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GAS-DYNAMIC LOSSES IN HIGH-SPEED HERMETIC COMPRESSORS FOR SHIPBOARD AIR-CONDITIONING SYSTEMS AND METHODS FOR THEIR REDUCTION

ГАЗОДИНАМІЧНІ ВТРАТИ У ВИСОКООБОРОТНИХ ГЕРМЕТИЧНИХ КОМПРЕСОРАХ СУДНОВИХ КОНДИЦІОНЕРІВ ТА ШЛЯХИ ЇХ ЗНИЖЕННЯ

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Abstract. The results of research gas dynamical performances of valving knots and gas ducts of high-turning hermetic compressors of marine air conditions were presented. The numerical values of resistance coefficients of valving knots and ducts, the values of gas velocities in it and dependence this performances from operating condition of compressor were conducted.

Purpose. Investigation of the gas-dynamic characteristics in the suction and discharge valve assemblies and in the corresponding gas-flow passages of high-turning reciprocating freon compressors is of particular importance for identifying ways to reduce energy losses.

Method. The gas-dynamic characteristics were investigated by simulating the working medium: instead of refrigerant, the valve assemblies and channels were blown through with compressed air. The modelling was approximate because the adiabatic exponents of air and the refrigerant are not identical. During testing, the gas-flow regime was maintained according to the Reynolds criterion Re_c, referred to the cylinder cross-sectional area; that is, the cylinder diameter d_c was taken as the determining dimension, since for a given design the geometrical ratios in the valve assembly depend only on the cylinder diameter.

The suction path of the compressor under study included the channels between the rotor and stator of the electric motor and the longitudinal grooves used for welding the stator stack; the channel in the crankcase block leading to the suction cavity of the cylinder head; the ports in the valve plate; and the suction valve. The space between the compressor casing and the compressor itself was not considered because losses in this part of the suction path are small.

The discharge path of the compressor consisted of the channels in the valve plate; the discharge valve with its buffer plate and lift limiter; the discharge cavity of the cylinder head, connected by a short pipe to the receiver; the receiver, which served as the compressor muffler; and the discharge tube up to the outlet connection.

Such configurations of the suction and discharge paths are typical of most marine hermetic compressors.

Results. The experimental results showed that, for the considered class of high-speed compressors, the value of ζ_v for annular valves lies within the range 2.4–4.2, which is approximately 1.5–2 times higher than the values recommended in the literature for annular valves of reciprocating refrigeration compressors.

As a result of the tests carried out with air and with refrigerant, the values of the resistance coefficient ζ_p for the compressor gas passages were obtained: 3.6 for the discharge path and 4.5 for the suction path

Scientific novelty. Using the obtained values of ζ_p and the recommended gas velocities in passage elements, the cross-sections of the discharge tube and the suction ports in the valve plate were calculated. The calculation showed that the cross-section of the flow channels in the electric motor was reduced and that the resistance of the suction path increased. The flow cross-sections of the channels in the electric motor can be increased by 15–20 %, and, as a consequence, compressor power consumption can be reduced by approximately 2 %, by widening the grooves used for welding the stator stack.

Practical importance. The study results were used in the development and refinement of new hermetic compressors with shaft rotational speeds of 50 s^{-1} and 66.7 s^{-1} for marine air conditioners.
Key words: hermetic compressor; gas dynamical performances; duct; valving knots.

