TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 131, 2016, 121–130

Władysław Kryłłowicz[b], Zbigniew Kozanecki Jr[a],*
Piotr Świder[a], Zbigniew Kozanecki[b]

Modernization of a high pressure synthesis gas turbocompressor

[a] *NEO-TEC Sp. z o.o., Kobiałka 11a, 09-411 Płock, Poland*
[b] *Lodz University of Technology, Institute of Turbomachinery, 219/223 Wólczańska, 93-005 Łódź, Poland*

Abstract

The paper describes several modernization aspects of a three-staged synthesis gas radial turbo compressor. The principal aim was to adapt the machine to new working conditions, resulting from the current state-of-art production technology. The required discharge pressure was changed from 21.1 MPa to 14 MPa. The paper covers the discussion on the scope of modernization, its variants and the final variant choice. It provides review of the thermodynamic calculation methodology, dedicated test stand results and also some of the on-site acceptance tests. The revamped compressor is in continuous service since October 2015.

Keywords: Syngas; Radial compressor; Modernization

1 Introduction

This paper includes the summary of experience gained during the revamp of a three-staged synthesis gas radial turbo compressor. The original compressor was an old construction from middle sixties of previous century. During the operation period, due to production technology change the required discharge pressure demand decreased by nearly 50%.

This change was possible because of a modern converter application. This new delivery pressure made the compressor to operate off design, away from its original

*Corresponding Author. Email address: z.kozanecki@neo-tec.pl

designed operating area. The principal aim of the revamp was do adapt the compressor to the new working conditions.

The modernisation was a joint venture of Neo-Tec and Institute of Turbomachinery, Lodz University of Technology.

2 Object description

Modernised machine is a three-staged, 'vertical split' type radial compressor driven by a steam turbine. Working fluid is a combination of hydrogen (74%) and nitrogen, with minor additives of methane and argone of total molar mass of 8.69 kg/kmol. Figure 1 presents the layout of the machine.

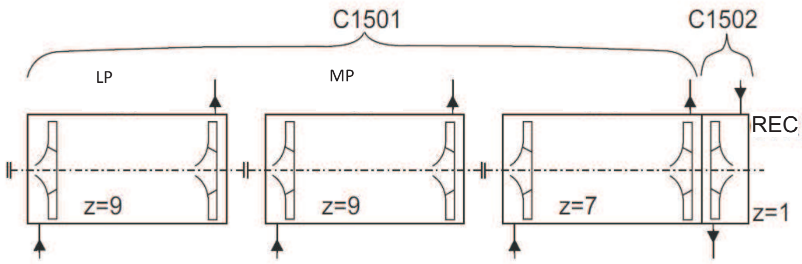


Figure 1: C1501/C1502 compressor layout: LP, MP, HP – low pressure, medium pressure, high pressure section, REC – circulating stage [7].

Nominal working point speed is 11 100 rpm (185 Hz), with total turbine coupling power consumption of 13470 kW. Number of stages in each section is as follows: $z_{LP} = 9$, $z_{MP} = 7$, $z_{HP} = 7$. In HP casing a single-staged 'synthesis recirculating' compressor (symbol C1502) is assembled. The gas at the discharge of HP (high pressure) section is directed to the synthesis circle, where a converter is located. The circulation stage provide the flow in the synthesis circle and through converter. In each cycle only part of NH_3 is condensing thus the circulation flow is several times higher than compressor discharge. Gas composition in the circle differ from the gas flowing through the compressor due to NH_3 existence; its molecular weight is approximately 10.68 kg/kmol.

Calculated and measured original polytrophic efficiencies were very low with respect to the current state of art, but were reasonable at the time when machine was built.

Table 1: Original compressor parameters at the nominal operation point and chosen reverse thermodynamic calculation results.

