PAPER ID: 0022
DOI: 10.18462/iir.cryo.2019.0022

Refrigeration Linde-Hampson Machine at 90K Operation by Use of Ozone Friendly Refrigerants

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ABSTRACT

The object of this study is a laboratory functional model of the low-temperature Joule-Thomson refrigerating machine operating on the Linde-Hampson cycle. A multi-component zeotropic mixture of hydrocarbons and some other ozone friendly substances is used as the working fluid. A hermetic lubricated compressor with corresponding after-cooling unit, an advanced oil-removing system, an internal heat exchanger of high-efficiency, a throttling device and evaporator make the main components of the functional model. Specific composition of the working fluid, hydraulic, temperature and thermal operating parameters of the refrigerator are calculated using an original algorithm of the mixed refrigerant Linde-Hampson cycle thermodynamic calculations elaborated at the ILK Dresden. Using the optimal calculated parameters, the functional model has been developed and experimentally tested. The computational procedure, some design features of the functional model and preliminary results of experimental testing are reported in the present paper.

Keywords: Refrigeration, Linde-Hampson machine, hermetic lubricated compressor, mixed refrigerant, ozone friendly substances, zeotropic mixture of hydrocarbons

1. INTRODUCTION

In recent years significant progress has been made in numerical computations of thermodynamic and transport properties of refrigerants, their mixtures, as well as of a considerable number of substances that had not been previously considered as working fluids for refrigerators (hydrocarbons, inert gases and their mixtures of all kinds). A number of quite reliable computer programs like NIST RefProp 9.1 (NIST, 2018a), NIST SUPERTRAPP 3.2 (NIST, 2018b) and PROSIM (prosim, 2018) became available. Now it is possible to implement the procedures calculating thermophysical properties of various substances and their mixtures directly into computer programs intended for optimization design of refrigerating systems.

This technology is especially relevant for the design of mixed refrigerant low-temperature refrigerating machines operating on either the Linde or the auto-cascade cycle (Rui, S., et al., 2016) − also known as the Kleemenko cycle. These machines are quite simple in their design and, to some extent, a modification of the classic Joule-Thompson (JT) refrigerating machine. In such machines, zeotropic mixtures of substances are used as the working fluid. The temperature glide along an isobar in two-phase envelope is a characteristic feature of those mixtures. That effect enables low temperatures (down to 90-100 K) at moderate values of pressure ratios of the working fluid. So now, depending on the required temperature level and cooling capacity of the low-temperature refrigerating machine the right working fluid and right thermal and hydraulic operational parameters can be calculated. Using such a method of optimization of mixed refrigerant refrigerating machines results in considerable improvement in power, dimensional, mass, and economic performances.

ILK Dresden has successfully developed the algorithms and computer models of the mixed refrigerant JT refrigerating machines. With these computer programs the laboratory functional model of the low temperature Linde-Hampson refrigerating machine designed for a gas stream cooling at the temperature level of 100…120 K has been successfully developed. The ecological compatibility of the system is ensured by the use of hydrocarbons as a component of the working mixture. The energy efficiency of the refrigeration machine is ensured by the selection of the optimum zeotropic working mixture of refrigerants (the number of components, the concentration of each component) and the thermal and hydraulic parameters of the system.

The optimal values of the operating parameters of the refrigeration machine will allow most efficient heat exchange equipment and thereby minimize the internal volume of the system and the mass of the refrigerant charge. So the system will be very compact and fire safe (the mass of hydrocarbon refuelling is planned to be less than 150 g, corresponding to the standard IEC 60335-2-89:2010 (IEC, 2010)). The maximum possible use of standard equipment (compressor, heat exchangers, filter-dryers, etc.) produced massively and widely available on the market was a main criteria in the development of such a refrigerating machine.

Such effective, space-saving and environmentally friendly low-temperature refrigerating machines can be in demand as an integral part of various technological processes, medicine, science, electronics, etc. In our case, the refrigerating machine has been developed with the purpose of a gas (N_2, hydrocarbons, etc.) stream cooling, as reflected in its specific design reported hereafter.

