Analysis of Two-stage High- temperature Heat Pump Efficiency

M. Jotanović, G. Tadić, J. Krope, D. Goricanec

***Abstract*—** The paper shows the potential benefit of using low temperature heat sources with high temperature pumps. Two-stage heat pumps with flash unit and heat exchanger are described. To determine the characteristic parameters of the two heat pumps a computer program was designed, with which we could determine the dependence of COP and compressor pressure ratios from evaporating temperature. An economic analysis of the justification of the use of heat pumps was made.

***Keywords*—** Heat transfer, high temperature heat pumps, computer programme, economic analysis

I. Introduction

FOR a long time now, people have experienced the negative consequences of the greenhouse effect. The situation is alarming and that is why we are searching for the ways how to avoid the approaching natural catastrophy. It threatens to destroy mankind unless we do something immediately. At various conferences and forums scientists are establishing that the only possible method of reducing air pollution is the efficient use of energy together with the development of new technologies and systems, as well as the use of renewable sources of energy.

The consequences of an intensive exploitation of energy and energy dependence refer decidedly to the countries with poor sources of energy and which are classified into a group of the most environmentally endangered countries in Europe. The complicated situation in these countries does not allow them to deal with energy issues by simply fulfilling demands and desires of energy consumers. With simple and cheap decisions on the rationalization, it is possible to achieve considerable savings and have unpolluted environment.

Studies and researches in industry usually show that disadvantages in energy conversion are on the side of technological processes; this means in the consumption of energy, rather than in the preparation of energy. When it comes to the operation of an energy plant, there is often a dilemma how to improve specific consumption of energy in processes, how to increase efficiency, how to direct and convert energy more efficiently, how to use waste heat and replace combustion of liquid and gaseous fuels with other sources of energy [1, 2, 3].

Researches have come up with an innovative solution of using waste low-temperature water (45°C) for the purpose of long-distance heating of buildings at the temperature of 90/70°C. In this respect, the Faculty of Chemistry and Chemical Engineering of University of Maribor developed an innovative high-temperature heat pump (TP) made by a Japanese company named Mycom in 2010 [4, 5, 6]. This is the first example in the world and has unseen application possibilities in industry, which is confirmed by the Intergovernmental platform for R&D in Europe reward.

The essence of this paper is focused on the following:

- Computer simulation of techno-economic effect of the use of different refrigerant,
- Computer simulation of the effect of different temperatures of low-temperature sources and different refrigerants on COP,
- Selection of technically and economically optimal heat pump,
- Production of a computer supported model which will enable efficient usage of different low-temperature sources for the purpose of long-distance heating, greenhouses heating, heating of dry-kilns, in process industry (technological water), sanitary water, etc. This would provide great financial benefits and environment protection.

The research was conditioned by:

- the necessity of intensive exploitation of alternative sources of energy and low-temperature sources,
- the necessity of a consistent and coordinated approach to the exploitation of the heat from low-temperature sources of energy,
- fulfilment of the recommendations and requests of the EU regarding efficient energy consumption and environment protection.

II. High Temperature Heat Pump

Heat pump is a process device which is used for heating. Its operation principles are based on the removal of low temperature from the environment, which is then given on a higher temperature level. The sources for the removal of energy are air, water or ground.

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The rapid development of heat pumps starts with the great oil crisis, when people started searching intensively for technological solutions for the replacement of fossil fuels with other sources of energy. Due to stricter laws on pollution, the awareness of energy, environmental awareness of consumers and the rise in the price of energy, heat pumps are becoming more and more interesting because they are energy productive and environmentally friendly. With the development of new technologies, the methods of application, improvement of performances, and reduction of the dimensions and mass of these devices on the other hand, the application of heat pumps increases. The most recent types of heat pumps have the possibility of operation in high-temperature systems of long-distance heating. They use refrigerants which do not harm the environment and have a good ratio between the electric power used and the heat obtained, which is even up to 1:9.

Scientists predict that heat pumps with different implementation methods in the future will be the fundamental part in low-temperature and high-temperature heating system as an environmentally acceptable solution. An important fact in the application of heat pumps is that they use much less primary energy compared to gas and oil boilers. This means that they can reduce the emission of CO_2 and other gases by 31 % to 60 %. For example, in cases where the operation of heat pumps uses electric power from renewable sources, the emission of greenhouse gases is reduced to minimal values, which will become a necessity sooner or later [7].

A. Operation of heat pumps

Compression heat pump consists of an evaporator, condenser, compressor and regulation valve. The operating principles of a one-stage heat pump (ETČ) are shown in Fig. 1 [8,9].

In the evaporator, the refrigerant, which is liquid, after the reduction of pressure on the regulation valve, is evaporated at low temperatures and saturation pressure. Then the refrigerant receives a heat flow from the environment Φ_U. Refrigerant vapour is then taken to the compressor where it is compressed to the pressure where the condensation temperature is higher than the temperature of the environment. Vapour is then cooled and condensed. It thus gives a flow of heat Φ_K to the surrounding area that is heated. The liquid refrigerant is then again taken to the evaporator through the regulation valve.

Since the compressor is the main process unit, it is necessary to consider the following when planning different implementations: the ratio between pressures, maximum allowed gas pressure maximum allowed temperature, rotation speed, power and volume flow of the compressor, as well as other characteristics. Table 1 shows characteristics of some compressors made by Mayekawa - Japan.

Table 1: Performances of the Mayekawa compressors [10]

Operating characteristics	**Type**	**Operating parameters**	**Comments**
Maximum allowed gas pressure (Pd) [bar]	K L, WA, WB WBH	Ammonia: 23 HFC: 24 23,6 20	
Maximim pressure of gas pumpimg out (Ps) [bar]	K L, WA, WBH, WB	6,86 5,88	
Maxium average gas pressure [bar]	42WBH, 62WBH 42WB, 62WB, 124WB	1000 revol./min; 8,22 1100 revol./min; 6,56 1200 revol./min; 5,38	Only for HFC
Minimum pressure of pumping out of the gas [bar]	All models	-0,733	Necessary for the introduction of oil
Maximum differential pressure [bar]	K L, WA, WB WBH WBH	19,6 14,7 20,0 15,2	=Pd - Ps
Maximum presuures ratio [/]	K L, WA, WBH, WB	Ammonia: ≤ 8 Ammonia: ≤ 9 HFC: ≤ 10	Allowed temperature of the gas
Minimum pressure of introduction of oil [bar]	K, WA L, WA,WBH, WB	9 8	
Maximum pressure of intro uction of water [bar]	All models	5	
Minimum pressure of the introduction of water [bar]	All models	2	
Maximum allowed gas temperature [°C]	K WA, WBH, WB,L	Ammonia: 140 HFC: 135 Ammonia:140 HFC: 120	With oil protection
Min. temp. of gas pumping out	All models	-60	
Maximum temperature of the introduction of water [°C]	K WA, WBH, WB,L	Ammonia: 50 HFC: 70 50	At the outlet of the refrigerant
Min. temperature of oil introduction [°C]	All models	30	
Maximum water temperature [°C]	All models	50	
Minimum temperature of water introduction [°C]	All models	15	
Maximum rotation speed [revol./min]	K L 2WA WA WBH, WB	1800 1750 (1500) 1100 1450 1200	
Minimum rotation speed [revol./min]	K L WA, WBH, WB	900 970 800	
Maximum strength of. drive [kW]	L, WBH, WB WA	145 77	
Volume flow of piston compressor [m^3/h]	K WA L WBH	35 – 250 80 – 380 220 – 650 250 – 770	

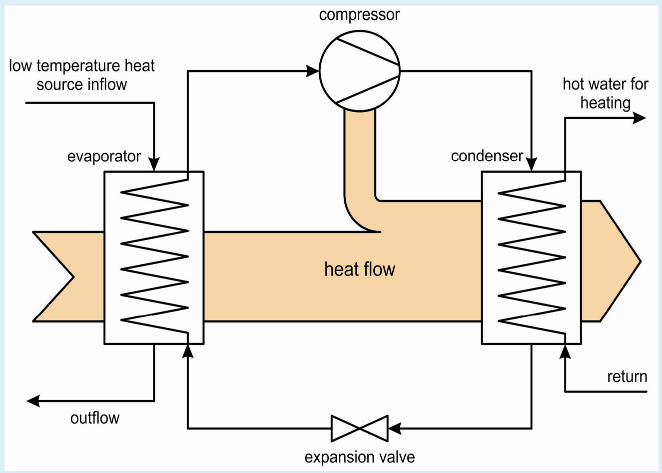

Fig. 1: High-temperature heat pump operating principles-diagram

B. Review of the sources for heat pumps

When it comes to the sources of energy for the operation of heat pumps, the most advantageous today are renewable sources such as wind, hydroenergy, solar energy, geothermal energy as well as waste energy from process industry.

