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Centrifugal refrigeration compressor design

Особенности проектирования центробежных холодильных компрессоров

В статье представлены материалы об особенностях проектирования центробежного холодильного компрессора для магистральных газоперекачивающих станций (КС).

Для обеспечения круглогодичного охлаждения газа на КС месторождения «Ямбургское» используется станция охлаждения газа (СОГ). Во избежание нагрева грунтов вечной мерзлоты температура природного газа, подаваемого в магистральный трубопровод, должна быть в интервале от 0 до минус 2°С. Центробежный холодильный компрессор является основным элементом СОГ, связанным с режимами работы холодильной установки и зависящим от параметров окружающей среды, охлаждаемого газа, состава хладагента и иных факторов.

***Ключевые слова**: центробежный холодильный компрессор, центробежная ступень, газодинамические характеристики, прочность, жесткость, массогабаритные показатели.*

У статті представлені матеріали про особливості проектування відцентрового холодильного компресора для магістральних газоперекачуючих станцій (КС).

Для забезпечення цілорічного охолодження газу на КС родовища "Ямбургское" використовується станція охолодження газу (СОГ). Щоб уникнути нагріву грунтів вічної мерзлоти температура природного газу, що подається в магістральний трубопровід, має бути в інтервалі від 0 до мінус 2°С. Відцентровий холодильний компресор є основним елементом СОГ, пов'язаним з режимами роботи холодильної установки і залежним від параметрів довкілля, охолоджуваного газу, складу холодагенту і інших чинників.

***Ключові слова**: відцентровий холодильний компресор, відцентрова ступінь, газодинамічні характеристики, міцність, жорсткість, показники маси та габаритів.*

The purpose of the paper is to design propane-butane refrigeration compressor for the natural gas cooling station (GCS) at the compressor station (CS).

A number of compressor configurations, such as single-flow gearless, single-flow geared and double-flow was considered, their advantages and disadvantage were analyzed. Variants thermodynamic, design and checking gas-dynamic calculations were performed. Also gas-dynamic calculation of the first compressor stage using computational fluid dynamics (CFD) was carried out. Using FEA modeling after the numerous redesigns and strength recalculations, the compressor casing, cover and impeller with maximal strength and stiffness properties at minimum mass-dimensional parameters were designed.

***Keywords**: centrifugal refrigeration compressor, centrifugal stage, gas-dynamic characteristics, strength, stiffness, mass-dimensional parameters.*

Introduction

To prevent soil thawing at the utmost North, natural gas temperature in a main pipeline has to lay over the range 0 to -2°C. The special gas cooling station (GCS) has been designed to provide the year-round gas cooling. The required temperature regime will be ensured in the following way: in winter, over the six months compressed gas will be cooled with the air coolers (AC), and in summer, when ambient temperature increases to the +13°C, natural gas will be cooled with both air coolers and evaporators of the GCS.

The main component of the GCS is the centrifugal refrigeration compressor. It has specific features, caused by the environment and cooled gas parameters, refrigerant composition and other factors.

Compressor characteristics

The designed compressor has to provide cooling of the 305 kg of the gas per second. It corresponds to the refrigerant mass flow of 61 kg/s. Polytropic efficiency has to be no less than 78%. Refrigeration compressor should work properly at maximum absolute discharge pressure 1,5 MPa, corresponding to the condensation temperature 50°C . In addition compressor casing has to stand maximum allowable working pressure 3,5 MPa, at minimum weight and overall dimensions.

Construction arrangements selection

Designing the compressor a number of construction arrangements, such as single-flow gearless, single-flow geared and double-flow were considered and analyzed. Analyzing these variants a number of unsatisfactory features has revealed.

As the unidimensional flow analysis showed there is a supersonic refrigerant flow in the single-flow gearless construction, so the shock loss minimization problem has to be solved. The single-flow geared configuration eliminates supersonic flow, but because of the significant rotor speed drop, rotor approaches its first critical speed.

Analyzing all faults of the single-flow configuration, the double-flow construction with two suction and one discharge pipes and with impellers back to back arrangement was chosen (Figure 1). Besides such arrangement has a number of advantages. Firstly, there is almost no axial thrust acting on the rotor and secondly the supersonic flow of the refrigerant is ruled out.

