Solar Turbines
A Caterpillar Company

POWERING THE GLOBAL ENERGY DEMAND

Centrifugal Gas Compressors

Table of Contents

Page

Caterpillar is a registered trademark of Caterpillar Inc.
Solar, Saturn, Centaur, Mercury, Taurus, Mars, Titan, SoLoNO$_x$, Turbotronic, InSight System, Insight Connect, and TurboFab are trademarks of Solar Turbines Incorporated. All other trademarks are the sole property of their respective owners and should be treated as such.
Specifications subject to change without notice. Printed in U.S.A.
© 2009 Solar Turbines Incorporated. All rights reserved.
SPCC/1009

Introduction

Solar Turbines Incorporated, headquartered in San Diego, California, is a wholly owned subsidiary of Caterpillar Inc., a Fortune 500 company and the world's largest maker of construction and mining equipment, diesel and natural gas engines, and industrial gas turbines. Solar Turbines is the world's leading manufacturer of industrial gas turbines in the 1 MW to 22 MW (1,590 hp to 30,000 hp) range. Solar has more than 13,400 units in operation, which have logged more than 1.35 billion operating hours onshore and offshore around the globe.

Solar® products play an important role in the sustainable development of oil, natural gas, and power generation projects around the world. These products include:

- Centrifugal natural gas compressors
- Gas turbine-driven compressor sets
- Electric motor-driven compressor sets
- Generator sets
- Mechanical-drive packages

Solar also manufactures *Turbotronic*™ microprocessor-based control systems that use programmable logic controller (PLC) technology. Solar builds complete turbomachinery packages that are ready to go to work, no matter where in the world the job might be.

Solar products are highly efficient, rugged, robust, durable and extremely reliable. Customers are operating *Solar* products in the toughest, most challenging environments, including remote arctic, desert, tropical and offshore areas. Solar designs and manufactures its products under various quality systems to ensure the highest reliability. Additionally, the company sells, manufactures, and services its products in more than 96 countries from over 43 locations worldwide.

Solar Turbines began manufacturing centrifugal gas compressors for applications in the oil and natural gas industries in 1960. The first *Solar* gas compressor package (*Solar* C16 gas compressor, driven by a *Saturn* gas turbine engine) was built and shipped in 1961 and installed on a transmission line in Mississippi (Figure i.1). Since then, the company has sold more than 5,000 gas compressors, which have logged more than 560 million operating hours onshore and offshore worldwide, making Solar a major compressor supplier and packager. Solar shipped the 5000th centrifugal gas compressor in October 2007, from Solar's San Diego, California manufacturing facility for installation at a compressor station near Frankfurt, Germany (Figure i.2).

Figure i.1 Solar's first gas compressor installation, 1961

Figure i.2 Solar president, Steve Gosselin, celebrates the shipment of Solar's 5000th gas compressor, 2007

To provide some perspective, in North America more than 80% of natural gas flowing through pipelines flows through *Solar* compressors. Currently, 100 to 150 compressors are produced per year. In addition to new compressors, more than over 200 overhauls and restages are completed each year.

COMPRESSOR PACKAGES

Solar's compressor packages include:

- Gas turbine or electric motor driver
- Single or multiple tandem-mounted gas compressor(s)
- PLC-based control system
- All required auxiliary systems

Solar's gas turbines, gas compressors, and turbomachinery packages are designed and manufactured under quality management systems certified by DNV to conform to ISO 9001 series of quality systems standards. In addition, the gas compressor casings are designed to comply with ASME Code Section VIII and API Standard 617 for materials, minimum strength, and maximum stress levels.

APPLICATIONS

Solar's gas compressors are used primarily for field compression. Applications for Solar's gas compressors demand efficient performance, high reliability, durability, and ease of maintenance, but field compression also imposes other requirements on equipment design. For example, refinery, petrochemical, and refrigeration compressors typically handle gases of relatively constant composition at nearly fixed pressures, while field compressors face operating conditions and mixtures of hydrocarbons that often vary considerably.

Field Applications

Field applications consist of two basic groups: upstream and midstream. The primary upstream applications include gathering associated gas, gas lift, gas injection/reinjection, and gas boosting. Compression at or near the wellhead frequently involves mixtures of various types of gases with differing compositions and pressure/flow requirements. Entrained liquids and solids are encountered regularly.

Common midstream applications include gas transmission, storage/withdrawal, tail-gas recompression, and local distribution. Gas in such applications is relatively clean, dry, and of fixed composition, usually with significant variation in flow and pressure.

Solar's family of compressors has been designed for a large range of applications. Standard impellers are designed with evenly spaced flow ranges. Impellers are built and tested in compressors before they are introduced for sale. This standard approach to compressor design and application yields accurate prediction and consistent performance, as well as lower cost and shorter lead times.

Due to their pre-engineered standard components, Solar's gas compressors can be restaged easily and quickly to restore or enhance performance and efficiency when onsite operating conditions change significantly. The numbers and types of stages can be varied to meet the new conditions, generally without modifying the auxiliary systems. As long as the new conditions are within a gas compressor's overall operating design envelope, the restage will be straightforward.

Solar's experience with modular aerodynamics and use of the same mechanical design in constantly increasing numbers of compressors contributes to our unmatched record of reliability and durability.

As the customer's single source, Solar also accepts total unit responsibility, providing commissioning, training, field service, parts support, repair, overhaul, upgrade, package refurbishment, operation and maintenance, engineering, procurement, construction, asset management, contract power, and financing. Figure i.3 shows typical onshore and offshore compressor stations that contain *Solar* gas compressor packages.

Figure i.3 Typical onshore and offshore compressor stations

This document describes Solar's compressor applications, features, systems, operating characteristics, assembly, testing, and performance as of the publication date.

1. Solar Compressor Features

1.0 OVERVIEW

Solar has been manufacturing compressors for 50 years and during that time has provided proven, reliable products to our customers. Solar, a provider of single source solutions, has a global presence and offers products with valuable, unique features that include:

- Innovative design created specifically for the oil and gas industry
- Cutting edge technologies and processes
- Maximized simplicity, flexibility, and value
- Modular rotor construction methods
- Standardized impellers and components

1.1 DESIGN PHILOSOPHY

Solar's gas compressor design philosophy emphasizes selection of standard components from an extensive database of pre-engineered designs that are combined to meet customer specifications. Pre-engineered designs allow much shorter production lead times and lead to a high degree of confidence in the performance predictions of the offered gas compressors. They also result in rotordynamic behavior that has been verified through pre-testing.

Pre-engineered configurations require the engineering of all components during the preliminary development cycle, including all impellers, guide vanes, seals, bearings, rotors, and materials consistent with the applications' requirements. Head and flow will vary significantly from project to project; a large database of standard components greatly simplifies the compressor configuration process. Prior to shipment, each unit is tested to verify the contract performance.

This approach yields predictable performance and a high degree of mechanical integrity. It also provides consistent parts for overhaul and restage efforts, making configuration management easier and more accurate. In addition, customers receive replacement parts of consistent design and quality.

Solar compressors are specifically designed to address the requirements of the oil and gas industry in upstream and midstream applications. Impeller aerodynamic design is optimized for natural gas, but it has enough flexibility for other gas applications.

Solar's gas compressors are developed to match the speed and power outputs of Solar's line of gas turbines, and thus may be used without gearboxes in direct drive applications (Figure 1.1). *Solar* compressors are also regularly used in applications that require electric motor drivers in place of conventional gas turbines (Figure 1.2).

Figure 1.1 Solar compressors matched to Solar gas turbines

Figure 1.2 Electric motor-driven compressor

Depending on the model, these compressors have one to ten stages to handle a variety of inlet flows and pressure ratios. Compressor packages with a single compressor can produce ratios of more than 4:1; multi-body arrangements produce pressure ratios of almost 40:1. Design features used in *Solar* compressors lead to excellent efficiency, range, and durability. These features include a low pressure-loss radial inlet that allows for an efficient beam-style machine supported by axially spaced radial bearings. Tilt-pad thrust and journal bearings, dry couplings, and high-pressure dry gas seals make the machines rugged and durable. Advanced aerodynamic impeller and diffuser designs along with volute discharge systems contribute to the efficiency of these compressors. When compared to plenums, volute discharges improve pressure recovery and performance by eliminating pressure dump losses associated with large area changes along the flow path.

Solar has two distinct compressor product lines: pipeline compressors and production compressors.

Figure 1.3 Compressor product families

The impellers for these two product lines are designed with different emphases:

- Pipeline compressors are focused on the needs of the natural gas transmission industry (Figure 1.4). These compressors are designed for unmatched high efficiency/wide flow range, and have a maximum of three stages. To improve efficiency, the impellers for pipeline compressors are longer than the impellers used in multi-stage compressors. Figure 1.5 shows a typical cross section of a pipeline compressor.
- Production compressors are focused for applications at or near the oil and gas fields (usually referred to as upstream applications), where high head or high-pressure ratios are required with wide operating ranges (Figure 1.6). These compressors are designed to optimize head as well as efficiency and flow range. Impeller back-sweep angles are lower to increase head per stage, and impellers are shorter than those found in pipeline compressors, enabling more impellers into a multi-stage compressor body.

Figure 1.4 C65 pipeline compressor

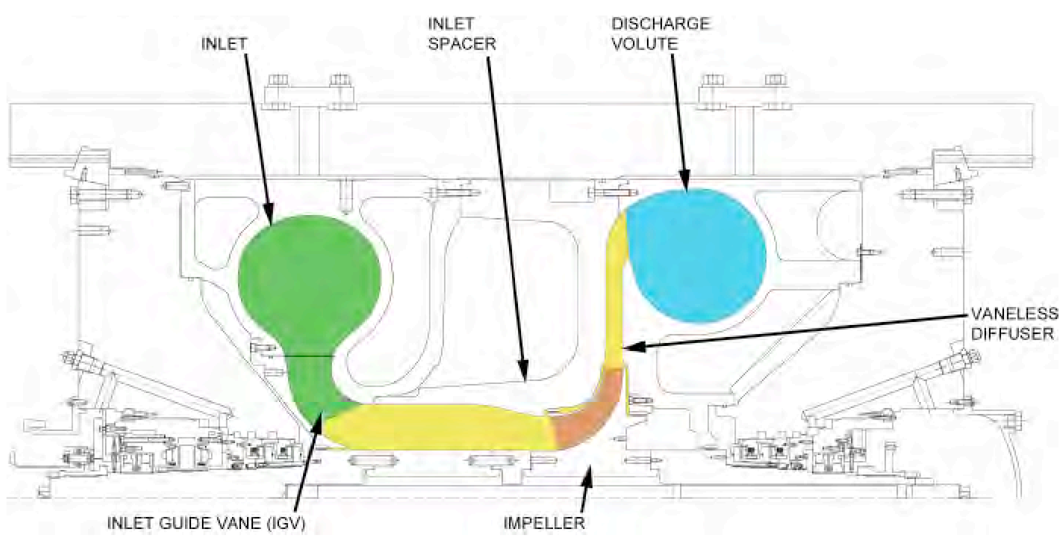

Figure 1.5 Pipeline compressor cross section

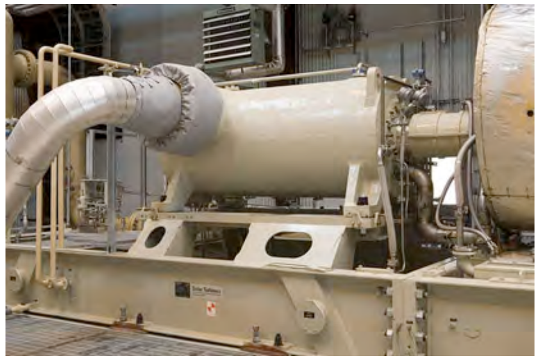

Figure 1.6 C51 production compressor

This standardized design approach results in lower initial costs and lower operating costs. It also reduces the lead-time for the initial product, waiting time for repair parts, and out-of-service time for compressor restages.

Another design consideration is tools that support field maintenance. A complete restage of a compressor has been accomplished in one work shift using Solar's specially designed field tools. Proper tools permit quick removal of the discharge end cap, rotor/stator assembly (module), insertion of the restaged module, and replacement of the endcap within eight hours. Figure 1.7 shows a typical cross section of a production compressor.

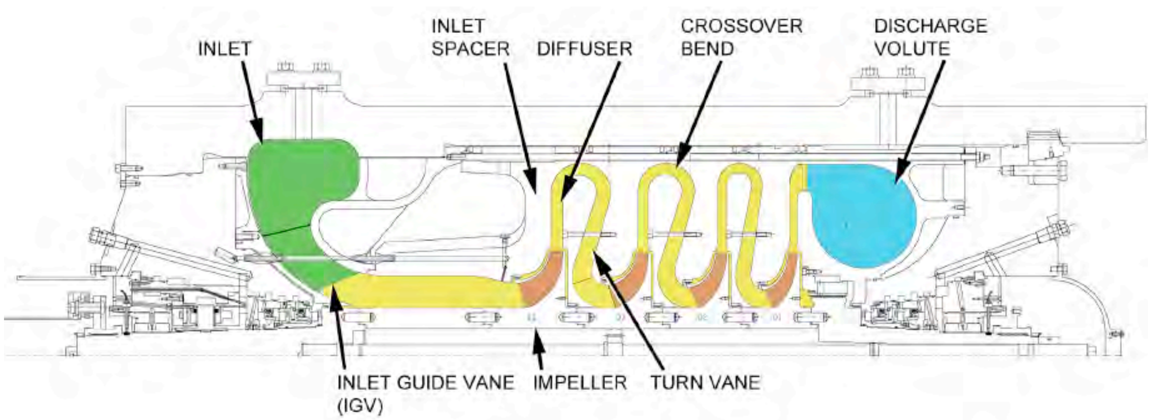

Figure 1.7 Production compressor cross section

1.2 MODULAR CONSTRUCTION

The centrifugal compressor/pump industry has traditionally used solid-shaft rotor with interference-fit impellers. This rotor assembly does not readily permit the substitution of different impellers on the shaft.

Aircraft turbojet engines, followed by industrial gas turbines, successfully use modular rotors. Solar transferred its experience of the gas turbine rotor design to gas compressors. The modular rotor has important advantages:

- Easy to assemble and disassemble, making restages fast and economical.
- Yields a stiff shaft, with favorable rotordynamic characteristics as explained below.
- Allows solid diaphragms— no gas leakage at the diaphragm split line, with associated loss of efficiency.

Figure 1.8 illustrates solid shaft and modular rotor construction. Comparing the two designs, and assuming that both use the same impeller inlet diameter, a solid shaft rotor design will end up with a smaller effective shaft diameter, which results in a more flexible shaft. From a rotordynamic standpoint, a stiffer shaft is a more desirable design.

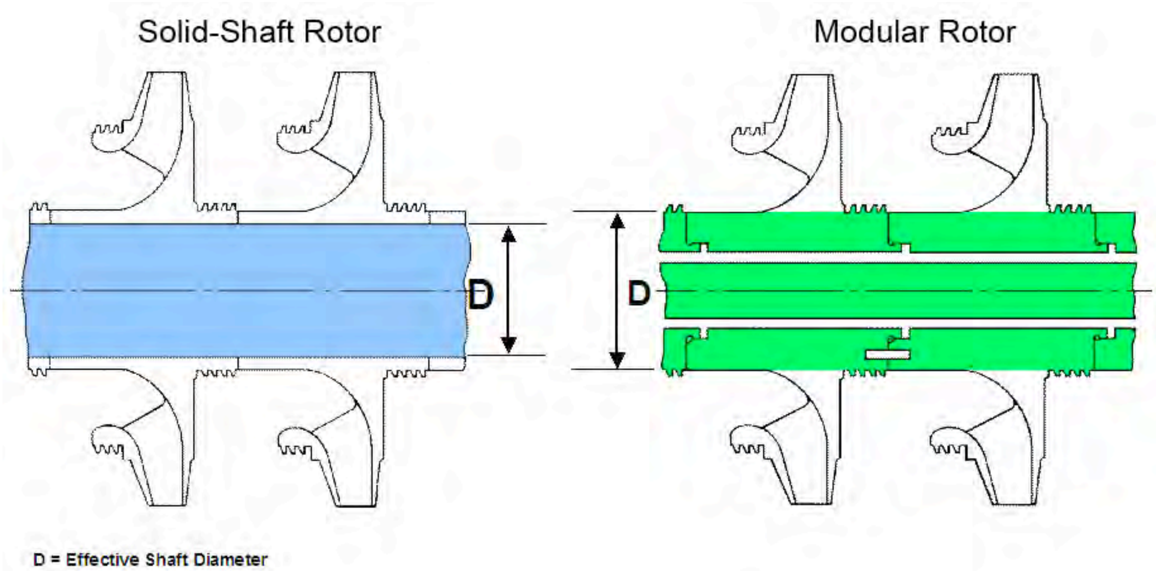

Figure 1.8 Solid shaft rotor vs. modular rotor

Finite element analysis (Figures 1.9 and 1.10) shows the benefit of the modular rotor design in using centrifugal force to maintain a tight radial fit during operation. As can be seen in Figure 1.9, this growth can be significant. The interference between the impeller and the solid shaft complicates assembly and disassembly operations. On the other hand, that same impeller designed for modular rotor construction uses the radial centrifugal growth of the impeller to tighten the interference fits. The right hand male pilot shown in Figure 1.10, grows radially more than the female pilot of the adjacent part, resulting in a tighter fit during operation. An example of a female pilot callout is shown on the left hand side of Figure 1.10. Lower interference fits and short pilots make the modular rotor easier to assemble and disassemble than a solid rotor.

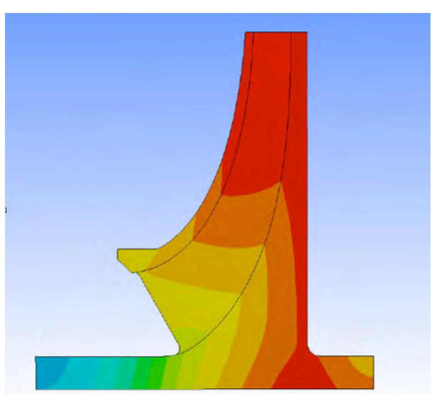

Figure 1.9 Solid rotor centrifugal growth

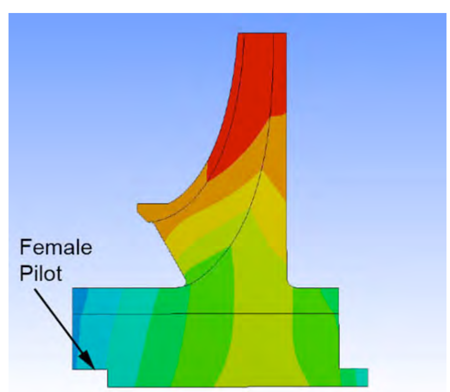

Figure 1.10 Modular rotor centrifugal growth

The American Petroleum Institute (API) recognizes Solar's modular design as compliant with API 617 requirements. API approved this design based upon Solar's field experience of the modular design in combination with the stringent, quality-focused manufacturing process of the modular rotor.

Figure 1.11 shows Solar's modular rotor assembly, which consists of stub shafts, impellers, rotor spacers (if required to maintain a constant bearing span) and a tie-bolt (center bolt).

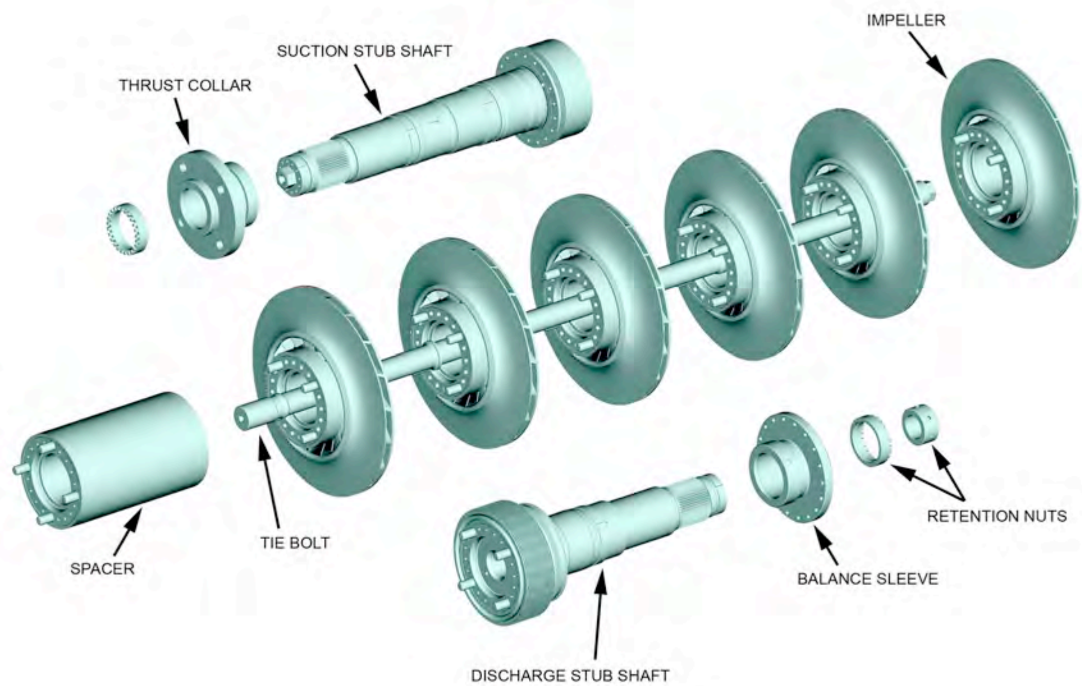

Figure 1.11 Typical modular rotor assembly

1.3 STANDARDIZATION

The aerodynamic design of the compressor makes use of standardized, pre-engineered, and pre-tested components, rather than customized stages. This standardization results in predictable high performance, improved reliability, improved delivery, and better economics. Combining a standardized, pre-engineered approach with modern computational tools, and with expanded test capabilities, Solar has been able to design standard compressor stages with high efficiency and a wide operating range by carefully optimizing each individual impeller and diffuser, together with the inlet system and the discharge volute.

Predetermined rotor spans eliminate most of the concerns regarding the rotordynamic performance because the rotordynamic behavior is known from previously built and tested, identical compressors. For example, the rotor system of a C40M compressor is identical to any other C40M with the same number of stages with regards to the bearing span, the bearings themselves, and the rotor design.

Scaling plays an important role in the standardization of Solar compressors. The C85, C65 and C45 pipeline compressors are geometric scales of the C40P compressor. Similarly, C41 and C61 production compressors are geometric scales of the C51 compressor. Scaling preserves compressor performance, shortens the time to market, and improves ability to predict performance. Figure 1.12 shows the non-dimensional compressor performance for the same staging in C40P, C45, and C65 size. As can be seen, scaling gives nearly identical performance.

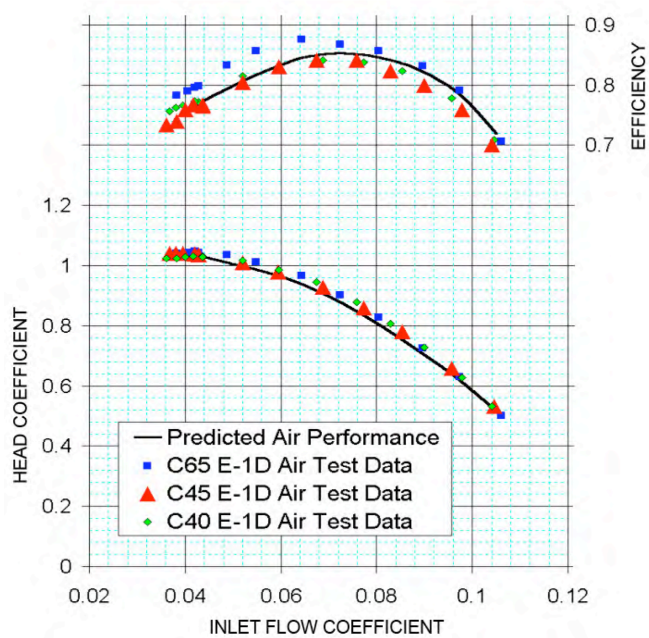

Figure 1.12 Scaling preserves compressor performance

Field testing under real conditions in carefully instrumented site performance tests (Figure 1.13) shows the advantages of our design methodology in meeting excellent levels of both efficiency and range, and matching compressor performance prediction.

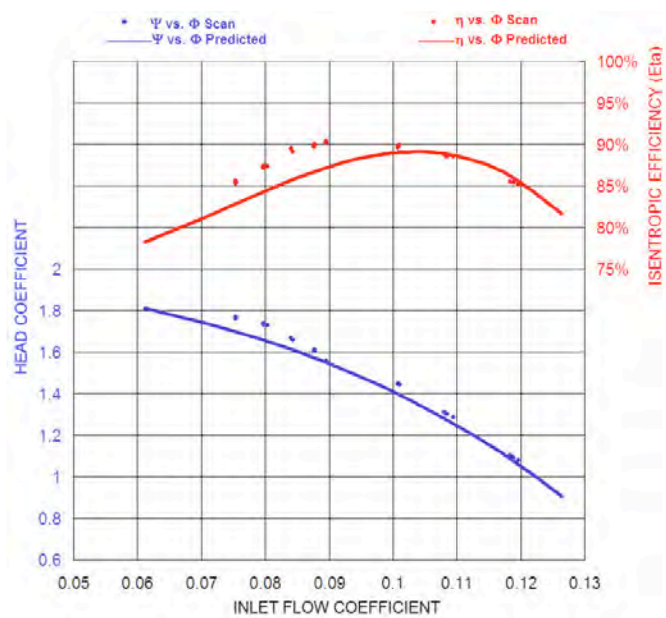

Figure 1.13 High efficiency and wide operating range verified at site conditions

Note: Solar defines inlet flow coefficient as follows:

$$\Phi = \frac{Q}{\frac{\pi}{4} x D_{tip}{}^{2} x U_{tip}}$$

Where Q = volume flow

D_{tip} = impeller tip diameter

U_{tip} = impeller tip velocity

Solar uses different manufacturing methods for the production of impellers: machining by five-axis milling technique (Figure 1.14), brazing together a machined shroud and a machined open-faced impeller (Figure 1.15), or precision investment casting (Figures 1.16 and 1.17). The appropriate method is selected based on size, flow coefficient, and manufacturing quality to ensure that the impeller meets stringent dimensional quality guidelines necessary for consistent aerodynamic performance. Due to no appreciable variance in aerodynamic geometry, as well as extensive data from more than 5,000 compressors in service, our application engineers provide highly accurate performance estimates for both new units and for restaged units.

Figure 1.14 Machined impeller

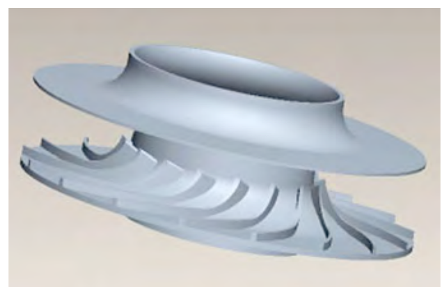

Figure 1.15 Brazed impeller

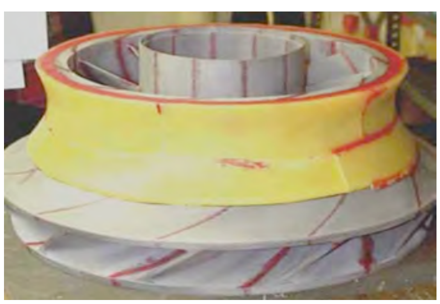

Figure 1.16 Impeller wax model

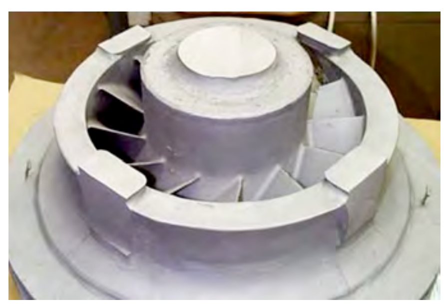

Figure 1.17 Investment cast impeller

Identically manufactured impellers exhibit consistently uniform performance and demonstrate close correlation between calculated and actual performance. Standardized aerodynamic components result in performance estimates that easily fall within acceptable industry ranges or even exceed the required level of accuracy. Figure 1.18 shows a family of impellers for a production compressor.

Figure 1.18 Impeller family

1.4 NEW TECHNOLOGIES AND CAPABILITIES

Design Challenges

The art of designing a gas compressor constitutes the effective blending of aerodynamic requirements with rotordynamic and structural constraints. Far more effort can be spent in careful optimization of standardized aero components than if the components were customized. The optimization process makes extensive use of three-dimensional (3-D) components modeling (Figure 1.19), the latest available computational fluid dynamics (CFD) tools, finite element analysis (FEA) and scaled model testing at Solar's aero test facility (ATF) described later in this brochure.

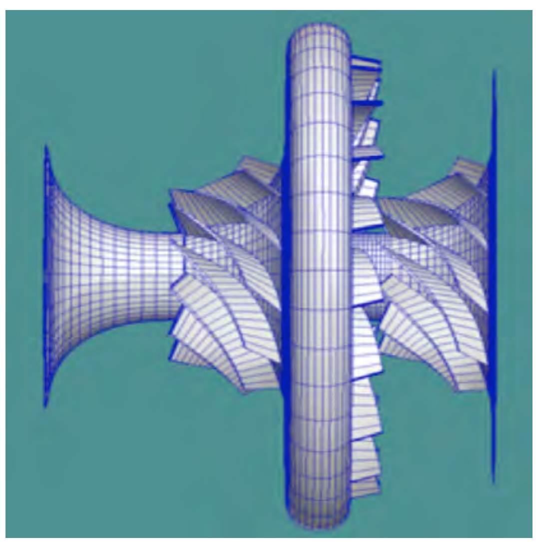

Figure 1.19 High efficiency component design

Aerodynamic Analysis Tools

Every compressor stage designed by Solar is analyzed using modern CFD tools. The CFD code simulates the 3-D flow field inside rotating or non-rotating components. Components are reviewed at design and off design conditions along a given speed line. Fluid interactions are studied with coupled analysis such as impeller with a volute (Figure 1.20 and Figure 1.21), with a return channel, or with a radial inlet. A detailed 3-D analysis provides a visualization of the 3-D flow field, enabling geometry adjustments to minimize secondary flow and thus pressure loss reductions such as drag. CFD analysis can also be used to calculate component performance.