Globally speaking, geothermal resources refer to the heat accumulated deep in the ground, i.e.in the mass of stone and fluids in the Earth's crust. A cross-section of a younger igneous intrusion lies on the surface, with conduction through deep tectonic discordances and through inlet volcanic channels. Geothermal energy is also the result of the decomposition of radioactive elements in different chemical processes that occur in the Earth's crust. The Earth's temperature is increased by 1°C (geothermal degree) to every 33 m of its depth.

If it increases more rapidly, we speak of a positive anomaly or increased geothermal degree, or geothermal gradient. The main indicator for an area to have a great potential for exploitation is the geothermal anomaly, which, hydrologically and hydrochemically, gives a clear description of the possibilities and methods for the exploitation of the energy potential. Considering the possibility of exploitation, geothermal energy is classified into:

- hydrogeothermal energy and
- petrogeothermal energy.

The first type refers to the geothermal energy of liquid and gaseous fluids, whereas the other is the geothermal energy of a mass of rocks. The significance of geothermal energy is best described by the two natural properties: the stability and consistency of the heat flow and heat, which is kept in rocks. If we say that a heat flow which is transferred from the inside of the Earth towards the surface of the Earth is considered to be geothermal energy, than we can say that this energy is a renewable source of energy.

Keeping energy enclosed in rocks is a complex process which depends on the three forms of heat transfer. Convection of water and magma is such a rapid energy transfer that we can classify geothermal energy into renewable sources, because the energy removed is compensated at the same speed. If, on the other hand, energy is transferred only with heat exchange, we can hardly classify it into renewable sources since the time constant of energy compensation is considerably longer than the constant of the exploitation of the same energy. All conventional methods of the exploitation of geothermal energy are based on the removal of energy from natural geothermal systems in which energy is supplied by water and transferred at the same time within the system itself. Water takes energy to the surface where it is exploited. During production, pressures begin to decrease, which causes the supply of additional water and energy to the system which is exploited. Such conditions are typical for renewable sources of energy, where compensation and exploitation of energy occur at the same rate. Geothermal energy results form three factors:

- Energy which is stored in rocks and fluids of the Earth's crust,
- Heat flow evolved by exchange and
- Energy flow through the Earth's crust in the form of the transfer of matter (magma, water, vapour, gases). Occurrences on the surfaces (hot springs) are usually the best indicator that geothermal sources exist in the depths of the Earth's crust; however, it is possible to recognize the sources sometimes even without them. It is essentially thought that the total value of uninvestigated or unidentified geothermal sources is greater than the known sources. Low-temperature sources are more commonly found than those higher -temperature. It should be known that the frequency of distribution of sources depending on temperature is the function of an ordinary exponential curve. In order to determine the lower limit of the potential of low -temperature sources on the Earth we must consider the following approximations:
- the frequency of occurrence of sources as the function of temperature has the properties of an exponential curve,
- the energy of a particular source is linearly dependent on its temperature and
- the volume of a geothermal reservoir is independent of its temperature.

Conventional exploitation of geothermal energy is usually classified into:

- high - temperature sources with the temperatures of water above 150°C, which are used for the production of electricity and
- low - temperature sources with the temperature of water below 150°C, which are mainly used directly for heating.

Considering the data for the Earth's geothermal gradient, the majority of European countries have a number of unexploited low - temperature geothermal energy sources. Only a small number of countries have high-enthalpy sources which are suitable for the production of electric power, on the depths that still justify the economic aspects of exploitation. Direct exploitation of geothermal energy may include different markets. In larger, industrially developed countries a large part of heat is used at temperatures between 50 and 100°C.

Researches into the usage of geothermal energy in the world regarding the purpose have shown that energy is mostly used for heating, half of it being for temperature regimes below 100°C. Distribution of exploitation purposes is the following: heating 35 %, pools (balneology) 15 %, greenhouses 14 %, fishponds10 %, industry 10 %, air-conditioning 1 %, and heat pumps 13 %, other 2 %.

In Europe and other parts of the world geothermal energy is used for the production of electric power, for utility heating of residences and industrial facilities, in agriculture for heating of greenhouses, in tourism, etc. The best conditions for the exploitation of geothermal energy in Europe are in Iceland, Italy and Greece.

Slovenia is also rich with low-temperature geothermal energy.

C. The possibility of using geothermal energy

The majority of flats in cities are old, with poor insulation and using high-temperature systems of radiators. When the temperature of geothermal water that we wish to use is 42°C, it is not high enough for high-temperature heating using a heat pump. By installing a heat pump into the heating system the energy of geothermal water is additionally used allowing simultaneously additional users of heat energy in the heating system, which has a positive effect on the environment and economic justifiability.

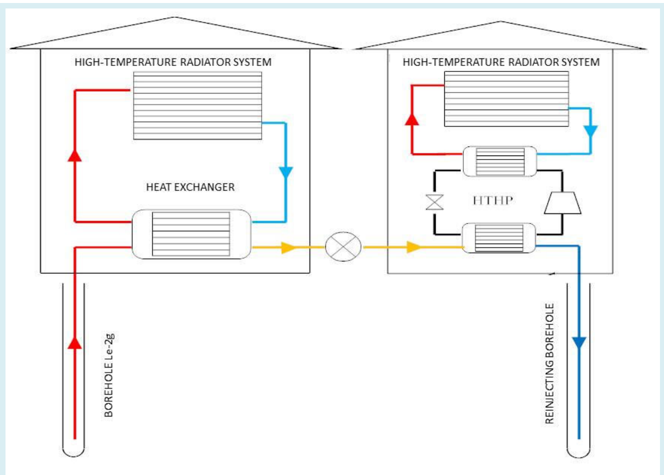

Fig. 2: Representation of the exploitation of geothermal water for long-distance heating

D. The two – stage high temperature heat pump with a flash expander

The two-stage high-temperature heat pump with a flash expander includes two compressors and two expansion valves. The two stages of compression are necessary due to the reduction of a high ratio of pressures, which has negative effects on the performances of the compressor (energy consumption, cooling...). Middle stage of compression is separated by the flash/separate flash expander, and the same refrigerant is used for heat transfer in both stages. The flash heat pump diagram is shown in Fig. 3.

Due to the two sages of compression there is a possibility of using lower temperature sources (10 to 30°C) for heating. In such implementation it is important which refrigerant will be used, because it has to have good thermo-physical characteristics as well as properly determined pressure.

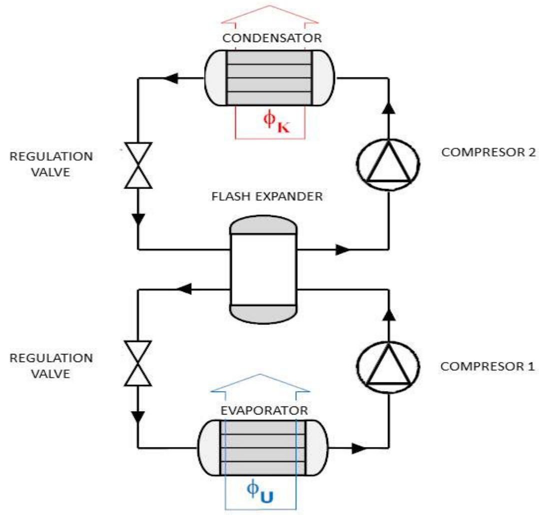

Fig. 3: Two-stage high-temperature heat pump with a flash expander

1) Calculation

When performing calculation for the two-stage high temperature heat pump with a flash expander it is important to determine correctly the intermediate pressure of the refrigerant, between the first and the second level/stage, which is equal to the following according to the general equation:

$$p_M = \sqrt{p_{S,1} \cdot p_{T,2}} \qquad \text{(Pa)} \qquad (1)$$

where:

$p_{S,1}$ - vapour pressure of the refrigerant on the inlet side of the first stage of compression (Pa)

$p_{T,2}$ - vapour pressure of the refrigerant on the DELIVERY SIDE of the second stage of compression (Pa)

Intermediate temperature T_M, which for the first stage of compression of the refrigerant is the condensation temperature $T_{K,1}$, and for the second stage of compression pa it is the temperature of evaporation $T_{U,2}$ is calculated using the following equation:

$$T_M = \left[\frac{-B \pm \sqrt{B^2 - 4 \cdot A \cdot \left(C - \log\left(\frac{p_M}{[Pa]}\right)\right)}}{2 \cdot A} \right]^{-1} \qquad \text{(K)} \qquad (2)$$

Heat flows, of the first stage $\Phi_{TČ,1}$ and of the second stage $\Phi_{TČ,2}$ are calculated using the equations:

$$\Phi_{TČ,1} = q_{m,S_1} \cdot (h_{g,2} - h_{l,3}) \quad \text{(W)} \tag{3}$$

$$\Phi_{TČ,2} = q_{m,S_2} \cdot (h_{g,6} - h_{l,7}) \quad \text{(W)} \tag{4}$$

where:

$h_{g,6}$ - specific enthalpy of refrigerant vapour on the DELIVERY SIDE of the compressor of the second stage of the heat pump (J/kg·K)

$h_{l,3}$ - specific enthalpy of the liquid refrigerant in the condenser of the second stage of the heat pump (J/kg·K)

The heat number of the two - stage heat pump with a flash expander is calculated using the following equation:

$$COP_G = \frac{\Phi_{TČ,2}}{P_{K,1} + P_{K,2}} \quad (\,/\,) \tag{5}$$

Other parameters of the high-temperature heat pump with a flash expander and the dimensions of the compressor for both stages are calculated according to the same principle as for one-stage high-temperature heat pumps [11, 12].

