Centrifugal Compressor Evolution

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Author’s Note: This article traces the evolution of centrifugal compressors from the relatively primitive machines of the early 20th century to the sophisticated turbomachines of the 21st century. The majority of the advances in centrifugal compressors are the result of two primary enablers: improved manufacturing techniques and advanced analytical tools.

Manufacturing methods

In early process centrifugal compressor development, design choices were limited by the manufacturing methods available. Machining techniques were limited to turning and three-axis milling that could produce fairly simple, 2-D style blades. While adequate for many compressor applications, they are inadequate for high-flow and/or high Mach number machines. For the sophisticated shapes for higher-flow applications, OEMs were forced to use welded fabrications or castings.

Through the 1950s, castings were used for compressor stationary components and cases. While offering some cost advantages, the rough surfaces and poor dimensional accuracy negatively impacted aerodynamic performance.

As energy costs increased, OEMs were forced to develop more efficient compressors. The general trend in peak compressor efficiency over the past 50 years is shown in Figure 1. Efficiencies are rapidly approaching the widely accepted efficiency limit of approximately 92% polytropic efficiency (the red line).

Impellers

Attaining high performance in a centrifugal compressor demands superior aerodynamics, and the most critical aerodynamic component is the impeller. A large portion of the efficiency gain in recent years has been due to the sophisticated impeller designs made possible by advanced manufacturing and analytical practices.

Impellers can be broken down in to two major categories: shrouded (or covered) and unshrouded (open). Most beam-style centrifugal compressors used covered impellers while integrally geared compressors tend to favor the open designs.

Impellers can be further subdivided based on the blading style with the blades being categorized as 2-D or 3-D. In general, the impeller flow rate drives the blade style. 2-D blading is appropriate for low-flow designs while high-flow designs require 3-D blades to achieve optimal performance. Impellers with 2-D blades can be fabricated using three-axis milling but it is very difficult to create a 3-D bladed impeller with a three-axis mill.

Initially, 2-D impeller blades were milled to the disk with the cover riveted in place. In the mid-1950s some OEMs began to weld such impellers but conventional fillet welding required an impeller blade height of 0.7 in. (15.25 mm) or greater. This limited the minimum flow rate achievable with the welded designs. This limitation was overcome with the development of brazing, electron beam welding and the like, as well as with electro-discharge machining (EDM), electro-chemical machining (ECM) and powdered metal technologies.

For high-flow coefficient impellers, the large flow angle variation at the impeller inlet mandates the use of three-dimensional blading. The variation is due to the change in peripheral and meridional velocity at the impeller inlet resulting from the change in radius and local curvature. Many blades were formed as sections of geometric shapes such as cones or cylinders. However, achieving the highest attainable efficiencies required more complex arbitrary (free-form) blade shapes defined by line elements in space or a mesh of points. Blades were cast or formed via die-pressing or the like and then welded into position. Deviations due to “spring-back” and heat input during the pressing and welding processes compromised performance.

The advent of flank and point five-axis milling eliminated

Figure 1. Efficiency trends

Figure 2. Low solidity vaned diffuser (LSD)
the need for die-pressing or forming as the blades are milled from a disk forging. Positioning of the milled blades was more precise and the amount of welding was also reduced.

Milling technology today allows covered impellers to be milled from a single forging, eliminating the need for welding. In general, the so-called “integral joint” yields a more robust impeller.

**Diffusers**

Diffusers, the second most important aerodynamic component, convert the kinetic energy (velocity pressure), leaving the impeller into static pressure (potential energy) and further reducing the volumetric flow. Diffusers fall in two broad categories: vaneless and vaned.

In general, vaneless diffusers offer the widest flow range because there are no vanes to interfere with the gas as it passes through the diffuser. However, the static pressure recovery in vaneless diffusers is not as high as in its vaned counterparts. Vaneless diffusers were dominant in early centrifugal compressors because of its ease of fabrication; i.e., the walls were formed via turning. However, the limited peak static pressure recovery restricted the peak efficiency achievable with vaneless diffusers.

In the late 1980s, some OEMs began applying low soli-dity vaned diffuser (LSDs), which have no true geometric throat (Figure 2). Researchers touted the advantages of LSDs (see Senoo et al [1983] and Osborne and Sorokes [1988]). LSDs provide nearly the same operating range as a vaneless diffuser yet provide greater pressure recovery and, therefore, higher stage efficiency. The introduction of the LSD provided a step change in stage efficiency without significantly reducing the flow range.

