Optimal Proportion of Refrigerants in Single-Stage High-Temperature Heat Pump

M. Jotanovic, M. Miscevic, B. Kulcar, D. Goricanec

Abstract**—** The paper presents types of refrigerants with their characteristics for use in single-stage high-temperature heat pump. A mathematical model is designed; based on the use of the modeling language GAMS a simulation of optimum mixture proportions of two refrigerants was made, which allows for the heat pump to operate the most efficient. Software package Math Cad Professional was used to make calculations of the characteristic properties of the heat pump.

Keywords**—** Heat transfer, single-stage high-temperature pump, refrigerant, simulation of effect, characteristic features of the pump

I. Introduction

According to the scientific research, it is known that the atmosphere has been rapidly warming since the mid-20th century. A lot of changes in almost each part of the world make us perceive the global warming. Most of the studies show that the man is the one who is guilty for the global warming. He and his unthinking actions (industry, traffic…) destroy the balance of nature. This is the reason why we have to prevent further warming by reducing gas emission. We also have to find new technological solutions in the fields of process and heating techniques [1, 2, 3].

According to Kyoto agreement, the gas emission that causes the greenhouse effects has to be reduced by EU in comparison to the 1990 emission in the same area. These gases include- CO_2, CH_4, N_2O, and anthropogenic F- gases. Substances that also make the ozone layer deplete include CFC, partly halogenated carbon hydrates and halons.

Having in mind the accepted agreements for reaching certain goals in the field of energy, we should get down to development of getting and using cheap energy and energy, based on low carbon technologies, that is acceptable for the nature.

Using high temperature heating pumps, which use various agents for refrigerating and enable energy transmission from lower temperature layers to higher is one of the ways for getting environmentally acceptable energy for heating.

The choice of refrigerating agent is based on physical-chemical characteristics of the existing agent and the ability of certain technological solutions that enable the wanted parameters of the operating heat pumps. Heating pump characteristics in regard to different agents are processed in MathCAD programme, the simulation of the optimal proportion of the mixture of agent for refrigerating is processed in programme package GAMS. Having in mind the choice of the agents, the comparison and the evaluation of positive and negative effects of the agent is done.

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II. Refrigerating Agent

These are functional liquids in heating pumps that absorb heat in lower temperature levels and transmit it to higher temperature levels. Functioning of the heating pumps is very dependent on the choice of refrigerating means. The goal of those agents is not just heating transmission, but they also have to fulfil some other conditions:

- Physical and chemical characteristics when changing the aggregate states should be very good, because they would be used for a long time
- They should have the minimum effect on nature, and they must not be harmful for humans or animals
- They should mix with materials that are found in the environment
- Their price should be acceptable

The ideal refrigerating agent has not been found yet, and probably it will never be found. Most of the agents that are used today are halogenated, which agent that they have radically tied halogenids (F, Cl, Br). Their effect on the environment is harmful because they destroy the ozone layer. This is why the new agents are being developed. These agents must have good thermodynamic characteristics, because adequate heating pump functioning is based on them.

A. *Refrigerating agent analysis*

It is proved that agents like CHCs and HCFC destroy the ozone layer. Other anthropogenic fluorinated gasses cause global warming. This is why the usage of CHC and CHFC in heating pumps is forbidden. As a current solution various fluorinated carbon hydrates are used, but they still have the negative effects. Developing new technologies and improving the pumps, organic means, which don't have harmful effects on environment, are increasingly used. The development of refrigerating agent is shown in Fig. 1.

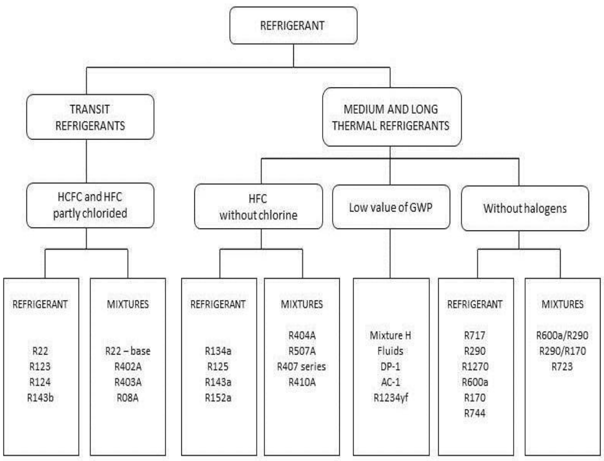

Fig. 1: Directions of the refrigerating agent development

The influence of the agent on depleting of the ozone layer is estimated by ODP index "Ozone Depleting Potential", which is determined for each agent separately and it is compared to the CFC 11 means, whose ODP is 1. The lower ODP is, the less ozone depleting is. Gases are also harmful for the ozone, that's why the ozone layer depleting is related to the process of global environment changing. GWP "Global Warming Potential" for each harmful gas shows the effect of the molecules of that gas on global warming and it is compared to molecule CO_2, whose value is 1. The effect of certain agent on ozone and global warming is shown in table 1 [4].

Table 1: The effect of certain agent ton ozone and global warming

Agent sign	Chemical formula	ODP	GWP
R-12	CCl_2F_2	1.00	8100
R-22	$CHClF_2$	0.055	1500
R-32	CH_2F_2	0.0	550
R-125	CHF_2CF_3	0.0	3400
R-134a	CF_3CH_2F	0.0	1100
R-143a	CF_3CH_3	0.0	4300
R-245fa	$CF_3CH_2CHF_2$	0.0	950
R-407c	R-32/R-125/R134a	0.0	1548.5

B. Marking and division of the refrigerating means

American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) standardized the division and marking of the agents by 34-92 standard. They are divided in five groups:

- halogenated carbohydrates,
- Isotropy
- zoetropes
- organic substances and
- inorganic substances.