2) Calculation programme

DATA

REFRIGERANT: R-600a (izo-butan)

CONSTANTS FOR THE CALCULATION OF THE PRESSURE FOR THE TEMPERATURE Tu AND Tk

$au := 0$

$AU := 0$

$bu := -1129.8$

$BU := -1112.6$

$cu := 9.3334$

$CU := 9.2784$

CONSTANTS FOR THE CALCULATION OF THE DENSITY OF THE REFRIGERANT

$at := -0.0026$

$AT := -0.0065$

$bt := 0.2975$

$BT := 2.7801$

$ct := 693.83$

$CT := 292.66$

CONSTANTS FOR THE CALCULATION OF THE DENSITY OF REFRIGERANT VAPOUR

$ap := 0$

$AP := 0$

$bp := 0.2583$

$BP := 0.6005$

$cp := -67.669$

$CP := -177.24$

CONSTANTS FOR THE CALCULATION OF THEENTHALPY OF REFRIGERNAT LIQUID

$xt := 2.463$

$XT := 2.8062$

$yt := -474.45$

$YT := -584.34$

CONSTANTS FOR THE CALCULATION OF THE ENTHALPY OF REFRIGERANT VAPOUR

$xp := 0$

$XP := 0$

$yp := 1.3407$

$YP := 1.2157$

$zp := 189.57$

$ZP := 230.14$

COMPRESSOR TYPE: 8RWH

NUMBER OF REVOLUTIONS [1/min]:

$N := 1000 \cdot \text{min}^{-1}$

OPERATION OF THE COMPRESSOR [%]:

$\mu := 1$

UTILIZATION OF THE COMPRESSOR [/]:

$\eta := 0.72$

BOILING TEMPERATURE OF THE REFRIGERANT [K]:

$Tu := 303.15K$

CONDENSATION TEMPERATURE OF THE REFRIGERANT [K]:

$Tk := 353.15K$

MEDIUM TEMPERATURE [K]:

$$Ts := \frac{(Tu + Tk)}{2}$$

$Ts = 328.15K$

COMPRESSIBILITY FACTOR:

$\chi := 1.188$

UNUSED SPACE FOR FILLING THE COMPRESSOR

$\varepsilon := 0.14$

TWO-STAGE HEAT PUMP WITH A FLASH EXPANDER

PRESSURE ON THE INLET SIDE OF THE COMPRESSOR [Pa]:

$$Pu := 10^{\left[au \cdot \left(\frac{K}{Tu}\right)^2 + bu \cdot \left(\frac{K}{Tu}\right) + cu\right]} \cdot Pa$$

$Pu = 404140.3Pa$

SPECIFIC ENTHALPY OF THE REFRIGERANT VAPOUR AT THE BOILING TEMPERATURE

$$Hpu := \left[xp \cdot \left(\frac{Tu}{K}\right)^2 + yp \cdot \left(\frac{Tu}{K}\right) + zp\right] \cdot \frac{J}{kg}$$

$$Hpu = 596.003 \frac{1}{kg} \cdot J$$

SPECIFIC ENTHALPY OF THE LIQUID AT THE BOILING TEMPERATURE

$$Htu := \left(XT \cdot \frac{Tu}{K} + YT\right) \cdot \frac{J}{kg}$$

$$Htu = 266.36 \frac{1}{kg} \cdot J$$

REFRIGERANT VAPOUR DENSITY AT THE BOILING TEMPERATURE

$$\rho pu := \left[ap \cdot \left(\frac{Tu}{K}\right)^2 + bp \cdot \left(\frac{Tu}{K}\right) + cp\right] \cdot \frac{kg}{m^3}$$

$$\rho pu = 10.635 \frac{kg}{m^3}$$

REFRIGERANT LIQUID DENSITY AT THE BOILING TEMPERATURE

$$\sigma tu := \left[at \cdot \left(\frac{Tu}{K}\right)^2 + bt \cdot \left(\frac{Tu}{K}\right) + ct\right] \cdot \frac{kg}{m^3}$$

$$\sigma tu = 545.077 \frac{kg}{m^3}$$

PRESSURE ON THE DELIVERY SIDE OF THE COMPRESSOR [Pa]:

$$Pk := 10^{\left[AU\cdot\left(\frac{K}{Tk}\right)^2 + BU\cdot\left(\frac{K}{Tk}\right) + CU\right]}\cdot Pa$$

$$Pk = 1342447.7Pa$$

SPECIFIC ENTHALPY OF REFRIGERANT VAPOUR AT THE CONDENSATION TEMPERATURE

$$Hpk := \left[XP\cdot\left(\frac{Tk}{K}\right)^2 + YP\cdot\left(\frac{Tk}{K}\right) + ZP\right]\cdot\frac{J}{kg}$$

$$Hpk = 659.5\,\frac{1}{kg}\cdot J$$

SPECIFIC ENTHALPY OF REFRIGERANT LIQUID AT THE CONDENSATION TEMPERATURE

$$Htk := \left(XT\cdot\frac{Tk}{K} + YT\right)\cdot\frac{J}{kg}$$

$$Htk = 406.67\,\frac{1}{kg}\cdot J$$

REFRIGERANT VAPOUR DENSITY AT THE CONDENSATION TEMPERATURE

$$\rho pk := \left[AP\cdot\left(\frac{Tk}{K}\right)^2 + BP\cdot\left(\frac{Tk}{K}\right) + CP\right]\cdot\frac{kg}{m^3}$$

$$\rho pk = 34.827\,\frac{kg}{m^3}$$

REFRIGERANT LIQUID DENSITY AT THE CONDENSATION TEMPERATURE

$$\sigma tk := \left[AT\cdot\left(\frac{Tk}{K}\right)^2 + BT\cdot\left(\frac{Tk}{K}\right) + CT\right]\cdot\frac{kg}{m^3}$$

$$\sigma tk = 463.805\,\frac{kg}{m^3}$$

INTERMEDIATE PRESSURE CALCULATION

$$Pm := \sqrt{Pu\cdot Pk}$$

$$Pm = 736571.2Pa$$

INTERMEDIATE TEMPERATURE CALCULATION Tm

$$Tm := \left(\frac{\log\left(\frac{Pm}{Pa}\right) - cu}{bu}\right)^{-1}\cdot K$$

$$Tm = 325.95K$$

SPECIFIC ENTHALPY OF REFRIGERANTE VAPOUR AT INTERMEDIATE TEMPERATURE Tm

$$Hpm := \left[XP\cdot\left(\frac{Tm}{K}\right)^2 + YP\cdot\left(\frac{Tm}{K}\right) + ZP\right]\cdot\frac{J}{kg}$$

$$Hpm = 626.4\frac{1}{kg}\cdot J$$

SPECIFIC ENTHALPY OF REFRIGERANT LIQUID AT INTERMEDIATE TEMPERATURE Tm

$$Htm := \left(XT\cdot\frac{Tm}{K} + YT\right)\cdot\frac{J}{kg}$$

$$Htm = 330.338\frac{1}{kg}\cdot J$$

REFIRGERANT VAPOUR DENSITY AT THE INTERMEDIATE TEMPERATURE Tm

$$\rho pm := \left[AP\cdot\left(\frac{Tm}{K}\right)^2 + BP\cdot\left(\frac{Tm}{K}\right) + CP\right]\cdot\frac{kg}{m^3}$$

$$\rho pm = 18.492\frac{kg}{m^3}$$

REFRIGERANT LIQUID TEMPERATURE AT THE INTERMEDIATE TEMPERATURE Tm

$$\sigma tm := \left[AT\cdot\left(\frac{Tm}{K}\right)^2 + BT\cdot\left(\frac{Tm}{K}\right) + CT\right]\cdot\frac{kg}{m^3}$$

$$\sigma tm = 508.253\frac{kg}{m^3}$$

PRESSURES RATIO FOR THE FIRST COMPRESSOR

$$rK1 := \frac{Pm}{Pu}$$

$$rK1 = 1.823$$

TEMPERATURE AT THE OUTLET OF THE FIRST COMPRESSOR [K]:

$$T2 := Tu\cdot\left(\frac{Pm}{Pu}\right)^{\left(\frac{\chi - 1}{\chi}\right)}$$

$$T2 = 333.358K$$

SPECIFIČ ENTHALPY OF REFRIGERANT VAPOUR AT THE TEMPERATURE T2

$$Hp2 := \left[XP\cdot\left(\frac{T2}{K}\right)^2 + YP\cdot\left(\frac{T2}{K}\right) + ZP\right]\cdot\frac{J}{kg}$$

$$Hp2 = 635.4\frac{1}{kg}\cdot J$$

FILLING RATE OF THE CYLINDER IN THE COMPRESSOR

$$\lambda := 1 - \varepsilon\cdot\left[rK1^{\left(\frac{1}{\chi}\right)} - 1\right]$$

$$\lambda = 0.908$$

THE REAL VOLUME FLOW OF THE COMPRESSOR $vK := Vk\cdot\lambda$

$$vK = 0.161\frac{m^3}{s}$$

$$qv = 0.092\frac{m^3}{s} \qquad Pad1 := vK\cdot Pu\cdot\left(\frac{\chi}{\chi - 1}\right)\cdot\left[rK1^{\left[\frac{(\chi - 1)}{\chi}\right]} - 1\right]$$

$$Pad1 = 40885.39W$$

COMPRESSOR STRENGTH

$$Pk1 := \frac{Pad1}{\eta}$$

$$Pk1 = 56785.26W$$

VOLUME FLOW AT THE INLET SIDE OF THE COMPRESSOR

$$vK = 0.161\frac{m^3}{s}$$

MASS FLOW AT THE INLET SIDE OF THE COMPRESSOR $qms1 := vK\cdot\rho pu$

$$qms1 = 1.709\frac{kg}{s}$$

MASS FLOW AT THE DELIVERY SIDE OFF THE COMPRESSOR

$$qv = 0.092\frac{m^3}{s}$$

VOLUME FLOW OF THE REFRIGERANT IN THE COMPRESSOR

$$qv := vK\cdot\frac{\rho pu}{\rho pm}$$

ADIABATIC COMPRESSION STRENGTH

MASS FLOW ON THE DELIVERY SIDE OF THE COMPRESSOR

$$qmt1 := qv\cdot\rho pm$$

$$qmt1 = 1.709\frac{kg}{s}$$

$$\Phi u1 := qms1\cdot(Hpu - Htm)\cdot 1000$$

HEAT FLOW OF THE BOILED PART OF THE HEAT PUMP THE FIRST STAGE [W]:

PRESSURES RATIO OF THE SECOND COMPRESSOR $rK2 = 1.823$

$$rK2 := \frac{Pk}{Pm}$$

$$\Phi k1 := qms1\cdot(Hp2 - Htm)\cdot 1000$$

$\Phi u1 = 453916.1W$

HEAT FLOW OF THE CONDENSATION PART OF THE HEAT PUMP THE FIRST STAGE [W]:

$\Phi k1 = 521234.5W$

HEAT NUMBER OF THE FIRST STAGE OF THE HEAT PUMP [/]:

$$qms2 := \frac{\Phi k1}{(Hpm - Htk)\cdot 1000}$$

$$qms2 = 2.372\frac{kg}{s}$$

MASS FLOW ON THE INLET SIDE OF THE SECOND COMPRESSOR

$$COPG := \frac{\Phi k1}{Pk1}$$

$COPG = 9.179$

TEMPERATURE AT THE OUTLET FROM THE SECOND COMPRESSOR [K]:

$T3 = 358.43K$

FILLING RATE OF THE CYLINDER IN THE COMPRESSOR

$$\lambda 2 := 1 - \varepsilon\cdot\left[rK2^{\left(\frac{1}{\chi}\right)} - 1\right]$$

$\lambda 2 = 0.908$

VOLUME FLOW OF THE COMPRESSOR IN THE SECOND STAGE OF THE HEAT PUMP

$$Vk2 := \frac{qms2}{\rho\, pm}$$

$$Vk2 = 0.128\,\frac{m^3}{s}$$

THE REAL VOLUME FLOW OF THE SECOND COMPRESSOR

$$vK2 = 0.116\,\frac{m^3}{s}$$

$$T3 := Tm\cdot\left(\frac{Pk}{Pm}\right)^{\left(\frac{\chi - 1}{\chi}\right)}$$

$$Hp3 = 665.9\frac{1}{kg}\cdot J$$

SPECIFIČ ENTHALPY OF THE REFRIGERANT VAPOUR AT THE TEMPERATURE T3

$$Hp3 := \left[XP\cdot\left(\frac{T3}{K}\right)^2 + YP\cdot\left(\frac{T3}{K}\right) + ZP\right]\cdot\frac{J}{kg}$$

$vK2 := Vk2\cdot\lambda 2$

VOLUMSKI PRETOK SREDSTVA U DRUGOM KOMPRESORU

$$qv2 := Vk2\cdot\frac{\rho\, pm}{\rho\, pk}$$

$$qv2 = 0.068\,\frac{m^3}{s}$$

ADIABATIC COMPRESSION STRENGTH IN THE SECOND STAGE

$$Pad2 := vK2\cdot Pm\cdot\left(\frac{\chi}{\chi - 1}\right)\cdot\left[rK2^{\left[\frac{(\chi - 1)}{\chi}\right]} - 1\right]$$

$Pad2 = 54020.19W$

$$Pk2 := \frac{Pad2}{\eta}$$

COMPRESSOR STRENGTH IN THE SECOND STAGE

$Pk2 = 75028.04W$

MASS FLOW ON THE DELIVERY SIDE OF THE SECOND COMPRESSOR

$qmt2 := qv2\cdot\rho pk$

HEAT FLOW OF THE HEAT PUMP IN THE SECOND STAGE [W]:

$\Phi k2 := qms2\cdot(Hp3 - Htk)\cdot 100($

$\Phi k2 = 614901.2W$

HEAT NUMBER OF THE HEAT PUMP [/]:

$$COPG2 := \frac{\Phi k2}{Pk2 + Pk1}$$

$COPG2 = 4.665$

3) Calculation results

The calculations of the two-stage high-temperature heat pump with a flash expander for refrigerants R-600a, R-290, R245fa and R-134a were conducted at two different temperatures of condensation (t_K = 70°C and t_K = 80°C). Results are shown in Fig. 4 and Fig. 5. Fig. 6 and Fig. 7 show the dependences of het number and ratio between pressures in the compressor.

The main objective of the implementation of the two - stage heat pump with a flash expander is to try to reduce the strength which is necessary for the compressor to operate in order to reduce the ratio between pressures and increase the useful heat flow of the condensation part of the pump.

When conducting the calculation, the fact that the compressors in both stages are equal, was taken into consideration. The compressors were piston compressors WBH chosen from the catalogue of the manufacturer called MYCOM. The volume flow of the compressor is 637 m^3/h, maximum operating pressure is $p_{max.}$ = 2,0MPaand maximum strength of the compressor is PK max. = 145,0 kW.

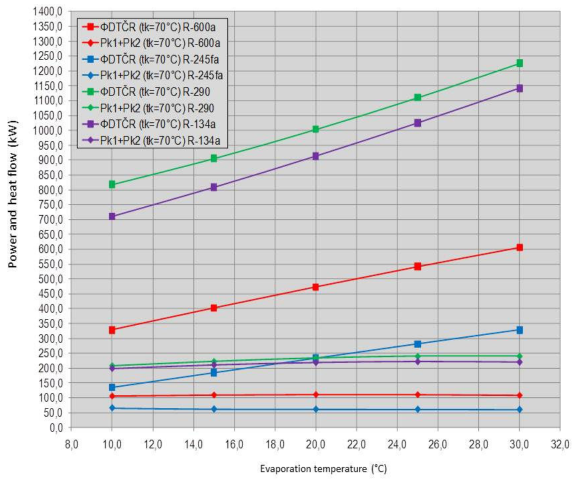

Fig.4: The two-stage high-temperature heat pump with a flash expander at the condensation temperature of the refrigerant of t_K = 70°C

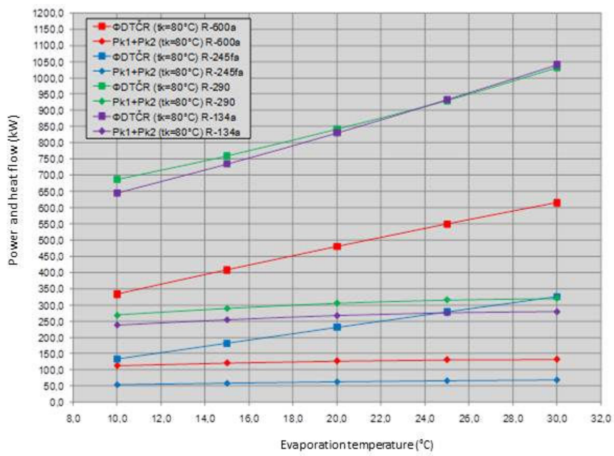

Fig. 5: The two-stage high-temperature heat pump with a flash expander at the condensation temperature of the refrigerant t_K = = 80°C

Analysis of the results shows that the use of low-temperature heat source is possible by using a two-stage high-temperature heat pump with a flash expander, but there are certain limitations regarding the operation of the compressor and the choice of the refrigerant.

