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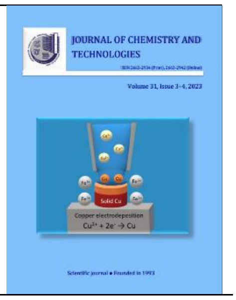

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THERMODYNAMIC CYCLE OF REGASIFICATION OF LIQUEFIED GASES WITH MECHANICAL ENERGY PRODUCTION

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Abstract

Transportation, storage and use of liquefied gases occupy a significant part in the world gas industry and maintain a tendency to increase. For economically justified maritime refrigerator transportation, the gases in the terminals of shipment are liquefied. Being delivered to the place of consumption for further use, they are converted into conventional low-pressure gas – regasified. Now this process is carried out in conventional heat exchangers, which are heated by natural sources of heat or specially preheated coolants. Liquefied gases similarly to the compressed mechanical spring contain a large amount of potential energy accumulated during liquefaction. This energy is not used in the current technological regasification process. The work offers the thermodynamic cycle of liquefied gases in which mechanical energy is obtained. Using the hydrodynamic method of transforming the liquid into a saturated steam and the isochoric process of its overheating in the proposed thermodynamic regasification cycle, the steam is repeatedly overheated and isoentropically expanding in the turbine. Combined, this allows for a lot of mechanical work to be done. Given the low temperature of the refrigeration transport of liquefied gases, the work considers the use of a combined hot heat source: the vapor is heated by the heat of the environment and then overheated to a higher temperature by specially heated water. The thermal calculations of the proposing the thermodynamic cycle of LNG (methane), which is transported at a temperature of minus 161.28 °C, and ethylene at minus 101.77 °C. The calculations performed for methane showed that the use of the method in its regasification at one of the largest regasification terminals in Europe, Barcelona, the capacity of which is 17.1 billion nm^3/year, will allow obtaining a hypothetical steam turbine plant capacity of 262 737.90 kW. Therefore, the annual energy production will be 2 295 278 302 kW. This amount of electricity requires 573 820 tons of fuel, provided that the specific fuel consumption of the diesel generator is 0.25 kg/(kW·h).

Keywords: liquefied gases, regasification terminals, regasification process, improvement of the regasification process.

ТЕРМОДИНАМІЧНИЙ ЦИКЛ РЕГАЗИФІКАЦИИ ЗРІДЖЕНИХ ГАЗІВ З ОДЕРЖАННЯМ МЕХАНІЧНОЇ ЕНЕРГІЇ

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Анотація:

Перевезення, зберігання й використання зріджених газів зараз займають значну частку у світовому газовому господарстві й зберігають тенденцію до подальшого збільшення. Для економічно виправданого морського рефрижераторного перевезення гази в терміналах відвантаження скраплюються. Будучи доставленими до місця споживання для можливості подальшого використання вони перетворюються у звичайний газ низького тиску – регазифікуються. Зараз цей процес здійснюється у звичайних теплообмінних апаратах, що обігріваються природними джерелами теплоти або попередньо спеціально нагріваємими теплоносіями. Зріджені гази подібно стиснутій механічній пружині містять велику кількість потенційної енергії, накопиченої при скрапленні. У використовуваному зараз технологічному процесі регазифікації ця енергія не використовується. У роботі пропонується термодинамічний цикл регазифікації зріджених газів, у якому отримується механічна енергії. Використовуючи гідродинамічний метод перетворення рідини в насичену пару, розглянутий нами раніше, і ізохорний процес її перегріву в пропонуємому термодинамічному циклі регазифікації, пара багаторазово ізохорно перегрівається та ізоентропно розширюється в турбіні. У сукупності це дозволяє одержувати багато механічної роботи. Враховуючи низьку температуру рефрижераторного перевезення зріджених газів, у роботі розглянуто варіант використання комбінованого гарячого джерела теплоти: спочатку пара нагрівається за рахунок теплоти навколишнього середовища, а потім перегрівається до більш високої температури спеціально нагріваємою водою.

