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ON-SITE DETERMINATION OF THE CONTROL LINE FOR INTEGRALLY GEARED COMPRESSORS FOR AVOIDANCE OF IMPELLER FATIGUE FAILURES

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Mr Süß has received his Diplom degree from the University of Cooperative Education in Ravensburg (2001).

ABSTRACT

In this lecture a root cause analysis (RCA) is presented for a fatigue failure of a semi-open impeller of an integrally geared compressor. The compressor had been in service for about one year in an air separation plant, when this failure occurred on the 3rd stage impeller.

A systematic root cause analysis confirmed that the impeller design and manufacturing was flawless, but revealed that the control line, as implemented as part of the usual commissioning process, did not protect the compressor from operation in a zone in which aerodynamically induced forces can lead to a fatigue failure.

The conclusion of the RCA, as outlined in the paper, is based on three major topics:

1. From R&D testing a flow regime could be identified where high blade vibrations occur due to an aerodynamically induced excitation.
2. From analysis of commissioning and trend data and the comparison with shop test results it could be verified, that the control line was not properly set on-site, and that the machine was indeed operated in a dangerous zone.
3. All other hypothetical root causes could be ruled out during the course of the RCA.

The implemented method for the correction of the identified deficiency is portrayed.

This contribution shows one example how compressor integrity can be endangered and offers a method that may be employed to end up with a setting of the anti-surge system which not only prevents the unit from falling into surge, but also from operation in potentially harmful flow regimes.

INTRODUCTION

There are a number of processes in industry which require air separation. Most frequently pure oxygen and pure nitrogen are required, but also the other elements are of economic interest.

Since ambient air is used as feedstock for air separation units (ASU), the raw material is free of charge. Energy consumption is the key factor, and therefore economics of such plants are directly driven by the power requirement of the compression train. Accordingly highest efficiencies are needed, what is even more emphasized in the ongoing trend for 'green technology'.

A compression close to an isothermal process provides lowest power consumption, accomplished by inter-cooling of the compressed air after each compression stage.

The use of semi-open (unshrouded) centrifugal impellers allows for high tip speeds, what leads to a high power density of the machine. This makes a small compressor frame size feasible, resulting in low capital expenditure. The integrally geared compressor as shown in Figure 1 has proven to be an ideal machine for such purposes. Maximum efficiency is provided not only by the aerodynamic stage design, but also by the favorable axial inflow to each impeller as well as the feature of optimum selection of the speed of each pinion.

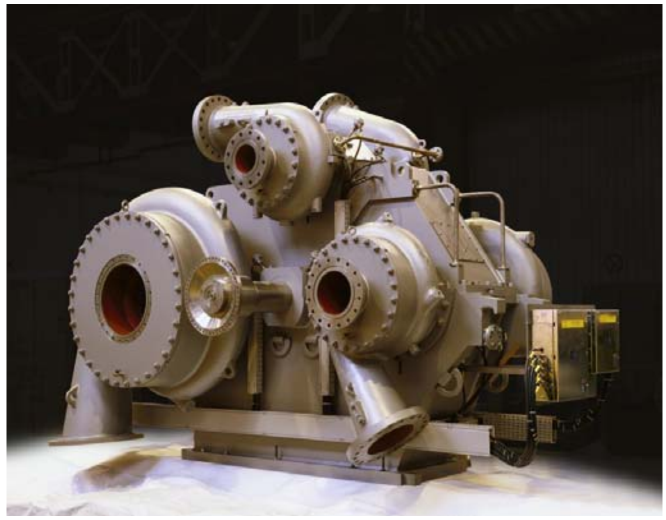

Figure 1: Integrally geared compressor

From time to time failures of blades of semi-open impellers were reported. For the required high availability figure for such ASU plants it is most important to have comprehensive knowledge on potential root causes for impeller failures. This knowledge has to cover all aspects from impeller design and material, fabrication, assembly, commissioning and operation.

While all of these aspects have to be addressed as part of a systematic root cause analysis (RCA) of an impeller failure, specific attention is paid in this paper to commissioning and operation of the compressor.

Aerodynamic excitation mechanisms, which can occur within the performance map, are often underestimated. Focus is often placed only on avoidance of deep surge. However, local stall phenomena can lead to excitation of destructive blade vibration, but detection of these phenomena is not always easy.

The current paper describes a case where the control line did prevent the machine from falling into deep surge, but nevertheless the compressor could be operated in a harmful zone.

SUBJECT AND TASK

In August 2009 one of the two main air compressors, which will be described in the next paragraph, was shut down due to high shaft vibrations. After opening of the machine a damage of one impeller blade at the 3rd stage was found, cf. Figure 2.

This impeller had been running from October 2008 till August 2009. The hours in operation till failure were 5272, the number of starts in this period was 35. The damaged impeller was sent to the OEM's shop for further investigation.

Figure 2: Photographs of failed impeller

Compressor Data

The compressor being subject of this paper is a three stage integrally geared compressor (STC-GV 100-3). It is the main air compressor (MAC) of an oxygen plant in India, which comprises two 50% units.
Each compressor is driven by a synchronous 50 Hz motor with a nominal power of 12000 kW. Typical for this region is a speed variation due to grid frequency perturbation by +3% to -5%. The suction pressure was specified with 0.976 bar at an inlet temperature between 8.5°C and 36.5 °C and the discharge pressure is 5.94 bar.

Findings from Macroscopic Investigation

The fracture surface of the failed impeller shows macroscopic characteristics of a fatigue fracture, cf. Figure 3. The crack start is located at the blade surface on the suction side, approx. 8 mm above blade ground and approx. 57 mm apart from the inlet edge. Referring to the direction of rotation, the crack started at the back side of the blade. The whole length of the fatigue crack is 88 mm.
6 arrest lines were counted on the fracture surface. The whole fracture surface is very smooth. The crack starts in the near of a machining mark and follows it at the blade surface till transition to forced fracture.
Material imperfections or corrosion in the near of the crack start were not found.

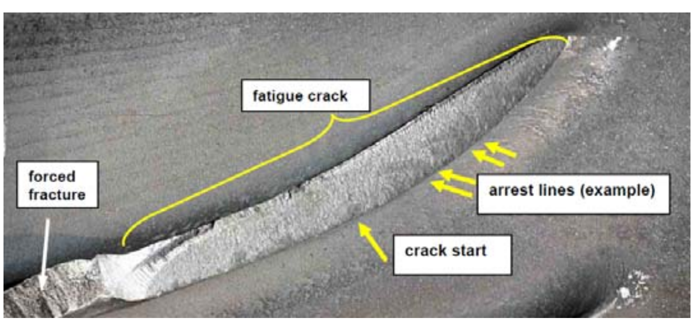

Figure 3: Photography of crack surface

Compressor Cross Section

A cross sectional view of the compressor is shown in Figure 4 zooming in on the 3rd stage. This view indicates the variable inlet guide vanes (IGV) in front of the impeller as well as the vaned diffuser, which is of the low solidity type (LSD). A contour ring secures a low tip clearance of the semi-open impeller.

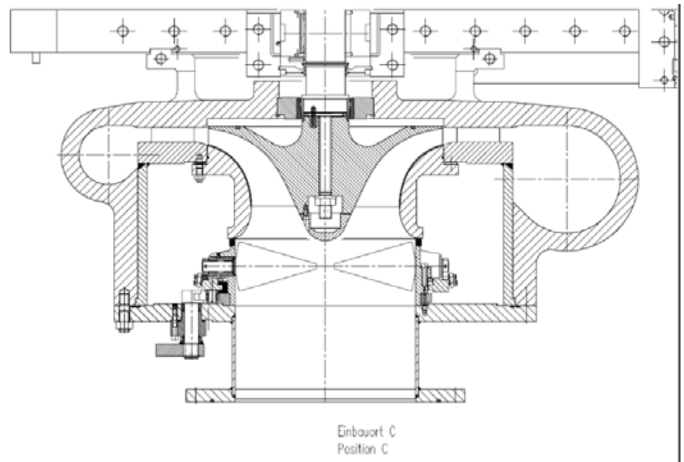

Figure 4: Compressor cross section

Mechanical Design Data

The mechanical key design data of the impellers are given in Table 1. They were fully referenced and do not form any risk in themselves due to extreme parameters.

