The flow research in vane diffusers of centrifugal compressors transonic stages

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Abstract. Abstract. Computational fluid and gas dynamics methods are used at all stages of design, including blade machine design: from the creation of a schematic design to a detailed study of all the main units [1, 2]. The industries using refrigeration centrifugal compressors (oil refining, chemical, food) development requires the machines unit capacity, efficiency and reliability increase, the weight and size characteristics and metal consumption decrease. Since this machine type production often has a small-scale or individual character, and the needs for machine specified characteristics maintain during the entire service life exists, design remains an important stage in the creation process. This article is devoted to the possibility of validating the calculations using the numerical gas dynamics methods research. The centrifugal compressor end stage numerical and a full-scale research were made during the investigation; refrigeration centrifugal compressor end stage full-scale investigation carried out by D.A. Kapelkin was chosen as a base of the research [3]. As a result, the diffuser flow numerical simulation results were analyzed and the calculated and full-scale characteristics comparison was carried out. Recommendations for the numerical gas dynamics methods application are given in a particular case, applied to refrigeration centrifugal compressors vane diffusers gas-dynamic calculations.

Keywords: vane diffuser, computational fluid dynamics, refrigerant compressor, centrifugal compressor.

1 Introduction

There are many centrifugal compressor flow path numerical investigations in general and vane diffusers research in particular [4-6]. However, all investigations have their

own specifics and contribute to the common view of the gas flow mathematical modeling in the compressor flow path process [7-12].

If there is the ideal gas calculations validation data for high-pressure stages diffusers, it's impossible to talk about the results reliability of the real gas numerical experiment carried out on the same stage, it's impossible to project one research results onto another due to the mathematical modeling methods high sensitivity to input data.

The need for similar specific investigations for each centrifugal compressor flow path specific section exists. This is true not only for the compressor industry, but for mechanical engineering in general. This determines the work relevance.

The work purpose is to prove or disprove the refrigeration machines vane diffusers design numerical methods usefulness and reliability at high M_{u2}.

To achieve the purpose, the following tasks were set:

1. The full-scale bench research results processing.

2. The gas flow simulation using numerical gas dynamics methods. Justification for the certain parameters choice.

3. The results processing.

4. The numerical and full-scale research results comparative analysis.

5. The numerical gas dynamics methods usage recommendations development in relation to the refrigeration centrifugal compressors vane diffusers gas-dynamic calculation.

2 Methods

A numerical experiment was used as the main research method in this work. The Ansys CFX software package as a platform for the flow mathematical modeling was chosen [13-15].

The numerical simulation was based on a field experiment described in [3]. The centrifugal compressors end stages vane diffusers experimental study was carried out on a specific test bench.

2.1 The vane diffuser full-scale investigation.

The experimental study objects were the variants of the end centrifugal stage model operating in a relatively wide range of M_{u2} numbers. The modification of the model version was carried out by changing the impellers with exit angles $\beta_{l2} = 22\degree\,30\,', 45\degree, 90\degree$ and changing the diffuser geometric parameters. The gas used in the research is R-12.

R-12 is a real gas, which state parameters do not obey the ideal gas state equation; this is the reason for the analytical apparatus choice [16].

The flow path schematic diagram corresponded to one of the schemes used for the experimental centrifugal stages of the CJSC "Nevsky zavod", the sketch is presented in Fig. 1. The stage model was made with an axial gas supply to the impeller.
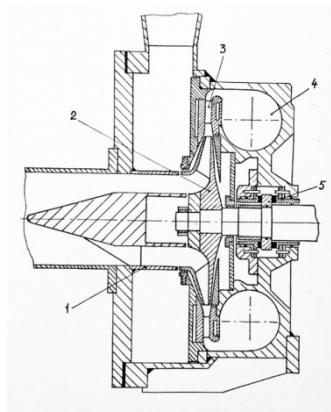
Fig. 1. The end centrifugal stage experimental model flow path sketch.
1 - inlet pipe
2 - impeller
3 - vane diffuser
4 - annular chamber
5 - double-sided mechanical seal

The 23 flow path modification were tested for the stages with vane diffusers. The experiment was carried out at $M_{u2} = 0.81; 1.015; 1.215; 1.42$. The M_{u2} constancy was maintained by the water amount varying supplied to the gas cooler.

