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OPERATION OF CENTRIFUGAL COMPRESSORS IN CHOKE CONDITIONS

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ABSTRACT

Centrifugal compressors are at times required to operate in or near the choke region. Various limits of the degree of allowable operation in choke have been established. Based on test data and numerical data, the behavior of centrifugal compressors in the choke region is studied. Changes in aerodynamic performance, thrust load, volute behavior and radial loading are considered. The issue of excitation of impeller vanes is addressed. Particular consideration is given to multistage machines, as well as dual compartment machines, in particular regarding the effects of impeller mismatch during operating conditions at flows significantly higher than the design flow.

Limitations in the overload operating range of a compressor not only impact the operational flexibility , but also can require more complicated control systems. The paper addresses aerodynamic, structural as well as rotordynamic issues related to the operation in choke.

INTRODUCTION

Centrifugal compressors are frequently used in the oil and gas industry in higher pressure applications (Rasmussen and Kurz, 2009). One of the characteristics of applications in the oil and gas industry is the requirement to use the compressors over a wide range of operating conditions (Figure 1). The machines, usually speed controlled, are expected to operate from low flows near surge to very high flows near or in choke. While the operation near surge, as well as issues to prevent compressors from surging have drawn significant amounts of attention (Moore et al, 2009). The potentially damaging effects of surging compressors are widely acknowledged.

Operating compressors in the high flow region , i.e. at higher flows than at the best efficiency point, often referred to as overload , choke or stonewall, has been identified by many manufacturers as a region of operation that needs to avoided, also (Brun and Kurz, 2007). Many compressor manufacturers will limit the operation of the compressors at high flows by defining a 'choke limit', 'overload limit', or 'sonic limit' on a compressor map, prohibiting the operation of the compressor at low pressure ratios. This can require installation of additional throttle valves, or more complicated recycle and anti-surge valve selections, and thus impacts the cost of operation, while at the same time limiting the operating range.

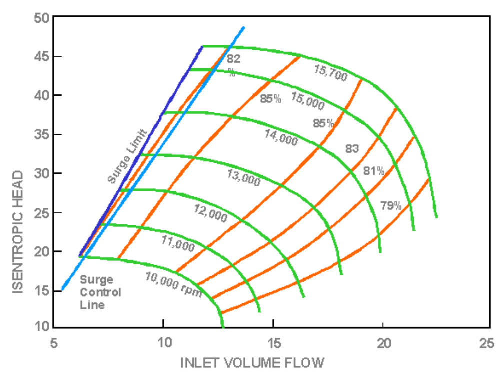

Figure 1: Typical Map of a Speed Controlled Centrifugal Compressor

In the literature, we find only a few papers (Sorokes, Miller and Koch, 2006, and Borer, Sorokes, McMahon, and Abraham, 1997, Brun and Kurz, 2007) discussing the issue of compressor operation in choke.

Borer et al (1997) discuss the impact of aerodynamic forces on impeller vanes. They point out that downstream flow non-uniformities, as they can be created by vaned diffusers or discharge volutes, can create fluctuating stresses in blade leading edges. Especially in high pressure compressors, the forces acting on blade leading edges can be quite large. Sorokes et al (2006) also focus on the risk of damaging the impeller leading edge or inlet region if flow fluctuations in combination with off-optimum incidence create dynamic forces that exceed the allowable stresses. In particular they point at the interaction between non uniformties created by compressor components such as diffuser vanes, volutes, guide vanes or return vanes, and the flow field at the impeller inlet.

The present paper will focus in particular on the impact of axial thrust loading of the compressor.

OPERATION IN CHOKE

Unfortunately, the setting of choke limits is somewhat arbitrary in many instances, and may be subject to questions, in particular when it limits operational flexibility. To be very clear: Prolonged operation in the choke region should be avoided, because if nothing else, the efficiency of the compressor is very low. Whether the compressor operates in choke or not is not as clearly defined, and not as obvious from an operational standpoint as the operation of a compressor in surge. A compressor map showing a single speed line will usually show a more or less steep drop in head and efficiency when the compressor is operated at flows higher than BEP. This behavior is more pronounced at Mach numbers approaching 1, and high mole weight gases (Figure 2).

