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Experimental characteristic simulation for two-stage pipeline centrifugal compressor

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Abstract. Up to now, the design methods for centrifugal compressors have been based on results of physical experiments. The possibility of correct CFD calculation of centrifugal compressors gas-dynamic characteristics could fundamentally change the situation. Gas-dynamic projects will become more reliable, cheaper, and the performance of compressors will be improved. However, the problem has not been solved yet. The experience of various authors shows that CFD calculations in many cases do not align well with experimentally obtained data. The aim of this paper is to check whether the commercial CFD software is capable of simulating centrifugal compressor characteristics using a 16 MW pipeline two-stage compressor as an example. The calculated characteristics of pressure ratio, efficiency and head coefficient are compared with characteristics measured using the manufacturer test bench on air. The calculations are performed in two ways: in the first case the flow in the impeller-stator gaps is taken into consideration, and in the second one it is not. In the first case, we do not take into account friction losses of the outer surfaces of impeller shrouds and leakage through labyrinth seals. The interface “STAGE” averages the parameters of flow at the impeller outlet and vaneless diffuser inlet. The flow leaving the impeller is averaged in tangential direction. This way of calculation excludes the mixing process of high and low-energy flow zones at the exit of the impeller. The calculated characteristic configurations are similar to the experimental ones, but they are shifted to the area of higher flow rates, and the difference in pressure ratio is out of the limits acceptable for engineering and research. Calculations that take into consideration gap losses underestimate the maximum efficiency by up to 2.5%. The characteristics calculated without gap losses are closer to the experimental characteristics. However, it is hardly possible to use this circumstance as a recommendation for the simulation methodology. The problem needs further investigation.

1. The aim of research

The authors have a long list of successfully designed powerful process compressors [1], [2]. Field-type Universal modelling method [3], [4] based on test results of model stages, guarantees the accuracy of pressure ratio and efficiency characteristics $\pm 1.5\%$ within the main range of the compressor characteristics.



The possibility to perform a similar or more accurate CFD-calculation would be of high significance for gas dynamics design. Any field-type method would be non-reliable without the source test data, and mathematical models cannot take into account all the flow path details.

CFD-calculations for centrifugal compressors are applied by the Russian and foreign specialists for centrifugal compressor characteristic simulation [5]-[8]. The experience of various authors shows that CFD calculations in many cases do not align well with experimentally obtained data. The Author who has managed to achieve matching results has been using proprietary software [9]. In other papers with successful matches between calculated and experimental data, the solver settings are not listed at all [10].

The aim of this paper is to check whether the commercial CFD software is capable of simulating characteristics of a multi-stage centrifugal compressor at various solver parameters settings.

2. The experimental setup

The object of the study is the 16 MW pipeline compressor (shown in Figure 1), which has the following characteristics: 2 stages, vaneless diffusers, impeller diameters are 800 mm each, frequency is 5,200 rpm, delivery pressure is 7.45 MPa, the pressure ratio is 1.35.

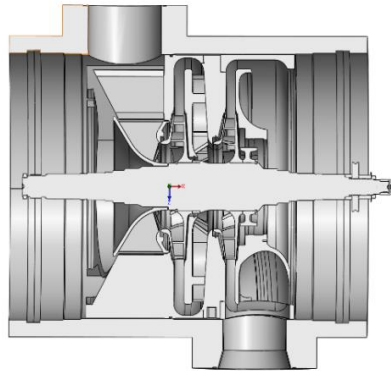


Figure 1. The cross-section of the 16 MW pipeline compressors

Figure 2 shows the flow path cross-section, solid models of inlet and exit nozzles.

