

Modernization of Centrifugal Chlorine Compressor

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Received (23 June 2018)

Revised (19 July 2018)

Accepted (23 August 2018)

In this paper modernization of a three-stage chlorine centrifugal compressor, in which chlorine is pressurized has been described. Machine nominal mass flow has been increased by 20% and the pressure rise has remained almost unchanged. New construction concept has been chosen basing on analysis of different design variants. Calculations were conducted with one-dimensional method with usage of in-house developed code. Algorithms implemented to this program are based on empirical studies and wide experience in compressor modernization and design. Moreover, results of modified machine acceptance test results, which satisfies VDI norm 2045/ISO 5389 has been presented.

Keywords: modernization, revamp, chlorine, compressor, electrolyze, chemical industry.

1. Introduction

At present, within Poland new centrifugal compressors are no longer in production. However, especially in petrochemical industry an unit modernizations (revamps, retrofits) of existing machines are very often performed, due to economical and timeline limitation reasons.

In this paper some of the aspects of modernization of a three-staged chlorine compressor is described. The compressor was installed in 1970 in one of middle European chemical plant.

According to Kryłłowicz [6] monograph a variety of modernizations advantages over new machine purchase can be listed, from which most significant are as follows:

- re-use of existing casings, piping and peripheral equipment,
- re-use of existing fundamentals,
- re-use of existing driving unit (if no speed change is required),
- relatively short timeline,
- relatively high potential outcome of modernizations; gas path along the machine can be completely replaced, which would result in both efficiency and flow increase.

Based upon the above mentioned it could be deducted that the planned modernization of the chlorine compressor would be economically proficient investment. It has also to be pointed out that the original equipment manufacturer (OEM) does not longer exist as an independent market operator, therefore the target user lost the capability of original maintenance and spare parts delivery. Modernization would guarantee supplies of spare components by modernization performing company.

2. Machine description

The compressor operates at chlorine manufacturing line, based upon the electrolyze of aqueous salt solution with mercury addition. Common technological scheme of the process is presented in Fig. 1.

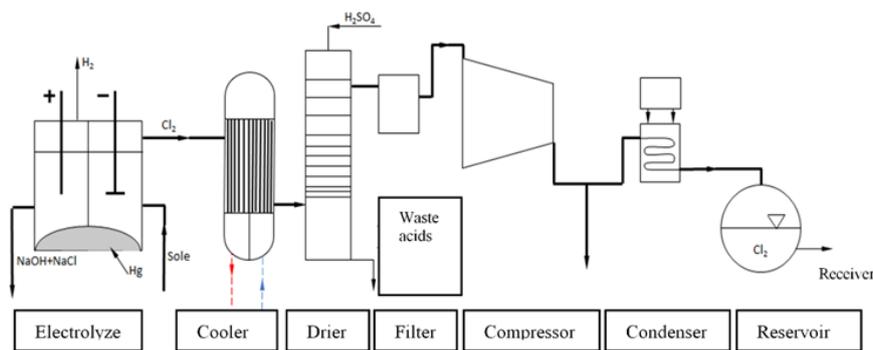


Figure 1 Simplified technological scheme of chlorine production by electrolyze of aqueous salt solution by Kramer [3]

The compressor sucks in the humid air from electrolyze, which is cooled down and dried using sulfur acid. The gas undergoes rough-filtration next using various coke-based filters.

The compressor should work in such a way that during the electrolyze a minor underpressure of few hundreds pascals (few mercury millimeters) is sustained. Too high underpressure would result in an inert gas suction, which would lead to chlorine pollution. Too low under-pressure would lead to chlorine leakages from electrolytes and site contamination.

Due to the resultant installation resistances (cooler, filter and piping) the suction pressure of the compressor is of approximately 82-97 kPa order. Inlet gas, apart from chlorine (97%-98% of volumetric flow), contains also inert gas and traces of Hg and H₂SO₄ (mercury and sulfur acid). In addition, some particles of 1-2 μm in diameter and mass share of 0.02-0.03 $\frac{g}{Nm^3}$ can be also included in the gas.

