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Modernization of a Nitrous Gas Turbine–Driven Turbocompressor

Władysław Kryłłowicz
Zbigniew Kozanecki
Institute of Turbomachinery
Technical University of Łódź
i10@adm.p.lodz.pl

Piotr Świder
Neo–Tec sp. z o.o
piotr.swider@neo–tec.pl

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The modernization of a nitrous gas four–stage centrifugal compressor is presented. The assumptions for modernization, its methodology and the results of thermodynamical, flow and dynamical calculations have been discussed. The technological aspects of the rotor modernization have been given.

Keywords: Nitrous gas, compressor, modernization, stability, rotordynamics

1. Introduction

Selected aspects of a C2102 nitrous gas centrifugal compressor modernization are presented. The basic aim of the modernization was to increase the flow by 20%, which allows for intensification of the nitric acid production line the compressor operates in. The C2102 compressor constitutes "a compressor train" together with three other machines (Fig.1).

The remaining machines in the train are: a C2101 axial air compressor and two drive turbines: a TC 2102 steam turbine and a TC 2101 expansion turbine. As a result, the C2102 compressor cannot be considered separately, without taking into account the properties of the remaining machines. The issues related to the modernization assumptions, the methodology of thermodynamic, flow and dynamical calculations and the technology of impeller manufacturing have been discussed. The problems connected to materials strength calculations as well as to the selection of structural materials have been neglected.

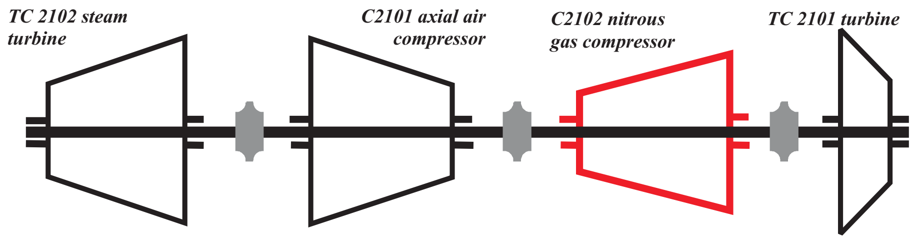

Figure 1 General arrangement of the "compressor train"

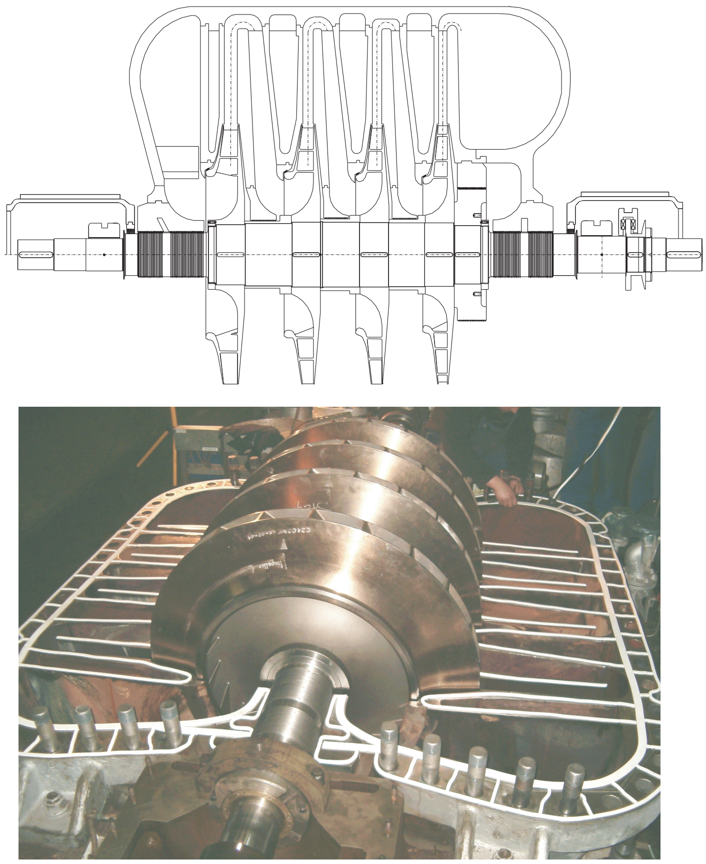

Figure 2 Longitudinal section and a view of the GHH compressor

2. Description of the object to be modernized

A large four–stage centrifugal compressor manufactured by GHH (Gutehoffnunshuette) was modernized. A longitudinal section and a view of the compressor are presented in Fig. 2.