2. BASIC UNITS OF THE LINDE–HAMPSON REFRIGERATING MACHINE

An admissible schematic of a system designed for cooling and partial liquefaction of a steady stream of gaseous nitrogen using a Linde–Hampson mixed refrigerant (LHMR) refrigerating machine as a source of cold is depicted in Figure 1. The LHMR refrigerating machine of the basic/simplest configuration can be used for example for liquefaction of N_2.

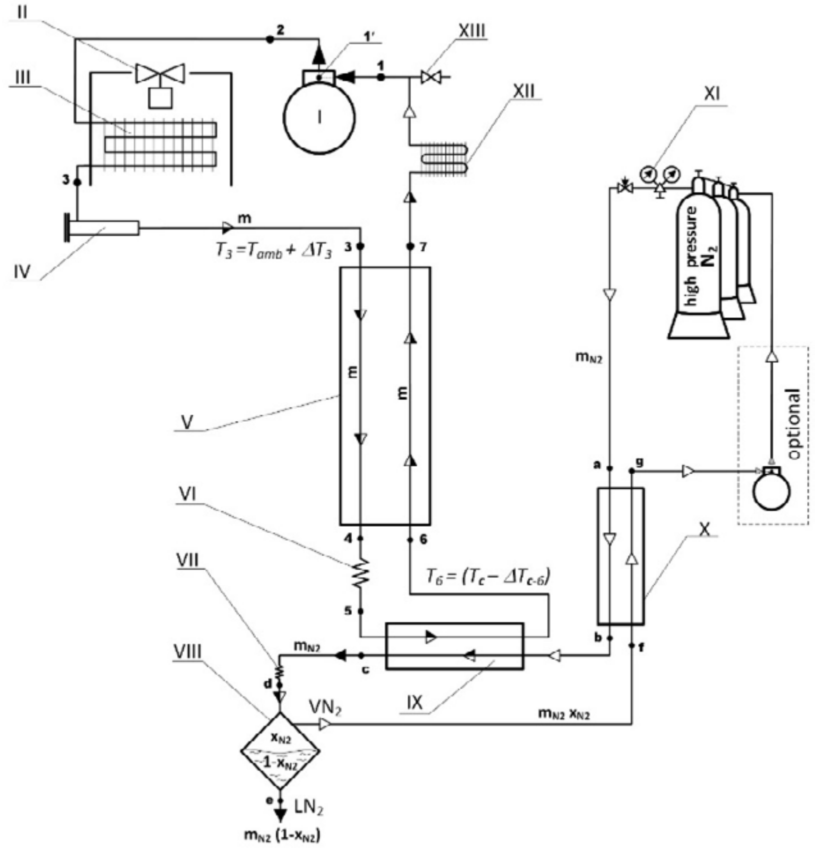

Figure 1:
Schematics of the Linde-Hampson mixed refrigerant closed-cycle designed for cooling and partial liquefaction of a steady stream of N_2. The refrigerating machine is comprised: I – hermetic lubricated compressor; II – fan; III – aftercooler (condenser); IV - filter-drier; V - recuperative heat exchanger; VI, VII - throttling unit's; VIII – liquid phase separator; IX - low-temperature heat exchanger «working mixture-nitrogen»; X - high-temperature heat exchanger «nitrogen-nitrogen», XI – pressure reducing valve, XII – super heater (optional); XIII - filling valve

While the hardware design of the gas cooler could be considered as a number of classical solutions, the algorithm for designing of a low-temperature refrigeration system is an innovative part of the project. The working fluid, temperature, thermal, hydraulic characteristics, and other parameters of the machine are calculated and defined purposely for the development of a working functional model of such low-temperature system for cooling of the gas flows (e.g. down to 77 K).

A multi component zeotropic mixture of hydrocarbons (HCs) is being considered and used as a working fluid in the LHMR refrigerating machine. A number of the working mixture components using in the machine's computer simulation varies within the limits of 2...20. The upper limit '20' comes from restrictions imposed by the computer routines (mentioned above) used to calculate thermodynamic and transport properties of mixtures of HCs. The maximal components number of 20 is a "theoretical" limit. (The computer simulation algorithm calculates the optimal working mixture composition, hydraulic parameters and sets the upper limit of the used components for the calculation.)