As a result of the mentioned issues, during the compression process the following hazardous factors can be listed:

1. high corrosion of the metal, especially above 100°C temperature,
2. high fire risk above 250°C, due to exothermical reaction between Cu-Cl₂,
3. Cl₂ reaction with machine lubricants – “soap” substance build up on the compressor parts.

Above mentioned factors generates the need of proper material selection and design solutions.

3. Original machine description – catalogue data

Fig. 2 presents the cross-sectional view of the original compressor and Fig. 3 represents the compressor location with respect to driver and the gearbox.

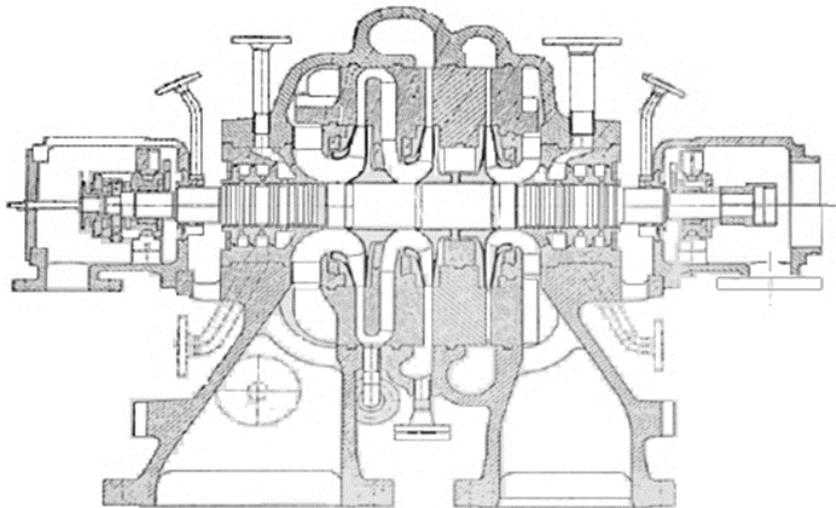


Figure 2 The cross-sectional view of the original compressor

Three-staged, two-sectioned, single shaft centrifugal compressor was driven by an electric motor through a gear-box. The compressor was equipped with an inter-cooler between first and second section (two-staged and single staged, respectively). Stage 2 and stage 3 ends up with the volutes extracting the gas outside the casing.

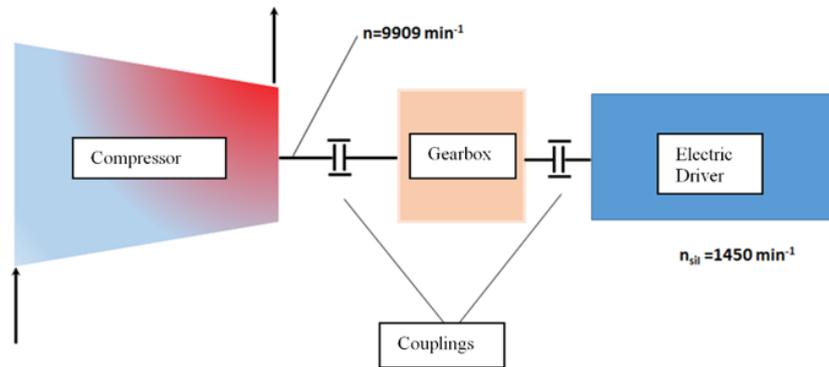


Figure 3 Compressor location with respect to the gearbox and electric motor

The impeller with backswept blades was manufactured by milling integrally with the hub and the cover was assembled by riveting. Downstream the impeller a set of vaneless diffusers was assembled. The design operating point parameters were listed in Table 1.