The C2102 compressor at the rated rotational speed n=4000 rpm is characterized by the flow of the working medium (measured at suction) equal to 53 800 Nm^3/h. The rated pressure rate is Π=3.

This machine (manufactured in the end of 1960's) belongs to the oldest generation of such machines from the point of view of its design. The 2D type impellers of the constant diameter D_2=1120 mm were riveted. They are characterized by low relative widths (b_2/D_2 from 0.054 for the first stage up to 0.0374 for the fourth one), which results in low efficiency of the machine. The internal isentropic efficiency η_{si} does not exceed the value η_{si}=0.73. All the diffusers are vaneless and long (D_4/D_2=1.75). The casing diaphragms are cast and characterized by very thin walls, which yields some limitations for the modernization.

The second feature that limits the modernization possibilities is a small "stage pitch", defined as a ratio of the stage axial span to the outer diameter of the impeller. This pitch equals approx. 0.25, which in practice makes the application of modern impellers impossible [6].

The working medium is a mixture of gases composed of N_2 (mole fraction 86.36%), NO_2 (mole fraction approx. 5.29%) and O_2 (4.1%) and some smaller quantities of other components (NO, N_2O_4, Ar).

The gas composition differs slightly in winter and summer, similarly as the parameters at suction. The molar mass of the gas fluctuates from M_m=29.747 to M_m=29.,84 kg/kmol, the suction pressure varies from 270 up to approx. 300 kPa, and the inlet temperature $T_A$changesfrom 310 up to 325 K.

3. Assumption for the modernization

As a result of the discussion on the experimental investigation outcomes, it was assumed that for the required mass flow increase by 20%, it would be necessary to attain the pressure ratio of Π=3.29, which would compensate for the increased pressure losses.

Two modernization variants were analyzed, namely:

1. a decrease in the number of stages to three, but with 3D type impellers, where the first stage would have an axial inlet impeller,

2. maintenance of four stages, but with total replacement of casing diaphragms, which would increase the useful length of the flow system and would enable an application of two 3D impellers.

Despite the fact that the estimated increase in the efficiency would be at least 5%, those solutions were abandoned due to high investment costs.

Finally, another solution was adopted:

1. an exchange of the rotor into a new one, with slightly wider 2D impellers (within the design limitations),

2. the required flow increase would be divided into the following proportions: 5% (design alternations) and 15% (an increase in the rotational speed from 4000 to 4200 rpm).

These assumptions are followed by the necessity to:

1. adapt the steam and expansion turbine to continuous operation under increased rotations and higher power consumption,

2. analyze the dynamics of the new rotor in detail and to introduce the required alternations in the bearing system.

4. Calculation methodology and characteristics of the modernized flow system

Owing to simplicity of the flow system, a 1D calculation method was applied in the calculations, and the idea of CFD computations was abandoned. A standard calculation package used at the Institute of Turbomachinery, TUL, was used.

The working gas properties were determined on the basis of the Berthelot equations of state. An interesting calculation problem consists in taking into account the specific reaction that occurs inside the flow system [7].

At the temperature under 147 °C (420 K), gaseous nitrogen tetroxide originates spontaneously from nitrogen dioxide NO_2, according to the scheme:

$2NO_2 = N_2O_4 + 61.45kJ$. At the temperature above 147 °C, N_2O_4 exists only in trace quantities, and the reaction of particle breakdown of N_2O_4 into two particles of NO_2 is endothermic (this effect causes a drop in the mole fraction of N_2O_4 from 0.97% at suction to 0. 01% at discharge in the C2102 compressor). However, due to a small mole fraction of N_2O_4, the actual temperature drop is imperceptible in practice.

Tab. 1 presents a comparison of the flow system geometry in the C2102 compressor in the original version and after modernization.

Fig. 3 shows a comparison of the impeller of the third stage in the original and modernized version. An increase in the impeller blade outlet angle $\beta_2$from the value $\beta_2 = 44^o$ to $\beta_2 = 45^o$(the solution was applied in stages 3 and 4) makes it possible to attain the required pressure ratio.

5. Dynamics of the new rotor

A physical and mathematical modeling of dynamic properties plays an important role in the designing process of new rotary machines and in modernization of the existing ones. A high level of the machine rotating system vibrations is often caused by the fact that values of synchronous excitations (connected with machine shaft rotations) as far as frequencies are concerned and values of natural frequencies of subsequent modes of rotating system vibrations or wide-range excitations, in whose range free vibration frequencies are included, are too close. Standards for rotary machines commonly used in chemical industry (API 612, 617, 610) comply very well with the requirements for the real dynamic characteristics of the rotating system and they determine the permissible separation margins between critical vibration frequencies with respect to the nominal operating range of the machine.