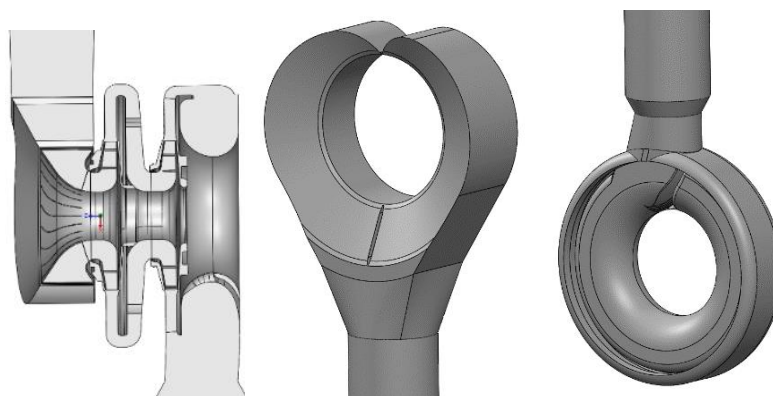


Figure 2. 16 MW pipeline compressor: the flow path cross-section (left), solid model of the inlet nozzle (centre), solid model of the exit nozzle (right)

The gas dynamic characteristics were measured during the course of unit tests on air. The measurements were repeated four times.

Tables 1-4 contain recalculated results of the unit tests (ψ_i, η, π characteristics as functions of \bar{V}_{inl}):

- \bar{V}_{inl} is the volume flow rate, m³/min;
- work coefficient:

$$\psi_i = \frac{T_{ex}^* / T_{inl}^*}{\frac{k}{k-1} R \cdot u_2^2}; \quad (1)$$

- polytropic efficiency:

$$\eta = \frac{\ln(p_{ex} / p_{inl})}{\frac{k}{k-1} \ln(T_{ex}^* / T_{inl}^*)}; \quad (2)$$

- pressure ratio:

$$\pi = p_{ex} / p_{inl}, \quad (3)$$

where: T is the temperature; k is the isentropic coefficient; R is the gas constant; u_2 is the blade speed, p is the pressure; * is for total parameters; subscript _{ex} denotes compressor exit; subscript _{inl} denotes compressor inlet.

Table 1. The results of the test No. 1

Volume flow rate, m ³ /min	435	364.2	306.5	237	181.8	129
Pressure ratio	1.25	1.36	1.43	1.46	1.46	1.45
Operation coefficient (two impellers)	0.74	0.95	1.12	1.26	1.36	1.58
Pol. efficiency	0.79	0.86	0.84	0.80	0.75	0.64

Table 2. The results of the test No. 2

Volume flow rate, m ³ /min	414.8	354.8	298.6	231.1	178.3	125.5
Pressure ratio	1.25	1.38	1.44	1.47	1.47	1.46
Operation coefficient (two impellers)	0.75	0.97	1.13	1.29	1.44	1.62
Pol. efficiency	0.77	0.86	0.85	0.79	0.71	0.63

Table 3. The results of the test No. 3

Volume flow rate, m ³ /min	408	397.3	387.3	362.3	340.9	316.6
Pressure ratio	1.25	1.28	1.30	1.35	1.39	1.41
Operation coefficient (two impellers)	0.75	0.79	0.85	0.93	1.02	1.07
Pol. efficiency	0.76	0.80	0.81	0.84	0.85	0.85

Table 4. The results of the test No. 4

Volume flow rate, m ³ /min	422.2	366.1	311.8	238.9	200.1	148.3
Pressure ratio	1.25	1.36	1.43	1.46	1.47	1.46
Operation coefficient (two impellers)	0.71	0.91	1.14	1.28	1.36	1.60
Pol. efficiency	0.81	0.88	0.84	0.79	0.74	0.65

Figure 3 graphically shows test results. Experimental points were approximated analytically. Efficiency value spread is mostly due to the inaccuracy of work input measured by total temperature rise in a compressor. However, the approximated characteristics are apparently reliable.