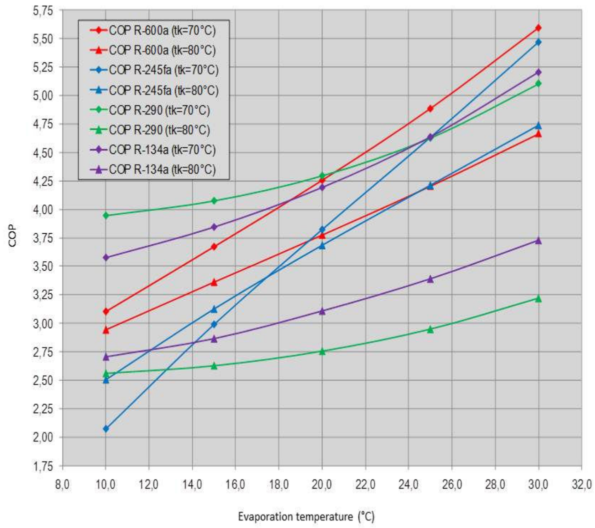

Fig. 6: Heat number of the two-stage heat pump

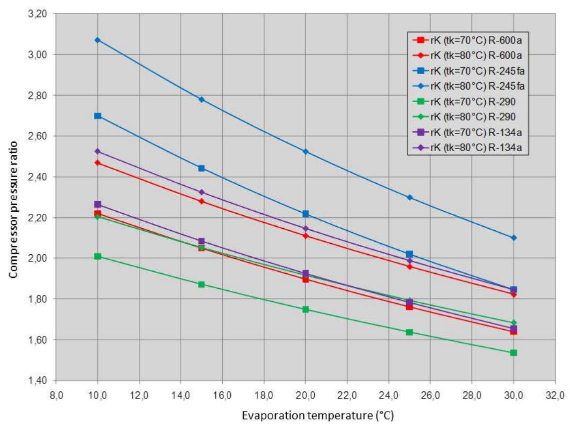

Fig. 7: The ratio between the pressures in the compressor of the two-stage heat pump with a flash expander

The diagram shows that using refrigerants R-290 and R-134a we achieve high values of heat flow, but for the proposed temperatures the consumption of energy necessary for the operation of the compressor is to high in relation to the chosen compressor. Refrigerant R-245fa gives low values of the heat flow as well as the low consumption of energy. For the refrigerant R-245fa the compressor is oversized. The most suitable characteristic parameters of the two-stage heat pump with a flash expander were obtained using the refrigerant R-600a, which enables a wide scope of operation and, on the other hand, conditions for optimal operation of the compressor are in accordance with the instructions from the manufacturer.

With the use of the refrigerant R-600a at the boiling temperature t_U = 20 °C and condensation temperature t_K = 70°C, the heat flow of the heat pump is $\Phi_{TČ}$ = 473,0 kW and the energy for the operation of the compressor is PK,1 = 44,7 kW or $P_{K,2}$ = 66,4 kW. Heat number is COP = 4,256, the ratio between pressures has the value of r_K = 1,898.

For the proposed type of the heat pump, it is understandable to use the refrigerant R-600a, which has good thermo-physical characteristics in a wider temperature range (low pressure and high enthalpy) For the operation of the heat pump to be practical, when other refrigerants are used, it is necessary to determine a suitable standard compressor for every level of compressing.

E. The two-stage heat pump with a heat exchanger

The two-stage heat pump with a heat exchanger consists of two one-stage heat pumps. The stages are separated by the heat exchanger, which serves as the condenser for the first stage and as evaporator for the second stage. The stages are separated in such way that refrigerants are not in contact. This means that each stage has its own refrigerant. The heat pump diagram is shown in Fig. 8. [13, 14].

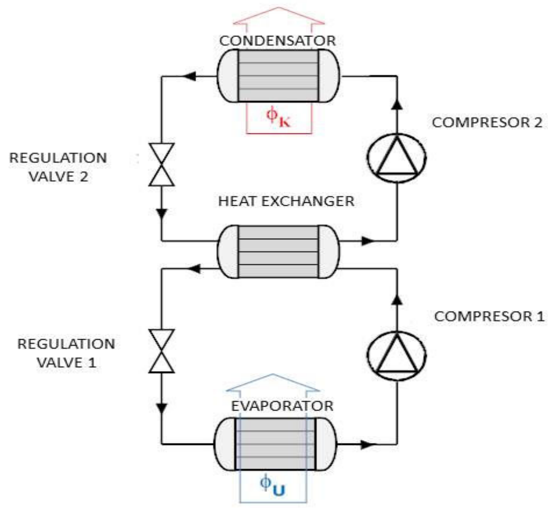

Fig. 8: The two-stage heat pump with a heat exchanger

1) Equations for the calculation

Calculation of the characteristic parameters of the operation of the two-stage heat pump with a heat exchanger is more simply conducted according to the principle of the two one-stage pumps, where it is assumed that the heat flow of the condensation of the first stage $\Phi_{K,1}$, is equal to the heat flow of boiling of the second stage $\Phi_{U,2}$. It holds that:

$$\Phi_{K,1} = q_{m,S,1} \cdot (h_{g,2} - h_{l,3}) \qquad \text{(W)} \qquad (6)$$

$$\Phi_{K,1} = \Phi_{U,2} \qquad \text{(W)} \qquad (7)$$

It follows that the mass flow of the refrigerant on the inlet side of the compressor of the second stage of the heat pump is equal:

$$q_{m,S,2} = \frac{\Phi_{U,2}}{(h_{g,5} - h_{l,8})} \qquad \text{(kg/s)} \qquad (8)$$

where:

$h_{g,5}$ - specific enthalpy of the refrigerant which enters the compressor in the second stage of the heat pump (J/kg·K)

$h_{l,8}$ - specific enthalpy of the liquid refrigerant which enters the evaporator in the second stage of the heat pump (J/kg·K)

Heat number of the two-stage heat pump with a heat exchanger is calculated:

$$COP_G = \frac{\Phi_{TČ,2}}{P_{K,1} + P_{K,2}} \qquad (/) \qquad (9)$$

Condensation temperature of the first stage $T_{K,1}$ and the boiling temperature of the second stage $T_{U,2}$ of the refrigerant are calculated using the method of the average value between the temperatures of boiling of the first stage $T_{U,1}$ and the temperature of the condensation of the second stage $T_{K,2}$.

$$T_M = \frac{T_{U,1} + T_{K,2}}{2} \quad \text{(K)} \qquad (10)$$

The possibility of using refrigerant in both stages requires that we determine the characteristics of the compressor for each stage separately according to the same principle as for the one-stage heat pump [11, 12].

2) Calculation programme

DATA FOR THE CALCULATION OF THERMODYNAMIC DATA Refrigerants
REFRIGERANT: R-717 Ammonia
CONSTANTS FOR THE CALCULATION OF PRESSURE AT THE TEMPERATURE Tu II Tk

$au := 0$
$AU := 0$
$bu := -1190.7$
$BU := -117$
$cu := 9.994$
$CU := 9.944$

CONSTANTS FOR THE CALCULATION OF THE REFRIGERANT LIQUID DENSITY

$at := -0.003$
$AT := -0.008$
$bt := 0.538$
$BT := 3.950$
$ct := 746.8$
$CT := 190.6$

CONSTANTS FOR THE CALCULATION OF THE REFRIGERANT VAPOUR DENSITY

$ap := 0.001$
$AP := 0.0000$
$bp := -0.69$
$BP := -0.010$
$cp := 81.25$
$CP := 1.941$

CONSTANTS FOR THE CALCULATION OF THE REFRIGERANT LIQUID ENTHALPY

$xt := 4.81$
$XT := 5.438$
$yt := -1117.$
$YT := -1319.$

CONSTANTS FOR THE CALCULATION OFTHE REFRIGERANT VAPOUR ENTHALPY

$xp := -0.010$
$XP := 0.145$
$yp := 7.007$
$YP := -43.24$
$zp := 349.9$
$ZP := 5732.$

TYPE OF COMPRESSOR: 8RWH
NUMBER OF REVOLUTIONS OF THE COMPRESSOR [1/min]:

$N := 1000 \cdot \mathrm{min}^{-1}$

OPERATION OF THE COMPRESSOR [%]:

$\mu := 1$

EFFICIENCY OF THE COMPRESSOR [/]:

$\eta := 0.72$

TEMPERATURE OF BOILING OF THE REFRIGERANT [K]:

$Tu := 298.15K$

TEMPERATURE OF CONDENSTATION OF THE REFRIGERANT [K]:

$Tk := 325.65K$

AVERAGE TEMPERATURE [K]:

$$Ts := \frac{(Tu + Tk)}{2}$$

$Ts = 311.9K$

COMPRESSIBILITY FACTOR I:

$\chi := 1.49$

UNUSED SPACE FOR FILLING THE COMPRESSOR $\varepsilon := 0.14$

VOLUME FLOWOF THE COMPRESSOR [m3/h]:

$$Vk := 0.0833\frac{m^3}{s}$$

PRESSURE ON THE INLET SIDE OF THE COMPRESSOR [Pa]:

$$Pu := 10^{\left[au\cdot\left(\frac{K}{Tu}\right)^2 + bu\cdot\left(\frac{K}{Tu}\right) + cu\right]}\cdot Pa$$

$Pu = 1002703.7Pa$

REFRIGERANT LIQUID DENSITY AT THE BOILING TEMPERATURE

$$\sigma tu := \left[at\cdot\left(\frac{Tu}{K}\right)^2 + bt\cdot\left(\frac{Tu}{K}\right) + ct\right]\cdot\frac{kg}{m^3}$$

$$PK := 10^{\left[AU\cdot\left(\frac{K}{Tk}\right)^2 + BU\cdot\left(\frac{K}{Tk}\right) + CU\right]}\cdot Pa \qquad \sigma tu = 605.176\frac{kg}{m^3}$$

PRESSURE ON THE DELIVERY SIDE OF THE COMPRESSOR [Pa]:

$PK = 2169860Pa$

SPECIFIČ ENTHALPY OF THE REFRIGERANT VAPOUR ENTHALPY AT THE CONDENSATION TEMPERATURE

$$Hpk := \left[xp\cdot\left(\frac{Tk}{K}\right)^2 + yp\cdot\left(\frac{Tk}{K}\right) + zp\right]\cdot\frac{J}{kg}$$

$$Hpk = 1486.6\frac{1}{kg}\cdot J$$

SPECIFIČ ENTHALPY OF REFRIGERANT LIQUID AT THE TEMPERATURE OF CONDENSATION

$$Htk := \left(xt \cdot \frac{Tk}{K} + yt\right) \cdot \frac{J}{kg}$$

$$Htk = 450.63 \frac{1}{kg} \cdot J$$

REFRIGERANT VAPOUR DENSITY AT THE TEMPERATURE OF CONDENSATION

$$\rho pk := \left[ap \cdot \left(\frac{Tk}{K}\right)^2 + bp \cdot \left(\frac{Tk}{K}\right) + cp\right] \cdot \frac{kg}{m^3}$$

$$\rho pk = 14.977 \frac{kg}{m^3}$$

REFRIGERANT LIQUID DENSITY AT THE CONDENSATION TEMPERATURE

$$\sigma tk := \left[at \cdot \left(\frac{Tk}{K}\right)^2 + bt \cdot \left(\frac{Tk}{K}\right) + ct\right] \cdot \frac{kg}{m^3}$$

$$\sigma tk = 561.662 \frac{kg}{m^3}$$

$$rK := \frac{PK}{Pu}$$

THE RATIO BETWEEN THE PRESSURES

$$rK = 2.164$$

THE TEMPERATURE AT THE OUTLET FROM THE COMPRESSOR [K]:

$$T2 := Tu \cdot \left(\frac{PK}{Pu}\right)^{\left(\frac{\chi - 1}{\chi}\right)}$$

$$T2 = 385.248K$$

SPECIFIČ ENTHALPY OF THE REFRIGERANT LIQUID AT THE BOILING TEMPERATURE

$$Htu := \left(XT \cdot \frac{Tu}{K} + YT\right) \cdot \frac{J}{kg}$$

$$Htu = 302.178 \frac{1}{kg} \cdot J$$

REFRIGERANT VAPOUR DENSITY AT THE TEMPERATURE OF BOILING

$$\rho pu = 8.275 \frac{kg}{m^3}$$

$$\rho pu := \left[ap \cdot \left(\frac{Tu}{K}\right)^2 + bp \cdot \left(\frac{Tu}{K}\right) + cp\right] \cdot \frac{kg}{m^3}$$

SPECIFIC ENTHALPY OF REFRIGERANT VAPOUR AT THE TEMPERATURE OF CONDENSATION

$$Hp2 := \left[xp \cdot \left(\frac{T2}{K}\right)^2 + yp \cdot \left(\frac{T2}{K}\right) + zp\right] \cdot \frac{J}{kg}$$

$$Hp2 = 1446.6 \frac{1}{kg} \cdot J$$

FILLING RATE OF THE CYLINDER IN THE COMPRESSOR

$$\lambda := 1 - \varepsilon \cdot \left[rK^{\left(\frac{1}{\chi}\right)} - 1\right]$$

$$\lambda = 0.906$$

THE REAL VOLUME FLOWOF THE COMPRESSOR

$$vK := Vk \cdot \lambda$$

$$vK = 0.075 \frac{m^3}{s}$$

$$qv := vK \cdot \frac{\rho pu}{\rho pk}$$

VOLUME FLOW OF THE REFRIGERANT IN THE CMPRESSOR

$$qv = 0.042 \frac{m^3}{s}$$

ADIABATIC COMPRESSION STRENGTH

$$Pad := vK \cdot Pu \cdot \left(\frac{\chi}{\chi - 1}\right) \cdot \left[rK^{\left[\frac{(\chi - 1)}{\chi}\right]} - 1\right]$$

$$Pad = 66551.62W$$

COMPRESSOR STRENGTH

$$Pk := \frac{Pad}{\eta}$$

$$Pk = 92432.8W$$

VOLUME FLOWON THE INLET SIDE OF THE COMPRESSOR

$$vK = 0.075 \frac{m^3}{s}$$

MASS FLOW ON THE INLET SIDE OF THE COMPRESSOR

$$qms := vK \cdot \rho pu$$

$$qms = 0.624 \frac{kg}{s}$$

VOLUME FLOWON THE DELIVERY SIDEOF THE COMPRESSOR

$$qv = 0.042 \frac{m^3}{s}$$

MASS FLOW ON THE DELIVERY SIDE OF THE COMPRESSOR

$$qmt := qv \cdot \rho pk$$

$$qmt = 0.624 \frac{kg}{s}$$

HEAT FLOW OF THE EVAPORATED PART OF THE FIRST STAGE OF THE HEAT PUMP [W]:

$$\Phi u1 := qms \cdot (Hpu - Htk) \cdot 100($$

$$\Phi u1 = 642010.4W$$

HEAT FLOW OF THE CONDENSATION PART OF THE FIRST STAGE OF THE HEAT PUMP [W]:

$$\Phi k1 := qms \cdot (Hp2 - Htk) \cdot 100($$

$$\Phi k1 = 621709.8W$$

HEAT NUMBER OF THE FIRST STAGE OF THE HEAT PUMP [/]:

$$COPG := \frac{\Phi k1}{Pk}$$

$$COPG = 6.726$$

HEAT FLOW OF THE CONDENSER IN THE FIRST STAGE IS EQUAL TO THE HEAT FLOW OF THE EVAPORATOR IN THE SECOND STAGE

$$\Phi K := \Phi k1$$

$$\Phi U := \Phi K$$

$$\Phi U = 621709.8W$$

DATA FOR THE CALCULATION OF THERMODYNAMIC DATA FOR REFRIGERANTS IN THE SECOND STAGE
HLADIVO: R-600a (izo-butan)