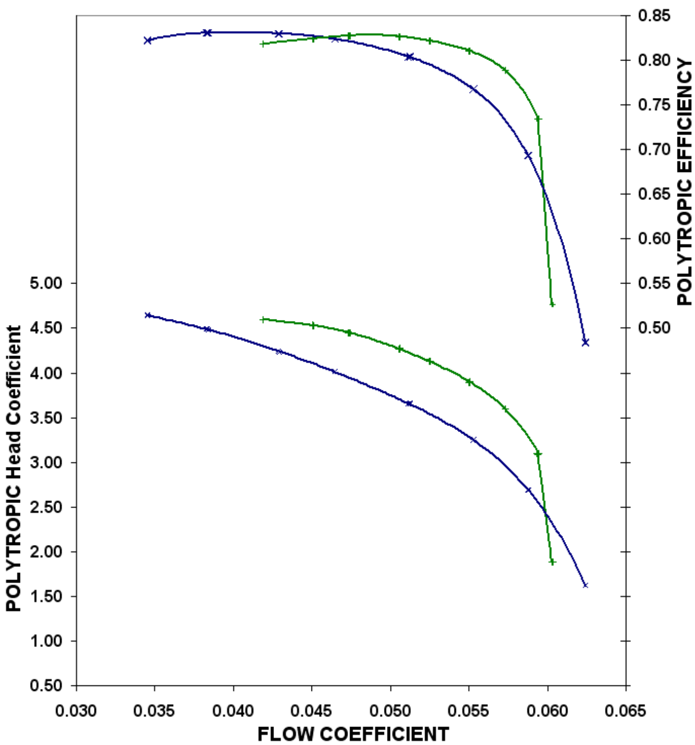

Figure 2: Stage map for M_n=0.56 and M_n=0.76

From an aerodynamic standpoint, choke refers to a situation where flow passages become blocked either due to the occurrence of compression shocks or due to massive flow separation. In centrifugal compressors, these flow passages can either be the impeller inlet, or the inlet to diffuser vanes.
In order to assess the flow through centrifugal compressor components, we have to introduce the concept of velocity triangles (Kurz, 2004, Brun and Kurz, 2000). The general behavior of any gas compressor can be gauged by some fundamental relationships: The vanes of the rotating impeller 'see' the gas in a coordinate system that rotates with the impeller. The transformation of velocity coordinates from an absolute frame of reference (c) to the a frame of reference rotating with a velocity u is by (Figure 3):

$$\vec{w} = \vec{c} - \vec{u} \quad (1)$$

where, for any diameter D of the impeller

$$u = \pi DN \quad (2)$$

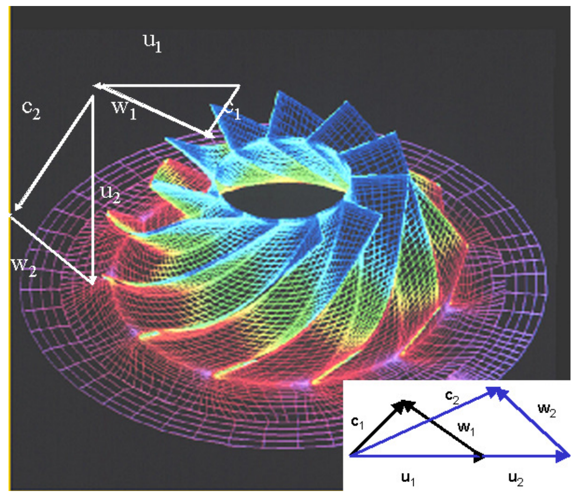

Figure 3: Velocity vectors in a Centrifugal Impeller

At the impeller inlet, the flow enters the impeller with a velocity c_1 in the stationary reference frame. An observer rotating with the impeller will see the same flow at a velocity w_1. At the inlet of the impeller, rotating at a speed N, the blade velocity u is higher at the tip or shroud of the impeller, than at its hub side.

The impeller exit geometry ('backsweep') determines the direction of the relative velocity w_2 at the impeller exit. The basic 'ideal' slope of head vs. flow is dictated by the kinematic flow relationship of the compressor, in particular the amount of backsweep of the impeller. Any increase in flow at constant speed causes a reduction of the circumferential component of the absolute exit velocity (c_{u2}) (Figure 3). It follows from Eulers equation, that this causes a reduction in head. Adding the influence of various losses to this basic relationship creates the shape of the head-flow-efficiency characteristic of a compressor: Whenever the flow deviates from the flow the stage was designed for, the components of the stage operate less efficient (Figures 2 and 4). This is the reason for incidence losses. Using an airfoil as an example, we see that at the 'design flow' the air follows the contours of the airfoil. If we change the direction of the incoming air, we see increasing zones where the airflow ceases to follow the contours of the airfoil, and create increasing losses. Furthermore, the higher the flow, the higher the velocities, and thus the friction losses .

A compressor, operated at constant speed, may be operated at its best efficiency point (Figures 2, 4 and 5). If we reduce the flow through the compressor (for example, because the discharge pressure that the compressor has to overcome is increased), then the compressor efficiency will be gradually reduced. At a certain flow, stall, probably in the form of rotating stall, in one or more of the compressor components will occur. At further flow reduction, the compressor will eventually reach its stability limit, and go into surge.

If, again starting from the best efficiency point, the flow is increased, then we also see a reduction in efficiency, accompanied by a reduction in head. Eventually the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke. (For practical applications, the compressor is usually considered to be in choke when the head falls below a certain percentage of the head at the best efficiency point).

Both operation near the surge limit and in choke leads to flow conditions that are severely different from the flow conditions at the compressors design point. If one uses the airfoil of an aircraft as an analogy, surge and choke would mark the stall points of the airfoil at very high positive flow incidence (i.e. high angles of attack) and at a very high negative flow incidence.

