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ENERGY EFFICIENCY, ENERGY SAVING POTENTIAL AND ENVIRONMENTAL IMPACT RESEARCH OF LPG CARRIER REFRIGERATION SYSTEM

Nowadays energy efficiency improvement and global warming are issues of current interest because of the natural resources depletion and extreme climate change. Thus, the problem of formation of strict regulations regarding emissions into the air arises. This paper presents the study of cascade refrigeration system for re-condensing of associated petroleum gas during sea transportation for LPG carrier. The structural optimization has been performed. LPG gas carriers with 266 000 m³ ethane capacity require 15 MW cascade refrigeration system for re-condensing if the temperature in the coastal LPG storage is -70°C, and the temperature for transported Ethan is maintained at -75°C. For current storage conditions the required system cooling capacity is only 1,078 MW intended for the heat gain rejection from the environment during Ethane transportation. The replacement of ozone-depleting refrigerant R22 to alternative agents: R407C, R404A, R402A, R717, R290, R1270 was estimated. The results of analysis have shown that the proposed improvements can be used to optimize the LPG carrier cascade refrigeration system.

Keywords: *Associated petroleum gas; Cascade refrigeration system; Environmentally friendly refrigerant; Energy efficiency; Energy saving potential; Ethane re-condensing.*
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ДОСЛІДЖЕННЯ ПІДВИЩЕННЯ ЕНЕРГОЕФЕКТИВНОСТІ, ЕНЕРГОПОТЕНЦІАЛУ, ВПЛИВУ НА НАВКОЛИШНЄ СЕРЕДОВИЩЕ ХОЛОДИЛЬНОЇ УСТАНОВКИ ДЛЯ СУДНА-ГАЗОВОЗУ

Питання підвищення енергоефективності та глобального потепління дуже актуальні у ці дні через виснаження природних ресурсів та екстремальну зміну клімату. Таким чином, стає задача в формуванні суворих правил щодо викидів в атмосферу. Ця стаття представляє дослідження каскадної холодильної установки для реконденсації попутного нафтового газу підчас транспортування морем за допомогою газовозу. Була виконана структурна оптимізація. Газовоз для транспортування 266 000 м³ попутного нафтового газу потребує 15 МВт каскадну холодильну установку для повторної конденсації, якщо температура зберігання в берегових сховищах дорівнює -70°C, а температура транспортування -75°C. Якщо умови зберігання однакові, то необхідна потужність установки складає лише 1,078 МВт для відводу теплоприпливів з навколишнього середовища при транспортуванні етану. Було оцінено можливість заміни озоноруйнуючого холодильного агенту R22 на альтернативні агенти R407C, R404A, R402A, R717, R290, R1270. Результати аналізу показують, що запропоновані вдосконалення можуть бути використані для оптимізації каскадної холодильної установки.

Ключові слова: *Супутній нафтовий газ; Каскадна холодильна установка; Екологічно безпечний холодильний агент; Енергоефективність; Енергетичний потенціал; Повторна конденсація.*

I. INTRODUCTION

Liquified Petroleum Gas (LPG) plays an important role in meeting global needs. LPG is produced by a mix of light hydrocarbons, mainly propane and butane, changing to a liquid state when compressed at moderate pressure or chilled. Due to CO_2 emissions (80% less than solid fuels), it is considered to be the clean fuel, respecting the environment.

The shipping industry is facing a lot of challenges today. For low freight rates period, fuel prices have increased to levels only seen during the oil crisis in the 70's.

Refrigeration systems are a large consumer of energy. It can contribute critically to the running costs with significant cooling requirements. Environmental regulations with new strict requirements put extra stress on the marine sector. For now, the latest Intergovernmental Panel of Climate Change report highlighted the increased confidence in the existence of an anthropic contribution to global warming although shipping contributing by an estimated 3% to global CO_2 emissions.

Excluding of fully pressurized gas carriers, must be provided to control cargo vapour pressure in cargo tanks during cargo loading and on passage. In the case of LPG

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as well as other chemical gas carriers, a reliquefaction plant is fitted for current purpose. The system is designed to cool down the cargo tanks and associated pipelines before loading. Also the system purpose to reliquefy the cargo vapor generated by flash evaporation, liquid displacement and boil-off during loading. Another important function to maintain cargo temperature and pressure within prescribed limits while at sea by reliquefying the boil-off vapor. The cascade refrigeration system for re-condensing LPG auditing is performed for the purpose of identifying opportunities for improving the system's energy efficiency and providing relevant technically and commercially sound recommendations.

