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THE INFLUENCE OF THERMAL LOADING ON THE LEAK TIGHTNESS BEHAVIOUR OF HORIZONTALLY SPLIT CENTRIFUGAL COMPRESSORS

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ABSTRACT

For the purpose of maintenance requirements or for some applications in the chemical industry (for instance chlorine) the centrifugal compressor must be designed with a horizontally (axially) split casing.

Beyond the decision about the type of manufacturing (cast or welded) or about the material selection one of the utmost issues regarding the design of the compressor lies in the leak tightness of the flanges. The design of the compressor casing needs detailed checking for tightness. In order to ensure a proper design with respect to integrity of stress and tightness under test and operating conditions several FE Analyses and resulting criteria have been developed by the OEMs. Furthermore in order to demonstrate the leak tightness the casing is subjected to a hydrostatic test prior to the assembly of the inner parts and rotor. According the API specification a hydrostatic pressure of at least 1.5 times maximum design pressure is applied. This standard procedure is usually considered to be sufficient for demonstrating the casing integrity, the tightness in operation and to check the accuracy of the FE analysis.

However some applications require the use of several sections (consisting of some stages) inside one casing. According to the stage configuration hot and cold casing sections might be close to each other. In operation the compressor casing is subjected not only to pressure but also to thermal loading. These potential high temperature gradients can considerably influence the compressor behaviour regarding its tightness. The conventional hydrostatic test can even be less critical than at some particular operating conditions of the compressor on site.

This paper describes some experiences of the author’s company with this type of compressors and the performed calculations. Different configurations are analysed and compared between each others. The paper shows the steps for the optimization of casings in order to develop an appropriate split line flange design supported by FE calculations. Some examples of different casing concepts are shown and discussed. At last the paper highlights some decisive issues which influence the tightness of compressor like:

- Design of inner casing
- Arrangement of sections
- Geometry of flange
- Design and arrangement of bolts.

INTRODUCTION

The casing of centrifugal compressors is designed according two different types: horizontally split (also called axially split) and vertical (“barrel type”). Generally a horizontally split casing is chosen for low pressure application (i.e. for pressure below 70 bar or 1,015 psi, see Figure 1).

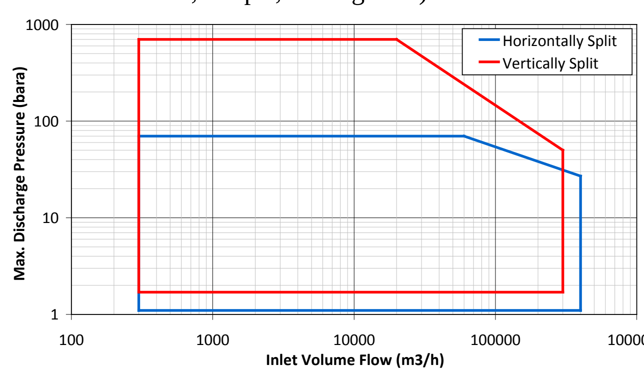


Figure 1. Operating Range of Horizontally and Vertically Split Casing Centrifugal Compressors (after Lüdtke, 2004).

The most common criterion for the evaluation of horizontally split casing vs. barrel type is the accessibility of compressor internals and rotor for the purpose of maintenance. A basic requirement put by maintenance people is to have a horizontally split casing in the middle of the train arrangements comprising of driver and two compressors (for instance for LNG Refrigeration application, see Salisbury, et al., 2011 and Meher-Homji, et al., 2011). In this example easy access to internals and rotor is given after loosening the split line bolts and removing the upper casing half. This advantage of horizontally split casings is improved if the nozzles are

downwards and no loosening of flanged connections to process piping is required.
Hence the accessibility for maintenance on site is ensured. Another example for maintenance requirements are the compressors for chlorine application in the chemical industry. Customers often insist to have a horizontally split casing due to light corrosion of surface after a service period without maintenance for five years or more. After this time it was experienced that bundles of barrel type compressors for chlorine application were only removable with increased effort. This problem can be avoided by using horizontally split casings. For these reasons horizontally split casing compressors are used in several different applications like natural-gas, refrigeration (i.e. propane), chlorine, methanol (synthesis) or LNG. Usually horizontally split casings have been designed for large centrifugal compressors with impeller diameters up to 1600 mm. Numerous examples for these applications can be found in Lüdke (2004). The mol weight of the process gas is a further criteria for the decision whether a horizontally or vertically split casing is the right choice. Light mol weights require vertically split casings due to required tightness of the casing which can preferably be realized by means of vertically split design. Another criterion from viewpoint of hydrogen application has to be noted from API 617, 2.3.1.3 of Chapter 2, 2002: "*Unless otherwise specified, casings shall be radially split when the partial pressure of hydrogen exceeds 1380 kPa gauge*." As a rough rule it can be stated that horizontally split casings are used for heavy gases.
In order to ensure the reliability of centrifugal compressors thorough analyses are performed by the OEMs with respect to the thermodynamic, lateral, torsional, stability, etc... Hence any already experienced cause of centrifugal compressor trouble (such as surge, shaft misalignment, vibration, etc...) is avoided. Additionally to these analyses the design of a horizontally split casing compressor requires detailed checking for tightness in the split line flange. In order to minimize the costs of production and maintenance the number of compressor casings for a specified application is reduced. Moreover to improve the efficiency the process gas is sometimes cooled during the compression process. Therefore some compressors are designed with several sections (among other with side stream, or in back-to-back arrangement, see Meher-Homji, et al., 2007). This configuration represents a challenge not only for the thermodynamics, rotordynamics and rotor thrust. Additionally for horizontally split compressor casing an arrangement with many sections requires special attention regarding the casing design, especially the flange and the casing tightness.