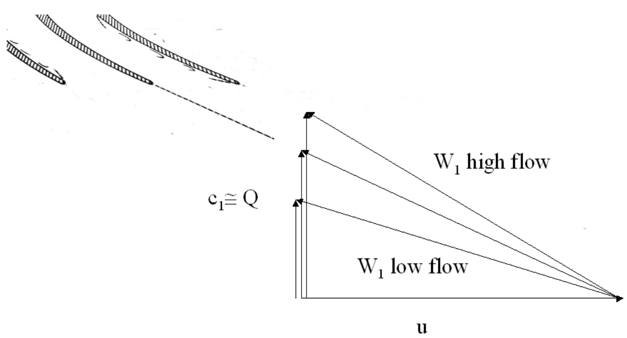

Figure 4: Inlet velocity triangles at different incidence

If the compressor impeller is operated at its design point, the flow will enter the impeller in the optimum direction. (Figure 4). If the flow is reduced, the direction of the flow into the impeller changes, increasing the losses, and ultimately leading to flow separations on the suction side of the impeller vanes. This is often referred to as stall. If the flow is increased starting at the design point, the losses are also increased, but so are the velocity levels entering the impeller. Eventually, the combination of flow separation (this time on the pressure side of the impeller), and the occurrence of compression shock waves will essentially limit the flow that can pass through the impeller (Figures 2 and 6). The shock waves themselves tend to be dynamic and fluctuating in nature.
The reality is a little more complicated, since the flow field is three dimensional. Therefore, compression shocks may only block part of the flow path (typically closer to the shroud side), and separated flow may form rather complex flow structures.

Vaneless diffusers offer no particular challenge for operations in choke (Figure 5), despite the fact that the exit velocity c_2 from the impeller in the stationary frame is rather high (Figure 3). In particular, there are no issues with compression shocks, and there is no particular danger of flow separation that has to be considered. Vaned diffusers, on the other hand, may show the same behavior described for the impeller inlet, and choke at high negative incidence angles and high velocities.

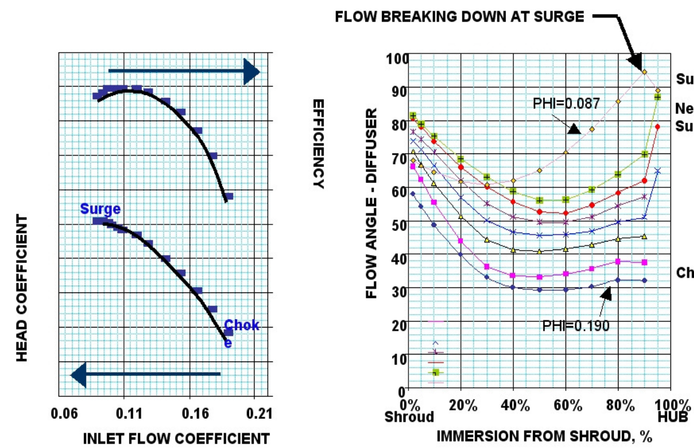

Figure 5: Flow in a vaneless diffuser from surge to choke

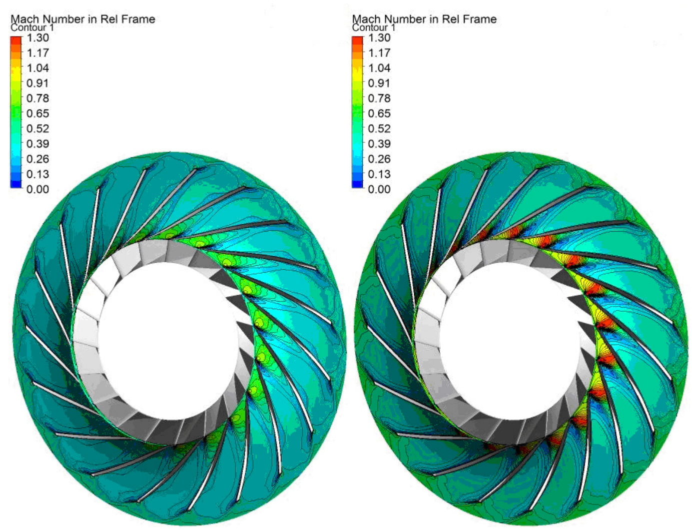

Figure 6: Mach number contours (relative frame) for impeller operating at M_n=0.56 (left) and M_n=0.76 and slightly lower flow coefficient (right). At the lower Mach number, stall due to negative incidence has developed. At the higher Mach number, a shock structure has formed at the pressure surface. Refer to Fig. 2.

In the compressor, choke is related to a flow regime at very high flows which means that the flow channels between blade rows may experience blockage effects, either from sonic flow shocks, wake areas, strong secondary flows, or simply by the fact that the disturbed flow uses the through flow area less efficiently. In reality, one often sees a combination of all of these.

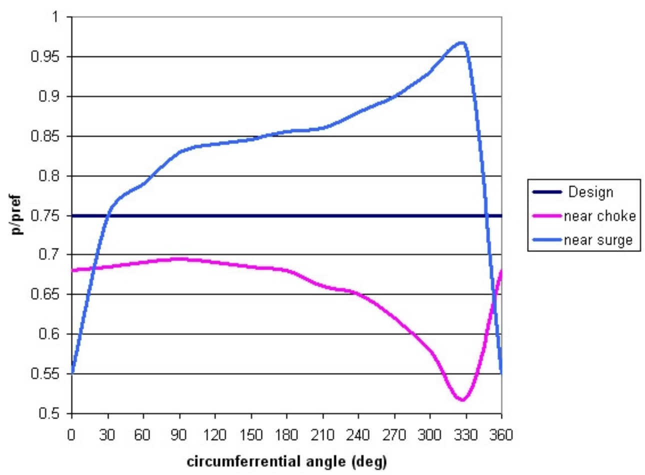

Figure 7: Circumferential Pressure Profile of a Volute (Fiedler, 1989)

Volutes, on the other hand show a distinct change in the circumferential pressure distribution when operated away from the design point (Figure 7), which has to be considered when the radial forces on the rotor are to be assessed.