Figure 1.20 Coupled analysis: impeller, diffuser and volute grid

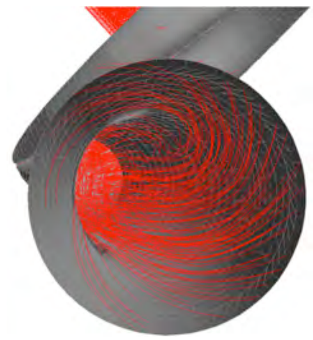

Figure 1.21 Coupled analysis: flow lines at volute exit

Advanced aerodynamic CFD analysis is being used to direct the design of the interstage discharge volute by incorporating a vaned diffuser into the support struts (Figure 1.22). This will maintain excellent performance for the two-compartment compressor. Figure 1.23 shows a typical result of CFD calculation of an impeller.

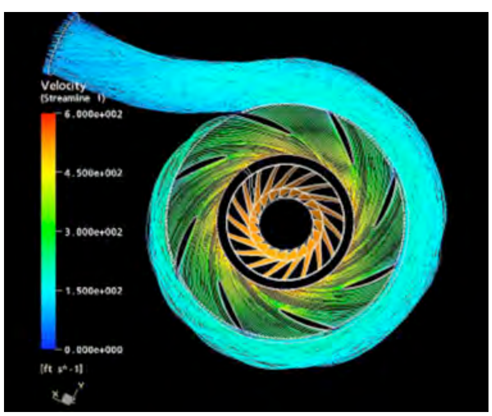

Figure 1.22 CFD of intermediate discharge of dual-compartment C51D

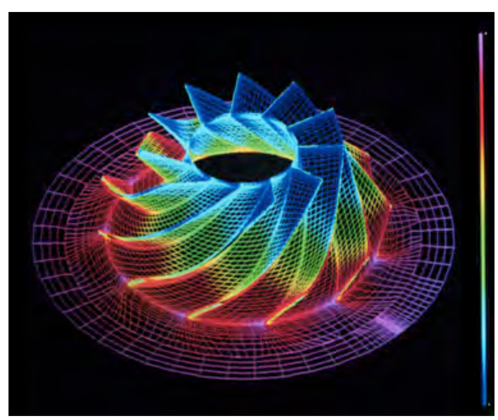

Figure 1.23 Impeller pressure distribution

Rotordynamic Analysis Tools

The primary design goal in rotordynamics is to ensure that compressor mechanical performance is stable under the operating conditions required by the customer. Solar ensures that all of its gas compressors meet or exceed API 617 response requirements for centrifugal gas compressors.

The following design goals are established in the conceptual design phase:

- Rotor insensitive to unbalance
- All bending modes above operating speed
- Lower modes critically damped
- Stability margin high

The primary rotordynamic considerations are the vibration level at operating speed and the location of resonant frequencies. The compressor rotor should be insensitive to unbalance throughout the operating speed range. Inevitably, rotors will have residual unbalance that appears as vibration during operation. The rotor system should be able to accommodate the residual unbalance limit established by API without exceeding the vibration limit. A set of finite element-based rotordynamic models, coupled with advanced fluid-film bearing simulations, comprise the basic tools for rotordynamic analysis.

Figure 1.24 compares the measured amplitude at the suction side X-Y vibration probes for the speed range of a C45 compressor during an unbalance sensitivity test and its simulation. It shows that different modeling techniques of a journal bearing can affect bearing stiffness and damping characteristics and thus the overall rotor vibration. The older model for bearing simulation (shown in brown, assuming an adiabatic bearing) overshoots at higher speeds. An 80/20 mixing model, (80% hot oil from the preceding pad mixed with cooler fresh oil at the inlet of the next pad, shown in blue), significantly improves the correlations between test and prediction; this model therefore describes the behavior of the lube oil once it enters the bearing.

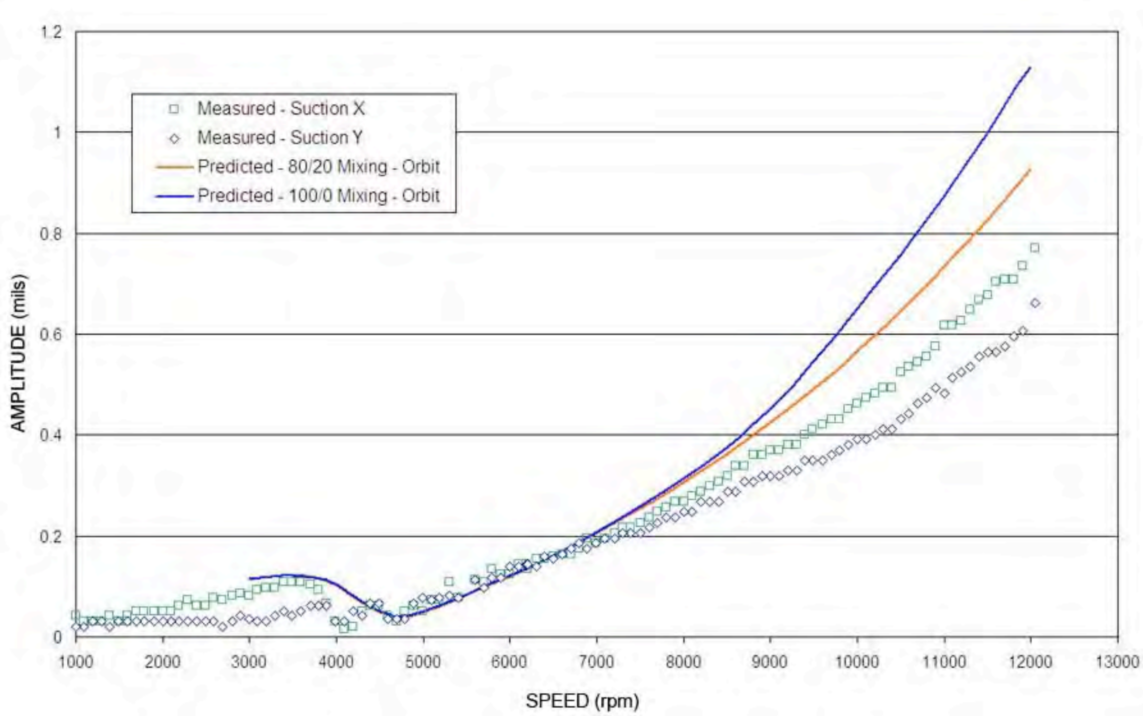

Figure 1.24 C45 unbalance sensitivity calculations

Because all C45 two-stage compressors use the same stub shafts and journal bearings, as well as the same bearing span, the test data presented here represent the response characteristics of all C45 two-stage compressors.

In order to accurately predict rotor response to unbalance, both rotor and bearing models need to be validated. As shown in Figure 1.25, modal testing is conducted to verify the accuracy of the rotormass-elastic model. The rotor is suspended by flexible straps and accelerometers are placed on the rotor to measure frequency as well as amplitude while a modal hammer strikes the rotor. The mathematical model of the rotor can thus be refined by matching both resonant frequencies and mode shapes.

Figure 1.25 Modal testing of a C61 rotor

The use of proven analysis tools combined with a large number of tested, pre-engineered rotor systems provides high confidence levels for predictions even during the project inquiry stage. All empirical data accumulated with the compressors are applied to all new projects. This validation leads to:

- High repeatability of performance levels
- High first-time test success rates for every new project
- Firm delivery commitments of hardware

Due to Solar's unique modular system, the rotordynamic characteristics of every compressor are not based on purely theoretical predictions, but rather on actual test data from an identical rotordynamic system. This takes a large amount of uncertainty out of any new project.

Lateral Analysis

All *Solar* gas compressors are pre-engineered products with extensive rotordynamic analyses early in the design stage. The predicted unbalanced response is validated during development test. Therefore, it is not necessary to perform individual lateral analysis on such a standardized product. However, Solar will provide a lateral analysis, as an option, to our customers based on the as-sold configuration. The content of this report satisfies the intent of a lateral analysis as described in API 617 or ISO 10439.

Solar's lateral analysis consists of three major calculations:

1) Undamped critical speeds and critical speed map
2) Damped unbalanced response
3) Damped stability

The undamped critical speed calculation provides natural frequencies of the system as a function of the support stiffness, neglecting any damping. A critical speed map overlaid with the expected bearing stiffness indicates potential critical speed locations of the drive train. Mode shapes of the lower modes are also shown in the plots.

The damped, unbalanced response analysis includes both stiffness and damping from the journal bearings. Various scenarios of unbalance lead to a prediction of rotor vibration amplitude as a function of speed (Bode Plot). In a typical analysis for a beam-type machine, three scenarios are selected such that the first three lower modes can be excited. This calculation shows how the rotor will behave under some of the worst unbalanced conditions.

For screening purposes, a stability analysis is carried out to identify the stability margin of the application. Based on the criteria established by API, either a Level I or Level II stability analysis will then be conducted.

Torsional Analysis

In case of electric motor drives, in multi-body trains or when otherwise requested, Solar will analyze the torsional dynamic behavior of the rotating shaft system as it undergoes forced vibrations to ensure the integrity of the shaft against torsional fatigue or overload.

The first step in a torsional analysis is to determine the undamped torsional natural frequencies of the drivetrain. The natural frequencies are then plotted on a Campbell diagram to determine if there are any torsional interferences with the 1X or 2X drivetrain order lines within +/- 10% of the operating speed range.

If any torsional interferences are found, a forced damped response analysis is performed per API guidelines to determine if the interferences are acceptable, using the Modified Goodman Diagram approach. Fatigue strength reduction factors are determined for the most highly stressed locations in the drivetrain taking into account such factors as stress concentration, temperature, shaft size and material strength. The calculated torsional fatigue strength is compared to the calculated alternating torsional shear stress. The drivetrain is considered to be torsionally acceptable if appropriate safety factors for infinite life are found, otherwise coupling changes or other modifications are evaluated to achieve a satisfying solution.

Structural Analysis Tools

The FEA method has become an integral part of Solar's structural design process. These tools can be used to complement traditional design methods and have the capability to analyze and design components that were previously sized by either iterative testing or conservative scaling from earlier designs. The major benefit derived from FEA is the appropriate sizing of complex parts.

Oversized structural components, such as casings, drive up both weight and cost of the compressor and, in the case of aero components, can degrade compressor performance. The most significant improvement in structural analysis has gone to the impeller. The aero requirements of high head, high efficiency, and a wide operating range generate high blade stress for a shrouded impeller. Prior to FEA, accurate analytical prediction of stress in the impeller's shroud and blades was very difficult. The current practice for new impellers is to design the initial impeller blade shape using the 2-D aero code. A computer file containing the proposed blade coordinates is loaded into a solid modeling program. Additional information, such as shroud thickness distribution, fillet radii and hub geometry, is entered. The geometry from the solid model is then read directly by the FEA analysis program. This process reduces the time it takes to generate this geometry from weeks down to minutes. The solid geometry is also used to generate detailed engineering drawings that dramatically reduce the opportunity for data entry errors.

The impeller geometry is solved and post-processed by the general purpose FEA program. Figure 1.26 shows a typical impeller FEA model. The result of the analysis is an accurate prediction of the stress and deflection distributions throughout the impeller. Based on the stress results, the geometry can be modified in the programs and another iteration can be run. Because of the rapid turnaround, the best compromise between aero performance and structural integrity can be found quickly.

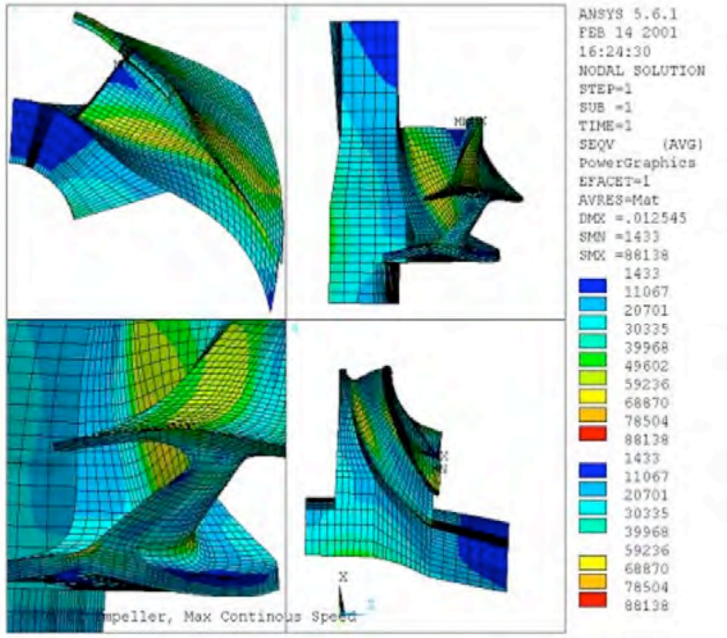

Figure 1.26 Impeller structural analysis

As an example, during the detailed design of a high flow stage impeller for the C40M compressor, FEA predicted unacceptable stress levels within the shroud. After consideration of many options, the final solution was a uniquely contoured fillet that reduced the stress to an acceptable level, without compromising the aero performance. The contoured fillet also improved the castability of the impeller.

FEA is also employed for the modal analysis of an impeller (Figure 1.27). Modal testing is conducted and compared to the prediction to validate calculations of natural frequencies and mode shapes.

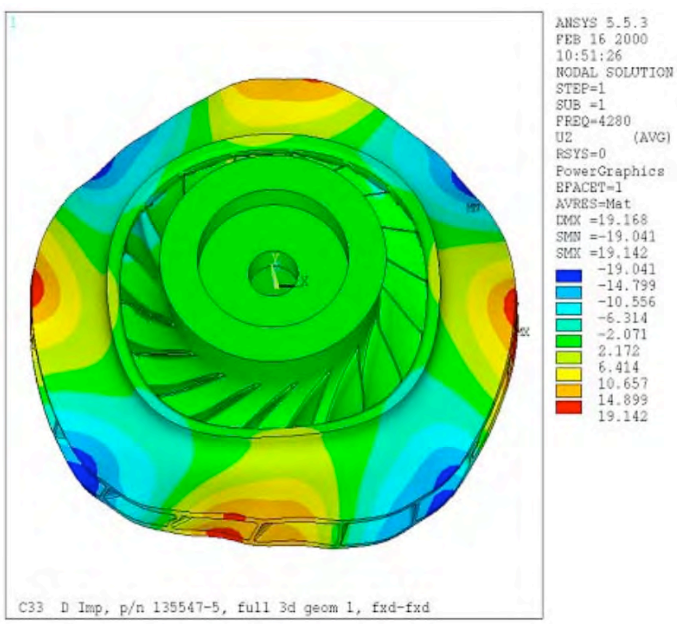

Figure 1.27 Typical FEA result

Figure 1.28 shows the actual mode shape and frequency using a production part. The mode shape and frequency show excellent agreement to the prediction. This validates the finite element model of the impeller and its ability to predict mode shapes.

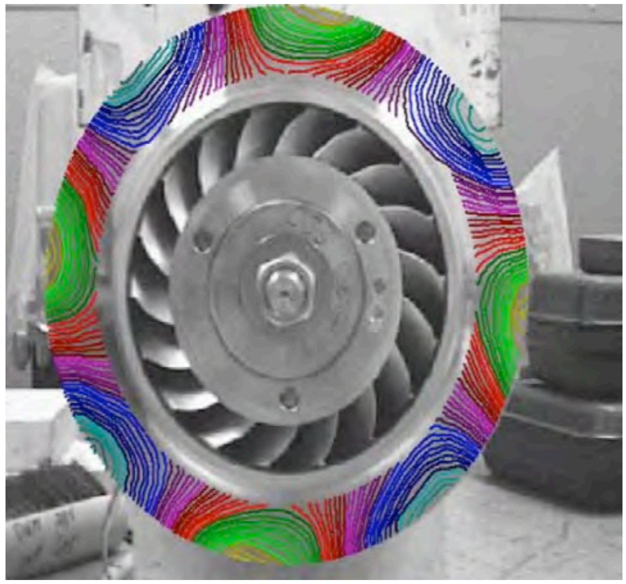

Figure 1.28 Mode shape and frequency

1.5 ONGOING DEVELOPMENTS

New compressors are being readied for production. The model C51D (dual compartment) compressor (Figure 1.29), a first in Solar's compressor product line, will offer four nozzles per casing, allowing for intercooling and side-streams that can be managed in a single casing, rather than requiring the traditional two-casing approach. The C51D will provide higher compression ratios and increased compression efficiency than previous solutions. It will serve in applications such as associated gas gathering, recompression, injection, storage and withdrawal, as well as certain high ratio/high-pressure applications.

The high-pressure C41 (Figure 1.30), will expand the line of successful production compressors to pressure ratings above 3500 psig.

These compressors combine the latest state-of-the-art technology with the experience and reliability that arise from building and installing over 5,000 compressors. They are designed for severe environmental and operating conditions in compliance with API 617.

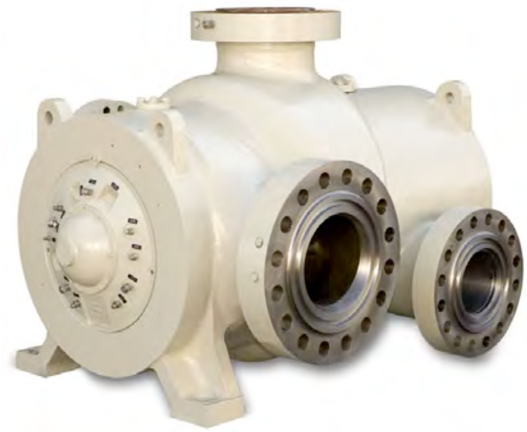

Figure 1.29 C51D dual-compartment production compressor

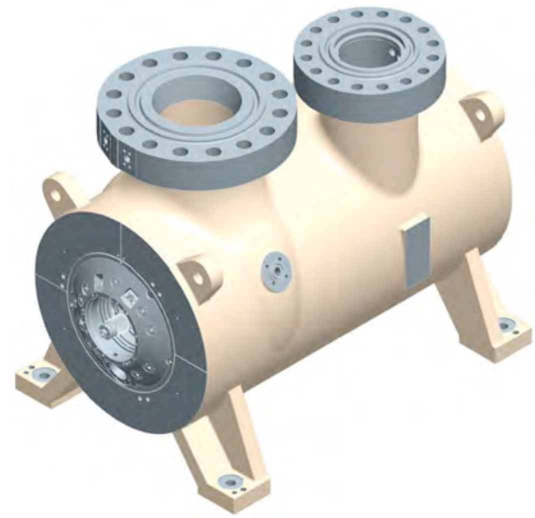

Figure 1.30 C41 production high pressure compressor

The C51D maximum case rating pressure is 3,000 psig, with maximum speed capability of 12,000 rpm. The C41 introductory case rating pressure is 3,750 psig with maximum speed capability of 14,300 rpm.

Solar's design process and test specifications comply with the latest edition of API 617. The new design tools are applied to all *Solar* gas compressors to provide comprehensive solutions for both the production and pipeline gas compression needs of the industry. By blending advanced design tools, experience and tested design practices, a new compressor with exceptional performance, rotordynamics and reliability can be developed in a remarkably short time.

2. Applications

2.0 OVERVIEW

In the production, transportation, and use of natural gas, compression applications are divided into three groups: upstream, midstream, and downstream. The majority of *Solar* compressors serve in upstream and midstream applications.

Applications at or near the oil and gas fields are known as upstream applications. Midstream applications handle the transmission and storage of gas. Downstream applications include compressors in chemical plants and refineries where they may process hydrocarbons and elemental gases, but may also serve as air compressors and compressors for refrigeration cycles. Solar does not offer compressors for downstream applications.

Solar has sold more than 5,000 compressors worldwide, which have logged more than 560 million operating hours in onshore and offshore installations around the globe, serving upstream and midstream applications. Some pipelines use only *Solar* centrifugal gas compressors driven by *Solar* gas turbines and electric motor drives

Figure 2.1 shows the overall head-flow coverage of *Solar* centrifugal compressors.

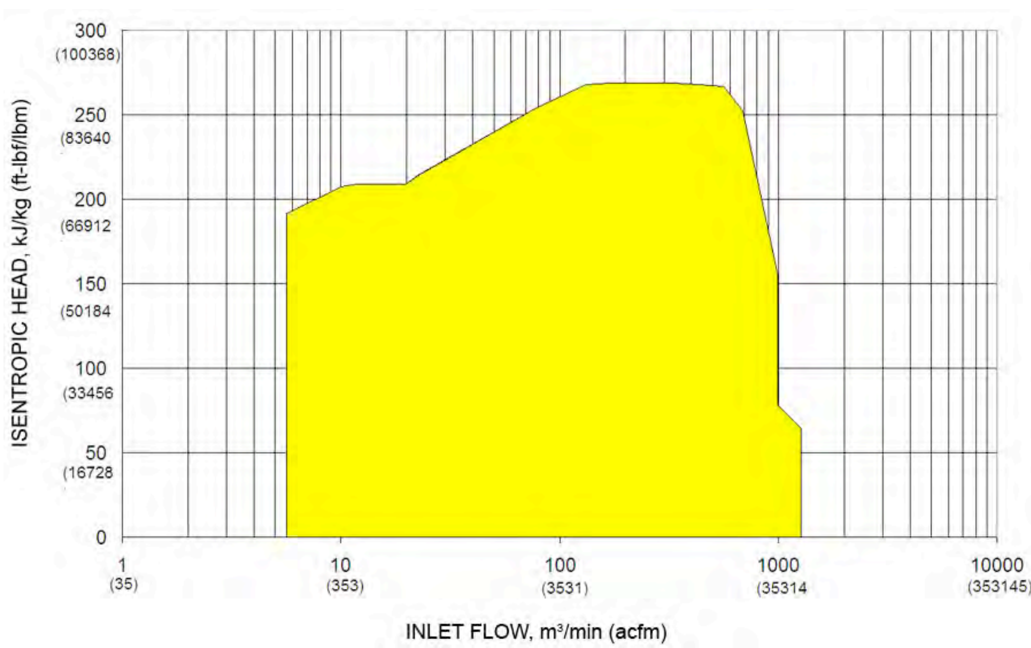

Figure 2.1 Overall head-flow coverage map

Solar's early gas compressor model lines, described as general purpose compressors (Figure 2.2), had flexible designs so that they could meet a wide range of applications. They were, and still are, widely used in midstream applications, as well as upstream applications. All of these products are matched to our gas turbine drivers in terms of power and speed.

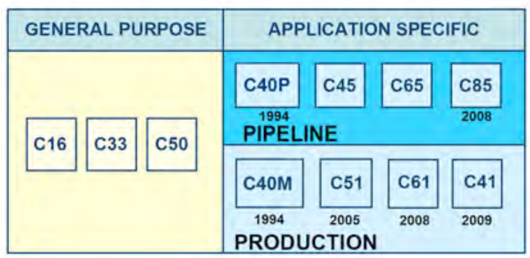

Figure 2.2 Solar family of compressors

Figure 2.3 C65 compressor

Solar's early C16 model was directly coupled to *Saturn* gas turbine engine operating at 22,000 rpm. The C30 and C33 compressor line was driven by *Centaur* and *Taurus* gas turbines, and the C50 was driven by the *Mars* gas turbine. As Solar's gas turbine products steadily grew in power, so did the range of oil and gas industry applications. The need for new, higher performance products, designed with state-of-the-art tools and materials, became evident.

In the late 1980s, Solar upgraded compressor designs and performance to meet the needs of pipeline customers; accordingly, Solar developed application specific compressor products for the pipeline industry for midstream applications. The C40P, the C45, and the C65 compressor model lines were developed in this endeavor.

In 2006, Solar announced the development of the C85, Solar's largest pipeline compressor. This compressor is specifically designed to match the *Titan* 250, Solar's largest gas turbine, rated at 30,000 hp or 22 MW. This compressor package will meet the needs of large diameter pipelines and long distance gas transportation systems.

Solar next focused on the development of specific gas compressors for production or upstream applications. The C40M, as well as the latest production compressor, the C51, are results of this endeavor. Building on our proven gas compressor product portfolio, Solar launched a major development initiative in 2004.

Solar's current emphasis is the development of new upstream-type compressors, such as the C61 compressor, which is based on the successful design and industry adoption of the C51 compressor. The C61 is an 18 percent geometric scale-up of the C51 and is well suited for *Mars*, *Titan* 130 and *Titan* 250 gas turbines. The C61 has a peak isentropic efficiency greater than 85%.

The C51D, a first in Solar's compressor product line, will offer four nozzles per casing, allowing for intercooling and side-streams that can be managed in a single casing, rather than requiring the traditional two-casing approach.

The ultimate goal of Solar's development initiative is to have a complete suite of gas compressor products for all applications in the power range of gas turbines Solar offers. Consequently, additional gas compressor products will be designed for specific oil and gas applications such as gas gathering and boosting, gas transmission, gas processing, storage, gas lift, and gas re-injection.

2.1 UPSTREAM

Applications at or near the oil and gas fields are referred to as upstream applications. Operations revolve around the gathering of gas from one or multiple wells, and subsequent compression to either feed it into pipelines, or to use the gas for increased oil recovery. The gas in upstream applications is generally minimally processed gas from gas wells or associated gas from oil wells.

Gas gathering may involve compression of a number of gas streams at various pressure levels from a number of gas wells. Suction pressures are often relatively low. The gas is then used for a number of purposes. It can be fed into a pipeline to be transported to a gas plant, to a liquefied natural gas (LNG) plant, or a pipeline grid. In offshore applications, this pipeline may operate at high pressures, and the gas may have to be boosted to this pressure. Compressors for this task are often called boost compressors, sales gas compressors, or export compressors.

The gas can also be used locally to enhance oil production. There are various techniques employed, gas injection being one of them. Gas injection maintains or increases the pressure in an oil reservoir by injecting gas into the reservoir, thus increasing the amount of oil that can be recovered from the reservoir. Gas lift injects gas into the well, thus aerating the crude for easier lift to the surface.

Production compressors are designed for high head per stage and high efficiency as shown in Table 2.1.

Table 2.1 Production compressor characteristics

English Units									
Compressor Frame Size	Maximum Number of Stages	Maximum Case Rating	Inlet Flow Range CFM X1000		Maximum Total Head ft-lbf/lbm X1000	Maximum Speed rpm	Maximum Weight	Suction Flange Size	Dischrge Flange Size
		psig	MIN	MAX		X1000	lb X1000	inches	inches
C16	10	3500	0.2	2.2	70.0	23.8	5.6	8	8
C33	12	2700	0.8	9.5	86.0	16.5	22.6	16	16
C40M	6	2500	1.0	9.5	80.0	14.3	26.6	16	16
C41 [1]	10	3750	1.5	18.0	90.0	14.3	40.4	20/20	16/14
C50	5	1500	2.0	20.0	80.0	12.5	42.5	24	24
C51 [1]	10	3000	2.0	25.0	90.0	12.0	53.9	24/20	20/16
C61 [1]	10	3000	2.8	35.0	90.0	10.2	83.3	30/24	24/20

SI Units									
Compressor Frame Size	Maximum Number of Stages	Case Pressure Rating	Inlet Flow Range m3/min		Maximum Total Head kJ/kg	Maximum Speed	Maximum Weight	Suction Flange Size	Discharge Flange Size
		kPag	MIN	MAX		rpmX1000	kg X100	mm	mm
C16	10	24132	4.2	62.3	209	23.8	2.5	203	203
C33	12	18616	22.7	269.0	257	16.5	10.3	406	406
C40M	6	17237	28.3	269.0	239	14.3	12.1	406	406
C41 [1]	10	25855	42.5	509.7	269	14.3	18.3	508/508	406/356
C50	5	10342	56.6	566.3	239	12.5	19.3	610	610
C51 [1]	10	20684	56.6	707.9	269	12.0	24.4	610/508	508/406
C61 [1]	10	20684	79.3	991.1	269	10.2	37.8	762/610	610/508

Notes:
1-Pressure Ratios are higher for the C41/C51/C61 Dual Compartment compressors depending on cooling capability.

Production compressors offer coverage over much more diverse operating conditions. Suction pressures, gas compositions, and heads vary between applications and require the compressors to be much more flexible in terms of stage count, speed, and power.

Solar production compressors can be used for single-body, geared or direct drive applications or in tandem (multi-body) with other compressors to meet a variety of duties and pressure ratios. Figure 2.4 shows *Solar* production family compressors and the corresponding *Solar* gas turbines for direct drive applications. Figure 2.5 shows applications of production compressors. Figure 2.6 shows an example of a triple-body tandem design and Figure 2.7 shows a high-pressure, double-body tandem arrangement. A single-body C51 compressor set is shown in figure 2.8.

