GAS DYNAMIC DESIGN OF THE PIPE LINE COMPRESSOR WITH 90% EFFICIENCY. MODEL TEST APPROVAL

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Abstract: Gas dynamic design of the pipe line compressor 32 MW was made for PAO SMPO (Sunny, Ukraine). The technical specification requires compressor efficiency of 90%. The customer offered favorable scheme - single-stage design with console impeller and axial inlet. The authors used the standard optimization methodology of 2D impellers. The original methodology of internal scroll profiling was used to minimize efficiency losses. Radically improved 5th version of the Universal modeling method computer programs was used for precise calculation of expected performances. The customer fulfilled model tests in a 1:2 scale. Tests confirmed the calculated parameters at the design point (maximum efficiency of 90%) and in the whole range of flow rates. As far as the authors know none of compressors have achieved such efficiency. The principles and methods of gas-dynamic design are presented below. The data of the 32 MW compressor presented by the customer in their report at the 16th International Compressor conference (September 2014, Saint-Petersburg) and later transferred to the authors.

Nomenclature

\[ A_0 \] - ratio of an impeller inlet diameter and diameter that corresponds to minimum of inlet velocity;
\[ c_2 \] - absolute velocity at an impeller exit;
\[ c_3 \] - flow velocity at vaneless diffuser exit;
\[ c_m \] - meridian velocity;
\[ c_{m1} \] - meridian velocity at a blade inlet;
\[ D_1 \] - blade inlet diameter;
\[ D_2 \] - impeller diameter;
\[ D_3 \] - vaneless diffuser exit diameter;
\[ D_{shaft} \] - shaft diameter of an impeller;
\[ H_T \] - Euler work of a compressor;
\[ h_T \] - Euler work of an impeller;
\[ K_F \] - ratio of area at blade inlet and an impeller inlet;
\[ K_n \] - specific speed;
\[ \tilde{l} \] - meridian relative length of an impeller blade channel;
\[ \tilde{m} \] - mass flow rate;
\[ M_u \] - impeller Mach number;
\[ n \] - RPM;
\( N_{stage} \) - shaft power of a stage;
\( \bar{V}_0 \) - volumetric flow rate at a stage inlet;
\( u_2 \) - blade velocity;
\( w_i \) - relative velocity at an impeller inlet;
\( z \) - number of stages;
\( \alpha_2 \) - flow angle at an impeller exit;
\( \Phi \) - flow rate coefficient;
\( \Phi_{des} \) - design flow rate coefficient;
\( \phi_0 \) - flow rate coefficient at an impeller inlet;
\( \phi_1 \) - flow rate coefficient at a blade inlet;
\( \phi_2 \) - flow rate coefficient at an impeller exit;
\( \eta_p \) - polytrophic efficiency;
\( \Delta \eta_{imp} \) - loss of efficiency in an impeller;
\( \Delta \eta_{stator} \) - loss of efficiency in a stator part of a stage;
\( \rho_0 \) - flow density at an impeller inlet;
\( \zeta_{imp} \) - loss coefficient of an impeller;
\( \zeta_{stator} \) - loss coefficient of a stator part of a stage;
\( \psi_p \) - polytrophic head coefficient;
\( \psi_T \) - loading factor;
\( \psi_{T_{des}} \) - design loading parameter.

1. OPTIMAL NON-DIMENSIONAL PARAMETERS

Pipe line centrifugal compressors consume huge amounts of energy. The highest possible efficiency of these machines is important economically and ecologically. A proper gas dynamic design is the first necessary step to the goal. There are several considerations that must be taken into account.