FLANGE DESIGN

General considerations

The elastic casing deformation and especially the tightness of the split line flanges under pressure is the most important criterion on tightness which is influenced by several parameters (casing size, design of flange, pressure level, number of nozzles, etc..) . Appropriate stiffness of split line flanges and sizing and pretensioning of bolts as well as the general casing design shall be considered in order to meet all requirements. In order to ensure the leakage tightness of the compressor in operation on site, the casing is subjected to a hydrostatic test with an inner pressure according to API 617, 4.3.2.1 of Chapter 1: "... *minimum of 1.5 times the maximum allowable working pressure*". Usually the design of the flange is investigated with respect to this hydrostatic test pressure. Until the middle of 90's the flange was designed with the help of analytical models (according Traupel, 2000 or VDI 2230, 2001). These models have been developed and validated since the 50's (Reuter, 1958 and Ullmann, 1965).
According to API 617, 2.3.1.5 of Chapter 2: "*Axially split casings shall use a metal-to-metal joint that is tightly maintained by suitable bolting*". Thanks to the pretension of the bolts a contact pressure at the joint area is created. The contact pressure area is limited by a 45 degree cone (Figure 2).

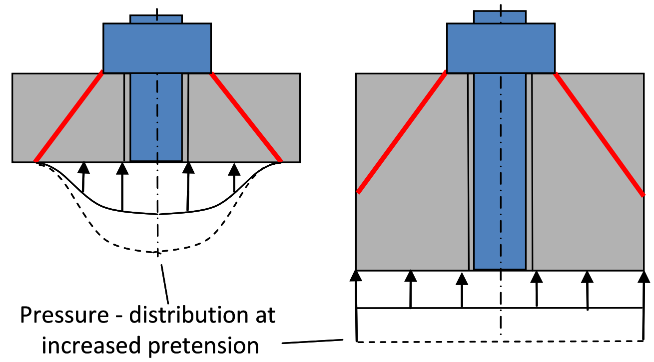


Figure 2. Contact Pressure Distribution at flange.

The contact pressure decreases like a "bell-curve" toward the cone edge. By increasing the pretension the contact pressure in the middle of the flange increases, but no difference is observed at the rim of the cone. Only with increased flange thickness (for the same flange width) the contact pressure becomes uniform and at the end of the cone a significant increase of the contact pressure occurs (see picture on the right). In the limit case the contact pressure for a sufficient thickness of the flange is uniform along the width. In this case the pretension of the bolt would be optimum for the leak tightness. It has to be noted that the optimum contact pressure distribution is reached when the bolts are inserted through the entire symmetrical flanges.
In this analytical study the flange is designed under the following assumptions:

- The loads from the inner pressure are carried by the bolts along the length of the casing.
- The bolts on the face side have the same dimensions as those at the length side.

The analysis is performed on the basis of a beam model theory (VDI 2230, 2001). The bore of the bolt is neglected and the relaxation of the bolt is assumed to be much higher than this of the flange.
Hence the minimum required pretension force of the bolts is determined in dependence on the geometry of the flange and the location of the bolt on the flange width according to Equation (1):

$$F_V = \frac{p \cdot t}{1 + 6\,s/c} \cdot \left[\left(1 + 6\,a/c\right) \cdot \frac{D_i}{2} + k \cdot c \right] \qquad (1)$$

It has to be noted that the distance "s" between the bolt bore and the center line of the flange shall be located on the side of the inner rim of the flange. Otherwise "s" becomes negative and higher pretension and bigger bolts are required.
The factor "k" describes the minimum required contact pressure at the inner edge of the flange. For k=1 the contact pressure is equal to the inner pressure inside the casing, hence no resistance to leakage is ensured. Furthermore the surface of the flanges shall be assumed to be "not perfect" (at least micro geometric non-plane). This is the reason why the contact pressure at the inner edge of the flange shall be higher than the inner pressure as shown in Equation (2):

$$p_F = -k \cdot p \tag{2}$$

This k-factor is not a safety factor but rather a physical-technological parameter.

FE Calculations

Even if this analytical analysis allows for a design of the flange with a reasonable accuracy it appears that the opening of the flange is underestimated. This is the consequence of some assumptions as a constant contact pressure along the flange width. Therefore a FE analysis of the contact flange is indispensable. Due to the improved hard- and software this calculation is nowadays a rather "straight forward" issue.
A 2D model is created based on geometrical parameters. The bolt is modeled with beam elements and the actual profile of the bolt is considered. The flanges are combined with contact elements. Figure 3 shows a typical meshed model of such a 2D FE analysis with the applied boundary conditions.

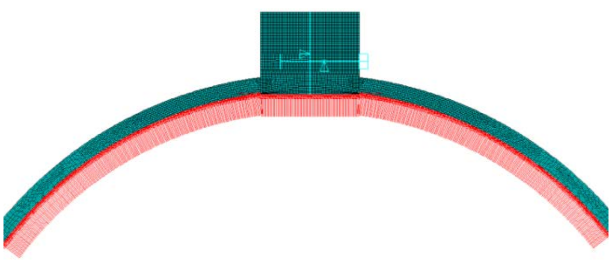


Figure 3. Meshed FE –model (2D) with applied boundary conditions.

The results (Figure 4) show the contact pressure distribution along the flange width as well as the opening (gap) in the flange.