Анотація. Наведено результати досліджень газодинамічних характеристик клапанних вузлів та газових трактів високооборотних герметичних компресорів суднових кондиціонерів. Наведено числові значення коефіцієнтів опору клапанних вузлів та трактів, величини швидкостей газу в них та залежність цих характеристик від режиму роботи компресора.
Мета. Дослідження газодинамічних характеристик у всмоктувальному та нагнітальному клапанних вузлах та відповідних газових трактах суднових високооборотних поршневих фреонових компресорів з метою відшукання шляхів зменшення втрат енергії.
Методика. Газодинамічні характеристики досліджували шляхом моделювання робочого середовища: замість хладону клапанні вузли та канали продували стисненим повітрям. Моделювання було наближеним, оскільки показники адіабати повітря та хладону не однакові. При випробуваннях витримували режим течії газу за критерієм Рейнольдса $Re_ц$, віднесеному до площі перерізу циліндра. за визначальний розмір приймали діаметр циліндра $d_ц$, оскільки геометричні співвідношення клапанному вузлі при однаковій конструкції залежать тільки від діаметра циліндра.
Досліджений всмоктуючий тракт компресора включав: канали між ротором та статором електродвигуна та поздовжні пази під зварювання пакета статора; канал у блок-картері, що веде у всмоктувальну порожнину головки циліндра, вікна в клапанній дошці та всмоктувальний клапан. Простір між кожухом компресора і самим компресором не розглядали через небагато втрат у цій частині тракту, що всмоктує.
Нагнітальний тракт компресора складався з каналів у клапанній дошці, нагнітального клапана з буферною пластинкою і обмежувачем підйому, нагнітальної порожнини головки циліндра, з'єднаної коротким патрубком з ресивером, ресивера (служив глушником компресора) і трубки нагнітальної до вихідного патрубка.
Такі схеми всмоктувального та нагнітального трактів притаманні більшості суднових герметичних компресорів.
Результати. Результати дослідів показали, що для розглянутого класу високооборотних компресорів значення кільцевих клапанів знаходиться в інтервалі 2,4...4,2, що приблизно в 1,5...2 рази перевищує дані, рекомендовані літературою кільцевих клапанів поршневих холодильних компресорів.
В результаті проведених випробувань на повітрі та холодоагенті отримані значення коефіцієнта опору $\xi_{оп}$ газових трактів компресора, дорівнюють для нагнітального тракту – 3,6 та для всмоктування – 4,5.
Наукова новизна. За отриманими значеннями $\xi_{оп}$ та рекомендованими швидкостями газу в елементах тракту були розраховані перерізи нагнітальної трубки та всмоктувальних вікон у клапанній дошці. В результаті зменшився переріз прохідних каналів в електродвигуні та підвищився опір всмоктуючого тракту. Збільшити прохідні перерізи каналів в електродвигуні на 15–20 % (і як наслідок, знизити споживану потужність компресора приблизно на 2 %) можна шляхом розширення пазів під зварювання пакета статора.
Практична значимість. Результати досліджень використані при створенні та доведенні нових герметичних компресорів із частотою обертання валу 50 $с^{-1}$ та 66,7 $с^{-1}$ для суднових кондиціонерів.
Ключові слова: герметичний компресор; газодинамічні характеристики; тракт; клапанні вузли.

FORMULATION OF THE PROBLEM

The experimental study was carried out on a KKhGV-14 (КХГВ-14) hermetic compressor designed for marine self-contained air conditioners [2]; for this compressor, losses in the valves and channels are the highest.

The compressor is a two-cylinder unit with a cylinder diameter of 50 mm, a piston stroke of 24 mm, a refrigeration capacity of 17 kW under nominal operating conditions, and a rotational speed of 66.7 s^{-1} at a power-supply frequency of 400 Hz.

The suction valves are annular valves with two support projections for the lift limiter. The discharge valves are also annular and are equipped with annular buffer plates of special configuration and a lift limiter. The valve plate contains a number of suction and discharge channels arranged around the circumference. The plate lift of the discharge valve is 1 mm, and that of the suction valve is 1.6 mm.

ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

Energy losses caused by hydraulic resistance in the suction and discharge valve assemblies and in the corresponding gas-flow passages of high-speed reciprocating Freon compressors used in marine air-conditioning systems amount to 19–22 % of the energy consumed by the compressor electric motor [1, 4]. The greatest losses are due to resistance in the discharge and suction gas passages (up to 53 %).

Accordingly, investigation of the gas-dynamic characteristics of these compressor components is of particular importance for identifying ways to reduce energy losses.