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Виконані теплові розрахунки пропонуємого термодинамічного циклу регазифікації ЗПГ (метану), який перевозиться при температурі *мінус* 161.28 °С і етилену – *мінус* 101.77 °С. Розрахунки, які виконані для метану, показали, що використання пропонуємого методу при його регазифікації на одному з найбільших в Європі регазифікаційних терміналі «Barcelona», продуктивність якого 17.1 млрд. нм³/рік, дозволить одержувати потужність гіпотетичної паротурбінної установки 262 737.90 кВт. Отже, річне виробництво енергії складе 2 295 278 302 кВт·г. Для виробництва такої кількості електроенергії потрібно 573 820 тонн палива за умови, що питома витрата палива використовуємим дизель-генератором складає 0.25 кг/(кВт·г).
Ключові слова: зріджені гази, регазіфікаційні термінали, процес регазіфікації, удосконалення процесу регазіфікації.

Introduction

Currently, the transportation of liquefied gases with refrigerator gas pipelines, as well as the related process of regasification, is widely used and intensively improved in the world gas industry. According to [1–4], in 2016, 25 regasification terminals of liquefied natural gases (LNG regasification terminal) with a total productivity of 236.5 billion nm^3/year were operated in Europe. By the end of 2025, it is planned to build 7 terminals, thereby increasing the total productivity to 308.15 billion nm^3/year.

Liquefied natural gas regasification terminal is designed to process liquefied natural gas into regular gas under pressure maintained in the pipeline of the gas distribution network. Currently, the regasification is carried out by simply heating LNG in evaporators - heat exchangers. Depending on the temperature of the hot heat source used, evaporators are:

- evaporators operating at the ambient temperature in which the hot source of heat is water (sea or river) or air (Open Rask Vaporizer – ORV);
- evaporators operating at temperatures above the ambient temperature.

The last evaporators-heat exchangers are:

- direct heating of liquefied gas;
- heat exchangers with a circulating LNG, which is heated directly by gas burners or electricity;
- evaporators of indirect heating;
- evaporators located in a water bath, which is heated by gas burners (Submerged Combustion Vaporizer - SCV); At the same time, heat exchangers of this type consume up to 1.5 % of regasification gas for their own operation.;
- Waste Heat Recovery LNG Vaporizer (WHRV), powered by the heat of gases exhausted in an electric generator turbine through an intermediate coolant circuit.

The most widely used evaporators are ORV and SCV, so they are used in this work.

On the one hand, it is known that the principle of operation of a heat machine, which turns heat into work, is based on the preliminary creation of a positive potential difference (temperature and pressure) between the working body and the environment. To do this, the working body in such heat converters is pre-compressed and heated, i.e. its pressure and temperature relative to the environment increase, thereby creating a positive potential difference. To realize this process, heat and work are consumed in thermal machines - the heat of combustion of fuel and the mechanical energy of the pump compressing the working body [5–7].

On the other hand, it is known that in order to optimize the process of transportation and storage of gases (ammonia, ethane, ethylene, methane, etc.) they are liquefied, consuming a lot of thermal and mechanical energy, which accumulates in the liquefied gases, i.e. a negative potential is created between the liquefied gas and the environment [8–12].

This potential is not used in the current liquefied gas regasification process. This work examines one of the possible options for using this negative potential to obtain mechanical energy when using a combined hot heat source in the heating process: first, the heat of the environment (water/air, ORV type evaporator), and then – SCV evaporator heat [13; 14].

The thermodynamic cycle of regasification of liquefied gases with the production of mechanical work and its schematic diagram

The proposed thermodynamic cycle of regasification of liquefied gases with the production of mechanical work is shown in Fig. 1, and the schematic diagram of the installation in which it is implemented – in Fig. 2. Let's consider the proposed cycle of regasification for liquefied natural gases (methane). Since methane is transported and stored in a saturated state at pressure p_1 = 0.103 MPa, its thermodynamic properties at point 1 (Fig. 1) are equal to: temperature t_1 = –161.28 °C, enthalpy h_1 = 0.69687 kJ/kg and density ρ_1 = 422.06 kg/m^3.

As a basic process of regasification, a pure ORV process is taken. In the proposed thermodynamic cycle we use a combined hot source of heat: in processes 2-3-4-4-4′, 6-6′ and 8-8′ we use the ORV heat exchanger with a coolant temperature at the entrance of +20 °C, and in the processes 4′-5, 6′-7 and 8′-9, which are appropriate extensions of these processes, the SCV heat exchanger with the

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temperature of the coolant at the entrance of +110 °C. We use the hydrodynamic method of transforming fluid into a saturated steam with the isochoric process of its overheating [2]. Finally, let us assumethe temperature and pressure of overheated steam, which enters the blades of the first steam turbine, equal to: $t_5 = 10$ °C and $p_5 = 3$ MPa (Fig. 1). Using the REFPROP program [15], we determine the thermodynamic properties of LNG (methane) necessary for further calculations. The determined thermodynamic properties of methane are summarized in Table 1.