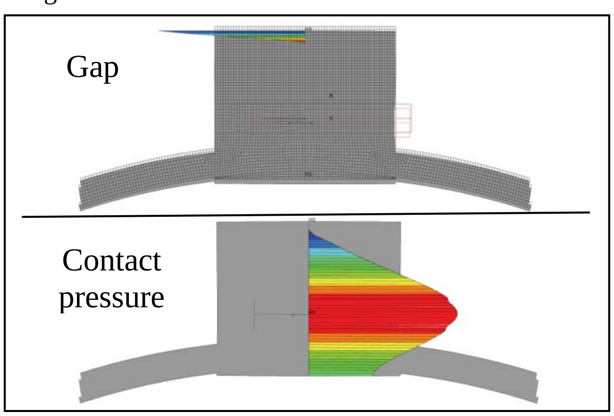


Figure 4. FE Results (gap and contact pressure).

In this example it can clearly be seen that a resulting contact pressure is present along the flange width with exception of the last 5 percent at the outside where a gap occurs. Hence the contact pressure is quantified and further evaluations are carried out with respect to the OEM's internal criteria. Thanks to this simplified procedure the design of the flange can be easily and quickly determined. Nevertheless for more complex structures this procedure may be too simplified. Moreover the 2D model considers perfectly symmetrical flange geometry; any influence from the nozzles is neglected. At last the front side and areas between front and length sides are not analyzed. Thus for some applications (for instance for a new flange design) a 3D analysis of the complete casing is required. The CAD model is imported directly into the FE-software. In comparison to the previous 2D analysis this calculation allows for evaluating the equivalent stress in the casing and the stresses at the bolts, too.
Figure 5 shows such a 3D meshed model. The investigated casing of this compressor consists of two welded casing halves with one-piece frontal covers on both face sides. The meshing at the bolts and the flange plane is refined (Figure 5, right).
The bolts are pretensioned and the inner pressure corresponding to the hydrostatic test pressure is applied.

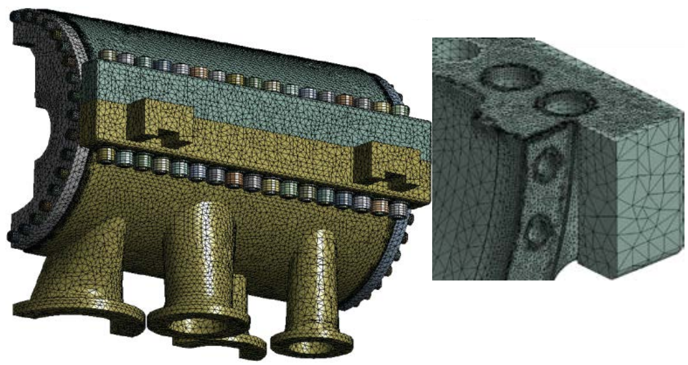


Figure 5. Meshed FE-Model (3D).

The results (Figure 6) represent the contact pressure distribution in the horizontal flange face. The scaling is chosen so that a contact pressure below 0 MPa (gap present) is illustrated in gray, whereas values below 1 time the inner pressure (k < 1) are in deep blue. It can clearly be seen that the overall contact pressure is high, demonstrating the leak tightness of the casing during the hydrostatic test.

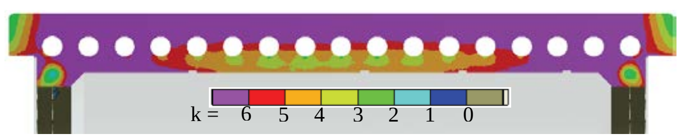


Figure 6. Contact pressure between flange faces (hydrostatic test).

VALIDATION OF FE-MODEL

In order to validate the above-mentioned FE-model several measurements were performed during the hydrostatic test of the casing. Three different topics were subjected to deeper investigations:

- Contact pressure after pretension of the bolts
- Deformation of casing during the hydrostatic test
- Minimum required pretension of bolt for leak tightness of flange.

The contact pressure distribution between the flange faces was measured with pressure indicating films. This measurement is principally qualitative. Red patches appear on the film if a contact pressure is present and the exhibited color density varies according to the variing contact pressure levels. The aim of this measurement is to compare the results of the FE analysis with the measured pressure distribution. Figure 7 shows the arrangement of the test with the opened casing and the films installed on the flange face.

Figure 7. Opened casing with pressure films.

Since the pressure indicating films show the pressure distribution in different red tones, the results of the FE analysis are represented in the corresponding color scheme with the same scale. From the comparison between the measurements and the results of the FE Analysis (Figure 8) it can be stated that the FEA delivers very satisfying results. The pressure distribution along the entire surface flange is well predicted. Especially the expected weakness at the end of the flange (denoted with a blue circle in the figure) is clearly visible at the measurement too.

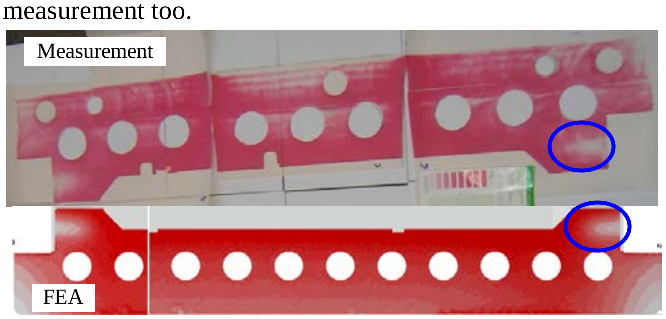


Figure 8. Pressure distribution after pretension (FEA and Film).