The general rule is that the specific speed of compressor stages must correspond to the optimum flow path dimensions. A specific speed of a compressor is [1]:

\[
K_n = 3,545 \frac{\bar{V}_0^{0.5} n}{H_T^{0.5}} \frac{60}{z}
\]

In most pipe line compressors all impellers are of the same diameter and design loading parameter. In this case the specific speed of a stage is:

\[
K_s = 3,545 \frac{\bar{V}_0^{0.5} n}{(H_T / z)^{0.5}} \frac{60}{z} = \Phi_s^{0.5} \]

The most important design parameter of a stage is a flow rate coefficient at a design flow rate \( \Phi_{des} \). The bigger the coefficient, the smaller the radial dimensions of a compressor are. The maximum efficiency of stages with 2D impellers is in a range of \( \Phi_{des} = 0.06 – 0.08 \). If \( \Phi_{des} < 0.06 \), the surface
friction losses in a flow path channels and losses in labyrinth seal and disc friction losses prevail. If \( \Phi_{\text{des}} > 0.08 \) the high level of a flow kinetic energy at the impeller inlet diminishes efficiency. The eq. (2) demonstrates non dimensional kinetic energy \( 0.5(w_1/u)^2 \) influence on the efficiency loss in an impeller:

\[
\Delta \eta_{\text{imp}} = \frac{\zeta_{\text{imp}}}{\psi_T} \left( \frac{w_1}{u} \right)^2
\]

The efficiency of stages with 3D impellers can be high at \( \Phi_{\text{des}} = 0.11 \sim 0.12 \) due to better organization of a flow in wide blade channels.

The second important design coefficient \( \psi_{T,\text{des}} \) is present at the eq. (3) too. Formally the bigger \( \psi_T \), the higher the efficiency. Its factual influence is quite opposite. The impeller loss coefficient \( \zeta_{\text{imp}} \) is much greater for impellers with high design loading parameters. Losses in stator parts are bigger too, as the bigger is \( \psi_T \) the bigger is non dimensional kinetic energy after an impeller:

\[
\Delta \eta_{\text{stator}} = \frac{\zeta_{\text{stator}}}{\psi_T} \left( \frac{c_2}{u} \right)^2 = 0.5\zeta_{\text{stator}}\psi_T \left( 1 + tg^2\alpha_z \right)
\]

The authors design practice demonstrates highest efficiency of stages with low values of \( \psi_{T,\text{des}} = 0.4 \sim 0.5 \).

It is important for GT-driven compressors to have maximum of consumed power \( N_i \) at a design flow rate. The power of a stage is proportional to two design coefficients product:

\[
N_{i,\text{stage}} \approx \bar{m} \cdot h_T = \rho_0 \cdot 0.785 D_2^2 u_2^3 \cdot \Phi \cdot \psi_T
\]

The maximum of \( N_{i,\text{stage}} \) at a design flow rate takes place if \( \psi_{T,\text{des}} \approx 0.5 \).

One more advantage of low Euler coefficients is steeper loading parameter of performance \( \psi_T = f(\Phi) \). The steeper the performance is, the smaller the flow rate coefficient is with highest delivery pressure that is a surge limit.

There are two most important constructive limitations that influence stage efficiency. The bigger is a hub ratio \( D_{\text{shaft}} / D_2 \), the bigger is an inlet blade diameter ratio \( D_i / D_2 \). It increases flow kinetic energy at an impeller inlet and diminishes efficiency of an impeller – eq. (3):

\[
w_1 / u_2 = \sqrt{\left( D_i / D_2 \right)^2 + c_{a1}^2}
\]

There is no problem to achieve necessary flow deceleration in a vane diffuser with limited radial dimension. Flow deceleration in vaneless diffuser is opposite to its radial length:

\[
c_3 / c_2 = 1 / \left( D_3 / D_2 \right)
\]

To obtain good efficiency the ratio \( D_3 / D_2 \) must be not less than 1.60 and better about 1.70.
2. ADVANTAGES OF THE FLOW PATH SCHEME PROPOSED BY THE CUSTOMER. DESIGN MAIN FEATURES

The pressure ratio of the designed compressor is comparatively low and RPM of GT drive is high. It was possible to realize a single stage solution for this compressor. The necessary compressibility criteria $M_u \approx 0.7$ is not too high.

Advantages of the single stage design:
- console disposition of the impeller with zero hub ratio make possible to diminish flow kinetic energy at the impeller inlet,
- an axial inlet nozzle with minimal head losses can be applied.

The vaneless diffuser radial length of the compressor is close to optimum. The exit internal scroll was designed by especially formulated method that appeared to be rather effective.