While operation at high flow is unattractive to the user due to the associated drop in efficiency, there are operational situations where it can be encountered:

-During start-up when the recycle valve is opened too much.
-Process upsets, for example if two compressors operate in parallel, and one of them has to be shut down (Ohanian et al, 2002)
-Compressor undersized for the desired operating conditions.
-Performance degradation due to fouling

Sometimes reducing clarity is the fact, that the Machine Mach Number M_n only indirectly reflects the aerodynamic situation at the sonic limit. The Machine Mach Number is tip speed of the impeller in relation to the speed of sound in the gas upstream of the impeller inlet. Therefore, the Machine Mach number describes not any actually occurring mach number in the machine. The part of the impeller that 'sees' the highest velocity is the impeller inlet on the shroud side in the relative frame (that is, the reference frame rotating with the impeller), which is w_1 in Figure 3 and 4.
While the machine Mach number, for constant inlet conditions, only changes if the impeller speed changes, the relative velocity w_1 also changes with different operating points at constant speed: In Figure 4, for a constant impeller speed, w_1 clearly increases for higher flows.
For this reason, there is no single machine mach number that identifies the onset of a inlet shock. The mach number effects are always a function of local Mach numbers. In particular high flow impellers, where the diameter of the impeller tip at the inlet is not much smaller than the impeller exit diameter, will show the impact of compressibility at relatively low Machine mach numbers.

OFF –DESIGN AERODYNAMICS

In multistage compressors impellers are in general selected such, that at the design point, all impellers operate at or near their best efficiency point (BEP) (or at about the same individual impeller surge margin. Surge margin is defined for a compressor operating at some flow Q, relative to the flow at surge Qs, for a constant speed:

$$SM = \frac{Q - Q_s}{Q} \Big|_{N=const} \quad (3)$$

This approach tends to yield a good efficiency at the design point as well as a wide operating range. When the compressor is operated away from the design point, for example at higher flows than BEP, the first impeller will create less pressure ratio, and therefore achieve less volume reduction than before (this can be seen, for example in Figure 2). Therefore, the second and subsequent impellers will increase their respective surge margin at a faster rate than the first impeller. In other words, the rear impellers will in general get to choke earlier than the impellers further in front (Figure 8). This means in particular, that in situations where the compressor appears to be is near choke, one or some of its stages may already be in deep choke.

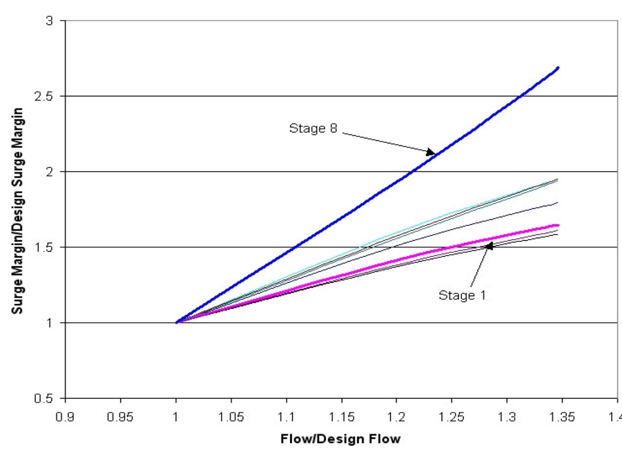

Figure 8: Shift in relative impeller operating points at increased flow for an 8 stage centrifugal compressor, expressed as distance from surge (Surge Margin).

THRUST LOADS

Thrust loads of centrifugal impellers are a result of a pressure imbalance between the front face and the rear face of the impeller. The sum of these forces over all impellers and the forces created by the balance piston are the resulting load on the compressor thrust bearing.

From the axial momentum equation, which takes into account the change of the axial momentum of the gas, and the forces due to the static gas pressure in axial direction:

$$\oiint \rho \vec{C}\,(\vec{C} \cdot d\vec{A}) = \oint p \cdot d\vec{A} + \vec{F} \quad (4)$$

we get the resulting forces on the impeller as (Figure 9):

$$\vec{F}_{impeller} = \vec{F}_{momentum}(c_{exit}, c_{inlet}) - \vec{F}_{pressure}(p_{cavity,front}, p_{cavity,rear}, p_{inlet}, p_{exit}) \quad (5)$$

The front and rear cavities are formed between the impeller tip and the labyrinth seals at the impeller inlet, and the impeller hub seals, respectively.