Stage No.	1	2	**3**
Impeller Outer Diameter [mm]	1030	835	**655**
Cover Disk	No	Yes	**No**
No. of IGV Vanes	12	12	**12**
No. of LSD Vanes	8	-	**8**
Operating Speed [rpm]	7181	7181	**11892**
Impeller Tip Speed [m/s]	387	314	**355**

Table 1: Impeller data

ROOT CAUSE ANALYSIS

Different methodologies may be applied for a systematic and comprehensive root cause analysis. The authors have good experience with the failure tree, which is basically the same as the fishbone method or the cause and effect diagram.
The failure tree as shown in Figure 5 covers the main topics for an impeller fatigue failure.

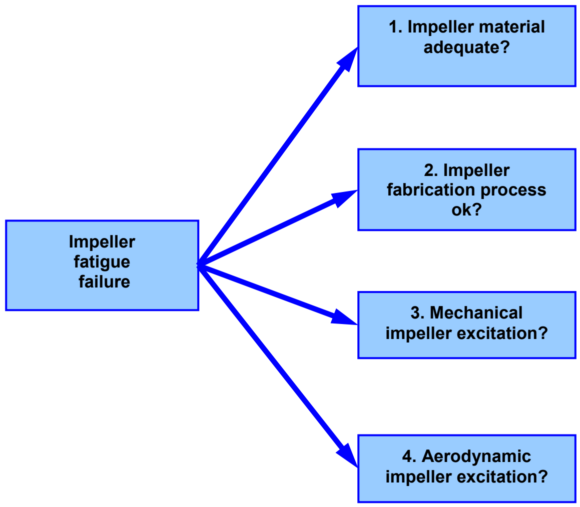

Figure 5: Failure tree for impeller fatigue failure

The first topic 'Impeller material adequate?' shall verify, whether the impeller material was suitable for this application. The potential causes for inadequate material are depicted in Figure 6. All sub-topics were checked and no indication was found, that the root cause of the failure was due to inadequate impeller material.

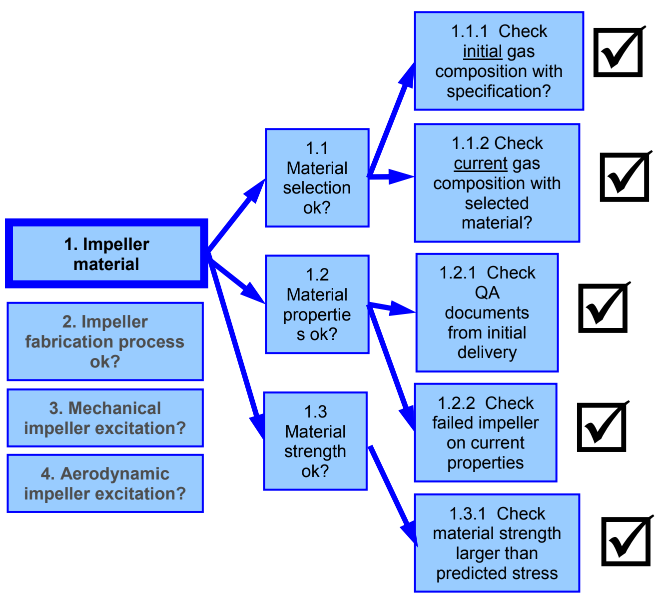

Figure 6: Impeller material topics

The second topic 'Impeller fabrication process ok?' shall verify, whether the impeller manufacturing process could have initiated the failure, cf. Figure 7. In fact, a machining mark was found at the blade root, but this determines only the *location of crack initiation* and is not considered as the root cause of the failure. Such a machining mark occurs unavoidably at the transition from flank milling of the blade to the radius at the blade root and is present on all milled impellers without causing trouble. On the basis of a huge available reference base it can be concluded, that another mechanism than stress concentration by the machining mark must have led to the failure. Naturally, any crack will have its origin in either a material inhomogeneity or in a geometry imperfection, both being unavoidable in a real world and harmless, if of minor extension.

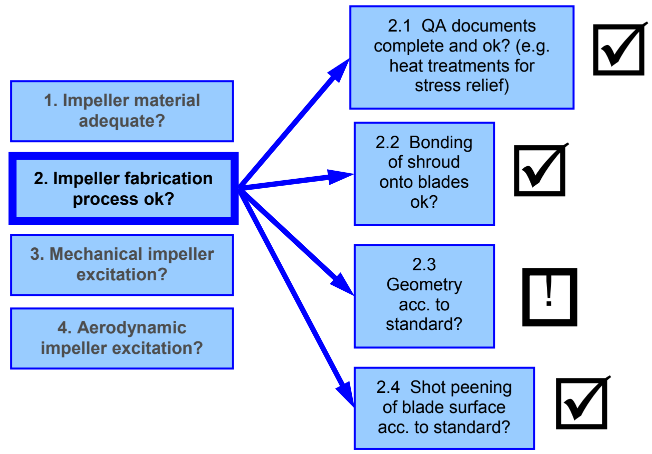

Figure 7: Impeller fabrication topics

The third topic 'Mechanical impeller excitation?' shall verify, whether there is any evidence that a mechanical impeller excitation could have been the excitation source leading to the fatigue, cf. Figure 8. In particular a possible rotor/stator rub must be considered, if the clearance of the contour ring was not properly set or if unexpected high pipe deformations generate large nozzle loads, which in turn can deform the volute and bridge the gap between blade and stator. Such a rub can cause a broad-band excitation and the location of excitation at the blade tip is ideal to excite the first blade bending mode shape. However, in this case no indication was found at all pointing to a rub as root cause. The damage found at the contour ring was evaluated as consequential damage of the blade failure.

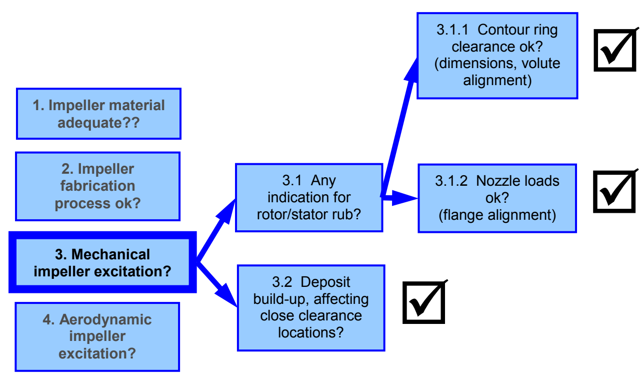

Figure 8: Mechanical impeller excitation topics

The fourth topic 'Aerodynamic impeller excitation?' (Figure 9, Figure 10) requires input from dedicated analysis.

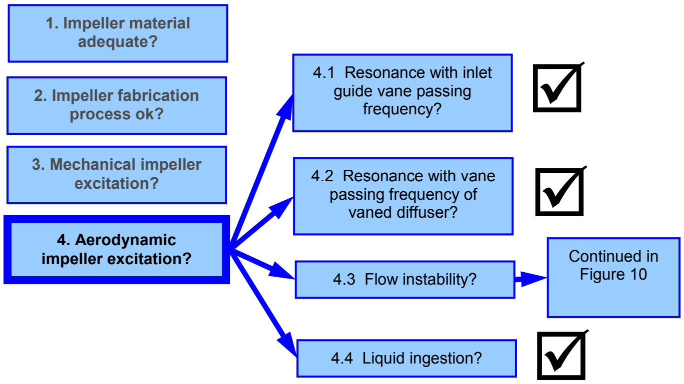

Figure 9: Aerodynamic impeller excitation topics

The sub-topic 4.1 checks whether or not a resonance of an impeller blade bending mode with the IGV passing frequency prevails. The Campbell-diagram as shown in Figure 11 confirms that the 12X excitation line of the IGV does not excite the first blade bending. Thus the design criterion for the IGV/impeller interaction is fulfilled.