The example for marking the agent is R 22, R-22, Refrigerant 22 or Trade name 22. The importance of marking is seen because of the complexity of chemical name giving. The group of halogenated carbohydrates contains halogen elements (Cl), (F), (Br) and (I) as well as (H). For marking the halogenated carbohydrates, the following system is used:

- the first number on the right shows the number of F atoms
- the second number is higher for one than the number of H atoms
- the third is for one lower than the number of C atoms

If the third number on the right is 0, we don't write it. Series 600 marks various organic mixtures, while the series 700 marks inorganic mixtures. Series 500 belongs to Isotropy [5].

C. Security of the refrigerating agent

Having in mind their being toxic and inflammable, these agents are divided in six groups (A1, A2, A3, B1, B2 and B3). Table 2 shows the division of the agents according to their security.

Table 2: The division of the agents into their security groups

Agent mark	Name of the agent	Chemical formula	Security group
R-12	dichlorodifluoromethane	CCl_2F_2	A1
R-22	chlorodifluoromethane	$CHClF_2$	A1
R-32	di-fluoromethane	CH_2F_2	A2
R-125	1,1,1,2,2-pentafluoroetan	CHF_2CF_3	A1
R-134a	1,1,1,2-tetrafluoroetan	CF_3CH_2F	A1
R-143a	1,1,1-trifluoroetan	CF_3CH_3	A2
R-245fa	1,1,1,3,3-pentafluoropropan	$CF_3CH_2CHF_2$	A2
R-290	propane	$CH_3CH_2CH_3$	A3
R-407c	mix R-32 / R-125 / R-134a		A1
R-600a	isobutene	$CH(CH_3)_3$	A3
R-717	ammoniac	NH_3	B2
R-718	water	H_2O	A1

Group A1 agents are the least dangerous, group B3 are the most dangerous. Capital letter refers to the toxicity of the agent when the concentration is less than 400 ml/m^3, the number refers to the flammable characteristics. The marks mean:

- grade A: toxicity is not identified
- grade B: proof of toxicity is identified
- grade 1: it is not flammable at 18°C and the pressure of 101 kPa
- grade 2: when the concentration is 0.10kg/m^3 at 21°C and the pressure 101kPa it shows certain flammable characteristics.

- Heating burning is less than 19.000kJ/kg.
- Grade 3:very flammable at concentration 0.10kg/m^3, at 21°C and the pressure 101kPa
- Heating burning is higher or equal to 19.000kJ/kg.
  Toxicity of the agents influences their usage.

D. Physical and chemical characteristics of the means

While the heating pump operates, the agent is exposed to great changes in pressure and temperature. This is why the physical and chemical characteristics of the agents are very important.

For good heat transmission it is good if the heating transmission and transfer coefficient between the refrigerator and the metal part is as bigger as it is possible, because it influences the size of the heat exchanger. Crystallization temperature has to be lower than the lowest possible temperature in the heating pump. Critical pressure and the temperature have to be as higher as possible above the operating zone of the device. The proportion among compressor pressures have to be as low as possible, which agent that the condensation pressure has to be as low as possible, while the steaming pressure has to be as high as possible. In order to have the minimum waste while passing through pipes, vents and heat exchangers, the viscosity has to be as low as possible.

The quantity of water in the agent has to be as low as possible, due to the lifetime of the pump. Chemical structure can be stable without other gases presence. Table 3 and 4 shows some important physical and chemical characteristics of the refrigerating agent that are important when planning the process.

Table 3: Basic chemical and physical characteristics of the agent [6, 7]

means	Chemical formula	Mol. weight (g/mol)	Cooking temperature (°C)	Crystallization temperature (°C)	Critical temperature. (°C)	Critical pressure (kPa)	Critical volume (L/kg)
R-22	$CHClF_2$	86.48	-40.76	-160.0	96.0	4974	1.904
R-32	CH_2F_2	52.02	-51.80	-136.0	78.4	5830	2.326
R-125	CHF_2CF_3	120.03	-48.57	-103.15	66.3	3630.6	/
R-134a	CF_3CH_2F	102.30	-26.16	-96.6	101.1	4067	1.81
R-143a	CF_3CH_3	84.00	-47.27	-111.81	72.71	3761	2.32
R-245fa	$CF_3CH_2CHF_2$						
R-290	$CH_3CH_2CH_3$	44.10	-42.09	-187.7	96.70	4248	4.53
R-407c	mixture	86.20	-43.79	/	86.1	4635	1.98
R-600	$CH3CH_2CH_2CH_3$	58.13	-0.5	-138.5	152.0	3794	4.383
R-600a	$CH(CH_3)_3$	58.13	-11.73	-160.0	135.0	3645	4.526
R-717	NH_3	17.03	-33.3	-77.7	1330.0	11417	4.245
R-718	H_2O	18.02	100.0	0	373.99	22064	3.11

Table 4: Other chemical and physical characteristics of the means

means	R-22	R-134a	R-245fa	R-290	R-407c	R-600a	R-717
Form	gas	gas	utek. gas	gas	utek. gas	gas	gas
Colour	colourless	colourless	colourless	colourless	colourless	colourless	colourless
Smell	Ether	Ether	Weak	Considera ble	Ether	Sweet	Ammoniac
Temperature of burning	Non-flammable	> 750 °C	/	470 °C	Non-flammable	450 °C	630 °C
Explosion danger	No.	No	No.	Yes .	No	yes.	yes.
Explosion limits upper / lower In the air	/	/	/	2.1 / 9.5 vol%	/	1.3 / 8.5 vol %	15 / 30 vol %
Density (kg/l) 0°C, p=1.013 bar	1,194	1,21	1.226	1.56	1.136	2.0	0.6
SolubilityIn water 20 °C	3.0 g/l	0.15 wt % mg/100g	0.13 g/l	75 mg/l	unimportant	5.4 mg/l	Hydrolyzed

E. Thermodynamic data on the means

Circular Kam's process can give good results if the specific cooking temperature is high. Specific enthalpy of the agent steaming influences the size of the heater exchanger as well as the pipes and vents dimensions. If the specific refrigerating heat is higher, the dimensions of the device are lower.