The Freon-12 superheated steam empirical equation of state proposed by N. Baer and E. Hicken, was used as a basis for the working fluid thermodynamic parameters connection.

$$p = \frac{RT}{v} + \frac{B_0 - B_1/T}{v^2}, \tag{1}$$
$$R = 68.7563 \frac{J}{kg * K};$$

$$B_0 = 24 \frac{n}{m^2}\left(\frac{m^3}{kg}\right)^2; \tag{2}$$

$$B_1 = 298 * 10^2 K \frac{n}{m^3}\left(\frac{m^3}{kg}\right)^2. \tag{3}$$

A number of the following assumptions were made in gas-dynamic calculations:
1. The stream is one-dimensional.

2. There is no heat exchange with the external environment.

3. Kinetic energy is determined from the particular section average flow rate.

4. The compression process in the stages and in the flow path elements is accepted to proceed along the polytrope of index n.

The loss coefficients ζ, the recovery coefficients ξ, the stage efficiency decrease magnitude due to losses in this element Δη were used for the flow path fixed elements efficiency rating [17, 18].

Parameters characterizing the flow path elements operating mode:

- The flow angle at the entrance or exit from the step section;

- The attack angle i_3 for blade cages;

- Diffusion coefficient of the oblique cut;

- Mach number in the element enter or exit at the corresponding characteristic speed.

The full-scale experiment results process is completed.

2.2 The vane diffuser numerical investigation

In the numerical experiment the impeller angle is $\beta_{l2} = 45°$. The vane diffusers angles $\alpha_{l3} = 11°; 20°; 23°$ were selected at the numbers $M_{u2} = 1.015; 1.215$, the angles $\alpha_{l3} = 14°; 17°$ were selected at the numbers $M_{u2} = 0.81; 1.015; 1.215; 1.42$. In total, 5 flow paths variants with different vane diffusers geometry were tested. In a full-scale experiment there were 26 variants, including vaneless diffusers variants and variants with different impellers.

The simplifications been established in the numerical experiment:

1 The working fluid temperature at the stage entrance is constant $T_n^* = 293°K$. The temperature was controlled by the water amount supplied to the gas cooler and was conditionally constant $T_n^* = 290 \div 296°K$ in the full-scale experiment.

2. The working fluid pressure at the stage entrance is constant $p_n^* = 6.8 * 10^4 \, Pa$, this is the average pressure according to the calculation results. The pressure fluctuated within $p_n^* = 6 \div 7 * 10^4 \, Pa$ in the full-scale experiment.

3. Compressor shaft rotation frequency are constant for each M_{u2} number.

4. The numerical model is simplified, the model does not have labyrinth seals, inlet and outlet pipes.

3 Results and Discussions

The vane diffusers gas-dynamic characteristics obtained in the numerical and full-scale experiments comparative analysis was made as a result of the work.

3.1 The loss ζ dependences on the flow path elements operating mode characteristic parameters (a_2, i_3)

Plots:

2-3 - for $\alpha_{l3} = 17°, M_{u2} = 0.81 \div 1.42$;

2-4 - for $\alpha_{l3} = 17°$, $M_{u2} = 0.81 \div 1.42$;
2-4 - for $\alpha_{l3} = 14 \div 23°$, $M_{u2} = 1.215$.

It is important to include the vaneless section of the vane diffuser, as there's always some vaneless annular space before the vane diffuser, which size depends on the stage design features, in the comparative analysis [19, 20].

The working fluid flow nature through the diffuser vaneless section is very complicated, especially at high M_{c2} numbers, and has a high flow irregularity degree. It is determined by the preceding it impeller design solutions and the behind located vane diffuser [21, 22, 23, 24, 25].

The full-scale and numerical experiments results for the $\alpha_{l3} = 17°$ are presented in fig. 2 and fig. 3.There is a smaller points scatter range and a smoother dependencies character observed in case of numerical experiment. This may be due to the loss coefficients determine method by measuring static pressure on the walls in a full-scale experiment is not sufficiently accurate due to the ζ_{2-3} relatively high sensitivity to errors in the "2-2" and "3-3" sections kinetic energy average values calculating.
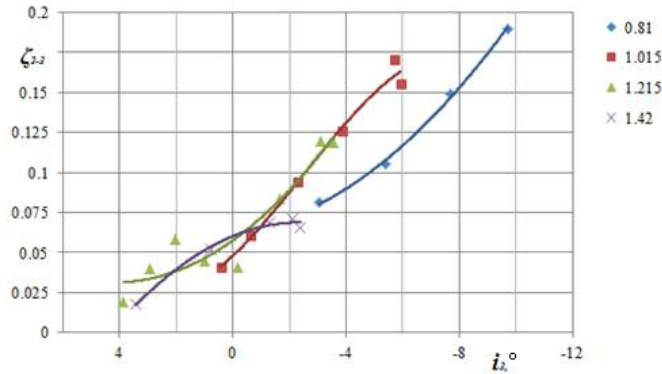