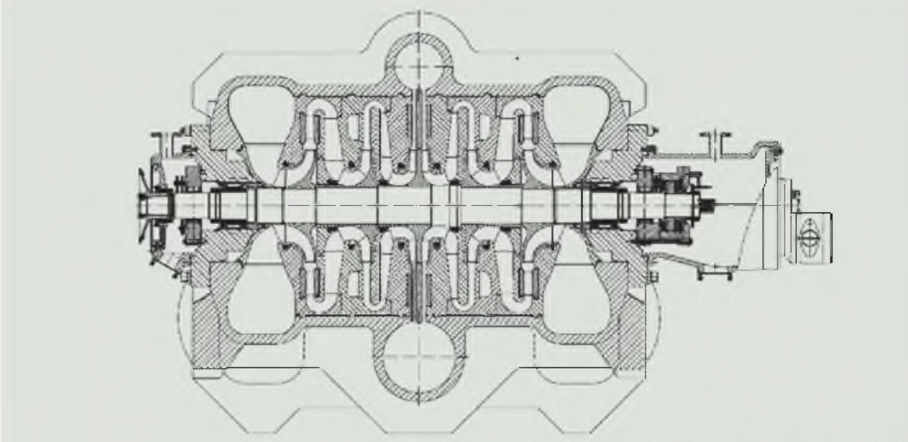


Figure 1. Compressor meridional section

Compressor setting design

Compressor setting was designed on the basis of the similarity theory.

To chose compressor stages, variants calculation has been done. According to the results of the calculation polytropic work, required to compress gas to the pressure p_{out}=0,965MPa at pressure ratio π=4,14 and polytropic efficiency equals H_p=69,3991 kJ/kg. Polytropic head coefficient, required at the taken rotor speed n=7800rev./min and impeller outlet diameter D_2=0,525 m equals ψ_p=1,509, and discharge coefficient Φ_0=0,1302. Using obtained discharge coefficient, polytropic head coefficient and polytropic compression work the compressor setting was designed. As the prototype model for the first stage the high-capacity stage was chosen. As the prototype models for the second and the third stages the licensed stages with vaneless diffusers with the outlet impeller diameter 0,482 m were chosen.

Calculation of the gas-dynamic characteristics, diffuser blade profile, return channel, gas-dynamic checking calculation and axial thrust calculation were performed using computer programs designed at the enterprise, which are based on the techniques, presented in Galerkin (2000) [1], Den (1973) [2] and Seleznyev and Galerkin (1982) [3].

Calculation and design of the first compressor stage

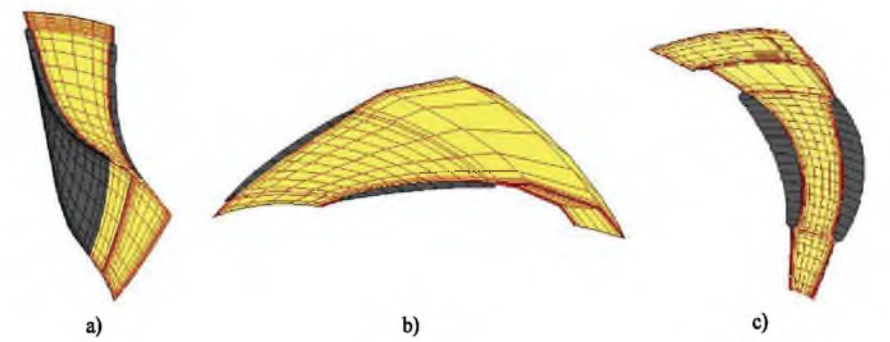
Up to the present moment one-dimensional gas-dynamic calculation was the main blade passage design tool. Unfortunally this method demands knowledge of the empirical coefficients, and what is more importantly it disregards the real flow pattern. Thereby it is impossible to design competitive compressors without modern three-dimensional flow analysis.

At the beginning, indispensable gas-dynamic calculations based on the quasi-three-dimensional model have been completed. Further a number of gas-dynamic checking calculations to estimate the real stage characteristics using CFD method have been completed.

The boundary conditions are total pressure, total temperature and inlet flow angle distribution at the inlet and the static pressure at the outlet. Riemann invariant was used for boundary conditions. Non-slip and adiabatic conditions were imposed all over the solid walls.