3. INNOVATIVE PART OF THE WORK

An innovative solution in the work is the algorithm for designing of a low-temperature refrigeration system in which the working fluid, temperature hydraulic characteristics, and other parameters of the machine are calculated and defined. The result of such calculations can be used for the development of such a low-temperature system for cooling of a gas flow.

During performing the calculation using the proposed algorithm, an optimal mixture from more than 20 different gas components is used. At the same time, a solution with a quantity of working mixture components from 2 up to 5, depending on the required temperature level and cooling capacity can be proposed. When using 20 different components for calculation, the number of combinations for selecting a working mixture can reach several hundred. Also, when selecting the working mixture, the optimal mass composition of the components is determined to ensure the best performance of the designed system, which means the minimum compressor power consumption.

The originality of this approach is that the required parameters are selected individually for each cooling object taking into account the required temperature level, heat load and operating conditions of the refrigeration machine. The arrangement of equipment in the system, which ensures the circulation of the charged mixture without deposition, is also an innovation.

4. CONCISE DESCRIPTION OF THE LINDE CYCLE CALCULATION PROCEDURE

Within the calculation procedure for the mixed refrigerant (MR) refrigerating machine the following parameters are considered. The object of cooling is a pressurized N_2-gas flow to be liquefied at a given mass rate (see Figure 1, points b-c). The cooling capacity Q_0 and the ambient temperature, T_{amb} has to be taken into account. The refrigerating machine should operate with a single-stage lubricated hermetic compressor. A mixture of hydrocarbons with as few components as possible should be used as the working agent.

By the use of the numbers in Figure 1, a steady-state energy balance for each cycle component is presumed as follows: A cooling capacity

$$Q_{eva} = H_6 - H_5 \,. \qquad \textit{Eq. (1)}$$

Since a non-azeotropic mixture is used as the refrigerant, the process of evaporation 5 - 6 proceeds under a varying temperature. I.e., Q_{eva} is being produced at the varying temperature level within the range of ($T_5 \ldots T_6$). The temperature of the N_2-flow in the evaporator is T_c, determined by the working fluid temperature of evaporation: $T_6 = T_c - \Delta T_{eva}$, where ΔT_{eva} is the heat exchange temperature loss.

The heat load of the aftercooler is

$$Q_{AFG} = H_2 - H_3 \,. \qquad \textit{Eq. (2)}$$

The heat load of the RHEX is

$$Q_{RHEX} = H_7 - H_6 = H_3 - H_4 . \qquad Eq. (3)$$

In the formulas presented above the mass flow rate and composition of the working fluid components are implied in the total enthalpies.

A special iterative procedure of the RHEX heat balance calculation is implemented in MR cycle calculations. Only a minimum allowable temperature difference $(T_3 - T_7)_{min}$ at the RHEX hot end is to be specified initially. The actual temperature difference $(T_3 - T_7)$ results from the iterative calculations and it may be greater or equal to the specified minimum difference.

In thermodynamic calculations of the RHEX designed for any non-azeotropic MR systems the q-T diagram analysis must follow the heat balance calculation. A procedure of the q-T diagram computation is well established and widely known (Venkatarathnam and Murthy, 1999) and (Kepler et al., 2004). The results of a MR refrigerating machine cycle calculation for current parameters are considered unsuccessful if the cooling curves in the diagram make no physical sense (have a pinch point).

The input power of the compressor is

$$W_{imp} = \frac{1}{\eta_m}\left(\frac{W_{is}}{\eta_{ind}} + W_{fr}\right), \qquad Eq. (4)$$

where W_{is} is the isentropic work of the compressor, W_{fr} the friction losses and η_m the efficiency of the compressor's electric motor. The efficiency of the compressor is evaluated from the following empirical equation (Plastinin, 1987):

$$\eta_c = \frac{T_6+(T_7-T_6)}{a\times T_3+b\times \Delta T_{c,sup}}, \qquad Eq. (5)$$

where a, b and b′ are the empirical coefficients reflecting influence of the temperature of condensation, superheating of the vapor in the compressor cylinder and the refrigerant flow temperature at the evaporator outlet port as a function of a specific compressor type (hermetic, open-type, vertical, compressor sizing, etc.) and refrigerant. $\Delta T_{c,sup}$ is the suction line superheat temperature difference.