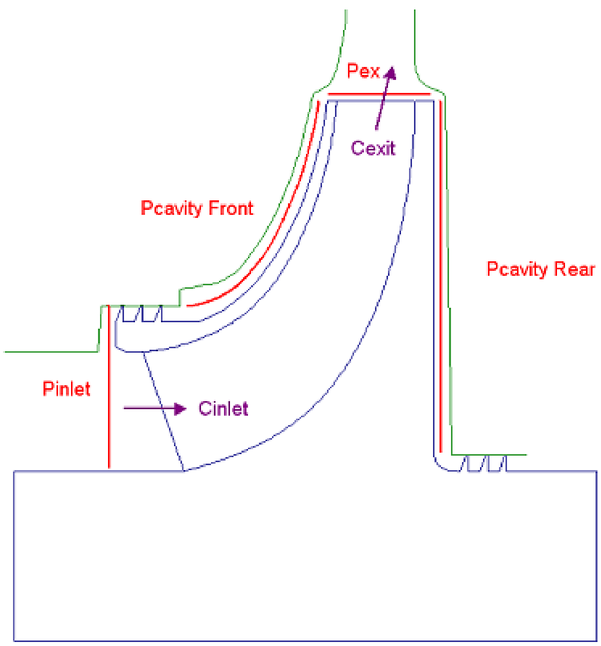

Figure 9: Forces on the impeller

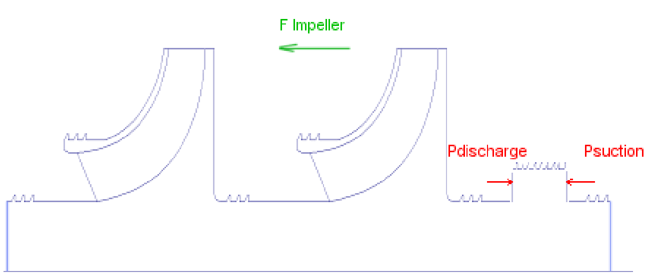

Figure 10: Balance Piston

The force on the thrust bearing is thus (Figure 10):

$$\vec{F}_{thrustbearing} = \vec{F}_{impeller} - \vec{F}_{balancepiston}(p_{discharge}, p_{suction}) \quad (6)$$

In the simplest approach to calculate the forces on the impeller, one would assume the pressure in the front and rear cavities to be equal to the pressure at the impeller tip. In a shrouded impeller however, the gas in the cavity is subject to swirl, and as a result, the static pressure at lower radii is lower than at the tip. The amount of swirl is a function of the cavity geometry, and the leakage flows through the labyrinths.

The cavity static pressure distribution can be calculated by:

$$p(r) = p_{tip} - \frac{1}{2}\rho(q\omega)^2(r_{tip}^2 - r^2) \quad (7)$$

which accounts for the cavity characteristics by introducing a cavity swirl coefficient q.

A simple approach would assume constant swirl coefficients for front and rear cavities. This approach is frequently used in the industry, but high pressure compressors require more accurate estimates. Correlations and CFD analysis (Figure 11 a) are being used for these, along with subscale test measurements for validation.

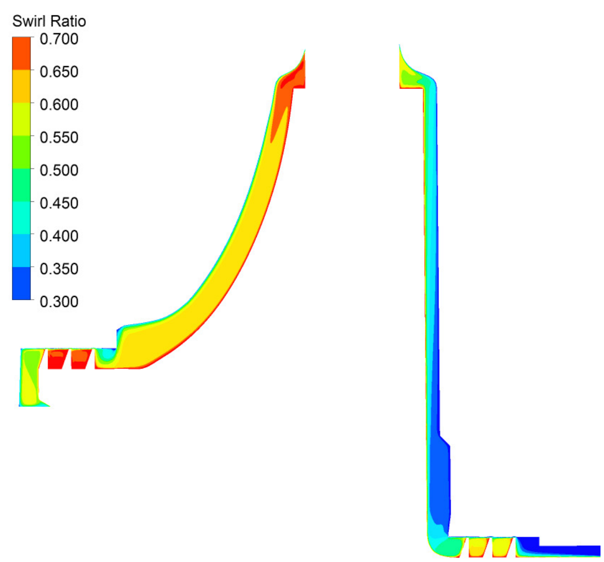

Figure11 a: Swirl ratio in the shroud and the backside cavity

Of particular importance for the topic of off design operation is the fact that the swirl coefficient changes when the impeller is operated away from its design point (Figure 11 b). Also, the magnitude of the swirl coefficient on the impeller backside changes in the opposite direction from the swirl coefficient on the front side of the impeller. This means that the thrust imbalance (for a given pressure level and a given speed) changes not just due to the pressure difference between the impeller eye and the corresponding backside, but also due to