ENH [4] scheme was chosen for the current difference scheme. This is the extra accuracy Godunov scheme.

For the calculation, the stage was divided into two computational domains. The first one includes the impeller, and the second one includes vaned diffuser and return channel. The grid contains 458752 cells in the impeller, 294912 in the vaned diffuser and 557056 in the return channel (Figure 2). H-type cells, recommended for the three-dimensional blade channels were used.



a) b) c)

Figure 2. Computational grid: *a) the impeller; b) the diffuser; c) the return channel*

A commercial solver package has been used for the current study. The solver adopted is a three-dimensional, viscous, time-accurate code to solve the unsteady Reynolds-averaged Navier-Stokes equations.

Menter's differential model of turbulence was chosen. Inlet turbulence level was given 7,5 % for current calculation.

The calculated characteristics are presented in the Figure 3. As the diagrams show, the results of

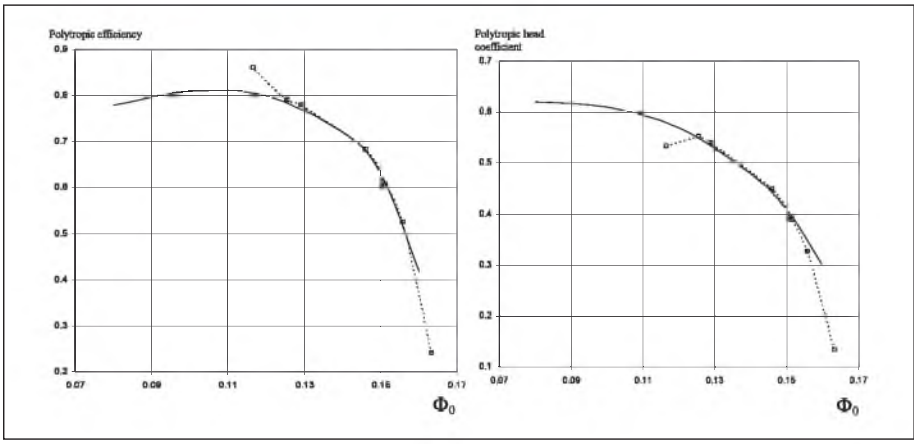


Figure 3. The first compressor stage characteristics: *—— – theoretical characteristics (quasi-three-dimensional model); □□□ – calculated characteristics (CFD calculation)*

The final construction with high and long ribs, which embraced compressor entry and end surface, is shown in the Figure 6. Also the horizontal rib, which together with the flow splitter, forms through stiffening rib was added. Besides, after the numerous calculations flow splitter shape was determined. Long fillet radius at flow splitter and compressor entry wall junction were used.

Results of the calculation of the final construction are shown in Figure 7. Apparently taken steps decreased axial movements and reinforce construction considerably.

the CFD calculations at the design conditions agree with the theoretical characteristics.

Compressor casing and cover design

Because of the complicated three-dimensional configuration of the constructs the three-dimensional computational model should be used. A commercial solver package has been used for the current investigation.

The design and the finite element model of the tentative casing construction are shown in the Figure 4. The casing is symmetrical, so to minimize the computational resources symmetric section was considered.

The construction is loaded with internal hydrostatic test pressure 4 MPa.

The results of the calculation are shown in Figure 5. Analyzing these results it was found, that axial tensile stresses in the flow splitter region of the volute reach the critical value (Figure 5,a). High axial stresses arise in the compressor inlet duct and horizontal parting flange connection point (Figure 5, a). In addition because of the compressor entry flat region and end surface buckling the bending stresses arise (Figure 5, b).

In the first calculation tightening torque for the stud joint of the horizontal parting flange was calculated. The contacting pair "casing-cover" was designed. The tightening torque was obtained from the tightness condition. It equals 50 tons.

