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RESULTS OF THE GAS CARRIER RELIQUEFACTION PLANT TRIAL

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Abstract. In the paper results of the gas carrier reliquefaction plant trial are considered. Safe transportation of liquefied gases is explained. The construction of the ship on trial is described. Designed parameters of the reliquefaction plant are presented. Heat gain into cargo tanks is obtained. Volumetric capacity, cooling capacity, volumetric efficiency and power consumption of the compressors are determined. Results of the main engine trial, diesel generator trial, reliquefaction plant trial, and calculations performed afterwards are represented in five tables. The results obtained may be used for optimisation calculations of gas carriers' reliquefaction plants.

Keywords: liquefied gas, trial, reliquefaction plant, heat gain, refrigerating capacity, volumetric efficiency, fuel oil consumption.

1. Introduction

The world carriage of liquefied hydrocarbon gases and their derivatives (methane, ethane, propane, butane, ethylene, propylene etc.) which are widely used in chemical and power engineering is mainly accomplished by sea, in bulk, on board the specialised ships called gas carriers [1, 2].

The recent growth in liquefied gas transportation results in intensified use of gas carriers and the construction of new vessels with sizeable cargo tank capacities.

In order to minimize the wall thickness and mass of the cargo tanks, and still ensure the safe transportation of the flammable substances, the gases are carried in a liquefied state, at atmospheric pressure, with corresponding low temperatures that are obtained by reliquefaction plants.

2. Ship's and reliquefaction plant's characteristics

The gas carrier used for the trial was built in 2006, at the Mitsubishi shipyard, in Japan. She is designed for the transportation of liquefied propane and butane, as well as their mixtures. The ship has four insulated cargo tanks, with a total capacity of 83 000 m^3. The cargo tanks are insulated with polyurethane foam, at a thickness of 100 mm.

Power of the main engine is: N_{me} = 11 660 kW, and the designed total mass of the ship, together with cargo, fuel and provision is: M_c = 74 000 t.

The reliquefaction plant (RP) is designed for the following working conditions: air ambient temperature: t_a +45 °C; seawater temperature: t_w = +32 °C; cargo temperature: t_c = –46 °C; air temperature in the upper part of the tank hold: t_{hu} = +36 °C; air temperature in the lower part of the tank hold: t_{hl} = +25 °C; the cargo tanks are filled to 98 %.

RP consists of five two-stage two-cylinder horizontal reciprocating double acting compressors. The technical data of the compressors is as follows: low pressure stage cylinder diameter: D_L = 445 mm; high pressure cylinder diameter: D_H = 240 mm; piston stroke: S = 200 mm; electric motor speed of rotation: n = 9.75 s^{-1} (585 rpm); swept volume of the low pressure stage: V_{hl} = 0.6 m^3/s (2155 m^3/h); swept volume of the high pressure stage: V_{hh} = 0.17 m^3/s (608 m^3/h). The cooling capacity at the design conditions is: Q_0 = 219.8 kW.

In order to avoid cargo contamination the compressors are oil-free. This means that there is no lubrication of the cylinders and gas passing is minimised by using teflon rings.

The compressors' electric motors are located in the gas-safe electric motor room and the shaft penetrates the bulkhead by means of a gas-tight seal. The electric motor power is: N_e = 220 kW.

Each compressor has its own sea water cooled condenser with a heat exchanging surface of 90 m^2.

There is no intermediate cooling of the gas between the stages of the compressors. After compression in the low pressure stage the gas is delivered directly to the high pressure stage for further compression.

In Table 1 the heat gain calculation results are given. The calculation is made for designed working conditions of the reliquefaction plant.

The total heat gain into the cargo tank is obtained from the following equation.

Table 1. Heat gain calculation results

Heat gain through	Cargo tank No				Sum total
	1	2	3	4	
General part Q_G, W	71 963	78 664	78 664	76 275	305 506
Anchor block, Q_A, W	981	912	912	912	3 717
Bearing block, Q_B, W	2 495	3 209	3 209	2 943	11 856
Trunk top, Q_{TR}, W	542	542	542	542	2 168
Anti-floating chock, Q_{AF}, W	1 466	1 690	1 690	1 466	6 312
Q_{TOTAL}, W	77 447	85 017	85 017	82 138	329 619

$$Q_{TOTAL} = Q_G + Q_A + Q_B + Q_{TR} + Q_{AF}, \quad (1)$$

where: Q_G – heat gain through general part of insulation, W; Q_A – heat gain through anchor block, W; Q_B – heat gain through bearing block, W; Q_{TR} – heat gain through trunk top, W; Q_{AF} – heat gain through anti-floating chock, W.

Heat gain Q was obtained from the heat transfer equation. The overall heat transfer coefficient for each element of the cargo tank construction where heat transfer takes place was preliminarily calculated. Their values in W/m²·K correspondingly are: $K_G = 0.216$; $K_A = 2.107$; $K_B = 1.615$; $K_{TR} = 0.230$; and $K_{AF} = 3.5$.