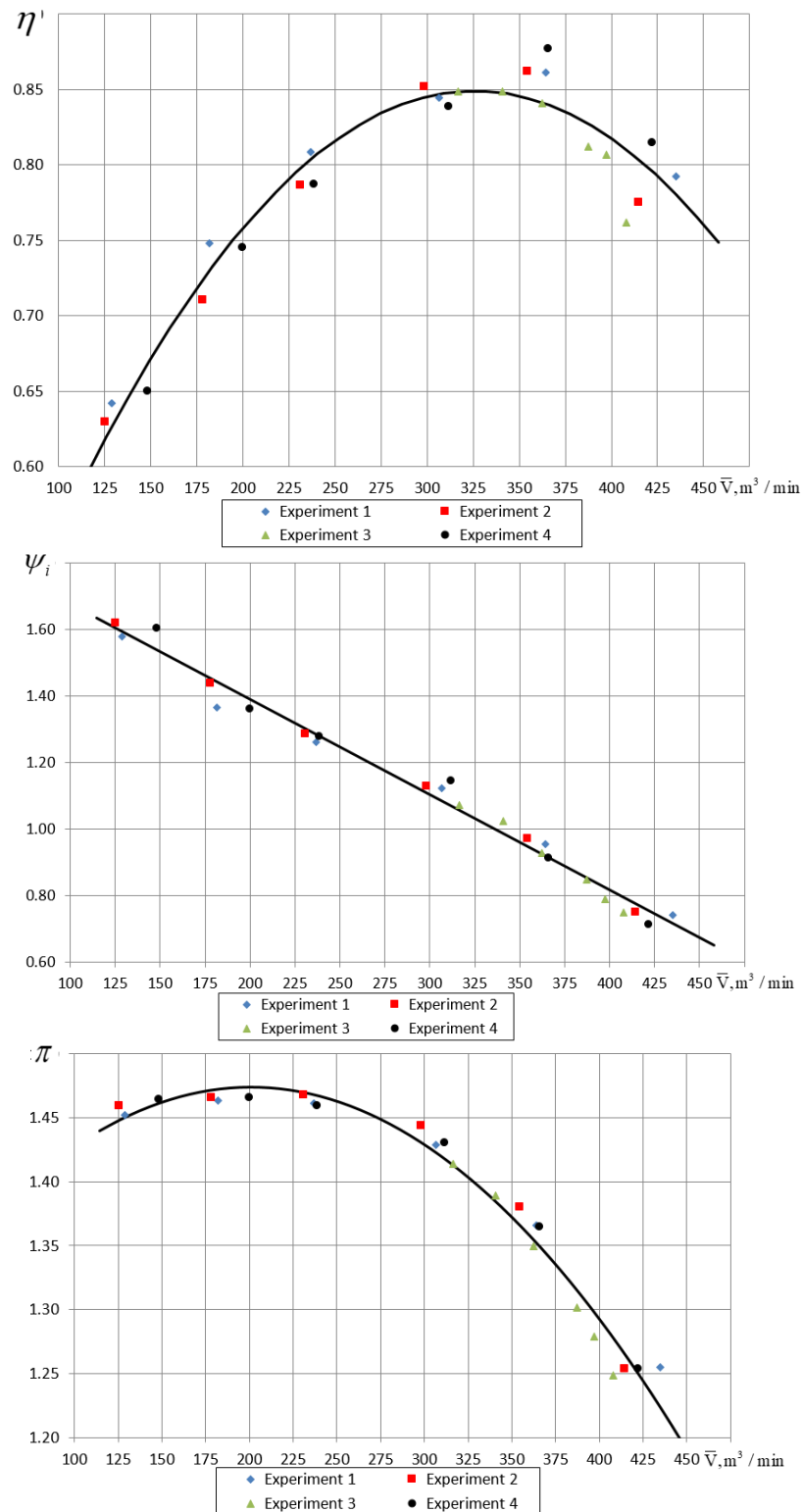


Figure 3. Approximation of test measurements.

3. CFD-calculation methodology

ANSYS/CFX program was used for calculations. Calculations were performed in steady mode with "STAGE" interface and flow averaging at impeller exit. SST turbulence model was selected for numerical modelling. The calculation includes the full flow path with all blade channels. It is necessary because of the circumferential asymmetry of the compressor flow path, namely, due to the presence of the inlet and the exit nozzles.