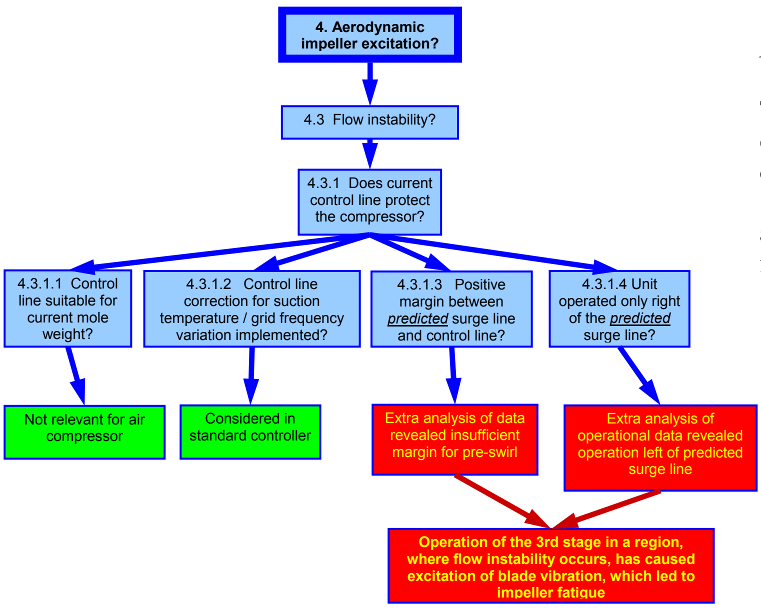

Figure 10: Aerodynamic impeller excitation sub-topics

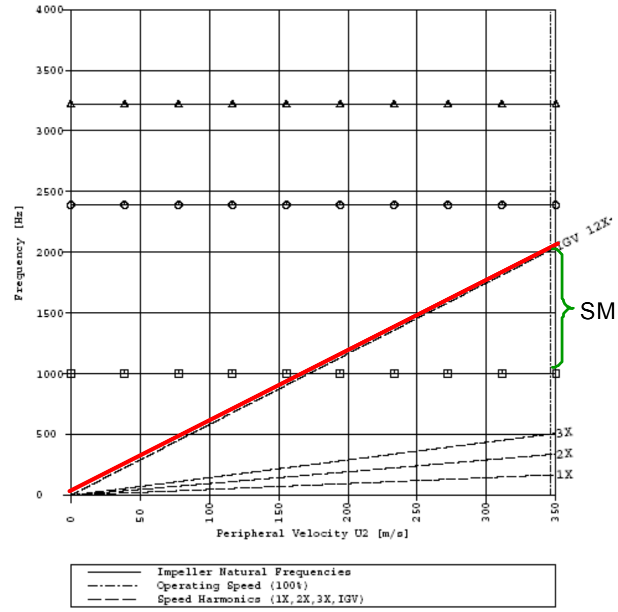

Figure 11: Campbell-diagram of the failed stage

The input for this Campbell-diagram is provided from a finite-element analysis of the impeller. Figure 12 displays the stress distribution for a vibration in the first blade bending mode. Similar plots can be generated for the higher blade modes but are not shown here, since they have never been involved in failures of this impeller type.

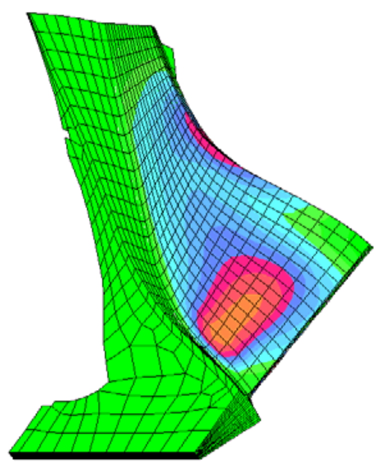

Figure 12: Dynamic stress distribution from impeller finite element analysis

While the excitation originating from the IGV acts primarily onto the impeller blades, the interaction of the diffuser vanes with the upstream impeller needs additional consideration with respect to excitation of the eigenmodes of the whole impeller structure including the back plate. The nodal diameters of these modes are preferably excited by a pressure pattern from the interaction between impeller blades and diffuser vanes, matching these nodal diameters. This mechanism is subject of

sub-topic 4.2 of Figure 9. The evaluation requires sorting all impeller modes according to their individual number of nodal diameters and plotting them in the so-called interference diagram as depicted in Figure 13. The underlying theory of this diagram can be found in [1]. Recent in-house research activities (compare König et al. [2]) revealed that the interference diagram may be insufficient under unfavorable conditions, but this is not considered relevant for the current investigation. In the interference diagram the harmonics of the shaft speed – a non-dimensional frequency - are plotted versus the number of nodal diameters of the structural eigenmodes. Each dot in the diagram represents one structural eigenmode (sorted by nodal diameter), whose respective eigenfrequency is scaled with the compressor shaft speed. The zigzag line - which is strictly spoken only a connecting line of discrete points - represents the aerodynamic excitation due to rotor/stator-interaction.
Resonance occurs if

1. the zigzag line crosses one of the discrete points representing the impeller eigenfrequencies, and furthermore this happens
2. at a harmonic order that corresponds to the number of vanes of a stator component (only the fundamental of the vane passing frequency is considered).

In the current case the only relevant harmonic order of the stator components is determined by the LSD vane count of z=8; a horizontal line is drawn at the respective harmonic excitation order. It can be deduced from Figure 13 that a comfortable safety margin (SM) of more than 24% exists. This safety margin easily covers line frequency variations.

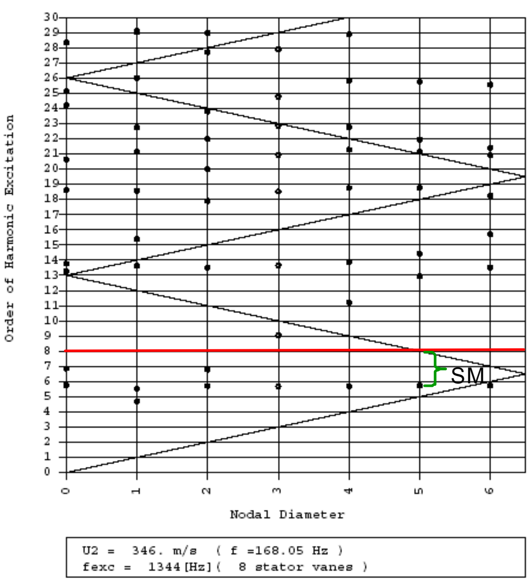

Figure 13: Interference diagram of failed impeller (Case: Design-Normal 47.5Hz)

Before we turn our attention to sub-topic 4.3 of Figure 9, we can quickly resolve sub-topic 4.4 'Liquid ingestion?'. If liquid is swallowed by an impeller, e.g. due to condensate being accumulated during stand-still in a piping sag, strong blade excitations will occur. In the current investigation no evidence of liquid ingestion was found, so that this topic can be ruled out as well.

The sub-topic 4.3 of Figure 9 'Flow instability' is the most complex one of all the listed sub-topics. For a better understanding of the criticality of different operating conditions an excursion is made at this place to results from R&D testing. With the awareness of these R&D results we afterwards will step back into the RCA.