The internal superheating caused by the compressor's electric motor is

$$\Delta T_{i,sup} = \frac{1}{C_p \times \eta_m}\left(\frac{W_{is}}{\eta_c} + W_{fr}\right)\frac{1}{\dot{m}}\left(\frac{1}{\eta_m} - 1\right)\varphi , \qquad Eq. (6)$$

where C_p is the specific heat, $\dot{m}$ is the refrigerant mass flow rate and φ is the empirical coefficient. The coefficient of performance of the cycle is

$$COP = \frac{Q_{eva}}{W_{imp}} . \qquad Eq. (7)$$

As with Q_{eva}, the *COP* should be related with the maximum temperature of the refrigerant flow in the evaporator (T_6).

The optimum component concentrations and pressures of evaporation and condensation have to be computed for every working mixture. The search for the optimum concentrations of a *N*-component working mixture has been conducted over the whole range of the components concentrations variations: $c_i \in (1..99)$ %, i = 1..N-1; $cN = 100 - \sum_{1}^{N-1} c_i$, with an accuracy of 1 %. The operating pressures of evaporation and condensation varied within the range of (1 … 20) bar with steps of 1 bar and has already been discussed above. The maximum pressure ratio (p_{cond}/p_{eva}) allowable in the calculations has been taken equal to 10.

5. THERMODYNAMIC RESULTS OF THE LINDE-HAMPSON MIXED REFRIGERANT CLOSED-CYCLE

A series of thermodynamic computations for various mixtures of hydrocarbons and conventional refrigerants have been conducted by the procedure presented above. The computational procedure determines both *the optimal values of the operating pressures* (within the given range) and *the optimal concentrations* of the given working mixture. The numerical simulation of the LHMR closed-cycle is conducted over the *whole range* of the variations of the concentration of the components ($c_i \in (1, \ldots, 99)$ %, $I = 1, \ldots, N_{cmpnt} - 1$; $c_{Ncmpnt} = 100 - \Sigma_1^{N-1} c_i$, % with an accuracy of 1%) and the operating pressures. The optimum values of the component concentrations and operating pressures are assigned in an additional procedure of comparative analysis of the compressor's input powers in all calculated operating modes.

A special computer program implying mentioned NIST's subroutines was developed in Intel Fortran to implement the full-scale numerical calculations with respect to the reported above parameters of optimization. The REFPROP subroutines are based on the most accurate pure fluid and mixture models currently available. It implements three models for the thermodynamic properties of pure fluids:

- Equations of state explicit in Helmholtz energy;
- Modified Benedict-Webb-Rubin equation of state;
- Extended corresponding states (ECS) model.

Mixture calculations employ a model that applies mixing rules to the Helmholtz energy of the mixture components; it uses a departure function to account for the departure from ideal mixing. Viscosity and thermal conductivity are modelled with either fluid-specific correlations, an ECS method, or in some cases with the friction theory method.

The SUPERTRAPP subroutines calculate phase compositions with the Peng-Robinson equation of state and offers a choice of the Peng-Robinson or the NIST extended corresponding states model (EXCST) for the calculation of phase properties. The calculations of thermodynamic phase properties of hydrocarbon mixtures used the recommended (default) NIST extended corresponding states mode; to calculate vapour-liquid equilibria the Peng-Robinson model was used.

6. FUNCTIONAL MODEL OF THE DESIGNED LOW-TEMPERATURE SYSTEM

In Figure 2 the functional model of the low-temperature system design to produce sufficient amount of cold to cool down and liquefy a gas flow is presented. The model has been developed using conventional units widely available in the market. The most sophisticated system is the low-temperature unit combining an internal recuperative heat exchanger of high efficiency, throttling unit and customized heat exchanger–evaporator. All three components are of modest cost as well, besides they are of special design and play the most important role in ensuring overall system operability and efficiency while producing cold at about 93 K. This functional model has being tested for several months and shown stable performance. Below, Table 1 shows the composition of several working mixtures that were tested on this functional model.