When using CFD calculations for research purposes, the flow in the impeller-stator gaps is often not simulated [8]. In this case, friction of shroud outer surfaces is neglected. Shroud friction transmits additional mechanical power to gas. Some amount of the gas passes through shroud and shaft labyrinth seals. Leakages in labyrinth seals and shroud friction reduce both efficiency and flow rate. Nevertheless, papers [11]-[12] present calculations in a simplified mode that show a good simulation of a stage maximum efficiency. On the other hand, the author [9] insists that the flow should be calculated in two gaps to perform correct parameter simulation.

Structured scheme with hexagonal elements for impellers, diffuser and return channel, as well as non-structured hybrid grid for the inlet and the exit nozzles were used for computation models. The first wall cell is sized from 10 to 50 micron. Dimensional factor is 1.2-1.5. The total number of computational cells amounted to 39,200,000. Figure 4 shows the computational grid of the compressor flow path.

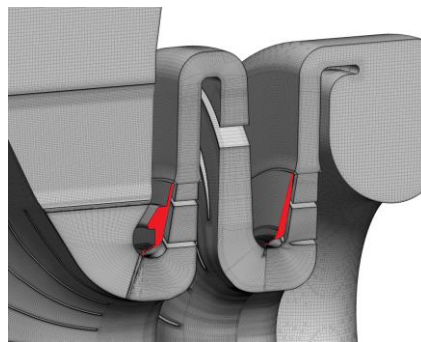


Figure 4. Section of the modelled compressor computational grid including labyrinth seals.

Elements indicating gaps and labyrinth seals (red in Figure 4) add 15 million cells to the computational amount. However, the gap modelling is not complete. The gaps between the main disk and its housing were not included.

4. Flow structure specifics

The most important flow specifics are linked to circumferential asymmetry of the compressor flow path. Pictured on the Figure 5 is an average cross-section of the flow structure corresponding to the maximum efficiency achieved at 306.5 m³/min.

A large low-energy zone exists in the toroidal collector. A scroll is a better solution for a design flow rate. However, there is an opinion that the toroid creates a lesser radial force applied to rotor at off-design flow rates. It is important for the high-pressure compressors. The flow separation occurs at the exit of the toroidal collector. The separation zone propagates into the exit pipe. Such gas dynamic imperfection is inevitable. This configuration of the exit nozzle is necessary to properly connect the compressor to a pipeline.

One of the recommendations listed in [13] is to orient the return channel vanes at +4° incidence angle at the design flow rate. Figure 6 shows separation zones appearing at the design flow rate due to the return channel vanes' orientation, which may lead to a decrease in efficiency.

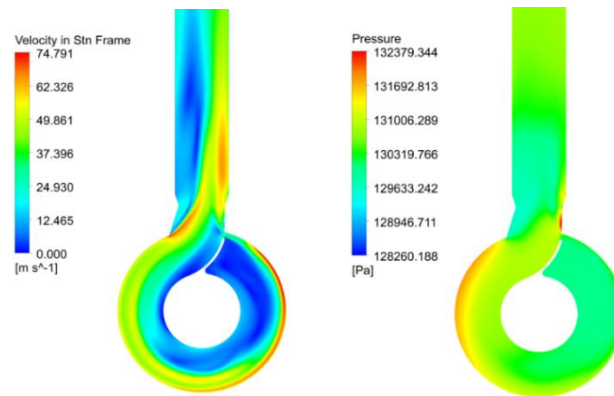


Figure 5. Velocity field (left) and static pressures (right) in the exit nozzle

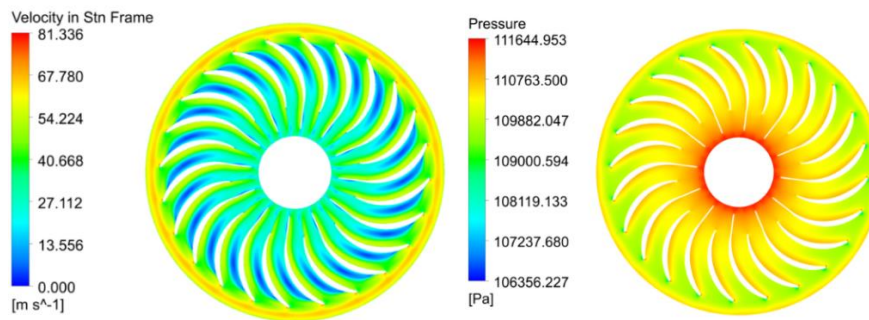


Figure 6. Velocity field (left) and static pressures (right) in the return channel vane cascade

5. Results and comparison of both types of the gas dynamic modelling

Figure 7 presents the calculated efficiency characteristic with and without flow in impeller-stator gaps. The hydraulic efficiency without losses in gaps is η_{hud} . Polytrophic efficiency η has to be less than hydraulic η_{hud} , because it does not consider shroud friction losses and leakages through labyrinth seals.

