THE CONSEQUENCES OF COMPRESSOR OPERATION IN OVERLOAD

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ABSTRACT

This work describes the potential consequences associated with operating a centrifugal compressor in overload. Nomenclature is offered to explain what is meant by overload operation, and methods that are used by original equipment manufacturers (OEMs) and end users to define overload limits are presented. The paper also describes the conditions that can lead to overload operation. Computational fluid dynamics (CFD) results are used to illustrate the forces acting on an impeller when it operates at very high flow. Finally, this paper suggests considerations that should be addressed when designing (or selecting) an impeller that could be subjected to extended overload operation.

INTRODUCTION

It is common knowledge that operating centrifugal compressors in surge can have detrimental effects on the mechanical integrity of parts. Prolonged excursions into surge have caused damage to impellers, bearings, seals, and other rotating components. The violent nature of surge events makes it necessary to install sophisticated control systems to prevent compressors from operating in surge.

While there are numerous publications in the open literature addressing the consequences of operating in surge, there are few, if any, articles on the potential dangers associated with operating a compressor in overload—the high flow region of a performance map. In the vast majority of applications, overload operation does not represent any cause for concern. In fact, many compressors spend the majority of their life operating in the high flow end of their performance envelope. Yet, in some applications, overload operation can be just as damaging as operation in surge.

This paper will discuss the potential consequences of operation in overload. The primary focus will be the potential damage that could occur in impellers, as they are typically the most expensive and difficult component to replace in a centrifugal compressor. The discussion will begin with definitions of the terms "overload" or "deep overload." These terms mean different things to different people. Therefore, it is important to understand the meaning of these terms in the context of this paper. The focus will next move to the types of situations that can lead to operation in overload. This is followed by an overview of some "real-world" examples of damage caused by overload operation. Computational fluid dynamics (CFD) and finite element analysis (FEA) results will then be used to provide a clearer picture of the forces imposed on impellers when they operate well above their intended design flow rates. Suggestions are offered on factors that should be considered when designing impellers that may be subjected to prolonged overload operation. Application of the general guidelines outlined in this discussion will in no way ensure the long-term mechanical integrity of the impellers. Given the unique factors associated with each situation, it is imperative that you seek the guidance and direction from engineering. However, proper application of these guidelines may increase the impellers' resistance to the forces imposed by overload operation that could lead to unwanted consequences. Finally, conclusions will be offered on the need to be cognizant of the consequences of overload operation when establishing the operating envelope for a new compressor application.

DEFINITIONS

Unlike surge, there is no commonly accepted definition for overload or "operation in overload." The compressor aeroperformance maps given in Figures 1 and 2 will be used to illustrate the various definitions. The maps show the head coefficient and polytropic efficiency plotted against the normalized inlet flow coefficient.

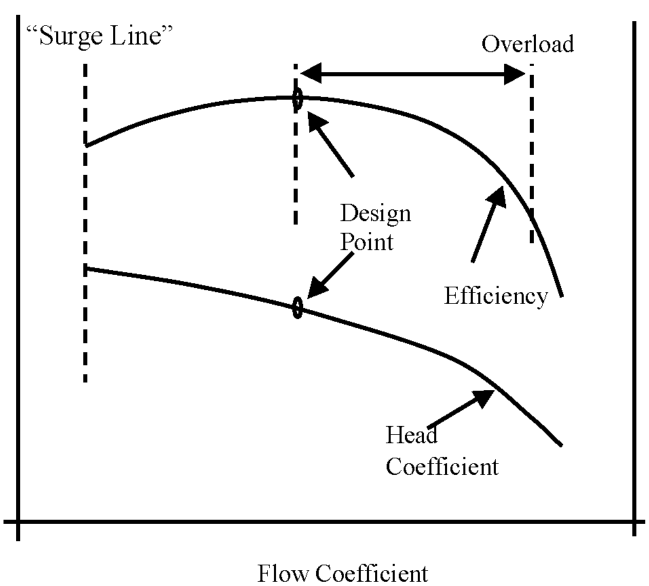

Figure 1. Definition of Overload.

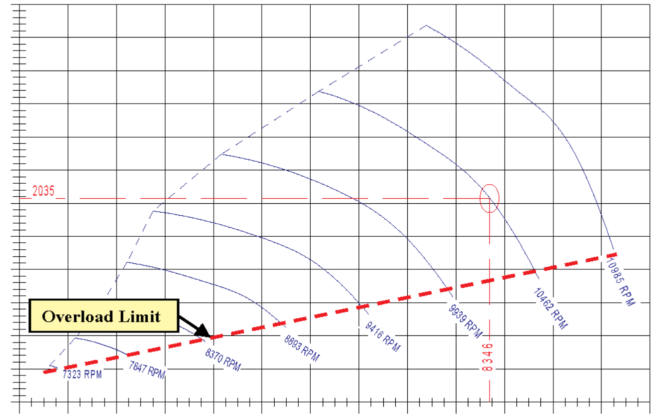

Figure 2. Setting Overload Limit.

In the broadest sense, the term "overload" is used to reference any condition under which the compressor's inlet flow exceeds its design flow rate (Figure 2). However, most compressors regularly operate at flow rates higher than design and most users do not consider this operating in overload. To them, this is normal operation. Instead, this group deems overload to be operating at any condition that exceeds the safe or recommended flow limit for the machine. This hazardous region to the extreme right on the performance map is often called "deep overload."

