating among prior experience; or (b) if the new component / stage can be shown to be a minor extrapolation of prior art. In this case, the OEM would resort to the same approach used for existing designs. The OEM might also take advantage of geometric similitude (i.e., scaling) or contour trimming to draw in additional related experience.

Relating this to the examples cited in the introduction, suppose a new large scale, high flow compressor requires a stage with a flow coefficient that is not shown in the OEM's experience listing. If the flow coefficient falls between two impellers in the listing, the OEM could justify the new stage by showing it to be an interpolation between the two existing designs; i.e., a contour trim – See *contour trimming* section in the AERO APPENDIX. If the flow coefficient is 5% higher than any previously tested by the OEM, the OEM might still justify the new stage based on the prior related experience by showing that the new stage is simply a contour extension of the existing design. Such an approach is valid as long as the contour extension is **SMALL**; i.e., 5% or less. However, one must use great care when attempting a contour extension beyond 5% because of issues related to Mach number, curvature, and leading edge incidence.

If the entire machine is larger than any previously built by the OEM; i.e., is a larger frame size; the OEM will certainly resort to geometric scaling to validate / justify the new design. Geometric scaling is used by all OEMs to create compressors of varying sizes in the same manner as is used to create scale model planes, trains, and automobiles. This approach has frequently been used to develop the very large, high capacity compressors required for modern LNG facilities. As the demand for higher capacity plants; i.e., from 1 to 5 to 8 to 10 MTPA (million tons per annum); has grown, OEMs and plant designers alike have resorted to scaling to create the new larger compressors, pressure vessels, piping systems and the like. The basic approach used is described in the Geometric Scaling section in the AERO APPENDIX.

Of course, situations will arise that the OEMs are not able to address with previous aerodynamic experience. Relating to the cited examples:

- Very large, high capacity compressor for LNG or GTL plants can require impeller flow coefficients that are beyond industry experience.
- CO2 compressors can require both very high flow coefficients as well as aerodynamic designs that can function properly while handling extremely dense fluids. CO2 compression also introduces issue regarding gas properties.
- Multi-phase flow applications demand novel methods to address the complex aerodynamics associated with such machines

Flow coefficient is a parameter used by both OEM's and end users to indicate the flow carrying capacity of a compressor stage or impeller. More detail and definition is included in the AERO APPENDIX.

When no prior operating or related experience is available, the OEM must resort to either: (a) analytical validation (virtual testing); or (b) scale model or prototype testing. Because testing a new prototype can be very expensive and time-consuming, OEMs will first offer an analytical justification that the new concept will perform as predicted. In the centrifugal compressor industry, the most common type offered today is 3-D computational fluid dynamics (CFD) results. Akin to Finite Element Analysis in stress analysis, the 3-D CFD code breaks the aerodynamic flow path into small elements and determines the velocities, mass flux, etc. through the small elements to mathematically simulate the flow through the machine or machine component. Tremendous advances have been made in modeling schemes built into such codes and they have become a major part of the design and analysis processes for all OEMs. For a more detailed discussion on CFD, please refer to **Analytical Validation – The Virtual Test Rig** under AERO APPENDIX.

Because they are not encumbered with built-in empirical loss / performance models, CFD codes can provide an unbiased assessment of any new configuration. That is, the aerodynamic assessment is based purely on flow physics and not on the so-called tuned correlations (i.e., "fudge factors") found in typical 1-D or 2-D codes. There are no simple adjustments (such as slip factors, blockage factors, loss coefficients, efficiency multipliers) that can be made to a CFD analysis. Granted, if one does not use the proper rigor, it is possible to obtain totally erroneous results from a CFD analysis. Therefore, it is imperative that these results be thoroughly examined to ensure that proper techniques were applied; i.e., sufficient grid density, appropriate turbulence models, necessary and sufficient boundary conditions, etc.

Obviously, interpretation of the "pretty pictures" generated by the CFD analyses requires that the reviewer have some knowledge of the strengths and weaknesses of the current CFD tools. An end user can rely on internal expertise within his/her company, can trust the OEM to provide an honest assessment of the results or can bring in a third-party consultant to interpret the results. The option chosen will depend on a number of factors but is typically driven by the level of confidence and trust that has been established between the end user and OEM (more on this later).

What is commonly done in CFD analysis is to compare the results predicted for a new design to the CFD results for an existing design for which actual test data also exists. This "comparative" approach provides a reference to better assess the validity of the new CFD results. Examples of such comparative analyses can be found in Sorokes (1993), Sorokes & Hutchinson (2000), and Sorokes, et al. (2007), as well as in numerous other publications in the open literature.

Validation via Testing

If analytical results provide insufficient validation of the new design or product, the only remaining option is to validate the design via testing.

Testing tends to fall into three basic categories: (1) model testing; (2) sub-scale testing; and (3) full or near-scale testing. Some might group model and sub-scale testing into the same category and suggest that there are two basic categories. It should be noted that in some cases, the model testing will not be a true scale of any particular machine but will be a configuration that mimics the aerodynamic conditions; e.g. diffuser vanes in a wind tunnel.

Model Testing

Model testing is treated separately herein and refers to those tests which approximate the actual operating conditions that would be present in a real machine. It is possible to test new stationary components for a centrifugal compressor without an impeller up or downstream. For example, Sorokes et al [3, 6] described a series of tests conducted to assess various sidestream arrangements using a model in which flow was sucked through a scale model of the sidestream using an air blower. Similarly, Simpson et al (2008) and Aalburg et al (2008) described work done to develop a low radius ratio stationary component using a model that consisted of a quarter segment of the return channel passage. Such models, if properly instrumented, provide invaluable data on the performance characteristics of the new designs and can be used to demonstrate the effectiveness of such.

Sub-Scale Testing

A more common approach is sub-scale testing of a one or more centrifugal stages in their entirety. Numerous examples of such rigs can be found in the open literature, including the work of Benvenuti (1978), Sorokes and Welch (1991), Sorokes and Welch (1992), and Sorokes and Koch (1996). The cross-section of a typical "stage-and-a-half" rig can be seen in Figure

12. Such rigs can be very valuable to gather validation data on new stage components. Of course, the use of sub-scale models has been commonplace throughout history in the development of a wide range of products from automobiles and airplanes to buildings or entire cities. However, much as scale model cities do not capture all of the details within the individual buildings in the city, sub-scale turbomachinery rigs do not capture all factors that influence stage or component performance. Most rigs include only one impeller either upstream or downstream of the stationary components so they do not have a true centrifugal stage upstream or downstream; i.e., they are not multi-stage. Furthermore, most scaled test vehicles are much smaller than the product compressors they are meant to replicate. Therefore, so-called "size effects" or transfer functions must be addressed to translate the performance from the small rig to the full-scale machine. Such adjustments can lead to conflict between OEMs and end users if they do not agree on the correction factors.

When validating new designs via sub-scale testing, OEMs must provide prediction curves for the new components and/or stages and compare test results to show the level of agreement achieved. The OEM might also provide details of the instrumentation used, assessment of measurement accuracy (i.e., uncertainty analysis) and details on the test conditions to demonstrate that they are aerodynamic equivalent to the full-scale application. The OEM might or might not apply a correction for size effects depending on whether the application of such provides more comfort to the end user. For example, it is fairly commonly held that scaling from a small to a large size will result in a slight increase in capacity for the full-size machine (i.e., because boundary layer effects have less influence on the full-size than the sub-scale). Therefore, it might be possible to include a capacity correction factor from small to full size. However, if the end user is more interested in overload capacity than surge margin, the OEM could provide the results with no size correction, thus insuring a conservative estimate for the capacity of the full-sized machine.

Full or Near-Scale Testing

If the end user is not satisfied or convinced by model or sub-scale testing, the next logical approach is to validate the new design via a near full size or full size vehicle or prototype. These are somewhat common in many industries, including the gas turbine and industrial centrifugal compressor industries and there are numerous publications describing such vehicles (Sorokes / Koch – 1996, Sorokes et al – 2009, OTHERS). Of course, depending on the size of the equipment involved, full or near-scale test vehicles can be quite expensive, with costs in the millions of dollars. However, in many cases, the cost is justified given the increased revenue stream to the OEM and the decreased risk to the end user should the prototype prove the viability of the new design. Given the potential to significantly reduce operating risks, it is not uncommon that end users would participate with the OEM in funding the construction and/or testing of the full or near-scale prototype. It is also possible that the OEM would offer exclusive rights or possible "first rights" to the end user that participates in (i.e. helps fund) the test program.

Other Considerations

Not all new requirements or applications are necessarily specific to the aerodynamic flow path or mechanical components. In some cases, the new development relates to the use of a new gas mixture or possibly the compression of a common gas mixture to new pressure / temperature levels. Recent examples of these include the increased interest in the so-called "high pressure, acid gas compressors" as well as the environmentally-driven interest in CO_2 capture / sequestration. The former has a greater impact on the mechanical integrity of a compressor because new materials are required to withstand the corrosive nature of the "acid gas." The latter has a greater influence on the aerodynamics of the compressor due to the high density of CO_2 at elevated pressures.

From an aerodynamic perspective, the new gas properties will certainly have an impact on the velocities in the gas passages and could result in other issues related to volume reduction and, therefore, stage or component matching. However, these factors are easily addressed by the OEM once accurate gas properties are known. Therefore, the OEM and end user must agree on the methods used to establish said gas properties and also agree on the real gas equations that will be used to model said properties in performance prediction and test data reduction tools.

Similarly, the OEM and end user must agree on methods to establish the ability of construction materials to withstand the corrosive effects of "acid gases" or the like.

In the above cases and others similar, most OEMs and end users rely on third parties to conduct and report on the research studies necessary to establish the new gas and/or material properties. Organizations such as NASA, SwRI, and NIST as well as universities that specialize in such research can provide the information necessary to validate the properties and provide confidence to both the OEM and end user that the gases or materials are being properly modeled.

Of course, the most obvious new applications where gas properties are somewhat of an unknown are the high pressure acid gas and CO_2 compression systems. In particular, there is a need for very precise gas properties for CO_2 when operating at elevated pressures to ensure proper matching of the aerodynamic components both within a stage and from stage-to-stage. If there is uncertainty on the gas properties, it is not possible to accurately predict the pressure and temperature ratio, and, therefore, the volume reduction from one compressor stage to the next. The consequence would be a mismatch between the incoming flow and the impeller sizing. The result would be either premature choke or premature stall / surge depending on the direction of the gas property error. For more on aerodynamic matching, see Sorokes (2003).

If justifying or validating a new CO_2 compression system to an end user, the OEM must clearly demonstrate that the gas properties used conform to the standards established by mutually accepted reference publications; i.e., those from NIST, SwRI, NASA, etc. Otherwise, there can be no confidence that the OEM calculations accurately reflect the aerodynamic performance on the individual stages and overall machine.

MECHANICAL CONSIDERATIONS

Buying Serial #1 – Mechanical Aspects

The primary concerns for the mechanical side of a new equipment evaluation are safety, operability, reliability, robustness, and maintainability.

A primary safety concern is effective containment of a flammable or toxic gas under pressure. Allowing flammable or toxic gases to leak to the atmosphere is totally unacceptable. Another safety concern is that the pressure containing envelope contains any debris resulting from the ingestion of foreign objects, and damage to internal rotating components. Obviously, any debris or foreign objects ejected from the casing would be an extreme hazard to plant personnel or other equipment near the compressor.

Operability relates to the ability to operate, start, stop, the unit without incurring any unacceptable vibration, rubs, leaks, or damage to the unit whereas reliability is more concerned with how long or how often the unit maybe operated without requiring a shutdown for maintenance or incurring an unplanned forced shutdown..

Robustness is closely related to reliability, but refers more to how tolerant the unit is to off-design operation, unexpected changes to operating conditions, unexpected gas composition changes, such as entrained liquids, or particulates.

Maintainability is related to how user-friendly the unit is to disassemble, repair or replace wearing parts, and re-assemble for return to service.

From a mechanical perspective, the key to successfully buying a serial # 1 of any piece of equipment is to perform a very thorough review of the equipment's design and manufacture, including a detailed review of all testing including component NDT. Safety is of paramount concern, followed by functional performance, and ultimately reliability and life.