		Compressor section			
Value	Unit	LP	MP	HP	REC
p_A	MPa	24.525	70.34	137.93	202.58
T_A	K	305	306	306	299
p_Z	MPa	71.12	139.30	210.91	218.77
T_Z	K	468	412	375	307.5
$\dot{m}$	$\frac{\mathrm{kg}}{\mathrm{s}}$	9.521	9.521	9.521	61.168
Π_{A_Z}	–	2.9	1.98	1.5292	1.0799
η_p	–	0.629	0.605	0.585	0.674

p_A – inlet pressure, T_A – inlet temperature, p_Z – discharge pressure, T_Z – discharge temperature, $\dot{m}$ - mass flow, Π_{A_Z} – pressure ratio, η_p – polytrophic efficiency

3 Modernization parameters

Original design (D) and modernized (M) parameters are listed in Tab. 2 below:

Table 2: Compressor design and modernisation parameters.

	Unit	Design condition 'D'	Modernization conditions 'M'
C1501 Compressor discharge pressure	MPa	21.091	13.975
Converter loop pressure loss	MPa	1.619	0.9
Converter inlet temperature	K	438 (165,°C)	413 (140 °C)
Recirculation compressor inlet flow	$\frac{\mathrm{Nm}^3}{\mathrm{s}}$	122.22 (440250 Nm^3/h)	95.48 (343735 Nm^3/h)
Compressor C1501 inlet flow	$\frac{\mathrm{Nm}^3}{\mathrm{s}}$	27.297 (98270 Nm^3/h)	28.083 (101098 Nm^3/h)

Comparing these parameters it is seen that the modernisation is in fact manufacturing a new machine. New pressure ratio is $\Pi_M \cong 5.75$, whereas the original was

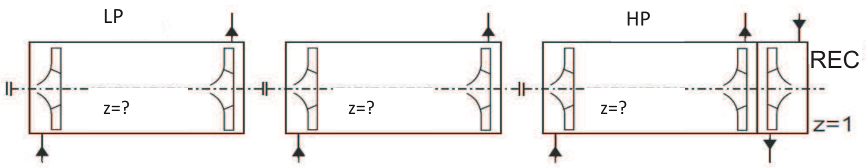
$\Pi_D \cong 8.60$. Volumetric flows are nearly preserved. The recirculating stage parameters are changed to a smaller extent; pressure ratio decreased from $\Pi_D \cong 1.083$ to $\Pi_M \cong 1.07$ and the volumetric flow increased by approx. 7%.

4 Choosing the modernisation conception and scope. Aerodynamic and construction limitations

After analyzing the requirements, the following modernisation scope was agreed:

- complete new flow channel (gas path) design,
- new diaphragms design and manufacturing,
- new bearings, seals and couplings,
- new vaneless diffusers manufacturing.

Four variants of modernisation were taken into consideration. These variants are presented in Fig. 2. In all variants A–D the recirculating stage remains the same.



A:	**z=9**	+	**z=9**	+	**z=7**	+	**z=1**
B:	**z=8**	+	**z=7**	+	**z=4**	+	**z=1**
C:	**z=9**	+	**z=9**	+	**z=0**	+	**z=1**
D:	**z=9**	+	**z=9**	+	**z=1**	+	**z=1**

Figure 2: Considered variants of modernisation C1501 compressor, where z is the number of stages on LP, MP and HP section, respectively [7].

Variant A – Basic. The number of stages (9 + 9 + 7 + 1 REC) is preserved, but the blade angles and geometry are different. Main disadvantage – significant leakage and internal flow losses, too many stages with respect to the requirements.

Variant B – Modified. Reduced number of stages (8 + 7 + 4 + 1 REC). Blade outlet angles are changed (Tab. 3), with preserved number of blades for LP (low pressure) and MP (medium pressure) sections and increased number of blades for HP section.

In this concept a new dynamics and stress analysis of the rotors was required. Advantages: design adequate to the new working parameters, original piping and cooling system can be preserved.

Variant C – Minimal. In this concept the number of stages is reduced (9 +9 + 1 REC), so only LP and MP section are in use. HP section is a 'blind' rotor, with recirculating stage only. This solution is potentially most energy efficient, but requires additional compensation for axial forces in HP section and also major changes in existing piping and cooling system.