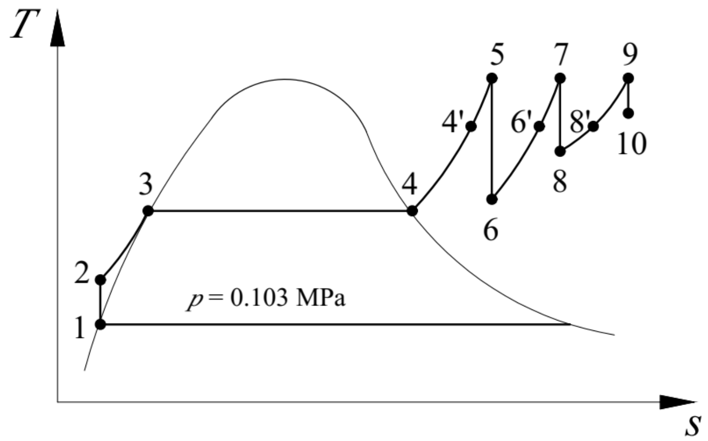

Fig. 1. Thermodynamic cycle of liquefied gas regasification with obtaining mechanical energy

Since at the hydrodynamic method of converting fluid into steam its overheating is carried out isochorically, on the T, S-diagram there is a "dip" from the accepted point 5 by ischor $v_5 = 0.045898$ m^3/kg to the boundary curve of saturated vapor (point 4, Fig. 1), we determine temperature and evaporation pressure: $t_{3,4} = -116.67$ °C, and $p_{3,4} = 1.38$ MPa. Since the conversion of boiling fluid to saturated steam is carried out isobarically, the liquid must be pre-compressed to this pressure and heated to saturation. Given the isentropic process of its compression, the enthalpy of the compressed fluid (point 2, Fig. 1) is 3.7161 kJ/kg, and the pump work is 3.02 kJ/kg.