Table 1 The operating point parameters of the original machine

Suction side	Ambient pressure	$p_A = 86.2 \text{ kPa}$
	Ambient temperature	$T_A = 234.95 \text{ K } (-38.2^\circ\text{C})$
Discharge side	Ambient pressure	$p_Z = 425.5 \text{ kPa}$
	Ambient temperature	$T_Z = 312.25 \text{ K } (39.1^\circ\text{C})$
Operating rotational velocity	165.15 Hz (9909 rpm)	
Driving unit power	360 kW	

4. Selecting the design of the modernized compressor

Principal modernization target was to increase the mass flow by 20% with reference to the current operating point. Hence, the new design mass flow is given by the formula:

$$\dot{m}' = \dot{m} \cdot 1.2 = 4.5595 \frac{\text{kg}}{\text{s}} \cdot 1.2 = 5.4714 \frac{\text{kg}}{\text{s}} \quad (1)$$

Mass flow increase would influence the installation resistance curve. In this case it was assumed that the underpressure at suction side will increase by approx. 1.4%. The temperature will remain unchanged. Hence:

$$p'_A = \rho_A \cdot 0.986 = 86.2 \text{ kPa} \cdot 0.986 = 0.85 \text{ bar} \quad (2)$$

$$T'_A = T_A = 234.95 \text{ K} \quad (-38.2^\circ\text{C}) \quad (3)$$

As the mass flow is increased the new higher pressure ratio would be required. In this case it was estimated that new pressure ratio should be greater by 4%. As a result a new designed discharge pressure was calculated:

$$\Pi' = \Pi \cdot 1.04 = 4.9362 \cdot 1.04 = 5.134 \quad (4)$$

$$p'_Z = p'_A \Pi' = 85 \text{ kPa} \cdot 5.134 = 436.39 \text{ kPa} \quad (5)$$

To sum up, the following design requirements were given:

- mass flow increase by 20%,
- pressure ratio increase by 4%.

In the aftermath of discussing the technical possibilities and analyzing the information available in the public domain, two variants of modernizations were taken into consideration:

Variant 1:

- Complete new gas path elements, including rotor, diaphragms with return channels and diffusers,
- Change of operating rotational velocity – new gearbox.

Variant 2:

- Gas path elements remain unchanged, except for minor re-machining
- Complete new rotor

The first variant is significantly more expensive, but its realization could result in greater increase in efficiency. Second variant is less expensive but has got construction limitations. In both cases it was decided that decreasing the number of stages to two would be too risky as the local supersonic speed reaches only 188 m/s at suction. Finally, the second variant was approved for realization.

5. Calculations of the modernized machine

Substantial information about the gas path and compressor components geometry was gained by the given OEM documentation and the on-site measurements. This information allowed for creation of calculation model of the compressor, presented in Fig. 4.

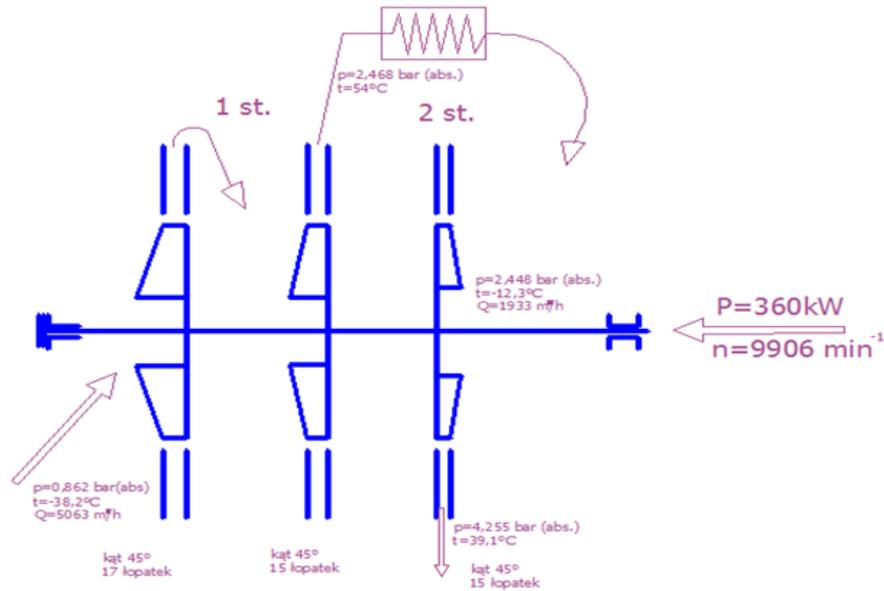


Figure 4 Calculation model of the chlorine compressor

Due to the small pressure on the suction side and relatively minor temperature ratio of the compressor (max temp. = 327 K) for the purpose of calculations it was assumed that the specific heat is a function of temperature only.