There is a substantial framework of industry standards that can be utilized to validate the equipment design and assess its worthiness and fit for service. For example, if the new piece of equipment has to contain pressure, then the pressure containing envelope maybe validate against the requirements of the ASME Pressure Vessel Code. If the equipment has any penetrations through the pressure containing envelope such as for instrumentation, electric power, a rotating shaft such as in a centrifugal compressor, and any connections such as for process gas inlet and discharge, lube oil supply and drain, seal gas or seal oil supply, drain, and vent connections, then the sealing mechanism utilized to prevent leakage at each location needs to be reviewed for integrity and redundancy.

Two strategies to utilize in the above review process are to assemble a capable group of experienced personnel to function as a "Design Review Team", and secondly, to review the new design against available experience. For example, if the item under consideration is a new compressor case design, then comparing items such as hoop stress, longitudinal stress, contact stress between components, stress concentrations, and deflections, especially in seal areas, for the new design against the range of existing design experience. Plots of where the new case design resides against the prior experience might reveal areas for further study when points fall near or beyond the edge of the prior experience.

The "Design Review Team" is invaluable in the validation of a new design validation. Such teams are best composed of experienced people from within and without the designer's organization who can be objective and provide constructive feedback to the designers. People from the end user community; such as operators, maintenance personnel, and lead rotating equipment engineers; can provide a wealth of input.

Another strategy to employ is to consider what is different about a new design. Also, assessing how the new design does its job better that the existing design is a clever tactic to highlight items for further investigation or validation.

Benchmarking a new design against what is done in an unrelated industries can also provide valuable insight and guidance as to whether a serial # 1 is safe to buy. For example, the automobile and aircraft industries provide a multitude of examples against which to check and compare.

Some tools that are needed for a thorough mechanical design review are:

1. Finite element analysis (FEA) of every component and of various assemblies of components. This might include 3-D FEA to determine stresses, deflections, and component natural frequencies. Elasto-plastic FEA's of components that are highly stressed, and dynamic FEA's to assess the fatigue loading are important for rotating components such as impellers. The dynamic pressures to be utilized in the dynamic FEA are usually generated by a transient 3-D CFD analysis or taken from dynamic pressure transducers during a test. A thermal analysis might also be warranted to determine the effect of operating temperatures on components fits and clearances.

2. Rotordynamic critical speed and stability analysis. This is critical to the proper functional operation of the compressor. If critical speeds with insufficient damping are present in operating ranges, high vibration and damage can occur. The requirement for critical speed separation margin is exacerbated in many of the design trade-offs encountered in the development of Serial #1 machines. Rotor dynamic instability is a devastating issue when experienced in a machine design. high pressures and high power levels have the ability to generated destabilizing mechanisms in machines. One of the significant problems encountered is that there are limits on the accuracy with which calculations may be executed in these new operating regimes. One of the conundrums encountered is that a good rotor dynamic design may be a poor capacity design or large footprint design, etc. These requirements must be accommodated through compromise and "smart" design.

3. Dynamic System Response Analysis. No compressor operates by itself. It is a part of a system that includes a driver, controls, various auxiliary support systems, and the end user's process plant. Simulating the start-up and shut-down of the compressor under normal, emergency, and off design operation is very important to meeting the reliability, safety, and functional performance aspects of the unit. This is typically done using very sophisticated

processing software that simulates the behavior of the various components in the process. The response of the system to different process upsets can be reviewed to ensure that the upsets to not result in dangerous operating conditions; i.e., rotor over-speed, high torque loads, etc.

4. Experience Plots

- Pressure vs. Flow
- Head per stage
- Delta-P per stage
- Case delta-P
- Impeller tip speed
- Sealing pressure
- Seal delta-P
- Power per stage
- Power per case
- Head per casing
- Impeller Flow Coefficient
- Etc,etc,etc.

The compressor casing is perhaps the most important of all compressor components as far as being consistently reliable in performing its function, namely, the safe containment of a flammable toxic gas under pressure. This record of high reliability can be attributed to the industry embracing the ASME Pressure Vessel Code and, in many instances, going beyond what the code requires due to other design considerations. These “other” design considerations can be:

- the desire to control or limit deflections of components,
- the need to establish a suitable value of support stiffness for rotordynamic considerations,
- the need to meet industry standards, such as the NACE standards regarding material compatibility with the gas being handled,
- the need to insure a 20 year life as required by API specs, and
- the need to include a corrosion allowance into the wall thickness as required by API.

Obviously then, one of the main items to check in the acceptance of any new casing design is compliance with appropriate industry standards. Other items to be checked are the casing material certifications which validate the appropriate material chemical composition, the correct hardness, the correct tensile strength, correct ductility, and the correct impact strength which is especially important for low temperature applications.

Another major item is non-destructive test results on the case material and especially on any welds that contain pressure. Many times the case design is such that the main process connections and many of the auxiliary connections are on the case. The ASME Code assigns a “joint efficiency” factor to any welded joints on pressure vessels that relates to the level of NDT performed on that weld. For example 100% radiographed joints receive a joint efficiency factor of 1.0. If spot radiography is performed, the joint efficiency factor maybe only 0.75, and if no radiography is performed the joint efficiency factor maybe just 0.50. This is important because these joint efficiency factors are utilized in a Code formula that determines the allowable maximum stress allowed for that joint.

The following sections describe examples of mechanical considerations in Buying or Selling Serial #1.

High-Pressure Sour Gas Injection Compressors

High-Pressure Sour Gas Injection is a relatively new application that has emerged due to the development of the oil fields in the Caspian Sea in Kazakhstan. These applications involve the compression of gas with a hydrogen sulfide (H_2S) composition of approximately 17% to 25% from relatively modest inlet pressures to a final discharge pressure of approximately 600 to 800 bar (8,000 psi to 11,000 psi).

When thinking about such applications to determine what is actually “new”, several items immediately come to mind. First and foremost is whether or not multistage centrifugal compressors have ever contained pressures of that magnitude. If not, then one needs to ask what the highest pressure that a particular multistage centrifugal compressor casings has contained, and also separately determine if any pressure vessels have ever contained such pressures. This will determine if a technology gaps exists for the entire industry or just for the manufacturer under consideration. Implicit in this thought about pressure containment is not just the casing proper, but also: (a) the sealing elements at any joint in the case, (b) any case penetrations and connections, and (c) the primary shaft seals. Each of these facets must be investigated and evaluated for pressure capability.

In trying to sell a case for such an application, items which are commonly shown to the purchaser can include the case FEA model, a diagram of the loads applied, pressure and deflection results, especially for high stress areas of interest such as around nozzle openings, and near any grooves or corners, the connection flange type and rating, the stationary sealing element design, and the primary shaft seal design and pressure rating for both dynamic and static sealing. Figure13 shows a model of a typical high pressure centrifugal casing. Such models are fundamental to the analysis of the stresses and deflections that the casing will experience due to normal operation and hydro-testing.

Determining which version of the ASME Pressure Vessel Code was utilized by the designer is also important. Division 1, Division 2, and Division 3 are all valid but have differing requirements and will results in diverse designs. The FEA deflection results for a typical case plus head (end closure) assembly model under certain loads are shown in Figure 14. The area of interest in this particular analysis was around two static o-ring seals to determine if the deflection would sufficient to all for potential extrusion of the polymer seal. A contact pressure analysis in this same area in shown in Figure 15 and indicates that positive contact remains under load. Such analysis work is useful in the evaluation of new designs as well as to determine if existing designs can be utilized for higher pressures.

This process is essentially somewhat like the proverbial “peeling the layers off the onion” until confirmation and validation of acceptability are achieved. If they are not, there is a strong case to achieve them by doing actual testing, which may include hydro-testing, strain gas testing, and even high pressure helium gas tightness testing. Helium is sometimes used for high pressure first article static pressure testing because it is inert, relatively safe, and the small size of the molecules tend to show any leak paths better than other inert gases or fluids. Performing pressure cycle tests is also important to assess the suitability of the case static seals relative to the effects of pressurization and depressurization cycles.

"Seal-less" Compressors

"Seal-less" compressors are gaining broader acceptance for severe toxic and harsh applications due to their inherent reduced potential for leaks. The term "Seal-less" might be considered a misnomer. Though the seal-less compressors have no shaft seals between the process gas and the atmosphere that would permit gas to escape to the atmosphere if a seal failure were to occur, they still have a static seal (o-ring, gasket, etc.) at every joint in the compressor and at every penetration through the pressure containing envelope. Such compressors usually employ magnetic bearings and are driven by a direct connected high-speed electric motor that is integrated with the compressor casing. Essentially all of the gas is contained by the casing and the connection flanges between the casing and the process gas piping. Since the electric drive motor and the magnetic bearings are internal to the pressure containing envelope, penetrations must be made through the pressure containing envelope for the electric power cables to the motor and magnetic bearings, the sensor cables for the magnetic bearings, and temperature sensors for the motor stator windings. One "Serial #1" item in these compressors compared to conventional compressors is the electric cable penetration. Such penetrators are made by a variety of companies but their application to multistage centrifugal compressors is relatively new. Care must be exercised to assure that the pressure, temperature, and sealing fluid experience with such connectors lies within the capability of the connector and that the connector is capable of meeting the pressure rating requirement of the compressor casing.

A typical "seal-less" compressor is shown in Figure 16 with the cable penetrators attached to the drive end of the casing and to the outboard end bearing housing. This particular connector carries the power for the magnetic bearings and the control cables for the magnetic bearing position and temperatures sensors. One of the pressure-containing control cable and sealing gland assemblies is shown in Figure 17 and one of the multi-pin connectors utilized for the control cables which are the atmospheric side of the cable penetrator is shown in Figure 18. The backside of the cable penetrator, the epoxy seal surrounding each individual cable connector on the pressure containing interior side of the connector and the o-ring which prevents gas leakage around the threads of the connector gland are pictured in Figure 19.

Also, magnetic bearings have a relatively limited load carrying capability compared to conventional oil lubricated journal and thrust bearings. Therefore, magnetic bearing load margins need to be scrutinized, as must the rotordynamic characteristics of the rotor on both the magnetic bearings and the conventional back-up (or drop-down) bearings; said back-up bearings might be of the rolling element or tilting pad type. Nevertheless magnetic bearings have been successfully applied in numerous multistage centrifugal compressors and their reliability has been proven when properly designed, manufactured, installed, and operated.

One final significant consideration is the cooling of the motor for these close coupled and single-enclosure systems. The motor must be cooled using some amount of process gas or a separate cooling gas supply system must be provided. In addition, care must be taken to ensure that no foreign materials or contaminants are allowed into the motor or the life of the motor could be compromised.

Very Large LNG Propane Compressor

In a very large LNG propane refrigeration compressor, the serial #1 item is most likely to be a new high flow coefficient impeller. Current world-class size LNG liquefaction trains utilize very large high horsepower industrial gas turbines driving very large multistage centrifugal compressors for the refrigeration process, with impellers up to nearly 2 meters in diameter and easily weighing over one ton. As mentioned above, the propane refrigeration application presents a challenge to the centrifugal compressor aerodynamic flow path designer. Due to the low inlet temperatures and the high molecular weight of the gas, the impeller inlet relative Mach number can be very high. The challenge to the mechanical designer is in gaining a thorough understanding of all the impeller natural frequencies, potential excitations, and stresses, both steady state and dynamic. These are all very important to ensure that the impeller will operate as specified over the life of the unit.

Typical results from an impeller natural frequency FEA analysis for four different natural frequency mode shapes are shown in Figures 20 through 23. It is important to plot these mode shapes on a Campbell Diagram of a "Safe Diagram" where these frequency can be studied for coincidence or nearness to any potential excitation within the compressor. Results of a principal stress analysis for a response at one of the natural frequency modes are given in Figure 24. This is an important step in the process of ensuring that the impeller design will have sufficient fatigue life. A full 360 degree FEA model of impeller displacements due to the 1D natural frequency mode is provided in Figure 25. Although it is much easier to visualize the results with a full 360 degree model, they take much longer to run, hence designers typically use "Pie Slice" models for analysis whenever possible to reduce time and cost. An impeller with a blade leading edge natural frequency is shown in Figure 26. This slide illustrates that not only the complete impeller structure has natural frequencies, but every component in the impeller as well. Such analyses are useful because such frequencies can many times be removed from the operating range by a change to the impeller or stationary designs, or can possibly be removed from the operating range if the excitation frequency can be changed.