According to the agent analysis and having in mind the environmental influence, characteristics, development direction the text processes:

- R-717 ammoniacNH_3
- R-600a isobutene $CH(CH_3)_3$
- R-290 propane $CH_3CH_2CH_3$
- R-134a 1, 1, 1, 2, tetraflourethane CF_3CH_2F
- R-245fa 1,1,1,3,3,-pentafluoropropane $CF_3CH_2CHF_2$
- R-407 mixture (R-32/R-125/R-134a)

The data on the agents are shown just for the intervals where the calculating of the heat pump is done [7, 8]

III. High temperature heating pumps

Using heating pumps with the renewable energy sources for heating, preparing sanitary water and producing electricity is technically limited due to relatively low temperatures of the sources.

The need for higher temperatures directed the development of the simple heating pump into the high temperature heating pump, where we achieve adequate proportion of the temperature of cooking and condensation.

Having in mind the temperature of the renewable energy sources technological capacity of the pump and economic justification, we can analyse following types of high temperature heating pumps:

- single phase high temperature heating pump
- two phase high temperature heating pump and
- Two phase high temperature heating pump with heat exchanger.

Functioning of the high temperature heating pump depends on the choice of the refrigerant in compressor capacity.

A. Single phase high temperature heating pump

Single phase high temperature heating pump is, basically, a common heating pump, where we can achieve the wanted temperature difference between the condensation temperature (t_k) and the cooking temperature (t_u) of 35°C to 50°C if we choose the adequate refrigerating agent and compressor. The scheme of the single phase heating pump used for heating is shown in Fig. 2. In order to function the single phase heating pump needs a relatively high heat source with the temperature range from 20°C to 40°C. The weak side of the pump is the fact that we cannot maximally use the energy source and its advantage is relatively small investment.

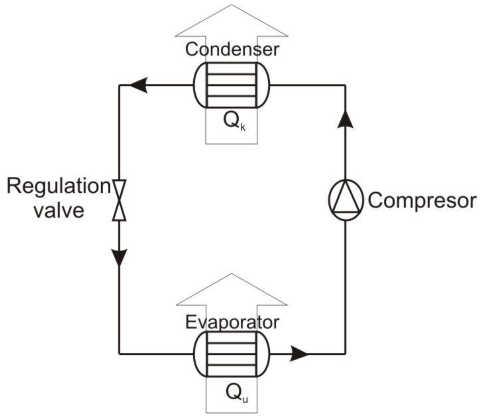

Fig 1: Single phase heating pump

B. Mathematics model

Developing a maths model for calculating the process characteristics of the pump and simulation of the refrigerating agent mixture is done on the base of known thermo physical data. For each agent, certain linear or quadratic equation for calculating the steam pressure, density and specific enthalpy is determined, based on the wanted functioning temperature.

The equation for calculating the steam pressure depending on temperature looks like this:

$$p = 10^{\left[A\cdot\left(\frac{1}{T}\right)^2 + B\cdot\left(\frac{1}{T}\right) + C\right]} \text{ (Pa)} \tag{1}$$

Where:

A, B, C are the constants for calculating the steam pressure depending on temperature

Equation for calculating the density of liquid ρ_l and density of steam ρ_g of the agent depending on temperature are:

$$\rho_l = \left[a_l \cdot T^2 + b_l \cdot T + c_l\right] \text{ (kg/m}^3\text{)} \tag{2}$$

$$\rho_g = \left[a_g \cdot T^2 + b_g \cdot T + c_g\right] \text{ (kg/m}^3\text{)} \tag{3}$$

where:

a_l, b_l, c_l - are constants for calculating the density of the liquid depending of temperature

a_g, b_g, c_g -are constants for calculating the density of the steam depending on temperature

Equation for calculating the specific enthalpy of the liquid h_l and specific steam enthalpy h_g of the agent depending on the temperature are:

$$h_l = \left[x_l \cdot T^2 + y_l \cdot T + z_l\right] \quad \text{(J/kg)} \tag{4}$$

$$h_g = \left[x_g \cdot T^2 + y_g \cdot T + z_g\right] \quad \text{(J/kg)} \tag{5}$$

Where:

x_l, y_l, z_l - are constants for calculating the specific enthalpy of the liquid depending on temperature

x_g, y_g, z_g - are constants for calculating the specific enthalpy of the steam depending on temperature

Based on the above mentioned equations, using the MathCAD programme, the programme for calculating the functioning characteristics of the heating pump was created. In the programme package GAMS, the programme for simulation of the determining of the optimal proportion between the two agents in the mixture under the chosen conditions, was created [7, 8, 9].

Heating flow of the single phase heating pump and refrigerating device Φ_{HN} are calculated from the following:

$$\Phi_{TČ} = q_{m,T} \cdot (h_{g,2} - h_{l,3}) \quad \text{(W)} \tag{6}$$

$$\Phi_{HN} = q_{m,S} \cdot (h_{g,1} - h_{l,4}) \quad \text{(W)} \tag{7}$$

Where:

$h_{g,1}$ - is a specific enthalpy of the steam at the entrance into the compressor (J/kg·K)

$h_{g,2}$ - is a specific enthalpy of the steam at the pressure side of the compressor (J/kg·K)

$h_{l,3}$ -is a specific enthalpy of the liquid agent in condensator (J/kg·K)

$h_{l,4}$ - is a specific enthalpy of the liquid agent that enters the stove (J/kg·K)

Heating number of the heating pump (TČ)

$$COP_G = \frac{\Phi_{TČ}}{P_K} \quad (\,/\,) \tag{8}$$

Refrigerating number of the refrigerating device

$$COP_H = \frac{\Phi_{HN}}{P_K} \quad (\,/\,) \tag{9}$$

Compressor characteristics have to be precisely calculated in order to correctly dimension the heating pump or the refrigerating device capacity. Having in mind the calculated data and standard dimensions of the compressor, we determine the capacity of the heating pump or the refrigerator device in theory [10, 11].