End users typically defer to the original equipment manufacturer (OEM) to establish an overload operating limit if the OEM believes such a limit is warranted. This is where the difficulties really begin because, unlike surge, compressor OEMs, in general, do not specify an overload limit. Because compressor performance drops off rapidly in the overload region, it is frequently assumed that an end user will not want to operate the compressor in that regime or will lack the driver capacity to do so. As will be seen, this can be a risky assumption to make.

The majority of OEMs address the issue of overload operation by putting a clause in the operator's manual stating that off-the-map operation must be avoided. In the strictest interpretation, this clause covers both the low-flow and high-flow ends of a performance map, i.e., do not operate at flow rates lower or higher than shown on the map. Still, surge control systems are installed to ensure no (or limited) operation in surge. It is somewhat uncommon to find control systems that limit overload operation. Again, this is quite often the case because the compressor driver cannot provide sufficient power to maintain prolonged operation in overload.

On occasion, OEMs will provide an overload limit on the right-hand side of the performance map (refer to Figure 2). This limit is set based on a variety of methods that are described in the following paragraphs. In fact, in rare instances, anti-overload control systems have been provided that operate based on algorithms very similar to an anti-surge control system, except that a discharge throttle valve is utilized instead of a recycle valve.

Percent of Design Flow

One of the simplest methods for establishing the overload limit is to specify the maximum amount of flow as a percentage of the design flow. Such limits are typically set based on experience and are a function of the machine Mach number (U_2/A_0), gas handled, number of stages, etc. It is well known that a compressor that runs at high machine Mach number or is handling high mole weight gases will have less overall flow range than a compressor that runs at low U_2/A_0 or handles low mole weight gases (Figure 3). Therefore, the allowable percentage increase from design flow must vary. For example, one might impose an overload limit of 120 percent of the compressor design flow rate for a high mole weight machine and allow as much as 140 percent of design flow for a low mole weight application.

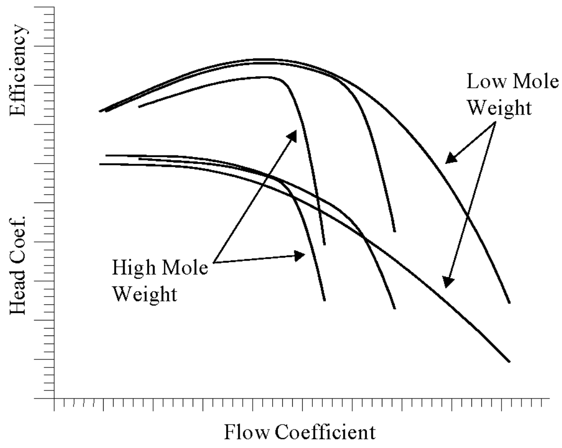

Figure 3. Variation in Flow Range with Mole Weight.

Impeller Mach Number

Some OEMs limit the overload capacity based on calculated values for impeller inlet relative Mach number. Calculations are performed with increasingly larger flow rates and the performance map is truncated (or “cut off”) when the Mach number in any stage within the compressor exceeds the specified level. Typically, the limit is a calculated Mach number of 1.0, the flow rate at which the impeller throat is choked. However, to be conservative, some OEMs set the overload limit based on an inlet relative Mach number of 0.96 or lower.

It is noteworthy that all OEMs do not use the same methods to calculate inlet relative Mach numbers. Further, inlet relative Mach number can be calculated at various locations on the impeller leading edge, i.e., at the shroud, at the mean. Therefore, agreement should be reached between the end user, contractor, and OEM regarding the exact definition of the Mach number being considered.

Another approach often employed by end users is to specify a maximum allowable inlet relative Mach number at the design condition for any impeller in a compressor. This method assumes that limiting the inlet relative Mach number at design will ensure a sufficient amount of overload capacity. However, this method does not ensure that the compressor is not operated in a risky portion of the performance map.

Drop in Head Coefficient/Efficiency

Another popular method for establishing overload limit is based on the rate of change of head coefficient or efficiency as a function of flow coefficient. This method takes advantage of the rapid drop in the compressor’s performance in the overload region as seen in Figures 1, 2, and 3. Often referred to as “stonewall” (because of its vertical nature and the fact that it limits compressor flow like a stone wall), the high flow end of the map has a highly negative slope. Some OEMs will limit the overload capacity to the flow rate corresponding to an “X” percent drop in head coefficient or efficiency for a 1 percent increase in flow. Common values of “X” fall between 10 percent and 20 percent.

An alternate form of this approach limits the overload capacity to the flow rate at which the efficiency or head coefficient is a given percentage of the design flow level, i.e., 15 percent, 20 percent, or 30 percent of the design efficiency or head coefficient. Beyond this point, the performance would be so low as to be unusable in most processes.

Section Summary

Clearly, it is important that the OEM and end user have a common understanding of how the overload limit was established and the potential implications of violating this limit. The latter will be addressed in subsequent sections herein.

CONDITIONS LEADING TO OVERLOAD OPERATION

Common circumstances that lead to prolonged operation in overload are:

- Loss of parallel compressor or compression train,
- Performance degradation (i.e., fouling) within a compressor,
- An undersized compressor (either due to changes in flow requirements for an existing compressor or misapplication of a new machine),
- Alternate operating conditions (i.e., summer and winter conditions in a pipeline application),
- Unanticipated changes in gas characteristics (mole weight, etc.),
- Process upsets.