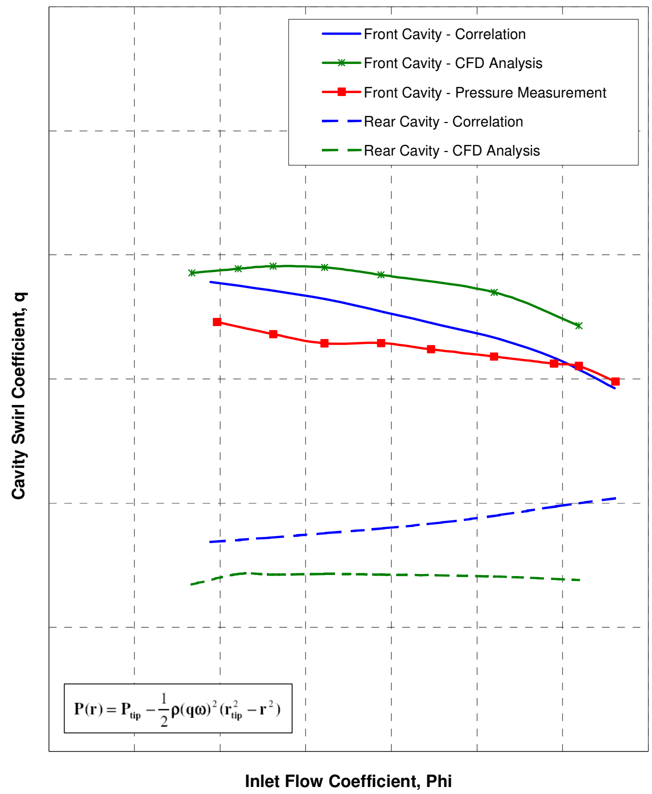

Figure11b: Cavity Swirl Coefficient for a Medium Flow Stage for different operating points