It is usual in design practice that nominal heat gain has about 15 % margin against calculated heat gain, due to various factors. So, nominal heat gain is obtained as follows:

$$Q^N_{TOTAL} = 1.15 \cdot Q_{TOTAL} = 1.15 \cdot 329\,619 = 379\,062 \text{ W}.$$

The dryness of propane, after throttling via the expansion valve into the cargo tanks, is: $x = 0.44$. So, the quantity of boil off gas produced under propane loaded condition, due to heat gain into the cargo tank from outside can be obtained from the following equation:

$$M_{BOG} = \frac{Q^N_{TOTAL}}{r \cdot (1 - x)},$$

where: $r = 425{,}900$ J/kg – latent heat of vaporization of propane at $t_c = -46$ °C.

$$M_{BOG} = \frac{379\,062}{425\,900 \cdot (1 - 0.44)} = 1.59 \text{ kg/s}.$$

If superheating of the propane vapour on the suction side of the compressor the low pressure stage is: $\Delta T_{SUC} = 15$ °C, specific volume: $v = 0.45$ m³/kg. These conditions require compressor volume capacity as follows:

$$V^R_C = M_{BOG} \cdot v = 1.59 \cdot 0.45 = 0.72 \text{ m}^3\text{/s}.$$

The sum of the swept volume of the low pressure stage is:

$$\sum V_{hl} = 5 \cdot 0.6 = 3 \text{ m}^3\text{/s}.$$

As we can see, the sum of the swept volume of the low pressure stage is significantly larger than that required by the compressor capacity. This fact shows that there is either a large reserve in the compressor's capacity or there are excessive volumetric losses in the compressor.

3. The trials

The reliquefaction plant, main engine and diesel generator trials were performed in order to establish technical characteristics of the reliquefaction plant compressors and fuel rates for the electrical power generation and ship propulsion.

Volumetric capacity, cooling capacity, volumetric efficiency and power consumption of the compressors are determined as a result of the reliquefaction plant trial.

During the trial all working parameters of the reliquefaction plant and the filling time of the condenser receiver with a fixed volume of: $V_f = 0.0126$ m³ were measured.

The trial took place with a seawater temperature $t_w = 14$ °C and an ambient air temperature $t_a = 25$ °C. Sea water heat transfer in the condenser was 3 °C. Transported cargo was commercial propane. According to the Certificate of Quality, the fraction of ethane gas of: $x_e = 3.1$ % and the fraction of propane gas of: $x_p = 96.9$ %. RP working parameters during the trial are shown in Table 2.

Table 2. Results of reliquefaction plant trial

Parameters	Compressor No				
	1	2	3	4	5
Low pressure stage					
Suction pressure, kPa	11	11	11	10	11
Discharge pressure, MPa	0.3	0.29	0.28	0.29	0.31
Suction temperature, °C	+2	−6	−1	−2	−7
Discharge temperature, °C	78	76	68	83	89
High pressure stage					
Discharge pressure, MPa	0.85	0.85	0.80	0.85	0.90
Condensate temperature, °C	15	15	16	15	16
Level increasing time, s	6.75	6.45	6.20	6.55	6.31

Condensate quantity is calculated by the following formula:

$$M = \frac{V_f \cdot \rho_c}{\tau}, \quad (2)$$

where: ρ_c – is cargo density given by loading terminal.

Volumetric capacity of compressor is determined as follows:

Table 3. Trial conditions and design sonditions of the reliquefaction plant

Compressor performance	Compressor number				
	1	2	3	4	5
Density of condensate in the condenser receiver part ρ_c, kg/m³	506	504	504	506	504
Condensate quantity, M, kg/s	0.944	0.984	1.024	0.973	1.006
Specific volume of ethane vapour at compressor suction v_e, kg/m³	0.672	0.648	0.658	0.670	0.645
Specific volume of propane vapour at compressor suction v_p, kg/m³	0.454	0.438	0.446	0.453	0.436
Volume capacity of the compressor at the trial V, m³/s	0.433	0.436	0.461	0.445	0.443
Estimated volume capacity of the compressor at the trial conditions V_1, m³/s	0.418	0.418	0.421	0.417	0.416
Estimated volume capacity of the compressor at the design conditions V_d, m³/s	0.392	0.394	0.414	0.403	0.402
Volumetric efficiency of the compressor at the trial λ	0.56	0.57	0.60	0.59	0.58
Estimated volumetric efficiency of the compressor at the trial conditions λ_1	0.54	0.54	0.55	0.54	0.54
Estimated volumetric efficiency of the compressor at design point λ_d	0.51	0.51	0.54	0.52	0.52
Enthalpy of ethane vapour at compressor suction h_{esuc}, kJ/kg	803.2	789.8	798,2	796.5	788.1
Enthalpy of liquid ethane in condenser receiver part h_{eliq}, kJ/kg	473.5	477.7	477.7	473.5	477.7
Enthalpy of propane vapour at compressor suction h_{psuc}, kJ/kg	492.7	480.6	489.0	487.3	478.9
Enthalpy of liquid propane in condenser receiver part h_{pliq}, kJ/kg	142.5	145.8	145.8	142.5	145.8
Refrigeration capacity of reliquefaction plant Q_0, kW	330.1	328.8	350.7	334.5	334.6
Refrigeration capacity of reliquefaction plant at design point Q_0, kW	227.8	229.2	241.1	234.7	234.2
Required power of compressor at the trial N, kW	152	152	152	152	156
Estimated required power of compressor at design point N_d, kW	203.7	203.7	207.8	206.4	206.4