Table 1. The composition of several working mixtures that were tested on the functional model.

Working mixture	The mass composition [kg/kg]	Pressure	
		Suction [bara]	Discharge [bara]
Isobutan – Ethylen – Methan	72 – 4 – 24	1.0	11.0
Isopentane – Methan	33 – 67	1.5	16.5
Isobutan – Propane – Methan	68 – 9 – 23	1.0	11.0
Isobutan – Ethane – Methan	72 – 4 – 24	1.0	11.0
Isobutan – Propane – Propylen – Methan	71 – 1 – 1 – 21	1.0	9.0
Isobutan – Propylen – Ethane – Methan	72 – 1 – 3 – 24	1.0	11.0

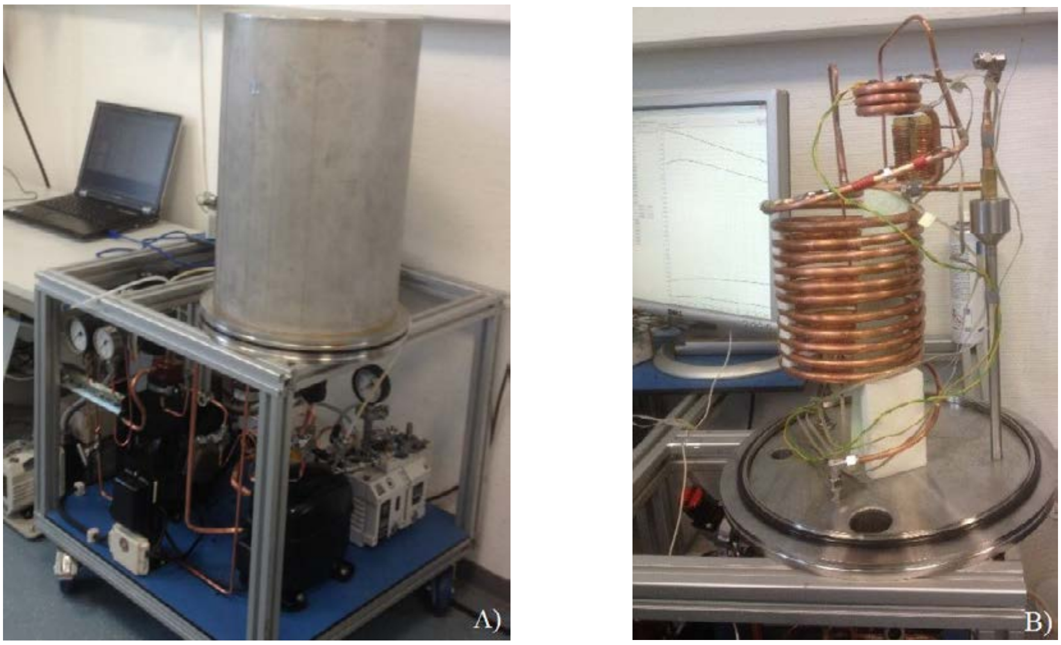

Figure 2:
A) General view of a low-temperature mixed refrigerant cooling system laboratory model design to operate at the temperature level of about 93 K.
B) Low temperature side of the laboratory model without insulating shell.

The cooling curve of the functional model down to about 100 K is shown in Figure 3. The functional model operated steadily over a significant period of time, which proves its operability under this temperature level.