Some remarks on phenomena when approaching the surge

When talking about stall and surge there is a considerable confusion about terminology in the turbomachinery community. It is normal with compressors that as the mass flow is reduced the pressure rise increases. Generally a point is reached at which the pressure rise is a maximum and further reduction in mass flow leads to an abrupt and definite change in the flow pattern in the compressor. Beyond this point the compressor enters into either a stall or a surge; regardless of the type of change occurring, the conventional terminology for the point of instability is the surge point and the line marking the locus of these points for different rotational speeds (or inlet pre-swirl) the surge line (compare Cumpsty [3]).
True surge is a condition where the overall averaged mass flow varies with time, so that the entire compressor changes more or less in phase from being unstalled to stalled and back again. The process may be so violent that the mass flow is reversed and previously compressed gas emerges out of the inlet; this is sometimes called deep surge. During some parts of the surge cycle the flow may be transiently in rotating stall.
The operating point at which the flow in the compressor becomes unstable and progresses to rotating stall or surge depends not only on the compressor but on the system in which it is operating: the duct, volume and throttle in a simple test configuration.
There are many uncertainties concerned with stall and surge: uncertainties of prediction for a given design and uncertainties associated with the conditions of operation. To cope with these it is normal to specify a quantity known as the *surge margin*. It should be noted that even though the compressor may not go into surge but into rotating stall it is conventional to refer to *surge margin* and *surge line* on the compressor operating map (compare Cumpsty [3]). Sinces the differences are crucial in the current investigation a distinction is made throughout this paper: the *surge line* is considered the line where true surge - usually deep surge - occurs, whereas the *acceptable operation point limit* gives the limit where arbitrary flow instabilities occur.
Usually, flow instabilities originate locally and do not directly cause the whole compression unit to become unstable; however, they may be a precursor of true surge. When approaching the surge line of a turbocompressor the likelihood of flow instabilities increases considerably. Under unfavorable conditions, local flow instabilities may also be the root cause of impeller failures. Their accurate prediction is very challenging. Even the application of state-of-the-art calculation tools like computational fluid dynamics (CFD) does not allow a reliable forecast of the exact onset of phenomena like rotating stall. Usually, OEMs rely on experience and empirical correlations to avoid critical compressor operating conditions. This implicates

that a certain distance from the acceptable operation point limit must be maintained throughout the entire performance map. This limit is generally predicted by the compressor vendor based on the predicted stage performance curves (from theoretical or stage experimental data) of the compressor. When the commissioning engineer determines the acceptable operation point limit experimentally as part of the on-site commissioning process, he typically faces the difficulty of having no accurate flow measurement available, which would allow for a direct comparison with the predicted values. Therefore deep surge, i.e. the flow instability of the overall system, is generally used to determine the acceptable operation point limit (in this case the surge line). However, with a close observation of all available signals (e.g. shaft vibrations, pressure transducers) and a lot of experience it is possible in many cases to detect indications of local flow instabilities already before the whole system is forced into deep surge. Proceeding in such a sensitive way is usually not applied, because:

- individual characteristics of each single machine make it hard to state generally valid quantities for the acceptable limit of operation,
- useful further sensor signals are not available due to practical reasons,
- operators usually have a strong economic interest to operate a compressor at the smallest possible turndown and
- the risk of causing an impeller failure due to an insufficient margin is not as well-known as desired.

For safety reasons the so-called control line is defined, which shows a significant margin (typically 5 to 10 percent) to the acceptable operation point limit. The compressor should not be operated beyond this line. A surge control system is applied that monitors the compressor's operating point in relation to the acceptable limits as established during commissioning and, if necessary, opens a recycle or blow-off valve to increase flow through the compressor to avoid surge. The ability of the control system to prevent surge depends on the accurate determination of the surge line and the correct measurement of the true operating point of the compressor, e.g. in some cases corrections for suction temperature or grid frequency variations are implemented. The system starts to open the recycle valve when the compressor gets close to the surge line within a certain margin (e.g. when crossing the control line).
If the control line is set according to 'normal practices' (for the practical constraints compare the bullet points before) local flow instabilities may establish if the compressor is operated at part-load. For a correctly designed compressor this is not considered to be a problem, but in very rare occasions even "correctly" (based on the state-of-the art) designed compressors may experience failures. These failures may be caused by local flow oscillations due to flow instabilities, and the additional accidental coincidence between excitation frequencies and structural eigenfrequencies of an impeller. Usually, such conditions cannot be predicted in the design phase, and even detailed root cause analyses may fail to unambiguously explain the origin of the problem. For multi-stage compressors the interactions are much more involved than for single-stage machines, and it is not a straightforward task to determine the stage that is most susceptible to critical flow instabilities. Also, it is known from analyses that the different stage characteristics of multi-stage machines may lead to conditions, where one stage is operated beyond its acceptable operation point limit, whereas the operating point of the whole compressor still shows an acceptable margin to the overall operation limitations. If such a condition occurs, pressure fluctuations caused by local flow instabilities may emerge and remain undetected, because the compressor as a whole is still able to deliver a stable output. To gain a better understanding of the flow mechanisms leading to the abovementioned unfavorable operating conditions detailed experiments have been carried out over the years at the OEMs R&D test bed. Two typical part-load flow instabilities found in the experimental data will be discussed in the following subsection. The first phenomenon is the well known inducer *stall* (which may be rotating or non-rotating), and the second phenomenon, which has attracted little attention in the centrifugal compressor community so far, is the so-called rotating *instability*.
For a more general overview on flow instabilities the textbooks of Cumpsty [3] and Japikse [4] are recommended to the interested reader. With respect to rotating stall an extensive amount of literature is available (e.g. [5-9]). This list is by far not exhaustive but may be a starting point for a deeper literature research.

RESULTS FROM R&D TESTING

Evidence of inducer stall

It is known from literature that inducer stall may create conditions sufficient to permit the stage to enter into a strong surge, which is especially true for high pressure machines. On the other hand, at very low pressure ratios, it is possible to stall the inducer massively while the stage continues to operate stably (compare Japikse [4]). In the OEMs database no impeller failures have been reported that can be solely explained with the existence of inducer stall. Nevertheless, inducer stall may lead to unsteady pressure fluctuations, which are especially strong if the flow state in the inducer changes from separated flow to attached flow in an intermittent manner. The separated inducer flow does not only constitute a direct excitation source, but may also interact with other flow instabilities. If inducer stall can be avoided within the whole operating map of a compressor this is definitely a means to reduce the risk of critical operating conditions.
Inducer stall can be detected at the inlet of a compressor stage by inserting a thermocouple into the contour ring (compare Figure 14). The thermocouples will read substantially higher than the inlet temperature due to the existence of backflow which heats up the thermocouple above the approaching air temperature. Frequently, the existence of the backflow or recirculation cell can be found a long distance upstream, equivalent to many impeller eye diameters (compare Japikse [4]).