CONSTANTS FOR THE CALCULATION OF PRESSURE AT THE TEMPERATURE Tu I Tk

$au := 0$

$AU := 0$

$bu := -1129.8$

$BU := -1112.6$

$cu := 9.3334$

$CU := 9.2784$

CONSTANTS FOR THE CALCULATION OF REGRIGERANT LIQUID DENSITY

$at := -0.0026$

$AT := -0.0065$

$bt := 0.2975$

$BT := 2.7801$

$ct := 693.83$
$CT := 292.66$
CONSTANTS FOR THE CALCULATION OF THE REFRIGERANT VAPOUR DENSITY
$ap := 0$
$AP := 0$
$bp := 0.2583$
$BP := 0.6005$
$cp := -67.669$
$CP := -177.24$
CONSTANTS FOR THE CALCULATION OF REFRIGERANT LIQUID ENTHALPY
$xt := 2.463$
$XT := 2.8062$
$yt := -474.45$
$YT := -584.34$
CONSTANTS FOR THE CALCULATION OF REFRIGERANT VAPOUR ENTHALPY
$xp := 0$
$XP := 0$
$yp := 1.3407$
$YP := 1.2157$
$zp := 189.57$
$ZP := 230.14$
TYPEOF THE COMPRESSOR: 8RWH
Broj obrtaja OF THE COMPRESSOR [1/min]:
$N := 1000 \cdot min^{-1}$
OPERATION OF THE COMPRESSOR [%]:
$\mu := 1$
UTILIZATION OF THE COMPRESSOR [/]:
$\eta := 0.72$
TEMPERATURE OF BOLINIHG OF THE REFRIGERANT [K]:
$TU := Tk$
TEMPERATURE OF CONDENSATION OF THE REFRIGERANT [K]:
$TK := 353.15K$
AVERAGE TEMPERATURE [K]:
$$TS := \frac{(TU + TK)}{2}$$
$TS = 339.4\ K$
COMPRESSIBILITY FACTOR:
$\chi 2 := 1.207$
PRESSURE ON THE INLET SIDE OF THE COMPRESSOR [Pa]:
$$PU := 10^{\left[au \cdot \left(\frac{K}{TU}\right)^2 + bu \cdot \left(\frac{K}{TU}\right) + cu\right]} \cdot Pa$$
UNUSED SPACE FOR FILLING OF THE COMPRESSOR
$\varepsilon := 0.14$
$PU = 731191.4Pa$
SPECIFIC ENTHALPY OF REFRIGERANT VAPOUR AT THE BOILING TEMPERATURE
$$HpU := \left[xp \cdot \left(\frac{TU}{K}\right)^2 + yp \cdot \left(\frac{TU}{K}\right) + zp\right] \cdot \frac{J}{kg}$$
$$HpU = 626.169 \frac{1}{kg} \cdot J$$
SPECIFIC ENTHALPY OF THE REFRIGERANT LIQUID AT THE BOILING TEMPERATURE
$$HtU := \left(XT \cdot \frac{TU}{K} + YT\right) \cdot \frac{J}{kg}$$
$$HtU = 329.499 \frac{1}{kg} \cdot J$$
REFRIGERANT VAPOUR DENSITY AT THE BOILING TEMPERATURE
$$\rho pU := \left[ap \cdot \left(\frac{TU}{K}\right)^2 + bp \cdot \left(\frac{TU}{K}\right) + cp\right] \cdot \frac{kg}{m^3}$$
$$\rho pU = 16.446 \frac{kg}{m^3}$$
REFRIGERANT LIQUID DENSITY AT THE BOILING TEMPERATURE
$$\sigma tU := \left[at \cdot \left(\frac{TU}{K}\right)^2 + bt \cdot \left(\frac{TU}{K}\right) + ct\right] \cdot \frac{kg}{m^3}$$
$$\sigma tU = 514.986 \frac{kg}{m^3}$$
PRESSURE ON THE DELIVERY SIDE OF THE COMPRESSOR [Pa]:
$$PK := 10^{\left[AU \cdot \left(\frac{K}{TK}\right)^2 + BU \cdot \left(\frac{K}{TK}\right) + CU\right]} \cdot Pa$$
$PK = 1342447.7Pa$
SPECIFIČ ENTHALPY OF THE REFRIGERANT VAPOUR AT THE CONDENSATION TEMPERATURE
$$HpK := \left[XP \cdot \left(\frac{TK}{K}\right)^2 + YP \cdot \left(\frac{TK}{K}\right) + ZP\right] \cdot \frac{J}{kg}$$
$$HpK = 659.5 \frac{1}{kg} \cdot J$$
SPECIFIC ENTHALPY OF THE REFRIGERANT LIQUID AT THE CONDENSATION TEMPERATURE
$$HtK := \left(XT \cdot \frac{TK}{K} + YT\right) \cdot \frac{J}{kg}$$
$$HtK = 406.67 \frac{1}{kg} \cdot J$$
REFRIGERNAT VAPOR DENSITY AT THE CONDENSATION TEMPERATURE
$$\rho pK := \left[AP \cdot \left(\frac{TK}{K}\right)^2 + BP \cdot \left(\frac{TK}{K}\right) + CP\right] \cdot \frac{kg}{m^3}$$
$$\rho pK = 34.827 \frac{kg}{m^3}$$
REFRIGERANT LIQUID DENSITY AT THE CONDENSTATION TEMPERATURE
$$\sigma tK := \left[AT \cdot \left(\frac{TK}{K}\right)^2 + BT \cdot \left(\frac{TK}{K}\right) + CT\right] \cdot \frac{kg}{m^3}$$
$$\sigma tK = 463.805 \frac{kg}{m^3}$$
THE RATIO BETWEEN PRESSURESOF THE COMPRESSOR
$$rK2 := \frac{PK}{PU}$$
$rK2 = 1.836$
TEMPERATURE AT THE OUTLET OF THE COMPRESSOR [K]:
$$T3 := TU \cdot \left(\frac{PK}{PU}\right)^{\left(\frac{\chi 2 - 1}{\chi 2}\right)}$$
$T3 = 361.413K$

3) *Results of the calculation*

Calculation of characteristic parameters was conducted for the following refrigerants:

- R-407c in the first stage and R-600a in the second stage
- R-717 in the first stage and R-600a in the second stage
- R-290 in the first stage and R-600a in the second stage

The results are given in Fig. 9 and Fig. 10, which show the strength which is necessary for the operation of the compressor and the heat flow of the heat pump. Calculations were conducted for different temperatures (t_U) of the refrigerant: from 5,0°C to 25,0°C and the constant temperature of condensation t_K = 70°C.

The values of the heat number of the two-stage heat pump with a heat exchanger were calculated for the condensation temperatures of the refrigerant t_K = 60°C and t_K = 80°C. Fig. 11 shows the dependence between the COP and the refrigerant boiling temperature.

For the calculation of all suggested combinations of refrigerants the compressors chosen were: a piston compressor with the flow of 300 m^3/h for the first stage, and a piston compressor with the flow of 700 m^3/h for the second stage.

The main objective of the two-stage pump with a heat exchanger is the use of a wider range of temperatures between the boiling temperature and the condensation temperature. With the proposed way of implementation of the heat pump, the low-temperature sources are maximally used. Moreover, the heat obtained can be used for long-distance or combined heating

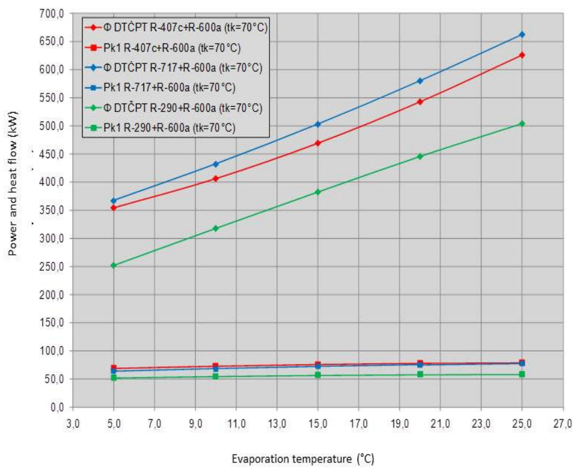

Fig. 9: The first stage of the two-stage heat pump with the heat exchanger

The following combinations of refrigerants are optimal for the operation of the two-stage heat pump with a heat exchanger:

- R-717 in the first stage and R-600a in the second stage,
- R-407c in the first stage and R-600a in the second stage.

The first combination gives the highest heat flow for the purpose of heating, the use of strength is equal or lower than that of the other proposed combinations of refrigerants. The problem of this combination is a narrow range of application , because at larger difference between the the condensation temperatures and boiling temperatures the operation of the compressor is relimited due to high operating pressures R-717. For wider operating conditions it is necessary to design the type of compressor which operates at higher operating pressures

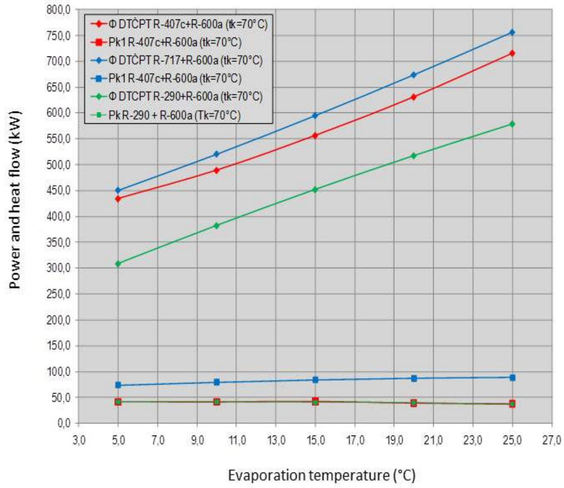

Fig. 10: The second stage of the two-stage heat pump with a heat exchanger

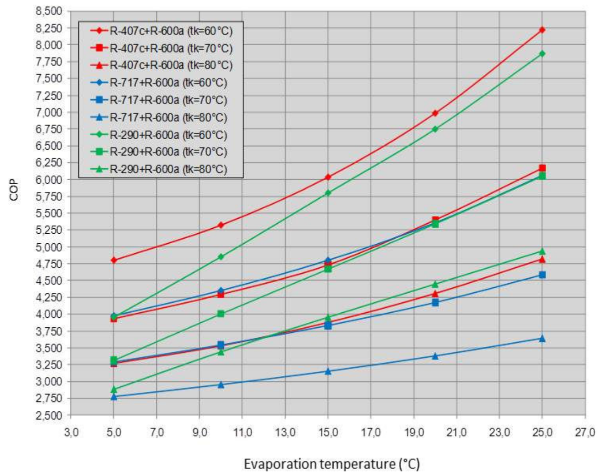

Fig.11: COP of the two-stage heat pump witha heat exchanger

The other combination of the refrigerants gives lower values of the heat flow. Here we are not restricted concerning the operation of the compressor in both phases. From the environmental aspect, we used F-gases in the first stage, which have a negative effect on global warming; however, R-407c refrigerant is currently used as a replacement for the HCFC refrigerant.