II. PERFORMANCE STUDY OF CASCADE RE-FRIGERATION SYSTEM FOR RE-CONDENSING LPG FOR GAS CARRIER

Ethan, evaporating from cargo tank CT under ambient heat gain influence, passes liquid separator LSLC, after that Ethan feed to the low temperature compressor of low temperature (pressure) cycle CLCLS where is compressed to intermediate pressure, and then boost by high temperature compressor of higher stage to condensing pressure. It is then passes through condenser-evaporator C-E where it gives heat from higher temperature. After condenser-evaporator C-E the liquid Ethan expands in throttling device EV-1, temperature reduces and pressure as well. After it returns to the cargo tank CT. In higher stage Ethan after condenser-evaporator C-E passes liquid separator LSUC then it is compressed in high temperature cycle compressor to intermediate pressure, then after economizer outlet vapor is mixed together with compressor outlet vapor of first stage high temperature (pressure) cycle after it is compressed by the compressor of second stage to condensing pressure.

Working medium of high pressure feed to the condenser where it rejects heat because of heat exchange with seawater, is condensed and is supercooled in the liquid subcooler. Then the stream is separated to the main stream and part stream (90%:10%)

Part of the stream (10%) expands in throttling device EV-2, temperature and pressure are reduced to intermediate level for main stream (90%) refrigerating purpose. Main stream passes throttling device EV-3 further feed to the condenser-evaporator where heat transfer between two refrigerants takes place, is evaporated and passes high temperature compressor of first stage. Part stream (10%) as the vapor mix with outlet vapor after compressor
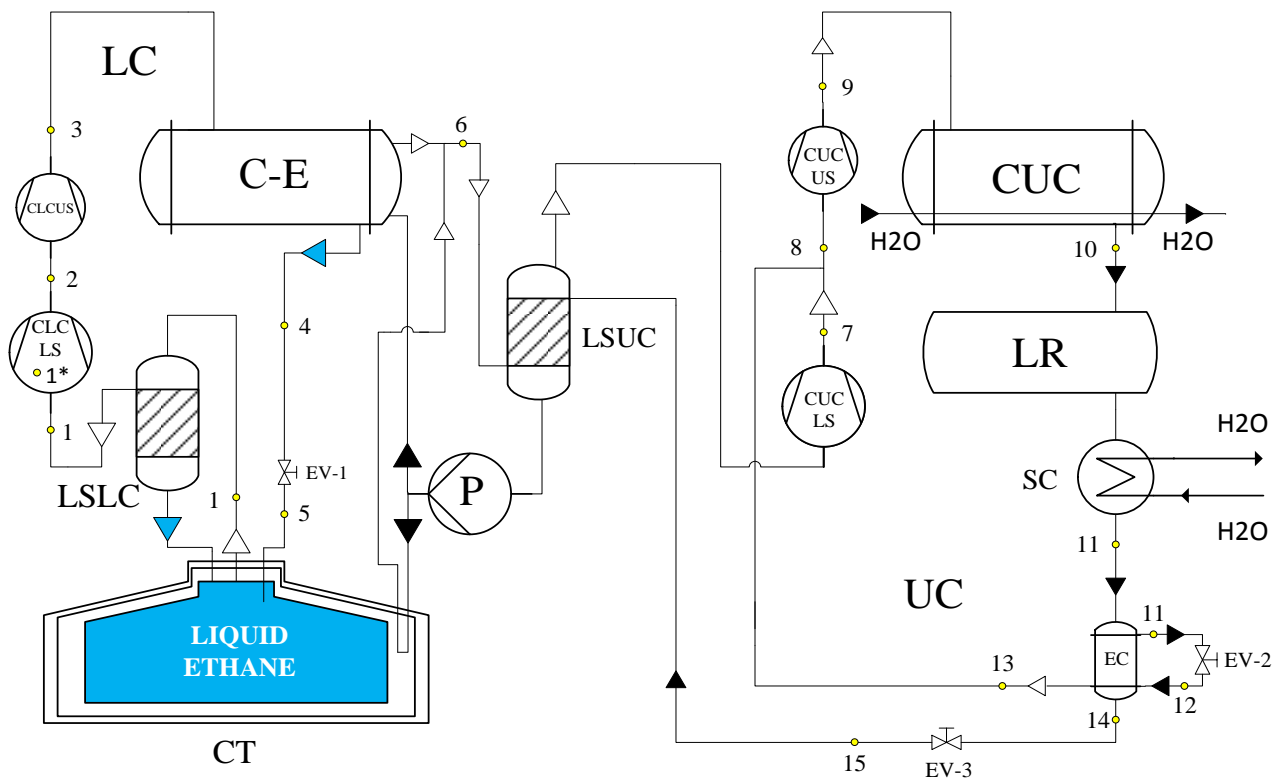