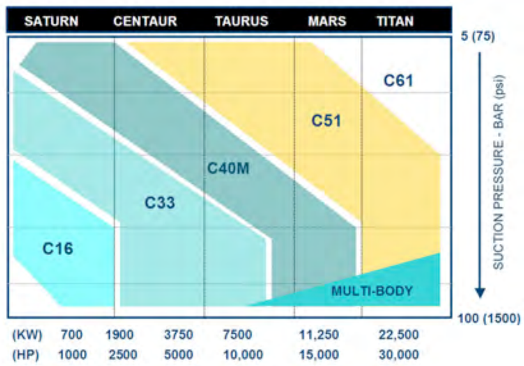

Figure 2.4 Solar production coverage

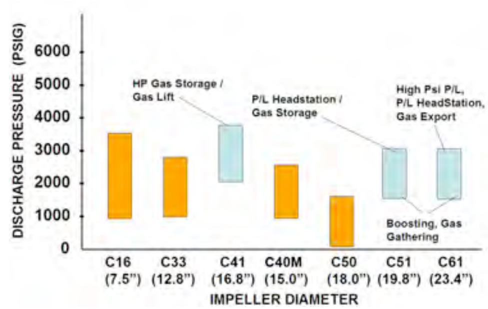

Figure 2.5 Upstream applications

Figure 2.6 Triple-body tandem compressor skid

Figure 2.7 Double-body tandem high-pressure compressors

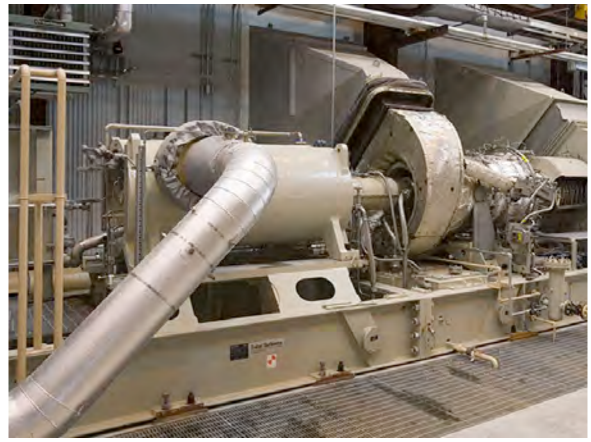

Figure 2.8 Single-body C51 compressor set

2.2 MIDSTREAM

Natural gas has to be transported, often over large distances, from the gas fields or LNG terminals to the ultimate users of the gas. To pump the gas through the pipeline, compression has to take place at regular intervals to compensate for the pressure drop in the pipeline due to friction. These pipeline compressors usually operate at relatively low pressure ratios, and are required to achieve wide operating ranges and a high efficiency. Gas pipeline systems may consist of long distance pipelines, as well as smaller distribution pipelines, and various sizes of interconnection facilities. Table 2.2 illustrates Solar's pipeline compressor characteristics.

Table 2.2 Pipeline compressor characteristics

English Units									
Compressor Frame Size	Maximum Number of Stages	Maximum Case Rating	Inlet Flow Range CFM X1000		Maximum Total Head	Maximum Speed	Maximum Weight	Suction Flange Size	Dischrge Flange Size
		psig	MIN	MAX	ft-lb$_f$/lb$_m$ X1000	rpmX1000	lbX1000	inches	inches
C40P	2	1600	1.5	11.0	36.0	15.5	26.9	20	20
C45	2 [1]	1600	3.8	17.0	36.0	12.0	60.0	24	24
C65	2	1600	5.0	24.0	36.0	10.5	76.0	30	30
C85	2	1600	10.0	45.0	36.0	7.0	170.0	42	42
SI Units									
Compressor Frame Size	Maximum Number of Stages	Case Pressure Rating	Inlet Flow Range m^3/min		Maximum Total Head	Maximum Speed	Maximum Weight	Suction Flange Size	Dischrge Flange Size
		kPag	MIN	MAX	kJ/kg	rpmX1000	kg X1000	mm	mm
C40P	2	11032	42.5	311.5	108	15.5	12.2	508	508
C45	2 [1]	11032	107.6	481.4	108	12.0	27.2	610	610
C65	2	11032	141.6	679.6	108	10.5	34.5	762	762
C85	2	11032	283.2	1274.3	108	7.0	77.1	1067	1067

<u>Notes:</u>

1- Three stage version is available for C45 compressor

Solar’s pipeline compressors are matched to the speeds and powers of their respective drivers and are optimized for typical pipeline conditions of head, flow, and operating pressures. Solar’s pipeline compressors are best-in-class due to their high efficiency and wide operating range with excellent durability and reliability. Solar currently offers four pipeline compressor models: C40P, C45, C65, and the C85. More than 530 of these units have been installed worldwide since their introduction in 1994.

The C40P is available in one or two stages and operates optimally with *Centaur*, *Taurus* 60 or *Taurus* 70 gas turbines. The C45 includes a three-stage option. It can be driven by *Taurus* 70, *Mars* or *Titan* 130 gas turbines. The C65 is available in one or two stages and is well matched to the *Mars* and *Titan* 130 gas turbines. The C85 is available in one or two stages and is well matched to Solar’s latest *Titan* 250 gas turbines.

Figure 2.9 shows Solar’s pipeline family compressors and the corresponding *Solar* gas turbines for direct drive applications. Pipeline sizes are shown on the vertical scale.

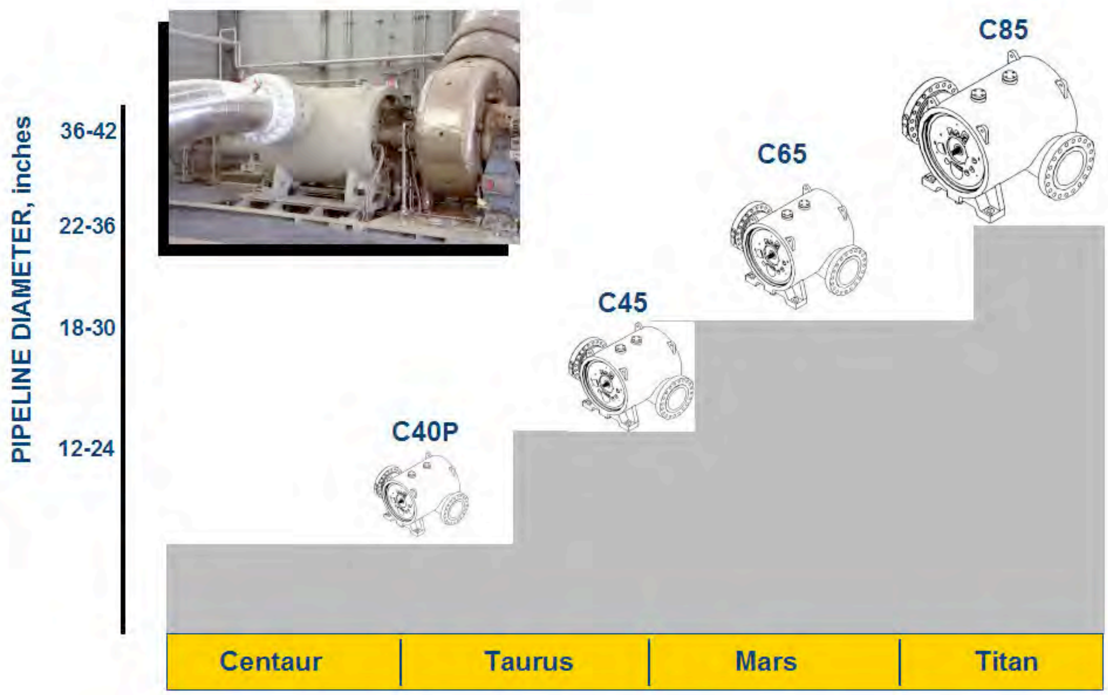

Figure 2.9 Solar’s pipeline compressor family

Before the gas is fed into the pipeline system, it often has to be processed in gas plants to remove potentially valuable heavier hydrocarbons such as ethane, or propane, as well as contaminants such as mercury. After processing, the gas may have to be compressed by a boost or sales gas compressor and fed into the pipeline system.

There are often compressor stations that bring the gas up to transmission pipeline pressure at the beginning of a pipeline. They receive gas from an oil/gas field or a gas plant. These stations are sometimes known as head stations.

Part of the operation of pipeline systems includes storage facilities, which are used to compensate for fluctuations in supply and demand. For storage operations, gas compressors compress gas from pipeline pressure to the higher pressures required in the storage facility. When gas is withdrawn from the facility and fed back into the pipeline, the gas remaining in the storage facility drops below pipeline pressure and has to be compressed again to bring it back to a pressure level higher than the pipeline. Gas plants, head stations, and storage facilities use multi-stage compressors in single or tandem arrangement to cover the required range of flow, pressure, and head requirements. Figure 2.10 shows typical gas compressor station layouts at various locations.

Figure 2.10 Typical gas compressor stations

3. Gas Turbine and Electric Motor Drives

3.0 OVERVIEW

Solar offers centrifugal gas compressors driven by gas turbines or electric motors**.** The gas turbine and electric motor drives that are described in this section are self-contained and completely integrated. Solar provides complete compressor solutions with both drives.

3.1 GAS TURBINE DRIVES

Solar gas turbines are manufactured to rigid industrial standards and are thoroughly tested in modern facilities. Solar's operations are currently certified by Det Norske Veritas (DNV) to conform to the International Standardization Organization (ISO) 9000 Series of Quality Systems Standards. Solar's smallest gas turbine, the *Saturn* 20, is shown in Figure 3.1 while Figure 3.2 illustrates Solar's largest gas turbine, the *Titan* 250.

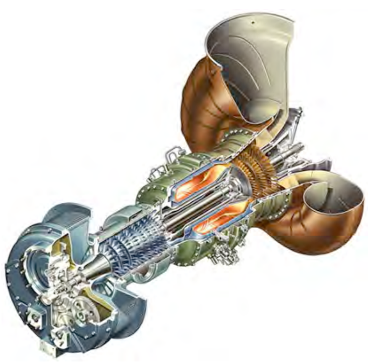

Figure 3.1 Saturn 20 gas turbine

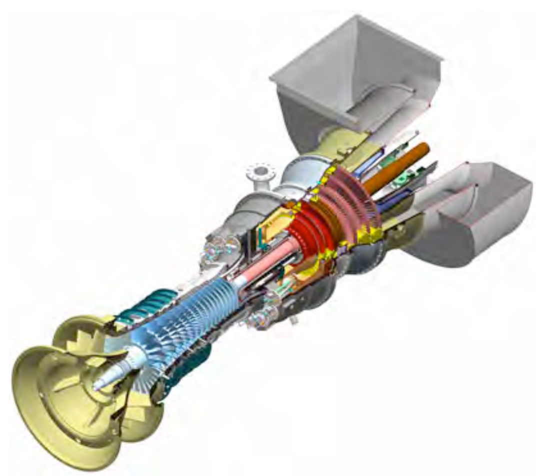

Figure 3.2 Titan 250 gas turbine

Solar designs its industrial gas turbines and gas compressors to be completely compatible. The gas turbines are a two-shaft configuration, which means the power turbine rotates independently of the gas producer; therefore, the driven equipment can be operated over a wide range of speeds and loads, which allows optimal adaptation to large changes in process conditions.

The continuous power cycle and rotary motion of a gas turbine provides several advantages over other types of engines, including relatively vibrationless operation, as well as fewer moving parts and fewer wear points.

Solar's gas turbines are designed around the fundamental principles of high efficiency, long life and low maintenance, ease of transport and installation, and high availability and reliability. This design philosophy combines the outstanding performance traits of an aero-derivative gas turbine with the construction and durability requirements of a long-life, rugged industrial machine. The gas turbine is also designed for a high degree of compliance with American Petroleum Institute (API) requirements.

The gas turbine incorporates Solar's advanced aerodynamic and mechanical technology and design. The structural concept of *Solar* gas turbines is unique in the engineering of gas turbines. With a few exceptions, contemporary machines have been designed to two extremes: either they are designed specifically to aircraft practices of highly sophisticated construction for light weight but short life, or they are designed with the massiveness of industrial steam turbines to ensure long life. In keeping with an optimum philosophy, the construction of Solar's gas turbines falls between the two extremes.

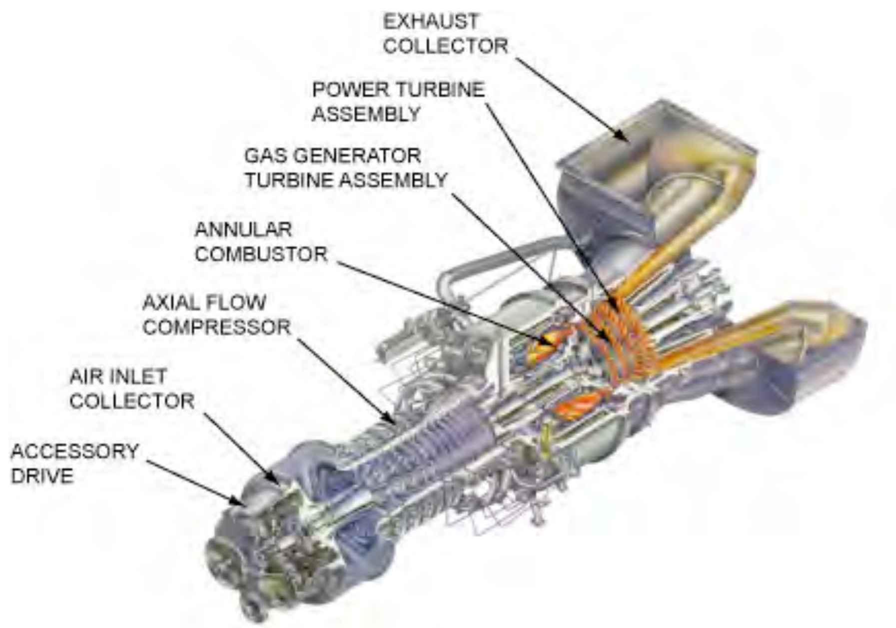

Figure 3.3 Typical components of a* Solar *gas turbine

One of the key design objectives of the gas turbine is operation at material temperatures and stress levels that provide maximum assurance of long life for the major rotating and stationary components. Another prime design objective is high availability and reliability. While many factors contribute to the dependability of the basic gas turbine, the selection of proper controls and gas turbine accessories is a major element.

The components of the gas turbine are maintained in accurate alignment by mating flanges, with pilot surfaces that are bolted together to form a rigid assembly. The accessory drive assembly, driven by the compressor rotor shaft, powers the main lube oil pump and other accessories, depending on the application. It also serves as the mounting point for the starter motor.

A vertically split compressor case design enables removal of either case half for greater access to internal compressor components for inspection, cleaning, or service. To further facilitate field service, the turbine incorporates numerous borescope access ports for inspection of internal flow path components.

With the exception of the *Saturn* 20, all of Solar's gas turbines are available with Solar's pollution prevention *SoLoNO$_x$*™ dry, low-emissions combustion system. Solar's experience with dry, lean-premixed combustion technology is unmatched by any other company in the gas turbine industry.

Solar's family of industrial gas turbine compressor sets includes:

Table 3.1 Solar gas turbine compressor sets

Turbine Model	Compressor Sets
Saturn® 20	C16, C33
Centaur® 40	C16, C33, C40M, C40P, C41, C50
Centaur 50	C16, C33, C40M, C40P, C41, C50
Taurus™ 60	C16, C33, C40M, C40P, C41, C50, C51
Taurus 70	C16, C33, C40M, C40P, C41, C45, C50, C51, C61
Mars® 90	C16, C33, C40M, C41, C45, C50, C51, C61, C65
Mars 100	C16, C33, C40M, C41, C45, C50, C51, C61, C65
Titan™ 130	C16, C33, C40M, C41, C45, C50, C51, C61, C65
Titan 250	C40M, C41, C51, C61, C85

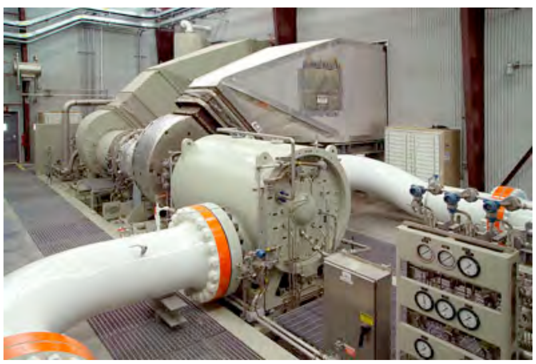

Figure 3.4 Mars 100 compressor set

Figure 3.5 Centaur 50 compressor set

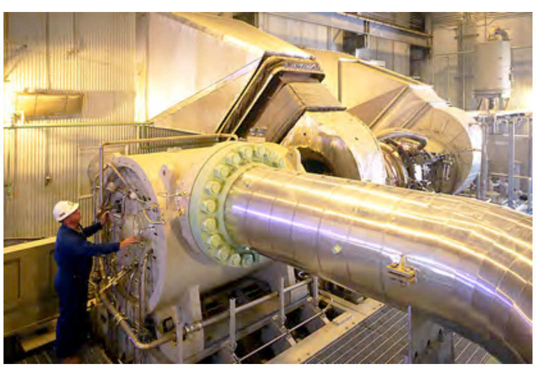

Figure 3.6 Titan 130 compressor set

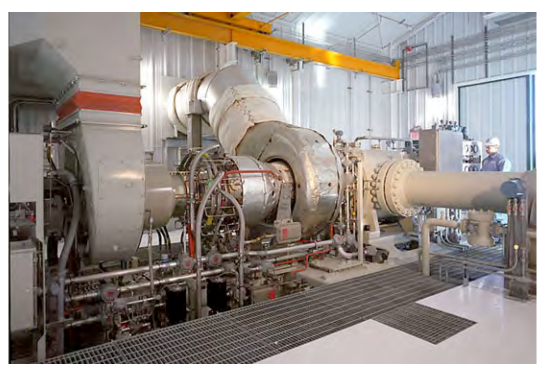

Figure 3.7 Taurus 60 compressor set

In the ongoing process of product improvements, Solar often will enhance the available performance of its gas turbines; therefore, for the latest performance on Solar's entire range of equipment, please visit our website at www.solarturbines.com or contact the sales office nearest you.

3.2 ELECTRIC MOTOR DRIVES

Solar provides electric motor drive (EMD) packages for compressors up to 40,000 hp. The availability of low cost, reliable electric power or regulatory requirements can make EMDs attractive compressor drive solutions.

There are three basic EMD types. Each is constructed differently and has a different scope of supply, a distinct cost structure, and its own unique advantages and disadvantages. The three types, used in combination with either synchronous or induction motors, offer potential solutions for most EMD applications. Unlike gas turbines, EMDs are not as flexible when it comes to off-design compressor conditions. To successfully couple centrifugal compressors and electric motors, it is critical to understand the characteristics of the different EMD configurations, especially their speed/torque characteristics, their starting characteristics, and their torsional harmonics.

The three main types of EMD for centrifugal gas compressors are:

- Variable frequency drive/electric motor (VFD-EMD)
- Variable speed hydraulic drive/electric motor (VSHD-EMD)
- Constant speed motor (CSM-EMD)

Variable Frequency Drive-Electric Motor Drive (VFD-EMD)

VFD-EMDs are attractive driver selections for centrifugal compressors. The variable speed and soft start characteristics allow for operational flexibility due to process variations and needs, and they also allow for reduced current draw and process impact during start up. The ability to control motor output torque during unit start-up makes a VFD-EMD more advantageous for the utility grid as well as for the compressor train compared to an EMD application that uses across-the-line motor starting.

There are several types of drive technologies developed by different manufacturers. The most common are load commutated inverters (LCI) and pulse width modulated (PWM) drives. Both synchronous and induction motors can be used with VFD controlled EMDs; however, beyond a certain horsepower range, only synchronous motors are cost effective and efficient.

A VFD eliminates the inrush current to the motor during starting, but it generates harmonics on the AC (alternating current) line that can affect the utility grid and the drive train. One method to reduce total harmonic distortion (THD) is to increase the number of pulses of the drive output waveform. As the VFD pulse rate increases, the power curve becomes closer to a true sine wave and the torsional and harmonic content decreases. In addition, VFD drives induce torque pulsation, which is seen at the motor air gap. The magnitude of the torque pulsation must be carefully examined as it can create severe torsional loads on the entire train. In order to minimize this risk, a variable frequency drive with very low torque pulsation is normally recommended to customers.

Variable Speed Hydraulic Drive-Electric Motor Drive (VSHD-EMD)

VSHD-EMDs have become popular over the last few years. A VSHD, sometimes referred to by the manufacturer's name, Voith, uses a mechanical means to vary the speed of the compressor. The VSHD is essentially a torque converter integrated with a gearbox and located between the motor and compressor.

VSHD-EMDs typically consist of a planetary or epicyclical gearbox with a hydraulic motor/turbine driven rotating planetary gear assembly that can vary the effective gearbox ratio depending on the speed and direction of rotation of the planetary gear assembly. Items that need to be addressed when selecting VSHD-EMDs are electric motor sizing, gearbox selection, process variations, VSHD cooler size, and power margin or service factor.

The motor and VSHD are mounted on a common driver skid. The biggest challenge with these drives is motor starting. Although the VSHD can vary speed and, in some cases start without driving the compressor, there is no provision for soft-starting the motor. This is a particular problem in sizes above 4,500 hp where utility companies typically do not allow across-the-line motor starting because of the large inrush currents generated by the motor. In one workaround method, a small starter motor (a pony motor) is mounted to one end of a double-shafted main motor. The starter motor drives the main motor up to speed before the main motor is engaged, thus eliminating most of the inrush current. However, this method adds cost and increases the scope of supply to include a start motor. There are other starting methods available such as autotransformer assisted, reactor assisted, and capacitor assisted starts. Another method involves the use of a hydrodynamic coupling (or hydraulic clutch) which can be drained during start-up, enabling the motor to start against very small inertia. However, in order to ascertain what system is best suited for the customer and the utility, a detailed starting study of the power system will be required.

Constant Speed Motor (CSM-EMD)

In this configuration, the motor is started either across the line directly from the utility or with some type of soft start mechanism. Starting of the motor needs to be carefully considered. The package includes the compressor, a gearbox, a lube oil management system, and a synchronous or induction motor. The standard configuration consists of a driver and driven skid. The motor, gearbox, and lube oil system are integrated into the driver skid and the compressor is mounted on a Solar standard-driven skid. If the motor is started across the line, inrush current and torque are typically very high, and special consideration is needed from the customer's electric utility supplier.

Constant speed EMDs often appear to be attractive, low-cost drive selections for centrifugal compressors, but the reduced operating range efficiencies compared to other EMDs and gas turbine drivers, compression train startup issues, and additional scope of supply often make them a less than desirable solution. Constant speed drives are best suited to low power applications where the compressor operating conditions do not vary from the design point. The main control mode for process control and load sharing with a centrifugal compressor is usually speed control. Since speed control is not available with a constant speed motor drive, another method of process control, load control, and load sharing is required. This is often done with a suction throttle valve. The additional pressure drop of the suction throttle valve, even when wide open, affects the overall efficiency and must be accounted for in the compressor operating conditions and motor selection.

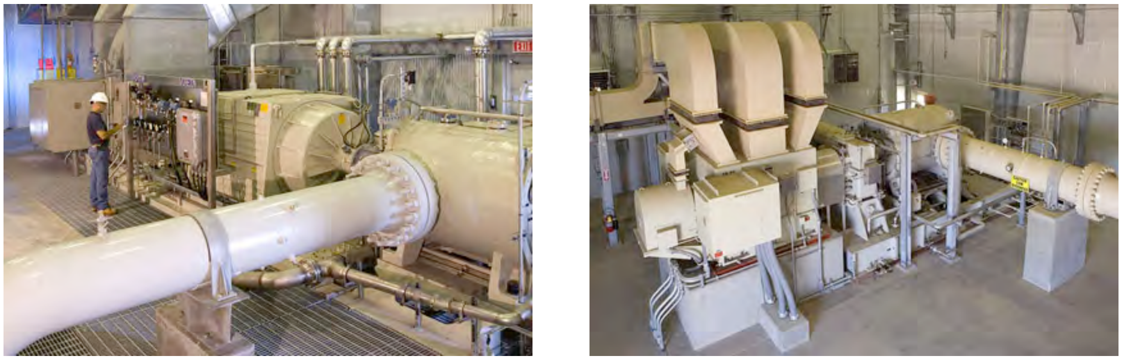

Figure 3.8 Typical electric motor-driven compressor sets

4. Centrifugal Gas Compressor Theory

All centrifugal compressors have one or more stages. Each stage consists of vanes, an impeller, and a diffuser. Vanes are designed to feed the gas flow uniformly into the impeller. A first stage has inlet vanes; following stages have return vanes. The impeller, the only rotating part of the stage, transfers the mechanical work fed into the compressor to the gas, and in this process increases the pressure, the temperature and the velocity of the gas. Because the desired outcome of all this work is a pressure increase, the diffuser converts most of the velocity, or kinetic energy, of the gas into pressure. The last diffuser in a compressor often feeds into a volute, which can be thought of as a diffuser in the form of a scroll, and it also converts kinetic energy of the gas into pressure. A single-stage compressor cross section is shown in Figure 4.1.

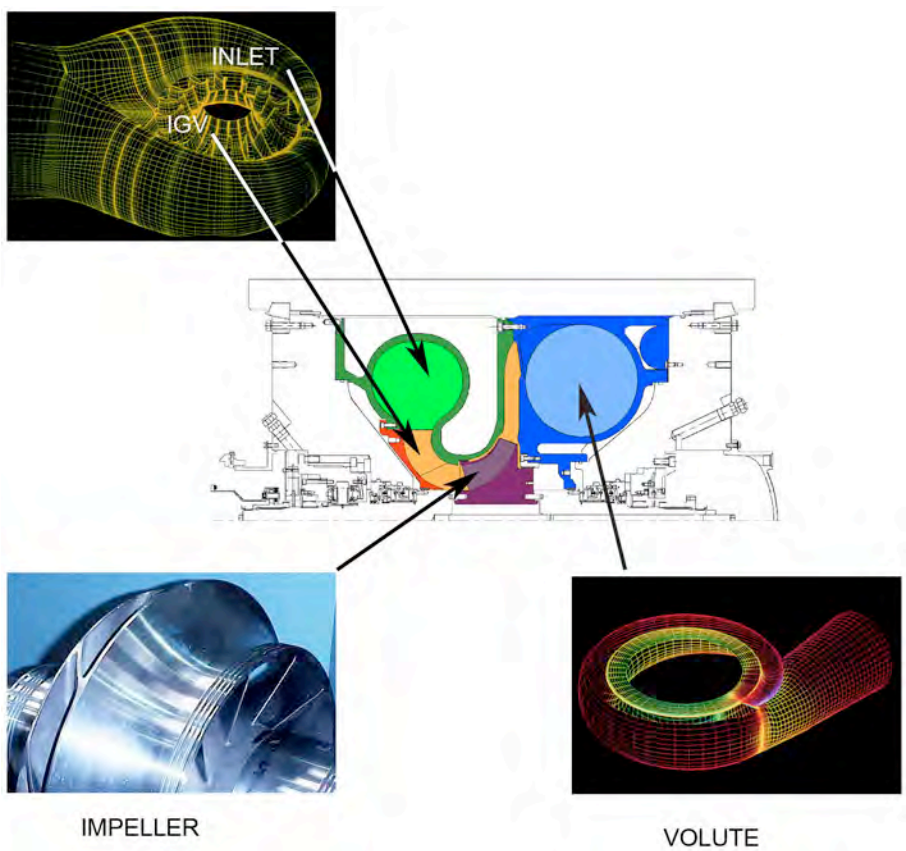

Figure 4.1 Compressor cross section

The energy content of the gas is expressed as its enthalpy. As shown in the Mollier diagram (Figure 4.2), all the mechanical work that is provided at the compressor shaft, except for a small amount lost to friction in the bearings and seals, is converted into an increase in enthalpy of the gas. The increased enthalpy manifests itself as an increase in temperature and pressure of the gas. If the composition of the gas is known, thermodynamic state variables such as pressure, density, temperature, enthalpy, entropy, etc., define the state of the gas as soon as two of the variables are known. Knowing the pressure and temperature of the gas at the compressor inlet fully defines the enthalpy and entropy, and defines the density of the gas at the inlet of the compressor.

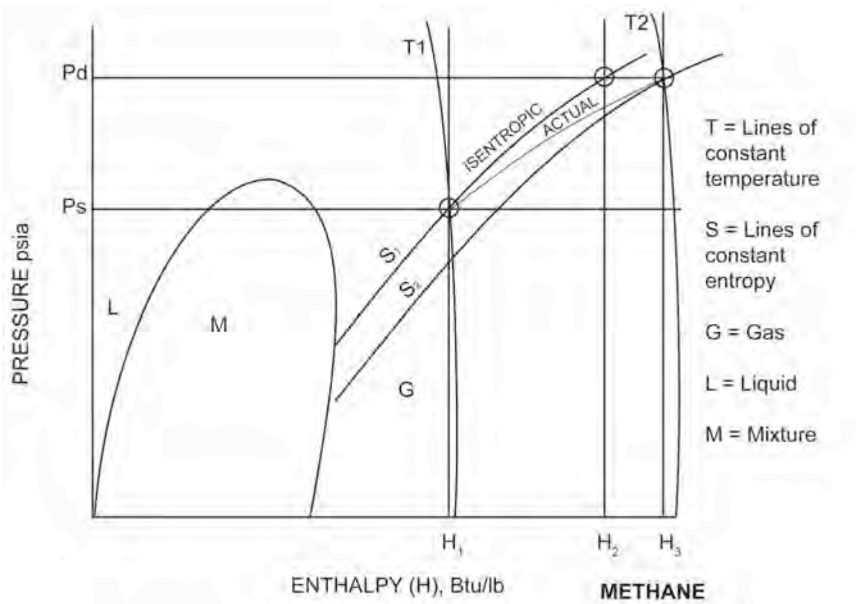

Figure 4.2 Mollier diagram

When testing a compressor, the pressure and temperature, and thus the enthalpy, are known at its inlet and exit. The difference between the enthalpy at the compressor exit and the enthalpy at compressor inlet is the actual head. Note that in customary units, the enthalpy difference is multiplied with a constant, the mechanical equivalent of heat, to adjust the units of measurement. When a compressor is designed, the discharge temperature is not initially known. Thus, a second state variable is needed in addition to the discharge pressure in order to define the exit condition of the gas. One way to do this is to assume an ideal isentropic compression process that assumes no losses are generated and no heat is exchanged with the surroundings. In such a process, the entropy at the discharge is the same as at the inlet, so the state of the gas can be defined at the exit by its pressure and its entropy. The difference between the enthalpy at this isentropic exit state and the enthalpy at the inlet, again, multiplied by the mechanical equivalent of heat, if customary units are used, is the isentropic head. As more is learned about the compressor during the design phase, its isentropic efficiency will be known, and the actual head can be determined from the isentropic head by dividing the latter by the isentropic efficiency.