Variant D – Miscellaneous. Machine layout as in Minimal variant, but in HP section, except for the REC stage, additional single stage is designed, from which the gas flows to the original suction pipe. The advantages is preserving both the original balancing system of HP section, piping and cooling system.

Mutual discussion about the benefits and disadvantages of each solution between the investor and the manufacturer resulted in variant B choice. Reduced number of stages at each section allowed for stage axial pitch increase by 6–7% for LP and 8–10% for remaining sections according to Lindner [12] axial pitch increase is beneficial for the stage efficiency.

5 Calculation methodology, flow path characteristics, test run results

Due to the lack of experience in field of high pressure radial compressor design the calculations were made with aid of numerical codes:

- 1D code, written and own by Institute of Turbomachinery, Lodz University of Technology. The algorithm of the code described in monography [11].
- Two computational fluid dynamics codes, including Ansys CFX [18].

Kinematics of the impellers were based upon the Benvenuti research [1], Paroubka and Cyrus [15]. MP and HP sections impellers are working within diffuser flow angles regime, for which according to Jansen [4] a rotating stall phenomenon may occur. Thank to Biba simulations [2] and his conclusions this phenomenon was eliminated.

Table 3: Blade characteristics.

Section		LP	MP	HP
Blade outlet angles $\beta_2^*[°]$	Design	45.5–30	36–36	30–30
	Modernization	42–30	42–32	28–29
Blade number z	Design	26–22*	22*	11
	Modernization	See above	See above	13

(*) Impellers with splitters

Original impeller diameters (from $D_2 = 387$ to $D_2 = 520.8$ mm) were not significantly changed. Taking into consideration flow coefficient, defined by Luedtke [13]:

$$\Phi = \frac{4\,\dot{V}}{\pi\, D_2^2\, u_2}\ , \tag{1}$$

where $\dot{V}$ is the inlet volumetric flow, D_2 is the impeller diameter, and u_2 is the peripheral velocity, it can be seen that all stages in all three sections are 'low flow coefficient stages'. For first stages of LP, MP, HP sections flow coefficients are as follows:

1st stage, LP $\longrightarrow$ $\Phi = 0.0163$,
1st stage, MP $\longrightarrow$ $\Phi = 0.0163$,
1st stage, HP $\longrightarrow$ $\Phi = 0.0163$.

The lowest value can be found for the last HP stage

$$(\Phi)_{\min} = 0.0112\ . \tag{2}$$

A representative test stand was built for each section. The layout of the stand, measurements methodology and results discussion is described in details in [5].

Figures 3 and 4 show the view of new rotors for modernized machine – LP, MP and HP section. Figures 5 and 6 present the acceptance on-site test results. Figure 5 refers to the compressor and Fig. 6 – recirculating stage.