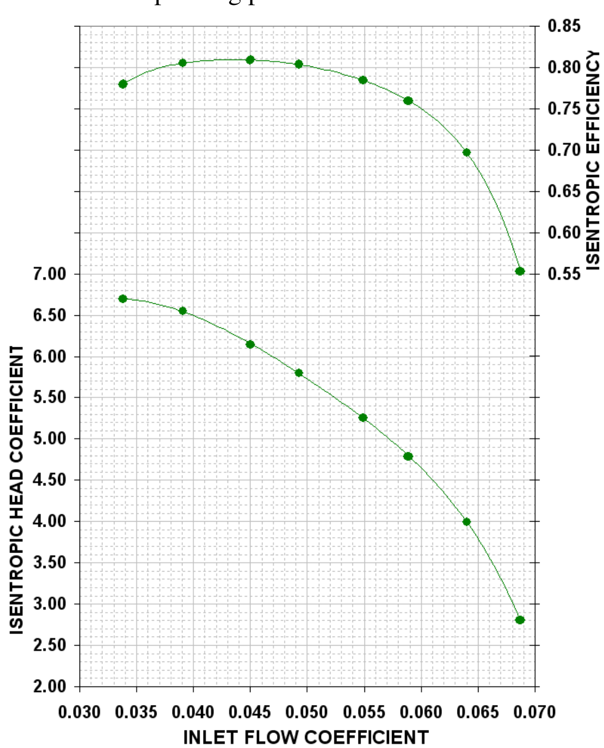

Figure 12 a: Non dimensional map for multistage compressor

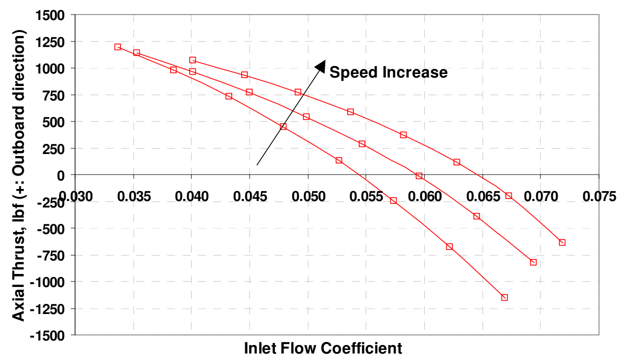

Figure 12 b: Change of axial thrust with operating point

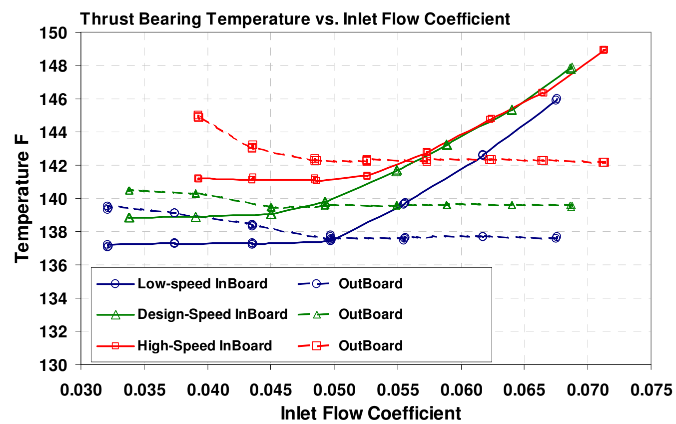

Figure 12 c: Thrust bearing temperature as a function of operating point

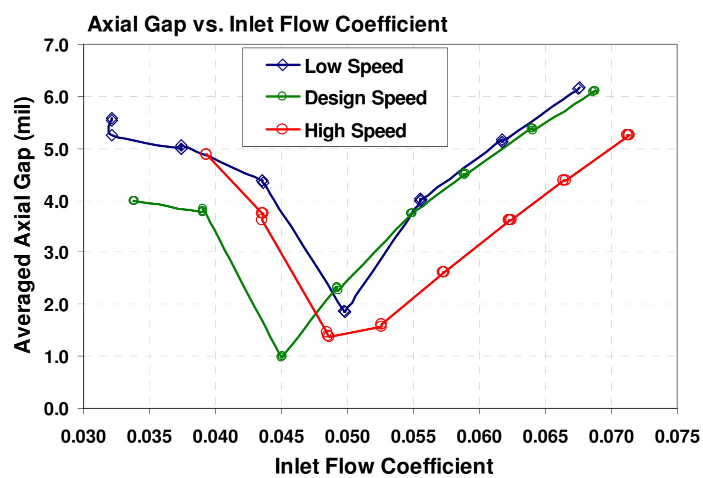

Figure 12 d: Axial position of the rotor as a function of the operating point

different swirl factors in the cavities in the front and back of the impeller. This imbalance, in particular, changes when the compressor moves from the design point to choke. In general, the shroud side swirl is higher than the backside swirl, a result also reported by Koenig et al, 2009.

Because the thrust load has a direct impact of the thrust bearing temperature, which can be measured conveniently, Figures 12 a –d establish the correlation between non dimensional operating point (Figure 12a), thrust load at different speeds (Figure 12 b), the resulting bearing temperature of the loaded and unloaded pads of the thrust bearing (Figure 12 c), as well as the axial position of the rotor as a result (Figure 12 d). The inboard bearing shows a significant increase in temperature (albeit not to a level that would cause concern) when the compressor enters the choke region. The outboard bearing only shows a much lower increase in temperature when the operating point moves towards surge. For this particular application, with the particular selection of the balance piston size, the thrust load reverses direction, which explains the behavior of the in board and out board bearing temperature. Of course, the bearing temperature increases also with speed. As a result of the thrust load changes and the changing load capacity of the thrust bearing with speed, the axial gaps for all speeds are fairly close together, but change significantly when the compressor is operated from design point to surge or into choke.

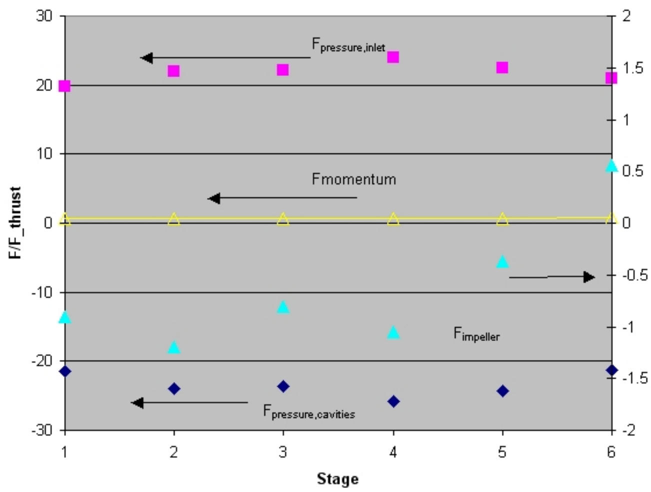

Figure 13: Contributing Forces to the Impeller Thrust in a 6 Stage Compressor.

If we compare the magnitude of the forces acting on the impeller (Figure 13), the pressure from the inlet eye and the pressures in the cavities are usually dominant, but act in opposite direction. In general, they generate a resulting force, much smaller than the pressure forces, in direction of the compressor inlet, but as can be seen in Figure 13, this is not always the case. The momentum force, generated by deflecting the gas form more or less axial to more or less radial direction, is usually much smaller than the pressure forces. At very high discharge pressures near choke, when the pressure differential over the impeller is rather small, the momentum force can become dominant, and create a net force towards the discharge end of the compressor.

The descriptions in this section are based on shrouded impellers. As opposed to open faced impellers, that have free standing blades, the blades in shrouded impellers are covered. Therefore, the pressure distribution on the front face of the impeller is governed by the impeller discharge pressure and the impact of swirl flow. In an open faced impeller, the pressure distribution would be governed by the pressure build up in the impeller flow passages.

MULTISTAGE MACHINES AND MACHINES WITH MULTIPLE SECTIONS

Another issue that has to be considered with multistage compressors operating near choke: The overall compressor may still produce head, when some individual stages are already reducing head, thus, acting as throttles. These impellers will consequently see a lower pressure on their discharge side than on their suction side, which can in some instances alter the axial thrust balance in the compressor, leading to increased load on the thrust bearings.

The mechanical design of compressors with multiple sections regarding the arrangement of impellers can either be a straight trough design, or a back to back design. In a back to back design, with the impellers in the first section facing in opposite direction of impellers in the second section, most of the axial thrust balance is accomplished by the impellers themselves. Usually, a relatively small balance piston can handle the axial thrust. For all applications, the axial thrust has to be determined for all operating conditions. Two section machines are particularly critical in this respect if they have to handle multiple streams that change flow independently of each other. Thrust limitations can create limits for the unbalance between the sections.

In highly transient conditions, such as emergency shutdowns (Moore et al,2009), this has to be considered. In some instances, hot gas recycle valves have to be employed to equalize the pressure between the sections before the thrust imbalance can cause damage.