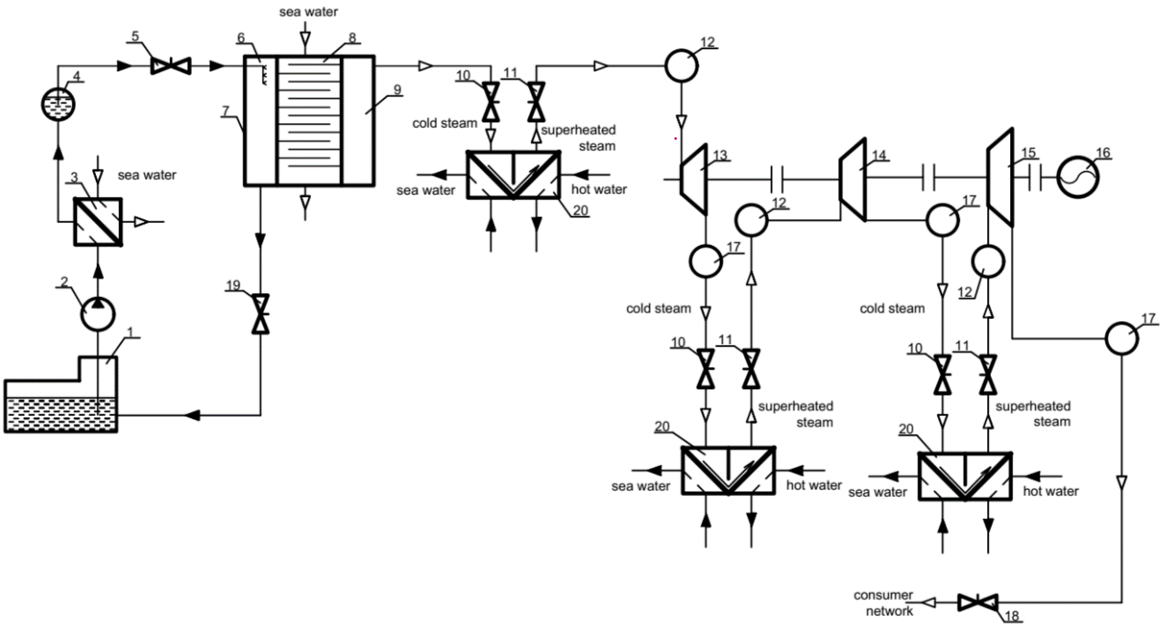

Fig. 2. Schematic diagram of the installation: 1 – gas storage tank; 2 – pump; 3 – heat exchanger; 4 – saturated liquid receiver; 5 – automatic pulse valve (the amount of liquid and the duration of its supply to the nozzle 6); 7 – a device for liquid boiling; 8 – ribbed from the middle of the chamber, which feeds a hot source of heat; 9 – the fluid to steam conversion chamber; 10 – automatic impulse valves (the amount and duration of steam feeding into two-stage overheaters 20); 11 – impulse valves (the amount and duration of superheated steam feeding to receivers 12); 13, 14, 15 – steam turbines; 16 – mechanical energy consumer (generator); 17 – recipients of the exhausted steam; 18 – pulse valve supplying low pressure gas to the main network of consumers; 19 – pulse valve that returns the fluid that has been evaporated to the gas storage tank

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The expansion of the steam in the first turbine 13 (process 5–6) is carried out isoentropically to such pressure that its temperature at the end of the expansion does not lower below −40 °C to ensure the cold resistance of the turbine blades. The same conditions should be observed in the expansion of steam in the following turbines - in processes 7–8 and 9–10. The overheating of the steam between the turbines is carried out in intermediate isochoric vapor overheaters, the principle of operation of which is considered in [2].

Certain thermodynamic properties of methane at all characteristic points of the process of its conversion from fluid into a gaseous state, in which it is transmitted to the gas distribution networks, with multi-stage use of its vapor in power turbines, are given in Table 1.

Table 1

Thermodynamic properties of liquefied methane at characteristic points of proposed thermodynamic cycle of regasification using its potential energy

points	*t*, °C	*p*, MPa	ρ, kg/m^3	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	−161.28	0.103	422.06	0.45283	0.69687	0.0061996
2	−161.04	1.0071	422.57	0.45578	3.7161	0.0061996
3	−123.86	1.0071	359.31	136.98	139.78	1.0576
4	−123.86	1.0071	15.810	491.19	554.89	3.8381
4′	10	2.2140	15.810	713.60	853.64	49023
5	100	3.0	15.810	876.39	1066.10	5.4006
6	−40	0.41167	3.4635	645.23	764.09	5.4006
6′	10	0.50295	3.4635	727.19	872.41	5.7189
7	100	0.66683	3.4635	888.60	1081.10	6.2123
8	−31.09	0.103	0.82399	662.75	787.76	6.2123
8′	10	0.12062	0.82399	730.13	876.52	6.4693
9	100	0.15918	0.82399	891.23	1084.4	6.9617
10	66.26	0.103	0.58617	828.41	1004.1	6.9617

Using tabular data, we calculate the energy characteristics of the thermodynamic cycle of regasification:

– specific work of the turbine of the regasification unit

$$l_{\text{turbine}} = l_{5,6} + l_{7,8} + l_{9,10} - l_{\text{н}} = (h_5 - h_6) + (h_7 - h_8) + (h_9 - h_{10}) - l_{\text{н}} =$$
$$= (1066.1 - 764.09) + (1081.1 - 787.76) + (1084.4 - 1004.2) - 2.14 =$$
$$= 673.51 \text{ kJ/kg};$$

– specific heat supplied in heat exchangers ORV and SCV

$$q_1^{\text{orv+scv}} = q_{2,4} + q_{4,5} + q_{6,7} + q_{8,9},$$

that is

$$q_1^{\text{orv+scv}} = (h_4 - h_2) + (u_5 - u_4) + (u_7 - u_6) + (u_9 - u_8) = (554.89 - 2.8377) +$$
$$+(876.39 - 491.19) + +(888.60 - 645.23) + (891.23 - 662.75) = 1409.10 \text{ kJ/kg};$$

– thermodynamic efficiency of steam turbine installation taking into account the total heat supplied in combined heat exchangers ORV and SCV

$$\eta_{\text{t}} = \frac{l_{\text{turbine}}}{q_1^{\text{orv+scv}}} = \frac{673{,}51}{1409.10} = 0.4780.$$

The peculiarity of the proposed thermodynamic cycle of LNG regasification is the use as hot sources of heat of two physical carriers: ORV and SCV.