Parallel Compressors

Many facilities have duplicate compressors or trains of compressors and the process flow is divided equally (or nearly equally) between these compressors or trains. Each machine in the train is sized such that the compressors operate near the middle of their performance map under normal conditions. However, if for some reason (process upset, regular maintenance, etc.) one or more trains are taken offline, plant operators may attempt to make up for the missing train(s) by forcing more flow through the remaining operational trains. This causes those remaining trains to operate at significantly higher flow rates, pushing them into the overload region of their performance maps. Assuming the drivers have sufficient power (and many do not), it would be possible to operate in the overload region for extended periods of time. The performance would suffer (i.e., low efficiency) but production requirements would be maintained.

Excess Driver Power

It is considered normal practice by many users to rate gas turbine drivers for the highest expected ambient air temperature expected, or at least an “average” of the historical high temperatures for that location. This is done so that the compressor and hence the “process plant” may be operated at close to design output on a hot day, or throughout the hot season. With typical aeroderivative gas turbines rated for a 120ºF inlet, the power output will increase from 25 percent to 75 percent when the ambient drops to 30ºF. It is also not uncommon for steam turbines to be rated for 110 percent of the compressor rated power with “minimum” steam conditions, only to be capable of providing 50 percent or more power when operated at “maximum” steam conditions.

Performance Degradation

In a multistage compressor, it is possible to drive the latter stages into overload if there is a reduction in the performance of the earlier stages or if there is a decrease in the mole weight of the gas being compressed. With regard to the former, the impellers or stages in a compressor are aerodynamically matched such that when all are functioning correctly, each operates near the middle of its respective performance map. However, if the performance of the early impellers or stages in the compressor begins to degrade, those impellers or stages will not provide the expected volume reduction. Therefore, any subsequent stages will be forced to “swallow” more flow than expected. If the performance degradation is substantial, the latter stages will operate in overload. Again, assuming there is sufficient driver capability, it would be possible for the stages to operate in overload for prolonged periods. Again, the horsepower consumption would increase significantly but production could be maintained.

The same scenario occurs if there is a change in the gas characteristics such that the volume reduction or overall flow range of the individual stages within the compressor drops. This will cause the impellers or stages to operate at successively higher capacity relative to design until the latter stages are operating at or near choke.

Undersized Compressor/Alternate Operating Conditions/Unexpected Changes

At times, compressors are purposely or inadvertently undersized in the selection process. For example, an end user may anticipate a sizeable reduction in flow rate during the life of the compressor. This user may choose, in the beginning, to operate the compressor at the high-flow end of its map, knowing that in coming years the flow rate will be reduced and the compressor will operate nearer the center of its map.

Occasionally, end users purchase equipment before finalizing their operating requirements. When the compressors are put into operation, they discover that the inlet conditions and/or gas mixtures are not as expected or that their production requirements exceed those originally projected. These changes can cause the compressor to operate at flow rates that are much higher than anticipated. In short, the compressor operates in the overload region of its performance map.

Finally, process conditions may also change during the life of a compressor, causing the flow rate through the machine to increase. The end user may lack sufficient funding to revamp or upgrade the compressor, choosing instead to operate existing equipment in overload despite the low performance.

POTENTIAL CONSEQUENCES OF OVERLOAD OPERATION

Two examples are presented to illustrate the potential consequences of overload operation. In both cases, the compressors were known to have operated for extended periods in the overload region. *Note:* Before proceeding further, it must be noted that end users and OEMs are often very reluctant to provide details on difficulties resulting from off-design operation. End users do not want details of their operating practices made public and OEMs typically do not want any suggestion that their products are anything but perfect. For that reason, limited details will be provided on the two sample cases.

The first example is taken from a high-pressure gas reinjection compressor. The compressor experienced repeated impeller fractures. The welded impeller was a low flow coefficient design, having a very short blade height relative to the impeller diameter (low b/r). Essentially, the impeller disk was fracturing at or around the leading edge region of the impeller. In the worst case, the inner portion of the impeller disk separated from the outer portion.

Following an investigation by the OEM and end user, it was reported that the impeller had been run extensively in the overload region of its performance map when a parallel train was taken offline. Dynamic forces caused by the combination of high impeller leading edge incidence, and a nonuniform pressure distribution caused by the downstream discharge volute, were sufficient to initiate cracks in the impeller, leading to the fractures. The compressor had to be taken out of service for several days to allow installation of the spare rotor.

It should be noted that the forces that led to the fractures were significantly lower at the design flow condition. That is, at design flow, the dynamic forces due to incidence and the volute pressure field were not sufficient to initiate the cracking. Further details on this case can be found in Borer, et al. (1997), and Sorokes, et al. (1998).

The second example is from a compressor processing heavy hydrocarbons. The subject impeller was a high-flow coefficient design, implying that the leading edge was quite tall (high b/r). Blade fractures occurred near the impeller's leading edge. In a few of the blades, a portion of the blade broke away as indicated by the crosshatched area in Figure 4. Again, based on an analysis conducted by the end user and the OEM, it was found that the compressor had operated in overload for long periods. In this case, the dynamic forces because of incidence caused a portion of the impeller blade's leading edge to fracture and separate from the impeller. The loss of material produced an unbalance on the rotor and the compressor had to be taken offline for repairs.