The method of manufacture is also a concern for the mechanical designer. High flow coefficient impellers are characterized by relatively large blade heights. The three most commonly used methods for manufacturing such large impellers are:

- multi-axis milling a bladed disc from a solid steel forging and attaching a shroud to the blade tips by welding,

- single piece multi-axis machining with an integrally machined shroud, or
- investment casting.

All three manufacturing methods have advantages and disadvantages but all three have been used to produce high quality, high performance impellers.

The mechanical designer also needs to validate that the material used to manufacture the impeller has the appropriate properties for low temperature service, and needs to be confident that the material properties will be acceptable over the temperature range the impeller will experience. Also, features such as the method used to attach the impeller to the shaft need to be assessed for acceptability, especially if the weight or center-of-gravity of the new impeller has changed significantly from the manufacturer's experience.

500 to 800 bar CO2 Compressor with traces of water

The presence of water together with carbon dioxide is potentially detrimental due to the formation of carbonic acid, which is very corrosive to carbon steels. In the author's experience, it only took 3 or 4 hours of running a closed loop compressor test on CO2 with a leaking cooler tube to coat the entire inside surface of the test loop and compressor with a fine red oxide rust powder. A gas compressor in a process plant running 24/7 for a 3 to 5 year production run could experience significant corrosion and erosion during that time, which might or might not be detectable with standard monitoring instrumentation.

As a countermeasure, most mechanical designers utilize various stainless steel alloys whenever CO2 and water appear in the gas composition. However, if H2S is also present, sulfuric acid might also form, further complicating the material selection process. The NACE code provides good direction to follow in this regard. In some instances nickel alloy materials such as Inconel 718, 625, and 738 might be needed to meet the NACE requirements.

An application such as an 800 bar CO2 compressor also raises some other potential issues which might approach an actual technology gap. Questions must be asked, such as:

1. Are there any real gas property tables or equations for CO2 at 800 bar?
2. How does the high gas density influence the impeller stresses and natural frequencies?
3. How does it impact the rotordynamics?

As an example, typical P-V-T data showing how the gas compressibility changes with pressure are given in Figure 27 for a typical gas mixture. In the top half of figures 28 through 30, comparisons of how three different commercially available gas "Equations-of-State" calculate the compressibility for the same gas mixture are shown. The deviation of the calculated compressibility from the measured compressibility is shown in the plot at the bottom of these figures. It can be seen that none of the equations-of-state match the actual P-V-T data perfectly, but some have relatively small deviations in certain ranges of pressure. The importance of matching the actual gas behavior lies in the accuracy of the designer's ability to predict the head and power required by the compressor. For gas mixtures at very high pressures data often does not exist. Therefore the owner and the manufacturer need to agree on what real gas equations or tables best approximate the gas properties for a given application.

Subsea Compressor

Subsea compression is one of the newest emerging applications for multistage centrifugal compressors. Much has been written and presented about the world's first subsea compressor application for Statoil's Ormen Lange project in Norway and the extensive qualification program undertaken by Statoil and Aker Solutions. The "heart" of the system is a "seal-less compressor" as discussed earlier, but, also involved the qualification of marinization of the whole system, including: (a) the marinization of a subsea VFD converter for a 12.5 MW motor driver; (b) marinization of the subsea magnetic bearing control system; (c) the wet mate-able power connectors, (d) the marinization of the instrumentation and controls, (e) the control valves, (f) the power supply, (g) etc. The list goes on for too long to do justice to it here. Needless to say, it is a huge undertaking. Much new ground is being broken, much is being learned, and hopefully it will be a technical and economic success.

There are a few methodologies for qualifying such new technologies that involve the determination of technology gaps, technology readiness levels, and the research and development required to advance the readiness levels and close the gaps. Ultimately, full scale prototype testing under installed environmental conditions maybe the only acceptable means to reduce risk. A model of one of the proposed subsea compression modules for the Statoil Ormen Lange Subsea Compression Pilot Project, which may be the world's first, is shown in Figure 31. One of the pressure containing cable and sealing gland assemblies is illustrated in Figure 32.

CONCLUSIONS

This tutorial has presented the various considerations facing an end user and an OEM when faced with the opportunity to buy or sell SERIAL #1 of a new centrifugal compressor product or component. Material was presented regarding the approaches typically taken to validate both the mechanical and aerodynamic viability of the new design / components.

Of course, a very important factor that must be present in order to either sell or buy serial #1 of any new technology is a high level of trust between the end user and OEM. If the end user does not trust the information being provided by the OEM or, equally important, does not trust the people providing said information, the end user certainly will not trust any new product or service offered by said OEM. Similarly, if the OEM cannot trust the end user to maintain confidentiality on new technologies or methods divulged in attempting to justify the purchase of serial #1, the OEM will be reluctant to divulge the very data necessary to prove his/her case. However, if a high level of trust is present, both the end user and OEM can have confidence in the information being exchanged.

Similarly, there must be honest, open communications between the OEM and end user. Of course, such is necessary to establish and maintain trust! The OEM must make the OEM aware of any surprises or issues that arose during the development and/or testing of the new product or technology and provide details on efforts to resolve said issues. Similarly, the end user must keep the OEM informed on changes in requirements (i.e., operating conditions, gas properties, process conditions) so that changes can be implemented and the impact of said changes can be assessed as early as possible in the design cycle.

One of the most effective ways to ensure effective communications and, therefore, a high degree of trust is to conduct design review or design audit meetings. If the new technology will be developed jointly between the OEM and end user, such meetings should occur at regular intervals during the execution of the project. In the situation where an OEM is attempting to justify the purchase of serial #1 after the development is complete, one or more on-site visits by the end user to the OEM facility should be arranged so that key decision-makers from the end user organization can meet directly with those involved in the development of the new technology / product. Such face-to-face meetings can significantly improve the effectiveness of communications and promote trust between the organizations.

In closing, buying or selling Serial #1 of a new style of compressor or compressor component presents many challenges to the end user and OEM. However, as technology evolves, those end users and OEMs that wish to remain competitive will eventually face the need to justify the purchase of or the technology in a Serial #1 application. Hopefully, this tutorial has provided both with a meaningful reference work that can be used when faced with such circumstances.

DISCLAIMER

The information contained in this document consists of factual data, and technical interpretations and opinions which, while believed to be accurate, are offered solely for informational purposes. No representation or warranty is made concerning the accuracy of such data, interpretations and opinions.

AERO APPENDIX

The following section provides additional material on the methods used to justify or validate the new aerodynamic components used in a SERIAL #1 application. The majority of this material is drawing from previous publications authored or co-authored by James M. Sorokes. Said publications can be found in the reference list at the end of this document.

Experience Data

Of course, one of the best ways to demonstrate experience on new designs is to relate them to prior experience. Therefore, the OEM will frequently provide charts or tables that show how the new designs relate to previously used / field proven equipment. The OEM will explain how the new design is either an interpolation, an extrapolation or the logical extension of a previously proven product.

The most common types of charts or tables include common parameters used to categorize or group centrifugal machinery, such as:

- Machine Mach Number, U2/A0 versus flow coefficient
- Shroud Inlet Relative Mach Number, MREL1T, versus flow coefficient
- Efficiency, η, versus flow coefficient
- Head Coefficient, μ, versus flow coefficient

Typical Parameters Defined -- Nomenclature

The following discussion is included to provide the reader with a better understanding of the typical aerodynamic parameters used to quantify experience.

Flow Coefficient

Flow coefficient is a parameter used by both OEM's and end users to indicate the flow carrying capacity of a compressor stage or impeller. Designers typically identify an impeller's design flow coefficient as the flow rate at which the impeller provide peak efficiency (i.e., work output / work input is maximized). Numerous flow coefficients are used in the turbomachinery industry but the two most common are:

- The ratio of the flow rate (Q) in actual cubic feet per minute (or cubic meters per minute) divided by the operating speed of the compressor (N) in rotations per minute (or rpm). Q/N is clearly dimensional parameter in that the units do not cancel, leaving the rather interesting relationship of Actual cubic feet per rotation as the "per minute" cancel. As a result, the term Q/N does change during the scaling process (more on this later).

- A relationship of the flow rate (Q) in actual cubic feet per minute (or cubic meters per minute) divided by the operating speed (N) and the impeller diameter cubed or Q/ND^3. When the proper units conversions are applied, the Q/ND^3 relationship does become dimensionless, such that it is not affected by the scaling process. The most common form is the so-called ϕ coefficient which in U.S. customary units is as follows:

$$\phi = 700.28 \frac{Q}{ND^3} \qquad (1)$$

Where:
Q = actual cubic feet per minute
N = compressor speed in rotations per minute (rpm)
D = impeller diameter in inches

Flow coefficient plays an important role in establishing the maximum swallowing capacity that can be achieved for an OEM's compressor. For many years, it was commonly held that the maximum flow coefficient that could be applied in a centrifugal compressor was 0.15. It was felt that any impeller with a flow coefficient greater than 0.15 would have to be of a mixed flow configuration; that is, the flow at the exit of the impeller is not directed radially outward but rather exits somewhere between a radial and axial direction (see Figure 33). Clearly, such stages would require greater stage spacing to accommodate the non-radial orientation of the stationary components. However, in recent years, some OEMs have begun to apply stages with flow coefficients greater than 0.l5 and stages with higher flow coefficients are appearing in some OEM's product offerings. For example, Dresser-Rand currently offers a flow coefficient of 0.17 as part of their standard radial product offering in beam-style machines. In axial inlet pipeline or integrally-geared machines, mixed flow impellers with flow coefficients as high as 0.24 or higher are common.

Regardless, if one knows:

a. the maximum flow coefficient allowable for a given OEM, ϕ_{MAX}
b. the nominal diameter of the impellers in the OEM's products, $D_{2\,Nom}$, and
c. the allowable operating speed by compressor frame size, N,

one can reasonably estimate the maximum allowable capacity for a given compressor frame size simply by solving the following relationship:

$$Q_{MAX} = \frac{\phi_{MAX}\, N\, D_{2\,Nom}{}^{3}}{700.28} \qquad (2)$$

For example, should the OEM have a product with 66" diameter impellers that can operate at 3600 rpm and should the maximum flow coefficient be 0.17, one could determine that the maximum frame capacity would be approximately 250,000 ACFM in a single inlet casing and 500,000 ACFM for a double-flow casing. If the OEM were to develop a 0.2 flow coefficient impeller, the maximum capacity in a single inlet casing would increase to nearly 300,000 ACFM.

The Mach Number Conundrum

One potential source of confusion in communications between OEMs and end users lies in the various Mach number that can be provided for centrifugal equipment. It is very important that there is a common understanding regarding which Mach numbers are requested and/or provided during the discussions of ANY turbomachinery application. The following discussions are provided to help with this understanding.

Machine Mach Number, U_2/A_0

The machine Mach number, also called the tip Mach number relates the impeller's physical tangential velocity to the sonic velocity of the gas at the inlet of a given impeller. The parameter is defined as U_2/A_0 where:

$$U_2 = \frac{N\,\pi\,D_2}{720} \qquad (3)$$

Where: N = operating speed (in RPM)
D_2 = impeller exit diameter (in inches)

$$A_0 = \sqrt{kzRTg_c} \qquad (4)$$

Where: k = ratio of specific heats
z = gas compressibility
R = gas constant (1545/mole weight)
T = inlet temperature in °R
g_c = gravitational constant

Please note that the machine Mach number is NOT based on the velocity of the gas exiting the impeller and must not be confused with the shroud inlet relative Mach number discussed below. Rather, machine Mach number gives the designer and the end user an indication of the overall flow range one might expect for an impeller. Typically, as the machine Mach number gets higher, the overall flow range gets narrower; i.e., less flow range from the choke point to minimum stable flow or surge. Therefore, the parameter can be of value when comparing an OEM's prior experience to the proposed compressor being selected. The greater the OEM experience with high U_2/A_0 applications, the greater is the knowledge that the OEM will have in developing the designs necessary to handle those applications.

Inlet Relative Mach Number

Inlet relative Mach number or more specifically shroud inlet relative Mach number is the highest Mach number and/or highest velocity within a centrifugal stage. It occurs along the cover or shroud of the impeller at or very near the blade leading edge. Unlike machine Mach number, shroud inlet relative Mach number is a gas velocity. Like machine Mach number, it gives the designer and end user insight into the flow range of an impeller, especially toward the overload or choke side of the flow map.