Thereafter the casing was assembled and the bolts (with size M72) were pre-stretched with a value of 1.5 per mil. The casing was then subjected to a hydraulic leakage test with an inner pressure according to API 617. As the design pressure of the compressor is 25 bara (363 psi), the hydrostatic pressure test was 1.5 · 24 = 36 barg (522 psig).

During the test the deformation of the casing was measured by means of several dial gauges arranged on different sides of the casing. The results of these measured values were compared to the values from the FE calculations. The differences were low, hence validating the FEA.

The determination of the necessary minimum bolt pretension to ensure the leak-tightness of the casing on the base of the FE analysis results is difficult. There are several differences between the built casing and the “ideal” FE-model (i.e. manufacturing tolerances, simplification of the FE-model) which affect the contact pressure in the flange. Due to these facts the necessary minimum bolt pretension was measured to validate the FE analyses. This minimum required bolt pretension was determined by gradually loosening the bolts in the flanges in steps of 0.1 per mil, from the original value of 1.5 per mil (obtained with a hydraulic pretensioning device). At last the location of leakage could be detected on the basis of the water leaking out of the casing.

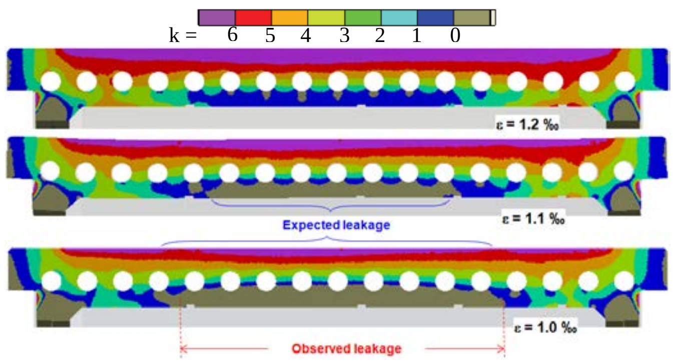


Figure 9. Expected and observed leakage at reduced pretension bolt.

During the test the casing remained tight until a bolt pretension of 1.0 per mil. Only at this low value some droplets were observed in the middle of the length flange. No trace of leakage could be observed on the cover sides of the casing. A comparison to the calculation (Figure 9) shows that the leakage was expected already at 1.2 - 1.1 per mil (contact pressure lower than 1x inner pressure). Obviously the FE analysis delivers satisfying values, though slightly conservative. The area of expected leakage matched with the measurements. Hence it could be stated that the FE model was well validated.

INFLUENCE OF THE TEMPERATURE

CASE A

General Arrangement

For a gas gathering application the authors’ company recently delivered two different skids (named “HP” and “Blower”) with centrifugal compressors driven by synchronous motors. The “HP” train consists of a synchronous motor driving, via a variable speed planetary gear, two centrifugal compressors (one horizontally split casing named “RH” one of barrel type or “RB”). The RH-compressor is the one whose casing was previously calculated and measured in the previous chapter of this paper. Figure 10 shows the described train arrangement.

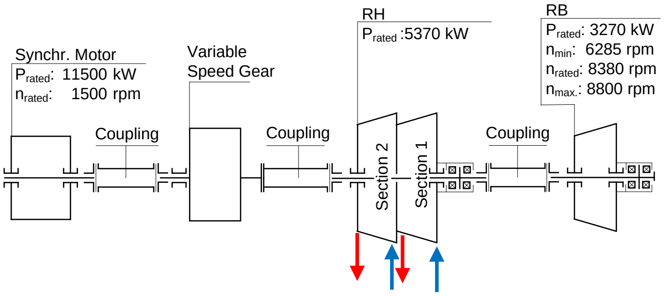


Figure 10. Schematic of the HP-Train Arrangement.

At relatively high mol weight (26 g/mol) the increase of the gas temperature inside the compressor for the specified pressure ratio (π= 6.8) is high. Without process cooling during the compression process the discharge temperature would reach an unacceptable level for the reliability of the compressor components like O-rings. Therefore two sections are needed and the gas has to be cooled between both sections in order to keep the temperature inside the compressor in an acceptable level. Table 1 summarizes the main operating conditions of the compressor for site operation. As shown in Figure 10 the arrangement of the stages is in-line, i.e. the gas enters one end of the compressor and is discharged part way through the casing (Section 1). The gas is then cooled and reintroduced in the middle of the casing near the discharge nozzle of Section 1. The hot gas exits at the discharge nozzle of Section 2 at the other end of the compressor.

Feature	Unit	Operating conditions	
		Section 1	Section 2
Suction pressure	bara (psi)	2.4 (35)	6.0 (87)
Discharge pressure	bara (psi)	6.5 (94)	16.2 (235)
Suction Temperature	°C (°F)	55 (131)	60 (140)
Discharge Temperature	°C (°F)	137 (279)	145 (293)
Rotor speed (100%)	rpm	8,367	8,367
Mass Flow	kg/h	63,550	61,260
Gas	g/mol	C_nH_m 25.8	C_nH_m 26.1

Table 1. Compressor Operating Conditions (Case A).

Findings on Site, Investigations

The compressor was delivered on site after successful mechanical and performance test at the author's testing facility. During the operation a leakage was observed at the horizontal flange (Figure 11).

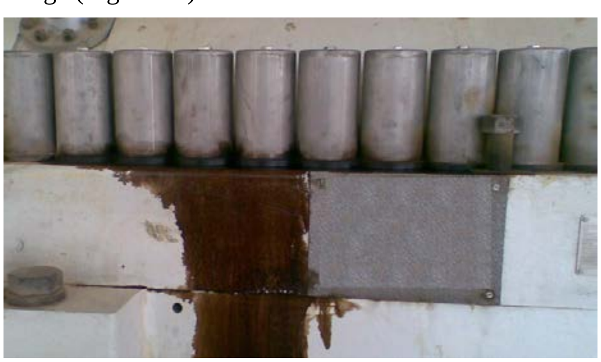


Figure 11. Observed Leakage during operation on site.