When the compressor is tested, the pressures and temperatures at both the inlet and the exit can be measured; therefore, both the actual head and the isentropic head can be calculated. The ratio of isentropic head over actual head is the isentropic efficiency. A high isentropic efficiency means that the compressor performs almost as well as the 'ideal', isentropic machine. For given process conditions, that is, a defined isentropic head, the compressor with the best isentropic efficiency will produce the lowest discharge temperature and consume the least power.

Mollier diagrams are usually only available for pure gases, but the gas compressor usually deals with gas mixtures; therefore, equations of state are used to calculate the relationships between pressure, temperature, density, enthalpy and the dew point temperature of gas mixtures. These equations are based on intermolecular gas behavior and empirical data, and can be considered semi-empirical. Solar uses the Redlich-Kwong equation of state for all calculations, and has a large body of data that indicates this method is accurate. Because the equation of state also allows calculation of the gas density, it is used to determine the actual flow into the compressor at inlet conditions based on standard flow, normal flow, or mass flow data.

It is important to know that any process condition can be converted into the actual flow and isentropic head. As shown in Figure 4.3, the performance of a centrifugal compressor is best displayed in a head versus flow map, which also conveniently includes lines of constant efficiency. This map is invariant to all but the most extreme changes in process parameters, and can thus be used to evaluate the compressor performance over a large range of conditions.

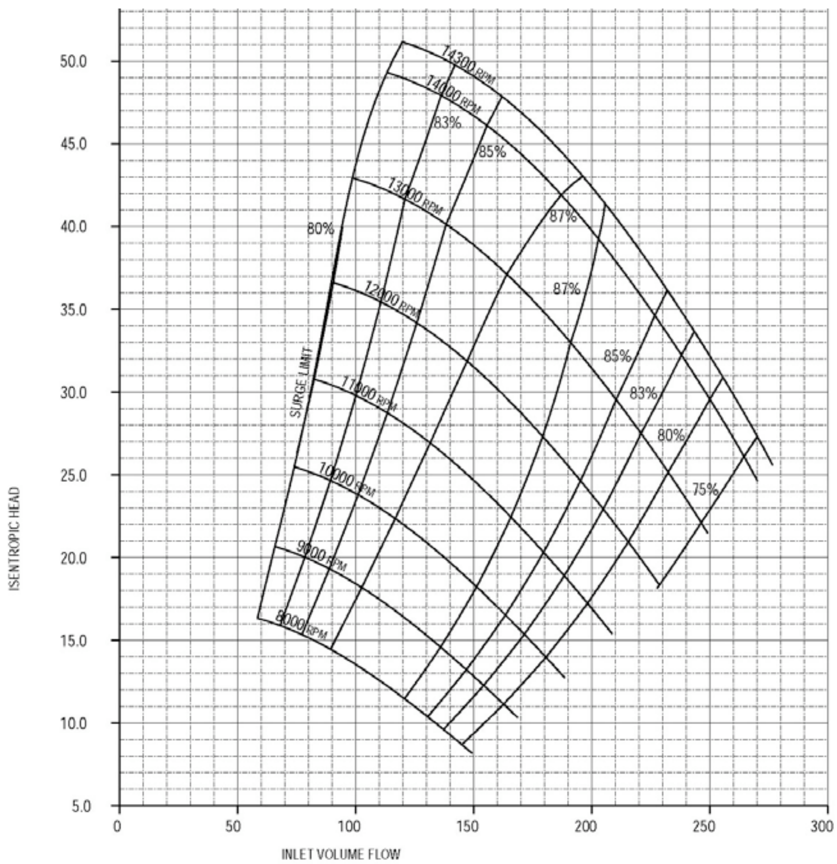

__Figure 4.3 Compressor map__

5. Performance, Prediction, and Testing

5.0 OVERVIEW

Because Solar supplies a standardized product, compressor stage performance is repeatedly verified. To verify the predicted performance, all of Solar's gas compressors are factory tested before shipment, so not only is the compressor's performance verified, but the prediction model is verified as well. The prediction can be verified in the aero test facility (ATF), in the open-loop test facility, in the closed-loop test facility, and can also be verified in a field testing process. The aerodynamic test results are added to the database used by the prediction model. As a result of this continuous capturing of new information, there is very little difference between the predicted performance and the actual performance of *Solar* compressors.

5.1 AERODYNAMIC STAGING

Each *Solar* compressor model is based on a standard, pre-engineered family of stages. Each stage consists of an inlet or return vane, an impeller, and a matched diffuser. A complete family of stages is developed at the introduction of a compressor model. A typical family of aerodynamic stages consists of at least 15 separate aerodynamic combinations of matched inlet guide vanes, impellers, and diffusers. The volutes are matched to the rear stages. The combination of pre-engineered impeller geometry and matched inlet guide vane geometry results in a wide range of inlet flow capability for each compressor model. The head-making capability of the compressor depends upon the number of stages and the compressor speed.

Application of the compressor consists of selecting the number of stages with their known characteristics that best meets the conditions and best matches the performance of the required driver.

Impeller diameters for each compressor family are selected to provide the best range of specific speeds, given the intended drivers and applications. The fundamental geometry of the compressor stages is conveniently expressed in terms of specific speed and is analytically expressed:

$$N_s = N\frac{Q^{1/2}}{H^{3/4}}$$

Where N_s= specific speed

N=compressor speed (rpm)

Q=inlet flow (ACFS)

H=head (isentropic, ft $-lb_f$ / lb_m)

The specific speed determines the general shape of an impeller. Low specific speeds identify impellers for lower flows. Reduction in efficiency at low specific speeds results from the relatively higher frictional losses in the narrow passages of the impeller and stationary parts. High specific speeds identify impellers for higher flows. The impeller cross section (Figure 5.1) shows the different flow path among the various specific speeds (higher to lower specific speeds shown from left to right). Most of the impellers have a radial exit pattern for the gas they process. The impeller (shaded in red lines) is a mixed flow impeller, with the flow leaving the impeller in a partially axial direction. On the backside, the labyrinth seals can be seen which are on the cylindrical surface of the balance drum. The diameter of the balance drum is chosen so that the axial thrust is balanced.

Figure 5.2 indicates a certain range of specific speeds where good impeller efficiency and flow range can be accomplished.

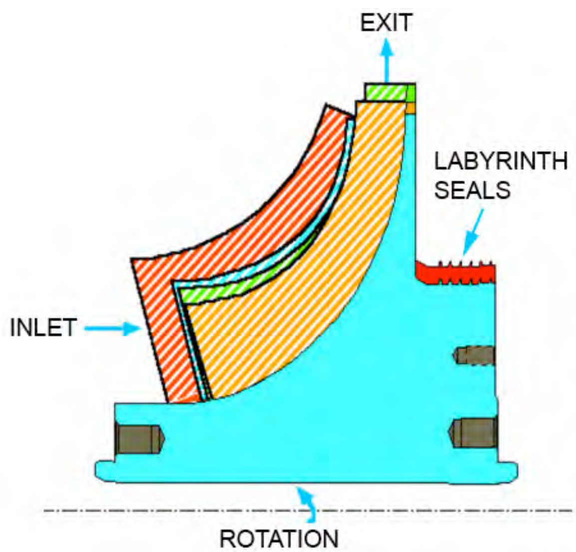

Figure 5.1 Pre-engineered impellers with different specific speeds

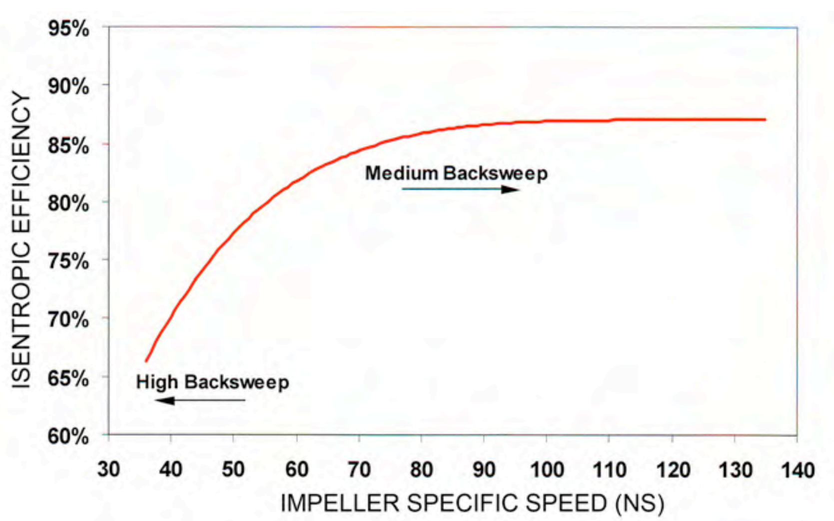

Figure 5.2 Effects of specific speed on efficiency levels

5.2 PERFORMANCE PREDICTIONS

Performance calculations for a centrifugal compressor require information about the gas that will be compressed and ideally, the gas composition is known. For natural gas, the gas behavior can be estimated by just knowing the specific gravity, but the gas temperature must always be known.

Performance calculations require data for at least three of the following variables:

- Suction pressure
- Discharge pressure
- Standard volumetric flow rate
- Compressor power consumption

Initial performance predictions are based on the mathematical model of head, flow, and efficiency characteristics of the individual stages. These characteristics are derived from tests in the ATF or the closed-loop facility, and the results are compared to the initial prediction.

Operating a compressor effectively for a given application requires, among other things, an understanding of its performance curve(s). There are many different formats to graphically show the expected performance of a compressor; however, the isentropic head versus capacity curve shown in Figure 5.3 is the most useful. It is drawn on coordinates of isentropic head and actual inlet volumetric flow rate with lines of constant speed and isentropic efficiency. A single line shows the approximate location of the surge limit.

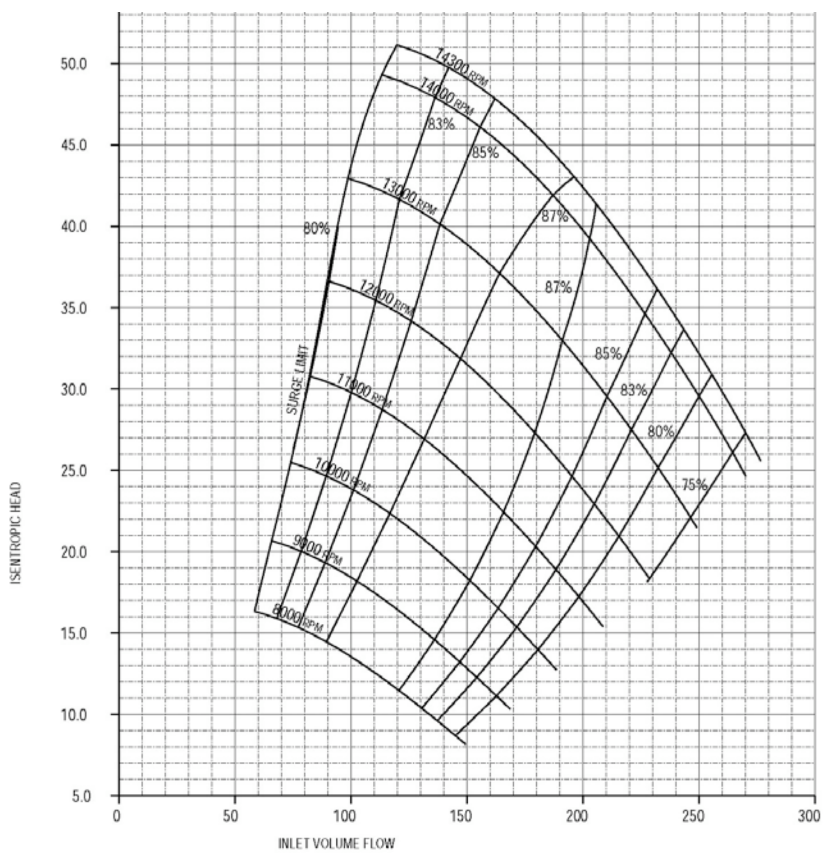

Figure 5.3 Typical head vs. capacity curve

Performance predictions can be made more accurately by the use of software, which Solar can provide as part of the computer that operates the compressor package.

5.3 FACTORY COMPRESSOR TESTING

Solar’s test facilities are designed for flexibility; all types of turbine packages can be tested with a minimum of time required for connection of fluid, electrical and instrumentation services needed for each test run. The computerized, real-time data acquisition system collects raw digital and analog data from the package and displays as well as prints out results in customary engineering formats with appropriate units of measure.

The control and display equipment shown in Figure 5.4 controls and monitors the power and test stimuli to operate the unit being tested and measures and evaluates its performance. The system establishes specified test conditions, constants, and operating limits. Test data are monitored by video displays as instructed by the test agenda, selecting various parameters for display, checking values and limits, and generating reports as needed. When acceptable performance levels are achieved, the test technician initiates a command to capture all instrumented points, which initiates automatic performance computations and prints the results for review for a permanent test record.

Figure 5.4 Compressor test facility control room

The gas compressor is tested in accordance with Solar's specifications and as generally outlined below. Prior to assembly of the internal components, all compressor casings receive a hydrostatic pressure test.

Hydrostatic pressure testing of all compressor casings and endcaps is conducted per API 617 for 30 minutes at 1.5 times the maximum casing design pressure, regardless of application, to check the integrity of the compressor housing and its internal passages. After the hydro test and final magnetic particle test, the casing is steam cleaned and bead blasted for surface preparation and painting. A nitrogen pressure test of the assembled compressor is performed to verify there are no gas leaks after it completes stringent and comprehensive mechanical performance testing.

As shown in Figure 5.5, every gas compressor that Solar builds is tested at our open-loop facility. Compressors are tested at ambient conditions to ensure that they meet strict performance guidelines for head, surge margin, and vibration.

Figure 5.5 Open-loop test facility

The mechanical testing is performed first. The dry gas seals are tested statically with nitrogen. After preliminary checks and static seal testing, the unit is operated at break-in speed, then at maximum continuous speed. Key mechanical parameters such as seal gas leakage and separation seal air flow, oil flow, and vibration levels are measured and evaluated against established limits.

After completion of the mechanical tests, aerodynamic performance testing on air is conducted. The primary objective of the test is to confirm the accuracy of the individual stage characteristics used for predicting compressor aerodynamic performance at the air-equivalent design speed. Accuracy is determined by comparing the overall head/flow speed line from choke to surge and the surge line position against prediction when operating at a speed equivalent to the site design speed. Surge points are determined at various speed points to validate the surge flow estimate for the entire operating speed range. Extensive instrumentation, together with the facility data acquisition and reduction system, validates mechanical and aerodynamic performance.

Testing is conducted on dedicated test stands using a facility driver. The suction and discharge nozzles are connected to an open-loop configuration that uses atmospheric air. The test evaluates mechanical and aerodynamic performance in accordance with test procedures and acceptance criteria as outlined in applicable test specifications.

A significant amount of development and production compressor testing has been done at Solar's closed-loop test facilities (Figure 5.6) where ASME PTC-10 Type I and II testing is conducted. The data gathered during these tests is fed into the prediction database to improve analytical models and improve compressor durability and reliability.

Figure 5.6 Closed-loop test facility

Aero Test Facility

The aero test facility (ATF), shown in Figure 5.7, allows for testing of scaled single stages while making detailed measurements in order to improve prediction capability and help understand component performance.

Figure 5.7 Aero test facility

In this facility, exclusively dedicated to development testing, Solar can test entire compressor stage impellers, diffusers, return vanes, and volutes. This process extends our aerodynamic design knowledge.

The objectives of all the ATF tests are to:

- Validate and establish limits of design codes (1-D, 3-D)
- Evaluate and understand stage component losses (impeller, return vane, etc.)
- Establish guidelines for aerodynamic parameters
- Develop and improve correlation
- Gather data to help design the next generation of compressors

All of which will enable Solar to:

- Design compressors with optimal performances that meet or exceed industry standards
- Improve performance prediction

Many different types of compressor stage components can be tested in the ATF. These stages can either be scales of existing production stages or experimental stages. For each test article, flange-to-flange performance is determined. Static and total pressures are measured at various locations throughout the stage to determine overall stage performance as well as the performance of each individual component.

The objective of the facility is not only to design and to test stages with the best efficiency for direct application to a production stage, but the facility also extends Solar's knowledge of critical parameters. A stage can therefore be designed deliberately with a sub-optimal surge margin and, by analyzing test data and predicted data, correlations can be derived as to what critical parameters, such as diffusion or flow angle, are responsible for the adverse effect on component stability. This extends the prediction capabilities of the design software and improves the ability to design stages with the best compromise between head, efficiency and range.

5.4 NEW GAS COMPRESSOR TEST FACILITY

Continued investment in development test facilities lies at the center of Solar's new product development efforts, which enables us to offer proven products to the marketplace. Shown in Figure 5.8, the gas compressor test facility (GCTF), which began operation in 2009, marked a substantial investment in new product development.

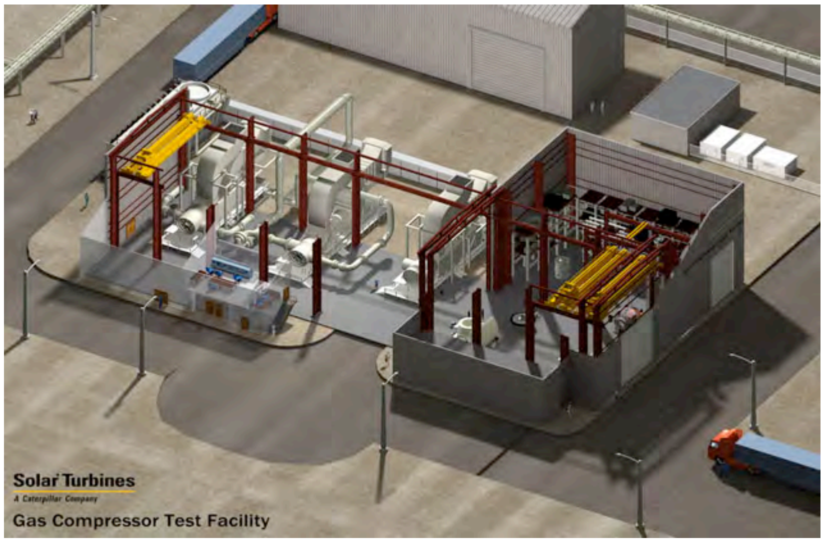

Figure 5.8 New gas compressor test facility

This world-class, state-of-the-art facility enables Solar to perform research, development testing, and production testing of compressors in accordance with ASME PTC 10 Type I and II, and uses natural gas at full load and full pressure condition with the contract drivers. The facility features three test bays; in the first two bays, *Mars* and *Titan* slave drivers equipped with universal driven-equipment skids will enable full load testing in a hydrocarbon or inert gas closed loop, initially up to 20,000 kPa (2900 psig). The third test bay will accommodate full package string testing and closed-loop testing up to 30MW (40,000 hp). This facility is available to test gas compressors after overhaul or restage.

Compressors can be tested under load to reproduce the design point and to accurately measure aerodynamic performance. Tests results are fed into the prediction database to improve analytical models, and improve compressor durability and reliability. In the GCTF Solar can:

- Test compressors under load
- Measure aerodynamic performance
- Update analytical models
- Determine rotordynamic performance
- Ensure trouble-free operation

The ASME PTC 10 Type II test uses an inert gas, such as CO_2, nitrogen or nitrogen-helium mixtures as the test gas. The PTC 10 code allows defining compressor operating conditions that are aerodynamically similar to the operating point in the field. The compressor is tested at the same flow and head coefficient, and the same ratio between inlet and discharge density as the field operating point while also satisfying the design machine Mach Number and Reynolds Number within certain tolerances. ASME PTC-10 Type I testing uses the same gas properties and the same operating conditions that the unit will experience in the field.

5.5 FIELD TESTING

Field testing of compressor packages driven by gas turbines or electric motors has become increasingly frequent because economic pressures demand that efficiency, capacity, head, power, and fuel flow be verified and continuously monitored to assure a project's return on investment. Test results may have significant financial implications for the compressor and gas turbine manufacturers and their customers. The results may be the basis of future decisions on plant modifications or extensions, or may serve as baseline data for monitoring purposes. Field tests can provide users with valuable operational and maintenance data and can also provide equipment manufacturers with information complementary to the data collected during factory testing.

The customer and Solar agree upon and document the parameters of interest, as well as the criteria (minimums and maximums) for acceptance. These parameters include compressor efficiency, compressor flow, compressor consumed power, but also gas turbine power and heat rate or fuel consumption.

Field testing has been made easier by widespread application of portable computers (shown in Figure 5.9), which have introduced powerful analytical tools directly to remote locations where standard digital controls measure and display all necessary parameters. Considerable numbers of customers choose field testing rather than the ASME PTC-10 closed-loop testing conducted in the factory. Field testing is much less expensive than factory testing and will not delay the delivery of the equipment to site.

Figure 5.9 Field test tools

The success of Solar's compressors in factory closed-loop tests and field tests proves the very close correlation between predicted performance and actually measured performance, both at design operating conditions and at off-design operating conditions. The vast data base of tested impellers and impeller combinations, in conjunction with Solar's pre-engineered aerodynamic components, is seen by many customers as a solid assurance of meeting performance; therefore, these customers need not specify factory closed-loop test on their machines.

Solar is proud that the proposed compressor is the compressor that is delivered. Solar achieves this through standardized, pre-engineered design and advanced design tools that repeatedly test those designs, and by continuously updating and fine-tuning the predictive models.

6. Compressor Assembly-Disassembly

6.0 OVERVIEW

A key advantage of the modular rotor is that it can be easily disassembled and reassembled without damaging the shaft or the impeller, unlike traditional shrink-fit, solid-shaft rotor construction. This is a very important feature if an impeller is damaged during operation and needs to be replaced, or operating conditions change and the compressor needs to be restaged. Figure 6.1 shows an assembled rotor ready to be installed in a compressor. All parts are manufactured for interchangeability so that seals, bearings, and complete stages can be replaced without any special fitting or shimming.

Figure 6.1 Assembled modular rotor

The *Solar* compressor modular rotor consists of stub shafts, impellers, and, if required, rotor spacers to maintain a constant bearing span, and a tie-bolt as shown in Figure 1.11 in Chapter 1. These components are individually balanced prior to assembly, and they are then assembled into a single, modular rotor, which is then degaussed and balanced again as a complete unit. The impellers, stub shafts, and spacers are rabbet-fit to each other for concentric alignment. The entire rotor is clamped together with the tie-bolt. Torque is transmitted through the interference fits, clamp faces, and dowel pins installed across the faces of the components.

The benefits from this type of construction are twofold. Impellers that can be reused in a "restaged" rotor are easily salvaged and downtime is minimized. Reusing old impellers that will match new operating conditions enhances the economic feasibility of restaging and maintains optimum compressor performance at the lowest possible operating costs.

Figure 6.2 Inspecting impellers during restage at Solar's Mabank facility

6.1 ASSEMBLY PROCEDURE

As shown in Figure 6.3, rotors are assembled in a build pit. Figure 6.4 illustrates how once assembled, the rotors are degaussed, and then the entire rotor undergoes a balancing operation.

Figure 6.3 Rotor assembly in build pit

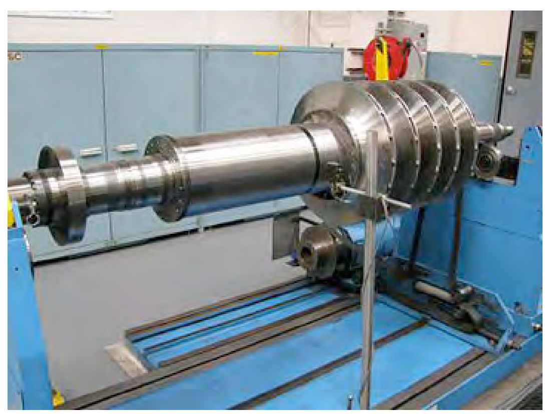

Figure 6.4 Rotor balancing

Figure 6.5 shows final rotor-check operations, including verification of mechanical run-out, electro-mechanical run-out, and other dimensional checks and verifications.

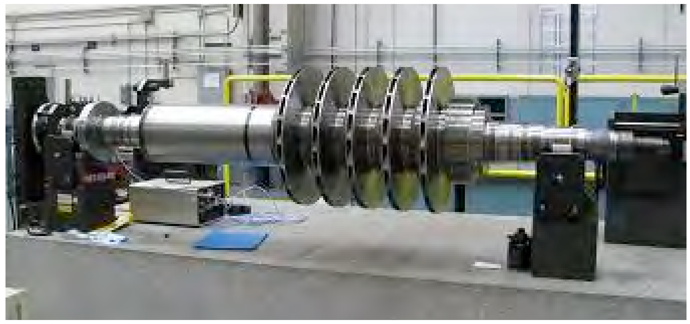

Figure 6.5 Electro-mechanical run-out checks

After the rotor has been balanced, it is disassembled so that it can be reassembled with the stator/diffusers (turn-vane/diaphragm assemblies), stacking each stage in sequence (Figures 6.6 through 6.12). This stacked assembly is known as a module. The modular rotor allows the use of solid stator/diffusers.

As shown in Figure 6.6, the rotor is first installed in the inlet portion of the module. In Figure 6.7, the rotor is then separated down to the first-stage impeller using separation tools, in preparation for module stack.

Figure 6.6 Installation of rotor into inlet system

Figure 6.7 Separation of rotor

The first stage stator/diffuser is installed. As shown in Figure 6.8, the first stage impeller is visible during installation of first stage diffuser. Figures 6.8 through 6.10 show how the procedure of sequentially stacking impellers and stators is repeated until all the stages are installed.

Figure 6.8 Stage 1 diffuser installation

Figure 6.9 Stage 4 diffuser installed

Figure 6.10 Discharge volute installation

Figure 6.11 Completed module assembly

Following volute installation (Figure 6.10), the completed module (Figure 6.11) is ready for installation into the compressor case (Figures 6.12 and 6.13) using specialized handling equipment.

Figure 6.12 Vertical module installation

Figure 6.13 Module installed

6.2 DISASSEMBLY PROCEDURE

Figure 6.14 demonstrates removal of a compressor module in the field.

Figure 6.14 C40P module being removed for restage

The module can be installed in the compressor housing either vertically or horizontally. Due to the availability of factory tooling and fixtures, the module is usually installed vertically in the factory while the module is usually installed horizontally at the customer's site. Figure 6.15 shows the ease of removing the module horizontally.

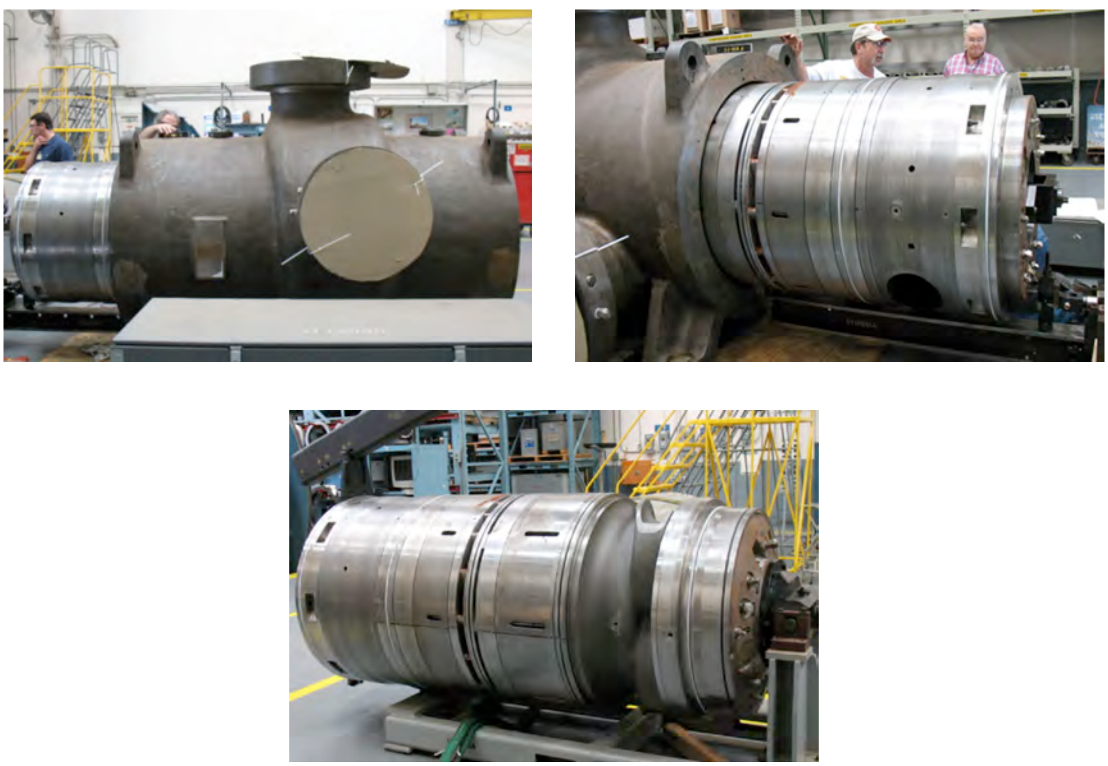

Figure 6.15 Horizontal module removal

Figure 6.16 shows the phases of a module removal and installation during an actual on-site module exchange. The entire process can be completed in one shift.