The end user is wise to request experience (or "cloud") plots showing the OEM's experience with high inlet relative Mach number stages. Said experience plots can demonstrate that the OEM has had sufficient experience applying such stages and, again, this experience demonstrates that the OEM has the technical expertise to satisfactorily develop these demanding stages. However, it is vitally important that the end user understand the methods used by the OEM to calculate the inlet relative Mach numbers. It is very possible to get different values dependent on the method and rigor used.

In general, three methods are used to calculate shroud inlet relative Mach number. The most common approach is to use a bulk flow or 1-D analysis code. Such codes perform velocity calculations along the impeller mean flow path based on flow-through-area and conservation of angular momentum. The codes then apply geometric correction factors or potential ratios to account for the local curvature and estimate the velocity and relative Mach number along the cover. The calculation procedure is quite straightforward but does allow for "tweaking" of the shroud inlet relative Mach numbers via the geometric correction factors or potential ratios. For that reason, Mach numbers should be treated as approximations rather than supremely accurate values. Also, when comparing Mach numbers calculated by different vendors, it is imperative that the assumptions and/or "adjustment factors" used by each are considered. Should two OEMs use different methods to estimate the "adjustment factors", it would be possible for the two OEMs to calculate two different Mach numbers for the same impeller geometry.

The other two methods, 2-D and 3-D analyses, eliminate the need for potential ratios or geometric correction factors because the code requires the user to input the full geometry of the impeller. The 2-D and 3-D codes calculate the velocities directly from the more detailed geometric model; i.e., they are not dependent on correction factors. The 2-D codes account for the curvature of the impeller shroud and blading. As such, the shroud inlet Mach numbers from a 2-D code tend to be more accurate than those from a 1-D calculation. The 3-D codes further enhance the accuracy by accounting for all curvature in the impeller inlet region including the shape of the blade leading edge. Consequently, the 3-D code also tends to yield the highest calculated Mach numbers; i.e., the more curvature that is taken into account, the higher the velocities.

It is possible to calculate inlet relative Mach numbers using 1-D, 2-D, or 3-D techniques. However, in the final analysis, when comparing Mach numbers received from various OEMs, one must inquire as to the methods used and as to what assumptions, modeling techniques, etc., were used in the analyses. Without insuring that a common approach was used, there can be no assurances that the comparison is valid. For a detailed discussion on the various methods, the reader is directed to Sorokes et al (2007].

It is important to note that the scaling process (to be address later herein) does not impact inlet relative Mach number and machine Mach number. Therefore, if an OEM has experience for a given stage in any compressor frame size, it can be used to demonstrate experience for any other compressor using the same stage operating at the same Mach number, regardless of size. This might be difficult for some to accept but were one to delve into the details of the aerodynamics; one would see that this is indeed the case. For that reason, OEM's regularly provide experience plots that include references from many different compressor sizes.

Efficiency

The most common efficiency term used by compressor manufacturers and/or users is polytropic efficiency. The equation is given below:

$$\eta_p = \left(\frac{k-1}{k}\right)\left(\frac{\ln(Pr)}{\ln(Tr)}\right) \qquad (5)$$

where: k = ratio of specific heats
Pr = pressure ratio
Tr = temperature ratio

Note that Equation (2) is only valid for a thermally perfect gas. Determination of polytropic efficiency for a real gas is a far more complicated effort.

Another popular expression for efficiency is the isentropic form which relates the isentropic enthalpy rise to the actual enthalpy rise (i.e., h_{is}/h_{act}). The perfect gas form of the equation is given below:

$$\eta_I = \frac{Pr^{\frac{k-1}{k}} - 1}{Tr - 1} \qquad (6)$$

Head Coefficient

The pressure generating ability of a compressor stage or section is typically expressed as pressure ratio or head rise. Pressure ratio is intuitively obvious and the equations for head and head coefficient, μ_P, are below:

$$Head_P = \frac{\eta_P}{g_c}\left(U_2 C_{U2} - U_1 C_{U1}\right) \qquad (7)$$

$$Head_P = \frac{\mu_P U_2^2}{g_c} \qquad (8)$$

$$\mu_p = \frac{Head_P \bullet g_c}{U_2^2} \qquad (9)$$

Where: η_p = polytropic efficiency
g_c = gravitational constant
C_{U1} = tangential velocity of gas entering impeller in feet per second
U_1 = peripheral velocity of impeller leading edge = $N\pi D_1/720$ in feet per second
C_{U2} = tangential velocity of gas exiting impeller in feet per second
U_2 = peripheral velocity of impeller trailing edge = $N\pi D_2/720$ in feet per second
D1 = impeller blade inlet diameter in inches
D2 = impeller blade exit diameter in inches
N = rotational speed in rotations per minute

To calculate the overall head generating capability of a compressor or compressor section, one must sum up the head generated by the each individual stage within the section or machine.

The head coefficient can also provide some insight into the flow range of a stage or, more importantly, the rise-to-surge. In general, as the head coefficient increases, the rise-to-surge will decrease and as the head coefficient decreases, the rise-to-surge will increase. This is illustrated in Sorokes (2003) and is replicated here for completeness.

To understand the parameters that influence head rise, consider the diagrams provided in Figure 34. A generic impeller exit velocity diagram is given in Figure34A with the critical velocity components and angles labeled. For the purist, these diagrams ignore the influence of slip or exit deviation. The following discussion also assumes a radial inlet guide upstream of the impeller.

Important variables to note are:

- The impeller exit flow tangential velocity, C_{U2}, and the impeller exit peripheral velocity, U_2. These two parameters are used along with the impeller efficiency, ηI, and gravitational constant, g_c, to calculate the head rise in the impeller. The equation for the typical case of an impeller preceded by a non-prewhirl guidevane is as follows:

$$Head = \frac{\eta_I}{g_c}\left(U_2 C_{U2}\right) \qquad (10)$$

- The impeller exit flow meridional velocity, C_{M2}, is a function of the impeller exit area, A2, in square inches and exit flow rate, Q2 in ACFM. This velocity can be estimate using the incompressible relationship:

$$C_{M2} = 2.4\ Q2/A2 \qquad (11)$$

- The flow angles β2 and α2 represent the relative and absolute exit flow angles, respectively. Since slip or deviation is neglected, β2 also is the impeller exit blade angle. The two exit velocity diagrams in Figures 34B and 34C represent impellers with 40° of backsweep (high head) and 60° of backsweep (low head), respectively. The black lines on each plot provide the velocities for the design flow condition. The red lines reflect operation at 110% of design flow while the blue lines reflect operation at 90% of design.

First, note the relative lengths of the C_{U2} vectors on the high and low head velocity triangles. The low head impeller generates less C_{U2} and, therefore, less head. Now note the change in the C_{U2} velocities between the high and low head velocity triangles for ±10% flow from design. Clearly, there is more change in C_{U2} for the low head. Therefore, there will be a greater head rise on the low head (or high backsweep) impeller. The result will be more useable flow range for the higher backsweep impeller.

The impeller head coefficient also plays an important role in determining the attainable head level of centrifugal stages. Obviously, the higher the head coefficient, the higher will be the attainable head level – see Equation 5 above. Therefore, as long as an impeller is capable of running at a very high tip speed, it is possible to generate very high head levels. For example, if the impeller having a head coefficient of 0.55 is able to operate at 1100 feet per second, that impeller will generate over 20,000 feet of head. If the same impeller is capable of running at 2200 tip speed, the attainable head level will be over 80,000 feet of head.

The ability to run at high tip speeds is limited by both mechanical and aerodynamic factors. The primary mechanical factor is the impeller stress levels. For example, it is commonly known that covered / shrouded impellers cannot run at as high a tip speed as open / unshrouded impellers because the cover causes a significant increase in the blade stresses, etc. The primary aerodynamic limits are Mach number based; i.e., inlet relative Mach number and Machine Mach number, U2/A0 (described above). In many cases, because of the gas properties or operating conditions, it is not possible to run an impeller at its mechanical tip speed limit so this limits the attainable head level for some applications. For example, assume a heavy mole weight gas has a sonic velocity of 700 feet per second. Assume further that an OEM has established a Machine Mach number limit of 1.2 to ensure that reasonable flow range is attained for a family of impellers having a 0.50 head coefficient. These stages will only be able to run at 840 tip speed, even if they are capable of running at higher speeds mechanically.

It is important to point out that all of the parameters described above are used to describe individual stage characteristics as well as overall compressor or compressor section performance.

Should this information be deemed insufficient by the end user, the OEM might also provide experience plots showing compressors with similar staging. These plots typically relate the impeller or stage design flow coefficient to parameters such as machine Mach number (U_2/A_0), impeller inlet relative Mach number (M_{rel1T}), molecular weight handled, head coefficient or head level per stage or other parameters of interest to the end user. These plots, often called “cloud diagrams” due to the large number of data points (or “cloud of data”) shown, illustrate the OEM’s experience over their full range of products. The intent of such “cloud diagrams” is to demonstrate that the centrifugal stages required for the end user application falls within the OEM’s design experience.

In addition to the “cloud diagrams” or tables summarizing prior experience, OEMs might provide plots of actual test data for compressor sections or individual stages showing how similar stages have performed in the past. These figures typically plot efficiency, head coefficient, and/or pressure ratio predicted versus actual test data. Such plots normally will not be given to the end user because they contain data gathered from a compressor for another end user or data that is considered proprietary by the OEM. Such curves will be shown during design audits or other such meetings but copies will not be distributed.

Related Experience

Quite often, the OEM might not have experience in a particular size of machine and will rely on the fundamental concept of scaling (or geometric similitude) and contour trimming to augment the data in the experience plots.

Geometric Scaling

Geometric scaling is used by nearly all turbomachinery OEMs. The principle of geometric scaling in turbomachinery is much akin to that of model railroads, model cars and/or model airplanes. A compressor or compressor component of a given size is geometrically scaled by some factor to make a smaller or larger version of that compressor or compressor component. The scale factor is typically established based the maximum (or nominal) exit diameter for the progenitor or parent impeller. By applying scale factor to the parent design, a new higher or lower diameter impeller can be created. Those interested in a more detailed discussion on geometric similitude are directed to Principles of Turbomachinery by D. G. Shepherd (1996)

The scale factor is applied as shown in table 1. Please note that the operating speed of the scaled impeller changes by the inverse of the scale factor. In that way, the machine Mach number or U2/A0 remains constant during the scaling process. That is, the diameter changes by the scale factor, the speed changes by the inverse of the scale factor. Therefore, the scale factor cancels and the parameter remains fixed between the original and scaled impeller.

It is critically important to note that angles do not scale. That means that the blade angle distribution from leading to trailing edge of an impeller or diffuser or other bladed/vaned component remains constant through scaling. Just as the contours of the scaled model car do not change from full size to ¼ scale, the aerodynamic component contours do not change.

An illustration of the result of geometric scaling can be seen in Figure 35. Note also the impact of scaling (or lack thereof) on the various parameters listed on the right of the figure.

The fundamental methodology behind scaling is to cause the velocities in the scaled impeller or other flow component to be the same as in the progenitor (or parent). Therefore, having experienced in a scaled version of any given impeller, diffuser, return channel, etc. means that one has experience at those velocities or Mach numbers. This is a critically important fact when considering scale model testing (addressed later).

Of course, there are "real world" limitations to the scaling process in that some dimensions do not scale; such as weld fillet sizes, minimum allowable blade thickness for various joining techniques, minimum wall thickness in stationary components, surface finish, corner radii for milling cutters, etc.

Contour Trimming / Contour Extensions

The basic principle of contour trimming (or contour extensions) is to reduce (or increase) the flow passage area to cause a stage or stage component to operate effectively at a lower (or higher) capacity. Contour trimming is typically used to fill out an impeller family after a progenitor or parent impeller has been designed. Using this approach, a series of impellers can be developed in a relatively short cycle time, with the bulk of the design time absorbed in the development of the progenitor stage.