$$V = \frac{M}{\dfrac{x_e}{v_e} + \dfrac{x_p}{v_p}}, \quad (3)$$

where: v_e and v_p – ethane and propane vapour specific volume at suction side of the compressor, m³/kg.

Refrigeration capacity of reliquefaction plant is computed by the formula:

$$Q_0 = V \cdot \left[\frac{x_e \cdot \left(h_{esuc} - h_{eliq}\right)}{v_e} + \frac{x_p \cdot \left(h_{psuc} - h_{pliq}\right)}{v_p} \right], \quad (4)$$

where: h_{esuc} – enthalpy of ethane vapour on suction side of the compressor, kJ/kg; h_{eliq} – enthalpy of liquid ethane in condenser receiver part, kJ/kg; h_{psuc} – enthalpy of propane vapour on suction side of the compressor, kJ/kg; h_{pliq} – enthalpy of liquid propane in condenser receiver part, kJ/kg.

Compressor volumetric efficiency λ is determined as the ratio of its volumetric capacity to the sum of the swept volumes of both stages of the compressors.

$$\lambda = \frac{V}{V_h}, \quad (5)$$

where: $V_h = V_{hl} + V_{hh} = 0.60 + 0.17 = 0.77$ m³/kg.

Outcome of the calculations made for the trial conditions and design conditions are shown in Table 3.

Results of the trial represented in Table 3 are evidence of a sizeable reserve of the reliquefaction plant capacity and of significant volumetric losses in the compressors. Volumetric efficiency at the trial conditions was: λ = 0.56 … 0.60, and λ = 0.51 … 0.54 at the design conditions; the difference caused by higher condensation pressures. Significant volumetric losses are explained by leakage of the compressed gas to the suction side due to oil-free compression and by increased volume of the

Table 4. Outcome of the main engine trial

Characteristics	Main engine load, %			
	50	75	90	100
Main engine power N_{me}, kW	6 810	10 110	12 410	13 670
Revolutions per minute n_{me}, rpm	87.3	99.8	106.7	110.3
Maximum cylinder pressure p_{max}, MPa	9.4	12.6	13.6	13.5
Exhaust gases temperature t_{exh}, °C	302	303	323	343
Specific fuel oil rate for power unit, $\frac{g}{kW \cdot h}$	181.9	173.3	172.7	174.5

Table 5. Fuel oil consumption based on trial data

Characteristics	Values
Fuel oil volume rate measured by flowmeter, m³/h	2.417
Correction factor for flowmeter	0.9975
Real fuel oil volume rate, m³/h	2.411
Fuel oil temperature, °C	89
Fuel oil density, kg/m³	997.5
Fuel oil mass rate, kg/h	2 214
Main engine power N_{me}, kW	11 660
Specific fuel oil rate for power unit, g^{fo}_{me}, $\frac{g}{kW \cdot h}$	181.3
Fuel oil calorific value, kJ/kg	40 560
Standard fuel oil calorific value, kJ/kg	42 700
Specific rate of standard fuel oil for power unit, $\frac{g}{kW \cdot h}$	171.6

dead space due to the large number of suction and discharge valves.

The reliquefaction plant trial proved that the actual compressor characteristics correspond to the design values and even exceed them. For example, in the actual compressor the design cooling capacity parameters are exceeded by 8 … 21 kW, whilst the actual power requirements are 12.2 … 16.3 kW less than those for the theoretical compressor.

Outcome of the main engine trial is presented in Tables 4 and 5.

Using trial data shown in Table 4, and considering full mass of ship: M_s = 7.4·104 tons, it is easy to calculate fuel oil consumption for moving this mass. Determined that, depending upon main engine load, this value is:

$$g_M^{fo} = 0.017...0.032 \frac{kg^{fo}}{t \cdot h}.$$

4. Conclusions

As shown by the main engine trial, the specific rate of standard fuel oil is 166.5 … 175.5 g/kW·h, with a minimum load of 90%.

During diesel-generators trials the obtained specific rate of standard fuel oil at 75 … 110 % is 193.7 … 196.6 g/kW·h with a minimum load of 100 %.

The specific values of standard fuel oil may be used for optimisation calculations of gas carrier reliquefaction plants.

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