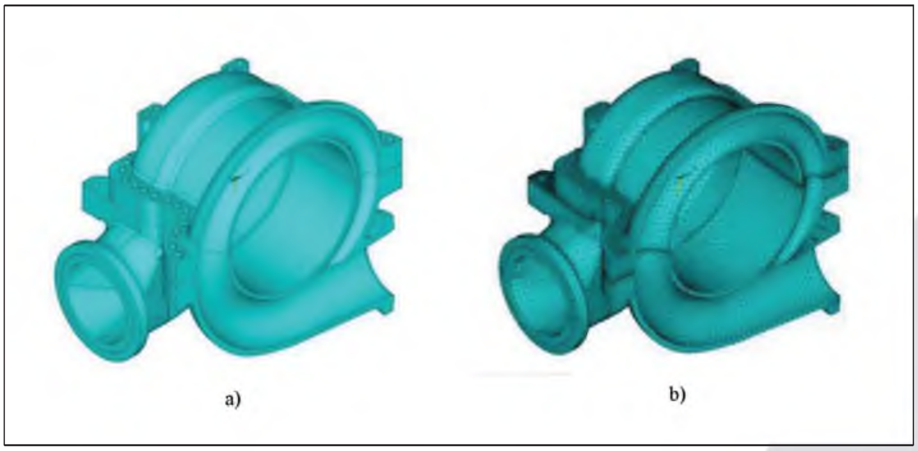


Figure 4. The design and the finite element model, tentative construction: *a) the design model; b) the finite element model*

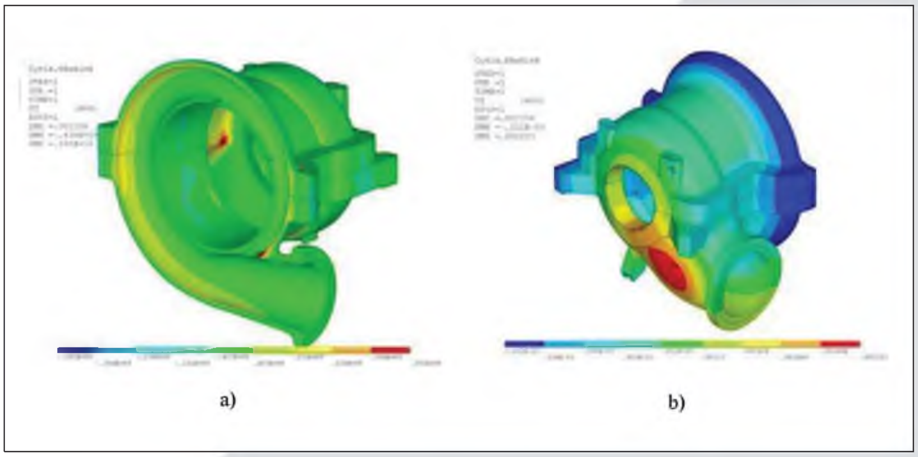


Figure 5. Results of the strength analysis, tentative construction: *a) axial stress; b) axial movement*

Further the tentative construction was ruggedized with strengthening elements. Stiffening ribs were installed from the outside, and the flow splitters in the compressor inlet duct. Different embodiments with different number, shape and arrangement of the ribs and splitters were calculated.

High tension regions decreased, displaced and became local.

Maximum axial stress equals σ=525 MPa (σ_z=1850MPa for the tentative construction).

Mechanical properties of the material after the final thermal treatment are:

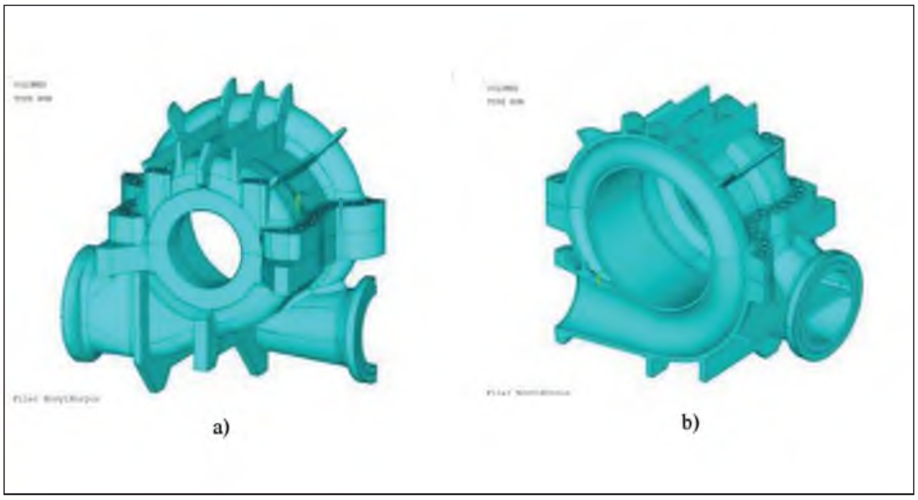


Figure 6. Final compressor case construction

- σ_y=540 MPa – yield strength;
- σ_U=525 MPa – ultimate tensile strength.