Figure 1 – **Technological flowsheet of modified cascade refrigeration system for LPG re-condensation**
CLCLS – Compressor of low temperature(pressure)of first stage; CLCUS - Compressor of low temperature(pressure) higher stage; C-E – condenser-evaporator; SC – water subcooler; EV-1 – expansion valve of low temperature (pressure) cycle; EV-2 – expansion valve of high temperature (pressure) cycle; LSLC – liquid separator of low temperature (pressure) cycle; CUCLS – Compressor of high temperature (pressure) cycle, first stage; CUCUS – Compressor of high temperature (pressure) cycle, higher stage; CUC – condenser of high temperature (pressure) cycle; LR – line receiver of high temperature (pressure) cycle; EC – economiser; LSUC – liquid separator of high temperature (pressure) cycle; P – Pump of high temperature (pressure) cycle; CT Cargo tank.

III. THERMODYNAMIC MODELING

The thermodynamic analysis of cascade refrigeration system for LPG re-condensation was performed based on the following general assumptions:

The temperature levels:

Evaporator temperature of low temperature cycle, $t_e^{LC} = -75\ °C$;

Condensing temperature of low temperature cycle, $t_c^{LC} = -21 °C$;

Evaporator temperature of high temperature cycle, $t_e^{UC} = -31\ °C$;

Condensing temperature of high temperature cycle, $t_c^{UC} = +35°C$;

During processes:

1) $1 - 1*$ - inlet pressure is reduced before input device;
2) $1* - 2$ – isentropic compression in the low temperature compressor of first stage;
3) $2 - 3$ – isentropic compression in the low temperature compressor of higher stage;
4) $3 - 4$ – isobaric heat rejection to the environment;
5) $4 - 5$ – isenthalpic expansion liquid working medium;
6) $5 - 1$ – isobaric heat gain to the liquid working medium;;
7) $6 - 7$ – isentropic compression in the high temperature compressor of first stage;
8) 8 – after gas mixing process point 13 and point 7;
9) $8 - 9$ - isentropic compression in the high temperature compressor of higher stage;
10) $9 - 10$ - isobaric heat rejection to the environment;
11) $10 - 11$ – isobaric cooling of liquid working medium in the liquid subcooler;
12) $11 - 12$ – isenthalpic expansion of part stream (10%) intended for main stream cooling;
13) $12 - 13$ isobaric heat gain to the liquid part stream (10%);
14) $11 - 14$ – isobaric cooling of main stream (90%) of liquid working medium in the economizer;
15) $14 - 15$ – isenthalpic expansion before condenser-evaporator;
16) $15 - 6$ – isobaric heat gain to working medium of high temperature cycle to condenser-evaporator.
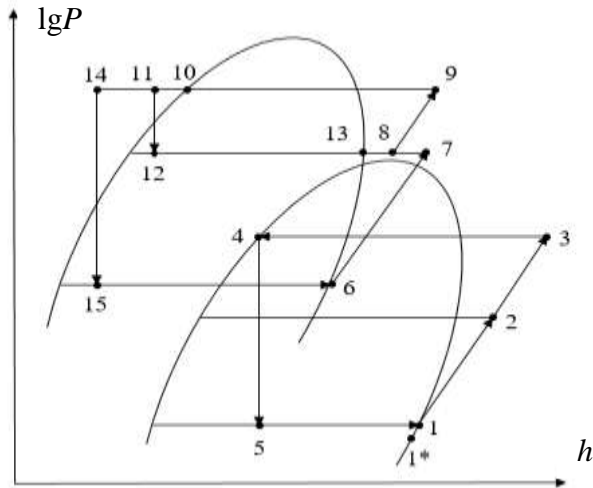