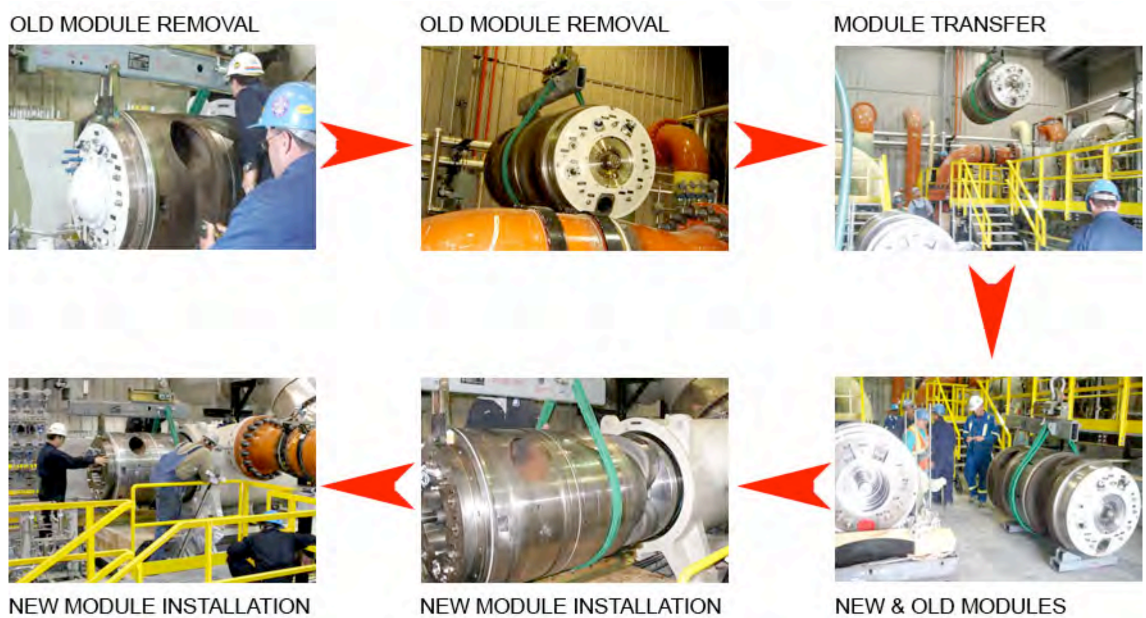

Figure 6.17 Module exchange on site

7. Compressor Sub-Assemblies, Components and Materials

7.0 OVERVIEW

All *Solar* compressors are composed of the module (sometimes referred to as the bundle), the casing, end caps, bearings, seals, and miscellaneous assembly hardware.

The module is composed of the rotor, stators, inlet housing and discharge volute. The module is the heart of a *Solar* compressor and as such, it determines the performance of the compressor. There are as many lengths of modules as there are bearing spans within housings lengths.

The modular rotor assembly consists of stub shafts, impellers, and, rotor spacers (if required to maintain a constant bearing span) and a tie-bolt, sometimes known as a center bolt. Figure 7.1 depicts some of the compressor components.

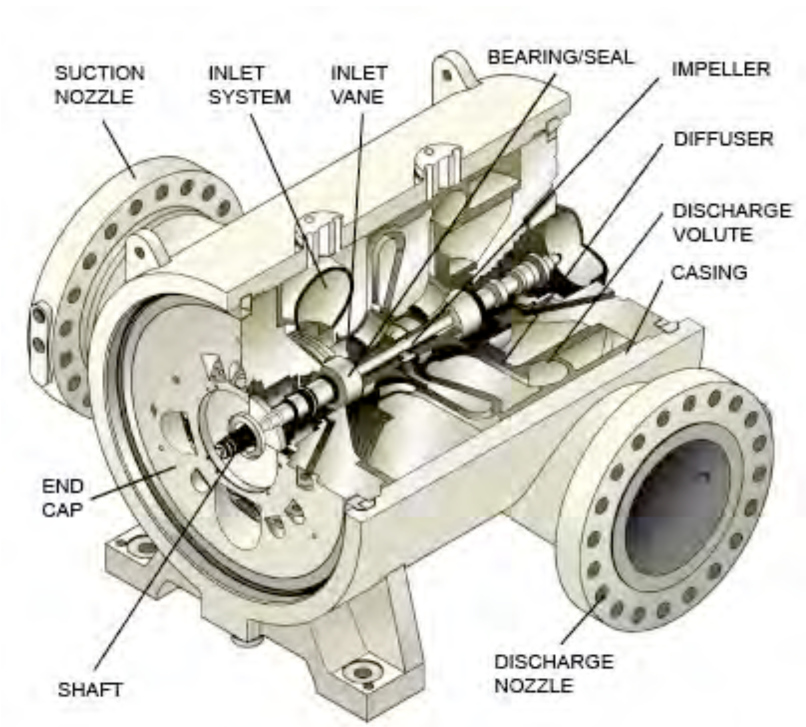

Figure 7.1 Gas compressor components

Solar compressors typically have:

- Cast steel centerbodies, heat treated to meet API 617 and ASME Section VIII codes
- Investment cast, machined, or brazed impellers (NACE compliant 15-5PH steel)
- Cast ductile iron inlet housing and exit volutes
- Steel stubshafts
- Cast nodular iron or fabricated steel vane and diaphragm components
- Dry gas seals for process gas isolation within the flowpath
- Carbon shaft seals for lube oil system isolation
- Tilt pad or flexible-pivot journal bearings
- Optional damper bearing for discharge end of shaft

Solar gas compressor materials that are in contact with the gas generally meet the requirements of National Association of Corrosion Engineers (NACE) standards for the most severe sour gas service without any modification. *Solar* compressors have established a long and successful record of service in the oil and gas industry, and experience has shown that the materials Solar uses prevent the sulfide stress corrosion problems caused by H_2S.

7.1 MANUFACTURING

Solar's gas compressor manufacturing processes are designed to provide high quality parts on time and at a competitive price. Although many of the gas compressor manufacturing operations are outsourced, the most critical final operations are performed in Solar's ISO 9000-2000 certified facilities. Outsourced components are purchased as supplier-code serialized parts, complete with traceable inspection reports and certifications, to provide full visibility and accountability of all outside manufacturing processes. This minimizes the requirement for in-house dimensional inspection of incoming product. Component inspection and quality records are available to implement quick corrective action should non-conforming material be encountered.

Outsourced components include castings and forgings. Outsourced operations include machining for housings, stators, impellers, shafts and fasteners. In-house component operations include final manufacturing, balancing, inspection, and testing operations for critical components, subassemblies, assemblies, and final product. The quality program, with its system of checks and controls in manufacturing, assembly, and test ensures a satisfactory, safe, and reliable product upon completion.

The methodology used achieves minimal process variation and, therefore, consistent results. Proven, practical methodologies allow for extremely tight tolerance (±0.0002 in. total run out) components and high precision multi-component (0.00025 in. maximum electromechanical run out) rotor assemblies in a clean environment, which does not have to be temperature controlled.

7.2 IMPELLERS

Solar's compressor family includes impeller sizes from 178 mm to 914 mm (7 in. to 36 in.). The impeller geometry is fixed, that is, no custom modifications are made to a given impeller. The impellers in a family have the same hub and tip diameters and, in some cases, the same axial width. Figure 7.2 shows a range of impellers for different flow requirements. The impeller on the far left is a high flow design and the impeller on the far right is a low flow design.

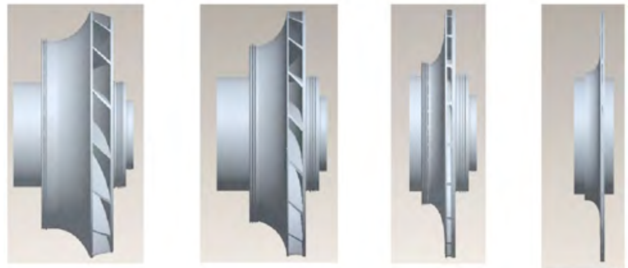

Figure 7.2 Impellers for different flow requirements

After introducing a compressor with a number of pre-engineered impellers and impeller combinations, ongoing development expands the coverage of each compressor. Diameters for the impellers are established to match the range of Solar's gas turbine family for use in direct-drive configurations.

The manufacturing operations include:

- Overspeed proof spin test at 115% of operating speed
- Non-destructive fluorescent penetrant inspection after spin
- Precision outside diameter and pilot grinding after spin to assure proper inter-stage fit up
- Precision drill holes for dowel pins
- Impeller detail balance
- Assembly and balance of the rotor
- Mechanical and electro-mechanical run-out inspection of the assembled rotor

Figure 7.3 shows the vacuum spin pit equipment on which all impellers are overspeed tested to verify the structural integrity of the part. After spin testing, the impellers undergo non-destructive inspection (Figure 7.4) using fluorescent penetrant inspection method to assure no propagating defects.

Figure 7.3 Overspeed spin pit

Figure 7.4 Non-destructive test facility

As shown in Figure 7.5, impellers undergo final grinding of the pilot diameters, mating faces, labyrinth seals. Dowel pinholes are drilled on the vertical machining center (Figure 7.6), and high-precision electronic probing ensures true position of the holes. Figure 7.7 illustrates how dowel pins are installed to ensure accurate rotor fit and the part is balanced.

Figure 7.5 Finish grinding

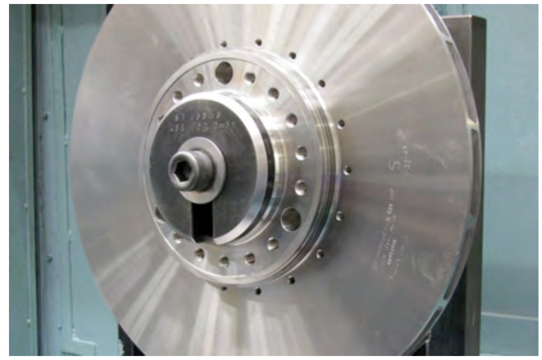

Figure 7.6 Precision drill

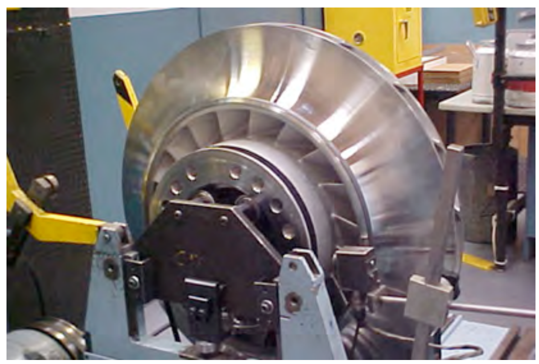

Figure 7.7 Detailed balance

7.3 BALANCE PISTON

The balance piston is located at the back of the last impeller. It creates an area of low pressure at the back of the last impeller that offsets the thrust of the rotor. Additionally, it reduces the pressure with which the shaft seals have to contend. The size of the balance piston is selected so that its force is greater than the sum of the unbalanced forces of the assembled rotor.

7.4 INTERNAL SEALS

The internal seals are labyrinth seals, which are cut on the shroud of the inlet to the impeller and on the hub on the back of the impeller. The stationary halves of the seals are babbit inserts in the diaphragms.

7.5 SHAFT END SEALS

Compressor shaft end seals and their attendant system prevent the escape of process gas along the shaft. Solar's current design approach is to use self-activated tandem dry gas seals. For a complete description of dry gas seals for *Solar* compressors and the seal system which supplies and monitors seal gas to the dry seals and the circumferential buffer seals, refer to Solar Product Information Letter (PIL) 140. Figure 7.8 shows the location of seal and bearings.

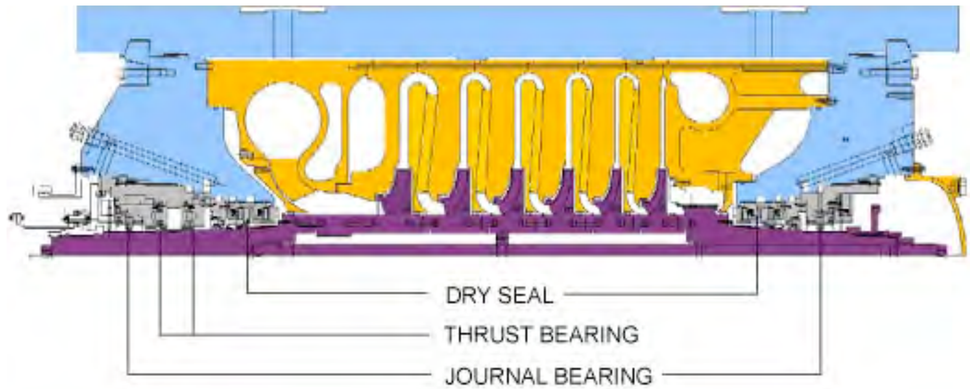

Figure 7.8 Location of seals and bearings

Dry Seal Configuration

Dry gas seals are mechanical face seals consisting of a mating (rotating) ring and a primary (stationary) ring. The mating ring rotates with the shaft, which is driven through the shaft sleeve. Tolerance rings center the shaft sleeve and mating ring. The primary ring is held against the mating ring by springs located in the housing. The design of the O-ring seal allows the primary ring to move axially with a minimum of frictional forces, while maintaining full seal differential pressure across it.

The tandem dry gas seal, with the total pressure drop taken across the inboard seal and using the second seal as a backup seal, is the standard configuration. These seals are installed as a single assembly.

Barrier Seal Configuration

The barrier seal is a segmented circumferential seal, buffered with air or nitrogen, which prevents oil leakage into the dry seal cavity.

Operating Principles

During static operating conditions, the seal dam area of the mating ring is in contact with the primary ring, up to a differential pressure of about 689 kPa (100 psi). Above this pressure, they separate from increased hydrostatic pressure between the sealing faces. During dynamic operation, the rotating mating ring lift geometry, in conjunction with the sealing dam, creates a pressure distribution that causes the primary ring to move away from the mating ring, forming a small, controlled gap between 0.0025 and 0.0050 mm (0.0001 and 0.0002 in.) wide. This very narrow gap allows a small leakage flow to pass through the seal. Different seal manufacturers will use different patterns for lift augmentation.

Contamination

For the dry seal to perform properly, the seal gas must be filtered to control the amount, type, and size of contaminants entering the seal. The seal gas must not have particles over two microns absolute in size nor must it have magnetized particles that will agglomerate. The seal gas should be free of liquids, and no liquid dropout should occur at the lowest site ambient temperature or seal gas supply temperature. There should be no liquid dropout, ice formation, nor hydrate formation due to pressure regulation from the seal gas supply pressure to the compressor case pressure. In situations where it is difficult or impractical to filter the process gas, a separate seal gas supply may be used. During start, pressurized hold, and shutdown transients, a separate seal gas supply is not required when the process gas is better than, or equal to, the cleanliness of pipeline quality natural gas. For seal gas requirements, refer to PIL 140.

Figure 7.9 shows a typical tandem dry gas face seal assembly and Figure 7.10 shows a tandem dry gas seal with intermediate labyrinth seal.

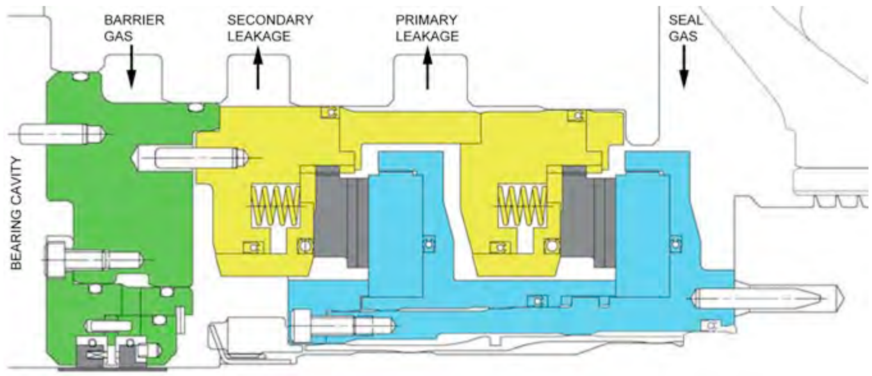

Figure 7.9 Tandem dry gas seal

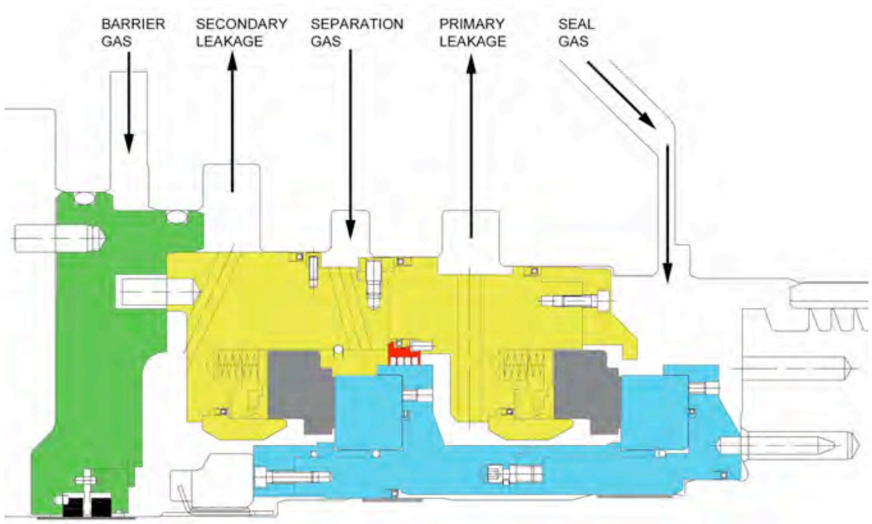

Figure 7.10 Tandem seal with intermediate labyrinth seal

Figures 7.11 and 7.12 show cutaways of tandem dry gas seal and tandem dry gas seal with intermediate labyrinth seal respectively.

Figure 7.11 Tandem dry gas seal

Figure 7.12 Tandem seal with intermediate labyrinth seal

7.6 BEARINGS

Hydrodynamic Journal Bearings

Figure 7.13 shows the tilt-pad journal bearings (whose location can be found in Figure 7.8) used by today's centrifugal gas compressors to make high operating speeds possible. Traditional oil film sleeve journal bearings suffer from fluid-flow induced instabilities, commonly referred to as oil whip or oil whirl. The tilt-pad bearing is not prone to these problems.

A typical tilt-pad journal bearing found in Solar's compressors consists of five pads. Each pad pivots on a hardened elliptical pin of variable diameter that is selected to achieve the desired clearance. The use of pins allows the clearance to be easily changed as the need arises. The pin makes line contact with the pad and housing. This contact offers high pivot stiffness, but it cannot self-align axially. Solar limits this type of bearing construction to a length/diameter (l/d) ratio of 0.25. Bearing edge loading may occur at higher ratios; consequently, different bearing geometries are required.

Since 1986, developments in compressor design have made use of larger l/d-ratio tilt-pad journal bearings, requiring a change in Solar's traditional mechanical design approach. Axial misalignment capability is now necessary to avoid journal bearing edge loading. This is accomplished by introducing a spherical pivot in place of the pin. The larger l/d-ratio bearings, combined with changes in rotor geometry, can improve rotor dynamics when compared with earlier configurations. For the same frame-size compressor, larger l/d-ratio bearings also reduce unit loading for improved durability.

Hydrodynamic Thrust Bearings

As illustrated in Figure 7.14, the thrust bearings are a tilt-pad design, same as with the journal bearings.

Figure 7.13 Journal bearings

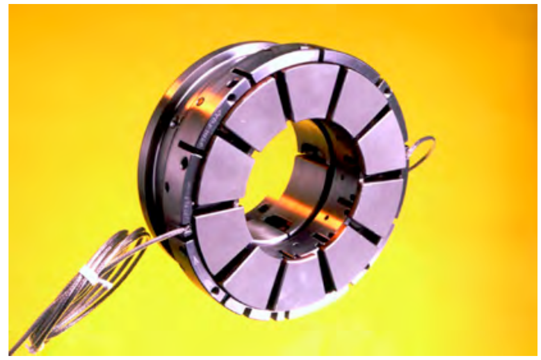

Figure 7.14 Tilt pad thrust bearing

In a tilt-pad thrust bearing, each thrust pad is an individual plate that is free to pivot. As the thrust collar rotates, each pad tilts, generating the load-carrying oil film. The bearing can accommodate slight misalignment between the bearing and thrust collar. Loads will be equally distributed among the pads through the movement of individual leveling links and disks. The self-equalizing tilt-pad thrust bearing has been adopted as the bearing of choice for new compressor designs. Its ability to accommodate misalignment and equalize uneven load distributions among the pads makes it more tolerant of high transient thrust loads caused by system upset.

7.7 CASINGS

The compressor's pressure-containing outer casing is a vertically split barrel-like assembly consisting of suction and discharge endcaps, which contain the bearing and seal assemblies, and a center-body, which holds the module. The endcaps contain all the service ports for oil and gas. All compressor casings are steel castings and conform to the design guidelines of ASME Section VIII and API 617.

Finite element analysis is used to determine the stress levels in the centerbody and the endcaps. While the center section is designed to keep the hoop stress levels in accordance with the ASME limits, the casing ends are made stronger to control the separating forces that tend to unload the casing-to-endcap seals. Likewise, the endcaps are designed to keep deflection from interfering with the operation of the compressor. Drains are provided on both suction and discharge sides of the casing. Figure 7.15 shows a raw cast casing before machining while Figure 7.16 shows the finished compressor.

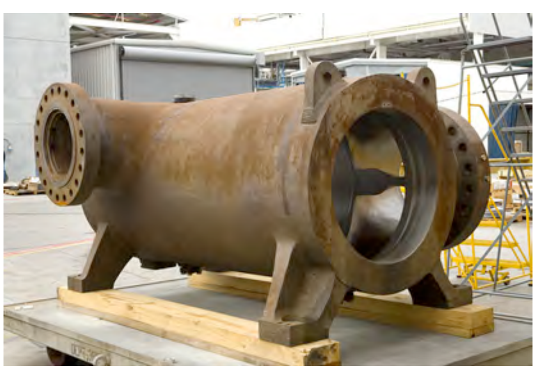

Figure 7.15 Raw cast casing

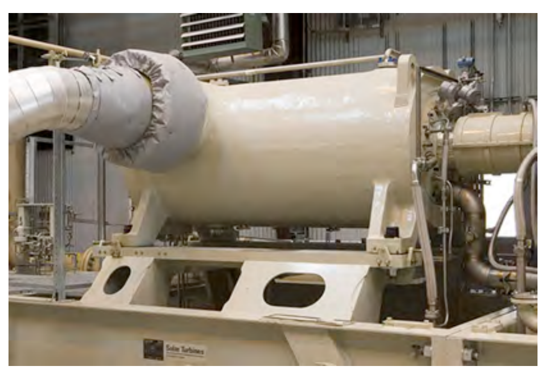

Figure 7.16 Finished C51 compressor

7.8 P2 INJECTION

Gas flowing under seals will create a destabilizing force if the gas retains the high tangential velocity as applied by the impellers when it enters the seal. This is of particular concern for the balance piston seal because of its length and high inlet pressure. For this reason, some production compressors with discharge pressures over 6,500 kPag (943 psig) use P2 Injection (Figure 7.17) into the balance piston seal to reduce the swirl from the gas before it enters the seal.

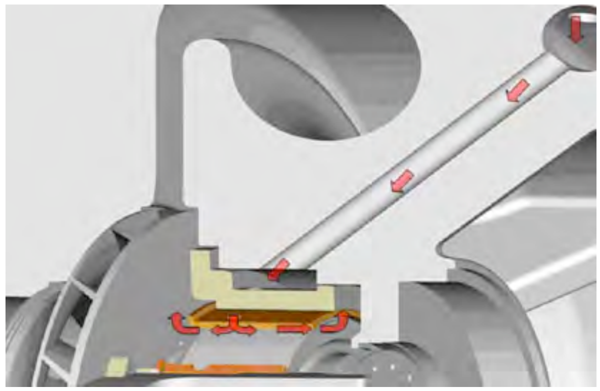

Figure 7.17 Balance piston seal P2 injection

8. Control of Centrifugal Gas Compressors

8.0 OVERVIEW

The three primary objectives for a compressor control system, listed in order of their priority, are:

1. Keeping the compressor within its operational boundaries
2. Meeting the process requirements
3. Distributing the load between multiple compressors

Meeting the process requirements should not cause the equipment to cross its operational boundaries. Distributing the load between multiple compressors should not cause the equipment to fail to meet the process requirements.

The three general methods of compressor control are:

1. Manipulation of the compressor's speed or its power
2. Use of recycle valves in parallel with the compressor
3. Use of suction or discharge throttling valves in series with the compressor

Manipulation of the compressor's speed or power is always more efficient than throttling or recycling the process gas flow; however, throttling and recycling are often used to augment driver control. Throttling and recycling are used when either the power or speed cannot be changed fast enough (transient) to respond to a transient event, or the process demands operation beyond the operational boundaries of the compressor.

8.1 OPERATIONAL BOUNDARIES

There are at least four boundaries of operation for all centrifugal compressors. They are:

1. Speed
2. Surge avoidance
3. Discharge temperature
4. Minimum pressure rise (choke)

Additionally, there may be a minimum operating speed, and there may also be conditions related to choke or stonewall, such as maximum turndown or minimum differential pressure.

Speed

The compressor is generally matched to the driver. As such, the maximum speed of the driver is the maximum speed of the compressor. If not, the maximum speed of the driver is reduced to the lowest maximum operating speed of the driven equipment. Compressors driven by gas turbines normally have a minimum operating speed below the minimum power turbine operating speed. If not, as with the electric motor drive, the minimum speed of the driver is increased to the highest minimum operating speed of the driven equipment. Management of the maximum minimum operating speed is accomplished within the driver control system.

Surge Avoidance

All centrifugal compressors have a maximum head they can achieve at a given speed and gas composition. At that maximum head, there is a minimum flow that must be maintained. Below that minimum flow, the head capability of the compressor will fall. When the head capability of the compressor is less than the prevailing system head, flow reverses and the compressor goes into surge. Recycling of the process gas is employed to ensure minimum flow with an acceptable margin.

Surge Avoidance During Start Up

In older designs, surge valves were left wide open during start-up. This approach, although effective at avoiding surge, created several problems. The greater the recycle flow, the lower the head for a given speed. Multiple recycle valves or ball valves often held the compressor in choke throughout the start sequence. As such, the compressor could not push the check valve open and get online until the recycle valves were closed. Constant recycling raises the temperature and the compressor is shut down because of an overheating condition. If the compression system does not include a cooler, the unit may shutdown before it can be placed online. To avoid this problem, modern surge avoidance systems hold the recycle valve capacity down to where the operating point of the compressor is held near surge. This is usually done with a fixed valve position because measuring head and flow at low compressor speeds is difficult even under the best conditions. Using a fixed valve position, a compressor's head/flow performance at surge can usually be matched within a few percent. An accelerating compressor will yield a greater surge margin as the discharge volume causes a lag in the pressure rise. A deceleration caused by a driver flame out during start-up causes an immediate emergency opening of the surge valve.

Surge Avoidance During Operation

During normal operation, two things will cause operation to move closer to surge: an increase in the system resistance or a reduction of the speed of the compressor. The latter may be a secondary effect of the first. An increase in the system resistance will be reflected at the compressor as an increase in discharge pressure or a decrease in suction pressure. For the same speed or power, increasing the head across the compressor results in a reduction in flow and surge margin. This is the typical "packing the line" scenario. A smooth, gradual opening of the surge control valve is appropriate. If, as a result of an increase in the system resistance, the suction or discharge pressure limit is reached, the process control will begin reducing compressor speed. As compressor speed falls, surge margin decreases and decreases more rapidly as it approaches surge. A more aggressive surge avoidance reaction will be required. In more sophisticated controls, reduction in speed is decreased or stopped entirely when the minimum surge margin is reached. This avoids undesirable control interaction.

Surge Avoidance During Shutdown

Again, in older designs, surge valves were quickly and completely opened with the initiation of shutdown. This is the most effective approach, for avoiding surge, once the fuel is shut off; however, with modern higher performance and higher efficiency gas turbines, a more controlled approach to shutdown is desirable to reduce thermal stresses and maintain tight clearances. To facilitate this, modern gas turbines are gradually decelerated to idle (cool down) before the fuel is completely shut off. If an already hot compressor is placed into full recycle, the compressor will overheat soon thereafter unless the recycle gas is cooled. Leaving the surge control engaged throughout this process has several advantages. With the surge control engaged, the turbine/compressor will decelerate with the operating point at the minimum surge margin, which is typically 10%. This operation reduces recycle flow while increasing head and subsequently discharge temperature. All of the energy put into the compressor is reflected in the mass flow and the temperature rise across the compressor. The energy balance doesn't change, however, the mass in the recycle loop is increased by the pressure ratio, and the heat loss through the recycle line is greater with the increased discharge temperature. If a recycle cooler is necessary to ensure reliable starting and completion of cool down, the cooler will be much smaller if its inlet temperature is higher.