Specific speed of the compressor – eq. (1), (2) - is quite good to apply traditional 2D impeller with $\Phi_{des} \approx 0.060$ and $\psi_{R,des} \approx 0.52$. The new 5th version of the Universal modeling method computer program was applied to optimize the flow path and to calculate the expected performances. Basic information about the Method was presented at the conferences earlier [2, 3, 4, 5]. The new 5th version with better modeling abilities was presented at [6, 7, 8].

The designed impeller has no hub at all. The impeller inlet diameter is much smaller than ones of the tested analogs. Some geometry and flow parameters of the closest analog type 055 (upper line) and of designed impeller variants are presented in Table 1.

| Variants | $K_F$ | $A\_D$ | $D_1/D_2$ | $\varphi_0$ | $\varphi_1$ | $w_2^\prime/w_1^\prime$ | $\varphi_2$ | $\alpha_2^\prime$ |
|----------|-------|--------|-----------|------------|------------|---------------------|------------|---------------|
| 055      | 0.825 | 1      | 0.592     | 0.300      | 0.285      | 0.729               | 0.250      | 24.9          |
| 1        | 0.80  | 1      | 0.435     | 0.318      | 0.302      | 0.941               | 0.153      | 15.0          |
| 2        | 0.825 | 1      | 0.445     | 0.316      | 0.309      | 0.921               | 0.153      | 15.0          |
| 3        | 0.85  | 1      | 0.444     | 0.306      | 0.308      | 0.924               | 0.153      | 15.0          |
| 4        | 0.90  | 1      | 0.452     | 0.295      | 0.313      | 0.908               | 0.153      | 15.0          |
| 5        | 0.95  | 1      | 0.460     | 0.285      | 0.318      | 0.892               | 0.153      | 15.0          |
| 6        | 0.825 | 0.975  | 0.436     | 0.328      | 0.323      | 0.920               | 0.153      | 15.0          |
| 7        | 0.825 | 1.025  | 0.458     | 0.299      | 0.296      | 0.915               | 0.153      | 15.0          |
| 8        | 0.825 | 1.05   | 0.469     | 0.284      | 0.282      | 0.911               | 0.153      | 15.0          |

The variants have different inlet dimensions. The other dimensions of the stage flow path - impeller and stator - were studied by the 5th generation program too. The final part of the impeller optimization is a qualitative one. The non viscid Q3D velocity diagrams were assessed by experts. The samples of the diagrams for the design flow rate are presented in Figure 1, 2.

The diagrams demonstrate several important features of the impeller design:
- non-incidence at inlet takes place along the leading edges height in spite of simple 2D blade configuration,
- flow deceleration along suction side is little. There is a hope that wake formation will not take place,
- velocity diagram area gives information about expected value of the Euler work coefficient.

3. MODEL TEST PROOF

The Universal modeling method has demonstrated reliability and the customers are realizing designs without any tests for two decades already. Because of new design features the customer of
this design has decided to check the project by tests of the model at the 1:2 scale. Figure 3 demonstrates the model and the part of the test rig cross section.

Figure 1. Velocity diagrams at three blade-to-blade surfaces (design flow rate).
Left – hub, middle mean, right – shroud

Figure 2. Meridian velocities along 7 blade to blade surfaces (design flow rate)
Figure 3. The model of the 32 MWt pipe line compressor at the test rig. Scale 1:2

The results of three tests in consequence are presented graphically in Figure 4.

![Graph](image)

Figure 4. Compressor 32 MWt model test results
1- design performance, 2- ANSYS CFX, 3- $M_u = 0.705$, 4- $M_u = 0.710$, 4- $M_u = 0.700$

4. CONCLUSION

The result of ANSYS CFX modeling by the customer is presented in Fig. 4. The performance curves shift right from factual but the maximum efficiency is predicted well. The authors experience with different CFD programs and different impellers demonstrate the same results usually.
Design and test data comparison demonstrate two remarkable results. The polytrophic coefficient calculated performance is in very good conformity with all three model tests. The previous 4th version is very effective to predict design flow rate efficiency and head but not all performance. Some non-conformity at flow rate left and right of the design point are usual for that model. Fig. 4 demonstrates better level of modeling by the new 5th version. The second important fact is that the optimistic design assessment of the 90% efficiency is proven with high probability.

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