PURPOSE OF THE STUDY

Investigation of the gas-dynamic characteristics in the suction and discharge valve assemblies and in

the corresponding gas-flow passages of high-turning reciprocating freon compressors is of particular importance for identifying ways to reduce energy losses.

MAIN MATERIAL

The gas-dynamic characteristics were investigated by simulating the working medium: instead of refrigerant, the valve assemblies and channels were blown through with compressed air. The modelling was approximate because the adiabatic exponents of air and the refrigerant are not identical. During testing, the gas-flow regime was maintained according to the Reynolds criterion Re_c, referred to the cylinder cross-sectional area; that is, the cylinder diameter d_c was taken as the determining dimension, since for a given design the geometrical ratios in the valve assembly depend only on the cylinder diameter.

The value of the resistance coefficient ζ depends on the reference velocity and, consequently, on the cross-section to which it is referred. In the experiments, it was determined for the minimum cross-section of the assembly under investigation (the valve slit, the discharge tube, the suction ports in the valve plate, etc.). The value of ζ for another, i-th, cross-section was generally recalculated, with gas density ρ varying along the flow, by the formula

$$\zeta_i = \zeta_a(\rho_i/\rho_a)^2(F_i/F_a)^2,$$

where F_i and F_a are, respectively, the areas of the i-th cross-section and of the assembly, and ρ_i and ρ_a are, respectively, the gas densities in the i-th cross-section and in the assembly.

The following quantity was used as the gas-flow velocity criterion in the valve:

$$M_v = \omega_v/\omega_s,$$

where ω_v is the gas velocity in the valve, m/s, and ω_s is the speed of sound in the gas, m/s.

Since the flow in the valves is subcritical under full-scale operating conditions, the condition $P_2/P_1 > 0.528$ was maintained in the model air-flow regimes. Here P_1 and P_2 are the air pressures upstream and downstream of the assembly under investigation.

The suction path of the compressor under study included the channels between the rotor and stator of the electric motor and the longitudinal grooves used for welding the stator stack; the channel in the crankcase block leading to the suction cavity of the cylinder head; the ports in the valve plate; and the suction valve. The space between the compressor casing and the compressor itself was not considered because losses in this part of the suction path are small.

The discharge path of the compressor consisted of the channels in the valve plate; the discharge valve with its buffer plate and lift limiter; the discharge cavity of the cylinder head, connected by a short pipe to the receiver; the receiver, which served as the compressor muffler; and the discharge tube up to the outlet connection.

Such configurations of the suction and discharge paths are typical of most marine hermetic compressors.

The tests were carried out in the following sequence: first, the valve assembly and its individual elements were blown through with air; next, the suction and discharge gas-flow passages were tested separately. The air blow-through data were compared with the results of compressor tests (indicator measurements) performed with refrigerant.

The experimental results showed that, for the considered class of high-speed compressors, the value of ζ_v for annular valves lies within the range 2.4–4.2, which is approximately 1.5–2 times higher than the values recommended in the literature [5] for annular valves of reciprocating refrigeration compressors.

Investigation of the assembled discharge-valve components made it possible to evaluate the influence of the buffer plate on its gas-dynamic characteristics. It was established that, when the buffer plate is present, the value of ζ_v increases by nearly 50 %.

The question naturally arises of improving valve characteristics. However, it appears that the characteristics of the discharge valve cannot be improved. As our experiments have shown [2, 4], increasing the valve-plate lift in the range of 1.4–1.7 mm led to its rapid failure during compressor operation, while reducing the stiffness of the buffer plate caused delayed valve closure at a condensing temperature of 50 °C and a discharge-to-suction pressure ratio P_d/P_s greater than 4.

The characteristics of the Tecumseh (USA) CL compressor discharge valve that we tested agree well with the experimental data obtained. The parameters of this compressor are close to those of the KHGV-14: its refrigeration capacity is approximately 14 kW at a power-supply frequency of 60 Hz, $d_c = 45.4$ mm, discharge-valve lift is 1.2 mm, and suction-valve lift is 2 mm. The value of ζ_v for the CL compressor suction valve is somewhat lower (by approximately 8 %), mainly due to the better design of the flow-through part of the channels in the valve plate.