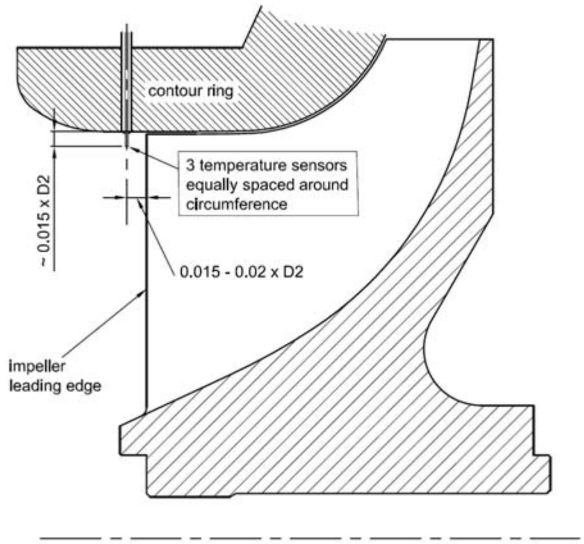

Figure 14: Inducer stall and instrumentation

The influence of inducer stall on the operating map of a centrifugal compressor can be exemplarily deduced from Figure 15 and Figure 16. Both figures are valid for constant speed and different settings of the inlet guide vane. The results are derived from a systematic investigation into the near-stall characteristics of an unshrouded impeller at the OEMs R&D test facility. This impeller is similar to the one under investigation in the current root cause analysis. In Figure 14 the isentropic head is plotted on the ordinate versus the inlet volume flow on the abscissa. In Figure 16 the total temperature rise of the test stage is plotted versus the same x-coordinate as used in Figure 15. In both figures the operating points where deep surge was detected are labeled with red stars, connected with a red solid line. This line is the true surge line. It can be clearly seen the true surge line is located far left of the blue borderline, which gives the acceptable operating point limit (the conventional “surge line”). The envelope between the blue and the red line gives the operating regime where local flow instabilities occur, but where the compressor still operates without falling into deep surge. This operating regime is highlighted as the transparent red area. The extent of this area indicates that a distinction between the true surge line and the conventional surge line is advisable for the impeller under investigation. The differences between the two lines are especially pronounced for positive pre-swirl (compare the curves for +50 and +75 deg swirl angle). In the R&D study the blue borderline was defined as the limit, where recirculation did occur. A typical indication for a recirculation zone can be deduced from the total temperature rise in Figure 16: if the total temperature curve traverses an inflexion point when reducing the volume flow this is a strong indication for backflow. In the R&D study this observation could be further confirmed by means of temperature measurements upstream of the impeller (not presented in the paper). Based on the experimental arrangement depicted in Figure 14 the signals of three thermocouples were evaluated, and clearly elevated temperatures were measured when approaching the blue borderline. It is important to note that for a positive pre-swirl of +75 deg the slope of the isentropic head curve to the left of the border line in Figure 15 does not give any indication of flow instability in the overall compression system. Nonetheless, to avoid inducer stall, the admissible operating range should be restricted to the blue borderline.

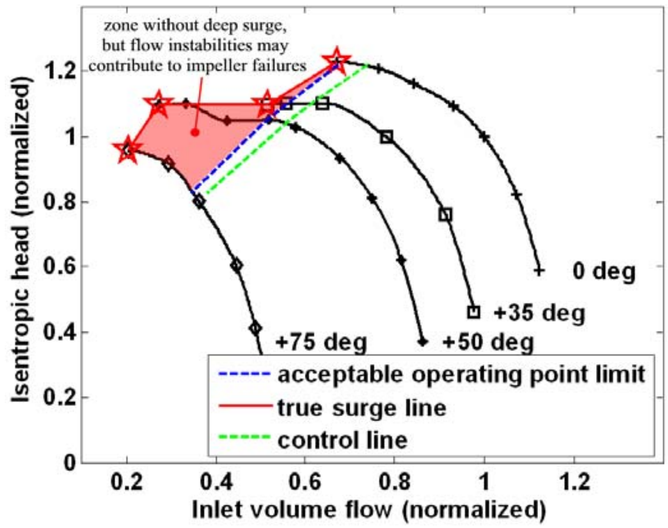

Figure 15: Compressor map from R&D testing

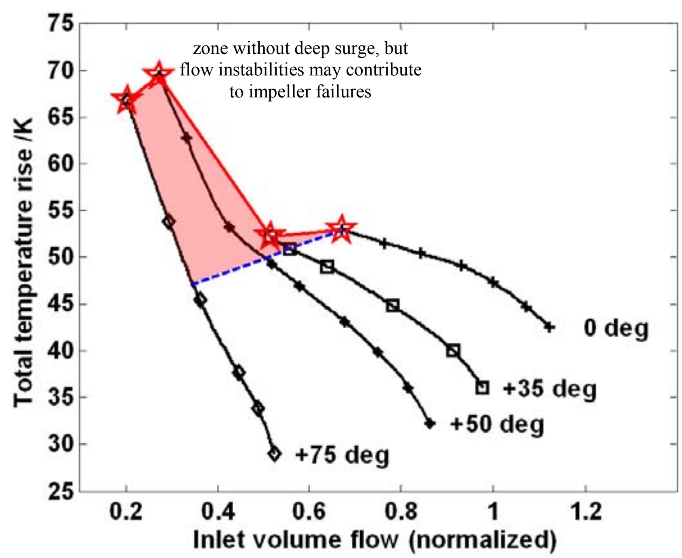

Figure 16: Total temperature rise from R&D testing

For multi-stage machines, as already mentioned, it may be possible that one stage operates beyond its single-stage operating point limit, whereas the overall machine still operates stably. In such a situation local flow instabilities - as inducer stall - are very likely to occur. Though uncommon, such conditions cannot be completely ruled out in the design of a multi-stage compressor. Temperature measurements upstream of the impellers are one way to minimize the risk of local flow

instabilities.
In this context two important issues should be emphasized: Firstly, the abovementioned effects cannot be captured by standard instrumentation; secondly, the instrumentation needed to resolve local flow instabilities is currently too susceptible to the harsh environment conditions experienced in the field to be used as a standard procedure.
For these reasons the approach to use thermocouples upstream of the compressor inlet is currently not suitable for widespread usage. An alternative way will evolve from the next chapters of this paper.

Rotating instability and high blade vibrations

Generally, stall occurs when the flow locally separates, leading to regions of reversed flow. In the case of rotating stall, one or more stalled passages (stall cells) rotate around the circumference at a fraction of the rotor speed. The stall cells are characterized by having reduced or no through-flow. Different types of rotating stall exist in centrifugal compressors, the most common one occurring in vaneless diffusors. Impeller rotating stall and rotating stall in vaned diffusors (uncommon) may also occur.
A generalized form of rotating stall is the so-called *rotating aerodynamic instability* as reported by Kameier [10] and Kameier and Neise [11]. Unlike rotating stall, the pressure fluctuations of a rotating instability are unsteady in a frame rotating with the instability pattern. An observer moving with an *instability cell* would measure a discrete frequency signal associated with the pressure fluctuation, whereas, moving with a *rotating stall cell*, the pressure would remain constant. One can imagine a rotating instability as a rotating loudspeaker that is moving relative to the impeller in the circumferential direction. In a stationary frame of reference the rotating instability manifests, according to Baumgartner [12], as a range of frequencies with a characteristic frequency signature; rotating stall, on the other hand, leads to a single discrete frequency with possible higher harmonics at most.
At the OEMs R&D testing facility a strong indication of a rotating instability was found for the same impeller that experienced inducer stall as described in the previous subsection. Correlating the measured signals from the stationary and the rotating frame of reference the typical signature of a rotating instability could be detected when the R&D compressor was operated close to the acceptable operation point limit. The detailed analysis of a compressor run-up close to this limiting line revealed that rotating stall characterized by four stalls cells occurred within a certain speed range. When further increasing the speed above a certain threshold the number of the rotating stall cells switched from four to three. The newly developed stall cells were characterized by a pulsating pressure field – a rotating instability - with a source frequency of 350 Hz. Further analysis revealed that this source frequency had been most likely caused by an acoustic resonance in the ring chamber between impeller and vaned diffusor. The instrumentation used for this test campaign is shown in Figure 17. A telemetry was used to transmit the data from the rotating frame of reference to the data acquisition system. Selected experimental results from the strain gauges are shown in Figure 18 and Figure 19. The diagram axes are known from the Campbell-diagram (compare Figure 11), but showing here experimental results superimposed on predictions. An additional quantity is the amplitude of the measured impeller strain, providing the spectral composition for each speed as recorded during a gradual speed increase.