In engineering practice for gap losses (“parasitic losses”) evaluation coefficients of disk friction β_{fr} and leakage β_{lk} are applied:

$$\eta = \eta_{\text{hud}} / (1 + \beta_{\text{fr}} + \beta_{\text{lk}}). \quad (4)$$

Empirical formulas to calculate loss coefficients $\beta_{\text{fr}} + \beta_{\text{lk}}$ are applied in engineering calculations. For the impeller-stator gap the authors apply the formula:

$$\beta_{\text{fr}} + \beta_{\text{lk}} = \frac{0,0004}{\Phi \cdot \psi_T}, \quad (5)$$

where Φ is flow rate coefficient, ψ_T is loading factor.

The authors cannot explain the peculiarity of the calculated and measured (Figure 3) points of efficiencies at flow rate 364 m³/min. The tested efficiency and both CFD-calculated efficiencies lie sufficiently higher than approximation curves.

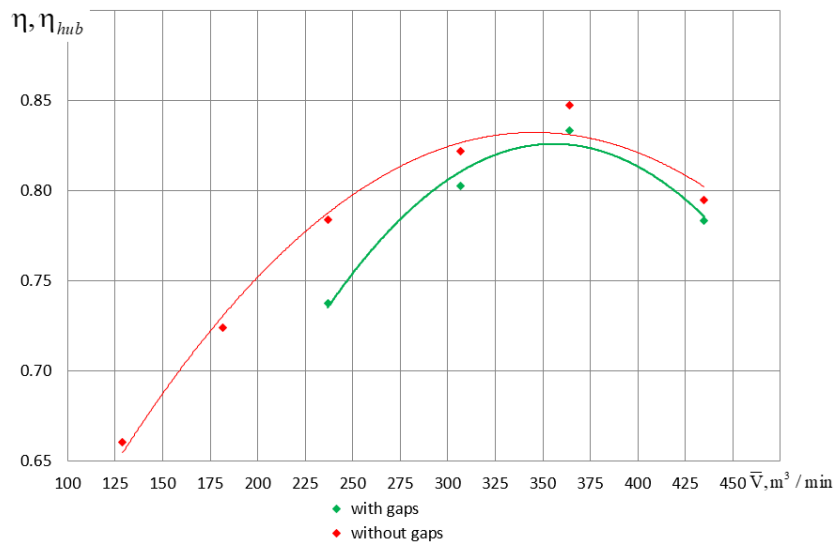


Figure 7. Calculated efficiency characteristics

The pressure ratio characteristics calculated using both ways are shown in Figure 8.