Discharge Temperature

If the selected compressor is required to operate near its maximum discharge temperature, strategies may be needed to avoid unnecessary shutdowns. Avoiding shutdowns often cannot be achieved by controls alone. Starting the second compressor in parallel on a station where no cooling is required for normal operation may be a problem and should be analyzed. Driving a compressor into recycle when it is operating near its discharge temperature limit can cause the discharge temperature to be exceeded.

All the power that is put into a compressor will be reflected in the heat rise across it and the mass flow through it. From a cursory examination of the operating points of a compressor, it may be thought that cooling will not be required. Coolers are expensive and the pressure loss across one can be as much as the pressure loss through two miles of pipe. A common and expensive mistake is to not examine heat of the gas across the entire operating envelope. Cooler and piping designs, as well as control strategies, must address these off-design operating conditions.

If an increasing head requirement for the compressor causes the discharge temperature to approach its limit, power should not be increased further. Reducing power will reduce the flow through the compressor and reduce the surge margin. Recycling will increase the inlet and subsequently the discharge temperature.

Minimum Pressure Rise (Choke)

Most compressors require the discharge pressure be higher than the suction pressure during operation. If the application does not provide sufficient flow resistance to maintain the minimum pressure rise across the compressor, throttling must be used. Control of the throttle valve is similar to surge avoidance. The throttle valve will be completely open as long as the minimum pressure rise across the compressor is maintained.

8.2 PROCESS CONTROLS

Process control imposes additional boundaries of operation on the compressor.

The most common process controls are:

- Minimum suction pressure
- Maximum discharge pressure
- Maximum flow

All of these limits can be imposed on a single compressor; however, only one will be in control of the process. All three are controlled in the same way. If the process moves through an operational boundary, power or speed is reduced until the parameter returns to within the operational boundary.

Driver Control

The most common and efficient method of controlling a compressor is through its driver. Power control is used most often with two-shaft gas-turbine drivers. The process controller output is applied to the gas producer. The power turbine and subsequently the compressor speed are not controlled, but rather they are allowed to find their natural equilibrium with the process. Minor changes in process conditions that cause the compressor/turbine speed to vary are not acted upon by the process control. This steady power setting reduces unnecessary wear on the gas producer.

Speed control is most often used with variable-speed electric-motor drives. The process controller output is applied to the speed set point of the motor controller. Because the process controller constantly corrects the speed of the motor to the set point, changes in process conditions cannot cause the compressor/motor speed to vary.

Recycling

Although only one process control can be in control of the process, multiple control elements can be used to control the process. When the power or speed limit is reached, or the change in the process condition is faster than the gas producer can respond, recycle is used. Within the operating speed and power range, recycle is used to ensure a transient error band (typically 2%) is not exceeded. This is accomplished by adding a small offset to the process control error to the valve control.

If power or speed is reduced to maintain the process, the surge margin of the compressor will be reduced. At the surge protection margin, the surge avoidance control will begin opening the recycle valve. As speed continues to be reduced to achieve the process control objective, the recycle valve will continue to open. To ensure stability, some controls reduce the rate of power reduction when the surge avoidance control intervenes. Modern high-performance, low-emissions engines use power change rates that are slow enough to not interact with the surge avoidance control.

If the driver has reached one of its limits and the process is then being maintained by recycling, the offset can be removed from the recycle control. In the rare instances when this is not acceptable, the offset can be removed when the engine is being controlled by another limit.

Recycle can be used for process control in an uncooled system; however, strategies must be used to ensure discharge temperature limits are not exceeded, which could cause subsequent compressor shutdown.

Throttling

Compressor throttling is used to maintain minimum head across a compressor and can be applied in the following situations:

- Avoiding operation of a compressor in choke
- Process control for constant speed compressors
- Compressors connected to multiple pipelines

Driver Control and Recycling

If a change in the process condition or set point is faster than the driver can track, recycling can be used to ensure a transient error band is not exceeded. This can be accomplished by adding a small offset to the process control error before conveying it to the valve control.

If the driver/compressor has reached one of its limits, the process can be maintained by recycling. If control is completely handed off to recycle, the offset can be eliminated. If the driver control can be restored, the offset must also be restored. It is imperative that two control elements controlling the same process only be allowed during a transient. The offset causes the secondary control element to fully close or open before the set point is met.

If power is reduced to maintain the process, the surge avoidance control may intervene. In this case the recycle valve is not opened to maintain the process, but rather to maintain the surge protection margin. As speed continues to be reduced to achieve the process control objective, the recycle valve may re-close. Some controls reduce the rate of power reduction when the surge avoidance control intervenes.

Control of Multiple Compressor and Process Parameters

The three dimensions of compressor operation are speed, head, and flow. The compressor's operation can be described on a plane bounded by maximum speed, minimum speed, maximum flow (choke), and minimum flow (surge). The system the compressor is installed in has only two dimensions: head, and flow. A line called a system characteristic curve can describe it.

There may be multiple inlets and outlets to a compressor. Pressure at any inlet or outlet of a compressor system is proportional to the flow in or out of it. The driver can control only one process. The maximum number of process controls that can be imposed is one less than the number of connections to the system.

For example, in a system with a compressor train connected to a suction header and two discharge headers, maintaining minimum suction pressure has caused reduced driver speed. In this situation, flow is less than the minimum compressor flow at the prevailing head (surge limit); therefore, minimum compressor flow is maintained by recycling. The discharge pressure of header 1 is controlled by throttling while the discharge pressure of header 2 is not controlled. If the pressure in discharge header 2 rises to its maximum, throttling could control this situation. However, additional throttling would reduce suction flow, and with suction pressure below its set point, process control would allow driver speed to rise. This is not the most economical control action as more power than necessary is being consumed to meet the process objectives.

The appropriate action would be to further reduce the driver speed, which would cause the suction pressure to rise. Subsequently, suction pressure is above its set point and is not being controlled. Flow is still less than the minimum compressor flow at the prevailing head (surge limit). Minimum compressor flow is maintained by recycling. Throttling still controls pressure in discharge header 1. Again, note that two processes and one operational boundary are controlled.

8.3 CONTROL OF MULTIPLE COMPRESSORS WITH A COMMON DRIVER (TANDEM)

The multiple-compressor configurations that may be found in a common driver system are:

- Compressors operated in parallel
- Compressors operated in series
- The back-to-back compressor
- The three-flange compressor
- Compressors operated in series/parallel (all in series or two into one)
- Tandems in parallel
- Tandems in parallel with control of the intermediate pressures

The rules that apply to a single compressor system also apply to multiple compressor systems. The driver can control only one process. There may be multiple inlets and outlets to a compressor system. Pressure at any inlet or outlet of a compressor system depends on the flow in or out of it. The maximum number of process controls that can be imposed is one less than the number of connections to the system.

Compressors, in series side streams (Figure 8.1), can be controlled in a variety of ways, depending on the conditions:

- If the suction pressure is low, power must be reduced to the train. With the reduced inlet flow, one of the outlets may be able to maintain pressure.
- The high-pressure compressor in the train can be either throttled or recycled. If the suction pressure is low, additional power must be applied to the train.
- If pressure is high, the low pressure compressor can be either throttled or recycled. If pressure between the compressors in the train is low, the high pressure compressor can be either throttled or recycled.
- If the system discharge pressure is high, the high pressure compressor can be either throttled or recycled. If the discharge pressure is low, additional power must be applied to the train.

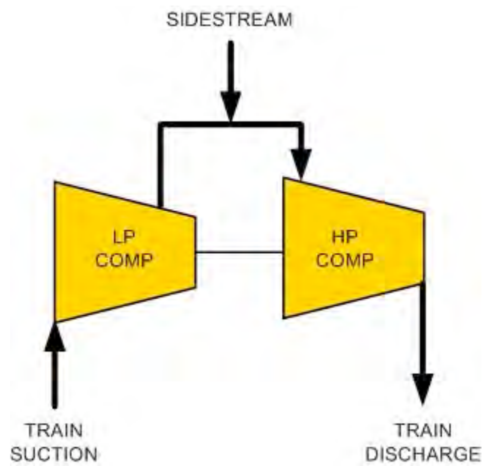

Figure 8.1 Compressors in series with side stream

8.4 CONTROL OF MULTIPLE COMPRESSOR SETS (LOAD SHARING)

The multiple-compressor set configurations that may be found in load-sharing systems are:

- Compressor sets operated in parallel
- Compressor sets operated in series
- Compressors operated in series/parallel (all in series or two into one)
- Tandems in parallel
- Tandems in parallel with control of the intermediate pressures

Load Sharing of Identical, Multiple Compressor Sets in Parallel

If the compressor sets are identical and the recycle valves are identical, the task of controlling the compressor sets is simplified. Load sharing can be accomplished by equalizing their driver power (N_{gp}), equalizing compressor speed (N_{pt}), or equalizing their turndown. Note that N_{gp} is the symbol for the speed of the gas producer turbine and N_{pt} is the symbol for the speed of the power turbine.

Equalizing the power (N_{gp}) of their drivers will keep the compressors operating the same as closely as the compressor sets (engine and compressor) are matched. Equalizing the speeds (N_{pt}) of their compressors will keep the compressors operating the same as closely as the compressors are matched. Equalizing their compressor turndown will keep the compressors operating the same as closely as the compressors and the surge avoidance system instrumentation is matched.

Load Sharing of Dissimilar, Multiple Compressor Sets in Parallel

There can be one or more large compressor sets in parallel with one or more smaller compressor sets. The larger units would form the baseline compression, as they typically are more efficient. The smaller, less efficient units operate as peaking units. Typically, the smaller units would be started and shut down as necessary and recycling would be avoided.

Load Sharing of Multiple Compressors Sets in Series

Multiple compressor sets in series may have the same type of drivers, but will not have the same compressors or operate at the same power. Again, multiple compressor sets in series can be operated at equal turndown. If part of the operational envelope includes recycling, an alternate control strategy must be adopted.

8.5 AUTO START/STOP

Some systems require a range of operation greater than can be achieved by a single compressor set. This range of flow can be achieved by starting additional compressor sets or shutting down unneeded compressor sets. The algorithm for starting and stopping compressor sets must not only consider optimum operation of the system, but must also consider the additional wear on the equipment.

If the process requires more than can be achieved with the on-line compressors operating at full speed or power, an additional compressor set should be started. If the process requires less than is being provided with the on-line compressors operating at minimum speed or power, one of the on-line compressors can be shut down.

The width of the power band between starting and stopping a unit is an additional consideration. Starting or shutting down one of two units (increasing capacity by 100% or decreasing capacity by 50%) will have a much more significant impact in overall operation than starting or shutting down one of three units (increasing capacity by 50% or decreasing capacity by 33%).

Automatic Start-up/Shutdown of Multiple Compressor Sets in Parallel

If the compressor sets are identical, the choice of which one to start or shut down can be based on many factors. The operator may want to keep the operating time on all the equipment about the same; therefore, the operator may choose to start low-time equipment and shut down high-time equipment. The operator may want to operate by consuming the least fuel and will keep the best performing equipment on line, shut down the low performers, or start the high performers.

A compressor station may have compressors with different flow capacities. It may have one or more large compressors to handle the base load and one or more smaller compressors to handle the peaking duties. The large base-load compressor(s) are intended to run all the time.

Operating high on the compressor map and starting a second compressor to meet conditions may cause both compressors to operate in recycle. This will happen if operation is required in the area between the maximum power line for one compressor and the surge line for both compressors.

Automatic Start-up/Shutdown of Multiple Compressor Sets in Series

As with the parallel compressors, power drives the decision to start or stop a unit. Compressors in series will have the same mass flow but not the same actual volumetric flow. As such they probably will not be staged the same. Increasing discharge pressure should cause a low-flow compressor to be started while a decreasing suction pressure should cause high-flow compressor to be started.

9. Surge Avoidance

9.0 OVERVIEW

Centrifugal compressors have a maximum head (peak head) that can be achieved at a given speed. At that peak head there is a corresponding flow. This is the aerodynamic stability limit (Figure 9.1a). Operation of the compressor is stable provided the head is lower (less resistance in series with the compressor) and the flow is greater than these values. That is, reductions in head result in increases in flow.

Increasing the system resistance or reducing compressor speed will decrease flow. Attempting to operate a compressor at a lesser flow than at peak head will result in unstable operation. As flow decreases, the head capability of the compressor will also decrease. As the head capability decreases, flow further decreases. Once the compressor can no longer meet the external head, it stalls and flow reverses. Surge is what happens after the compressor stalls.

Not only is surge detrimental to meeting the process objectives, the resulting axial and radial movement of the rotor can cause damage to the compressor. Surge can be avoided by ensuring that the flow through the compressor is not reduced below the flow at peak head.

A surge avoidance system prevents surge by modulating a valve in a recycle loop around the compressor. This is referred to variously as the recycle, bypass or anti-surge valve. A typical system (Figure 9.1b) consists of pressure and temperature transmitters on the compressor suction and discharge lines, a flow differential pressure transmitter across the compressor flow meter, an algorithm in the control system, and the recycle valve with corresponding accessories.

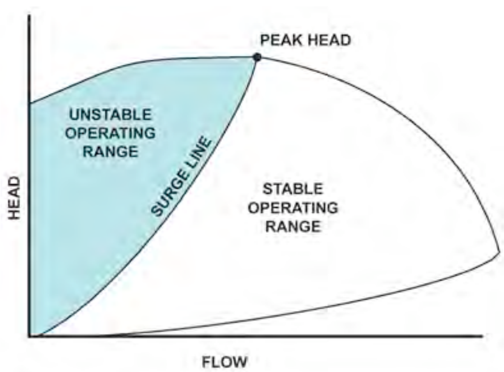

Figure 9.1a Compressor operating range

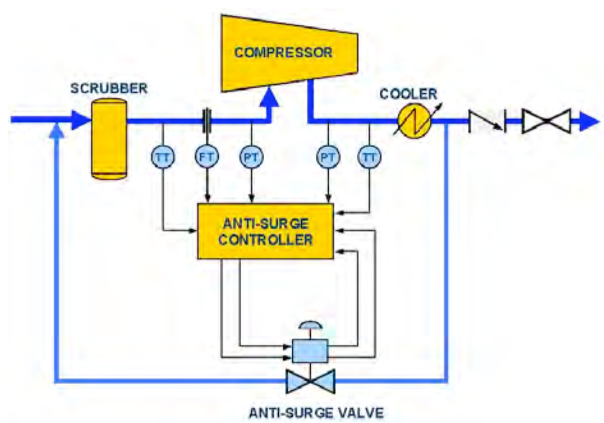

Figure 9.1b Basic surge avoidance recycle configuration

The system determines the compressor's operating point using the pressure, temperature, and flow data provided by the instrumentation. The system compares this operating point to the compressor's surge limit. The difference between the operating point and the surge limit is the control error. A PID (proportional, integral, derivative) control algorithm acts upon this difference, or “error,” to develop a control signal to the recycle valve. When opened, a portion of the gas from the discharge side of the compressor is routed back to the suction side and head across the compressor is prevented from increasing further. When the operating point reflects more flow than that required to maintain the protection margin, the surge control valve moves toward the closed position and the compressor resumes normal operation.

It is important to note that any surge avoidance system has to be configured for two fundamentally different situations:

- In the control mode, the system has to be able to adjust the recycle valves to usually relatively slow process changes such that the process is not disturbed.
- In the emergency mode, the valve has to be able to react fast enough to avoid compressor surge during rapid transients, for example during emergency shutdowns.

9.1 THE ESSENTIALS FOR SURGE CONTROL

There are five essentials for successful surge avoidance:

1. Surge Limit Model—must precisely predict the surge limit over the applicable range of gas conditions and characteristics.
2. Control Algorithm—must ensure surge avoidance without unnecessarily upsetting the process.
3. Instrumentation—must be selected to meet the requirements for speed, range, and accuracy.
4. Recycle/Anti-surge Valve—must fit the compressor and be capable of large and rapid, as well a small and slow, changes in capacity.
5. Review of System Volumes—recycle valve must be fast enough and large enough to ensure the surge limit is not reached during a shutdown. The piping system is the dominant factor in the overall system response. It must be analyzed and understood. Large volumes may require more complex valve arrangements with more than one valve.

9.2 THE SURGE LIMIT MODEL

In order to avoid surge, it must be known where the surge will occur. The more accurately this is predicted, the more of the compressor’s operating range that is available to the user. A compressor’s operation is defined by three parameters: head, flow, and speed. The relationship between the compressor’s operating point and surge can be defined by any two of the three.

In Figure 9.2, three different models are shown for expressing the distance of the compressor’s operating point from the surge limit: surge margin, head rise to surge, and turndown.

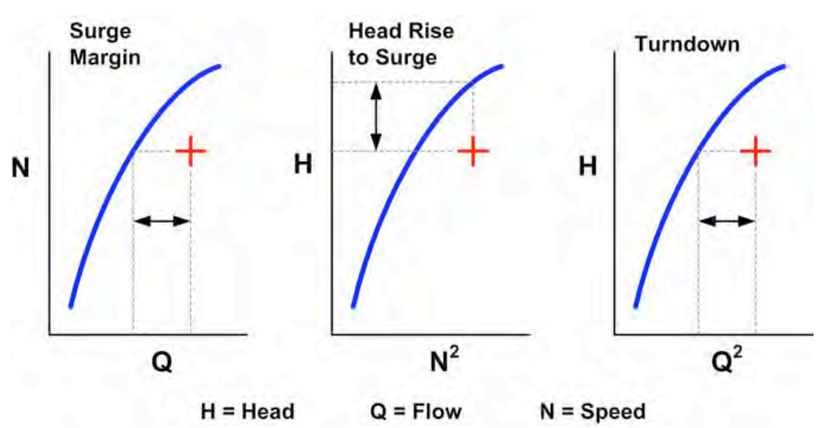

Figure 9.2 Surge limit models

Surge margin and head rise to surge involve speed and, as a result, the surge limit will change with changes in the composition of the gas. The turndown uses a head-versus-flow relationship to provide a means of modeling the surge limit without being affected by gas conditions or characteristics. The parameters of the surge limit model using turndown can be measured in terms of the head across the compressor and the head across the flow-measuring device.

$$H = K \cdot \frac{\left(\frac{P_D}{P_S}\right)^{\sigma} - 1}{\sigma} T \cdot SG \cdot Z$$

$$\sigma = \frac{n-1}{n}$$

This is the basic equation for polytropic head (H). For the head across the compressor and the head across the flow-measuring device, there are common terms. These terms, units (K), gas temperature (T), specific gravity (SG), polytropic exponent (n) and compressibility (Z) are the same in both equations and can be cancelled out.

$$H_{REDUCED} = \frac{\left(\frac{P_D}{P_S}\right)^{\sigma} - 1}{\sigma}$$

A similar reduction can be performed on the equation for flow. The result is a simplified model that is referred to as reduced head versus reduced flow.

9.3 THE CONTROL ALGORITHM

A surge avoidance control must be able to react appropriately to changes in power and the process. There are two very different situations to which the system must respond.

If the operating point slowly crosses the control line, opening of the recycle valve should be slow and the valve size should be small. The interdiction of the surge avoidance control should be unnoticeable. It should be as though the compressor had infinite turndown.

Conversely, if the operating point races across the compressor map in response to a sudden change, the recycle valve should begin opening before the operating point crosses the protection line. Reaction of the control should be aggressive to protect the compressor. In this case the process is not the major concern, as it has already been affected.

A sudden change in the operating conditions produces a control response. This is a standard control test. Figure 9.3 reflects reactions of variously tuned controls: a low gain produces a slow response; a critically damped control produces an aggressive response but settles down quickly; too high a gain causes the system to oscillate.

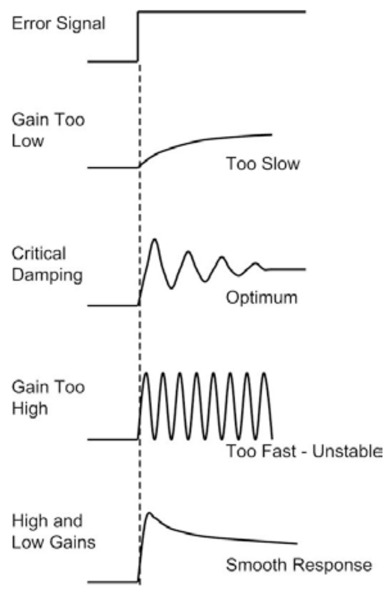

Figure 9.3 Typical control responses

Most of the time a surge avoidance system is idle. At times, it must react aggressively to protect the compressor, probably requiring gains too high for stability. Once open, the gains are reduced to avoid instability in closing the valve. That is, once surge has been avoided, the control system should bring the process back on line slowly and smoothly without further upsets.

The need for extremely high gains is driven by the fact that surge avoidance systems are normally built using commonly available process control plant components. As such, these components are designed for ruggedness, reliability, and low maintenance. In general they are not focused on the speed of data acquisition. Information about changing process conditions often lags by a tenth of a second. Significant advances in surge control valve action have been made in recent years; however, the response of the valve is still typically the dominant lag in the system.

9.4 INSTRUMENTATION

To avoid surge, the control system needs to know where the compressor is operating in relation to the compressor's surge line in real time. Again, how close the protection margin can be placed to surge depends on how accurately and how quickly the change in flow is reported to the control. Correctly selected instrumentation is essential. The system must have accurate measurements of the suction and discharge pressures and temperatures, and the rate of flow. Flow is the most important parameter, as it will move the fastest and farthest as the surge limit is approached. Ideally, the flow transmitter should be an order of magnitude faster than the process. Unfortunately, compared to pressure and temperature transmitters, flow transmitters tend to be slow. Even the best surge avoidance control will allow a compressor to surge if it is connected to a slow transmitter.

9.5 FLOW MEASURING DEVICES

Most commercially available flow measuring devices are accurate enough for surge avoidance; however, it is the transmitter that slows the control system's response times. A differential pressure transmitter's response time is inversely proportional to its range; thus, the stronger the signal, the faster the response.

Devices that develop high differential pressure (DP) signals are desirable. Those with low signal levels tend to have low signal-to-noise ratios. Transmitters for low DP signal ranges typically have slow response times. Devices that create an abrupt restriction or expansion to the gas, such as orifices, cause turbulence and, subsequently, create noise.

It is preferable to place the flow measuring device on the suction side of the compressor. Typically, variations in pressures, temperatures, and turbulence of the gas are less upstream of the compressor. Also, the device must be inside the innermost recycle loop (see Figure 9.1). Failure of the device will cause the compressor set to be shut down.

Low permanent pressure loss (PPL) devices are often recommended; however, their benefits may be marginal. The cost impact of the power lost due to operating a PPL device can be calculated. For example, a flow meter developing a 100-inch H_2O signal and a 50% PPL flowing 100 MMSCFD (50 lb/sec) is equivalent to about 20 horsepower.

As noted, strong signal devices are highly preferred. Pitot types (Annubars & Verabars) have a relatively low signal level, around 25 inches H_2O. In the middle are orifices and venturis with a moderate signal of around 100 inches H_2O. Compressor suction-to-eye provides a strong signal, around 700 inches H_2O, with the added benefit of not causing any additional pressure loss.

Suction-to-Eye Flow Measuring. This method uses the inlet shroud or inlet volute of the compressor as a flow-measuring device. This feature is now available on many compressors. The design requirements for the inlet volute and the flow-measuring device have several things in common. Performance of the first stage impeller and the device is dependent on the uniform direction and velocity of the flow presented to it.

Critical to the operation of suction-to-eye flow measurement is the placement of the eye port. As the impeller approaches surge, an area of recirculation begins to develop at the outer perimeter of the inlet to the impeller. If the eye port is placed too close to the impeller's outer perimeter, the relationship of the DP to flow will be affected. Fortunately the calibration factor typically remains nearly the same for the same surge margin. Hence, selecting the meter factor at the desired surge protection margin will contribute to effective surge avoidance.

In a typical pipeline application (600 psi suction pressure) suction-to-eye will develop 25 psid (692 inches H_2O). This is nearly seven times the differential of an orifice plate. Typically the signal-to-noise ratio is low and there is no additional permanent pressure loss. For surge avoidance the suction-to-eye method is strongly recommended.

9.6 THE RECYCLE/ANTI-SURGE VALVE

Earlier it was discussed how the control should react differently to gradual and rapid approaches to surge. Likewise, the valve must address these two very different requirements. For the gradual approach, it should behave like a small valve and produce smooth throttling. For the rapid approach case, it should act like a large fast valve to handle sudden major changes.

There are three general valve characteristics:

- Quick opening, where most of the valve's capacity is reached early in its travel
- Linear, where capacity is equal to travel
- Equal percentage, where most of the capacity is made available towards the end of the valve's travel

All three types of valve have been used in various configurations as recycle valves. These types of valves are shown in Figure 9.4.

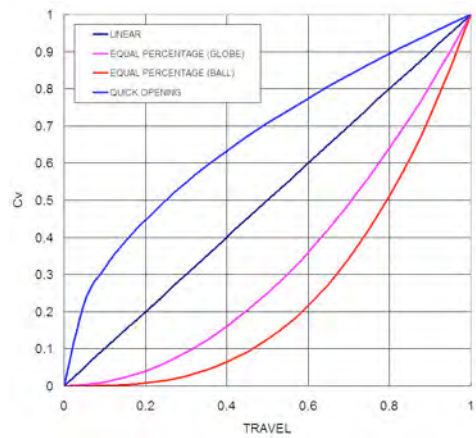

Figure 9.4 Valve types

Equal percentage valves, and in particular noise-attenuating ball valves, are recommended for surge avoidance systems with a single surge control valve. They perform like smaller valves when nearly closed and bigger valves when close to fully open. Figure 9.5 is a comparison of two types of equal percentage valve. For a given valve size, the noise-attenuating ball valve is often twice the cost of the globe valve, but it provides approximately three times the flow coefficient (Cv) or capacity. It is also more reliable as it is less susceptible to fouling and improper maintenance.

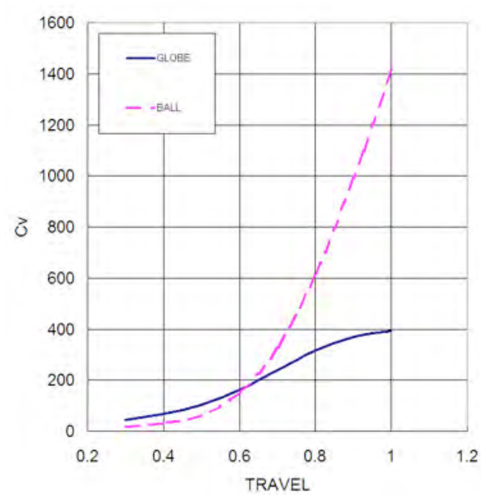

Figure 9.5 Ball and globe valves compared

Employing a valve with an equal percentage characteristic may provide the capacity needed to avoid surge during a shutdown while maintaining enough resolution at less than 50% capacity to provide good control at partial recycle. With an equal percentage characteristic, the valve typically has greater resolution than a single linear valve selected to fit the compressor.

Multiple Valves

If the volumes on either side of the compressor are large, a multiple valve approach may be needed. If an integrated approach is used, the total valve capacity will be reduced.

Figure 9.6 shows the most common hot and cold recycle configuration. Usually the cooled (outer) valve is modulating and the hot (inner) valve is a quick opening on-off type. Generally, the two valves are sized independently. If the cooled valve has a solenoid, its capacity can be considered with that of the shutdown valve; subsequently the shutdown valve can be smaller.

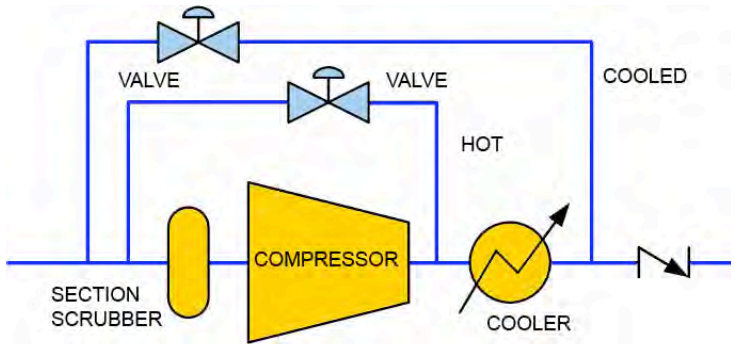

Figure 9.6 Hot and cold recycle valve arrangement

An alternate to this configuration is to have a second cooled valve in parallel with the first as Figure 9.7 shows. This arrangement provides some measure of redundancy. In the control the two valves are operated in cascade. That is, they have different set points, for example 9% and 10% surge margin. Under normal movements of the operating conditions only the 10% surge margin valve (primary valve) will open. If movement is fast enough to push the operating point down to 9%, the second valve (secondary valve) will open. If the primary valve becomes fouled and no longer positions properly, the control can place it in the secondary position and the secondary becomes the primary valve. This change can be made without taking the compressor off line.

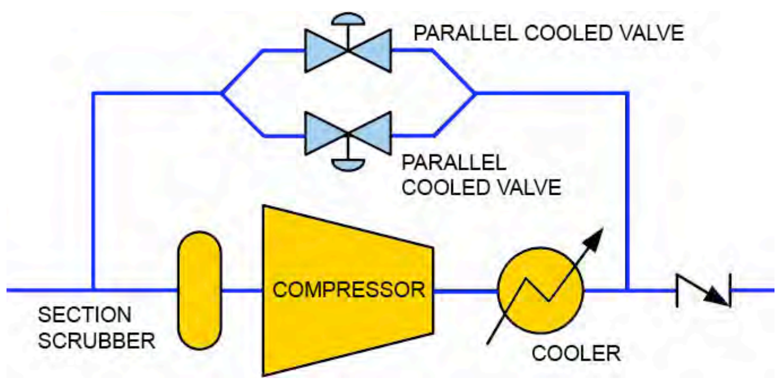

Figure 9.7 Parallel recycle valves

The advantages of the two parallel valves do not come without a price. In normal operation 2% to 5% of the pressure rise across the compressor will be lost across the cooler. In the shutdown scenario, the required flow through the cooler to avoid surge may be two or three times the normal flow. This will result in four to nine times the pressure drop across the cooler. This additional pressure drop may increase the needed recycle valve capacity significantly.