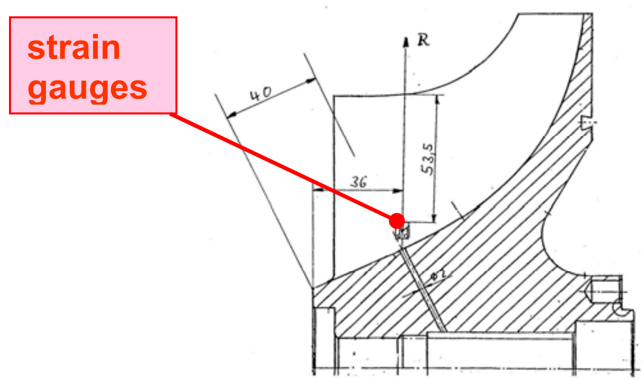

Figure 17: R&D test impeller instrumented with strain gauges for measurement of blade vibrations

For an IGV pre-swirl angle of 0 deg the vibration amplitudes were found to be smallest in the middle of the operating map, and highest close to the acceptable operation point limit. The latter case is depicted in Figure 18.

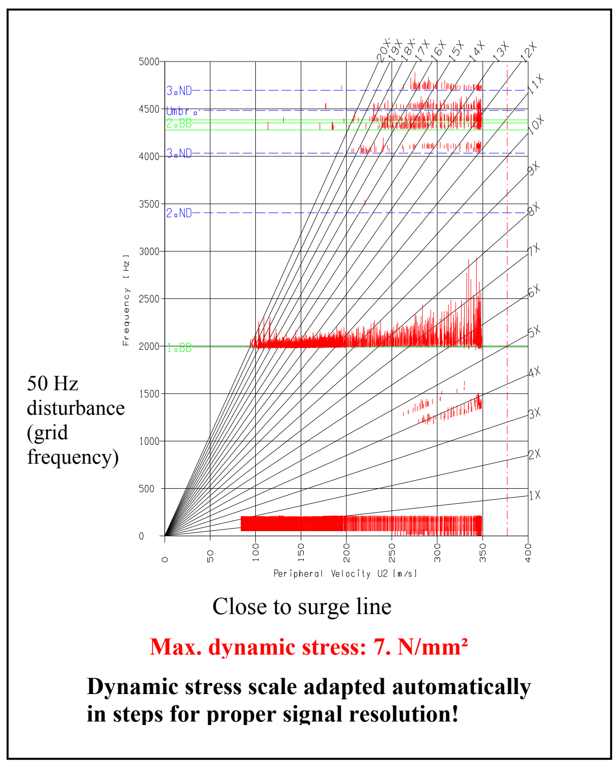

Figure 18: Measured dynamic loads in performance map, typical results for IGV:0°

Though elevated amplitudes were measured for this setup it was found that the maximum dynamic stresses were way too small to cause any damage to the impeller. The situation changed dramatically when the pre-swirl angle was set to 50 deg (Figure 19). For this IGV position the measured maximum dynamic stresses reached values sufficient to lead to impeller fatigue failures. It is assumed that the root cause for the high vibration amplitudes in the R&D study was the abovementioned rotating instability.

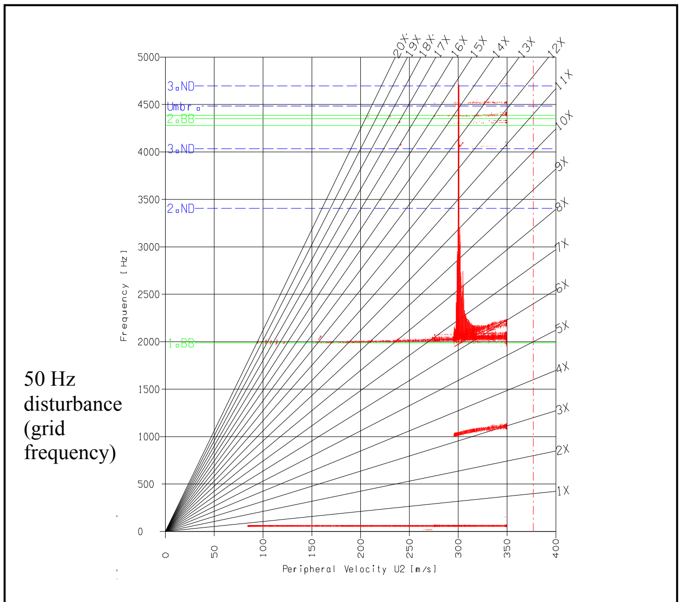

Figure 19: Measured dynamic loads in perfor-mance map, typical result at surge line for IGV:50°

Conclusions from R&D testing

From R&D testing two examples of flow instabilities were presented that may occur when a compressor is operated close to or even beyond its acceptable operation point limit. Both phenomena are characterized by different physical source mechanisms and may occur separately or in combination. If both instabilities occur in parallel it can be suspected hat they influence or even reinforce each other.
It is important to note that the deviations between the true surge line and the line that characterizes local flow instabilities increase for large positive pre-swirl. This led to the observation that high dynamic impeller stresses are more likely for positive pre-swirl if the compressor is operated to the left of the acceptable operation point limit.
With the awareness of the importance of these flow instabilities we can now step back to the RCA, namely to Figure 10, where sub-items of the topic ‘flow instabilities’ are listed. While a change of the gas mole weight is irrelevant for an air compressor, corrections for suction temperature shift and grid frequency fluctuations were considered adequately in the controller.
Therefore focus was on the verification of a positive margin between predicted acceptable operation point limit and the implemented control line and whether or not the unit was operated only right of the predicted acceptable operation point limit. For this verification operational data from site needs to be analyzed.

Analysis of operational data (trend data from site)

After such an important failure event as discussed in the current paper all available trending data has to be analyzed in order to find noticeable problems and a hint at the damage mechanism. When dealing with impeller damage the first emphasis is usually placed on the evaluation of shaft vibration data. The aim is to find out at which point in time a first deviation from the usual level in the vibration amplitude can be observed. This can give an indication under which operating conditions the failure mechanism did progress to that phase, where a loss or major deformation of impeller material becomes effective as unbalance, causing a change of the synchronous shaft vibration.

In the current case the usual shaft vibration level belonging to the corresponding radial bearing was about 7-8 µm. It was overshot at about 8:15 PM, more than two hours before the breakdown of the 3rd stage’s impeller.
It can be seen in Figure 20 that the vibration amplitude rose progressively before it jumped above the shutdown limit at 10:59 PM.

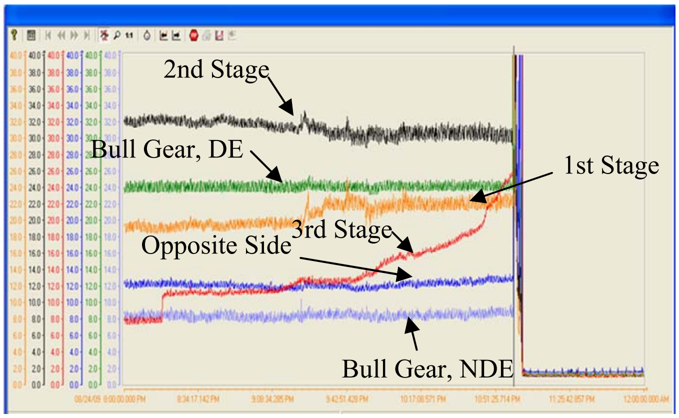

Figure 20: Trend data from site indicating the instant of impeller failure, shaft vibration amplitude

