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EVALUATION OF THE EFFICIENCY OF CYCLES OF MARINE HERMETIC VAPOR COMPRESSOR REFRIGERATION MACHINES

ОЦІНКА ЕФЕКТИВНІСТІ ЦИКЛІВ СУДНОВИХ ГЕРМЕТИЧНИХ ПАРОКОМПРЕСОРНИХ ХОЛОДИЛЬНИХ МАШИН

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Abstract. The question relating to the assessment of marine hermetic vapour compressor refrigeration machine (VCRM).

Due to the use of climate control equipment on ships with sealed VCRM with air condensers, condensing temperatures have increased and cycle limits have widened, with correspondingly lower reversibility coefficients.

In a real VCRM, expansion of the cycle temperature limits and an increase in the pressure ratio result in a drop in the compressor unit efficiency and delivery ratio, which must have a serious effect on the VCRM characteristics.

The aim of the study is to provide a method of evaluating the effectiveness of the cycles of modern hermetic VCRM ship equipment air conditioning and refrigeration in the absence of irreversibility and the expansion temperature limits cycle with the ideal and the real hermetic compressor units and the impact of losses in the heat exchangers (evaporator and condenser) and the compressor unit on the characteristics VCRM.

The losses caused by temperature differences in the heat exchangers have the same effect on the performance of the VCRM as losses in the compressor unit.

It was noted earlier, that in order to assess the actual VCRM its values are compared with the characteristics of the model cycle and the theoretical cycle.

In the particular example shown in the ship's actual temperature limits VCRM expansion cycle and increased pressure ratio leads to a drop in efficiency and flow rate of the compressor unit, which has a serious impact on the performance of VCRM. The study found that the loss of energy caused by the temperature difference in the heat exchangers (evaporator and condenser), have the same effect on the characteristics of the ship VCRM ¸ as losses in compressor unit. The proposed method makes it possible to assess the effectiveness hermetic VCRM depending on design parameters and operating conditions of the machine. The research results can be applied in the design of the marine equipment hermetic VCRM air-conditioning and refrigeration.

Key words: cycle of hermetic vapor compressor refrigeration machine; efficiency of cycles.

Анотація. Обговорюється питання, пов'язане з оцінкою досконалості суднових герметичних парокомпресорних холодильних машин (ПКХМ).

У зв'язку із застосуванням на суднах кліматичного обладнання, що має у своєму складі герметичні ПКХМ з повітряними конденсаторами, підвищилися температури конденсації та розширилися межі циклів, відповідно знизилися коефіцієнти оборотності.

У дійсній ПКХМ розширення температурних меж циклу та збільшення відношення тисків призводить до падіння ККД та коефіцієнта подачі компресорного агрегату, що має серйозно впливати на характеристики ПКХМ.

Метою дослідження є створення методу оцінки ефективності циклів сучасних герметичних ПКХМ суднового обладнання кондиціювання та рефрижерації за відсутності незворотності та при розширенні температурних меж циклу з ідеальним та реальним герметичним компресорним агрегатом та впливу втрат у теплообмінних апаратах (випарнику та конденсаторі) та компресорному агрегаті на характеристики ПКХМ.

Втрати, спричинені різницею температур у теплообмінних апаратах, мають такий самий вплив на характеристики ПКХМ, як і втрати в компресорному агрегаті.

Для оцінки дійсної ПКХМ її показники порівнюють із характеристиками зразкового циклу та теоретичного циклу.

На конкретному прикладі показано, як у дійсній судновий ПКХМ розширення температурних меж циклу та збільшення відношення тисків призводить до падіння ККД та коефіцієнта подачі компресорного агрегату, що серйозно впливає на характеристики ПКХМ.
В результаті дослідження встановлено, що втрати енергії, спричинені різницею температур у теплообмінних апаратах (випарнику та конденсаторі), мають такий самий вплив на характеристики судової ПКХМ як і втрати в компресорному агрегаті. Запропонований метод дозволяє проводити оцінку ефективності герметичної ПКХМ залежно від конструктивних параметрів та умов роботи машини. Результати дослідження можуть бути застосовані під час проектування герметичних ПКХМ суднового обладнання кондиціювання та рефрижерації.
Ключові слова: цикли герметичної парокомпрессорної холодильної машини; ефективність циклів.

Formulation of the problem. The only possible cycle of a vapour compressor refrigeration machine (VCRM), in which reversible conditions are observed, is the inverse Carnot cycle [4]. To estimate perfection of VCRM along with exemplary cycle (Carnot cycle), comparative theoretical cycle is applied, which includes adiabatic compression of dry steam, isobaric cooling of superheated steam, condensation along isotherm, throttling of liquid and its evaporation along isotherm [5]. It is known that a workable VCRM was created after replacement of detander by a throttling body and transition from compression of wet steam to compression of dry steam. Losses from replacement of adiabatic expansion by throttling and wet steam by dry steam depend on properties of low-temperature working fluid (LTWF) and are summarized by reversibility factor [3, 4]

$$\eta_c = \frac{\varepsilon_T}{\varepsilon_c},$$

where $\varepsilon_т$, ε_c – are the refrigeration coefficients of the theoretical and exemplary cycles.