In the previous considerations, the impact of seal leakage flows on impeller and balance piston thrust forces has been mentioned. Impeller seals, as well as balance piston seal clearances can increase over time, although designs where the labyrinth teeth are facing abradable material are less prone to this problem. Nevertheless the fact remains that increased seal clearances can change the trust balance. However, monitoring bearing temperatures seems a good way of protecting the compressor from thrust load issues, especially when operating at extreme points of the map.

ACTIVE MAGNETIC BEARINGS

For the discussions above it was tacitly assumed that the compressor uses hydrodynamic radial and thrust bearings. The changes in uncompensated axial thrust when the compressor is operated form surge to deep choke can usually be accommodated by the load capacity of this type of bearings. Active magnetic bearings only have about one tenth of the load capacity of hydrodynamic bearings. This may cause restrictions for operating the compressors in the choke region, as the resulting unbalanced thrust load can no longer be accepted. Also, the radial forces generated by a discharge volute in off-design conditions (Figure 7) have to be carefully reviewed.

IMPELLER VANES

Borer et al (1997) and Sorokes et al(2006) have specifically discussed in more detail the issue of damage to the impeller vanes due to high cycle fatigue. With respect to the operation in choke, the frequency of the exciting forces on the impeller vanes can be captured in a Campbell diagram (an example is given in Figure 14). Comparing the exciting frequencies (in this case the vane passing frequency from the inlet vane) with natural frequencies of the impeller and its vanes reveals the separation margin between excitation and response. If the separation margin is deemed too small, impeller modifications such as increases in vane thickness can reduce stress levels and modify impeller natural frequencies. Figure 15 shows the result of such a redesign, where it was possible to achieve the stress reduction and a larger separation margin, while retaining the impeller performance.

It should be noted that in some axial compressors, but also in centrifugal compressors with free standing blades, a phenomenon called choke flutter can be observed when they are operated in the choke region (Fottner, 1989). The mechanism is different from the mechanism mentioned above, as it is the flow around the blade that causes the excitation. However, the frequencies that are excited are, just as in the case of disturbances in the inlet or exit flow, the blade natural frequencies. The aerodynamic excitation of the blade is usually caused by boundary layer separations, transonic shock patterns, and the related vortex frequencies. This problem has, to our knowledge, not been encountered in centrifugal compressors with shrouded impellers.

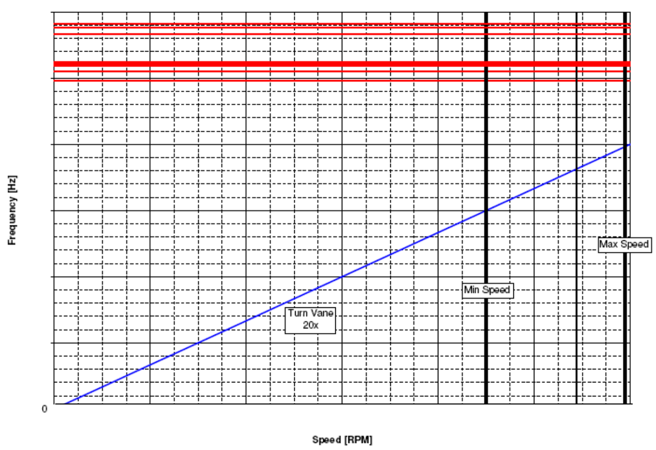

Figure 14: Campbell Diagram for an Impeller.

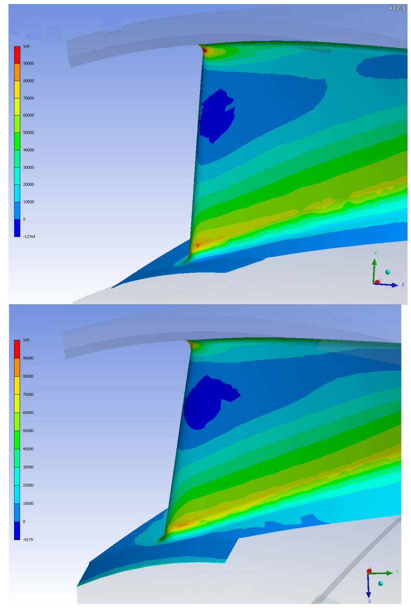

Figure 15: Principal stress distribution near the leading edge of an impeller vane. The lower picture shows the improved version of the same impeller with reduced stress in the hub and shroud region.

CONCLUSIONS

Besides the performance penalties, the study indicates that operating in choke is often not a problem for the compressor provided:

-The balance piston is sized correctly to provide adequate thrust load balance over the entire operating range

-Issues like blade strength to deal with alternating stresses are considered, or the occurrence or strength of alternating stresses is reduced.

NOMENCLATURE

$$D = Diameter$$

$$\vec{C} = Velocity \quad stationary \quad frame$$

$$\vec{A} = Area$$

$$\vec{F} = Force$$

$$Q = Flow$$

$$M_n = Machine \quad Mach \quad Number$$

$$N = Speed$$

$$p = \Pr essure$$

$$SM = Surge \quad M \arg in$$

$q =$ cavity swirl coefficient

$r =$ radius

$tip =$ impeller tip

$\vec{u} =$ blade velocity

$\vec{w} =$ velocity rotating frame

$\rho =$ density

$\omega =$ impeller rotational speed

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