A Root Cause Analysis was carried out, investigating all the possible reasons leading to such a clear leakage. Although the hydrostatic test was successful and validated the calculations the FE analysis was checked and the calculations were re-performed taking into account more boundary condition like the piping forces, internal forces inside the machine and the temperature distribution in the casing. It appeared soon that the temperature had a large influence on the leakage tightness behavior of the casing. As can be seen in Figure 12 the temperature gradient in the flange is very large thus leading to large deformations of the casing. Although the hydrostatic tests had proven a very high safety regarding the leakage (actual bolt pretensioning 1.5 per mil, minimal required pretension 1.0 per mil), the behavior is totally different with consideration of the temperature.

Figure 12. Temperature distribution at flange in operation.

The temperature at the surface of the casing was also measured along the rotor axis at different circumferential locations and compared to the results of the FE Analysis (Figure 13). As the ambient temperature was lower than the specified value, the overall gas temperature in the compressor is lower, too. Nevertheless the temperature distribution along the axial length corresponds to the expected one. Especially the large temperature gradient through the sections is clearly recognizable. It has to be noted that this temperature behavior is not strictly homogenous along the circumference. This is mainly due to the arrangement of the nozzles in the casing which influence the temperature distribution.

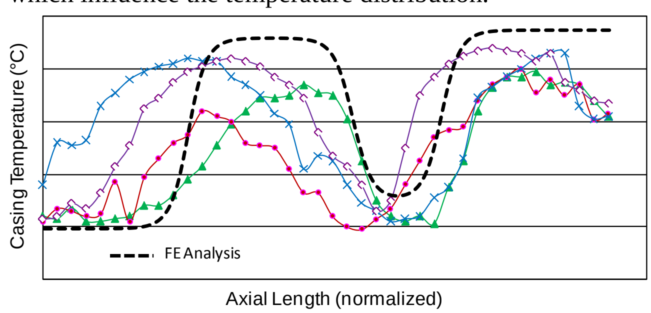


Figure 13. Measured Temperature on casing during Operation.

A review of the FE Analysis taking into account not only the pressure but also the temperature under actual conditions was carried out. Figure 14 shows the gap distribution in the horizontal flange. The scale is chosen such that the absence of gap is represented in red. The results clearly demonstrate the presence of leakage in the region of the suction nozzle of section 2. Hence the calculation could confirm the findings on site.

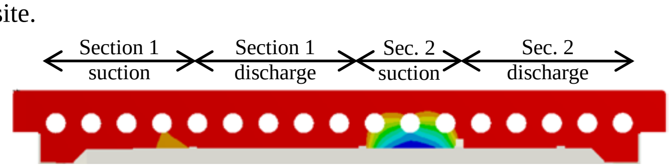


Figure 14. Calculated gap at flange under operating conditions.

CASE B

General Arrangement

For a refinery application a horizontally split compressor was supplied in a coker plant. The compressor consists of two sections with seven stages and operating conditions listed in Table 2. The arrangement of the sections and the operating conditions are similar to Case A. The compressor is driven by an asynchronous motor via a variable speed planetary gear. The casing is welded with main and intermediate nozzles downwards. In comparison to the casing presented as Case A the volute here are integrated in the casing by means of welded metal plates (Figure 15). Hence the temperature gradient between the sections is expected to be lower than the experienced value in Case A. Moreover the size of the compressor is bigger.

Feature	Unit	Operating conditions	
		Section 1	Section 2
Suction pressure	bara (psi)	1.1 (16)	3.2 (46)
Discharge pressure	bara (psi)	3.9 (57)	13.1 (190)
Suction Temperature	°C (°F)	49 (120)	51 (124)
Discharge Temperature	°C (°F)	121 (250)	137 (279)
Rotor speed (100%)	rpm	7,120	7,120
Mass Flow	kg/h	85,430	72,750
Gas	g/mol	C_nH_m 36.5	C_nH_m 33.3

Table 2. Compressor Operating Conditions (Case B).

FE Analysis

Similar to Case A a FE analysis was performed taking into account the pressure and the temperature inside the casing. The temperature distribution was applied according to Figure 15.

Figure 15. Temperature distribution in operation (steady state).

The size of the bolts at the horizontal flange is M72 with a pretension of 2.0 per mil. The forces induced by the internal part as well as the nozzle forces are superimposed.
The results shown in Figure 16 represent the gap distribution in the horizontal flange. Here, once again, the presence of a gap in the flange located at the suction of section 2 is obvious.

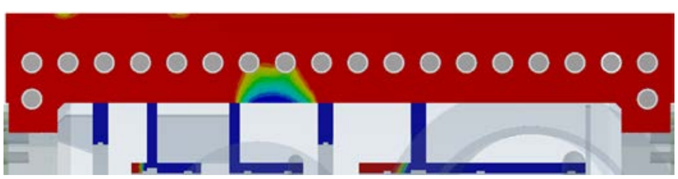


Figure 16. Gap distribution in the horizontal flange.