Then, the specific heat that is supplied in heat exchangers SCV:

$$q_1^{\text{scv}} = \text{q}_{4',5} + \text{q}_{6',7} + \text{q}_{8',9},$$

that is

$$q_1^{\text{scv}} = (u_5 - u_{4'}) + (u_7 - u_{6'}) + (u_9 - u_{8'}) = (876.39 - 713.60) + (888.60 - 727.19) +$$
$$+(891.23 - 730.13) = 324.2 \text{ kJ/kg}.$$

The ratio of the heat input entering the heat exchangers of SCV to the total heat entering both types of ORV and SCV heat exchangers in series:

$$\delta q_1^{\text{scv}} = \frac{q_1^{\text{scv}}}{q_1^{\text{orv+scv}}} = \frac{324.20}{1409.10} \cdot 100 = 23.0 \text{ \%}.$$

Thus, the heat input is less than ¼ of the total heat generated in the proposed thermodynamic LNG regasification cycle. The use of secondary low-temperature heat sources as a hot heat source (cooling water or engine exhaust gases) will significantly reduce the cost of the LNG regasification process and increase the capacity and efficiency of the thermodynamic

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regasification cycle of all liquefied gases transported by sea [16-18].

Lrt us Calculate the theoretical mechanical capacity of a conditional terminal with a capacity of 1.0 billion nm3/year equipped with a device manufactured in accordance with the proposed thermodynamic regasification cycle. The mass gas throughput of such terminal is equal to

$$M_1 = \frac{1 \cdot 10^9}{365} = 0.002739726 \cdot 10^9\ \mathrm{m^3/day} = 0.00011416 \cdot 10^9\ \mathrm{m^3/hour} =$$
$$= 0.031709792 \cdot 10^3\ \mathrm{m^3/sec} = 22.751\ \mathrm{kg/sec},$$

where the coefficient 0.71746 kg/nm3 – methane gas density under normal conditions (0 °C and 1.0325 bar).

Then the mechanical power of such a terminal

$$N_1 = M_1 \cdot l_{\mathrm{turbine}} = 22.751 \cdot 673{,}51 = 15323.03\ \mathrm{kW}.$$

Currently, one of the largest regasification terminals in Europe is Barcelona in Spain, with an annual productivity of 17.1 billion nm3/year of LNG. Then the mechanical capacity of the hypothetical steam turbine installation that could be installed at this terminal is 262 737.90 kW (263 MW). Thus, the annual energy production will be 2 295 278 302 kW. To produce this amount of electricity, 573,820 tons of fuel are needed, assuming that the specific fuel consumption of a diesel generator in operation is 0.25 kg/(kW·h).

The total productivity of the LNG terminals in Europe by the end of 2025 will reach 308.15 billion nm3/year. Then the total capacity of hypothetical steam turbines that can be installed at all regasification terminals of Europe equipped in accordance with the proposed thermodynamic regasification process, will be 4 734 656.957 kW, which corresponds to the annual energy production of 4,136196318·10[10] kW.

To produce this amount of electricity requires 10 340 491 tons of fuel.

Thermodynamic process of regasification of liquefied ethylene with production of mechanical work

In addition to LNG, other liquefied gases (ethan, ethylene, butane, etc.) are transported by gas carriers. They are transported/stored at higher temperatures. It is interesting to find out how much energy can be obtained in the regasification of such gases [19, 20].

For example, let's look at the regasification of liquefied ethylene, which is carried out in the proposed method. To find the dependence of the efficiency of the proposed regasification method on the type of liquefied gas, we preserve the conditions of transportation (p_1 = 0.103 MPa) and parameters of operation of the first turbine (p_5 = 3.0 MPa i t_5 = 100 °C).

Thermodynamic properties of ethylene in characteristic points of offered thermodynamic regasification cycle were also determined using the REFPROP program [15] and are shown in Table 2.

Table 2

Thermodynamic properties of ethylene in characteristic points of proposed thermodynamic regasification cycle

points	*t*, °C	*p*, MPa	ρ, kg/m3	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	–103.77	0.103	567.65	–0.17850	0.0	0.0
2	–103.31	1.5997	568.53	–0.17621	2.6376	0.0
3	–36.726	1.5997	453.99	170.73	174.25	0.84811
4	–36.726	1.5997	29.598	468.74	522.79	2.3223
4′	10	2.0969	29.598	526.39	597.23	2.5447
5	100	3.0	29.598	651.47	752.85	2.9267
6	–40	0.3022	4.5425	494.10	560.63	2.9267
6′	10	0.37165	4.5425	550.20	632.02	3.1443
7	100	0.49554	4.5425	671.20	780.29	3.5134
8	6	0.10347	1.2597	548.64	630.78	3.5134
8′	10	0.10497	1.2597	553.35	636.69	3.5302
9	100	0.13879	1.2597	673.84	784.01	3.8977
10	82	0.10360	1.0126	647.68	752.58	3.8977