Table 1 Comparison of the geometry before and after modernization

No.	Value	Symbol	-	1st stage	2nd stage	3rd stage	4th stage
1	Impeller outer diameter	$\mathbf{D}_2$	-	1120	1120	1120	1120
2	Outlet blade width	$\mathbf{b}_2$	new	67.5	53.3	47.52	43.5
			original	63.5	50.5	45.00	41.2
3	Blade outlet angle	β_2^*	new	50	48	45	42
			original	50	48	44	39
4	Impeller eye diameter	D_{0z}	new	674.5	652.7	632.0	611.0
			original	660	640	620	600
5	Blade inlet dia–meter (average)	$\mathbf{D}_1$	new	693.5	662	646	620
			original	705	669	648	621

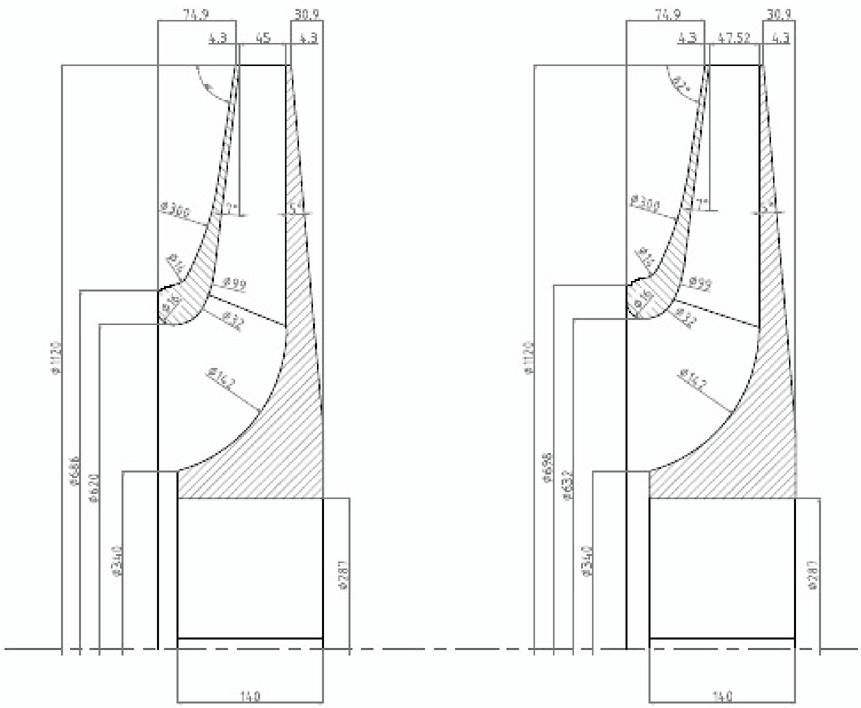

Figure 3 Comparison of the impeller of the third stage in the original and modernized version