COUNTERMEASURES

To solve the problem different countermeasures are possible. Some measures as applying a sealant in the split line (thin layer) or machining a groove (internal, vented) to recycle the leakages are rather emergency or temporary solutions and are hence not further considered here. The first countermeasure, actually the most simple, consists in increasing the pretensioning of the bolts. However the bolts cannot be endlessly pre-stretched as the elasticity limit of the bolt material in operation (i.e. at actual gas temperature) has to be considered.

CASE A

In this case the problem was solved by replacing the bolts located around the suction nozzle of section 2 by inconel bolts of same size with increased yield strength, allowing a much higher pre-stretch than the original value. Table 3 summarizes the minimum required pretension in operation, considering the gas pressure alone, and together pressure and temperature as well as for the hydrostatic test (increased pressure). What strikes the eyes is that the conditions in operation are by far much severe than those during the hydrostatic test. To reach an equivalent loading in operation (with consideration of thermal loading) a pressure test of 100 bar (1,450 psi) would be required! For structural reason this is evidently not possible.

Case		Applied Conditions		Min. Required Pretension	
		Pressure bara (psi)	Temperature °C (°F)	Bolt Force (N)	Force Ratio %
Hydrostatic Test		37 (537)	-	8.06E+05	100
In Operation	p - only	2.4 - 16.0 (35 - 232)	-	2.93E+05	36
	p + T	2.4 - 16.0 (35 - 232)	55 - 145 (131 - 293)	2.20E+06	273

Table 3. Minimum required bolt pretension for different conditions.

CASE B

For this case two different alternatives are investigated. The basis for these studies is a new design of the original casing with the aim of a considerably reduced temperature gradient along the inner rim of the flange.

Back-to-Back Arrangement

The first proposal consists in permuting the arrangement of the sections. Instead of the originally in – line arrangement (Section 1 suction / Section 1 – discharge / Section 2 suction / Section 2 discharge), a back-to-back arrangement (Section 1 suction / Section 1 – discharge / Section 2 discharge / Section 2 section) is designed. Hence both warm sections are close to each other and the thermal loading is reduced.

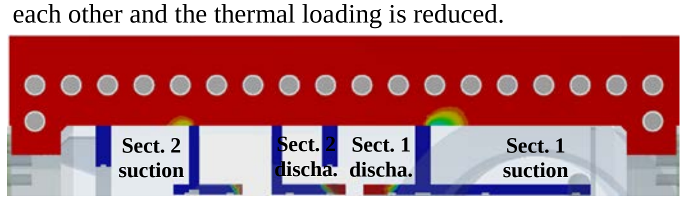


Figure 17. Gap distribution in the horizontal flange (back-to-back).

The resulting gap caused by the thermal and pressure loading in operation is shown in Figure 17. It turns out that the previously observed gap now completely vanished. There is no leakage expected. However a new, small gap at the inner rim of the flange appears in the vicinity of the suction nozzle of the first section. This is due to the large deformation in the middle of the casing where the higher discharge temperatures of the gas are concentrated. From this study it can be concluded that the back-to-back arrangement improve the tightness behavior of the casing with respect to the thermal loading. Nevertheless this configuration has still some room for optimization.

Optimized Configuration

In order to get rid of the thermal loading inferred by the large temperature gradient it was decided to add a partial ring at the suction of section 2 as shown in Figure 18. The base of this study is the original in-line arrangement. The aim of this additional ring is to separate the cooled flow coming from the suction nozzle of section 2 to the flange. Thanks to holes on the left ring warm flow from the discharge of section 1 comes inside the narrow room. Hence the flange is only in contact with warm gas.

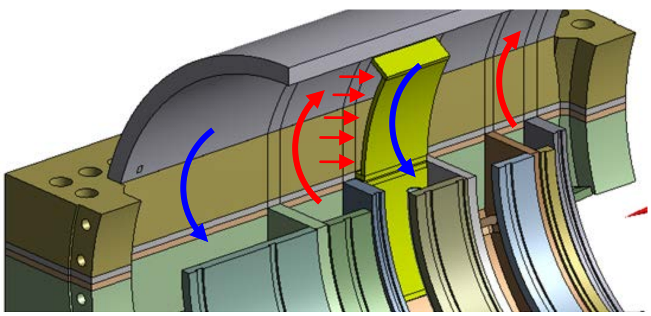


Figure 18. Scheme of implemented additional ring.

The steady state temperature distribution is shown in Figure 19. It can clearly be seen that the intermediate ring prevent the cooled gas from suction of section 2 to come into contact with the flange. Hence the flange in the middle of the casing is only in contact with increased temperature, reducing the thermal loading on bolts and flange.

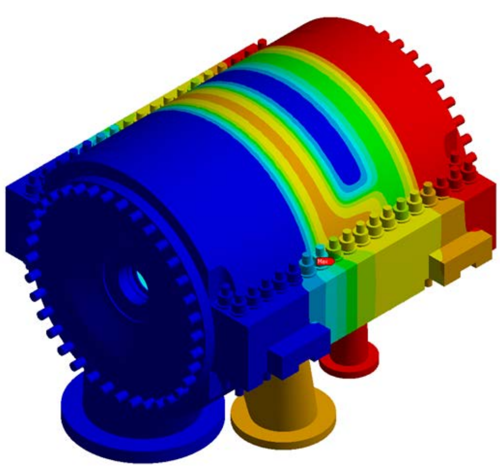


Figure 19. Temperature distribution with implemented additional ring.

Consequently the gap distribution, represented in Figure 20, clearly shows that no gap is present in the flange anymore. This is exactly the behavior which is looked for using the intermediate ring which acts as a buffer zone between the flange and the cooled flow of section 2.