The refrigeration factor of a theoretical hermetic VCRM is [5]

$$\varepsilon_{т.h} = \varepsilon_c \eta_c \eta_h , \qquad (1)$$

where η_h is the coefficient taking into account the influence of the built-in electric motor,

$$\eta_h = 1/(1+\bar{\theta}_{em}),$$

$\bar{\theta}_{em}$ – the relative overheating of the steam in the electric motor;

$\bar{\theta}_{em} = \theta_{em}/T_0$ (here θ_{em} is the superheat of the steam in the electric motor; T_0 is the absolute boiling temperature of the LTWF).

In a real VCRM, the heat transfer apparatus (HTA) have finite dimensions (compared to a theoretical VCRM where the HTA have an infinite surface) and the boiling T_0' point is lower than the cold source temperature $T_{c.s.}$

$$T_0' = T_{c.s.} - \theta_e ,$$

and the condensing temperature T_c' is higher than the ambient temperature $T_{a.t}$

$$T_c' = T_{a.t} + \theta_c ,$$

where θ_e, θ_c are temperature heads in evaporator and condenser.

This expands the temperature limits of the cycle:

$$\Delta T_C = T_{a.t} - T_{c.s} ,$$

$$\Delta T_c' = T_{a.t} - T_{c.s} + \theta_c + \theta_e .$$

The energy coefficients entering equation (1) vary,

$$\varepsilon_{т.h}' = \varepsilon_c' \eta_c' \eta_h' .$$

As a result $\varepsilon_c' < \varepsilon_c$; $\eta_c' < \eta_c$; $\eta_h' < \eta_h$ and, respectively, $\varepsilon_{т.h}' < \varepsilon_{т.h}$ [5]

The energy coefficients included in equation (1) are functions of the temperature difference ΔT_c, whose relative increment will be written in the form [5]

$$\frac{\left(T_c' - T_0'\right) - \left(T_{a.t} - T_{c.s}\right)}{T_{a.t} - T_{c.s}} = \frac{\theta_c + \theta_e}{T_{a.t} - T_{c.s}} .$$

In work [5] it is shown (Fig. 1), that expansion of temperature limits (ambient and cold source) has a great influence on refrigerating factor of backward Carnot cycle ε_c (line 1), smaller influence on reversibility factors of theoretical VCRM with glanded compressor unit (CU) η_c (line 2) and still smaller influence on factor η_h, considering influence of suction vapour heating in hermetic compressor units (HCU) motor (line 3).

Analysis of recent research and publications. From the analysis of researches and publications it follows that in works [3, 4] an analysis of theoretical cycles and evaluation of efficiency of large VCRM with open (gland) compressor units is given. In work [5] refrigeration coefficients and cycle reversibility coefficients were determined mainly for small hermetic commercial VCRM and it was concluded that losses caused by temperature difference in HTA affect characteristics of VCRM much more strongly than losses in hermetic compressor units (HCU).

For ship hermetic VCRM such researches were not conducted. Therefore the results given in [3, 4, 5] require specification and, as researches [1] have shown, can't be fully used at creation and perfection of ship hermetic VCRM.

The objective is to propose a method for assessing the cycle efficiency (reversible and in the presence of irreversibility) of modern hermetic VCRM of shipboard equipment air conditioning and refrigeration (EACR)

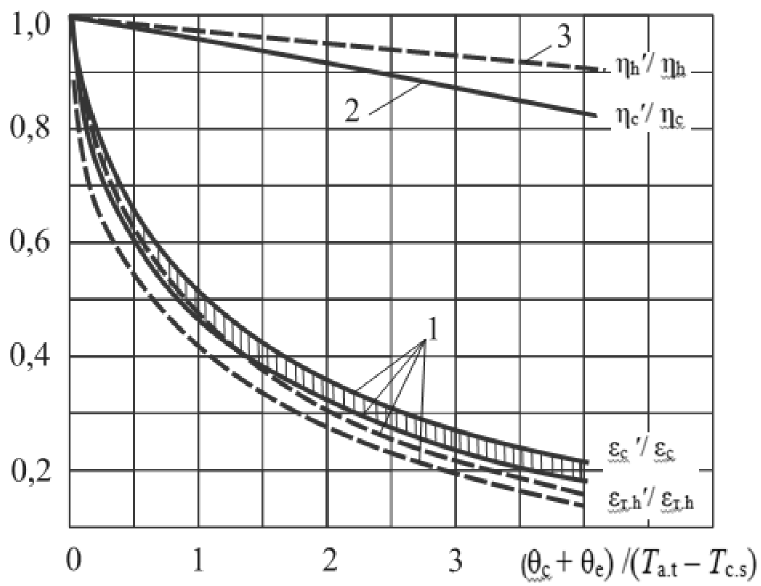

Fig. 1. Influence of expansion of the cycle temperature limits on the refrigerating factor of a theoretical VCRM: ——— with a glanded CU; - - - - - - with HCU

with ideal and real compressor units and to evaluate the degree of influence of losses in HTA and HCU on VCRM performance.