Figure 2 – *The cycle of cascade refrigeration machine in the lg P–h diagram*

Inlet stream point for Ethane to the input device of centrifugal compressor is calculated by the equation:

$$h_{1*} = h_1 + (C_a^2 / 2*1000) - (C_1^2 / 2*1000). \quad (1)$$

Specific cooling capacity of low temperature cycle, q_0':

$$q_0^{LC} = h_1 - h_5. \quad (2)$$

Specific adiabatic work of compression for the 1 stage of low temperature cycle, W_a^{LC-1}:

$$W_a^{LC-1} = h_2 - h_{1*}. \quad (3)$$

Specific adiabatic work of compression for the higher stage of low temperature cycle, W_a^{LC-2}:

$$W_a^{LC-2} = h_3 - h_2. \quad (4)$$

Specific condensing heat of low temperature cycle, q_c^{LC}:

$$q_c^{LC} = h_{3*} - h_4. \quad (5)$$

Specific actual work of compression of the 1 stage of low temperature cycle, W_R^{LC-1}:

$$W_R^{LC-1} = W_a^{LC-1} / \dot{\eta}_a'. \quad (6)$$

Specific actual work of compression for higher stage of low temperature cycle, W_R^{LC-2}:

$$W_R^{LC-2} = W_a^{LC-2} / \dot{\eta}_a'. \quad (7)$$

Point of end of the actual compression, 1-stage in the centrifugal compressor of low temperature cycle h_{2*}:

$$h_2* = h_{1*} + W_R^{LC-1}. \quad (8)$$

Point of end of the actual compression, higher-stage in the centrifugal compressor of low temperature cycle, h_{3*}:

$$h_3* = h_{2*} + W_R^{LC-1}. \quad (9)$$

Mass flow of ethane, Ma^{LC}:

$$M_a^{LC} = Q_0^{Full} / q_0^{LC}. \quad (10)$$

Adiabatic power of the centrifugal compressor, 1-stage, low temperature cycle, N_a^{LC-1}:

$$N_a^{LC-1} = M_a^{LC} * W_a^{LC-1}. \quad (11)$$

Adiabatic power of the centrifugal compressor, higher-stage, low temperature cycle, N_a^{LC-2}:

$$N_a^{LC-2} = M_a^{LC} * W_a^{LC-2}. \quad (12)$$

Actual power of the centrifugal compressor, 1-stage, low temperature cycle, N_R^{LC-1}:

$$N_e^{LC-1} = N_a^{LC-1} / \dot{\eta}_a'. \quad (13)$$

Actual power of the centrifugal compressor, higher-stage of low temperature cycle, N_R^{LC-2}:

$$N_e^{LC-2} = N_a^{LC-2} / \dot{\eta}_a'. \quad (14)$$

Specific cooling capacity of low temperature cycle, q_0^{UC}:

$$q_0^{UC} = h_6 - h_{14}. \quad (15)$$

Specific volumetric cooling capacity, high temperature cycle q_v^{UC}:

$$q_v^{UC} = q_0^{UC} / V_1. \quad (16)$$

Specific adiabatic work of compression, 1 stage, high temperature cycle, W_a^{UC-1}:

$$W_a^{UC-1} = h_7 - h_6 . \qquad (17)$$

Specific actual work of compression, 1 stage, high temperature cycle, W_R^{UC-1}:

$$W_R^{UC-1} = W_a^{UC-1} / \dot{\eta}_e , \qquad (18)$$

where $\dot{\eta}_e$" – effective efficiency of screw compressor;
Point of end of the actual compression (screw compressor), 1-stage, of high ttemperature cycle, h_{7*}:

$$h_7^* = h_6 + W_R^{UC-1} . \qquad (19)$$

Refrigerant's part required for cooling of the main flow in the economizer, y:

$$y = (h_{11} - h_{14}) / (h_{13} - h_{12}) . \qquad (20)$$

Point of start of the actual compression process (screw compressor), higher-stage, high temperature cycle, h_8:

$$h_8 = (y*h_{13} + h_7^*) / (1+y) . \qquad (21)$$

Specific adiabatic work of compression, higher- stage, high temperature cycle, W_a^{UC-2}:

$$W_a^{UC-2} = (1+y)*(h_9 - h_8) . \qquad (22)$$

Specific actual work of compression, higher- stage of high temperature cycle, W_R^{UC-2}:

$$W_R^{UC-2} = W_a^{UC2} / \dot{\eta}_e . \qquad (23)$$

Point of end of the actual compression (screw compressor), higher-stage, high temperature cycle, h_9^*:

$$h_9^* = h_8 + W_R^{UC2} . \qquad (24)$$

Specific heat from condensation process, high temperature cycle, q_c^{UC}:

$$q_c^{UC} = h_{9**} - h_{10} . \qquad (25)$$

Mass flowrate of refrigerant (ammonium), Ma^{UC}:

$$Ma^{UC} = (Ma^{LC} * q_c^{LC}) / q_0^{UC} . \qquad (26)$$

Adiabatic power (screw compressor), 1-stage, high temperature cycle, N_a^{UC-1}:

$$N_a^{UC-1} = Ma^{UC} * W_a^{UC-1} . \qquad (27)$$

Adiabatic power (screw compressor), higher-stage, high temperature cycle, N_a^{UC-2}:

$$N_a^{UC-2} = Ma^{UC} * W_a^{UC-2} . \qquad (28)$$

Actual power (screw compressor), 1-stage, high temperature cycle, N_R^{UC-1}:

$$N_e^{UC-1} = Ma^{UC} * W_R^{UC-1} . \qquad (29)$$

Actual power (screw compressor), 2-stage, high temperature cycle, N_R^{UC-2}:

$$N_e^{UC-2} = Ma^{UC} * W_R^{U-2} . \qquad (30)$$

Point of end of the actual lubricating compression (screw compressor), higher-stage, high temperature cycle, h_{9**}:

$$h_{9**} = ((1 + y)*Cp_{R717} *t_{9*} + g_м*Cp_м) / ((1 + y) * \\ * Cp_{R717} + g_м * Cp_м), \qquad (31)$$

where Cp_{R717} - isobaric heat capacity of ammonium refrigerant, kJ/ kg*°C; g_o – oil sprayed over the rotor lobes gets carried away, kg/kg; Cp_o - isobaric heat capacity of oil kJ/ kg°C.

Coefficient of performance, cascade refrigeration system, Carnot cycle, $COP_к$

$$COP_к = T_0^{LC} / (T_c^{UC} - T_0^{LC}) . \qquad (32)$$

Theoretical specific coefficient of performance, cascade refrigeration system, COP_T^{SP}:

$$COP_T^{SP} = q_0^{LC} / (W_a^{LC1} + W_a^{LC2} + W_a^{UC1} + W_a^{UC2}) \qquad (33)$$

Actual specific coefficient of performance, cascade refrigeration system, COP_R^{SP}:

$$COP_R^{SP} = q_0^{LC} / (W_R^{LC1} + W_R^{LC2} + W_R^{UC1} + \\ + W_R^{UC2}) . \qquad (34)$$

Theoretical full coefficient of performance, cascade refrigeration system:

$$COP_T^F = Q_0^F / (N_a^{LC1} + N_a^{LC2} + N_a^{UC1} + N_a^{UC2}) . \qquad (35)$$

Actual full coefficient of performance, cascade refrigeration system:

$$COP_R^F = Q_0^{пов} / (N_e^{LC1} + N_e^{LC2} + N_e^{UC1} + N_e^{UC2}) \qquad (36)$$

Theoretical specific thermodynamic perfection degree of the cascade refrigeration system:

$$\dot{\eta}_{tpd} T^{SP} = COP_T^{SP} / COP_к . \qquad (37)$$

Actual specific thermodynamic perfection degree, cascade refrigeration system:

$$\dot{\eta}_{tpd} R^{SP} = COP_R^{SP} / COP_к . \qquad (38)$$

Theoretical full thermodynamic perfection degree, cascade refrigeration system:

$$\dot{\eta}_{tpd} T^F = COP_T^F / COP_к . \qquad (39)$$

Actual full thermodynamic perfection degree, cascade refrigeration system:

$$\dot{\eta}_{tpd} R^F = COP_R^F / COP_к . \qquad (40)$$

IV. RESULTS AND DISCUSSIONS

In this section, structural optimization has been performed, the effect of the condenser subcooling on the performance of the refrigeration system operating with R717, R290, R1270, R407C, R404A, and R402A is theoretically investigated. Current refrigerants were appropriately tested in the same system and operating conditions.