Recycle valves need to be fast, and accurately positionable. They also need to be properly sized for both the compressor and the piping system. A valve well suited for modulating recycle around the compressor may not be suitable for a shutdown. (A further explanation can be found in the Review of Piping Volumes Section 9.7).

For some two-valve applications, single-purpose valves may be suitable, one for controlled recycling and one for shutdown. A linear characteristic valve is appropriate for the controlled recycling and a quick-opening characteristic globe or ball valve for shutdown.

For the applications where the compressor speed lines are fairly flat (little increase in head for a decrease in flow) from the design conditions to surge, extra-fast depressurization may be required. To achieve this, two quick-opening valves may be employed. In this case, a single 6-inch linear characteristic valve is replaced by two 4-inch quick-opening valves. The two 4-inch valves should have slightly less flow capacity (Cv) but they will open nearly 45 milliseconds faster. For linear valves, 50% travel equals 50% capacity. For quick-opening valves, capacity approximately equals the square root of travel. As such the two 4-inch valves will have 70.7% of their fully open capacity at 50% open. Comparing the two arrangements 250 ms after the shutdown is initiated, the two 4-inch quick-opening valves will have 56% more flow capacity than the single 6-inch linear valve.

For throttling, the valves are operated in cascade or split range. For most controlled recycling only one valve is opened. Although the valves have a quick-opening characteristic, the valves are smaller, thus the capacity per percent travel is less. The two quick-opening valves operated in cascade or split range will have the same Cv as the 6-inch linear at 25% travel.

Valve Actuation

As previously discussed, there are two operational scenarios for the surge avoidance system: modulating (minimum flow control) and rapid depressurization for shutdown. By inserting a three-way solenoid valve into the positioner's output, the valve can be made to open with either a proportional (4-20 mA) signal for modulating control, or a discrete (24 VDC) signal for total fast opening.

The primary difference between a surge control valve and a standard control valve is in its actuation system. The preferred actuator for surge avoidance is spring return, fail open. This design is simple, reliable, and ensures the compressor is protected in the event of a power failure. Both spring and diaphragm and spring and piston actuators are used. The spring and diaphragm actuator is most commonly used on globe valves. The spring and piston actuator is more often used on ball valves. The more powerful spring and piston actuators are required on rotary valves due to the greater forces required to accelerate the mass of the ball. Some ball valves are not suitable for surge control applications because their shafts and attachments to the ball are not strong enough to transmit the torque required to open these valves at the required speeds.

Surge control valves need to be able to open very quickly. As such, their actuators will have strong springs, very large air passages, and shock absorbers at their end of travel. This must be considered when sourcing recycle valves for surge avoidance.

The accessory unique to a sound surge control valve assembly is the single-sided booster or exhaust booster. This is essentially a differential pressure relief device. Opening the booster vents the actuator pressure to atmosphere. The threshold for opening is about 0.5 psid. There is a small restriction (needle valve) between the control pressure from the positioner via the three-way solenoid valve and the top of the booster. Small slow reductions in pressure (opening the valve) do not cause the booster to open. Large fast reductions in pressure developing more than 0.5 psid across the restriction cause the booster to open. If the solenoid valve is de-energized, the top of the booster is vented to atmosphere and the booster fully opens.

Standard industry quick-exhausts are not recommended for this application. They have a high threshold for opening (typically 2 – 4 psid) and an equally high threshold for re-closing. Although they may work well for fully opening the valve they will not work well with the positioner.

Positioners should be selected for high capacity and quick response to changes in their control signals. Most of the major valve manufacturers have released second and third generation smart positioners that are suitably fast for this application. All surge avoidance valve assemblies supplied by Solar are provided with smart positioners.

Figures 9.8 and 9.9 show globe and ball valves with their preferred instrumentation configurations.

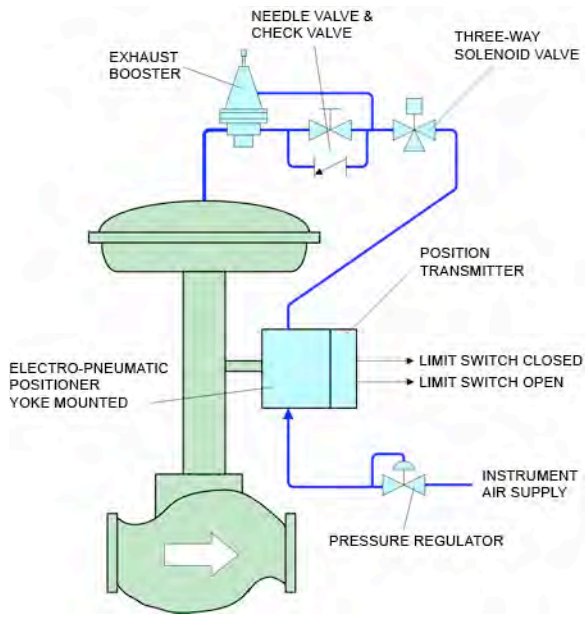

Figure 9.8 Globe valve assembly

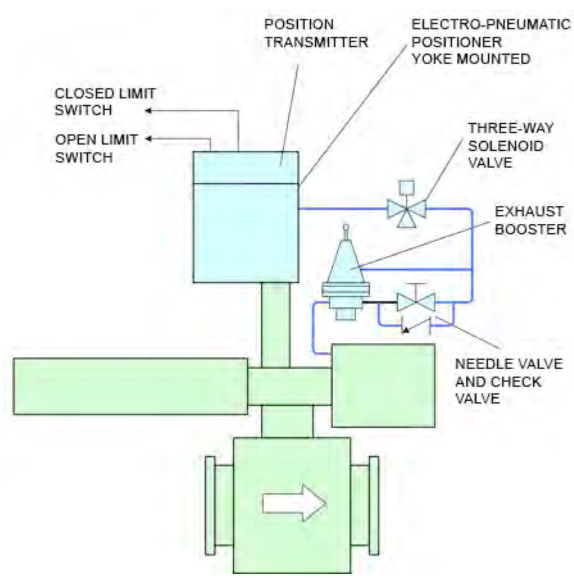

Figure 9.9 Ball valve assembly

Recycle Valve Sizing Tool

Solar has developed a valve-sizing program to facilitate matching a recycle valve to a compressor. The compressor data is entered into the tool in its normal form (pressures, temperatures, heads, speeds and flows). Various operating conditions for a specific application are then entered, such as the minimum and maximum operating speeds, pipe operating pressures, temperatures, relief valve settings, and cooler data if applicable. The tool calculates the equivalent valve capacities or Cvs from that data.

Typically the surge limit of a compressor equates to a single valve capacity or Cv (Figure 9.10). The valve tool has an extensive library of valve Cg, Cv, and Xt tables from the major surge control valve suppliers. As previously described, a single surge control valve application will have an equal percentage characteristic. Once a valve is selected, several performance lines of a specific opening can be developed and overlaid on the compressor map. The equal percentage characteristic valve should be at about two-thirds travel at the surge conditions. The valve evaluation in Figure 9.11 shows such a valve with its 60% and 70% open performance lines straddling the compressors surge line. Multiple valve configurations will typically have less reserve capacity.

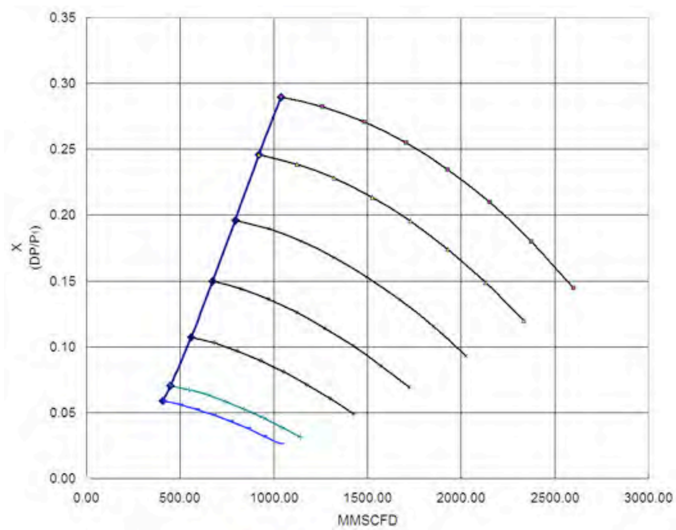

Figure 9.10 Almost constant Cv at the surge limit

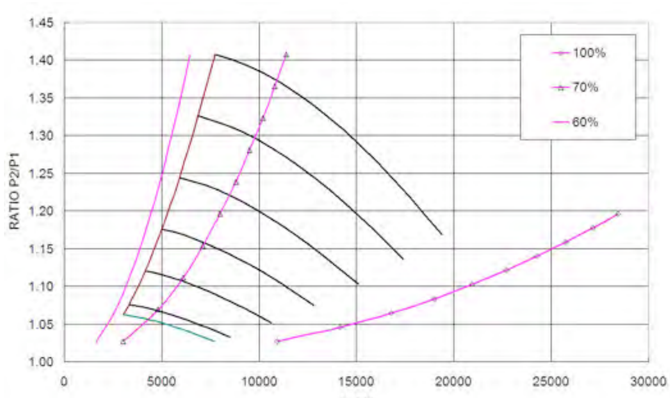

Figure 9.11 Valve matched to compressor

The tool enables the surge control engineer to quickly compare surge control valves from different manufacturers to find the optimum fit.

9.7 REVIEW OF SYSTEM VOLUMES

Design of the piping and valves, together with the selection and the placement of instruments will significantly affect the performance of an anti-surge control system. This should be addressed during the planning stage of a project because the correction of design flaws can be very costly once the equipment is installed and in operation.

As described previously, the control system monitors the compressor operating parameters, compares them to the surge limit, and opens the recycle valve as necessary to maintain the flow through the compressor at a desired margin from surge. In the event of an emergency shutdown (ESD), where the fuel to the gas turbine is shut off instantly, the surge valve opens immediately, essentially at the same time the fuel valve is closing.

In a simple system (Figure 9.12), the boundaries for the gas volume on the discharge side are established by the discharge check valve, compressor, and anti-surge valve. The volume on the suction side is usually orders of magnitude larger than the discharge volume and, therefore, can be considered infinite. Thus, to simplify the analysis of a system, the suction pressure can be considered constant. This is not a general rule, but is used to simplify the following considerations. This yields the simplified system, consisting of a volume filled by a compressor and emptied through a valve.

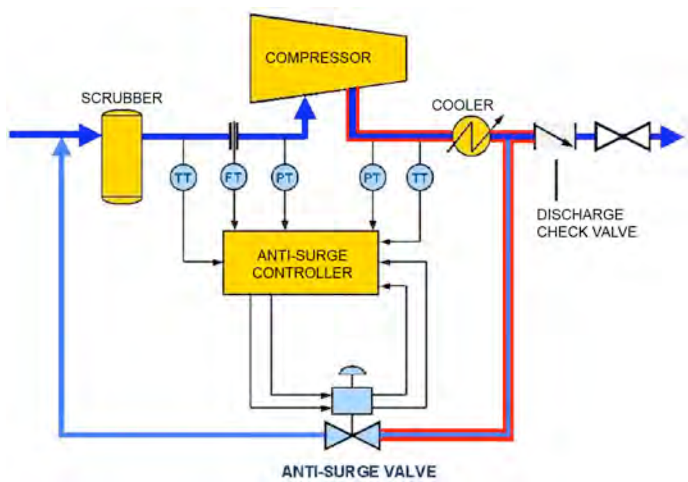

Figure 9.12 Piping volume (red highlight)

The basic dynamic behavior of the system is that of a fixed volume where the flow through the valve is a function of the pressure differential over the valve. In a surge avoidance system, a certain amount of the valve's flow capacity will be consumed to recycle the flow through the compressor. Only the remaining capacity is available for de-pressurizing the discharge volume.

The worst-case scenario for a surge control system is an ESD particularly if the compressor is already operating close to surge when the engine shutdown occurs. If an ESD is initiated, the fuel supply is shut off immediately and the compressor will decelerate rapidly under the influence of the fluid forces counter-acted by the inertia of the rotor system. Figure 9.13, which displays data based on test data and theoretical considerations, indicates a 30% drop on compressor speed within the first second after shutdown. A 30% loss in speed equates to approximately a loss in head of about

50%. The valve must, therefore, reduce the head across the compressor by about half in the same time as the compressor loses 30% of its speed.

Figure 9.13 also shows the deceleration of the engine's power turbine, and thus the driven compressor, following an ESD. Also depicted is the response of the recycle valve.

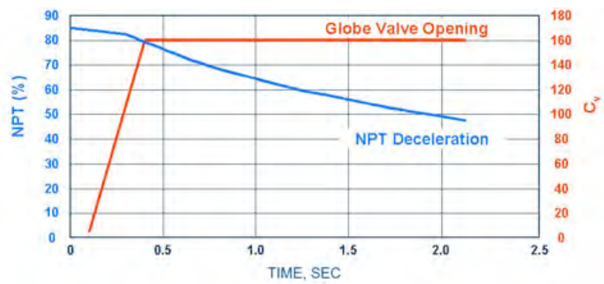

Figure 9.13 Train deceleration and valve opening

The larger the volumes are in the system, the longer it will take to equalize the pressures. Obviously, the larger the valve, the better its potential to avoid surge will be. However, the larger the valve, the poorer its controllability at partial recycle. The faster the valve can be opened, the more flow can pass through it. There are, however, limits to the valve opening speed, dictated by the need to control intermediate positions of the valve, as well as by practical limits to the power of the actuator. The situation may be improved by using a valve that is only boosted to open, thus combining high opening speed for surge avoidance with the capability to avoid oscillations by slow closing.

If the discharge volume is too large and the recycle valve cannot be designed to avoid surge, a short recycle loop (hot recycle valve) may be considered, where the recycle loop does not include the after-cooler.

While the behavior of the piping system can be predicted quite accurately, the question about the rate of deceleration for the compressor remains. It is possible to calculate the power consumption for a number of potential steady-state operating points. The operating points are imposed by the pressure in the discharge volume, which dictates the head of the compressor. For a given speed, this determines the flow that the compressor feeds into the discharge.

Referring back to Figure 9.12, another area of concern is the closing behavior of the check valve downstream of the compressor. Check valves are self-actuating, and the faster they close in the case of an emergency shutdown event, the better. Spring assisted-to-close check valves (all other things being equal) should close faster. Many check valves are available with an anti-slam feature. This feature slows the final travel of the check valve to avoid slamming. This feature will negatively impact surge avoidance.

9.8 Application

Solar has developed a software simulation program to rapidly evaluate the allowable piping volume (Figure 9.12) for the selected valve size.

The model iteratively determines the maximum allowable discharge volume for a given valve configuration. This is important, because the valve size can be determined early in the project phase. With a known valve configuration, the station designer can be provided with the maximum volume of piping and coolers between the compressor and the check valve that allows the system to avoid surge during an ESD.

The calculation requires specification of the head-flow-speed relationship of the compressor, and the definition of the surge line as a function of compressor speed, compressor head, or compressor flow. Further, the valve needs to be described by its maximum capacity (Cv), as well as by its capacity as a function of valve travel and the opening behavior, including the delay. The discharge check valve is assumed to be closed as soon as the recycle flow exceeds the compressor flow, that is, once the depressurization begins.

The fast stop simulation treats the check valve as ideal, and it is assumed to shut instantly when the flow through the check valve stops or is reversed.

The calculation procedure is started by initiating the deceleration of the train and the valve opening. For each time step, the compressor head and flow (based on speed and system pressures) and the flow through the valve (based on system pressures and valve opening) are calculated. The mass of the gas trapped between the recycle valve and compressor discharge is subsequently determined, yielding a new discharge pressure. If surge occurs, that is, if the flow drops below the flow at the surge line, the backwards flow through the compressor is assumed to increase with time in surge, with a recovery once the required head drops 1% below the head at surge. The modeling of the backwards flow is not critical, and is only made to avoid numerical instabilities, because the only information that is expected from the model is whether or not the compressor will surge for the given configuration.

Figures 9.14, 9.15, and 9.16 show typical results of these simulations. In Figure 9.14, the discharge volume is small enough, and while the actual flow of the compressor approaches the minimum allowable flow (surge flow) at about 500 ms after the initiation of the ESD, so surge can be avoided.

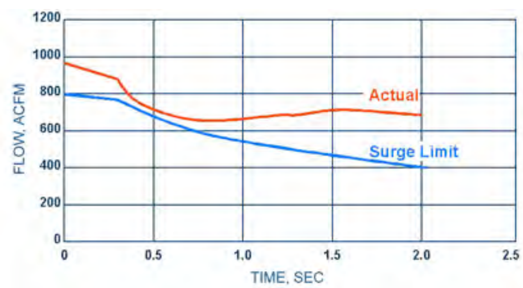

Figure 9.14 Actual flow and flow at the surge line during ESD – surge avoided

In Figure 9.15, the compressor surges about 700 ms after the initiation of the ESD. For this configuration, either the valve size has to be increased, or the discharge volume has to be reduced, to avoid compressor surge during an ESD.

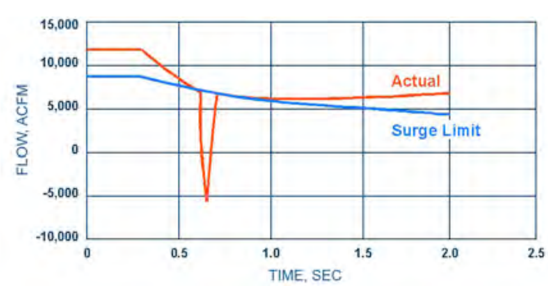

Figure 9.15 Actual flow and flow at the surge line during ESD – compressor surge at 0.7 sec.

In Figure 9.16, the system is severely under-designed and will require significant changes including the possible addition of another valve in a hot bypass mode.

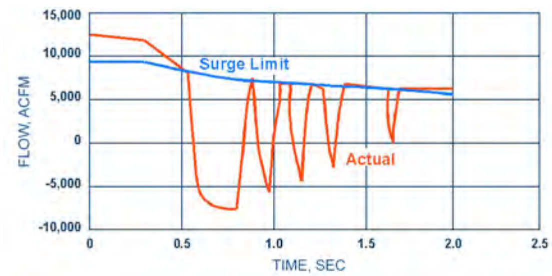

Figure 9.16 Actual flow and flow at the surge line during ESD – multiple surges

The most important point to consider in the design of surge avoidance systems is the realization that surge avoidance must be viewed in terms of the total system and not as an isolated item looking only at the compressor itself. Solar has made significant development in its surge avoidance systems and their application and can provide solutions for almost all conceivable valve and piping arrangements.

10. Compressor Ancillary Systems

10.0 OVERVIEW

All *Solar* compressors are sold in a complete compressor set configuration including driver, gearbox (if required), controls, and associated ancillary systems. The following describes the parts and ancillary systems provided with *Solar* compressor sets.

10.1 SEAL SYSTEMS

Dry gas seals are standard in all new *Solar* compressors. A dry gas seal gas system supplies and monitors seal gas to the dry gas seals; it also supplies and monitors buffer air or inert gas to the circumferential buffer seals. Figure 10.1 illustrates the system, consisting of filtration of the seal gas and buffer air or inert gas, regulation of their pressure and flow, and monitoring leakage.

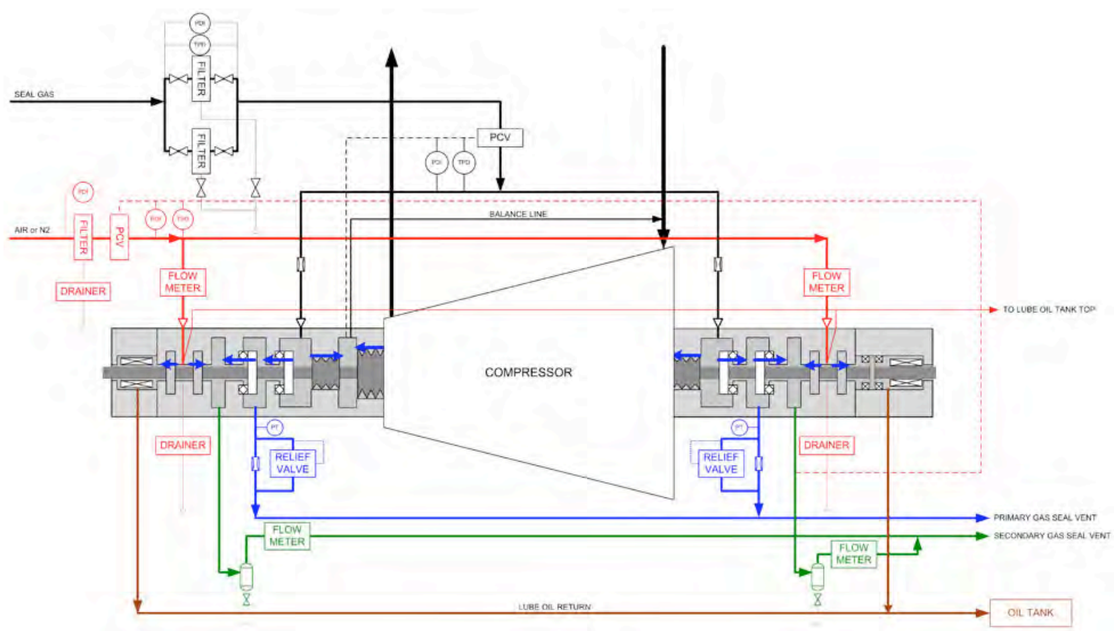

Figure 10.1 Typical dry gas seal system

The facility must be able to supply filtered buffer air or inert gas to the circumferential buffer seals at a rate that exceeds the controlled leakage of the two seal assemblies. The required pressure is

517 to 1345 kPag (75 to 195 psig). For details, see PIL 140. Buffer air is injected between a pair of seal rings at a regulated pressure of 172 to 207 kPa (25 to 30 psi) above the secondary seal/buffer air vent pressure. High flow alarm and shutdown switches can be used to protect the unit in the event of seal failure.

Leakage flow for the dry gas-face seal is a function of sealing pressure and rotational speed. Leakage flow for the circumferential seal remains constant with speed, but increases with the pressure across the seal.

10.2 LUBRICATION SYSTEM

When a turbine engine drives the compressor, the engine and the compressor share a lubrication system. When an electric motor drives the compressor, a lubrication system must be provided for the motor, gearbox and the compressor.

The lube oil system circulates pressurized oil to the various working parts of the gas turbine and driven equipment to provide lubrication and cooling. The system supplies lubricating oil, which conforms to Solar's engineering specification ES 9-224, from the lube oil tank located in the steel base frame. Proper oil temperatures are maintained by thermostatic oil control valves and an oil cooler.

The oil filters are supplied with a six-way transfer valve, a differential pressure indicator, and a differential pressure alarm. The system includes all piping and manifolds internal to the skid.

10.3 FLEXIBLE DISC COUPLINGS

Modern disk couplings (Figure 10.2) eliminate the need for lubrication, although lube oil may be present in the shaft cover. Hubs are attached to the rotating equipment with splines and interference-fit pilots. Hubs, sleeves and spacers are made of alloy steel, while flexible elements are corrosion-resistant stainless steel disks. The flexible disks transmit torque while accommodating axial offset and angular misalignment. The flexible disks are pre-stretched at installation to accommodate the axial growth during operation.

Figure 10.2 Typical compressor set coupling (image courtesy of Kopflex)

10.4 OPERATIONAL MONITORING

Full instrumentation for compressors is provided to monitor vibration, temperature, and pressure.

Bearing temperature is measured by 12 resistance temperature detectors (RTDs) installed in each compressor. There are four RTDs installed in each of the radial bearings and the thrust bearing, two in the upper portion and two in the lower portion of each bearing. One of each RTD pair is monitored; the other is a spare.

Six proximity probes are installed in each compressor to measure bearing vibration. Each radial bearing has two proximity probes; one is installed 45 degrees left of vertical and the other is installed 45 degrees right of vertical. The thrust bearing also has two proximity probes; one is installed 90 degrees left of vertical and the other is installed 90 degrees right of vertical. A seventh proximity probe, a Keyphasor signal, provides speed and phase rotation data for the compressor control system.

Suction and discharge pressure and temperature are monitored and evaluated for the need to annunciate a warning or to initiate a shutdown.

10.5 COMPRESSOR CONTROL

All compressor set control systems contain the basic yard valve sequencing and logic and basic operational monitoring fault detection, annunciation, and executing the appropriate response (shutdown). Additionally process control and surge avoidance controls are offered as an option.

Typical on-skid and remote control components are shown in Figures 10.3 through 10.5.

Figure 10.3 Typical on-skid control

Figure 10.4 Typical control room

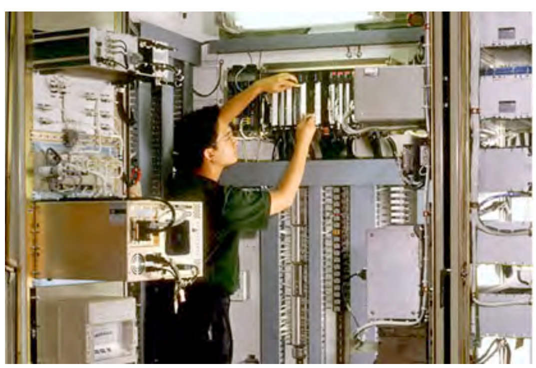

Figure 10.5 Control console interior components

Process Control

Standard compressor control options are: minimum suction pressure control, maximum discharge pressure control, and flow. Primary control is achieved by manipulating the driver. Secondary control can be achieved by recycling or throttling. Where multiple controls are selected decoupling and integrator anti-wind up is achieved by current frame error calculation and least-gating.

Surge Avoidance

Solar is a premier provider of surge avoidance systems and has been supplying surge avoidance systems since 1985. Detailed descriptions of surge avoidance and surge control options are contained in section 9.0 of this brochure.

10.6 PACKAGING

The gas turbine package is a completely integrated, fully operational package including a driver skid that mates to the compressor skid. The package is equipped with all accessories and auxiliary systems needed for normal operation when installed in, and connected to, suitable facilities. In addition, many optional features can be supplied to meet various installation and operation requirements. Figure 10.6 shows package assembly area in Solar's San Diego plant.

Figure 10.6 Package assembly area

Designed specifically for industrial service, the gas turbine package is a compact, lightweight unit that requires minimum floor space for installation. Proven packaging features greatly reduce installation costs, time, materials, and labor. Over the years, there has been a dramatic increase in the number of components requiring installation on a package to meet the technical, commercial and certification requirements. Figure 10.7 demonstrates how Solar accommodates this uniqueness while enhancing safety, accessibility, maintainability and commonality of its packages.

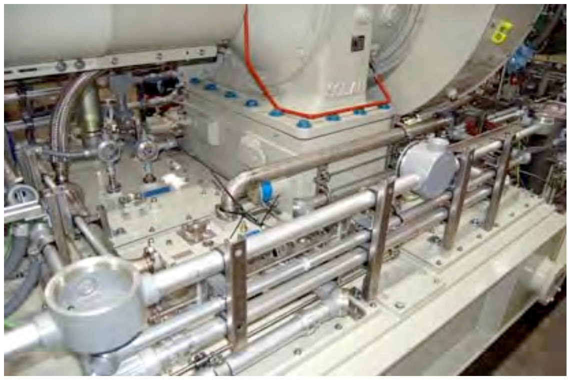

Figure 10.7a Previous package configuration

Figure 10.7b Improved access and maintainability

The basic gas turbine compressor set includes:

- Two-shaft industrial gas turbine
- Centrifugal gas compressor
- Gas turbine air inlet and exhaust collectors
- Gas turbine/compressor control
- Start system
- Fuel system
- Lubricating oil system
- Seal system
- Base skid with drip pans
- On-skid electrical system and wiring

The base frame is structural steel assembly with beam sections and cross members welded together to form a rigid foundation. Mechanical interface connection points for fuel, air, and water (for gas turbine cleaning) are conveniently located on the outer skid edge. Electrical connection points are made in on-skid junction boxes and terminal strips.

Package piping and manifolds are 316L stainless steel. This applies to all package piping systems including the start, fuel, and lube oil systems, as well as supply, drain and vent lines. In addition, the associated flange assembly hardware is 316 stainless steel.

The following items are not stainless steel, but may be specially ordered in stainless steel:

- Valve bodies or blocks and system functional components
- Pipe supporting hardware such as cushion clamps and brackets
- Oil tank cover assemblies with connection piping and fittings welded in place
- Sliding lube oil drain couplings and plates
- Pipe flexible couplings
- Filter housings
- Lube oil tank
- Lube oil cooler

Solar's *Turbotronic* control system is microprocessor-based and used for sequencing, control, and protection of the gas turbine package, and for providing an extensive range of options for package monitoring and plant control.

The control system is based on a commercially available programmable logic controller (PLC) configured to Solar's requirements. It provides an optimum combination of control and display features, reliability and maintainability, and is designed specifically for the control of gas turbines and centrifugal gas compressors. The control system also includes a number of sensors, transducers, and monitoring devices. Data are collected and sent to the PLC for computation and for generation of the required control actions and indications.