Refrigerating device temperature on the pressure side of the compressor

$$T_T = T_S \cdot r_K^{\left(\frac{\chi-1}{\chi}\right)} \quad \text{(K)} \tag{10}$$

The proportion of the compressor pressure is:

$$r_K = \frac{p_T}{p_S} \quad (\,/\,) \tag{11}$$

where:

T_S - Temperature of the agents at the entrance side of the compressor (K)

χ - Pressure factor of the agent (/)

p_T - Steam pressure of the agent at the pressure side of the compressor (Pa)

p_S - Agent steam pressure at the suction side of compressor (Pa)

The power needed for compressor functioning P_K at adiabatic agents compressed in the heating pump or refrigerating device is calculated by the following equation:

$$P_K = \frac{P_{ad}}{\eta_K} = \frac{\frac{\chi}{\chi-1} \cdot p_S \cdot V_K \cdot \left[r_K^{\left(\frac{\chi-1}{\chi}\right)} - 1 \right]}{\eta_K} \quad \text{(W)} \tag{12}$$

Where:

P_{ad} - is the adiabatic power of agent compression (W)

η_K - Usefulness of compressor (%)

Level of filling the compressor cylinder λ is determined this way:

$$\lambda = 1 - \varepsilon_o \cdot \left[r_K^{\left(\frac{1}{\chi}\right)} - 1 \right] \quad (\%) \tag{13}$$

The real cubic flow of compressor:

$$q_{V_K dej.} = q_{V_K} \cdot \lambda \quad (\mathrm{m^3/s}) \tag{14}$$

Volume flow of the refrigerating device in the compressor:

$$q_H = q_{V_K dej.} \cdot \frac{\rho_{g,S}}{\rho_{g,T}} \quad (\mathrm{m^3/s}) \tag{15}$$

Mass flow of the refrigerating agents at the suction side of the compressor :

$$q_{m,S} = q_{V_K dej.} \cdot \rho_{g,S} \quad \text{(kg/s)} \tag{16}$$

Mass flow of the refrigerating agents at the pressure side of the compressor:

$$q_{m,T} = q_H \cdot \rho_{g,T} \quad \text{(kg/s)} \tag{17}$$

Where:

ε_o - harmful area (%)

q_{V_K} - comrpessor volume flow (m^3/s)

$\rho_{g,S}$ - agent steam density at the suction side of the compressor (kg/m^3)

$\rho_{g,T}$ - agent steam density ustina at the pressure side of the compressor (kg/m^3)

C. Determining the optimal proportion in the two refrigerating agent mixture

Programme GAMS (*General Algebraic Modelling System*) is used for simulating of the determining the optimal proportion in the two refrigerating agent mixture. Simulating in the above mentioned programme determines the mixture of the two refrigerating agents, where the maximum efficiency for them can be achieved when the heating pump is functioning.

Optimal proportion of the mixture is determined by Z function, for the heating flow value of the heating pump:

$$\max Z = q_{m,S} \cdot (h_{g,1} - h_{l,4}) \cdot 1000 \quad \text{(kW)} \tag{18}$$

Optimal proportion of the two agents in the mixture is determined for the agent share (X_1) and agent share (X_2), which means that:

$$X_1 + X_2 = 1 \quad (\%) \tag{19}$$

When simulating, we have in mind the basic characteristics of the compressor (functioning power, maximum pressure and pressure proportion).

Basic characteristics of the computer programme, which enables optima mixture determination, is presented in Fig. 3.

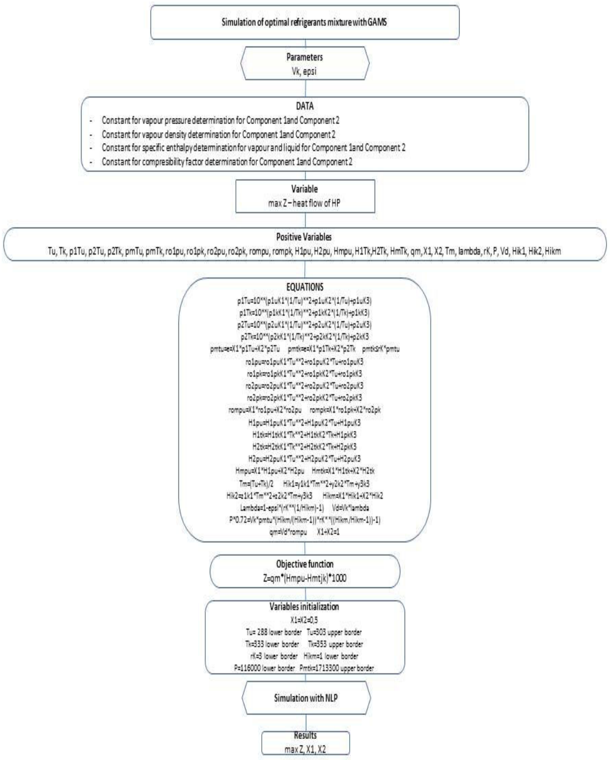

Fig. 3: Diagram for determining the optimal proportion of two agents in the mixture