Using tabular data, we calculate the energy characteristics of the thermodynamic process:

– the total specific work of turbines of a regasification steam turbine installation

$$l_{\mathrm{turbine}} = l_{5,6} + l_{7,8} + l_{9,10} - l_{\mathrm{н}} = (h_5 - h_6) + (h_7 - h_8) + (h_9 - h_{10}) - (h_2 - h_1) =$$
$$= (752.85 - 560.63) + (780.29 - 630.78) + (784.01 - 752.58) - (2.6376 - 0.0) =$$
$$= 370.52\ \mathrm{kJ/kg};$$

– specific heat that is supplied in heat exchangers ORV and SCV

$$q_1^{\mathrm{orv+scv}} = q_{2,4} + q_{4,5} + q_{6,7} + q_{8,9},$$

that is

– thermodynamic efficiency of steam turbine installation taking into account the total heat supplied in combined heat exchangers ORV and SCV

$$q_1^{\mathrm{orv+scv}} = (h_4 - h_2) + (u_5 - u_4) + (u_7 - u_6) + (u_9 - u_8) =$$
$$= (522.79 - 2.6376) + (651.47 - 468.74) +$$
$$+(671.20 - 494.10) + (673.84 - 548.64) = 1005.18\ \mathrm{kJ/kg}.$$

The peculiarity of the proposed thermodynamic cycle of LNG regasification is the use as hot sources of heat of two physical carriers: ORV and SCV.

Then, the specific heat that is supplied in heat exchangers SCV:

$$q_1^{\mathrm{scv}} = q_{4',5} + q_{6',7} + q_{8',9},$$

that is

$$q_1^{\mathrm{scv}} = (u_5 - u_{4'}) + (u_7 - u_{6'}) + (u_9 - u_{8'}) = (651.47 - 526.39) + (671.20 - 550.20) +$$
$$+(673.84 - 553.35) = 366.57\ \mathrm{kJ/kg}.$$

The ratio of the heat input entering the heat exchangers of SCV to the total heat entering both types of ORV and SCV heat exchangers:

$$\delta q_1^{\mathrm{scv}} = \frac{q_1^{\mathrm{scv}}}{q_1^{\mathrm{orv+scv}}} = \frac{366.57}{370.52} \cdot 100 = 98.9\ \%.$$

Thus, the heat input is almost 100 % of the total heat generated in the proposed thermodynamic cycle of ethylene regasification.

We calculate the theoretical mechanical capacity of the conditional ethylene terminal with gas productivity of 1.0 billion nm³/year, equipped with an installation, manufactured in accordance with the proposed thermodynamic cycle of regasification. Mass productivity on gaseous ethylene of such terminal is equal to

$$M_1 = \frac{1 \cdot 10^9}{365} = 0.002739726 \cdot 10^9\ \mathrm{m^3/day} = 0.00011416 \cdot 10^9\ \mathrm{m^3/hour} =$$
$$= 0.031709792 \cdot 10^3\ \mathrm{m^3/sec} = 39.989\ \mathrm{kg/sec},$$

where the coefficient 1.2611 kg/nm³ – the density of gaseous ethylene under normal conditions (0 °C and 1.0325 bar).

Then the mechanical power of such a terminal

$$N_1 = M_1 \cdot \mathrm{l_{turbine}} = 39.989 \cdot 370{,}52 = 14816.72\ \mathrm{kW}.$$

Thus, the temperature of transport of ethylene is 58.5 % higher than that of methane, and the molecular weight of ethylene is 65% higher, so the capacity of their conditional regasification terminals with a gas productivity of 1.0 billion nm³/year differs only by 3.4 %.

That is, the lower the temperature of transportation of liquefied gas and the higher its molecular weight, the more energy can be obtained using the thermodynamic cycle of its regasification.

Conclusion

The thermal calculations suggested by the thermodynamic cycle of the liquefied gases regasification indicate large reserves of the potential energy contained in them but not currently utilized. The design capacity of LNG terminals in Europe alone is impressive. If one takes into consideration the entire world's regasification terminals and all liquefied gases transported by sea, the efficiency of the proposed regasification method would be enormous.

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