The concept of contour trimming is illustrated in Figure 36. The approach used in this example was to reduce the area in the impeller passage by a constant percentage. Adjustments were made to the blade or passage height from leading edge to trailing edge so that the passage area of the contour trim was 80% of the area of the progenitor impeller. Therefore, the design flow (or optimal flow) of the contour trim impeller would be 80% of the design capacity of the progenitor. The meridional (or through flow) velocity is directly proportional to flow through area; i.e. $\vec{V} = f(Q, A)$. Because the area and flow were both reduced by 20%, the velocity levels in the contour trimmed impeller will be nearly the same as the velocities in the progenitor, though not identical.

The contour trimmed impeller will have different velocity levels along the cover due to the changes in curvature of the contour trimmed cover as well as the changes in leading edge angle along the shroud. However, these effects tend to be minor so long as the contour trim or extension is 40% of less. Further, the curvature of the trimmed impeller typically reduces and the shroud path length tends to increase, leading to a more optimal shroud profile and a reduction in the shroud relative Mach number at the leading edge (assuming incidence levels are acceptable).

It is likewise possible to do a contour extension; i.e., increase the flow passage by a certain percentage to increase the capacity of the impeller or other component. However, greater care is required for contour extensions, especially for impellers, because: (a) this will result in an increase in the shroud inlet relative Mach number; (b) the curvature along the cover will increase; (c) the resulting change in shroud inlet angle can result in adverse incidence effects; (d) etc. As a general rule, if the required increase in flow exceeds 10%, it would be more prudent to develop a new impeller design.

Because the hub contour and the blading are the same between different contour trims of an impeller, it is reasonable to expect very similar performance characteristics between the various members of an impeller family generated via contour trimming. Again, if an OEM can provide references for impellers that are of the same family, this typically is acceptable to demonstrate experience for that impeller.

Analytical Validation – The Virtual Test Rig

CFD software was developed to assist in the solution of complex turbomachinery flow fields. Initially applied only by high end gas turbine or jet engine designers, these sophisticated tools eventually found their way to engineers at industrial turbomachinery manufacturers. However, it has only been in the last 15 to 20 years that industrial users have begun to make more widespread use of CFD. There are a variety of reasons for this slow adoption.

Most early codes were limited in their applicability in that they were only two-dimensional, or in three dimensions capable of analyzing a single bladed passage or one bladed element. Analyses could include stationary or rotating elements but not both. The loss and/or turbulence models built into the codes were inadequate to properly model the complex flows encountered in centrifugal compressors. The codes were limited to relatively simple geometric and topological (mesh) configurations. Finally, solutions provided were steady state or time averaged, and unsteady effects were not usually considered.

Beyond the limitations of the codes themselves, there were several resource issues that also hindered the use of CFD. First, the time required to prepare input for CFD runs was prohibitive. An analyst could spend several weeks setting up a single run. Second, once the input was prepared, it might take a week or more for the analysis to execute on the available computer resource. Third, after the computer finished the calculation, several more weeks were spent interpreting the results. In short, a single solution could take in excess of a month. Clearly, this was not the type of analysis that could be completed often or as part of a day-to-day design cycle. Unlike the aircraft industry where it may take a year or more to develop a new aero flow path, the industrial compressor designer often must engineer a solution in two or three days. Fortunately, large increases in available, economic computing power together with development of improved computational methods now provides the industrial designer with much improved analytic capability.

Past Practice

Many industrial turbomachinery vendors began applying CFD as part of their design cycle as early as 1988. Typically, such analyses focused on individual components in isolation. Sorokes (1993) provided a general overview of how CFD was used in the design of industrial centrifugal impellers. Similarly, Dalbert and Ruetti (1993) described how CFD was being applied in the analysis of impellers and diffusers. Casey (1994) provided a detailed review on the application of CFD to a wide range of industrial turbomachinery, including radial and axial compressors. In this work, he provided very helpful guidance in the proper use of CFD, in the form of "CFD Aerodynamic Design Rules". Other works, such as those of

Amineni et al (1995) and Camatti et al (1995), concentrated on the analysis of various diffuser types while Japikse et al (1996) and others performed extensive analyses on various return channel configurations. Still others {i.e., Koch et al Koch et al (1995) and Hillewaert et al (1998)} addressed CFD as applied to inlet or discharge systems.

These isolated component analyses provided a major step forward from streamline curvature and other 2-D analyses. It was now possible to visualize in three dimensions the flowfield within compressor passages so that qualitative assessments could be made of component designs. Flow anomalies that contribute to poor performance could be identified and alternate configurations pursued to eliminate said shortcomings. While the quantitative results from individual CFD analyses were not considered overly accurate, the relative changes or performance trends estimated by such codes were believable. In short, the designer now had what amounted to a "virtual test rig" for judging the advantages / disadvantages of alternate configurations or for identifying shortcomings in existing designs.

Since conducted on the computer, a virtual test does not require the "cutting of metal." Components are created as computational grids rather than physical pieces and the test bench is the computer rather than a test facility. Virtual test rigs will never totally replace "real world" test vehicles. Still, the analyst can "virtually test" many different configurations and choose the best alternative before proceeding with physical rig testing.

The more complex the components, greater are the savings that are gained through virtual versus physical testing. For example, compressor inlets, discharge volutes, and sidestreams (re-entries) can be very difficult to manufacture and often are castings or complicated fabrications. Clearly, in these cases, physical rig testing must be kept to a minimum and use of a virtual test rig is imperative.

CFD Developments

Industrial designers and analysts require CFD software to provide accurate solutions to their applications of interest in a cost-effective and timely manner. With the widespread availability of affordable workstation class computing, the main issues are therefore accuracy and productivity (productivity also being directly related to cost). The information provided by the CFD must be of sufficient accuracy to allow the designer to make appropriate decisions based on the available information, and it must be available rapidly enough to fit within the time scales of the design cycle.

For the industrial designer, “accuracy” means providing reliable information of the following type:

1) Qualitative: correctly reproduces the important flow features, such as swirl, boundary layers, shocks, wakes, separation zones, stagnation points, mixing layers etc.
2) Quantitative: efficiency, work input, pressure rise, hub-to-shroud profiles, component loss coefficients, flow distortion parameters, incidence, deviation or slip etc.

While it is desirable that such features are perfectly reproduced, this expectation is unrealistic given limitations due to mesh size, turbulence modeling and other modeling assumptions such as steady state flow. To be used confidently, it is therefore of importance that CFD provide:

1) “Sufficient” level of accuracy
2) Repeatability and consistency.

The two factors that influence accuracy most for turbomachinery applications are the discretization accuracy of the flow solver and the computational mesh. Only after these two are adequately treated do other issues such as the turbulence model become of primary significance. In particular, the turbulence model is the usual scapegoat for poor CFD predictions, while in reality other causes are often more significant. Hence it is often the case that “numerical errors” exceed “model errors”.

Use of CFD to enhance or improve bulk flow (or 1-D) models has become commonplace amongst suppliers of industrial turbomachinery. Again, due to the high cost of rig or model testing, CFD offers a more economical means for developing performance or loss models for new configurations. Clearly, the models must be tuned to reflect empirical data. Still, CFD can be used to generate and assess basic models and significantly reduce the matrices of tests that must be run to validate such.

To make the most effective use of CFD, it is important to calibrate the computational models against sound test data. An example of this type of exercise is presented in Sorokes et al [3, 4]. As described in these works, the authors’ companies undertook an extensive test program on typical sidestream configurations and augmented the testing by conducting CFD analyses on the test configurations. The analytical and test results were then compared to improve the OEM’s knowledge in the use of CFD to design sidestreams and predict their performance.

Full CFD analyses should be run with the upstream and downstream components included in the analysis; i.e., the upstream IGV or axial inlet and the downstream diffuser or volute. This ensures that the proper velocity, pressure, and temperature profiles are present at the impeller inlet and exit. It is possible to get meaningful results with the so-called “impeller only” analysis but even these solutions typically include a portion of the upstream and downstream flow passages. Said portions are necessary simply to generate the computational mesh.

Regardless of the CFD code used, the following are some guidelines to assist in the assessment process. Again, these are only suggestions and the designer should develop his or her own criteria based on experience; i.e., results that yield good performance versus those that yield unacceptable performance such as rotating stall, premature choke, low efficiency and the like.

Flow profiles must be reviewed along the following planes:

- Near pressure surface
- Near suction surface
- Near shroud
- Near hub
- Mid-pitch (i.e., midway between two adjacent blades)
- Mid-span (i.e., midway between hub and shroud)
- Impeller exit plane

Flow profiles are typically assessed using:

- Velocity vectors colored by relative Mach number
- Static pressure contour plots
- Entropy generation
- Relative Mach number contours
- Efficiency contours

When assessing impeller flow profiles, adverse qualitative features to note include:

1. High relative Mach number or shock formation near the impeller leading edge. This can be indicative of high losses or limited flow range, especially if the high Mach number region occupies a large percentage of the flow passage.
2. Any regions of low momentum or flow reversal. These might indicate stalled fluid or an increased potential for premature stall, low flow range, or high losses.
3. Large variations in suction surface versus pressure surface velocities or pressures. This is also indicative of stall potential or low flow range.
4. Large variation in velocity or static pressure at impeller exit. This almost certainly suggests possible diffuser performance problems, premature stall, etc.
5. A large wake or low momentum region at exit. This is particularly bothersome if the wake region is passage-centered as opposed to the more classic location near shroud and suction surface. A large wake or low momentum region will cause poor stall / surge margin, low rise-to-surge, and poor diffuser performance.

REFERENCES / BIBLIOGRAPHY

Aalburg, C., Simpson, A., Schmitz, M., Michelassi, V., Bacciottini, C., "Design and Testing of Multistage Centrifugal Compressors with Small Diffusion Ratios", GT2008-51263.

Amineni, N., Engeda, A, Hohlweg, W.C., and Boal, C.F., 1995, "Flow Phenomena in Low Solidity Vane Diffusers of an Air Packaging Compressor," ASME Paper No. 95-WA/PID-1.

Bibet, Pierre-Jean, Quoix, Bernard, and Grimstad, Haakon, 2009, "Hybrid Pump-A New Type of Pump for the Pazflor Deep Sea Project", Proceeding of the Twenty Fifth Pump Users Symposium.

Brenne, L.,Bjorge, T.,Gilarranz, J.,Koch, J., Miller, H., 2005, "Performance Evaluation of a Centrifugal Compressor Operating Under Wet-Gas Conditions', Proceedings of the 34th Turbomachinery Symposium, pp.111-120, Houston, USA

Camatti, M., Betti, D., and Giachi, M., 1995, "Vaned Diffusers Development Using Numerical and Experimental Techniques," ASME Paper No. 95-WA/PID-4.

Casey, M.V., 1994, "The Industrial Use of CFD in the Design of Turbomachinery", AGARD Lecture Series 195, Turbomachinery Design Using CFD.

Dalbert, P. and Ruetti, K., 1993, "Application of three dimensional viscous flow calculations in the aerodynamic design of industrial turbocompressor components," IMechE Paper No. C449/024/93.

Gould, B., and Loaiza, C., "Seabed Production Boosting Systems Push the Limits", www.EP.com, July 2010.

Gupta, Shailendra K., Boomer, David L., and Fulton, Lynn, "FCC Hot Gas Expander Reliability Upgrade Evaluation", Proceedings of the 29th Turbomachinery Symposium, 2000.

Hillewaert, K., and Van den Braembussche, R.A., 1998, "Numerical Simulation of Impeller-Volute Interaction in Centrifugal Compressors," ASME Paper No. 98-GT-244.

Hopper, B.L., Baldassarre, L., Detiveaux, I., Fulton, J.W., Rasmussen, P.C., Tesei, A., Demetriou, J., and Mishael, S., 2008, "World's First 10,000 Psi sour Gas Injection Compressor", Proceedings of the Thirty-Seventh Turbomachinery Symposium.

Japikse, D., Wight, S., Yoshinaka, T., and Karon, D., 1996, Final Report For The Return Channel Consortium, Concepts ETI, Inc. (Proprietary)

Koch, J.M., Chow, P.N., Hutchinson, B.R., and Elias, S.R., 1995, "Experimental and Computational Study of a Radial Compressor Inlet," ASME Paper No. 95-GT-82.

Mankins, J.C., 1995, "Technology Readiness Levels," Office of Space Access and Technology, NASA

Miller, H.F., 1998, "Benefiting from Efficiency Improvements to Gas Compression," Dubai Energy Technology Conference – Technical Paper, Dubai, UAE.