D. Computer programme for determining the optimal proportion of two agents

```
Vk=0.17695;
epsi=0.14;
scalar
*constant for calculating the pressure when cooking or condensation COMPONENT 1 (R-600) iso-butane
p1uK1 /0/,              p1kK1 /0/,
p1uK2 /-1129.8/,        p1kK2 /-1112.6/,
p1uK3 /9.3334/,         p1kK3 /9.2784/,
*constant for calculating the pressure of cooking and condensation COMPONENT 2 (R-290) propane /0/,   p2kK1 /0/,
p2uK2 /-1002.3/,        p2kK2 /-964.18/,
p2uK3 /9.3344/,         p2kK3 /9.2103/,
*constant for calculating the steam pressure when cooking and condensation COMPONENT 1
ro1puK1 /0/,            ro1pkK1 /0/,
ro1puK2 /0.2583/,       ro1pkK2 /0.6005/,
ro1puK3 /-67.669/,      ro1pkK3 /-177.24/,
*constant for calculating the density of the steam when cooking and condensation COMPONENT 2
ro2puK1 /0/,            ro2pkK1 /0/,
ro2puK2 /0.5439/,       ro2pkK2 /1.9109/,
ro2puK3 /-141.0/,       ro2pkK3 /-585.88/,

*constant for calculating the enthalpy of the COMPONENT 1 (R-600a) iso-butane
H1puK1 /0/,             H1tkK1 /0/,
H1puK2 /1.3407/,        H1tkK2 /2.8062/,
H1puK3 /189.57/,        H1tkK3 /-584.34/,
*constant for calculating the enthalpy of COMPONENT 2 (R-290) propane
H2puK1 /0/,             H2tkK1 /0/,
H2puK2 /0.9718/,        H2tkK2 /3.6472/,
H2puK3 /310.77/,        H2tkK3 /-844.7/,
*constant for calculating pressuring factor
y1k1 /0/,               z1k1 /0/,
y2k2 /0.0012/,          z2k2/0.0052/,
y3k3 /0.7985/,          z3k3 /-0.2713/;
variables z;
*z max is heating flow of the heating pump W
positive variable  Tu    temperature of steaming ,
Tk    temperature of condensation,
p1Tu,
p1Tk,
p2Tu,
p2Tk,
pmTu,
pmTk,
ro1pu,
ro1pk,
ro2pu,
ro2pk,
rompu,
rompk,
H1pu,
H2pu,
Hmpu,
H1Tk,
H2Tk,
HmTk,
qm    mass flow at the suction side of the compressor kg/s X1,
X2,
Tm    medium temperature,
lamda    level of filling the compressor cylinder,
rK   pressure proportion ,
P     compressor power,
Vd   real compressor volume,
Hik1  pressure factor of component 1,
Hik2  pressure factor of component 2 ,
Hikm  pressure factor of the mixture
TT    heating flow;
equations
e1,e2,e3,e4,e5,e6,e7,
e8,
*e9,e11,e13
e10,e12,
e14,e15,e16,e17,e18,e19,e20,e21,e22,e23,e24,e25,e26,e27,e28,e88,obj;
*ontext
e1..   p1Tu=e=10**(p1uK1*(1/Tu)**2+p1uK2*(1/Tu)+p1uK3);
e2..   p1Tk=e=10**(p1kK1*(1/Tk)**2+p1kK2*(1/Tk)+p1kK3);
e3..   p2Tu=e=10**(p2uK1*(1/Tu)**2+p2uK2*(1/Tu)+p2uK3);
e4..   p2Tk=e=10**(p2kK1*(1/Tk)**2+p2kK2*(1/Tk)+p2kK3);
e5..   pmtu=e=X1*p1Tu+X2*p2Tu;
e6..   pmtk=e=X1*p1Tk+X2*p2Tk;
*pressure proportion
e7..   pmtk=g=rK*pmtu;
e8..   ro1pu=e=ro1puK1*Tu**2+ro1puK2*Tu+ro1puK3;
e10..  ro2pu=e=ro2puK1*Tu**2+ro2puK2*Tu+ro2puK3;
e12..  rompu=e=X1*ro1pu+X2*ro2pu;
e14..  H1pu=e=H1puK1*Tu**2+H1puK2*Tu+H1puK3;
e15..  H1tk=e=H1tkK1*Tk**2+H1tkK2*Tk+H1tkK3;
e16..  H2tk=e=H2tkK1*Tk**2+H2tkK2*Tk+H2tkK3;
e17..  H2pu=e=H2puK1*Tu**2+H2puK2*Tu+H2puK3;
e18..  Hmpu=e=X1*H1pu+X2*H2pu;
e19..  Hmtk=e=X1*H1tk+X2*H2tk;
e20..  Tm=E=(Tu+Tk)/2;
e21..  Hik1=E=y1k1*Tm**2+y2K2*Tm+y3K3 ;
e22..  Hik2=E=z1k1*Tm**2+z2K2*Tm+z3K3 ;
e23..  Hikm=E=X1*Hik1+X2*Hik2;
* level of filling the compressor cylinder
e24..  lamda=E=1-epsi*(rK**(1/hikm)-1);
*real cubic
e25..  Vd=E=Vk*lamda;
*compressor power
e26..  P*0.72=e=vK*pmtu*(hikm/(hikm-1))*rK**((hikm/(hikm-1))-1)
*mass flow
e27..  qm=E=Vd*rompu;
*heating flow
e28..  TT=e= qm*(Hmpu-Hmtk)*1000;
e88..  X1+X2=E=1;
obj..  z=e=TT;
X1.l=0.4;
X2.l=0.7;
Tu.l=283;
Tu.lo=293;
Tu.up=293;
Tk.up=343;
Tk.lo=343;
```

```
rK.up=3;
Hikm.l=1.2;
Hikm.lo=1.0001;
X1.fx=0.4;
P.up=100000;
*pmtk.up=1713300;
option decimals=6;
model topcrp /all/;
solve topcrp using NLP maximizing z;
Display "R E Z U L T A T"
display z.l,X1.l,X2.l,pmtk.l,pmtu.l,rk.l;
```

E. Simulation results

The main condition for mixing the refrigerating agents to be used in the heating pump is their capability to mix them physically without chemical reaction. That is confirmed for R-600a and R-290.

The aim of the optimal proportion simulation of the mixture is to determine the maximum capacity usage of the compressor and the maximum usage of the high temperature heating pump heating flow.

The limiting conditions for the efficacious simulation are determined:

- volume compressor flow is constant and equals q_{Vk} = 637 m^3/h,
- upper limit of the compressor power type WBH is P_K = 116 kW, which is 80% of the maximum power of the compressor and equals 145 kW,
- upper limit of the allowed pressure of the agent mixture is 1,7133 MPa, which is 85,6 % of the maximum allowed pressure of the refrigerating agent which is 2,0 MPa,
- the proportion between the pressures is higher than 3
- the temperature of steaming t_U = from 15 to 35°C,
- The temperature of condensation t_K = from 60 to 80°C.

The results of the simulation are presented as a graphic in Fig. 4, where we can see the heating flow, the compressor power and single component shares. Fig. 5.rigerating number values for the single phase heating pump.

The simulation shows that the power for the three functioning conditions is equal P_k= 116,0 kW . The heating flow of the steaming part of the single phase high temperature heating pump is lower if the condensation temperature t_K is higher. The diagram also shows the share of both agents X_1 and X_2, at different condensation temperatures.