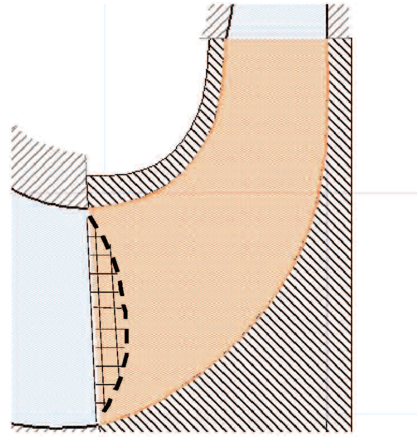

Figure 4. Schematic of Blade Leading Edge Fracture

While the mechanisms that ultimately led to the impeller fractures were different (more on this in the discussion to follow), generally speaking, the root cause for the fractures was the same—operation in overload.

DESCRIPTION OF FORCES

Although the forces resulting from operation in overload have a similar root cause to those found in surge or stall, the true nature of those forces is radically different. The violent forces associated with surge are typically very low frequency (6 Hz or less) and result from the flow reversal through the compressor when the impeller or impellers can no longer overcome the downstream static pressure. When in surge, the inability of the compressor to overcome such pressure is directly related to the increase in incidence or other losses in the compressor components, i.e., impellers, diffusers, return channels, etc. That is, as the flow rate in a compressor is reduced from design toward surge, the angle at which the flow impinges on the bladed or vaned components increases, thus increasing the incidence (or delta angle between the flow angle and the blade/vane angle) (Figure 5). At some point, the incidence angles lead to flow separation or other anomalies that cause very high losses within all of the compressor components, making it impossible for the compressor to overcome the downstream pressure. Because flow moves from the region of higher pressure to a region of lower pressure, the flow reverses direction and goes backward through the compressor. The resulting forces on the compressor internals can be destructive.

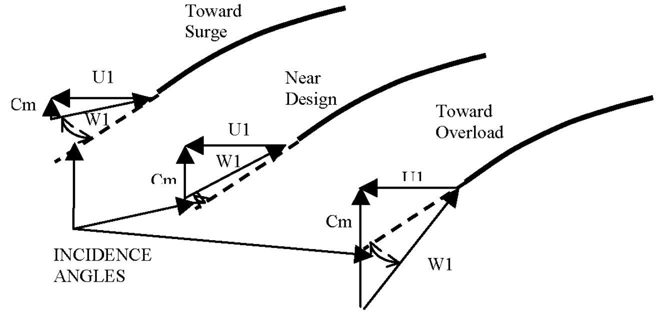

Figure 5. Impeller Incidence.

The basic driver of the forces associated with overload operation is also incidence. However, the frequency of the force mechanism tends to be high frequency. Whereas surge phenomenon has a frequency of 6 Hz or less, the forces associated with overload tend to be more on the order of blade passing frequency (1 to 3 kHz), i.e., the number of impeller blades times compressor running speed. This distinction is important as one considers the nature of the potentially damaging forces.

As with surge, when the compressor is operated at flow rates above design, there is an increase in blade incidence on the impeller. Eventually, the incidence becomes high enough that, as in surge, the flow separates from the impeller blade. In or near surge, this separation occurs on the suction surface; while in overload, it occurs on the pressure surface of the blade. The flow separation causes a reduction in the effective throat (or minimum) area in the impeller, leading to a significant increase in the flow velocity. If the velocity gets high enough, shock waves will form and choke will occur. The pressure fields associated with shock waves are highly dynamic and cause excessive forces on the impeller blades and walls. These forces alone could be sufficient to damage an impeller. However, when these forces are further exacerbated by other nonuniformities within the compressor flow path, one can rapidly reach conditions that can quickly lead to impeller fractures. Two such situations are described in the following discussion.

Blade/Vane Interactions

It is well known that interactions occur between impellers and stationary vanes upstream (inlet guidevanes or IGVs) and/or downstream (diffuser vanes). Numerous technical papers have been written on this subject including Kushner (1980), Fisher and Inoue (1981), and Eckert (1999). Summarizing, the upstream or downstream vanes are surrounded by fields of varying pressure through which the impellers rotate causing pressure fluctuations and, therefore, varying forces on the impeller blades. The frequency of the pressure disturbance is determined by multiplying the rotational speed of the impeller by the number of stationary vanes. Of course, if the upstream component comprises multiple vane rows, the vane wakes from all of the upstream vanes must be considered because it may be possible for the wakes from the first row to propagate through the second row and reach the impeller.

Problems arise when the frequency of the pulsations align with a natural frequency in the impeller, thereby exciting the impeller, resulting in a resonance condition. When the forces are of sufficient magnitude, it would be possible to initiate a fracture in the impeller.

Blade/vane interactions during overload operation are of particular concern because of the high level of energy in the gas stream. In overload, the higher-than-nominal flow rate causes the velocities within the aerodynamic components to be higher than at design. Therefore, the static pressure variation at the exit of the upstream inlet guide vanes will be less uniform than it is at design flow. That is, the high core flow velocity in the IGV passages will cause regions of low static pressure, while the static pressure in the stagnated (or wake) region immediately downstream will be very high (Figure 6). If the inlet guide vanes are sufficiently close to the impeller blade leading edges (as is often the case with full inducer-style impellers), each blade will be subjected to this highly nonuniform static pressure field. Because the magnitude of the pressure variation is higher in overload than at design, the forces during overload operation may be sufficient to create problems, even though the forces at design are not.