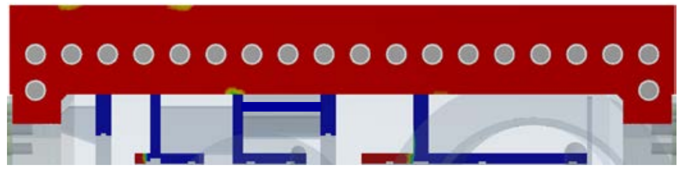


Figure 20. Gap distribution in the horizontal flange (additional ring).

CASE C

General Arrangement

For a Nitric Acid plant the author's company supplied two identical centrifugal compressors (with slightly different conditions) for a full refurbishment of the existing compressor trains. Both trains consist of one centrifugal compressor with integrated expander driven by an asynchronous motor via a gear and flexible and solid couplings. After nearly forty years of operation the two NOx compressor trains were in rather bad condition and proper maintenance was increasingly difficult and time consuming. In view of this situation it was decided to recondition the units such that up to twenty years of further operation could be achieved. Moreover both compressors shall be equipped with some other components. To reduce downtime and to avoid any problem it was specified to procure new casings of new geometry. All compressor stator parts were replaced. The flow channels of the new internals are made to the same geometry as the existing ones to prevent any changes in the aerodynamic or thermodynamic behavior of the compressors. In combination with the new compressor casings the stator guides, the inlet walls and the return channels were supplied machined to final dimensions. The design of the rotor was not changed. Only the riveted impellers were replaced with welded impellers. Figure 22 shows a cross section of the original compressor.

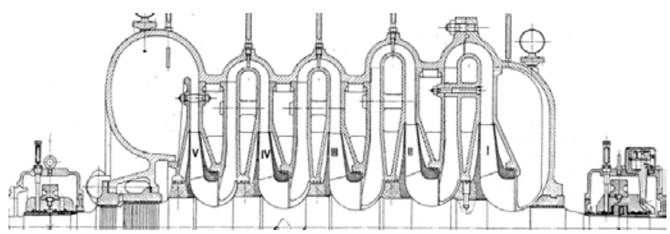


Figure 22. Cross Section of original NOx-compressor (Courtesy of MAN Diesel&Turbo Schweiz AG)

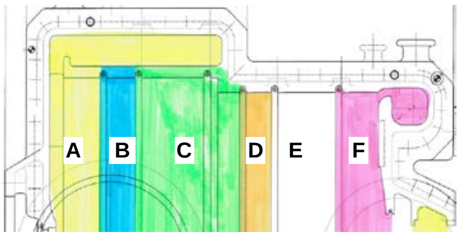
Due to the relatively high volume flow the size of the compressor is rather large, with large suction and discharge volutes. The configuration of the compressor is in-line (no intermediate nozzle). However the heating of the gas inside the casing is rather large (from 35°C up to 192°C (95°F – 378°F) for compressor 1, 56°C to 227°C (133°F – 441°F) for compressor 2). Table 4 summarizes the main characteristics of the compressors. Due to the specified restricted area around the compressor the dimensions of the new casing were limited.

Feature	Unit	Operating conditions	
		Compressor 1	Compressor 2
Suction pressure	bara (psi)	0.92 (13)	0.92 (13)
Discharge pressure	bara (psi)	3.15 (46)	3.14 (46)
Suction Temperature	°C (°F)	35 (95)	56 (133)
Discharge Temperature	°C (°F)	192 (378)	227 (441)
Rotor speed (100%)	rpm	5,395	5,395
Mass Flow	kg/h	52,733	45,342
Gas		NOx	NOx
	g/mol	28.7	29.0

Table 4. Compressor Operating Conditions (Case C).

FE Calculations

The casing and the flanges were first designed according to the results of the 2D-FE analysis. After modelling in 3D, the model was imported in FE-software and calculated according the boundary conditions shown in Figure 23.



Area	p (bara)	T (°C)
A	0.9	37
B	1.3	77
C	1.8	121
D	2.3	157
E	3.0	193
F	3.7	227

Figure 23. Applied pressures and temperatures on draft casing.

The calculation was performed in three steps. In the first step the bolts are pre-stretched, and then the pressure applied. In the last step the temperature distribution is applied in the casing. The size of the bolts is of M30 with a pretension of 1.8 per mil (corresponding to a force of 160,790 N). The results regarding the leak tightness are represented in Figure 24. On the left side the gap between the half casings is shown during the hydrostatic test. The picture on the right shows the gap resulting from the pressure and temperature in operation applied together.

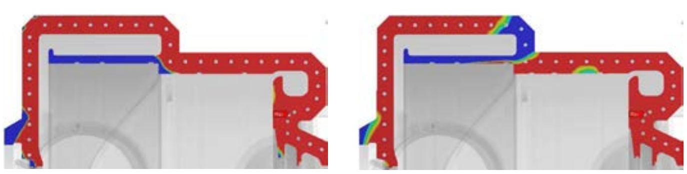
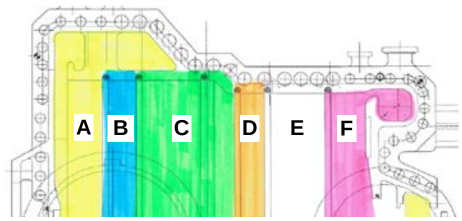


Figure 24. Gap at flange for draft casing with applied test pressure (left) and pressure + temperature in operation (right).

The results from the calculation with test pressure (left) indicate clearly that the flange is leak proof during the hydrostatic test. However under the considerations of pressure and temperature in operation the flange is not tight any more. This is mainly due to the large temperature gradient between the zones A and D. Therefore it can be stated that this configuration is not acceptable, the design of the casing must be considerably improved with respect to the temperature influence.