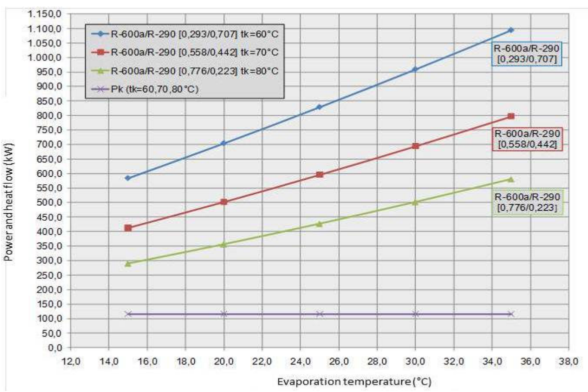

Fig. 4: Single phase heating pump with the optimal proportion of R-600a and R-290 in the mixture.

The results of calculating functioning characteristics of the heating pump using R-290 agent show that the needed power of the compressor is too high. The heating flow of the heating pump with R-290 agent is high. If we compare the values of the heating flow and compressor energy consumption with R-600a agent, we are reserved when we talk about the usage of the compressor power having in mind the above mentioned things.

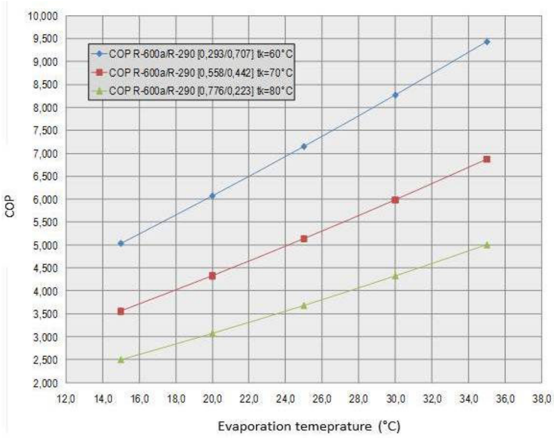

Fig. 5: Refrigerating number of the single phase heating pump with the optimal mixture proportion of R-600a and R-290

R-600a andR-290 mixture achieves the maximum, having in mind the maximum achieved by the compressor and the heating pump heating flow. There are some restrictions with some agents, and because of the thermo limitations the process is not possible to be taken out at all. Fig. 6 shows the dependence of the heating pump heating flow on the component share 1 (R -600a) in R-600a and R-290 agent mixture, at the constant temperature of steaming t_U = 20°C and at different condensation temperatures. The compressor power in each form is the same, P_K = 116,0 kW , the maximum pressure of condensation equals p_{mTk} = 1,7133 MPa.

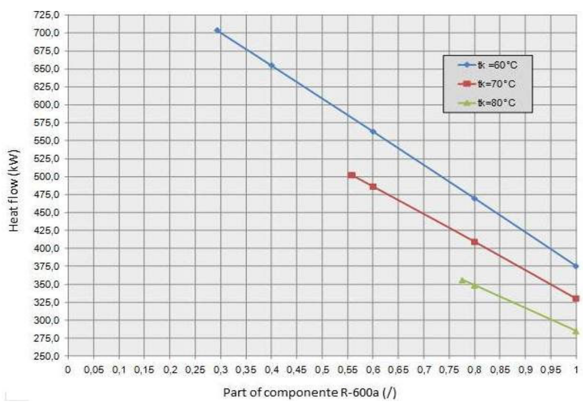

Fig. 6: Heating flow of the cooking part of the heating pump depending on the share

The results of calculating heating flow depending on agent (R-600) share in two-agent mixture (R-600a/R-290) show that the share of (R-600a) increases if the temperature is getting higher. The simulation proves that functioning of the heating pump with R-290 is impossible because of the high pressures and great compressor power consumption, despite the great values of the heating flow. The optimal usage of the high temperature heating pump and the compressor is achieved when the mixture is R-600a/R-290 (0.558/0.442), at condensation temperature $t_K = 70°C$.

IV. Calculating the Characteristic Features of the Heating Pump

Calculating the characteristic features of the heating pump used for heating is processed in MathCAD Professional. Having in mind the thermo physical features, the simulation of determining the optimal mixture proportion is presented. The Programme is based on the mathematic model, which enables calculating the characteristic features of the pump and compressor, for various conditions. Due to the choice of refrigerating agents, the results are given as graphics.

The aim of the research is the optimal choice for refrigerating agents, based on the fact that the heating pump would be used for central heating. The optimal type of the compressor is chosen having in mind the given results and the adequate catalogue.

The maximum compressor functioning at 1000 rotations per minute was taken in consideration when calculating. The volume flow was processed and chosen on the basis of the chosen compressor. Based on these results, the refrigerating agent was also chosen. And it must be in accordance with the compressor characteristics.

Calculation for the single phase high temperature pump is done for four different refrigerating agents. R-134a, R-245fa, R-290 and R-600aat steaming temperatures (t_U = 15, 20, 25, 30 and 35°C) and at condensation temperatures a (t_K = 60°C, 70°C and 80°C). The calculated heating flow values and compressor energy consumption are presented in Fig. 7, 8, 9. Piston compressor with cubic flow 637 m^3/h was used in calculating. The diagram shows that the highest flow is with R-290 and R-134a agents, but they both use great quantity of energy for compressor functioning. R-245fa has low values of heating flow and low consumption of energy by the compressor. The best results are guaranteed by R-600a.