**Other components: Inlet guides, return channels, volutes, inlets, sidestreams and casings**

The stationary components in most early centrifugals were constructed via casting. Typical associated issues included core shifts, varying vane thicknesses and rough surface finishes — all contributed to increased losses and greater uncertainty in performance predictions.

To address this and other similar issues, OEMs began machining the stationary flow path components. By the year 2000, OEMs were building compressors with nearly 100% fabricated/machined internal components compared to the 1950s when nearly 100% of the components were cast.

**Aerodynamic analytical techniques**

The evolution in analytical methods has also been a major enabler in advancing aerodynamic technology, which, in turn, is the result of the advances in computer technology. Sophisticated analyses that took days can now be completed in hours, resulting in more precise modeling of individual components or entire compressors.

**1-D methods**

The most common approach used in 1-D models is the so-called “velocity triangle” methodology. Formulations based on the Euler turbomachinery equation, the Bernoulli equation, conservation of mass, conservation of angular momentum, and other empirical performance models are used to solve for the meridional, tangential, and relative velocities and flow angles at various key locations within a centrifugal stage; i.e., the inlet and/or exit of flow path components.

**2-D methods**

Introduced commercially in the late 1950s, two-dimensional methods improved the designer’s ability to access the aerodynamic quality of a new design. The most common 2-D method is the streamline curvature approach that breaks the flow passage into “streamtubes” of constant mass flow, such as shown in Figure 3. Velocities are calculated based on the local curvatures in the meridional (or hub-shroud) profiles and the mass flow through the streamtube area.

**3-D methods**

Three-dimensional computational fluid dynamics (CFD) codes are the most rigorous analytical techniques used to calculate flow through aerodynamic components. First widely available to the industrial compressor industry in the late 1980s, such codes provided a major step forward in
the ability to understand the flow physics inside the rotating impeller and stationary components as well as the interactions between rotating and stationary components.

CFD analyses are performed using computational grids that break the flow passage up into small polyhedrons (e.g., hexahedrons or tetrahedrons), essentially the aerodynamic equivalent of finite element analysis. Such codes account for all facets of the aerodynamic component geometry and provide far more insight into the flow physics in the compressor flow path, leading to higher performance. Some typical CFD results are shown in Figure 4.

The discussion will now turn to the evolution of compressor mechanical technology.

Undamped critical speed analysis

In the mid-1940s, Myklestad developed a method of calculating modes of uncoupled bending vibration of an airplane wing (Myklestad, 1944). One year later, Prohl developed a general method for calculating critical speeds of flexible rotors (Prohl, 1945). The Myklestad-Prohl method is a transfer matrix solution technique at the heart of the undamped critical speed map commonly used today. A rotordynamic analyst uses an undamped critical speed map to analytically determine the location of the rotor’s natural frequencies as a function of the bearing support system (Figure 5). From the late 1940s though the 1960s the calculation of the first critical speed (nC1) was conducted by hand to ensure the running speed range of the compressor did not coincide with nC1.

Synchronous unbalance response

In 1965, Lund released Part V of a report prepared for the U.S. Air Force Aero Propulsions Laboratory (Lund, 1965). This landmark publication contained a computer program to determine the unbalance response analysis of a rotor on fluid film bearings. These tools, along with bearing and seal programs that are used to determine oil film stiffness and damping, enabled more advanced analysis on rotors. The forced response tool allowed the designer to apply unbalance to the rotor to calculate the rotor’s response to unbalance, the location of NC1 and the amplification factor (API 617, 2002) and (Childs, 1993).

Rotor stability analysis

In the early 1970s, a series of field stability problems arose in high-pressure compressors. In 1974, Lund published a breakthrough paper on the analysis of rotor stability (Lund, 1974). The rotordynamics stability code provided the added insight of logarithmic decrement of the first vibration mode. The log dec of a system can be characterized in the time domain as the amplitude of successive amplitude peaks. If the amplitudes are decaying over time the log dec is positive (see Figure 6); if growing the log dec is negative (Wachel, 1975), (Kuzdzal, 1994), (Ramesh, 2004). Certainly, a negative log dec is undesirable.

Hydrodynamic oil film bearing

In the 1970s and 1980s, the sources of instability were difficult to quantify. As hydrodynamic bearing technology advanced, designers recognized the need to optimize the plain sleeve bearing. In general, a plain sleeve bearing has a comparatively large load carrying capacity, but performed poorly in rotor dynamics stability, often being the source of the issue. Oil whirl (Pinkus, 1956), (Newkirk, 1956) was a common term used in the 1960s to characterize the maximum speed the bearing could operate at before it generated undesirable instability forces. Resonant whip was used to describe when the unstable frequency locked on to the first natural frequency. Designers in that era worked to modify the inner geometry of the journal bearing to control the oil whirl and whip characteristics with the intent of increasing the bearing instability threshold speed (Figure 7).