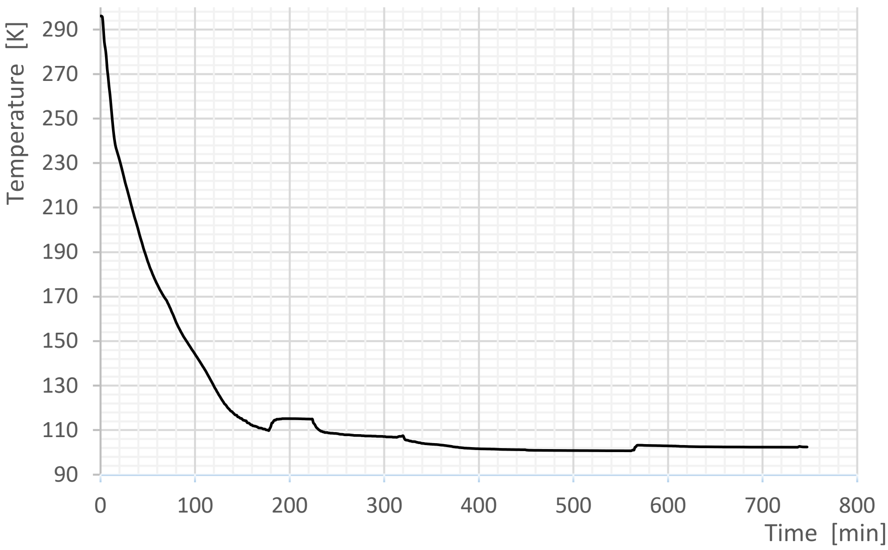

Figure 3: Cooling curve of the functional model down to about 100 K.

Figure 4 shows the result of experiments for four different mixtures I, II, III, and IV in the form of operating heat load dependent on the temperature after the Joule-Thomson and the temperature on the outlet of the evaporator.

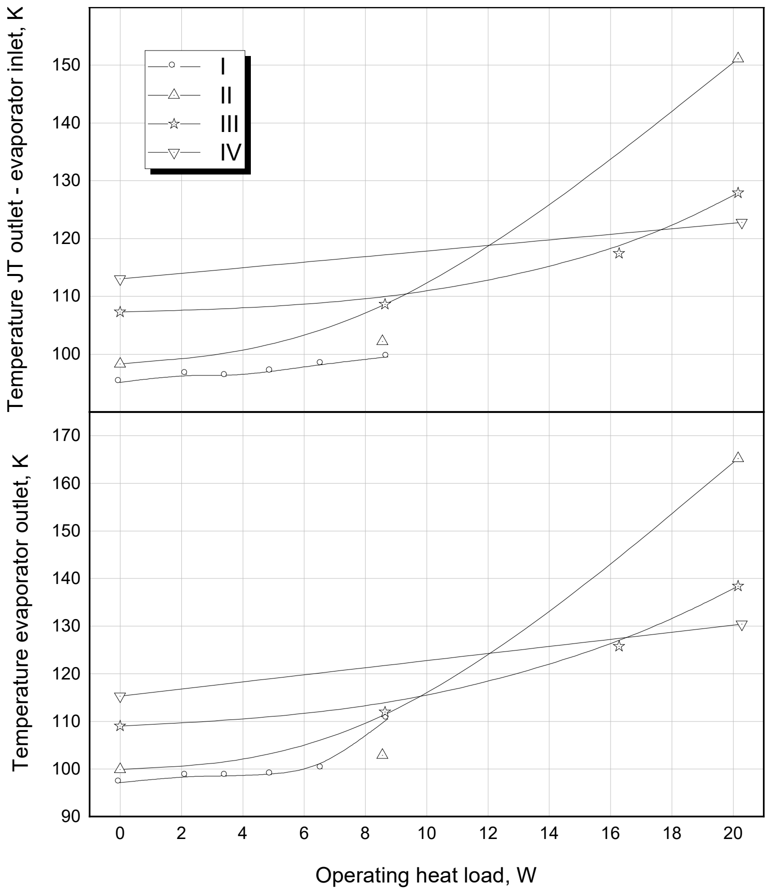

Figure 4: The result of experiments with four different refrigerant mixtures I, II, III, and IV. Used compressor type: Danfoss SC18CL. As a throttle, the copper capillary was used. The capillary length needs to be adjusted for each mixture composition.

7. CONCLUSIONS

A low-temperature refrigeration system for cooling and possible subsequent liquefaction of a gas flow at a temperature level of about 100 K was developed. The shown technical solution provides a set of advantages over existing systems: comparatively high power efficiency, environmental friendly, fire safety, compactness and economic competitiveness in the current market.

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