Moses, K.D., Malone, R.W., 2005, "Development of Risk Assessment Matrix for NASA Engineering and Safety Center," NASA Marshall Space Flight Center

O'Neill, J., Thakur, N., Duus, A., 2007, "Technical Risk Assessment: a Practitioner's Guide," Defence Science and Technology Organisation, Australia

Shepherd, D. G., Principles of Turbomachinery, MacMillan Publishing Co., 1956

Simpson, A., Aalburg, C., Schmitz, M., Pannekeet, R., Larisch, F., and Michelassi, V., 2008, „Design, validation and application of a radial cascade for centrifugal compressors," ASME Paper No. GT2008-51262.

Sorokes, J.M., "The Practical Application of CFD in the Design of Industrial Centrifugal Impellers," Turbomachinery Symposium Proceedings, Texas A&M, pp 113 - 124 (1993).

Sorokes, J.M., "Industrial Centrifugal Compressors -- Design Considerations," ASME paper no. 95-WA/PID-2 (1995).

Sorokes, J.M., Nye, D.A., D'Orsi, N., and Broberg, R., "Sidestream Optimization Through the Use of Computational Fluid Dynamics and Model Testing," *Turbomachinery Symposium Proceedings*, Texas A&M (2000)

Sorokes, J.M. and Hutchinson, B.R., "The Practical Application of CFD in the Design of Industrial Centrifugal Compressors," Challenges and Goals in Industrial and Pipeline Compressors, PID-Vol. 5, pp 47 – 54, 2000

Sorokes, J.M., "Range Versus Efficiency – A Dilemma for Compressor Designers and Users," ASME paper no. IMECE2003-55223, 2003

Sorokes, J.M., Kopko, J.A., Koch, J.M., "Sidestream optimization for large compressor applications," *Ninth European Fluid Machinery Congress*, pp. 25-38, 2006

Sorokes, J.M., Kopko, J.A., "High Inlet Relative Mach Number Centrifugal Compressor Impeller Design," ASME paper no. GT2007-27864, 2007

Sorokes, J. M., Kuzdzal, M. J., "Centrifugal Compressor Evolution," Turbomachinery Symposium Proceedings, Texas A&M (2010)
Vannini, G., Canmatti, M., Masala, A., Svetti, F., Neri, M.O., Bondi, S., and Evangelisti, S., 2011, Full Loop Testing of a 12-MW Vertical High Speed Subsea Motorcompressor", Proceedings of the Fortieth Turbomachinery Symposium.
Waterfield, T., Looper, L., and McDonald, C., "The World's Highest Pressure Injection Pump", Sulzer WEB site publication (www.sulzerpump.com), April 2002.

ACKNOWLEDGEMENTS

The authors thank all of the pioneers through the year who advanced the state-of-the-art by being willingly buying or selling Serial #1. The authors also thank all (too numerous to list herein) who contributed to this tutorial. Finally, Mr. Miller and Mr. Sorokes thank Dresser-Rand for allowing us to present this tutorial.

TABLES

Table 1: Rules for Geometric Scaling

Dimension	**Multiply by:**
Linear (i.e., diameters, widths, heights)	Scale Factor
Area (i.e., throat areas, inlet areas, channel areas)	Scale Factor Squared
Volume (i.e., passage volume)	Scale Factor Cubed
Operating Speed	Inverse of Scale Factor
Angles (i.e., blade angles, vane angles, slope angles)	DO NOT SCALE

FIGURES

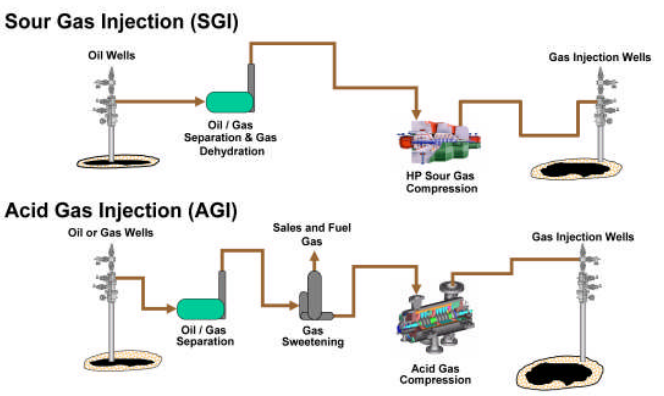

Figure 1. Configuration of Sour Gas Injection System and Acid Gas Injection for Tengiz

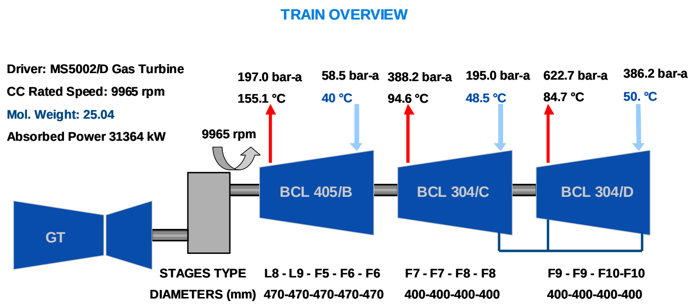

Figure 2. Tengiz Injection Compressor Overview

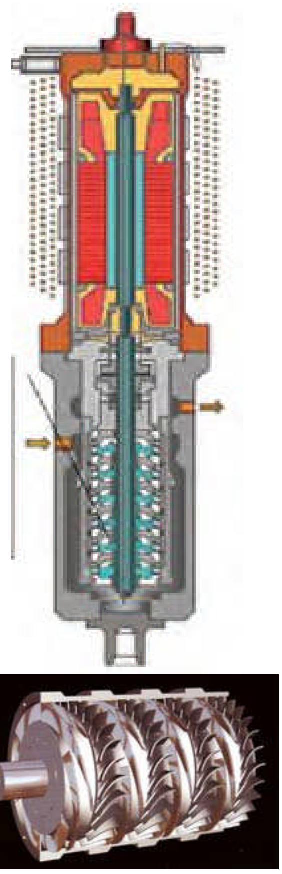

Figure 3. View of a typical Framo helico-axial pump (entire pump and section of impeller

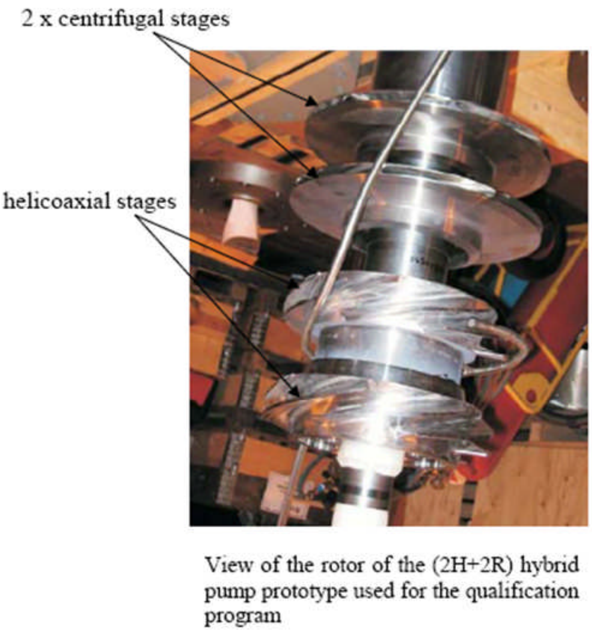

Figure 4. View of the "Hybrid" composition of the Pazflor pump rotor (two helicoaxial stages followed by two radial stages).

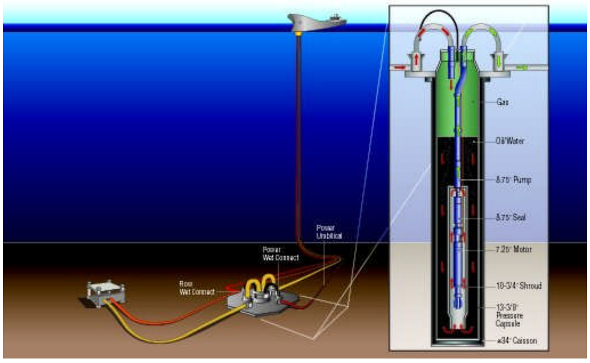

Figure 5. Baker Hughes Centrilift Pump as Installed in Perdido Field (Courtesy of Centrilift, Baker Hughes, Inc.)

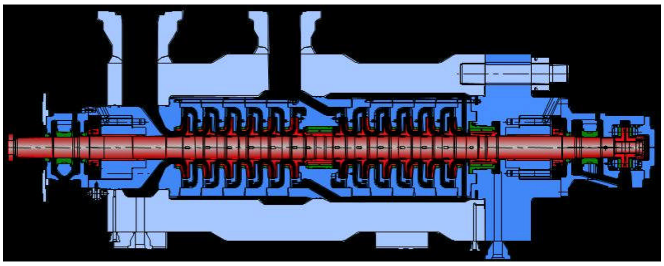

Figure 6. Thunderhorse Water Injection Pump by Sulzer. 338M3/hr at 605 bar discharge. (Courtesy of Sulzer Pump)

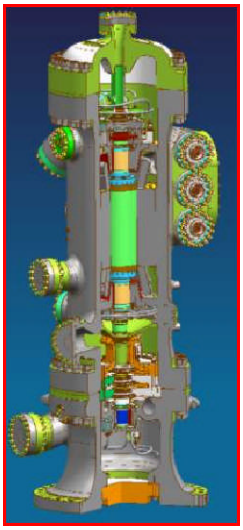

Figure 7. Ormen Lange Subsea Compressor 3D View.