Countermeasures

Many attempts were carried out based on this draft casing geometry. Especially the design of the bolt was changed (larger bolts) and the thickness of the flange was modified. Also the span between the bolts was reduced to a minimum and their pretension was increased. Unfortunately none of these modifications could lead to any satisfying result.
Thus the general design of the casing had to be changed.
For this new casing geometry special attention was placed on the temperature distribution along the flange in order to achieve the lowest possible gradient.
The results of this investigation led to the geometry represented in Figure 25 with the applied pressures and temperatures.



Area	p (bara)	T (°C)
A	0.9	37
B	1.3	77
C	1.8	121
D	2.3	157
E	3.0	193
F	3.7	227

Figure 25. Applied pressures and temperatures on optimized casing.

The temperature gradient is now clearly much “smoother” than for the original configuration. Especially the zone “C” acts as a buffer between zones “A” and “D”. Hence the deformations at the flange induced by the thermal loading could be considerably reduced. Additionally the size of the bolts was increased as well (M48 and M56), leading to a much higher pretensioning force than original (final: 356,190 N and 490,371 N). As shown in Figure 26 no leakage is expected any more not only with respect to the pressure but also under consideration of the temperature.

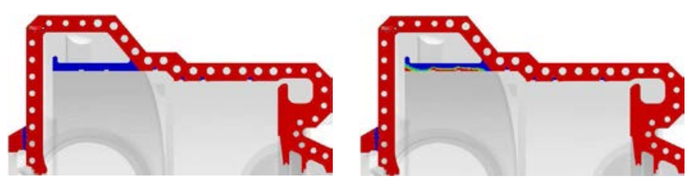


Figure 26. Gap at flange for optimized casing with applied test pressure (left) and pressure + temperature in operation (right).

CONCLUSION

The casing design and its configuration for a horizontally split centrifugal compressor is essential to ensure the tightness of the machine in operation. Especially for compressor subjected to large temperature gradients (i.e. with intermediate flow, as shown in Cases A and B) or compressors with casings without constant outer diameter (Case C) the thermal loading in operation can not be neglected anymore. Any abandonment of the temperature influences in the casing design can lead to inconvenient issues in operation on site. The thermal loading is actually not handled in the API 617 requirement which considers the working pressure, only.

Even an increase of the pressure during the hydrostatic test can not reflect the conditions on site and ensure the leak tightness of the compressor.

To determine the influence of the thermal loading on the casing standardized analytical calculations and even parametrized 2D-FE simulations are not sufficient. The simulation of thermal loading requires a 3D FE analysis.

Several calculations have to be performed prior to the casing manufacturing and even for its final design in order to take countermeasures at the right time and adapt the geometry of the casing according to the boundary conditions. A simple countermeasure consisting in increasing the pretensioning of the bolts may be not enough.

With comprehensive FE analyses an optimization of the geometry of the flange and of the casing is obtained, ensuring the reliability of the compressor regarding its tightness on site.

NOMENCLATURE

F_v	= Pretension Force of Bolt	[N]
S	= Safety Factor	[-]
p	= Inner Pressure	[bar]
p_f	= Contact Pressure at inner rim of flange	[bar]
k	= Contact Pressure to Inner Pressure Ratio	[-]
a	= Distance between inner rim and center line of flange	[m]
c	= Half width of flange	[m]
s	= Distance between bolt bore and center line of flange	[m]
t	= Bolt Span	[m]
D_i	= Inner Diameter of Casing	[m]

REFERENCES

API 617, 2002, “Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services”, Seventh Edition, American Petroleum Institute, Washington, D.C.

Lüdtke, K. H., 2004, “Process Centrifugal Compressors, Basics, Function, Operation, Design, Application”, Springer-Verlag, Berlin, Germany

Meher-Homji, C. B., Matthews, T., Pelagotti A., Weyermann, H. P., 2007, “Gas Turbines and Turbocompressors for LNG Service”, Proceedings of the Thirty-Sixth Turbomachinery Symposium, Turbomachinery Laboratory, Departement of Mechanical Engineering, Texas A&M University, College Station, Texas, USA.

Meher-Homji, C. B., Messersmith, D., Weyermann, H. P., Richardson, G., Pattrick, P., Biagi, F., R., Gravame, F., 2011, “World’s First Aeroderivative Based LNG Liquefaction Plant – Design, Operational Experience and Debottlenecking”, Proceedings of the First Middle East Turbomachinery Symposium, Doha, Qatar.

Reuter, H., 1958, „Die Flanschverbindung im Dampfturbinenbau“, BBC-Nachrichten, Mannheim, Germany

Salisbury, R., Rasmussen, P., Griffith, T., Fibbi, A., 2011, “Design, Manufacture and Test Campaign of the World’s Largest LNG Refrigeration Compressor Strings”, Proceedings of the First Middle East Turbomachinery Symposium, Doha, Qatar.

Traupel, W., 2000, “Thermische Turbomaschinen, 2. Band”, Springer-Verlag, Berlin, Germany

Ullmann, K., 1965, „Experimentelle Untersuchungen zur Flächenpressung in Teilfugen von Turbinengehäuse-flanschen“, Maschinenbautechnik, Berlin, Germany

VDI 2230, 2001, “Systematische Berechnung hochbeanspruchter Schraubenverbindungen – Zylindrische Einschraubenverbindungen“ (Systematic calculation of high duty bolted joints – Joints with one cylindrical bolt), Verein Deutscher Ingenieure, Düsseldorf, Germany.