In comparison with the other trended values, no direct hint could be found which would explain the failure, especially no other noticeable signal fluctuations were observed at the same time when the first increase in the vibration signal was recorded. The fatigue fracture must have been caused by earlier or longer prevailing loads which did not necessarily persist during this period of progressive crack growth.
In that respect it must be noted that only the deep surge becomes manifest in a peculiar signature of trending signals. Other aerodynamic excitation mechanisms can not be detected

within the limitations of typical process instrumentation and recording cycle time. A systematic quest for surge incidents in the trending data was carried out. If a (deep) surge cycle occurs, the flow through the whole machine is reversed. The duration of this event depends on the system's stiffness and typically is in a range between 0.5s and 20s. The resulting drop in the differential pressure can be identified in the trending system of the process control system if its cycle time is below the duration of a surge event. Furthermore the characteristic signature of a surge cycle contains a significant drop of the compressor's discharge pressure, a delayed rise of suction temperature and vigorous response of the anti-surge controller. Even if the DCS (Distributed Control System's) trending is not capable to resolve the typical flow characteristics of a surge event, the response of the anti-surge controller will inevitably be recorded.
The available trend data was examined with respect to the characteristic indications discussed in the previous paragraph. A typical event which was found during this evaluation is displayed in Figure 21. The anti-surge control valve (ASV) opened relatively fast at 5:26 PM, but in case of a surge event the reaction would be much stronger. The anti – surge controller executes a comparison between the actual differential pressure at the flow measurement device and a setpoint. The difference between these values is amplified by a PI controller. If the flow decays to zero, the control deviation will become negative (< -40%, which means the minimum flow requirement is undercut by this amount) and this leads to an immediate opening of the valve by at least 40%, because the proportional gain of the controller had been adjusted to 1.0 for this specific application. The advanced OEM controller delivered for this unit limits the maximum closing rate, and even an immediate recovery of normal flow conditions could not restore the original valve position. Figure 21 exhibits a rapid increase of the ASV signal by only 15%. This value is too small to be attributed to a surge cycle. Presumably a fluctuation in line frequency or a sudden drop of the process demand caused this event, but the surge limit was not exceeded. The slight change in vibration amplitude simultaneous to the valve step (notice the scale) might be attributed to a shift of the load on stage 3. The analysis of all available trending data revealed no surge incident. This confirms that the control line as implemented with common practice prevents the unit from falling into deep surge.
Though deep surge has not been observed, Figure 21 shows another remarkable fact. The compressor is operating in turndown condition, on the verge of the control line or even on this limit. Such operation cases are quite common and not suspected to be harmful. However, it is an area of the operation envelope where potential local flow instabilities (e.g. stall phenomena) may appear without the entire machine falling into deep surge. Evaluation of the whole amount of available DCS charts supports the finding that the machine did run quite often in this operational area.

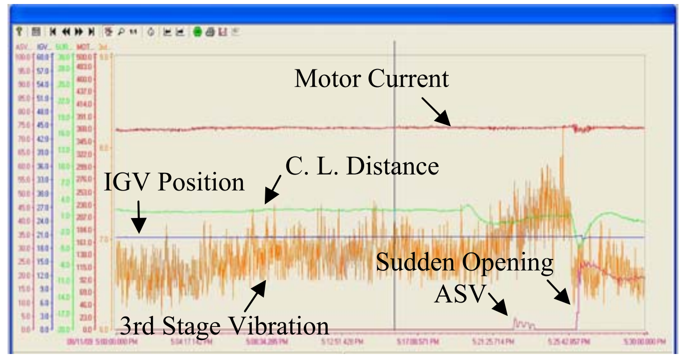

Figure 21: Trend data from site indicating operation near control line

Analysis of commissioning data

During the thermodynamic test run of this compressor at the OEM's facilities the single stage characteristics had been determined. Also the stability limit, often called the surge line of each stage is known. These individual limits are identified during shop testing by different criteria such as head rise, sub synchronous vibration and noise.
When single stages are combined to a complete multi-stage compressor (stage stacking procedure), the overall performance map is calculated through a computational procedure by which all individual stages are added up in consideration of the specified working conditions, defined by rotational speed, suction pressure and temperature, gas composition and recooling temperatures. The calculation program identifies the machine's overall surge limit when the first individual stage reaches its respective operation envelope. The performance map obtained by this procedure ('as built' documentation) characterizes the real-world compressor closely.
In the further course of the RCA it is essential to identify exactly in which part of the operation envelope the compressor had been operated. For this purpose it is helpful to display some of the identified operating points in the above mentioned performance map – but in practice this turns out to be difficult because one decisive information is missing: the exact delivery rate.
The Anti – Surge controller of course needs a flow measurement, but for gear type compressors in most cases the inlet cone of the 1st stage is used for flow metering so that no additional device is needed. Before entering the machine, the suction line's diameter is reduced by a tapered section. By this the gas flow is accelerated and according Bernoulli's equation it is possible to gauge a differential pressure:

$$\dot{V} = \alpha \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} \quad \Rightarrow \quad \dot{V} = XBL \cdot \sqrt{\frac{\Delta p \cdot T}{P_{abs}}}$$

Where $\dot{V}$ denotes the volumetric (effective) flow, A the cross – sectional area of the reduced diameter, Δp the differential pressure, ρ the density. T and P_{abs} stand for absolute temperature respectively pressure upstream of the contraction. Strictly speaking this law applies only to incompressible flow, but in certain limits it also is suited for gases; the precondition

is that the differential pressure is low in comparison to the absolute pressure. See EN ISO 5167 for reference.
The 2nd equation implicitly contains a definition of the orifice constant “XBL”. Under the assumption of ideal gas law it only depends on the gas composition. If we regard an air compressor, the volumetric flow can be calculated from the above mentioned three measurements and this constant. Unfortunately, the exact value of XBL can only be obtained if the contraction factor α is known. Literature provides numbers for standardized geometries such as Venturi tubes or standard nozzles. The inlet cone of a gear type compressor is not of such a kind and therefore it is not possible to determine XBL based on literature values. In order to estimate the pressure drop for transmitter rating approximations exist, but for an exact determination of the operating point within the performance map – which is crucial for the current RCA - those approximations are not accurate enough. It should be noted that for the purpose of anti-surge control the main requirement for flow measurement is repeatability and not predictability. In practical applications it is usually regarded as needless to determine the exact value of the proportionality constant as long as the actual surge limit is recorded and the control line is defined on the basis of differential pressure values (as result of surge tests).
In the current study the only feasible way to identify XBL was to superimpose the performance map from shop test with measurement data collected during the surge test on site.

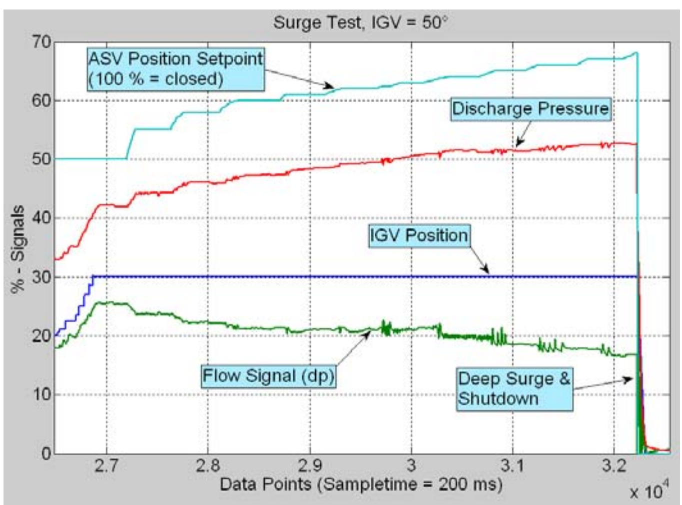

Figure 22: Commissioning data from site, surge tests at 50° IGV angle

During commissioning, decisive signals had been recorded with high sampling rate (a feature of the deployed controller) and on this basis the further analysis will be conducted.
Figure 22 displays the course of the most interesting raw signals during surge test at one fixed IGV position. This diagram is in the time domain, so we can see that the test has been conducted very carefully and ends with a deep surge cycle event. During the test, the ASV is closed stepwise (increasing signal). The response of the valve actuator causes slight signal fluctuations in the discharge pressure and flow signal, noticeable by the ripples on these curves which follow each increase of the ASV setpoint. In a careful post-failure analysis of these curves the increasing ripple of these curves could also be interpreted as indication of beginning flow instability, although deep surge with flow reversal did not yet take place.
The compressor performance curve and the surge limit depend on the IGV position and therefore surge tests were carried out on-site at different IGV settings. Detection of deep surge was used to determine the surge line.