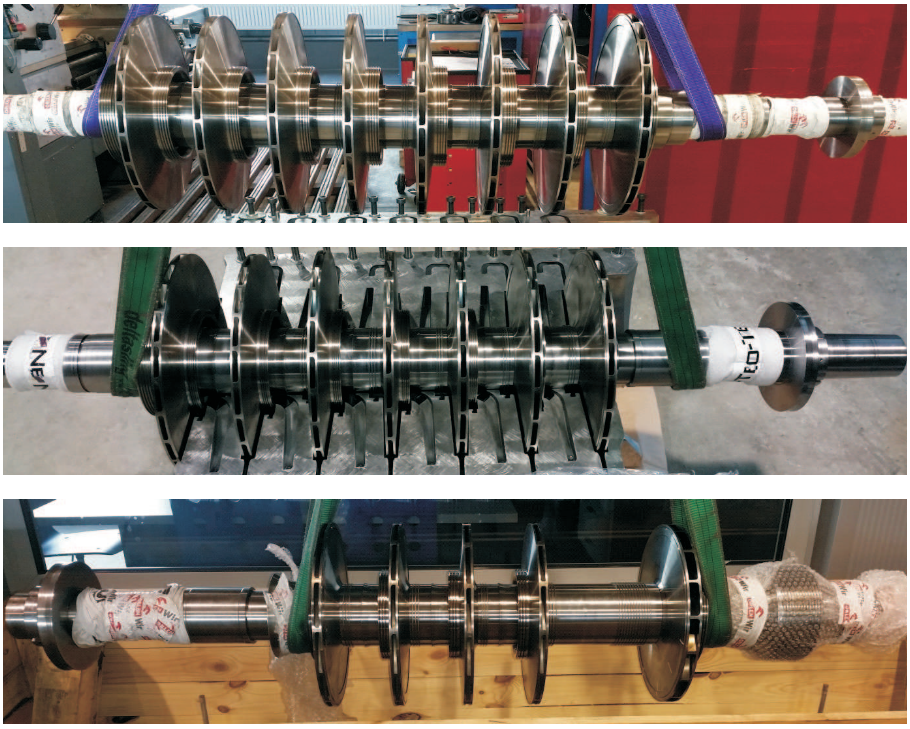


Figure 3: New rotors for modernized machine – LP, MP, and HP section, respectively.

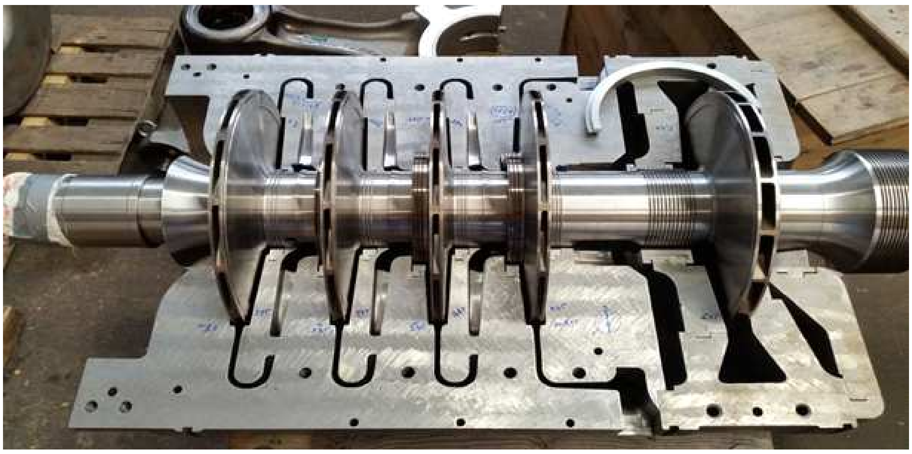
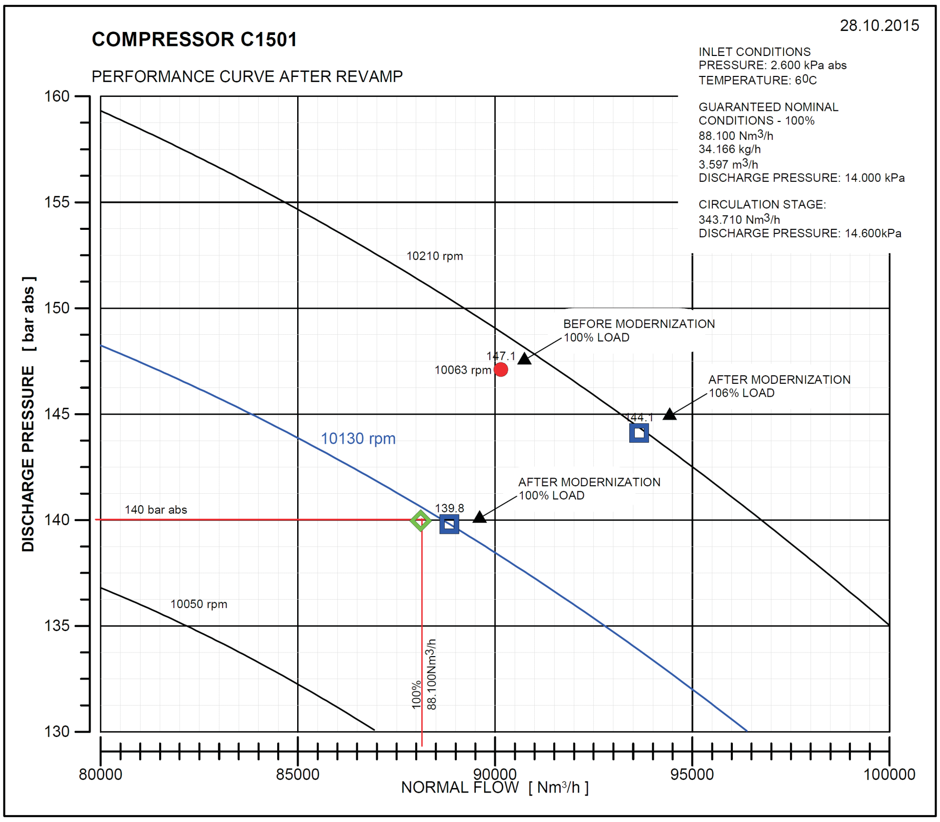