Determining the pressure ratio of the stages is done according to the following formula:

Total pressure ratio of the compressor:

$$\Pi_{A-Z} = \frac{p_Z}{p_A} = \frac{425.5 \text{ kPa}}{86.2 \text{ kPa}} = 4.9362 \quad (6)$$

Pressure ratio of the first section:

$$\Pi_I = \frac{p_{Z_I}}{p_A} = \frac{246.8 \text{ kPa}}{86.2 \text{ kPa}} = 2.863 \quad (7)$$

Pressure loss in the inter-cooler

$$\Delta p_{ch} = p_{Z_I} - p_{A_{II}} = 246.8 \text{ kPa} - 244.8 \text{ kPa} = 2 \text{ kPa} \quad (8)$$

Pressure ratio of the second section:

$$\Pi_{II} = \frac{p_Z}{p_{A_{II}}} = \frac{425.5 \text{ kPa}}{244.8 \text{ kPa}} = 1.738 \quad (9)$$

Thermodynamic calculations of the original machine were based upon the PARAM software, available thanks to Institute of Turbomachinery, TUL. This 2D software

calculates the medium parameters at the given control surfaces of the compressor stage. The results of the calculations are presented in Table 2 in form of blading characteristics.

Table 2 Blading characteristics of the original and modernized machine

Stage		1	2	3
D ₂ [mm]		374	374	374
b ₂ [mm]	orig	17	13.5	9.5
	mod	20.72	14.26	9.45
β ₂ [°]	orig	45	45	45
	mod	50	48	45
Z [-]	orig	17	15	15
	mod	17	15	15

where:

- D₂ – impeller diameter,
- b₂ – impeller top width
- β₂* – blade outlet angle,
- Z – blade number in form of the blading characteristics,
- orig -original machine,
- mod – machine after modernization

6. Energy balance of the modernized compressor

The transmitted power of the original compressor is determined from the given formula:

$$P_u = \dot{m} \sum_{i=1}^3 e_{u_i}$$

$$P'_{u_{\text{mod}}} = 4.5595 \frac{\text{kg}}{\text{s}} \cdot \left(21.406 \frac{\text{kJ}}{\text{kg}} + 20.84 \frac{\text{kJ}}{\text{kg}} + 22.30 \frac{\text{kJ}}{\text{kg}} \right) = 294.3 \text{ kW} \quad (10)$$

(values of the transferred Euler energy e_{u_i} were taken from PARAM code).

Assuming following efficiencies, reflecting the state of art of compressor manufacturing, estimated power consumption at coupling location can be calculated.

- Compressor internal efficiency $\eta_i = 0,98$
- Compressor mechanical efficiency $\eta_m = 0,97$
- Compressor total efficiency = 0,965

$$(P_{\text{coupl}})_{\text{mot}} = \frac{P_u}{\eta_i \eta_m \eta_p}$$

$$(P_{\text{coupl}})_{\text{mot}} = \frac{294.3 \text{ kW}}{0.98 \cdot 0.97 \cdot 0.965} = 320.8 \text{ kW} \quad (11)$$

The transmitted power of the modernized compressor is determined from the formula:

$$P'_{\text{u}_{\text{mod}}} = \dot{m}' \sum_{i=1}^3 e'_{\text{u}_i}$$

$$P'_{\text{u}_{\text{mod}}} = 5.4714 \frac{\text{kg}}{\text{s}} \cdot \left(23.103 \frac{\text{kJ}}{\text{kg}} + 22.866 \frac{\text{kJ}}{\text{kg}} + 21.79 \frac{\text{kJ}}{\text{kg}} \right) = 370.4 \text{ kW} \quad (12)$$