A comparison of the experimental results with the data obtained from operation of the valves as part of a Freon compressor (Fig. 1) showed satisfactory agreement; the scatter of individual points did not exceed 15–20 %. On this basis it was concluded that, for quantitative and qualitative assessment of the gas-dynamic quality of valves and gas passages in high-speed hermetic compressors, these components can be tested with air. As a result, the experiment is substantially simplified and, consequently, so is the compressor development process.

Based on the compressor tests with refrigerant, the gas-dynamic characteristics of the valve assemblies and gas-flow passages were determined as functions of compressor operating mode.

Figure 2 presents the dependences of ζ for the suction and discharge valve assemblies and of the average velocities

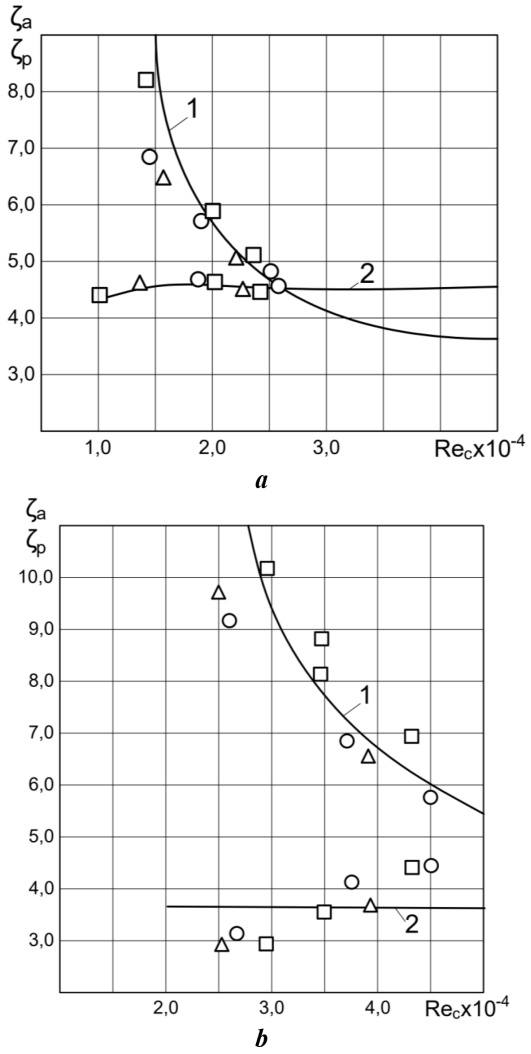


Fig. 1. Gas-dynamic characteristics of the compressor's valve assembly (**1**) and gas tract (**2**) on the discharge line (*a*) and suction line (*b*): solid line – air blowing; dots – operation of the compressor on refrigerant (Δ – t_c = 30 °C; ○ – t_c = 40 °C; □ – t_c = 50 °C)