During investigation following results were derived:

a) actual power of high refrigeration cycle is reduced depends on refrigerant at 3-12% respectively (Fig.3) where upper curve shows basis system and lower curve - modified system, due to reducing part stream at 20-30 % required for main stream cooling;

b) diesel fuel consumption is decreased depends on refrigerant at 3-12% respectively (Fig.4) where left column represent the basis system and right column - modified system;

c) condenser heat load is reduced at 3-13% Fig.5;

d) ozone depleting substances emission were reduced (Fig.6);

e) mass-dimensional characteristics is reduced 15 times; f) thermodynamic perfection degree is increased up to 2% Fig. 7.
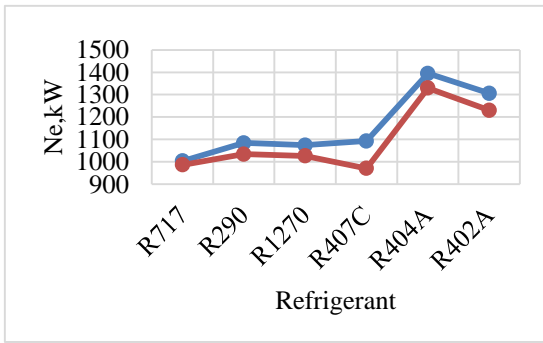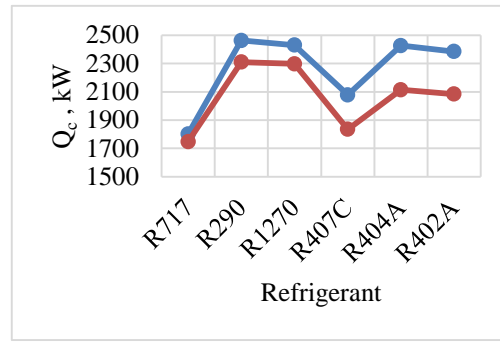
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Figure 3 – *Ne = f (R)*

Figure 5 – *Q_c = f (R):*
upper curve – base system;
lower curve – modified system.
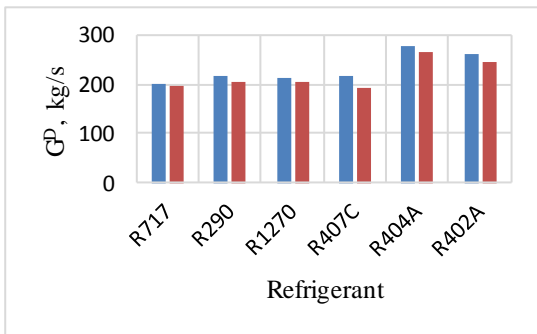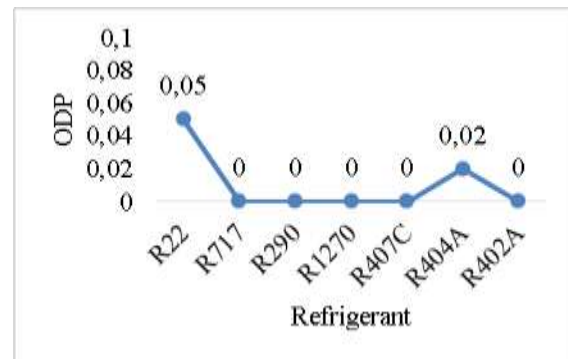