Values of the componential energy processed by each stage were taken from data included in table 2. Power loss to the leakages was calculated according to the formula below:

$$P_p = \sum_{i=1}^3 \dot{m}_{\text{Lab}_i} e_{\text{u}_i}$$

$$P_p = 0.0517 \frac{\text{kg}}{\text{s}} \cdot 23.103 \frac{\text{kJ}}{\text{kg}} + 0.0685 \frac{\text{kg}}{\text{s}} \cdot 22.866 \frac{\text{kJ}}{\text{kg}} + 0.085 \frac{\text{kg}}{\text{s}} \cdot 21.729 \frac{\text{kJ}}{\text{kg}} = 4.607 \text{ kW} \quad (13)$$

Disc friction losses of the impellers were calculated according to simplified formula by [2], assuming constant friction loss coefficient $B_T = 910\text{-}4$

$$P_B = B_T \rho_2 u_2^2 D_2^2$$

Since the impeller diameters and so the peripheral velocities are equal for each stage the resultant power of wading losses can be given by the formula:

$$P_B = B_T u_2^2 D_2^2 (\rho_{2_1} + \rho_{2_2} + \rho_{2_3})$$

$$P_B = 9 \cdot 10^{-4} \cdot 119 \frac{\text{m}^2}{\text{s}} \cdot 0.374 \text{ mm}^2 \cdot \left(4.29 \frac{\text{kg}}{\text{m}^3} + 6.235 \frac{\text{kg}}{\text{m}^3} + 11.428 \frac{\text{kg}}{\text{m}^3} \right) = 4657 \text{ W} \cong 4,6 \text{ kW} \quad (14)$$

To cover the losses the internal power of the compressor is expressed as follows:

$$P_{\text{u}_{\text{mod}}} = P'_u + P_p + P_B = 370.4 \text{ kW} + 4.6 \text{ kW} + 4.6 \text{ kW} = 379.6 \text{ kW} \quad (15)$$

Taking into account the previously calculated efficiencies η_m and η_p the compressor total power demand is calculated:

$$(P_{coupl})'_{mod} = \frac{379.6 \text{ kW}}{0.97 \cdot 0.965} = 405.5 \text{ kW} \quad (16)$$

Assuming the power excess factor $k = 1.12$ driving unit power is determined:

$$P_s = (P_{coupl})'_{mod} \cdot 1.12 = 405.5 \text{ kW} \cdot 1.12 = 454 \text{ kW} \quad (17)$$

Finally, a driving unit with 465 kW power was chosen and installed.

Fig. 5 shows the result of modernized machine acceptance test according to VDI 2045/ISO 5389 standards.