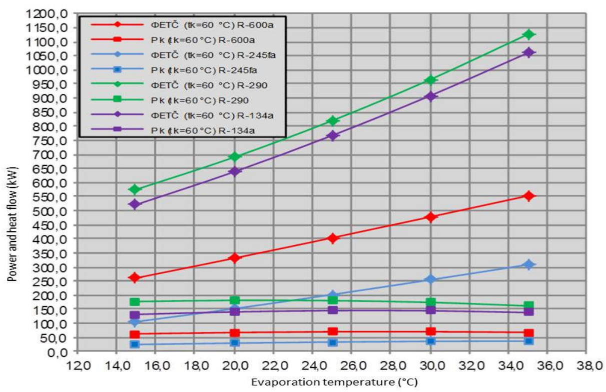

Fig. 7: Single phase heating pump at condensation temperature t_K=60°C

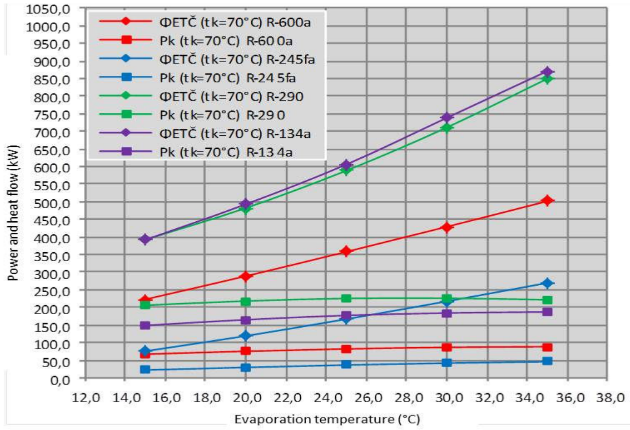

Fig. 8: Single phase heating pump at condensation temperature t_K=70°C

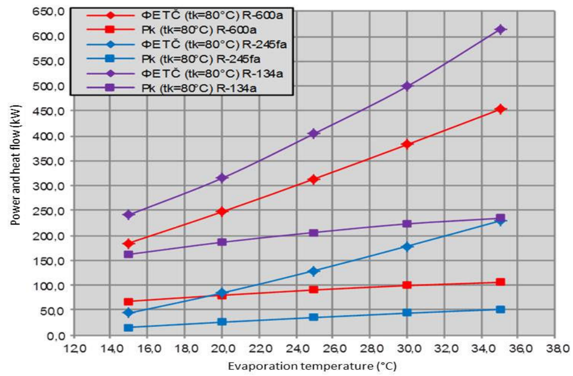

Fig. 9: Single phase heating pump at condensation temperature t_K=80 °C

The heating pump number value is shown in Fig. 9 and it is calculated by the 8 equation. The calculation and the results show that the biggest heating number is for R-245fa and R-600a. R-134 a and R – 290 achieve lower heating numbers. The pressure proportion is the proportion between the pressure pressure and the pressure at the suction side of the compressor, and it is dependant on temperature. The pressure proportion is shown in Fig. 10 an it is calculated by 10 equation. Considering the steaming temperature, it is obvious that lower proportions are achieved with R-290, higher with R-245fa. The proportions of R-600a and R-134 are very similar.

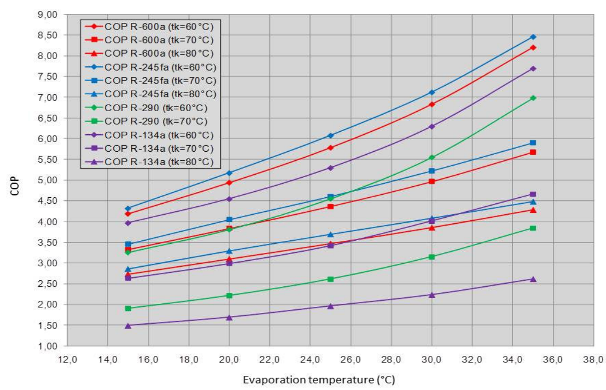

Fig. 10: Heating number of the single phase heating pump

With refrigerating agentR-600 at steaming temperature t_U = 25°C and condensation temperature t_K = 70 °C , the heating flow is $\Phi_{TČ}$ = 357,9 kW and the power used for compressor functioning is P_K = 82,0 kW. Heating number is COP= 4,365 and the pressure proportion is r_K = 3,105.

Based on the calculation, we can see that refrigerating agent R-600a is adequate for MYCOM WBH compressor. Other agents do not supply the optimal usage.

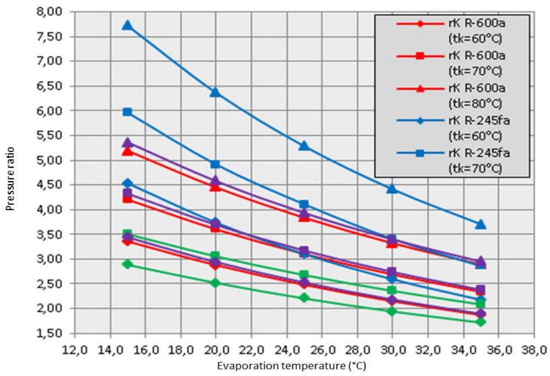

Fig. 11: Pressure proportion of the single phase heating pump

A. *Calculation programme for single phase heating*

DATA FOR CALCULATION OF THE THERMO DYNAMIC DATA FOR REFRIGERATING AGENTS

AGENT: R-600a (izo-butan)

CONSTANTS FOR CALCULATION OF THE PRESSUR AT Tu AND Tk TEMPERATURES

$au := 0$

$AU := 0$

$bu := -1129.8$

$BU := -1112.6$

$cu := 9.3334$

$CU := 9.2784$

CONSTANTS FOR CALCULATING LIQUID DENSITY OF THE AGENT

$at := -0.0026$

$AT := -0.0065$

$bt := 0.2975$

$BT := 2.7801$

$ct := 693.83$

$CT := 292.66$

CONTSTANTS FOR CALCULATING STEAM DENSITY OF THE AGENT

$ap := 0$

$AP := 0$

$bp := 0.2583$

$BP := 0.6005$

$cp := -67.669$

$CP := -177.24$

CONSTANTS FOR CALCULATING LIQUID ENTHALPY OF THE AGENT

$xt := 2.463$

$XT := 2.8062$

$yt := -474.45$

$YT := -584.34$

CONSTANTS FOR CALCULATING STEAM ENTHAPY OF THE AGENT

$xp := 0$

$XP := 0$

$yp := 1.3407$

$YP := 1.2157$

$zp := 189.57$

$ZP := 230.14$

COMPRESSOR TYPE: 8RWH

COMPRESSOR ROTATION PER MINUTE [1/min]:

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

COMPRESSOR TOTATION [%]:

$\mu := 1$

COMPRESSOR USAGE [/]:

$\eta := 0.72$

AGENT STEAMING TEMPERATURE [K]:

$Tu := 308.15\ K$

AGENT CONDENSATION TEMPERATURE [K]:

$Tk := 333.15\ K$

MEDIUM TEMPERATURE [K]:

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

$Ts = 320.65\ K$

PRESSURING FACTOR:

$\chi := 1.174$

UNUSED PART OF THE FILLING TANK OF THE COMPRESSOR $\varepsilon := 0.14$

VOLUME FLOW OF THE PISTON COMPRESSOR [m3/s]:

$$Vk := 0.17695\ \frac{m^3}{s}$$

PRESSURE ON THE SUCTION SIDE OF THE COMPRESSOR [MPa]:

$Pu = 464519.3\ Pa$

SPECIFIC AGENT STEAM ENTHALPY AT STEAMING TEMPERATURE

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

$$Hpu = 602.707\ \frac{1}{kg} \cdot J$$

SPECIFIC AGENT LIQUID ENTHALPY AT STEAMING TEMPERATURE

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

$$Htu = 280.391\ \frac{1}{kg} \cdot J$$

AGENT STEAM DENSITY AT THE STEAMING TEMPERATURE

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

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

$$\rho pu = 11.926\ \frac{kg}{m^3}$$

AGENT LIQUID DENSITY AT THE STEAMING TEMPERATURE

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

$$\sigma tu = 538.618\ \frac{kg}{m^3}$$

PRESSURE ON THE PRESSURE SIDE OF THE COMPRESSOR [MPa]:

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

$Pk = 868486.8\ Pa$

SPECIFIC AGENT STEAM ENTHALPY AT CONDENSATION TEMPERATURE

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

$$Hpk = 635.2\ \frac{1}{kg} \cdot J$$

SPECIFIC AGENT LIQUID ENTHALPY AT CONDENSATION TEMPERATURE

$$Htk = 350.546\ \frac{1}{kg}\cdot J$$

AGENT STEAM DENSITY AT CONDENSATION TEMPERATURE

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

$$\rho pk = 22.817\ \frac{kg}{m^3}$$

AGENT LIQUID DENSITY AT CONDENSATION TEMPERATURE

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

$$\sigma tk = 497.422\ \frac{kg}{m^3}$$

PRESSURE PROPORTION OF THE COMPRESSOR

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

$$rK = 1.87$$

AGENT STEAM ENTHALPY AT CONDENSATION TEMPERATURE

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

$$Hp2 = 641.2\ \frac{1}{kg}\cdot J$$

LEVEL OF FILLING THE COMPRESSOR CYLINDER

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

$$\lambda = 0.901$$

THE REAL CUBIC FLOW OF THE COMPRESSOR

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

$$vK = 0.16\ \frac{m^3}{s}$$

AGENT CUBIC FLOW IN THE COMPRESSOR

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

$$qv = 0.083\ \frac{m^3}{s}$$

POWER OF HT ADIABATIC PRESSURE

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

$$Pad = 48582.83\,W$$

COMPRESSOR POWER

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

$$P = 67476.16\quad W$$

VOLUME FLOW AT THE SUCTION SIDE OF THE COMRESSOR

$$vK = 0.16\ \frac{m^3}{s}$$

MASS FLOW AT THE SUCTION SIDE OF THE COMPRESSOR

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

$$qms = 1.902\ \frac{kg}{s}$$

VOLUME FLOW AT THE PRESSURE SIDE OF THE COMPRESSOR

$$qv = 0.083\ \frac{m^3}{s}$$

MASS FLOW AT THE PRESSURE SIDE OF THE COMPRESSOR

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

$$qmt = 1.902\ \frac{kg}{s}$$

HEATING FLOW OF THE STEAMING PART OD THE HEATING PUMP [W]:

$$\Phi k := qms\cdot(Hpu - Htk)\cdot 1000$$

$$\Phi k = 479692\quad W$$

HEATING FLOW OF THE CONDESATION PART OF THE HEATING PUMP [W]

$$\Phi ETC := qms\cdot(Hp2 - Htk)\cdot 1000$$

$$\Phi ETC = 552848.8\quad W$$

HEATING NUMBER OF THE HEATING PUMP [/]:
TEMPERATURE AT THE COMPRESSOR EXIT [K]:

$$COPG = 8.193$$

V. CONCLUSION

The way of choosing the most adequate refrigerating agent for adiabatic circular process functioning for various usages is presented in the text. The choice is based on reducing negative effects on the environment and on thermo dynamic characteristics. Compressor characteristics also have to be considered. The article includes the detailed presentation of the optimal choice of the agents for heating pump functioning, which is used for central heating.

Simulation model for determining adequate agent R-600a and R-290 mixture for the most efficacious functioning of the pump is presented. With maximum heating flow and considering the determined limiting conditions, it is proven that the optimal mixture of the two agents at crystallization temperature t_K = 70°C equals R-600a/R-290 = 0,558/0,442. Process functioning conditions limit the way of using the mixture.

For heating households, three different pump types are suggested:

- Single phased high temperature heating pump
- Double phase high temperature heating pump
- Double phase high temperature heating pump with heat exchanger.

Each pump type has a programme written in MathCAD Professional, which enables calculating characteristic features of the heating pump and compressor. The calculating is done at different temperatures of steaming and condensation. The results are presented as graphics, which enables easier reading and choosing the adequate compressor.

The main aim of the simulation is determining the maximum heating flow and determining the usage of compressor. When doing this, we have to consider all the limitations that can appear when functioning. The simulation of determining of the optimal mixture is done by GAMS programme.

According to these data, using R-600a agent and MYCOM, type WBH compressor is recommended when using single phase heating pump. The heating pump functioning is reasonable at steaming temperature t_U = 25°C and condensation temperature t_K = 70°C. Heating flow of the single phase heating pump is, in such a case, 357,9 kW, and compressor energy consumption 82,0 kW.

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