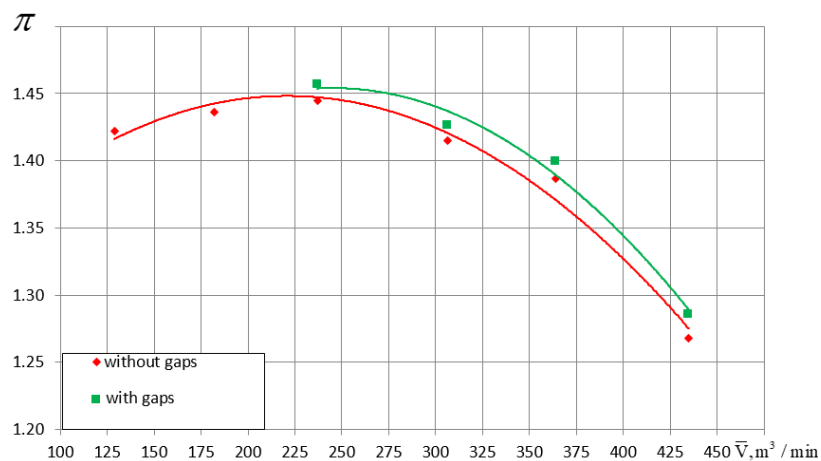


Figure 8. Calculated pressure ratio characteristics

One-dimensional analysis predicts an opposite influence of gaps on pressure rate characteristics. Due to labyrinth leakage, flow rate of the impeller \bar{m}_{imp} is higher than flow rate of a compressor \bar{m} :

$$\bar{m}_{imp} = \bar{m} + \bar{m}_{lk} = \bar{m}(1 + \beta_{lk}). \quad (6)$$

Leakage flow circulating in the impeller diminishes flow rate of the stage. On the contrary, the calculation with gaps shifts the characteristic to higher flow rate.

6. Comparison of measured and calculated parameters

Figure 9 presents information on calculated and measured work input. The convergence condition was set as follows: the deviation of the outlet temperature must not exceed 0.5% of its average value.

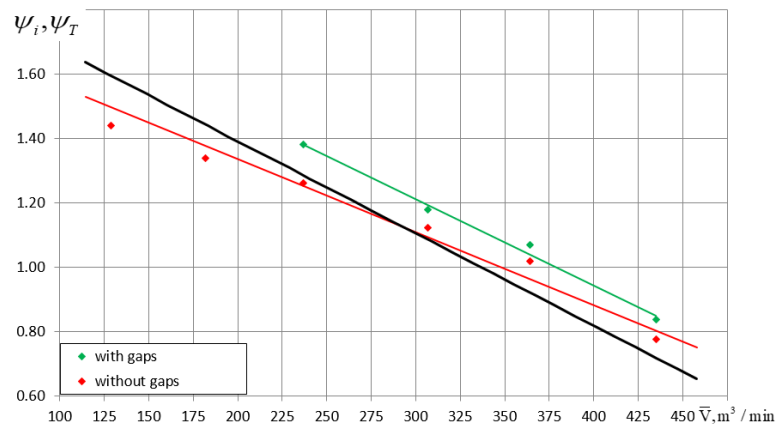


Figure 9. Work coefficient and loading factor (two impellers' coefficients)

The linearity of obtained characteristics agrees with experimental results listed in [14]. The calculated work coefficient characteristic is naturally located above the calculated loading factor characteristic due to parasitic losses. In accordance with 1-D theory, operation coefficient and loading factor are connected by the formula:

$$\psi_i = \psi_T (1 + \beta_{fr} + \beta_{lk}). \quad (7)$$

In accordance with formula (5) at the flow rate equal to 316.5 m³/min, parasitic losses coefficients are $\beta_{fr} + \beta_{lk} = 0.016$. The difference between calculated values is $\psi_i - \psi_T = 0.09$. It is 5.6 times higher than that obtained from 1-D theory. Evidently, the CFD-calculations demonstrate more deep and sufficient influence of flow in gaps on characteristics in comparison with that predicted by one-dimensional theory.

Measured and calculated work coefficient characteristics are parallel, but the difference in ψ_i values is drastic: 8 – 9%. The error of simulation is unacceptable and quite typical for CFD-calculations. In paper [6], a comparison of the calculated and experimental pressure characteristics shows a similar result.

The graph of the work coefficient calculated without gaps has smaller angle of inclination than that of the experimentally obtained characteristic.

Figure 10 presents efficiency characteristics. Simplified calculation (no gaps) better simulates the measured efficiency characteristic. However, this comparison is not correct, as in any case $\eta \neq \eta_{hud}$. Maximal calculated efficiency η is 2.5% lower than the measured maximal efficiency. The calculated characteristic η shifts to higher flow rates.

In paper [8] the calculation of intermediate-type stages also showed a shift of the calculated characteristics to higher flow rates. On the other hand, the measured efficiency was lower than the calculated one. Similar results are presented in the paper [15].