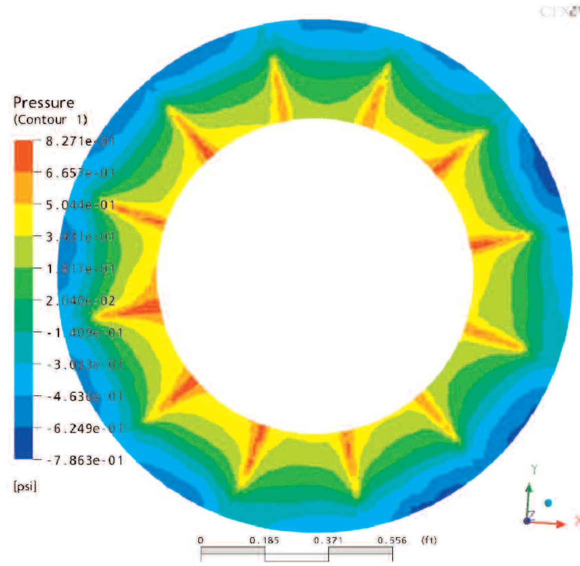

Figure 6. CFD Results Showing IGV Wakes.

The situation with downstream diffuser vanes is somewhat different. Rather than interacting with wakes, the excitation forces associated with vaned diffusers are a consequence of the pressure fields forming around the diffuser vanes because of potential flow effects (Figure 7). Again, as the impeller blades pass the diffuser vanes they pass through the "lobes" in the pressure distribution. The resulting variation in pressure imposes a dynamic force on the impeller that could excite an impeller's natural frequency. As with the IGV wakes, the magnitude of the pressure variation surrounding the diffuser vanes will be greater during overload operation than at design. This is due in large part to the increased incidence on the diffuser vanes and probable separation of the flow from the vane pressure surface. The increased velocities resulting from the higher flow rates also increase the kinetic energy or velocity pressure ($\frac{1}{2}\rho V^2$). This causes even higher loads across the impeller blades. As a result, the dynamic forces in overload will be greater and might be sufficient to initiate a fracture whereas those at design are not.

Figure 7. CFD Results Showing Pressure Field Around LSD Vanes.

Interaction with Other Pressure Nonuniformities

The forces acting on the impeller during overload operation can be further exacerbated by the presence of other pressure nonuniformities in the compressor flow path. A prime example of this is the nonuniformity caused by a discharge volute or collector. Again, several technical papers have been published that document this nonuniformity, including Hagelstein, et al. (1997), and Sorokes and Koch (2000). Essentially, an impeller upstream of a discharge volute or collector will be operating in a nonuniform circumferential pressure field as seen in Figure 8. The impeller can be somewhat shielded from the effect of the volute though the use of vaned diffusers. However, the nonuniformity persists nonetheless.

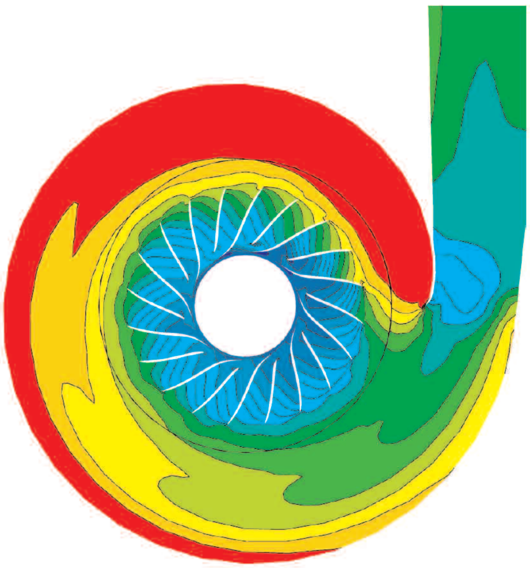

Figure 8. Nonuniform Pressure Field Due to Volute.

As reported in Sorokes and Koch (2000), such nonuniformities, whether caused by a volute, compressor inlet, or other driving mechanism, further increase the level of dynamic forces acting on the impeller because the circumferential pressure distribution tends to become even more skewed when operating in overload. The dynamic strains shown in Figure 9 are for an impeller upstream of a volute operating at various flow rates. As can be seen, the strains are highest in the overload region of the performance map. It should be noted that these strains were obtained during an extensive test program to identify the root cause of the impeller fracture.

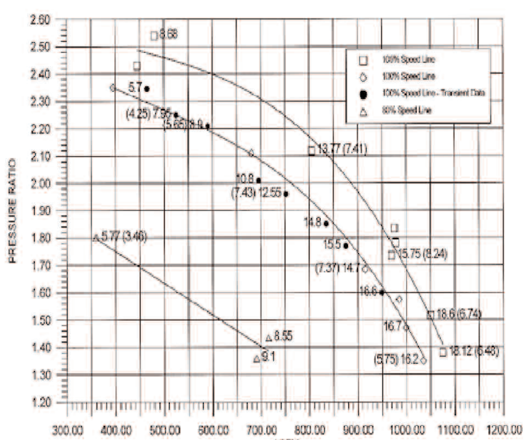

Figure 9. Variation in Impeller Dynamic Strain with Flow.

Wake Effects

There is much conjecture as to whether it is possible for an impeller to excite itself into resonance. Although there is no definitive proof that this can occur, it has been theorized that the blade wakes shed by an impeller could provide the mechanism for self-excitation. Under normal operation, the wake regions are sufficiently small and of reasonably limited pressure magnitude (i.e., the difference between the static pressure in the wake region and the static pressure in the core flow is small). Therefore, there is not enough force or "delta pressure" to cause any difficulties. However, as in the inlet guide vane, the difference between the static pressure in the core flow and static pressure in the secondary zone and blade wakes is higher. It is possible that an impeller blade could interact with the wake shed by the preceding blade (Figure10), resulting in an excitation at blade passing or other frequency (i.e., number of impeller blades times the rotational speed).