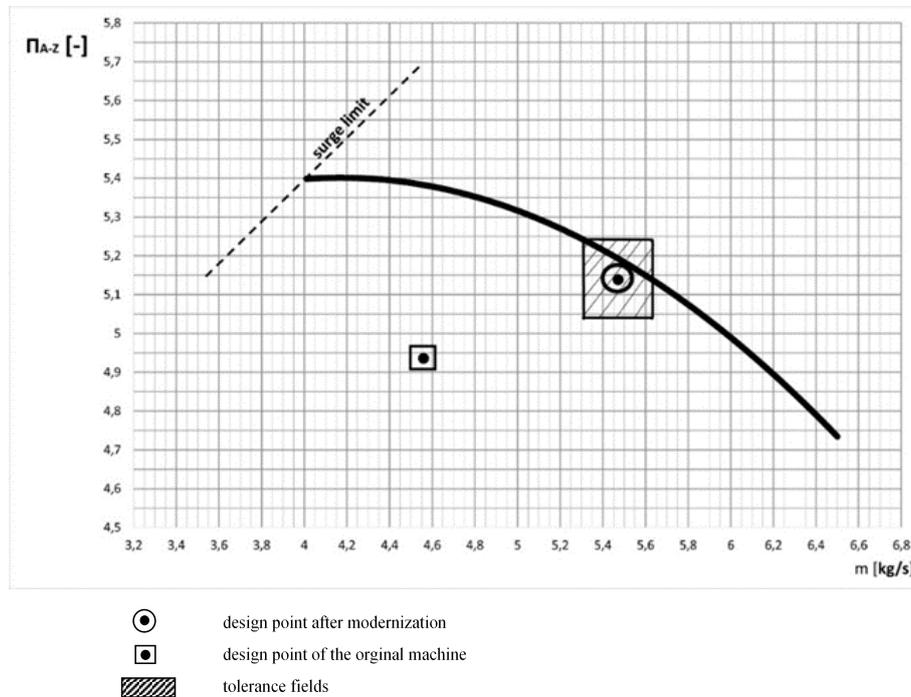


Figure 5 Results of modernized machine acceptance test

No matter what is the agreed scope of the modernization, the manufacturer should also take care about the proper material analysis and selection, adequate to the working medium. This refers to both rotating and stationary parts including the casing. Re-used parts should be in acceptable condition with no evident cracks and free of corrosion areas. One of the problematic issue when re-using the stationary parts is the repair/regeneration. If during NDT (non-destructive testing) some minor cracks or erosion areas are detected they can be repaired by welding. As for the described modernization the casing base material was GS26CrMo4 cast steel, which is difficult to weld, therefore without appropriate technology application

(specific welding material selection and post-welding heat treatment) the repair will be unsuccessful.

7. Summary and conclusions

The described case of the chlorine compressor modernization was a typical design issue, followed by a vast economical and technical analyses. Similar studies are getting more and more frequent in Poland due to operation timeline limitations of compressors used especially in petrochemical industry. The required increase of compressor mass flow was 20% with simultaneous increase of pressure rise by 4%. In this paper two variants of modernization have been presented. First variant needed complete replacement of gas path elements such as rotors, diffusers etc. What is more, increase of operating rotational velocity was necessary and thus replacement of current gearbox. This modification would result in better compressor efficiency than in variant 2, but also in significantly longer overhaul and what is more important, it would be much more expensive. In second variant only rotors had to be changed and the rest of gas path elements remained the same. As an effect lower cost of modernization and shorter stoppage in compressor operation were needed. Despite the fact that this variant is much more complicated in terms of design, it was decided to conduct this version of modernization. Important limitations of mass flow rate increase was diameter of inlet pipe (resultant high Mach number in this region) and required 20% of increase was maximum possible value.

Chemical plants rarely accept long overhauls allowing for extended scope of modernization. Simultaneously, plants are not willing to spend money for more expensive enterprises, as Variant 1 described above. Hence replacing the rotors and usage of existing stationary parts are a fair compromise, usually meeting the user principal expectations. Majority of cases with such a scope is completely satisfactory, as in this particular case.

References

- [1] **Cumpsty, N. A.:** Compressor Aerodynamics, *John Wiley & Sons, New York, 1989.*
- [2] **Eckert, B., Schnell, E.:** Axial – und Radial – Kompressoren, *Springer Verlag, 1961.*
- [3] **Kramer, W.:** Chlogas-Kreiselverdichter-Entwicklung, *Pumpen und Verdichter*, 1/68, 37–42, Leipzig, **1968.**
- [4] **Kryłłowicz, W., Hanausek, P.:** Aspekty konstrukcyjne oraz przepływowe modernizacji sprężarki promieniowej ČKD, *ZN PL, CMP*, 116, 69–77, **2000.**
- [5] **Kryłłowicz, W., Kozanecki, jr Z., Świder P., Kozanecki Z.:** Modernization of a high pressure synthesis gas turbocompressor, *Transactions of the Institute of Fluid-Flow Machinery*, 131, 121–130, **2016.**
- [6] **Kryłłowicz, W.:** Teoria I praktyka modernizacji sprężarek promieniowych, *Mono-graph, Łódź, 2013.*
- [7] **Lüdtke, C.:** Rerate of Centrifugal Compressors – an Alternative for Production Increase and Efficiency Improvement, *ZN PL, CMP*, 108, 239–251, **1995.**
- [8] **Münger, F., Näf C.:** Neues Innenleben für bestehende Verdichter, *Sulzer Technical Review*, 1/2000, 28–31, **2000.**
- [9] **Świder, P., Kozanecki, Z., Graczykowski, M., Kryłłowicz, W.:** Technical aspects of a large size industrial process turbo compressor revamp, *Open Engineering*, 5, 1, 438–446, **2015.**