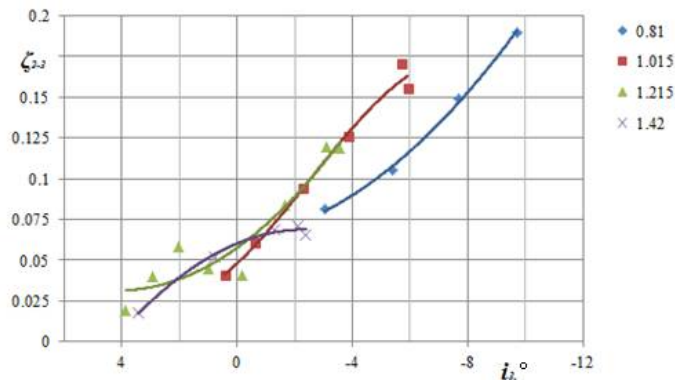Fig. 2. The full-scale experiment results, $\alpha_{l3} = 17°$, $M_{u2} = 0.81 \div 1.42$.

Fig. 3. The numerical experiment results, $\alpha_{l3} = 17°$, $M_{u2} = 0.81 \div 1.42$.

The full scale and numerical experiments results approximation comparison for all M_{u2} is presented in fig. 4. The dependences have a similar dynamic character and numerical values in the section $\zeta_{2-3} = 0.09 \div 0.18$, however, the i_3 angle values a little different. This may be due to calculation methods differences and the recalculated values large amount for a numerical experiment.
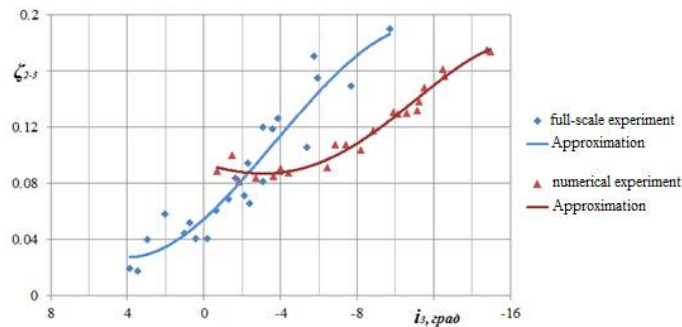

Fig. 4. The full-scale and numerical experiments results comparison for all M_{u2}.

The full-scale and numerical experiments results comparison for different diffuser vanes angles α_{l3} and $M_{u2} = 1.215$ is presented at in fig. 5. The trends coincidence is also traced with the α_2 angle lag for the numerical experiment. It can be noted that the characteristics smoothness for a full-scale experiment is higher than for a numerical one.
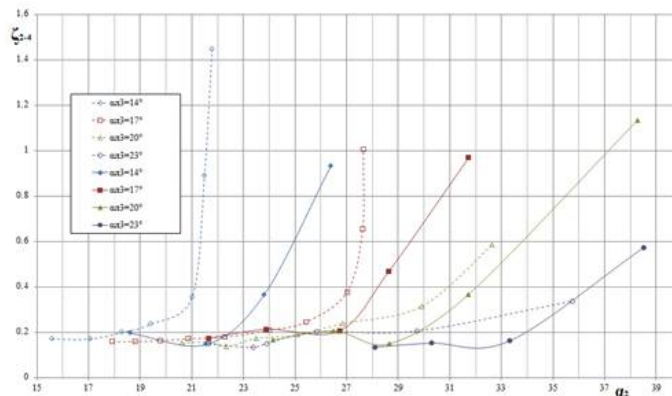

Fig. 5. The full-scale and numerical experiment results comparison. Loss in section 2-4 for $M_{u2} = 1.215$ and $\alpha_{l3} = 14 \div 23°$.

The discrepancy between the variant points is explained by the multiple value recalculation and the additional variants inclusion in the calculation as additional variants with different mass flow rate were added. The best connection between natural and numerical experiments is traced for the angles $\alpha_{l3} = 17 \div 23°$.