The PLC works in conjunction with the video display terminal (VDT) and permits a variety of advanced software and control options, condition and trend monitoring, and supervisory control. The system provides the operator with information necessary for operation of the equipment and offers a variety of communications options for data exchange with the customer's supervisory system. For applications in a Class I, Group D, Division 2 area, an on-skid control system is provided. This can be supplemented with an auxiliary control console that is mounted in a non-hazardous area, or it can be connected directly to the station control system. For applications in a Class I, Group D, Division 1 area, an off-skid control system is provided in a console that is mounted in a non-hazardous area.

10.7 FLOATING STRUCTURE EXPERIENCE

Solar’s rugged gas turbine compressor sets, generator sets and mechanical-drive packages have proven their outstanding reliability and availability on floating vessels and other offshore structures for more than 30 years. Solar has in-depth expertise gained by millions of turbomachinery operating hours on tension leg platforms (TLPs), spars, floating production storage and offloading (FPSOs), barges, semi-submersibles and compliant towers (CTs) in environments ranging from offshore Australia to Indonesia, the North Sea, Brazil, and the Gulf of Mexico.

Solar Turbines understands the unique demands placed on turbomachinery operating on floating production systems. Solar knows how to engineer turbomachinery packages and key systems to avoid possible complications caused by structural twisting and other constantly changing dynamic forces imposed upon floating structures or vessels by wind, waves and currents. For more details, refer to Solar publication SPFPS (Floating Production Systems). Figures 10.8 and 10.9 show floating production systems with *Solar* gas turbines and compressors installed.

Figure 10.8 Platforms in Gulf of Mexico

Figure 10.9 Floating production, storage and offloading vessel

11. Construction Services and Customer Support

11.0 CONSTRUCTION SERVICES

Solar's Construction Services organization is uniquely qualified to design turnkey systems for fluid compression, liquid pumping, and power generation systems, with single-source responsibility, engineering expertise, optimal economic designs, and detailed attention to quality and safety to ensure the customers are satisfied with their complete system.

Construction Services offers a comprehensive range of equipment and services to assist customers in successfully meeting their complete power system expectations. Solar has global experience, onshore and offshore, managing various types of power configurations and compression systems. Solar's services, based on years of experience and expertise in power system engineering and compression systems development, as well as complete project management, include:

- Feasibility studies
- Proposal preparation
- Design and engineering
- Material procurement
- Fabrication
- Onsite construction
- Quality control
- Scheduling
- Budget control
- Shipping
- Installation, testing, and commissioning

11.1 TurboFab

Solar's integrated fabrication and machining facility TurboFab (Figure 11.1) is located in Chanelview, Texas. TurboFab takes advantage of its 1,200 feet of waterfront access and 50-acre state-of-the-art facility to provide customers with total package solutions.

Figure 11.1 TurboFab facility

Solar's TurboFab facility can fabricate a wide variety of equipment support of gas compression and transmission applications. These include pipe and valve skids (Figure 11.2), process equipment skids (Figure 11.3), process modules and control rooms (Figure 11.4), and arctic modules (Figure 11.5). Land base compressor stations, pipe-rack skids, rotating equipment skids, pump skids, fuel gas skids, fuel gas conditioning systems, liquid fuel treatment and processing systems, rotating equipment bases, gas turbine bases, power generation modules, steam processing systems, as well as cost effective refurbished equipment can also be constructed at this facility.

Figure 11.2 Pipe/valve skid

Figure 11.3 Process skid

Figure 11.4 Process module and control room

Figure 11.5 Arctic enclosure

In 2007, TurboFab shipped the largest single lift module Solar had ever constructed (Figure 11.6). The module measures approximately 100’ wide by 110’ long by 45’ high, and weighs 1,430 short tons. Onboard are two *Mars* 100 gas turbine compressor sets that drive *Solar* centrifugal gas compressors.

Figure 11.6 Single lift module

Figure 11.7 shows TurboFab’s production capabilities, including fabrication, pipe spooling, process skid and turbine base assembly, painting, testing, electrical assembly, and instrumentation. Production facilities include: 2,580,000 square feet of all-weather work space, heavy fabrication bay (180'L x 100'W x 80'H), pipe spooling bay (60'L x 60'W x 40'H), process skid bay (65'L x 60'W x 80'H), CNC milling machine (150-ton working capacity), and the paint and blast shop (8,800 square feet).

Figure 11.7 TurboFab production facilities

11.2 CUSTOMER SUPPORT

In conjunction with Solar gas compressors and gas compression packages, Solar can provide commissioning, training, compression service, field service, parts support, restaging, repair, overhaul, upgrade, package refurbishment, operation and maintenance, engineering, procurement, construction, asset management, contract power, and financing.

Solar is committed to expanding the gas compressor product portfolio to match the power and speed of Solar's gas turbine products. Solar's latest gas compressor products are designed for greater efficiency and wider performance ranges. This expansion of performance and flexibility means the number of possible impeller configurations has evolved into an array of staging choices for the latest gas compressor models.

Gas Compressor Spares Philosophy

Solar has developed three spares packages for gas compressors to increase system availability and reduce the risk of extended outages. Depending on equipment criticality, customers can select the appropriate support package to meet their target availability. It is recommended that customers carefully evaluate the business risk associated with compressor emergencies to decide which spares package meets the operational needs and availability targets of the application since Solar does not stock all possible compressor components. Each increasing level of gas compressor spares packages provides for more comprehensive compressor hardware and tooling that coincides with increasing availability of the gas compressor package.

All three packages include the gas compressor tooling to support the execution of installing the chosen spares option. Customer-owned tooling is an integral part of each compressor spares package. *Solar* gas compressor tooling sets are often in use to support the demand for overhauls and restages worldwide. Therefore, the availability of Solar's tooling sets is limited as the population of these compressors continues to grow.

Table 11.1 summarizes the spares packages components with corresponding tooling while Table 11.2 lists the latest generation compressor models supported by the spares packages.

Table 11.1 Gas compressor spares support packages

Level	Spare Bundle	Spare Impeller	Spare Bearings, Seals & Maint. Items	*Module Tooling	Bearing & Seal Tooling
Spares Package. III Spare Bundle	**X**	Included In Module	**X**	**X**	Included In Module Tools
Spares Package. II Spare Impeller		**X**	**X**	**X**	Included In Module Tools
Spares Package. I Spare Bearings, Seals & Maint. Items			**X**		**X**

**Module tooling provides for removal and installation of the module and includes Bearing and Seal Tooling*

Table 11.2 Compressor models supported by spares packages

Pipeline Compressors	**C40P**	**C45**	**C65**	**C85**
Production Compressors	**C40M**	**C51**	**C61**	

Spares Package III - comprises a spare gas compressor bundle, dry gas seals and bearings, all bundle removal tooling including bearing and seal tools. Figure 11.8 shows bundle removal /installation with bundle removal tooling.

Figure 11.8 Bundle removal/installation with tooling

The spare gas compressor bundle consists of a balanced rotor assembly with shafts, impellers and spacers as well as stationary components including guide vanes and stators. Exchanging a spare bundle can minimize downtime to the few days that are required to remove and install the spare bundle. Solar offers long-term storage containers with handling fixtures for spare bundles.

Spares Package II - includes a spare first stage impeller, bundle removal tooling, and spare bearings and dry gas seals. This package offers lower capital expenditures than a spare bundle, but requires longer corresponding outage time. This downtime may vary depending on the customer's geographic location, as the bundle must be returned to a Solar overhaul facility to complete the repair.

At the time of repair the customer's bundle, along with the spare impeller, is sent to a Solar overhaul facility. The bundle is disassembled and the rotor is rebuilt and dynamically balanced with the spare impeller. The bundle is re-assembled and returned to the customer site.

Experience has shown that catastrophic compressor failures due to foreign object ingestion are typically on the suction side of the compressor. While stationary components are usually repairable, damaged impellers (Figure 11.9), cannot always be successfully repaired. To mitigate the risk of these occurrences, Solar offers Spares Package II as an option. Due to the modular design of *Solar* gas compressors, spare impellers can be exchanged for the damaged component and re-assembled into a module in a shop environment.

Figure 11.9 Damaged impellers

Spares Package I - components are typical maintenance-type spares and are a standard offering for all compressor set Recommended Service Parts Lists (RSPLs).

This package includes gas compressor spare bearings and dry gas seals with the tooling for removal and installation. Spare bearings and dry gas seals may take several weeks for delivery, therefore, maintaining Spares Package I components is recommended to increase availability. Spares Package I also includes compressor skid maintenance items such as dry seal filters, package and instrumentation components.

Another option is stocking a dynamically balanced spare rotor. This option does not include all the stationary components that are provided in a complete spare bundle. Therefore this option requires a more extensive set of tooling, procedures and expertise to incorporate the rotor into a bundle configuration that can be installed in the compressor case.

A spare rotor (Figure 11.10), can require less initial capital expenditure than a bundle (Figure 11.11), however the increased cost for the higher level of tooling, and the time required to assemble the rotor into a bundle configuration can quickly offset the initial capital savings. Also, during the rotor-to-bundle build process, a specialized set of assembly procedures must be followed to ensure proper rotor balance and operation of the gas compressor. It is highly recommended that a qualified *Solar* gas compressor technician supervise this task in a shop environment.

Figure 11.10 Rotor configuration

Figure 11.11 Bundle configuration

11.3 GAS COMPRESSOR RESTAGING

Normal usage and change in operating conditions – gas flow, pressure, temperature and composition – affects equipment productivity or range of operation. Restaging restores compressors to optimum performance by changing the gas flow and head characteristics to optimize efficiency around a new set of operating conditions (Figure 11.12). It is a highly cost-effective way to optimize compression systems.

Figure 11.12 Restaging procedure

How Restaging Improves Performance

Improve Surge Margin (Figure 11.13) - Flow through the compressor is insufficient for the current staging, and the operating point has moved left toward the surge line. To maintain stability, the compressor must recycle gas back through the gas cooler and return it to the suction side. Consequently, the unit operates at a higher flow than it is actually sending downstream and fuel is wasted. This compressor would be restaged to a lower-flow configuration, thus placing the operating point in the middle of the map with satisfactory surge margin.

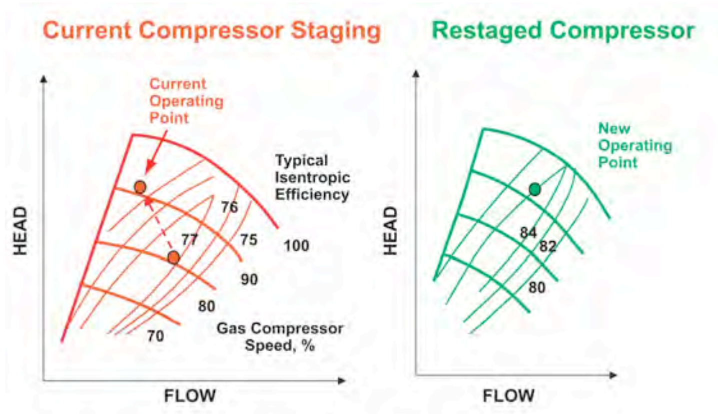

Figure 11.13 Improve Surge Margin

Improve Efficiency (Figure 11.14) - The performance scenario detailed below shows higher flow demand and lower head requirement than the original design. There are now too many stages and the speed has declined, forcing the compressor to operate at lower efficiency and the turbine to operate with a higher, off-optimum speed loss. A restage would drop one or more stages in combination with a change to the remaining stages, moving the map over the current operating point and restoring lost compressor and turbine efficiency.

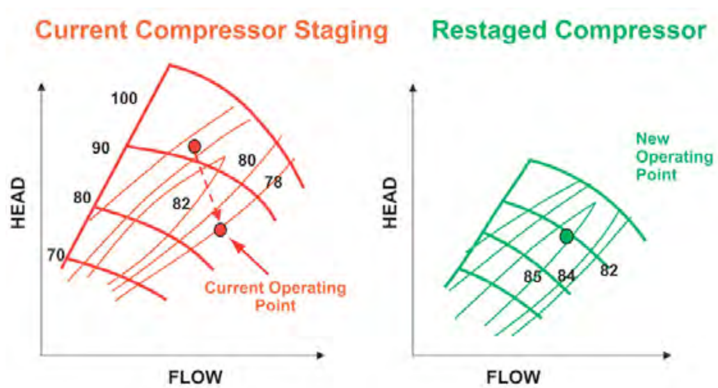

Figure 11.14 Improve Efficiency

Increase Head-Making Capability (Figure 11.15) - Inlet pressure is decreasing, yet discharge pressure must be held constant. The unit, therefore, must operate at increasing speed. Eventually, the unit will reach its maximum speed and will be unable to make the required head. The solution is to add more stages to accommodate the lower inlet pressure and higher head requirement while maintaining the required discharge pressure.

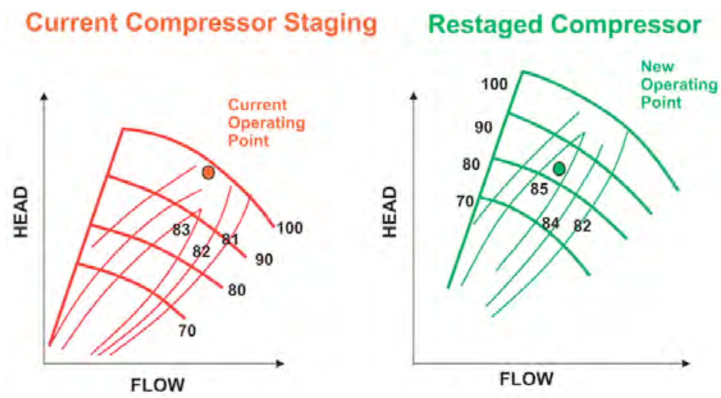

Figure 11.15 Increase head-making capability

11.4 MACHINERY MANAGEMENT SOLUTIONS

Solar's Machinery Management Services (Figure 11.16) is now employing *InSight System*™ to provide the most comprehensive approach to equipment health management ever developed, enabling a transition from time-based to a condition-based maintenance program. *InSight System* employs web-enabled technologies, processes and people in order to deliver high value Machinery Management Solutions. These solutions are custom fit to the customer's needs and business objectives and are designed to increase Life Cycle Value while decreasing Life Cycle Costs for customer owned Solar equipment. *InSight System* monitors, gathers and analyzes information to help customers make decisions by delivering these enabling capabilities.

Figure 11.16 Machinery Management Services

Remote Monitoring and Diagnostics (RM&D) - combines advanced diagnostics, condition monitoring, remote troubleshooting, email alert notifications, predictive recommendations, and equipment operation summary reports. Dianostic data from gas turbine and related equipment is automatically analyzed, trended, and posted on the web utilizing a centralized data management process. This data provides early warning of adverse trends and reduces unplanned shutdowns and repair trips The system provides automatic updating of new or updated data allowing rapid deployment best practices and fleet knowledge.

Troubleshooting (LiveView*)* - is accomplished through real-time graphical imaging from LiveView^{SM}, which provides a remote view the control panel and operations in progress. This capability reduces the requirement to send a field engineer to the site, which allows immediate technical and troubleshooting support. This system allows download of high-resolution, real-time data files over the Internet for advanced analysis. This information facilitates more confident and objective decision-making and provides the ability to get simultaneous input from experts in other locations.

Certified Condition Assessments - OEM certified condition assessment visits are conducted to assess equipment physical condition and operating environment. The reports generated by these visits, make recommendations for extended operation and/or corrective actions.

Secure Connectivity Solutions via InSight Connect - *InSight System* tools rely upon a dedicated connectivity solution that allows for secured access and standardizes data acquisition and transmission. *InSight Connect* delivers a secure, standard connectivity solution ensuring the highest level of reliability over a wide variety of connections.

Information Management - provides owners, operators, and maintenance and support personnel immediate web access to accurate and reliable technical information, such as operation and maintenance manuals, service bulletins, project photographs, illustrated parts listings, and additional types of technical documentation.

Collaborative Communication - provides communication and information management. The Collaborative Workspace can be thought of as a web-based virtual global conference room that permits geographically dispersed personnel, from both the customer and Solar, to communicate efficiently. The system stores and manages fleet data and statistics, management reports, technical reports, key performance indicators, schedules, drawings, software, test reports, and equipment assessments.

Maintenance Management - capabilities include the implementation of fleet-wide best practices and equipment-specific maintenance plans within a computerized system. Solar has developed the ability to link diagnostic outputs to prompt maintenance events. Advanced diagnostics and condition assessments combined with targeted maintenance result in extended run periods and lengthened overhaul cycles.

Appendix - Measurement Conversions

Equivalences Between Customary, Metric, and S.I. Units

Basic Equivalences			
1 in.	= 0.02540 m	1 m	= 39.37 in.
1 sq. in.	= 6.452 cm^2 = 0.0006452 m^2	1 cm^2	= 0.1550 sq. in.
1 ft.	= 0.3048 m	1 m	= 3.281 ft.
1 $ft.^2$	= 0.09290 m^2	1 m2	= 10.76ft^2
1 $ft.^3$	= 0.02832 m^3	1 m^3	= 35.32 ft^3
1 lb_m	= 0.4536 kg_m	1 kg_m	= 2.205 lb_m
1 lb_f	= 4.448 Newton	1 Newton	= 0.2248 lb_f
1 lb/sec	= 0.4536 kg/s	1 kg/s	= 2.205 lb/s
1 lb/sec	= 1633 kg/hr	1 kg/hr	= 0.0006124 lb/sec
1 ft-lb_f/$lb_{m°}$°R	= 0.005381 kJ/kg°K	1 kJ/kg°K	= 185.8 ft.lb_f/lb_m°R
1 ft-lb	= 1.356 J	1 J	= 0.7376 ft-lb
1 ft-lb	= 0.1383 kg.m	1 kgm	= 7.231 ft-lb
1 ft-lb/sec	= 0.1383 kg.m/s	1 kgm/s	= 7.231 ft-lb
1 ft/sec	= 0.00508 m/s	1 m/s	= 196.8 ft/sec
1 BTU/lb_m	= 2.326 kJ/kg	1 kJ/kg	=0.4299 BTU/lb_m
1 Btu/lb_m	= 0.5556 kCal/kg	1 kcal/kg	= 1.800 Btu/lb
1 Btu/ lb_m °R	= 1 kCal/kg.°K	1 kcal/kg.°K	= 1 Btu/lb°R
1 kCal/kg. °K	= 4.186 kJ/kg °K	1 kJ/kg. °K	= 0.2389 kCal/kg.°K
1 J/m	= 1 Newton	1 Newton	= 1 J/m
1 J/kg. °K	= 0.1020 m. kg_f/kg_m°K	1 metric slug	= 1 kg_f/m.s^2
1 lb_m	= 32.17 slugs	1 kg_m	= 14.59 slugs
1 lb-mol	= 379.56 SCF		
1 MMSCFD	= 109.81 lb-mol/hr		
1 Ton Refrigeration = 200 Btu/min = 3.333 Btu/sec			

Pressure Equivalences

1 psi	= 27.70 in. H_2O = 6.895 kPa = 6895 Pa	= 13.85 King = 0.06895 bars = 0.07030 kg/cm^2	= 0.06805 atm = 0.1069 in. T.B.E. (Tetro-Bromi-Soultions) = 6895 N/m^2	= 20.36 in. Hg = 0.07030 ata
1 in. H_2O (at 60°F)	= 0.03609 psig = 0.00249 bar	= 0.07349 in. Hg = 0.249 kPa	= 0.5 King = 249 N/m_2	= 0.002456 atm
1 in. Hg (at 32°F)	= 0.4912 psig = 0.0339 bar	= 13.61 in. H_2O = 3.39 kPa	= 0.03342 atm = 3390 N/m^2	= 6.805 King
1 atm	= 14.70 psi = 1.013 bar	= 407 in. H_2O = 101,300 N/m^2	= 29.93 in. Hg = 1.033 kg/cm^2	= 101.3 kPa
1 bar	= 14.504 psig = 100 kPa	= 402 in. H_2O = 100,000 N/m^2	= 29.5 in. Hg = 1.020 kg/cm^2	= 0.987 atm
1 ata	= 1 kg/cm^2	= 14.223 psig		
1 Pa	= 0.000145 psig	= 1 Newton/m^2	= 0.1019 kg/m^2	= 1.019 x 10^{-5} kg/cm^2
1 kPa	= 0.145 psig = 0.0100 bar	= 4.20 in. H_2O = 1000 N/m^2	= 0.295 in. Hg = 0.01019 kg/cm^2	= 9.87 E-03 atm = 101.0 kg/m^2
1 MPa	= 145 psig			
1 kg/cm^2	= 14.22 psig	= 0.9810 bar	= 98.10 kPa	

Temperature	To Convert Inch Hg to PSI Multiply by	To Convert Inch H_2O to PSI Multiply by	To Convert Inch Hg to Inch H_2O Multiply by
At 32°F	0.4912	0.03612	13.6
50°F	0.4903	0.03611	13.6
60°F	0.4898	0.03609	13.6
70°F	0.4893	0.03605	13.6
80°F	0.4888	0.03600	13.6
90°F	0.4883	0.03594	13.6
100°F	0.4878	0303586	13.6

PSI = in. Hg x [0.49272 – T (°F) x 0.000049]
PSI = in. H_2O x [0.03604 + T x 4.7 x 10^{-6} - T^2 x 6.5 x 10^{-8}]

Mass Flow Equivalences

1 kg/hr	=	0.0006125 lb/sec
1 kg/min	=	0.03675 lb/sec
1 kg/s	=	2.205 lb/sec
1 lb/sec	=	1633 kg/hr
1 lb/sec	=	27.22 kg/min
1 lb/sec	=	0.4536 kg/s

Volume Flow Equivalences

1 m^3/s	=	2119 CFM
1 m^3/min	=	35.32 CFM
1 m^3/hr	=	0.5887 CFM
1 CFM	=	0.0004719 m^3/s
1 CFM	=	0.02831 m^3/min
1 CFM	=	1.699 m^3/hr
1 CFM	=	0.001440 MMCFD
1 MMCFD	=	694.4 CFM

Head Equivalences

1 m-kg_t/kg_m	=	3.281 ft- lb_f/lb_m
1kJ/kg	=	334.6 ft-lb_f/lb_m
1 J/kg	=	0.3346 ft-lb_f/lb_m
1 N-m/kg	=	0.3346 ft-lb_f/lb_m
1 kN-m/kg	=	334.6 ft-lb_f/lb_m
1 ft-lb_f/lb_m	=	0.3048 m-kg_f/kg_m
1 ft-lb_f/lb_m	=	0.002989 kJ/kg
1 ft-lb_f/lb_m	=	2.989 J/kg
1 ft-lb_f/lb_m	=	2.989 N-m/kg
1 ft-lb_f/lb_m	=	0.002989 kN-m/kg
1 kJ/kg	=	102 m-kg_f/kg_m
1 m-kg_f/kg_m	=	0.009806 kJ/kg

Work Equivalences

1 Btu	= 778.3 ft-lb = 0.0002931 kW-hr = 107.6 kg-m	= 0.2520 kcal = 1055 J = 0.0003985 CV	= 0.0003930 hp-hr
1 ft-lb	= 0.001285 Btu = 5.050 x 10^{-7} hp-hr = 1.356 J	= 0.000238 kCal = 3.766 x 10^{-7} kW-hr = 0.1383 kg-m	
1 kCal	= 3088 ft-lb = 0.001163 kW-hr = 426.9 kg-m	= 3.968 Btu = 4187 J = 0.001581 CV-hr	= 0.001560 hp-hr
1 hp-hr	= 2544 Btu = 0.7457 kW-hr = 2.737 x 10^5 kg-m	= 1.98 x 10^6 ft-lb = 2.684 x 10^6 J = 1.014 CV-hr	= 641.2 kCal
1 kW-hr	= 3412 Btu = 1.341 hp-hr = 3.671 10^5 kg-m	= 2.655 x 10^6 ft-lb = 3600 kJ = 1.360 CV-hr	= 859.9 kCal
1 kg-m	= 9.807 J = 0.03704 x 10^{-5} CV-hr = 0.002342 kCal	= 7.233 ft-lb = 0.3653 x 10^{-5} hp-hr = 0.009295 Btu	= 0.2724 x 10^{-5} kW-hr
1 J	= 0.1020 kg-m = 0.3777 x 10^{-6} CV-hr = 0.2388 x 10^{-3} kCal	= 0.7376 ft-lb = 0.3725 x 10^{-6} hp-hr = 0.9478 x 10^{-3} Btu	= 0.2778 x 10^{-6} hp-hr

Power Equivalences

1 hp	= 0.7457 kW = 0.1781 kCal/s	= 0.7068 Btu/sec = 76.06 kg-m/s	= 550 ft-lb/sec = 1.014 C.V.
1 kW	= 1.341 hp = 0.2388 kcal/s	= 0.9478 Btu/sec = 102.0 kg-m/s	= 737.6 ft-lb/sec = 1.360 C.V.
1 Btu/sec	= 1.415 hp = 0.02520 kCal/s	= 1.055 kW = 107.6 kg-m/s	= 778.3 ft-lb/sec
1 ft-lb/sec	= 0.00182 hp = 0.0003238 kCal/s	= 0.001356 kW = 0.1383 kg-m/s	= 0.001285 Btu/sec
1 kg-m/s	= 7.231 ft-lb/sec = 0.002343 kCal/s	= 0.01315 hp = 0.009295 Btu/sec	= 0.009804 kW
1 kcal/s	= 3.968 Btu/sec = 3088 ft-lb/sec	= 5.615 hp = 426.9 kg-m/s	= 4.187 kW
1 C.V. (France, Latin America) 1 C.V.	 = 0.9863 hp = 542.5 ft-lb/sec	= 1 PS (Germany) = 0.7355 kW = 0.1757 kCal/s	 = 75 kg-m/s = 0.6971 Btu/sec

Temperature Equivalences

$$°F = °C \times 1.8 + 32$$

$$°C = (°F - 32) \times \frac{5}{9}$$

$$°R = °F + 459.7$$

$$°K = °C + 273.2$$

$$°R = °K \times 1.8$$

Standard Gravity Acceleration Equivalence

$$g = 32.17 \frac{ft}{sec^2} = 9.807\ m/s^2$$

Note: g appears in equations using customary or metric units to harmonize the units for mass (kg_m, lbm) and force (kg_f, lbf)

Viscosity Equivalences

Absolute or Dynamic Viscosity:			
1 lb/sec.ft	= 1.488 kg/s.m	= 1.488 Pa.s	= 1488 centipoise
1 kg/s.m	= 0.672 lb/sec.ft	= 1 Pa.s	= 100 centipoise
1 centipoise	= 0.000672 lb/sec.ft	= 0.001 kg/s.m	= 0.001 Pa.s
1 kPa.s	= $1x10^6$ centipoise		

Kinematic Viscosity:		
1 cm^2/s	= 0.001076 ft^2/sec	= 1 stoke
1 m^2/s	= 10.76 ft^2/sec	= 10,000 stokes
1 ft^2/sec	= 929.0 cm^2/s	= 0.001076 ft^2/sec
1 centistoke	= 1 x 10^{-6} m^2/s	= 0.00001076 ft^2/sec

Gas Constant Equivalences

m-kg_f/kg_m °K	= 1.823 ft-lb_f/lb_m °R	= 9.804 J/kg °K
kJ/kg °K	= 185.9 ft-lb_f/lb_m °R	= 102 m-kg_f/kg_m °K
Ft-lb_f/lb_m °R	= 5.379 J/kg °K	= 0.5485 m-kg_f/kg_m °K

Typical Performance Data in Customary, Metric, and S.I. Units

P_1	145 psia	10.00 bara	1000 kPa
P_2	373.4 psia	25.74 bara	2574 kPa
H_{isen}	44,233 ft-lb_f/lb_m	13,482 m-kg_f/kg_m	132.212 kJ/kg
Q_1	3129 CFM	5316 m^3/h	1.477 m^3/s
SQ	45.3 MMSCFD	1214 nm^3/d –e3	1214 nm^3/d –e3
HP	2733 hp	2038 kW	2038 kW
N	15153 rpm	15153 rpm	15153 min^{-1}
$P_{2/}$ P_1	2.574	2.574	2.574
T_1	59.0°F	15.0 °C	15.0 °C
T_2	218.6°F	103.7 °C	103.7 °C
Ç isen	0.783	0.783	0.783
SM	0.336	0.336	0.336
S.G.	0.650	0.650	0.650
k_{50}	1.315	1.315	1.315
k_{300}	1.257	1.257	1.257
Z_1	0.9838	0.9838	0.9838

Typical Performance Data in Customary, Metric, and S.I. Units (cont'd)

Z_2	0.9846	0.9846	0.9846
k_1	1.313	1.313	1.313
k_2	1.273	1.273	1.273
1 bara	= 100 kPa	1 kPa	= 0.01 bara
1 bara	= 14.50 psia	1 psia	= 0.06895 bara
1 m-kg_f/kg_m	= 3.281 ft-lb_f/lb_m	1 ft-lb_f/lb_m	= 0.3048 m-kg_f/kg_m
1 J/kg	= 0.3345 ft-lb_f/lb_m	1 ft-lb_f/lb_m	= 2.989 J/kg
1 m^3/hr	= 0.5886 CFM	1 CFM	= 1.699 m^3/hr
1 m^3/s	= 2119 CFM	1 CFM	= 0.0004719 m^3/s
1 nm^3/d – e3	= 0.03733 MMSCFD	1 MMSCFD	= 26790 nm^3/d
1 kW	= 1.341 hp	1 hp	= 0.7457 kW
1 K	= 1.8 °R	1 °R	= K x 5/9

Solar Turbines
A Caterpillar Company

FOR MORE INFORMATION

www.solarturbines.com

Solar Turbines Incorporated
P.O. Box 85376
San Diego, CA 92186-5376 USA
Tel: (+1) 619-544-5352