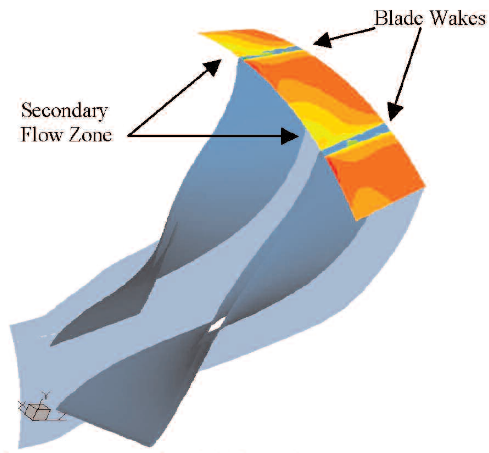

Figure 10. Impeller Blade Wakes.

ANALYTICAL METHODS

Computational fluid dynamics can provide valuable insight into the flow physics and the detrimental forces associated with operation in overload. For example, Borer, et al. (1997), used CFD to investigate possible excitation mechanisms in a problematic stage in a high-pressure reinjection compressor. Their work showed that a high static pressure load was occurring near the leading edge of the impeller while operating in overload. The static pressure distribution for the impeller is shown in Figure 11. The root cause of the pressure load was high negative incidence on the impeller blade caused by operating at a high flow rate (i.e., overload). When the impeller was operating nearer design flow, the pressure load was significantly smaller. In short, the analytical results were helpful in identifying the adverse conditions within the impeller that contributed to the fractures.

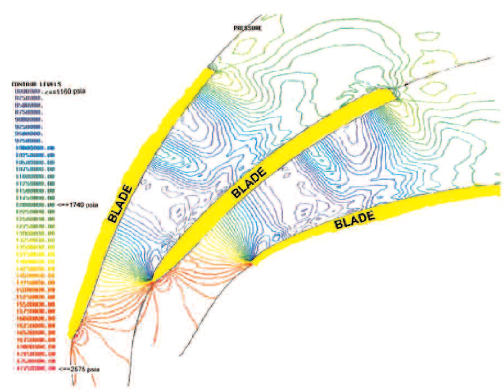

Figure 11. Impeller Static Pressure Distribution During Overload Operation.

CFD can also be used to develop pressure loads or net impeller blade forces that can be applied when conducting finite element analyses to assess the structural integrity of designs. CFD can also be used to determine how said forces vary with flow rate. Historically, CFD was limited to providing steady-state (or time averaged) pressure distributions that could be applied as pressure loads in FEA studies. However, in the past decade, advances in CFD have made it possible to generate unsteady or transient pressure distributions. These unsteady pressures can be translated into a force at a frequency and imposed as a boundary condition in an impeller natural frequency analysis. This offers designers the opportunity to assess possible excitation and determine if changes are needed to avoid a resonance problem.

High-flow coefficient impellers that may be subjected to overload operation are of particular concern. For clarity, high-flow coefficient is defined herein as any impeller having a flow coefficient, ϕ, of 0.12 or greater, where ϕ or phi is an internationally accepted, nondimensional flow coefficient relating flow, speed, and diameter. By their nature, such impellers have very tall leading edges, that is, the length of the blade from hub to shroud at the leading edge is large. The excessive length makes it possible for the blade to flex or flutter if it is subjected to a sufficient dynamic force. The impeller blade leading edge is most susceptible to flexing because it is the tallest and often the thinnest portion of the blade. If the magnitude of the vibrations becomes sufficiently large, a fracture could occur. Therefore, it is imperative that designers understand the magnitude and the frequency of the forces acting on such impellers.

The above issue is of even greater concern on unshrouded or open impellers. Without a cover or shroud to help hold the blades in position, the blades have significantly more freedom to flex or flutter. As a result, operation near stonewall on open impellers is strongly discouraged.

As noted, operation in overload causes high levels of negative incidence on the impeller blade leading edges. This negative incidence causes unbalanced pressure forces between the two sides of the blades. If these forces become dynamic or unsteady because of the presence of upstream IGVs or some other circumferential asymmetry in the flow field, the fluctuating pressure forces could be more than sufficient to induce alternating stresses in the impeller. Assuming the excitation force does not change in frequency, and assuming the frequency aligns with an impeller natural frequency, resonance could occur and ultimately result in the loss of mechanical integrity.

As an illustration, the pressures acting on the leading edge of a high-flow coefficient impeller are shown in Figures 12 and 13. The analytical results given in Figure 12 are for the impeller operating at its design flow rate while the results in Figure 13 are for overload operation. In both cases, the plot on the right is for the pressure surface of the blade, while the plot on the left is for the suction surface. Also, the pressure scales are consistent among all of the figures.

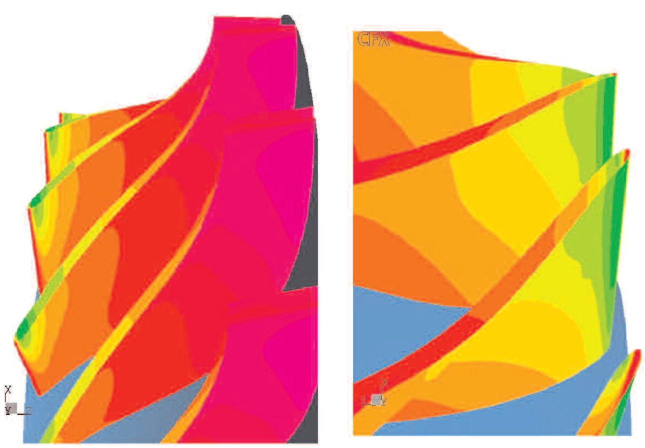

Figure 12. Static Pressure Load on High Flow Coefficient Impeller Operating at Design Flow.