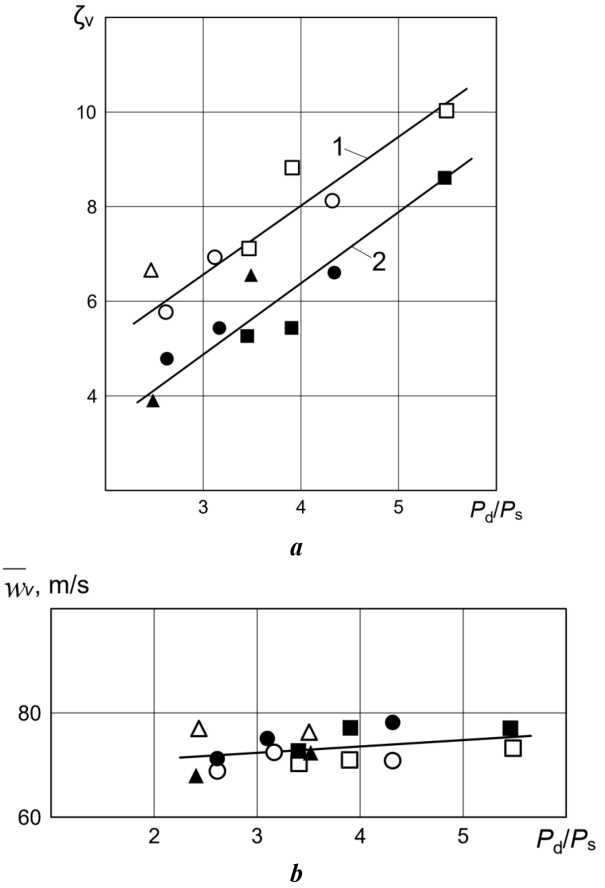


Fig. 2. Dependence of the compressor's valve assembly resistance coefficient ζ_v (*a*) and the average velocity in the valve $\bar{w}_v$ (*b*) on the pressure ratio P_d/P_s (Δ – t_c = 30 °C; ○ – t_c = 40 °C; □ – t_c = 50 °C)

in the valves, referred to the valve slit, on the compressor operating mode, i.e., on the pressure ratio P_d/P_s.

It was established (see Fig. 2, *a*) that, as P_d/P_s increased from 2.5 to 5.5, the value of ζ_v for the discharge valve assembly increased from 5.5 to 10.2, while that for the suction assembly increased from 4 to 8.5. This can be explained by a decrease in the equivalent valve cross-section as P_d/P_s increases and by a decrease in the valve flow coefficient.

The average velocities in the valves (see Fig. 2, *b*) changed only slightly with compressor operating mode. For both suction and discharge valves, as P_d/P_s increased, the average velocity rose approximately from 70 to 78 m/s. In this case, M_v did not exceed 0.4. The maximum gas velocity in the valves, at the moment of their opening, reached 106–110 m/s, and M_v was 0.6. It should be noted that the obtained average valve velocities and, accordingly, M_v were approximately 1.5–2 times higher than the recommended values [5]. In our opinion, this situation is characteristic of the class of high-speed compressors considered here.

As a result of the tests carried out with air and with refrigerant, the values of the resistance coefficient ζ_p for the compressor gas passages were obtained: 3.6 for the discharge path and 4.5 for the suction path (see Fig. 1).

These values indicate high gas velocities in the passages. For example, the average velocity in the discharge tube, from the receiver to the outlet connection, under nominal compressor operation reached nearly 60 m/s. This led to unjustified losses: the pressure drop over this section was about 0.1 MPa, and the compressor power consumption increased by approximately 6–7 %. A similar situation was also observed in the compressor suction path.

Since it is practically impossible to substantially reduce ζ_p of the gas-flow passages with the existing compressor design, the only way to reduce losses is to increase the flow cross-sections of the gas-passage components that cause the largest losses.

Using the obtained values of ζ_p and the recommended gas velocities in passage elements, the cross-sections of the discharge tube and the suction ports in the valve plate were calculated. The calculation showed that the cross-section of the flow channels in the electric motor

was reduced and that the resistance of the suction path increased. The flow cross-sections of the channels in the electric motor can be increased by 15–20 %, and, as a consequence, compressor power consumption can be reduced by approximately 2 %, by widening the grooves used for welding the stator stack.

CONCLUSIONS

Tests of the compressor with an enlarged discharge-tube cross-section, its internal diameter having been increased from 10 to 14 mm, showed that power consumption was reduced by a further 0.35 kW, i.e., by more than 6 %. No pressure pulsations were observed downstream of the discharge connection, and the noise level remained unchanged.

To assess the influence of reducing the discharge-valve plate lift on compressor performance, the compressor was tested with a plate lift of 0.9 mm; a 0.5 mm plate was installed instead of the standard 0.4 mm shut-off plate. It was found that the compressor refrigeration capacity remained practically unchanged, while power consumption increased by less than 2 %.

The study results were used in the development and refinement of new hermetic compressors with shaft rotational speeds of 50 s^{-1} and 66.7 s^{-1} for marine air conditioners.

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