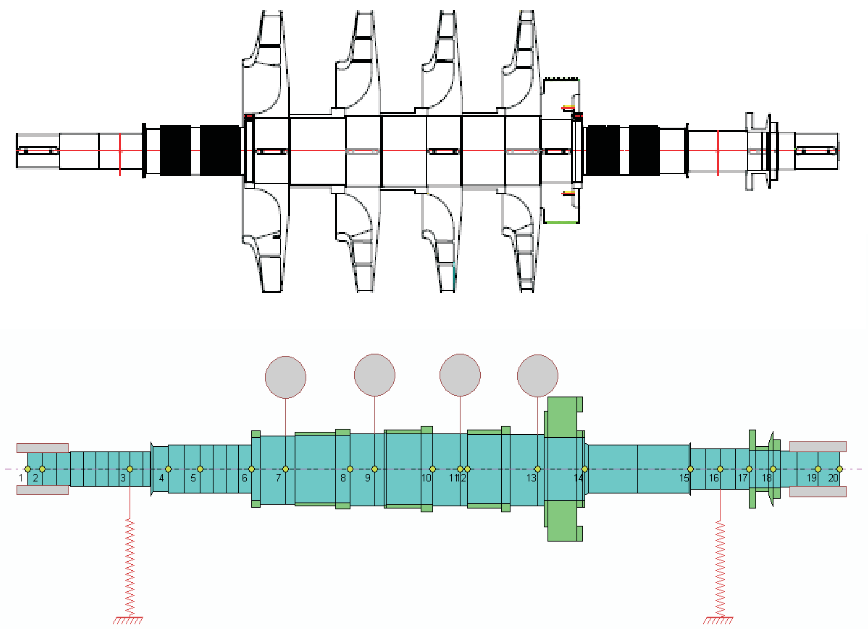

Figure 4 Design and a theoretical model of the compressor rotor

Fig. 4 shows a theoretical model of the compressor rotor used for rotor–dynamic calculations along with the modernized rotor design realization. Tab. 2 presents a modernized rotor equivalent rigid body properties connected to the important change in the impeller design of the C2102 compressor after modernization.

Table 2 Modernized rotor equivalent rigid body properties

rotor length [mm]	rotor mass [kg]	CM location [mm]	diametral inertia [kgm^2]	polar inertia [kgm^2]
3298	2506.8	1667	1061.0	157.82

In Fig. 5, theoretical results of the stability analysis of the modernized machine, including the hydrodynamic elliptical bearing operation, are shown.

The theoretical results presented in Fig.5 show the stable operation of the machine supported in the applied elliptical hydrodynamic bearing system, in the whole range of operational speeds.

The only way to verify the reliability of the presented concept of the modernization was to retrofit the real machine according to the design idea presented in this study.

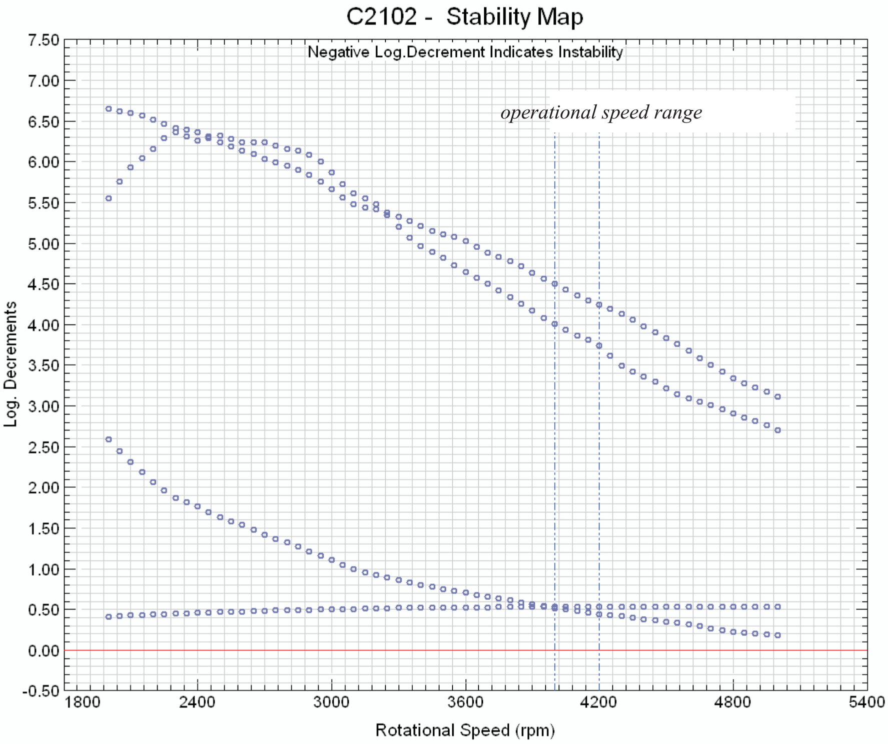

Figure 5 Stability map of the modernized compressor rotating system

6. Conclusion

The modernization of a nitrous gas compressor discussed in the present paper is a typical example of rerating undertaken recently in Poland, see [2, 5]. Typical features of such rerating consist in:

- the scope of work is limited only to an exchange of the rotor into a new one (which potentially enables a further assembly of the old rotor),
- the design of new impellers is based only on the calculated results (3D in general, sometimes – as in the case discussed here – 1D),
- elements from the retrofit scope are introduced (exchange of bearings and sealings).

In many cases of industrial compressor modernizations, the unsatisfactory results are caused by a wrong design of dynamical properties of the rotating system (i.e., operation in the vicinity of the resonance). Modern calculation methods employed in the design of components like bearings and seals, as well as rotor geometry modifications can change effectively machine disadvantageous dynamic characteristics.

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