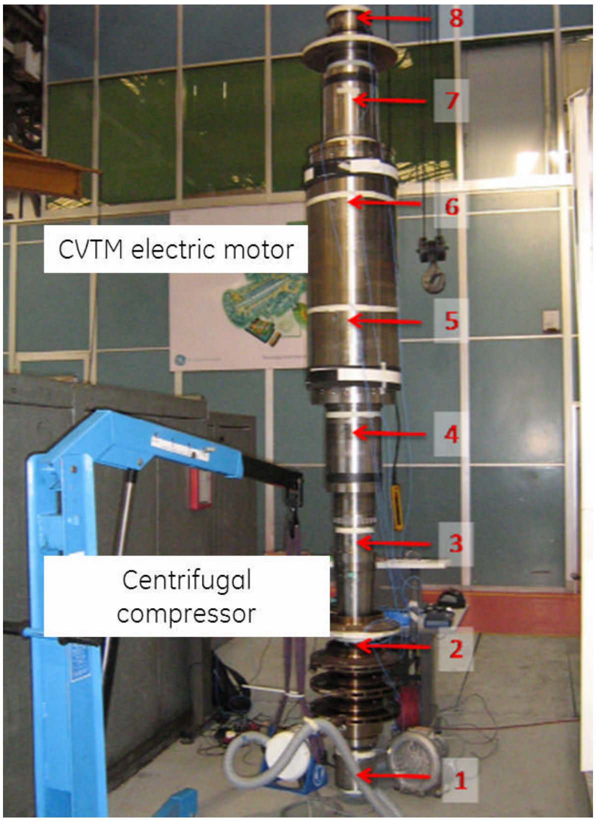

Figure 8. Ormen Lange Compressor Hanging for Free-Free Test

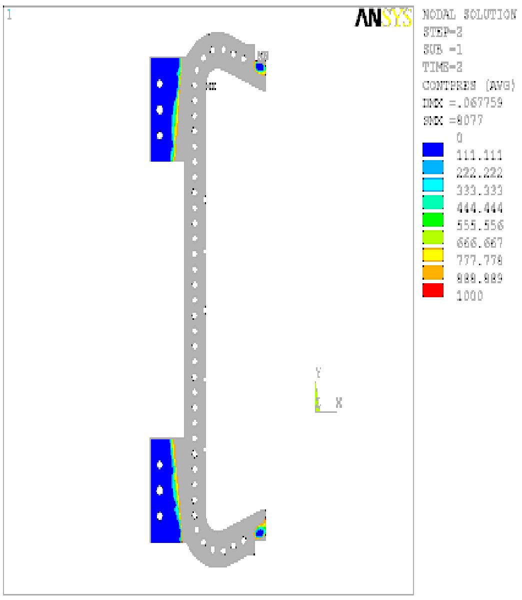

Figure 9
Finite Element Analysis – Case Split Contact Pressure

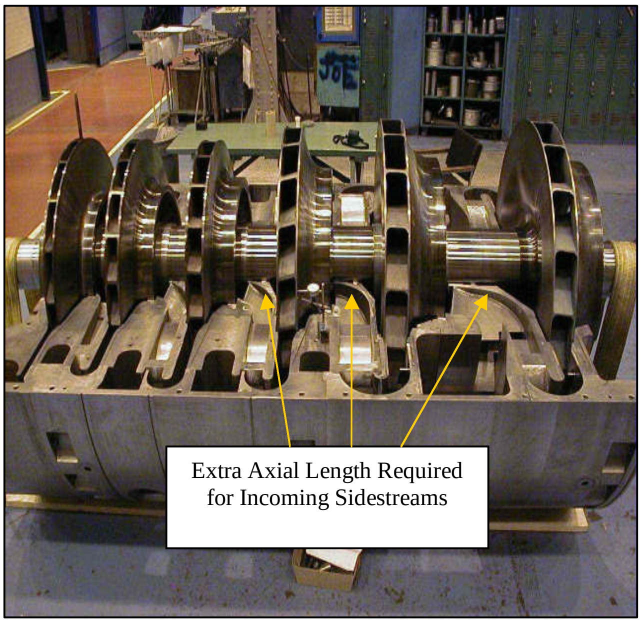

Figure 12. Sidestream Compressor

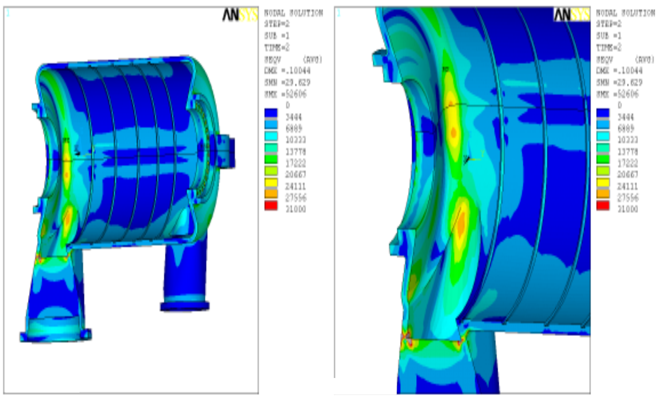

Figure 10. Finite Element Analysis – Case Von Mises Stress

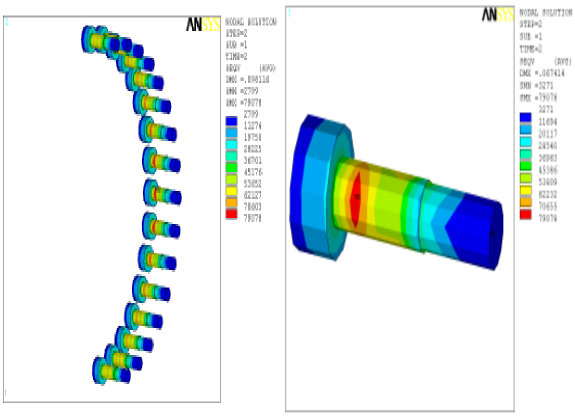

Figure 11. Finite Element Analysis - Bolt Von Mises Stress

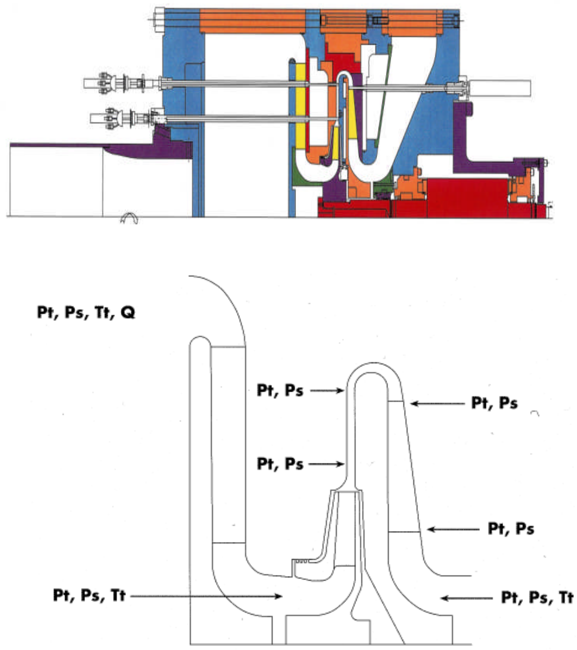

Figure 13. Cross Section of a Single Stage Test Rig

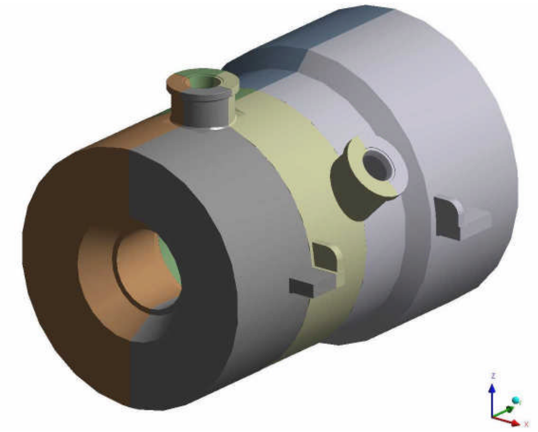

Figure 14 High Pressure Compressor Case

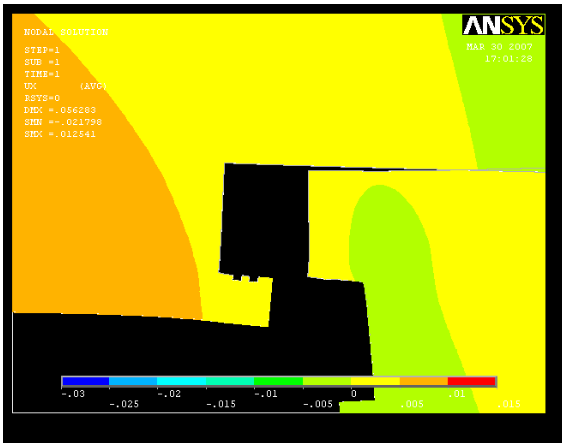

Figure 15 Case Assembly FEA - Deflection analysis (Zoomed-in view of static seal area)