Figure 11 shows that the measured pressure ratio significantly exceeds the calculated one in the practically important range of flow rates, i.e. to the left from the optimum flow rate of 306.5 m³/min.

Pressure ratio calculated taking into consideration the impeller-stator gaps is shifted to the right. This resulted from the overestimated head coefficient value. The calculated blade load (pressure difference between the pressure and suction sides of blades) is higher than the actual load. This causes increased flow transformation at the blade cascade inlet. This transformation shifts characteristic to the higher flow rate [13]. Despite the higher calculated head coefficient, lower efficiency makes the maximum calculated pressure ratio much less than the measured one.

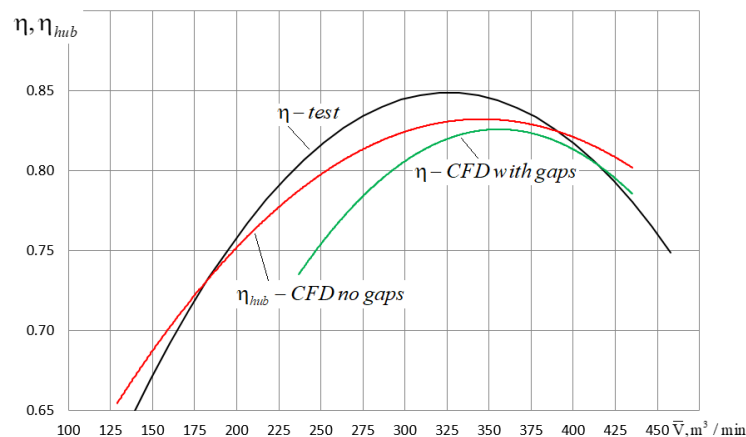


Figure 10. Efficiency characteristics

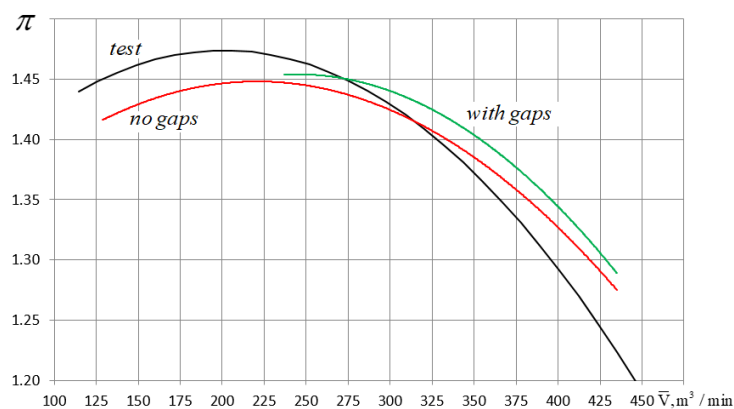


Figure 11. Pressure ratio characteristics

7. CONCLUSION

The experience gained by the authors and their colleagues demonstrates the validity of CFD-calculations usage for analysis of centrifugal compressor stator parts and formulation of design recommendations [7], [11].

The author of [9] presents results of an excellent simulation of centrifugal compressor stages' characteristics. The problem is that he has applied his own CFD program that is not available to general users. The description of the program code and approaches to solving the Navier-Stokes equations do not allow one to judge how those results were obtained. The SST turbulence model used in the TACOMA program was used by the authors of [6] when calculating with ANSYS CFX and showed poor results.

The majority of authors report on unacceptable mismatch of centrifugal compressor characteristics calculated by commercial CFD programs [7], [16]. The presented calculations also demonstrate unsatisfactory results of CFD-simulation of the centrifugal compressor characteristics. This fact is important for future investigation: flow in the impeller-stator gaps influences general flow to the greater extent that it is predicted by 1-D theory. Simplified calculations of flow paths without impeller-stator gaps are impractical.

Empirical coefficients in turbulence models should be further investigated. The authors state that achieving a match between head characteristics is necessary. The higher the loading factor is, the higher is the blade load and the bigger the flow deviation at the leading edge. If we learn how to calculate the head characteristic correctly, the range of efficiency characteristic will become closer to the actual range.

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