The full-scale and numerical experiments loss comparison in section 2-4 for $\alpha_{l3} = 17°$ and the range of numbers M_{u2} is presented in fig. 6. The general trends coincide, but the values differ in α_2 and loss in this case. Moreover, the losses in section 2-4 are higher for a numerical experiment than for a full-scale one on the average. This can be explained by the numerical calculation algorithm features.
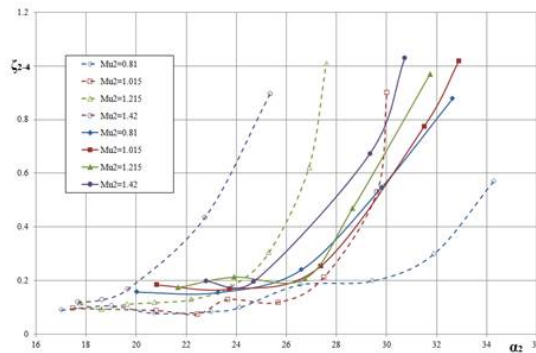

Fig. 6. The full-scale and numerical experiment results comparison. Loss in section 2-4 for $M_{u2} = 0.81 \div 1.42$ and $\alpha_{l3} = 17°$.

The dependences sequence in M_{u2} is generally preserved, however, the numerical experiment is characterized by a significantly smaller α_2 scatter range for the same loss value for different M_{u2}. You can see that the highest quality dependencies coincidence is observed for $M_{u2} = 1.015$, this indicates that the settings were accepted are optimal just for such M_{u2} numbers.

3.2 The refrigerant stage diffusers flow numerical study results.

The flow rates in the vane diffuser for different angles α_{l3}, $M_{u2} = 1.015$ and $M_{u2} = 1.215$, are presented in fig. 7, a,b. Fig. 7a: Mass flow rates are chosen so they are approximately the same $m \cong 1.5\ kg/s$. On average, the difference is 2.5%, but its values are higher and can reach 5% at the extreme angles. Fig. 7b: $m \cong 2.1\ kg/s$. On average, the difference is 2%, but its values are higher and can reach 10% at the extreme angles.

а) $\alpha_{l3}=11°$
б) $\alpha_{l3}=14°$
в) $\alpha_{l3}=17°$
г) $\alpha_{l3}=20°$
д) $\alpha_{l3}=23°$

а) $\alpha_{l3}=11°$
б) $\alpha_{l3}=14°$
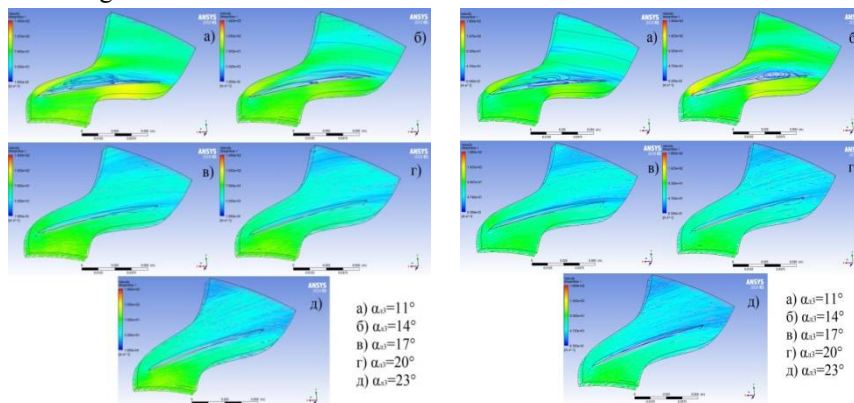в) $\alpha_{l3}=17°$
г) $\alpha_{l3}=20°$
д) $\alpha_{l3}=23°$

Fig. 7, a,b. Velocity distribution in the vane diffuser for variants with $\alpha_{l3} = 11 \div 23°$, $M_{u2} = 1.015$, $m \cong 1.5 \, kg/s$ the speed scale ranges from $1/10^{-4} \div 1.4/10^2 \, m/s$ and for variants with $\alpha_{l3} = 11 \div 23°$, $M_{u2} = 1.215$, $m \cong 2.1 \, kg/s$, the speed $6.5/10^{-5} \div 1.9/10^2 \, m/s$.

The numerical experiment results for the diffuser with $\alpha_{l3} = 14°$, $M_{u2} = 0.81 \div 1.215$ are presented in fig. 9. The solutions convergence was extremely unsatisfactory for variants with $\alpha_{l3} = 14°$, $M_{u2} = 1.42$. It is possible that the M_{u2} value increase significantly affects the numerical study results quality and the solutions convergence deteriorates. Satisfactory convergence was obtained at various mass flow rates for variants with $\alpha_{l3} = 17°$, $M_{u2} = 1.42$.