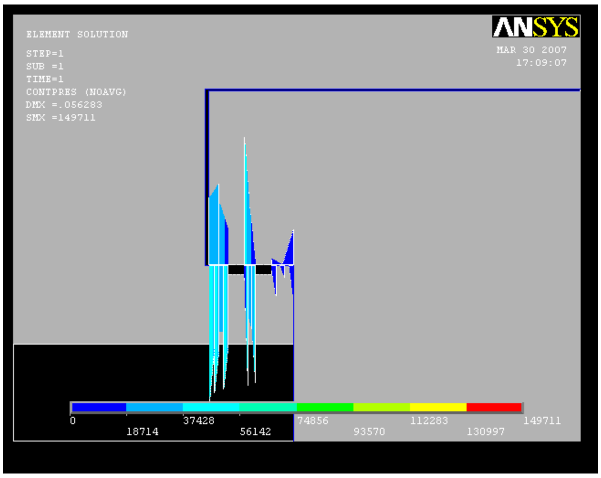

Figure 16 Case Assembly FEA – Contact Pressure Analysis

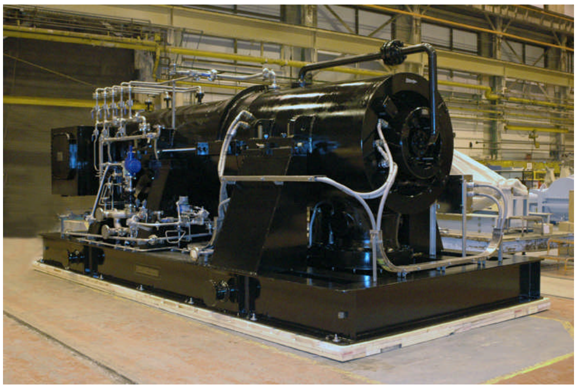

Figure 17   Seal-less Compressor Assembly

Figure 18 Pressure Rated Cable Penetrator

Figure 19 Multi-pin Connector

Figure 20 Cable Seals

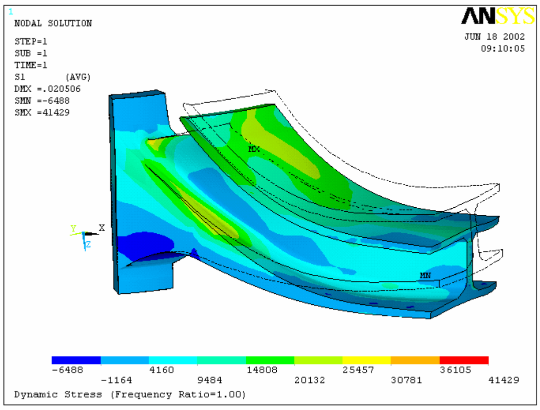

Figure 21 Impeller Natural Frequencies – Mode A

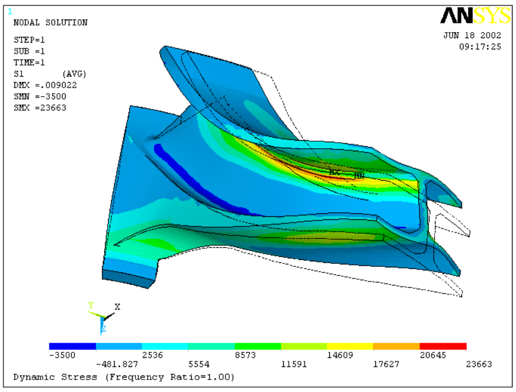

Figure 22. Impellers Natural Frequencies - Mode B

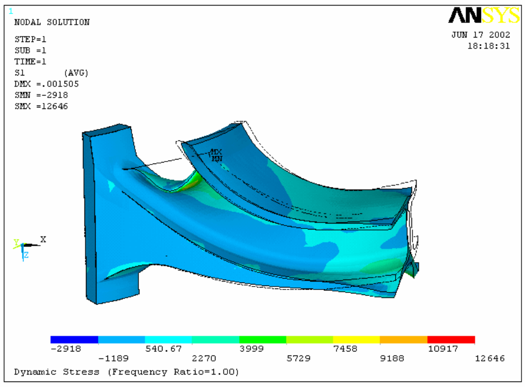

Figure 23. Impeller Natural Frequencies - Mode C

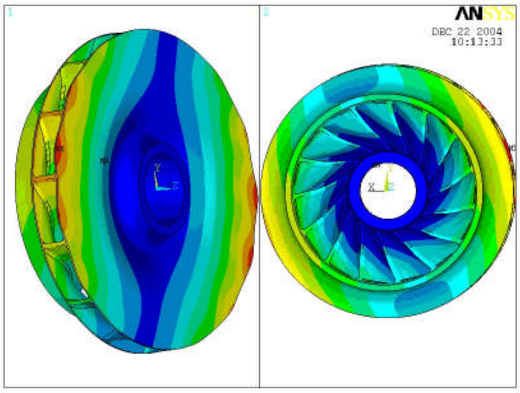

Figure 24. Impeller Natural Frequencies - Mode D

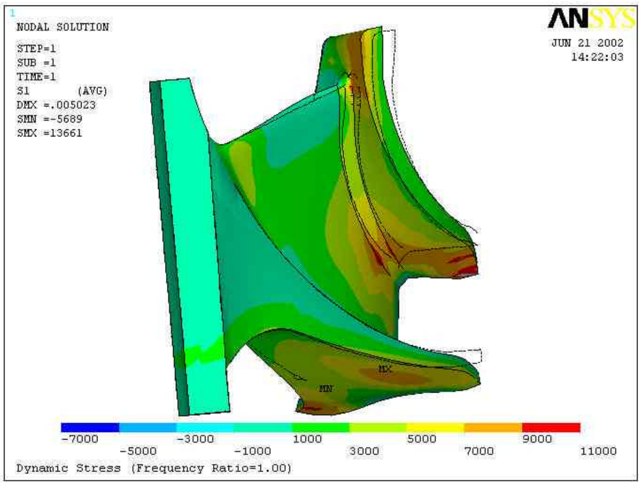

Figure 25. Impeller Dynamic Stresses

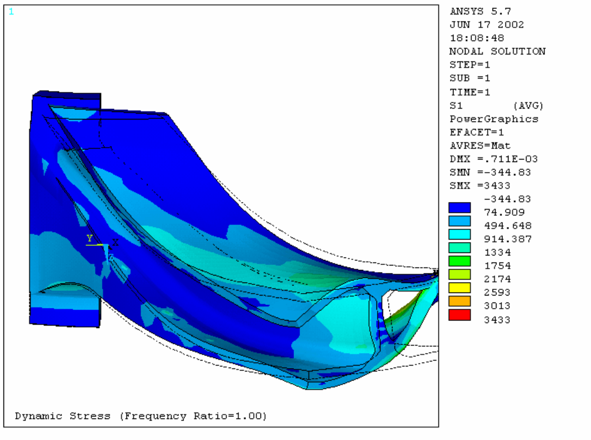

Figure 26 Impeller Mode Shape

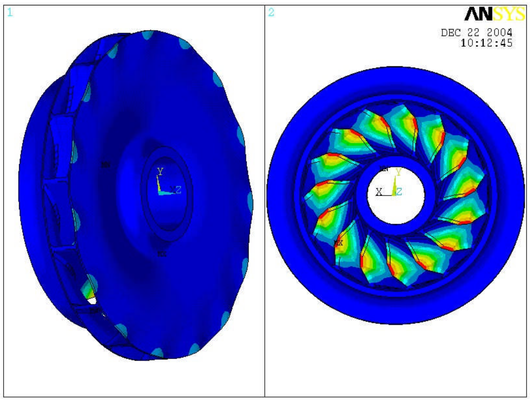

Figure 27 Impeller Mode Shape – Blade Leading Edge Mode

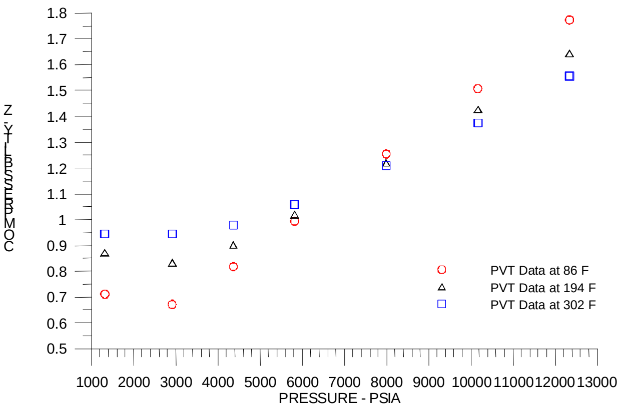

Figure 28 Typical Gas Mixture P-V-T Data

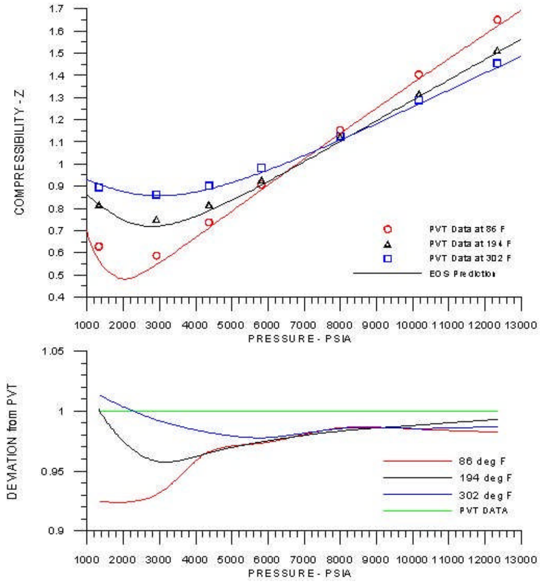

Figure 29 P-V-T Comparison to LKP Equation of State Prediction

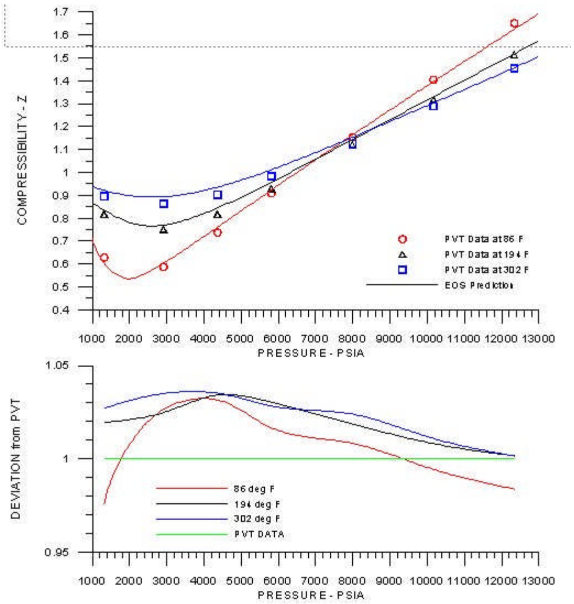

Figure 30 P-V-T Data Comparison to API-Soave Equation of State Prediction

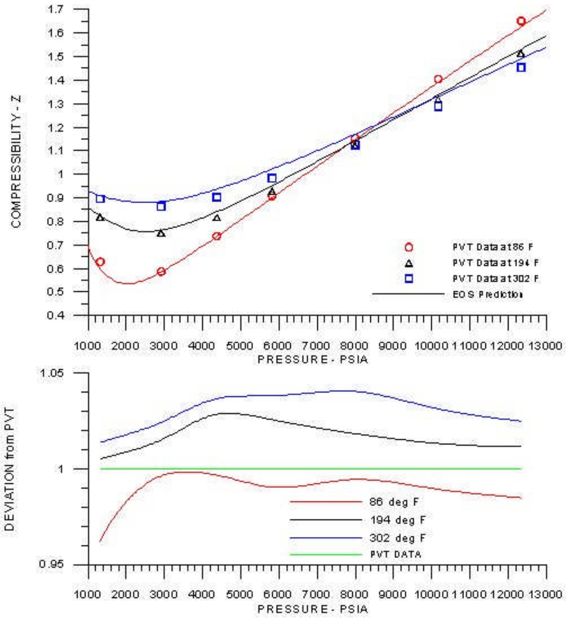

Figure 31 P-V-T Data Comparison to BWRS Equation of State Prediction

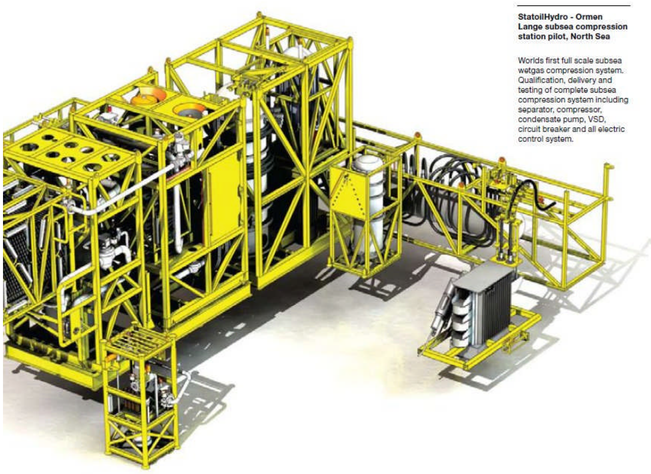

Figure 32 Subsea Compression Qualification Test
Image Source: Aker Solutions – Subsea Processing and Boosting brochure, 2008, (used courtesy of Aker Solutions)

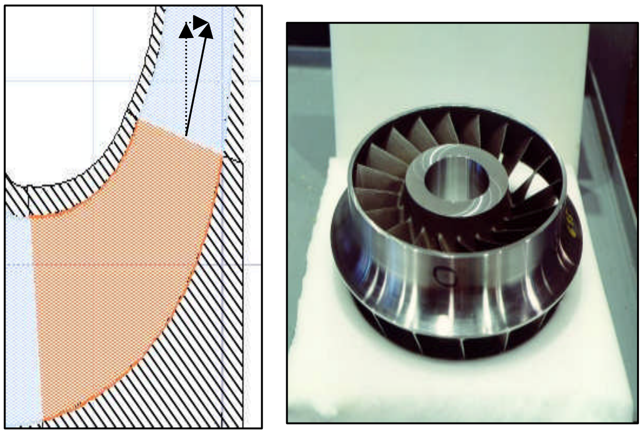

Figure 33. Mixed Flow Impeller - Non-Radial Orientation at Impeller Exit

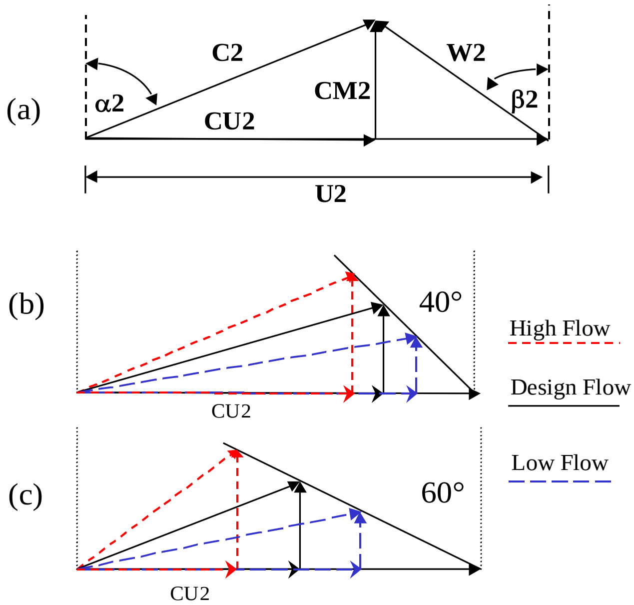

Figure 34. Impeller Exit Velocity Triangles - (a) nomenclature. (b) 40 deg backsweep, (c) 60 deg backsweep

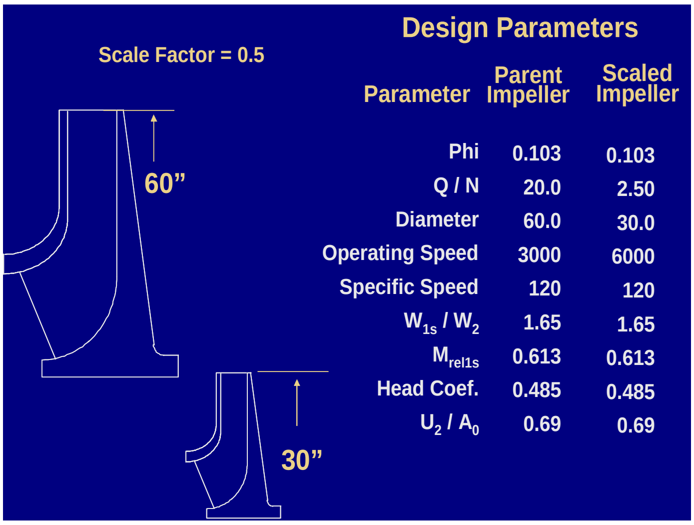

Figure 35. Geometric Scaling

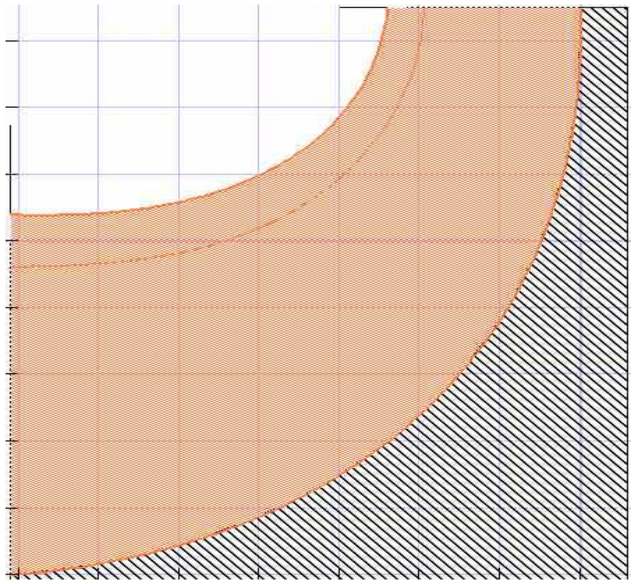

Figure 36. Contour Trimming