Evaluation of commissioning data relative to shop performance results

As explained above, the volumetric flow can be represented by a formula that contains differential pressure, absolute pressure and temperature. In this case we have atmospheric suction conditions, so the influence of the ambient pressure is insignificant. By the choice of a suitable proportionality constant it was possible to approximate both performance maps: the theoretically combined shop test data and the experimentally gained curves from the on-site surge tests.
The result is reproduced in Figure 23.
Note that during the surge tests the IGV positions were not exact the same as in the calculated performance map. The main finding when interpreting this composed chart is that some of the evaluated surge points are located left of the predicted surge line. This particularly applies to the lines with positive pre-swirl, IGV close to minimum opening.
This finding does not oppose the stability limit found during single – stage testing. Obviously the stages can transgress their individual operation margin without giving cause for deep surge of the complete machine.

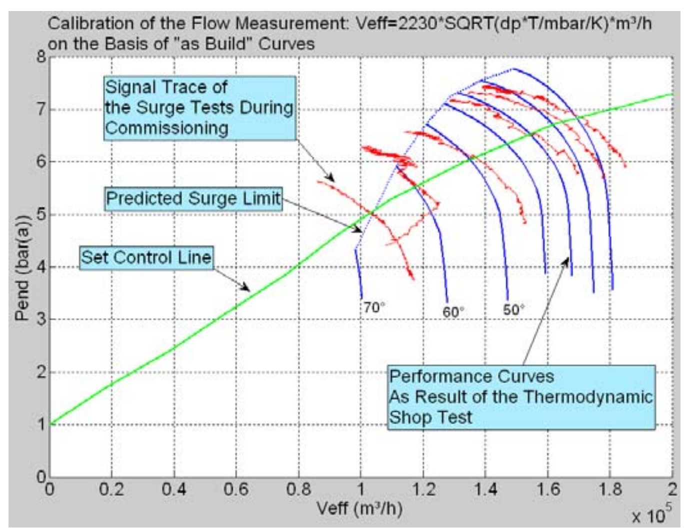

Figure 23: Comparison of test run results with commissioning data

Figure 23 also shows the set control line of the anti surge controller. At the lower edge this line cuts the stability limit according to shop test results and so the controller definitely does not enforce enough safety margin to protect the machine against local flow instabilities. As a result of operational data analysis we have already stated that turndown operation was quite common in the time before the failure happened. That means the machine had been run in exactly the area in which protection against local flow instabilities was not ensured.

RCA closure

With these insights from analysis of trend data and from review of the commissioning data we can eventually step back into the RCA, where we stopped at sub-topic 4.3 of Figure 9.
We know that the surge line was determined on-site according to common practices. In the post-failure analysis indications of flow instabilities could be detected already before the whole unit went into deep surge. By an excursion to R&D results it was shown that in particular in this region flow excitations may exist with the potential to generate high, destructive blade vibrations.
Evidence was found that the unit was indeed operated repeatedly in this zone. Since the comprehensive root cause analysis did not reveal any other deficiency, it can be deduced that this phenomenon is a possible explanation for the impeller damage.

Corrective action

With the above mentioned result of the root cause analysis the corrective action is quite straightforward. With the direct comparison of the predicted and the on-site measured performance curves as shown in Figure 23 a modified control line can easily be defined for implementation as provided in Figure 24. The steeper slope of the new control line reduces the turndown for lower compressor head, but gains some more turndown for higher head. With this corrected control line operation in the dangerous zone is no longer possible.
As for the impeller failure discussed in the current paper, local flow instability is assumed to be the root cause, so that any indication of local flow instability should be avoided for the failed stage of this compressor in the future. In mutual agreement between OEM and operator it was decided to install a set of thermocouples upstream of the failed impeller to indicate possible inducer stall (compare Figure 14). In addition to the correction of the control line this measure shall provide assurance that the risk of local flow instabilities – which could endanger future reliability - is minimized for this specific machine.

The added instrumentation is only used during test and implementation of the corrected control line.

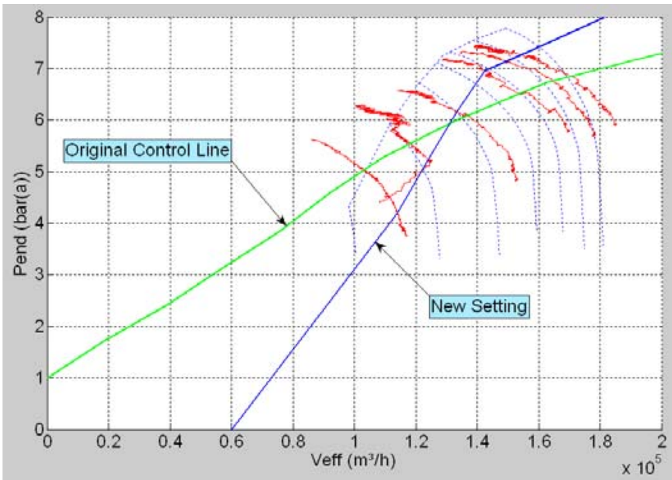

Figure 24: Proposed new control line

CONCLUSIONS

A blade failure was experienced on the 3rd stage semi-open centrifugal impeller of the main air compressor of an air separation unit after less than one year of operation.

The OEM of the compressor performed a comprehensive root cause analysis and concluded that operation of the 3rd stage in a region (turndown at high pre-swirl), where flow instability may occur, has caused excitation of blade vibration, which led to impeller fatigue.

A machining imperfection, which was found at the blade root, determines only the *location* of crack initiation, but is not considered as the root cause of the failure.

A proper correction of the control line will exclude re-occurrence of this issue.

The lesson learnt from this failure can be summarized as follows:
The common practice for on-site determination of the surge line had, in the past, proven to be completely sufficient for almost all machines. Since no calibrated flow signal is usually available during commissioning, observation of flow reversal (‘deep surge’) is taken for definition of the surge point. However, from R&D test results it is known that in particular at an individual stage of a multi-stage compressor local flow instabilities may occur without the whole machine falling into surge. Under unfavorable conditions such local flow instabilities may form a rotating instability, which has the potential to generate destructive blade vibrations. Frequency and intensity of the exciting forces onto the blades are not yet predictable. In fact we learned from our experiments, that these exciting forces (or harmonics thereof) also need to match the natural frequency of the impeller blade in order to cause blade vibration amplitudes, which are high enough in order to be destructive. The pure existence of inducer stall is not sufficient to cause impeller fatigue; if not also further specific conditions

prevail.
In mathematical terms the operation of a compressor stage in an unstable flow regime is seen as a necessity, but not sufficient to cause an aerodynamically induced fatigue failure.
A safe way to avoid such failures is the elimination of the necessary condition, i.e. avoidance of operation of a compressor stage in this unstable flow regime. This can be achieved by proper setting of the control line of the compressor. Further compressor instrumentation may provide supporting signals for detection of inducer stall, although this is not the first choice for commissioning of each and every new machine.
A software tool was developed in order to allow the commissioning engineer in future jobs to conduct directly an on-site comparison between predicted and tested compressor curves. This tool in combination with a careful evaluation of all available signals allows the commissioning engineer to implement a control line, which protects the unit not only from falling into surge, but also minimizes the risk of such very rarely occurring impeller failures as discussed in this paper.

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