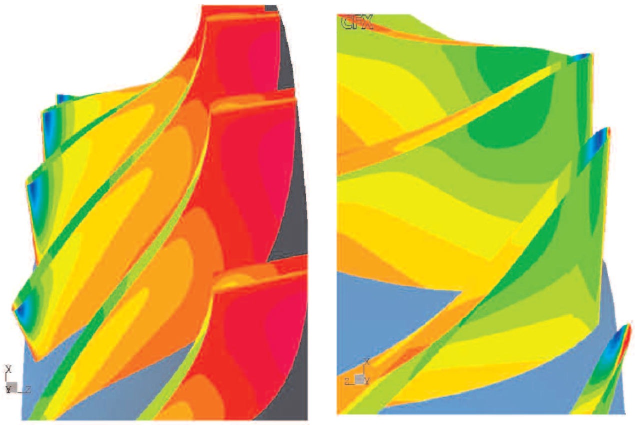

Figure 13. Static Pressure Load on High Flow Coefficient Impeller Operating at Overload.

As can be seen, the variation in pressure loads is much higher when the impeller is operating in overload. One can note the wide variation in pressure on the pressure side leading edge at overload versus the considerably lower variation at design. Since the pressure on the suction side of the leading edge is fairly consistent between design and overload, it is also clear that the delta pressure suction to pressure surface is higher in overload than at design. Note further that the pressure distribution in Figure 13 bears a resemblance to the schematic of the blade leading edge fracture in Figure 4. This alone is not definitive evidence that a fracture would occur. However, were the pressure forces on the impeller unsteady (which they certainly would be), and were the unsteadiness to be at a frequency that aligns with the natural frequency of the blade leading edge (i.e., the blade leading edge mode), a blade fracture like the one shown in Figure 4 could occur.

DESIGN OR OPERATIONAL CONSIDERATIONS

It is critically important that end users and OEMs discuss the full range of conditions that a compressor will face in operation. For example, if the process engineers or end user know that there is a chance for extended operation in overload, the OEM may want to undertake more detailed analyses of the proposed equipment. These analyses may include detailed CFD and FEA analyses as described above to understand the potential ramifications of overload operation. These analyses can be used to:

- Establish the maximum safe operating limit,
- Help the end user understand the risks of operating in that flow regime, or
- Provide guidance to the OEM regarding how to modify component designs to minimize the risks by increasing the robustness of the design.

Regarding the latter, it may be possible to change the number of vanes in adjacent stationary hardware to avoid natural frequency interference issues. Alternatively, the design or manufacturing technique used for an impeller could be changed to make it less susceptible to interference issues. For example, the OEM may choose to apply more stringent criteria for welds (i.e., weld shape requirements, more rigorous inspection techniques) or possibly resort to single-piece impeller fabrications. Of course, a natural frequency in the speed range is acceptable as long as its response does not exceed allowable limits imposed by the end user or OEM. In short, by having full knowledge of the potential modes of operation, the OEM can undertake efforts to minimize the risks.

One additional example illustrates how this happened on a recent compression project. The client, a major oil and gas producer, needed more compression on an existing offshore platform to boost gas going to shore. Because of the declining pressure nature of the field, the compressors were designed with both the present and several future conditions in mind to allow for change-out of internal parts over the life of the field to accommodate the changing operating conditions. Because the application required very high horsepower drivers (>40 MW), and because the early years of operation had the highest inlet pressures, only a few stages (impellers) were required to meet the specified discharge pressure. This resulted in a fairly high horsepower per stage relative to the end user's and OEM's experience for this particular size unit. In contrast, operation in future years at much lower inlet pressures would require more stages (impellers) to meet the same desired discharge pressure. Therefore, in the later years of operation, the horsepower per stage would drop into more a comfortable range.

The end user also desired that if one unit had to be taken out of service, the remaining units would need to accept more inlet flow; that is, operated in overload, a flow rate 30 percent to 50 percent higher than the design flow rate. Because the stage power was high at the normal design point, the OEM was concerned about the prospect of even higher powers being absorbed in an off-design mode. When the end user consulted the OEM about the potential operation at higher flows, the OEM responded with an engineering study to determine the magnitude of the dynamic stress. The study included an innovative combination of transient CFD and FEA. The resulting analyses indicated that the original impeller geometry was acceptable but with relatively low safety margins. The OEM then determined that the calculated stresses could be significantly reduced with very minor changes to the geometry of the impeller, notably in the blade to shroud fillet weld. This substantially increased the factor of safety, and providing confidence that the impeller integrity would not be at risk due to overload operation.

If the end user had not consulted the OEM, it is uncertain whether or not a problem would have occurred. However, by having an open dialog, and by being proactive in performing an analysis, the risk of a problem was much more remote.

Of course, it may not be possible to eliminate all of the risks associated with overload operation. Even the most advanced analyses are but approximations of the real world. Therefore, one cannot be assured that such analyses will capture all of the potentially damaging phenomena within the compressor flow path. Further, even the most robust designs will fail if subjected to the right excitation mechanism, i.e., one that aligns with the natural frequency of the impellers. Even a solid ring would fail if subjected to the right excitation. At some point, common sense must prevail. The end user and OEM must face the reality that the safest approach is to avoid operating in the overload region of the performance map. If the compressor might break if you run there, do not run thereóor be prepared to undergo regular overhauls to check for internal damage.