Thereby, as the results show, local stresses for the final case construction are safety, because they are lower than the yield strength.

Compressor first stage impeller design

In consequence of the gas-dynamic calculation the impeller blade profile for the first stage was obtained. Further impeller hub and shroud plates were designed. After the numerous re-designs and strength recalculations the impeller construction (Figure 8,a), provided minimum stresses (Figure 8,b) has been designed.

Designed blade welding slot and blade profile further were used to design CNC machine part program.

Conclusions

Following tasks were solved in this paper:

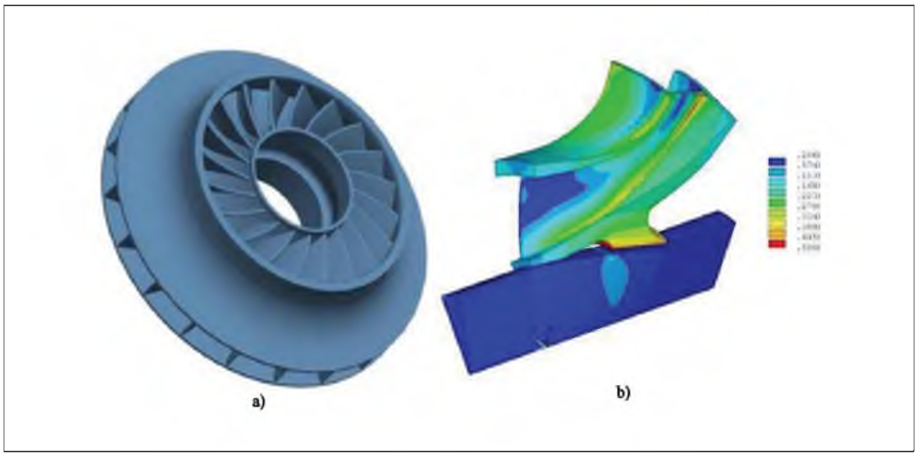


Figure 8. Impeller strength calculation: *a) 3D impeller model; b) von Mises equivalent stress*

1. A number of construction arrangements were considered and analyzed. Double-flow construction with two suction and one discharge pipes and with impellers back to back arrangement was chosen as the most suitable from the engineering and gas-dynamic points of view;

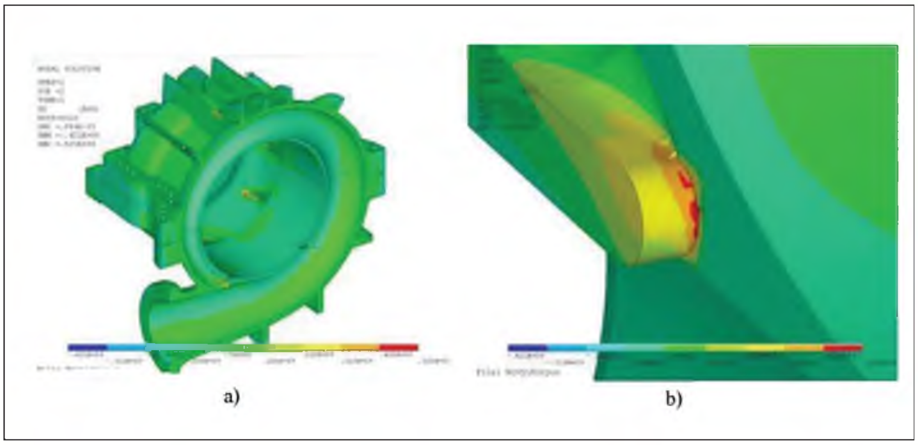


Figure 7. Axial stress distribution, final construction: *a) compressor casing; b) flow splitter*

2. Using CFD the dimensionless characteristic of the first stage were calculated;

3. By means of the numerous strength recalculations the casing and cover constructions, provided maximum strength and stiffness at minimum mass-dimensional properties have been designed. Besides impeller construction provided minimum stresses has been designed.

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