Figure 4: HP section of the modernized machine.



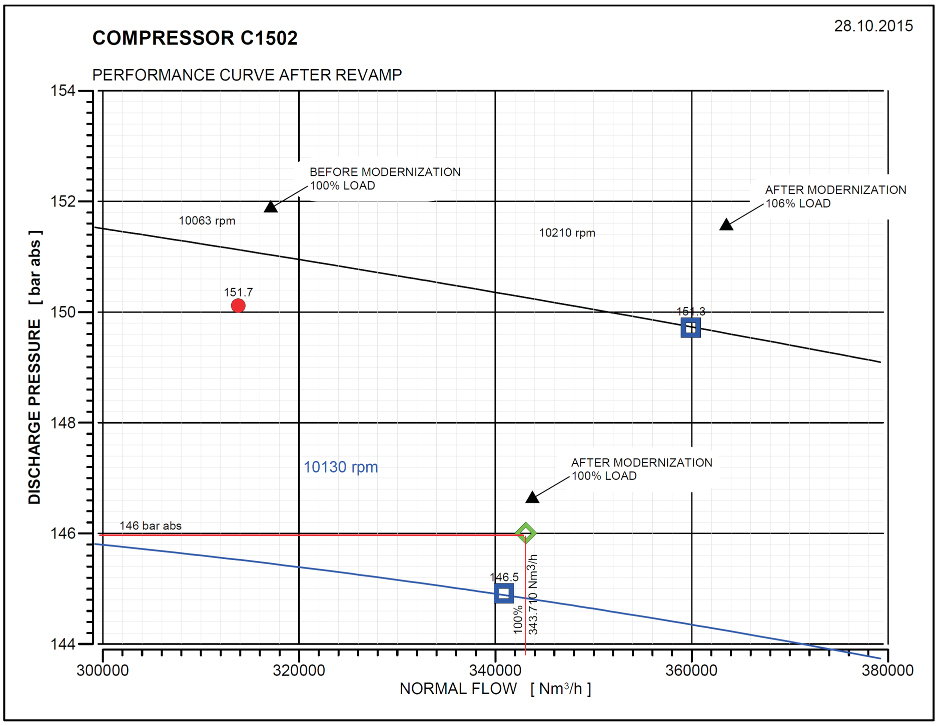
● – working point before the modernization – test data; ◇ – guaranteed working point after modernization – predicted, ◻ – guaranteed working point after modernization – test data

Figure 5: Compressor C1501 characteristics after modernization.

6 Summary

Looking at the plot in Fig. 5 and 6, it can be seen that the position of tested working point corresponds to the guaranteed point. Achieved polytrophic efficiencies for sections 1, 2, and 3 are equal to $\eta_{p_{LP}} = 0.691$, $\eta_{p_{MP}} = 0.68$, $\eta_{p_{HP}} = 0.63$, $\eta_{p_{REC}} = 0.703$ respectively. These efficiencies are substantially greater than the original. Assumed steam consumption reduction was approximately 7.5 t/h, but the test runs revealed that in fact the real steam savings were nearly 12.9 t/h. This corresponds to the compressor power requirement drop by 1670 kW.