Purpose of the study. The objective is to propose a method for assessing the cycle efficiency (reversible and in the presence of irreversibility) of modern hermetic VCRM of shipboard equipment air conditioning and refrigeration (EACR) with ideal and real compressor units and to evaluate the degree of influence of losses in HTA and HCU on VCRM performance.

Main material. Due to the use of climate control equipment on ships with sealed VCRM with air condensers, condensing temperatures have increased and cycle limits have widened, with correspondingly lower reversibility coefficients.

In a real VCRM, expansion of the cycle temperature limits and an increase in the pressure ratio result in a drop in the compressor unit efficiency and delivery ratio, which must have a serious effect on the VCRM characteristics. Let's show it on the example of ship hermetic VCRM monoblock unit for provisional storerooms, maintaining in the provisional chamber 0 °C at ambient air temperature 30 °C. The volume described by pistons of HCU (HGV-14) of the unit is 21 m^3/h ($5{,}83\cdot10^{-3} m^3/s$), low-temperature working body is R22 [1].

In a real VCRM, expansion of the temperature limits of the cycle and an increase in the pressure ratio results in a drop in efficiency

θ_c, θ_e – temperature head in condenser and evaporator, °C,

$\theta_c = t_c - t_{a.t}$, $\theta_e = t_{c.s} - t_0$;

λ, η_e are feed ratio (supply coefficient) and electrical efficiency of the hermetic compressor unit.

Let us comment on the results of the calculations shown in the table.

In the absence of external irreversibility, the VCRM of the monoblock unit would operate in the following mode (see Table 1): boiling temperature t_0 = 0 °C, condensation t_c = 30 °C, subcooling t_{sub} = 30 °C, suction t_{suc} = 0 °C. Under these conditions, in a theoretical cycle, the cooling capacity of HCU is 20730 W, and the refrigerating factor is 7,48. In reality, there are volumetric and energy losses of the compressor unit. The ratio of condensing pressure to boiling pressure equal to 2,39 for HCU HGV-14 corresponds to the feed ratio λ = 0,77 and electric efficiency η_e = 0,53 [1]. Taking this into account, the cooling capacity of VCRM with the actual compressor unit, but without external irreversibility, will be 15960 W, and the refrigerating factor will be equal to 3,96, i.e. the cooling capacity will decrease by 1,3 times, and the refrigerating factor – by 1,9 times.

In a VCRM with external irreversibility (at typical average temperature heads in the condenser and evaporator $\theta_c = \theta_e$ = 10 °C for ship refrigeration) the cycle temperature limits.

Will expand considerably: the machine will operate at t_0 = -10°C, t_c = 40°C, t_{sub} = 35°C, t_{suc} = 20°C. Under these conditions, cold-productivity of the cycle with a theoretical compressor unit makes 13940 W, refrigerating factor is equal to 4,02. Under the influence of external irreversibility, cold-productivity decreases 1,49 times, and energy indices – 1,86 times. From the above it follows that influence on energy characteristics of ship hermetic VCRM from external and internal irreversibility takes place in equal degree.