Figure 4 – *G^D = f (R)*

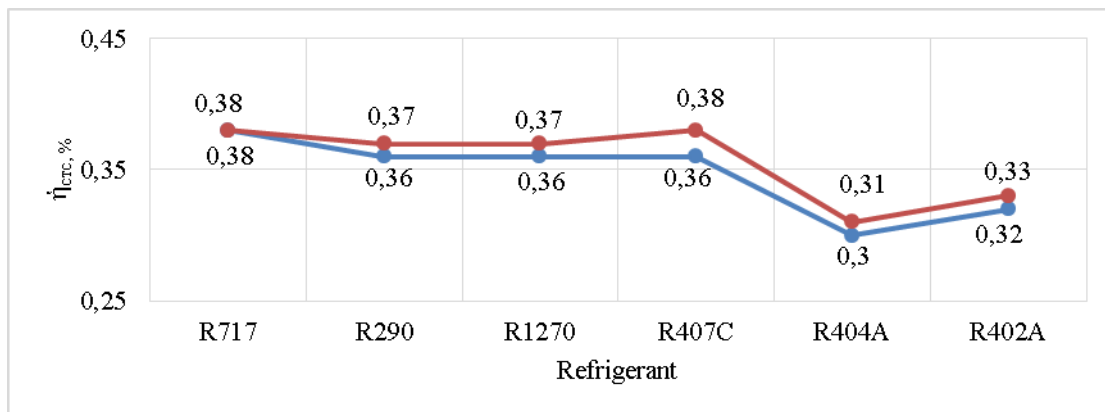**Figure6** – *ODP = f (R)*

Figure 7 - *η̇_cmc = f (XA):*
upper curve – base system; lower curve – modified system

V. CONCLUSIONS

Cascade refrigeration system for LPG re-condensation has been studied. Theoretical study on energy efficiency, energy potential and pollution prevention for marine environment has been analyzed. LPG gas carriers with 266 000 m³ ethane capacity is required 15MW cascade refrigeration system for re-condensing. The temperature in the coastal LPG storage is -70°C, and the temperature for transported Ethan is maintained at -75°C. For current storage conditions the required system cooling capacity is only 1,078 MW intended for the heat gain rejection from the environment during Ethane transportation. Numerical analysis

of the energy efficiency has shown the diesel fuel consumption reduction by 3 – 12 % (Fig.3, Fig.4). Refrigerant choice plays significant part due to adding subcooler after condenser, that in own turn helps to reduce the part stream intended for main stream cooling in the economizer by 20-30%. The viable alternative for the ozone-depleting refrigerant R22 replacement with one from chosen environmentally friendly alternative: R407C, R404A, R402A, R717, R290, R1270 was analyzed. Relevant technical and commercial opportunities for improving the energy efficiency of investigated refrigeration system have been presented.

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ИССЛЕДОВАНИЯ ПОВЫШЕНИЯ ЕНЕРГОЭФФЕКТИВНОСТИ, ЕНЕРГОПОТЕНЦИАЛА, ВЛИЯНИЯ НА ОКРУЖАЮЩУЮ СРЕДУ ХОЛОДИЛЬНОЙ УСТАНОВКИ ДЛЯ СУДНА-ГАЗОВОЗА

Вопросы повышения энергоэффективности и глобального потепления очень актуальны в эти дни из-за истощения природных ресурсов и экстремального изменения климата. Таким образом, перед нами стоит задача о разработке правил касательно выбросов в атмосферу. Эта статья представляет собой исследование каскадной холодильной установки для повторной конденсации попутного нефтяного газа при транспортировке морем газовозом. Была выполнена структурная оптимизация. Газовозу для транспортировки 266 000 м³ попутного нефтяного газа необходима 15 МВт каскадная холодильная установка для повторной конденсации, если температура хранения в береговых хранилищах равна -70°С, а температура транспортировки -75°С. Если условия хранения одинаковые, то требуемая мощность установки составляет лишь 1,078 МВт для отвода теплопритоков при транспортировке этана. Проведена оценка возможности замены озоноразрушающего хладагента R22 на альтернативные агенты R407С, R404А, R402А, R717, R290, R1270. Результаты анализа показывают, что предложенные усовершенствования могут быть использованы для оптимизации каскадной холодильной установки.

Ключевые слова: Попутный нефтяной газ; Каскадная холодильная установка; Экологически безопасный хладагент; Энерегоэффективность; Энергетический потенциал; Повторная конденсация.

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