Other combinations of refrigerants give poorer results. The operating conditions of the compressor are limited or completely impossible.

III. Economic Analysis

Business decisions which determine business conditions of a company affect market competition and innovation and usually direct our business decisions as to how to make the best business policy in a certain period of time. When deciding whether to refuse or accept an industrial project, the key factor is the economic analysis which is based on the two assumptions:

- investment, which is a single sum of money for the implementation of the project in practice
- the surplus income in comparison to the expenses, which means a difference between the revenues and the expenses for maintenance and operation .

The economic analysis of a project refers to the determination of the values of the above mentioned assumption using economic methods which differ in requirements and accuracy. The choice of the method depends on the stage of the development of the project. For the economic analysis of the justification of the application of the high -temperature heat pump for heating the method of net present value was used (NPV). The annual inflation rate is also taken into consideration.

NPV represents the sum of all values of money flows. The rule on deciding on the investment based on NPV is that the investment is accepted if NPV is higher than 0 or vice versa. Apart from this method, the succefulness of the investment can be assessed using a profitability coefficient (K), which represents the relation between the net present value from selling and the sum of values of all expenses, on one side, and the deadline of return on the other [15].

A. *Economic justifiability of the application of the high-temperature heat pump*

The decision on the investment is much easier and simpler if it is based on the calculated parameters for individual implementations of heat pumps. The calculation included the models with most suitable refrigerants. The economic analysis stated that the costs of the investment are covered from personal sources and from loans in 30% to 70% ratio. The present value of the investment costs C_{INV} je was calculated according to the equation;

$$C_{INV} = C_0 + \sum_{j=0}^{N} \frac{a_n \cdot C_{T\check{C}}}{(1+r)^j} \text{ (EUR/year)} \tag{11}$$

where:

C_0 - own funds (EUR)

$C_{T\check{C}}$ - the cost of the heat pump (EUR)

r - discount rate (/)

N - lifetime of the system (years)

Annuity factor $a_n$is calculated using the equation:

$$a_n = \frac{r_a \cdot (1+r_a)^{n_1}}{(1+r_a)^{n_1} - 1} \text{ (/)} \tag{12}$$

r_a - discount stage annuity (/)

n_1 - time of return annuity (/)

Maintenance costs of the heat pump were evaluated at 2 % of the purchase price. Net present value of expenses with inflation rate included was calculated using the equation:

$$C_S = \sum_{j=0}^{N} \frac{0.02 \cdot C_{T\check{C}} \cdot (1+r_j)^j}{(1+r_j+r)^j} \text{ (EUR/year)} \tag{13}$$

Net present value of electric power costs for the operation of the compressor of the pump were calculated using the equation:

$$C_{PS} = \sum_{j=0}^{N} \frac{C_E \cdot P_K \cdot t_1 \cdot t_2 \cdot (1+r_j)^j}{(1+r_j+r)^j} \text{ (EUR/year)} \tag{14}$$

Net present value of the income from the heat produced considering the inflation rate is calculated using the equation:

$$C_P = \sum_{j=0}^{N} \frac{C_T \cdot \Phi_{T\check{C}} \cdot t_1 \cdot t_2 \cdot (1+r_j)^j}{(1+r_j+r)^j} \text{ (EUR/year)} \tag{15}$$

Net present value of the heat income, considering the investment costs and maintenance costs , as well as the consumption of electric energy for the operation of the compressor is calculated using the equation:

$$C = C_P - (C_{INV} + C_S + C_{PS}) \text{ (EUR/year)} \tag{16}$$

where :

r_j - inflation rate (/)

C_E- electric power cost (EUR/kWh)

C_T -the cost of heat for heating (EUR/kWh)

t_1 - operation time per day (h/day)

t_2 - operation time per year (days/year)

The calculation of economic parameters was conducted for all three heat pumps. The cost of the heat pump was calculated based on the costs of individual process devices. The cost of electric power is a known value. The time of operation of the pump is 2160 working hours [12].

Table 2 contains data for the calculation of different implementations of heat pumps. Due to easier economic evaluation and comparison of the very method of implementation, the results for high-temperature pumps are shown in Fig. 12. It shows NPVs at different temperatures of boiling of refrigerants, and at a constant condensation temperature t_K=70°C.

Table 2: Data for the calculation of the economic analysis of different implementations of the heat pump

Implementation of the heat pump	ETČ	ETČ	DTČR	DTČP	DTČP
Refrigerant	R-600a	R-600a/R-290	R-600a	R-717/R-600a	R-407c/R-600a
Temperature of boiling (t_U) [°C]	20,0	20,0	20,0	20,0	20,0
Temperature of condensation(t_K) [°C]	70,0	70,0	70,0	70,0	70,0
Heat flow of the heat pump ($\Phi_{TČ}$) [kW]	288,6	585,36	473,0	674,2	631,1
Compressor strength (P_K) [kW]	75,3	116,0	111,1	161,6	116,9
Operation time per day (t_1) [h/day]	18	18	18	18	18
Operation time per year (t_2) [days/year]	120	120	120	120	120
Heat pump lifetime (N) [year]	20	20	20	20	20
Own funds (C_o) [EUR]	16.800	16.800	28.200	31.800	31.800
Cost of the heat pump ($C_{TČ}$) [EUR]	56.000	56.000	94.000	106.000	106.000
Electric power costs (C_E) [EUR/kWh]	0,07	0,07	0,07	0,07	0,07
Cost of heat for heating (C_T) [EUR/kWh]	0,0325	0,0325	0,0325	0,0325	0,0325
Discount rate (r) [%]	7,00	7,00	7,00	7,00	7,00
Inflation rate (r_j) [%]	1,20	1,20	1,20	1,20	1,20

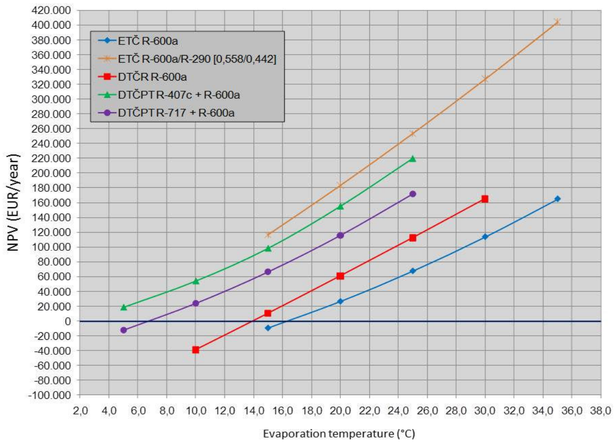

Fig. 12: NPV depending on boiling temperature of the refrigerant

IV. Conclusion

With a two-stage heat pump with a flash expander, the best operating conditions are with the refrigerant R-600a. For the proposed implementation of the heat pump, the ratio between the pressures of the compressed refrigerants decreases. Boiling of the refrigerant at the temperature of t_U = 20°C is economical. The condensation temperature is t_K = 70°C. Then the heat flow is 408,0 kW, the consumption of energy for the operation of both compressors is 111,1 kW. It is suggested to use the same compressor in the first and the second stage as with the one-stage pump.

The two-stage heat pump with a heat exchanger consists of two one-stage pumps. Its advantage is that we can use, in each stage, different refrigerants.

The analysis of thermo-physical properties of refrigerants established that the operation of a two-stage heat pump with a heat exchanger is the cheapest using refrigerants R-407c in the first stage and R-600a in the second stage of the heat pump. If the condensation temperature of the refrigerant in the second stage is t_K=70°C, the operation of the heat pump of the first stage is economically justifiable at the boiling temperature of t_U = 10°C. In the first stage, the type of the compressor which is recommended is WA or WBH, in the second stage only WBH. The heat flow of the two-stage heat pump with a heat exchanger in that case is 489,4 kW, the total consumption of energy for the operation of both compressors is 113,9 kW. The suggested implementation of the heat pump is one of the methods for maximum utilization of low -temperature sources of energy for heating of residential units.

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