Comparing the results with data collected by Connor [3] or Muenger [14] it can



● – working point before the modernization – test data; ◇ – guaranteed working point after modernization – predicted, □ – guaranteed working point after modernization – test data

Figure 6: Recirculating stage characteristics after modernization.

be stated that the quality of the modernization was coherent with the world state-of -art. The revamp experience described in [6–10, 16] significantly contributed to the overall success of this modernization.

Received in July 2016

References

[1] Benvenuti E.: *Research on centrifugal compressor stages with low flow coefficients.* Quaderni Pignone **15**(1972), 11–19.

[2] Biba Y.I., Nye D.A., Zheji L.L.U.: *Performance evaluation and fluid flow analysis in low flow stages of industrial centrifugal compressor.* Int. J. Rotat. Mach. **8**(2002), 5, 309–317.

[3] O'Connor J.: *Uprating feasibility studies for turbocompressors in refinery and chemical processes-improving existing equipment*. Sulzer Techn. Rev. **3**(2007), 10–13.

[4] Jansen W.: *Rotating stall in a radial vaneless diffuser*. J. Basic Eng. 86(1964), 4, 750–758.

[5] Kabałyk K., Kryłłowicz W.: *Numerical modeling on the performance of a centrifugal compressor impeller with low inlet flow coefficient* Transactions IFFM, **131**(2016), 41–53.

[6] Kozanecki Z.: *Rotating Systems of the Low and Medium Power Turbomachinery.* ITE Publ. House, Łódź-Radom 2008 (in Polish).

[7] Kozanecki Z., Kryłłowicz W.: *Analysis of technical modernization possibilities of the synthesis gases compressor C1501–C1502 in Anwil SA Ammonia Plant, part I, part II.* Opinion on the viability of already operating machine in conditions of cooperation with the new converter D 160, Łódź 2012 (in Polish).

[8] Kozanecki Z., Kryłłowicz W., Świder P., Kozanecki Z. jr: *Complex modernization of large size industrial process compressors-technical aspects and examples.* [In :] Proc. VETOMAC VIII Int. Conf. on Vibration Engineering and Technology of Machinery, R. Rzadkowski (Ed.), Wydawnictwo IMP PAN, Gdańsk 2012, 155–166.

[9] Kryłłowicz W., Kozanecki Z.: *Complex modernization of the ammonia compressor.* ZN PŁ, CMP, **133**, Łódź 2008, 165–174.

[10] Kryłłowicz W., Kozanecki Z., Świder P.: *Modernization of a nitrous gas turbine – driven turbocomressor.* Mechanics and Mechanical Eng. **15**(2011), 3, 273–288.

[11] Kryłłowicz W.: *Theory and practice of modernization of the radial compressor.* Monografie Politechniki Łódzkiej, Łódź 2013 (in Polish).

[12] Lindner P.: *Aerodynamic tests on centrifugal process compressors – influence of diffusor diameter ratio, axial stage pitch and impeller cutback.* ASME J. Eng. Power-T ASME **105**(1983), 4, 910–919.

[13] Luedtke K.H.: *Process Centrifugal Compressors. Basics Function, Operation, Design, Application.* Springer, Berlin Heidelberg 2004.

[14] Muenger F., Naef C.: *Neues Innenleben fuer bestehende Verdichter*, Sulzer Techn. Rev. **1**(2000), 28-31.

[15] Paroubek J., Cyrus V., Kynel J.: *Experimental investigation and performance analysis of six low coefficient centrifugal compressor stages.* J. Turbomach **117**(1995), 4, 585–592.

[16] Świder P., Kozanecki Z., Graczykowski M., Kryłłowicz W.: *Technical aspects of a large size industrial process turbocompressor revamp.* Open Eng. **5**(2015), 1, 438–446.

[17] Yongsheng Wang, Feng Lin, Chaogun Nie: *Design and performance evaluation of a very low flow coefficient centrifugal compressor.* Int. J. Rotat. Mach. **2013**(2013), ID 293486.

[18] Ansys website (2015), www.ansys.com , (accessed on 15 March, 2015).