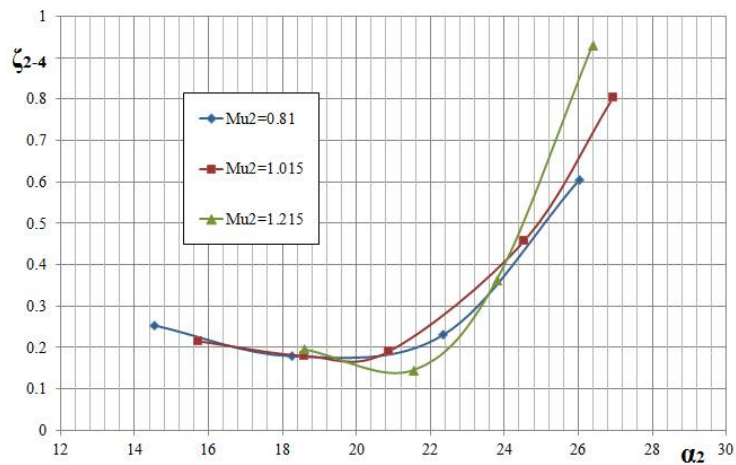Fig. 9. The numerical experiment results. Gas dynamic characteristics for $\alpha_{l3} = 14°$, $M_{u2} = 0.81 \div 1.215$.

The different angle α_{l3} diffusers gas-dynamic characteristics comparison, M_u2 = 1.015 is presented in fig.10.
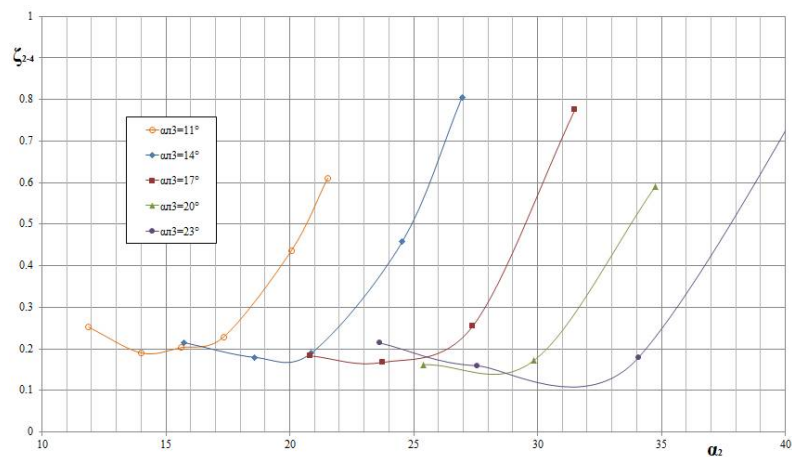
Fig. 10. The numerical experiment results. Gas dynamic characteristics for $\alpha_{l3} = 11 \div 23°$, $M_{u2} = 1.015$.

Conclusions

The sufficient difference in loss values obtained in the full-scale and numerical experiments was revealed As a result of the work carried out. It is difficult to use numerical gas-dynamics methods using such values for serious calculations, which design studies will be based on. However, this result is connected with certain limitations that are unavoidable when carrying out this level work. Firstly, CFD programs are technically and structurally complex computing systems, the work, settings and capabilities study of takes a great amount of time. Secondly, some assumptions and simplifications were made in this work that influenced the result. One of the simplifications most detrimental to the result is the lack of seals in the design model. Thirdly, the numerical gas dynamics methods always have some error with respect to real processes take place in turbomachines.

Considering all these factors, we can conclude that the work results are satisfactory and they can be significantly closer to excellent if the above reasons are eliminated.

Also, the results could be influenced by the particular gas (R12) properties, included in the computing complex material properties general base. For example, the working fluid density ρ influence at certain parameters and the gas heat capacity c_p influence.

It was revealed that the variants with $M_{u2} = 1.015; 1.215$ have the best the numerical calculation results convergence in the course of the calculations, which is confirmed by the full-scale and numerical experiments results comparison presented in Fig. 5 and 6. It is necessary to look for other CFD modeling approaches and try to carry out more accurate verification calculations at high relative Mach numbers. This will require a fine enough grid tuning, turbulence models, etc.

However, before focusing on the work results, it is necessary to conduct a more detailed numerical gas dynamics methods study in relation to the centrifugal compressors operating on refrigerant and at high Mach numbers vane diffusers.

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