Although tilt pad thrust bearings were invented by Mitchell in 1905 and Albert Kingsbury in 1907, tilt pad journal bearings did not become popular until the late 1960s. Tilt pad bearings offered a distinct advantage over a fixed profile bearing, in that the bearing had movable pads that significantly reduced the oil film cross-coupling stiffness, thereby increasing the rotor stability.

The first usable program to analyze tilt pad bearings was
based on the ground-breaking paper by Lund (1964). To this day, tilt pad bearings are the workhorse hydrodynamic bearing for centrifugal compressors used in the oil and gas industry. Experience has shown successful operating of tilt pad journal bearings with light load at speeds as high as 570 ft./sec (174 m/sec) and with unit loads as high as 775 psi (5434 kPa) at moderate surface speeds.

Main compressor seals

Main compressor seal technology has also advanced over the years. In the 1910s, centrifugal compressors were generally used to supply air to fire blast furnaces in the steel industry. The main seals in these air compressors were aluminum labyrinth type seals that typically allowed gas to leak to the atmosphere. Later, as centrifugal compressors were used to compress methane (CH₄) and other combustible gases, leakage to the atmosphere was intolerable (Kirk, 1986).

Oil seal bushings were utilized for main compressor seals for higher-pressure natural gas applications. Records show oil film seals being used in the 1950s (Figure 8). These seals used oil supplied at a pressure higher than the compressor suction pressure to ensure the volatile gas did not leak to the atmosphere. Not until the late 1970s and early 1980s is there evidence in the open literature of OEMs’ working to understand seal forces and how they impact machinery vibration characteristics (stability) (Kirk, 1977).

Today, oil film seals are typically only used for revamp and repair activities (Figure 9). Nearly 100% of new compressors sold in the oil and gas industry feature dry gas seals (DGS). The first known application of a DGS in the author's company was in a 30 psig (210 kPa) single-stage overhung compressor in 1962. Since that time, the DGS industry has worked hard to gain market acceptance (Stahley, 2005). Current practical seal running experience is near 3625 psi (250 bar) delta pressure with positive laboratory experience to 5800 psi (400 bar).

Internal seals

While tooth labyrinth seals have been used for decades to seal impeller stage rise from “bleeding” back to low-pressure areas in the machine, a recent enhancement to the eye and balance piston/division wall seal is the swirl brake (Figure 10). When properly designed, these vane-like devices can substantially reduce cross-coupling stiffness generated inside a tooth laby by controlling the gas tangential velocity (Moore, 2000). These swirl breaks can be used on tooth laminths as well as damper seals.

Damper seals (Figure 11) introduce more desirable...
direct damping than undesirable cross-coupling stiffness, thereby improving the stability of the rotordynamic system. The industry certainly has come a long way in understanding the forces exerted on a rotor since the famous Ekofisk field rotor instability in the North Sea in 1974.

Impeller analysis
Advances in structural dynamics have also occurred. One such area that has greatly improved machine reliability is impeller dynamics. In the early years a designer might perform simple hand calculations to ensure that impeller did not yield or slip on the shaft. This was sufficient when impeller tips speeds were 50% of today’s typical speeds. Nevertheless, some users endured the occasional impeller incident. In many instances, the failure might have resulted from high-cycle fatigue cracks caused by an impeller running in a resonance condition. Today, impellers are analyzed using modal and forced response analysis in much the same way as rotors have been for four decades.

Understanding impellers’ natural frequencies and the aerodynamic forces exerted on impellers has greatly reduced the number of impeller-related incidents in recent years (Schiffer, 2006).

Closing remarks
Tremendous advances have been made in the design and manufacturing of industrial centrifugal compressors. Many can be traced to the evolution of the analytical tools used in the design process. However, these analytical methods would be wasted were it not possible to build the complex designs. One must recognize the critical role that improved manufacturing methods played in enhancing compressor performance.

In closing, the industry has not reached the end of the evolutionary process. Further improvements are possible and given the continuing demand for more energy efficiency equipment, OEMs will strive for still higher performance levels.

Nomenclature
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\phi = \text{flow coefficient} = \frac{Q}{ND_2^3}
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- **Q** = flow in cubic feet per minute
- **N** = speed in rotations per minute
- **D_2** = impeller exit diameter in inches