Table 1. Methodology for calculating the efficiency of VCRM cycles

Magnitude, dimensionality	Designation, calculation formula	Cycle	
		at absent external irreversibility	**at available at external irreversibility**
1. Cold source temperature (air at the evaporator inlet), °C	$t_{c.s}$	0	0
2.Ambient temperature (air at the condenser inlet), °C	$t_{a.t}$	30	30
3.Subcooling temperature, °C	t_{sub}	30	35
4. Suction temperature, °C	t_{suc}	0	20
5. Boiling point (pressure), °C (mPa)	$t_0\,(p_0)$	0 (4,98)	−10 (3,55)
6. Condensing temperature (pressure), °C (mPa)	$t_c\,(p_c)$	30 (11,91)	40 (15,32)
7. Volume described by pistons, m³/s	V_h	$5{,}83\cdot10^{-3}$	$5{,}83\cdot10^{-3}$
8. Low temperature working fluid (LTWF)	R22 (t_{cr} = 96,13 °C)		
9. Refrigeration factor of an ideal cycle	$\varepsilon_c = \frac{T_{c..s}}{T_{a.t} - T_{c..s}}$	9,1	–
10. Specific volumetric cooling capacity, kJ/m³	q_v	3554,6	2390,7
11. Cooling capacity of the theoretical cycle, W	$Q_T = V_h q_v$	20730	13940
12. Reversibility factor (in the absence of external irreversibility); a = 0,75; n = 0,15 [5]	$\eta_c = a\left(\frac{t_{cr} - t_c}{t_c - t_0}\right)^n$	0,822	–
13. Refrigeration factor of a theoretical cycle (in the absence of external irreversibility)	$\varepsilon_T = \eta_c \varepsilon_c$	7,48	–
14. Refrigeration factor of an ideal cycle (in the presence of external irreversibility irreversibility)[5]	$\varepsilon'_c = \varepsilon_c \frac{1 - \frac{\theta_e}{T_{c.s}}}{1 + \frac{\theta_e + \theta_c}{T_{a.t} - T_{c.s}}}$	–	5,26
15. Performance indicator HTA	$F = \frac{\theta_c + \theta_e}{T_{a.t} - T_{c.s}}$	–	0,667
16. Attitude $\frac{\eta'_c}{\eta_c}$ (see figure 1)	$\frac{\eta'_c}{\eta_c} = f_1\left(\frac{\theta_c + \theta_e}{T_{a.t} - T_{c.s}}\right)$	–	0,97
17. Reversibility factor	η'_c	–	0,797
18. Attitude $\frac{\eta'_h}{\eta_h}$ (see figure 1)	$\frac{\eta'_h}{\eta_h} = f_2\left(\frac{\theta_c + \theta_e}{T_{a.t} - T_{c.s}}\right)$	–	0,98
19. Reversibility factor	η'_h	–	0,96
20. Refrigeration factor of the theoretical cycle (in the presence of external irreversibility)	$\varepsilon'_{T..h} = \varepsilon'_c \eta'_c \eta'_h$	–	4,02
21. Pressure ratio	p_c/p_0	2,39	4,31
22. HCU feed ratio [1]	λ	0,77	0,64
23. Electrical efficiency of HCU [1]	η_e	0,53	0,53
24. Cooling capacity (with actual compressor unit), W	$Q = Q_T \cdot \lambda$	15960	8920
25. Refrigeration factor (with actual compressor unit)	$\varepsilon_e = \varepsilon_T \eta_e$	3,96	2,13

Table 2. Values of reversibility coefficients ηc of VCRM cycles

Temperature condensation t_c, °C	Low temperature working body (LTWF)							
	Ammonia R717	Propane R290	Bhutan R600	R134A	R404A	R407C	R410A	R22
30	0,85	0,80	0,87	0,80	0,74	0,78	0,75	0,80
50	0,78	0,71	0,80	0,72	0,64	0,69	0,65	0,71

Under the influence of external irreversibility, the operating conditions of the actual compressor unit will also change. The pressure ratio will increase to 4,31, accordingly the delivery ratio will decrease to 0,64; the electrical efficiency will not change (η_e = 0,53) [1]. As a whole cold-productivity will be equal to 8920 W, i.e. it will be decreased more than in 2,3 times, and refrigerating factor will be 2,13, i.e. decreased more than in 3,5 times. The cycle calculation results agree (discrepancy no more than 3 %) with the experimental data of [1].

This example shows that losses caused by temperature differences in the heat exchangers have the same effect on the performance of the VCRM as losses in the compressor unit.

It was noted earlier, that in order to assess the actual VCRM its values are compared with the characteristics of the model cycle and the theoretical cycle. Table 2 shows the values of the coefficients η_c determined by the authors for the VCRM operating on the main modern LTWF at t_0 = -15 °C and t_c = 30 and 50 °C.

It follows from the table that if a refrigerating factor of a theoretical cycle ε_T is considered equal to the limit theoretically achievable in real VCRM, then depending on applied LTWF from 15 to 35 % of energy is lost even at perfect HCU and HTA. The most promising LTWF (in terms of efficiency) should be considered R600, R717, R22, R290, R134A.

Some words should be said in favour of LTWF R22, which belongs to hydrochlorofluorocarbons (HCFC), which use is regulated by Montreal Protocol (their step-by-step restriction with complete prohibition by 2030 is planned) [2]. Environmentalists do not even want to hear that LTWF R22, influencing so little on destruction of ozone layer, has indisputable advantages (one-component, safety, low cost, etc.) which modern LTWF do not possess and its application could be permitted in sealed systems.

Conclusions. A method is proposed to calculate and evaluate the efficiency of shipboard pressurized VCRM cycles in the absence of irreversibility and in the extension of cycle temperature limits with ideal and real HCU. It is shown that energy losses caused by temperature differences in the HTA (evaporator and condenser) have the same effect on the performance of the VCRM as losses in the compressor unit.

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