End users can employ overload control systems to ensure that a compressor does not operate beyond some agreed upon maximum capacity. These are implemented by incorporating algorithms in the control system that limit driver operation (speed, load, etc.) or restrict the movement of control valves to keep the compressor in a safe region on its performance map. Some argue that such overload controls limit production and decrease profitability. However, when this reduction is weighed against the costs and lost production associated with equipment failure, limiting overload operation does not seem to be a bad choice.

CONCLUSIONS

The most important conclusion to be derived from this work is that, contrary to many commonly held beliefs, operation in overload can subject a centrifugal compressor to adverse forces. In fact, in some circumstances, overload operation can be just as detrimental to component structural integrity as surge.

Of course, it is critical that the end user and OEM come to an understanding on the meaning of overload operation. The term "overload" has many connotations, so it is important that all parties adopt a common definition. Because one person's "overload" might well be another's normal operation, a more rigorous definition must be applied when specifying compressor flow range requirements or discussing how the compressor is being operated in production. It might also be possible to adopt an industrywide standard that defines "overload" as operating a compressor at flow rates that exceed the maximum flow rates shown on the performance map provided by the OEM. This would put the onus on both the OEM to provide an accurate prediction of overload capability and the end user to properly assess their need to operate at such high flow rates.

As seen herein, in some reported cases, prolonged operation in overload can lead to impeller fractures. Overload operation can also exacerbate structural natural frequency interference issues that may exist within a compressor flow path. Forces and/or pressure nonuniformities tend to be greater when operating at flow rates much higher than design. Such increases are caused in large part by the increased incidence levels on impeller blades or adjacent stationary vanes (i.e., vaned diffusers).

Advanced analytical tools such as computational fluid dynamics or finite element analysis help quantify the magnitude of the forces associated with overload operation. Such analyses can also be used to mitigate risk. Designs can be modified to reduce the potential for harmful interferences, or operating limits can be derived so as to avoid risky portions of the performance map. However, the most sophisticated analyses and most advanced manufacturing methods cannot eliminate the risk of component failures due to overload operation. Common sense dictates that the most effective way to eliminate such risk is to avoid high risk operating conditions. Simply put, if there is increased risk of mechanical failure by running at a portion of the performance envelope, the risks must be weighed against the potential gains and the cost of maintenance or replacement of the equipment.

In conclusion, though not receiving as much attention as surge, prolonged operation in overload can have very detrimental effects on a centrifugal compressor. End users and OEMs alike need to be cognizant of the potential risks associated with operating in this portion of the performance map. End users accept the risks associated with surge and, despite the extra horsepower consumed, often run their compressors on recycle so as to avoid surging the units. As noted, the forces associated with surge and overload are similar, yet the industry has not taken steps to protect equipment from overload operation. Failure to recognize the risks associated with overload operation can have a devastating impact on compression equipment, production, profitability, and engineering careers.

NOMENCLATURE

ACFM	= Actual cubic feet per minute
A_0	= Sonic velocity of gas at impeller inlet
b	= Impeller blade height
Cm	= Impeller meridional velocity
D_2	= Impeller exit diameter
IGV	= Inlet guide vane
LSD	= Low solidity vaned diffuser
N	= Rotational speed in rpm
Q	= Inlet volumetric flow in ACFM
r	= Impeller radius
U_1	= Impeller inlet peripheral velocity
U_2	= Impeller exit peripheral velocity
U_2/A_0	= Machine Mach number
V	= Gas velocity
W1	= Impeller inlet relative velocity
ϕ	$= \dfrac{700Q}{N D_2^3}$
ρ	= Gas density

REFERENCES

Borer, C., Sorokes, J. M., McMahon, T., and Abraham, E. A., 1997, "An Assessment of the Forces Acting Upon a Centrifugal Impeller Using Full Load, Full Pressure Hydrocarbon Testing," *Proceedings of the Twenty-Sixth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 111-122.

Eckert, L., 1999, "High Cycle Fatigue Cracks at Radial Fan Impellers Caused by Aeroelastic Self-Excited Impeller Vibrations, Part 1: Case History, Root Cause Analysis, Vibration Measurements," Proceedings of DETC99, ASME Design Engineering Technical Conference.

Fisher, E. H. and Inoue, M., 1981, "A Study of Diffuser/Rotor Interaction in a Centrifugal Compressor," IMechE Journal Mechanical Engineering Science, *23*, (3), pp. 149-156, London, England.

Hagelstein, D., Van den Braembussche, R., Keiper, R., and Rautenberg, M., 1997, "Experimental Investigation of the Circumferential Pressure Distortion in Centrifugal Compressor Stages," ASME Paper No. 97-GT-50.

Kushner, F., 1980, "Disc Vibration—Rotating Blade and Stationary Vane Excitation," ASME Journal of Mechanical Design, *102*, pp. 579-584.

Sorokes, J. M. and Koch, J. M., 2000, "The Influence of Low Solidity Vaned Diffusers on the Static Pressure Nonuniformity Caused by a Centrifugal Compressor Discharge Volute," ASME Paper No. 00-GT-454.

Sorokes, J. M., Borer, C. J., and Koch, J. M., 1998, "Investigation of the Circumferential Static Pressure Nonuniformity Caused by a Centrifugal Compressor Discharge Volute," ASME Paper No. 98-GT-326.

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