IOP Conf. Series: Materials Science and Engineering 463 (2018) 022012 doi:10.1088/1757-899X/463/2/022012

A Static Mathematical Model for Cascades of Heatpump Installation of Multipurpose Thermal Point

V I Parshukov[1], V V Papin[2], R V Bezuglov[3], A D Ovechkin[4]

[1]Director, LLC SPE "Donskie technologii", Michailovskaya str., 164A, Novocherkassk 346400, Russia
[2]Researcher, Research department, DSTU, Gagarin Square, 1, Rostov-on-Don 344010, Russia
[3]Researcher, Research department, DSTU, Gagarin Square, 1, Rostov-on-Don 344010, Russia Country
[4]Undergraduate, Department of thermal power plants and heating, Platov SRSTU, 132, Novocherkassk 346400, Russia

E-mail: romanbezuglov@inbox.ru

Abstract. In the article the static mathematical model of work of heatpumping part of multifunction thermal point on the basis of the cascade thermal pump is described. Thermal point provides the consumer with heat energy for heating and hot water supply, refrigerating energy for conditioning. Also the complex can provide the consumer with both types of energy (seasonally) for ensuring ventilation of the building. The main advantages of thermal point is high coefficient of performance in all range of work of installation, and also in high utilization coefficient of primary fuel. The coefficient of performance remains high due to optimization of heatpumping cycle - in selection of certain temperatures for each mode. Increase in utilization coefficient of primary fuel is reached by utilization of waste warmth from sewer drains, exhaust air of the ventilation unit, and also useful use of heat and refrigerating energy as heating/conditioning implementation coproducts.

1. Introduction

The mathematical model is created taking into account target orientation of research problem (the analysis of the processes in multifunction thermal point), the required accuracy and previously known information [1-4]. At mathematical modeling analytical and statistical models are applied . Analytical models are harsher, consider smaller number of factors, always demand some assumptions and simplifications. However results of calculation for them are more indicative, clearly reflect the main patterns inherent in the phenomenon and, the main thing are more adapted for search of optimal solutions [5]. For the best understanding of the processes happening during the work of multifunction thermal point it is necessary to construct and solve mathematical model of all installation contours.

2. Relevance of mathematical model the cascade heat pump

Now in Russia there are practically no complex researches on creation of domestic systems of power supply on the basis of heat pumps [6-11]. At the same time features of climatic zones are practically not considered, and it is very important for operating conditions of installation. It is impossible to create product which equally well works in the conditions of the southern regions and Far North [12-16]. Thus, relevance of development the cascade heatpumping installation, in particular development of mathematical model of the processes in it is proved [17-20].

IOP Conf. Series: Materials Science and Engineering 463 (2018) 022012 doi:10.1088/1757-899X/463/2/022012

3. Scientific importance of development the mathematical model of the cascade thermal pump

The scientific importance of the developed designs consists in essentially other approach to development of the thermal scheme of multifunction thermal point. This scheme provides overflows of heat energy from sources to receivers in and out the consumer at the expense of step structure of vapor-compression heatpumping installations. This scheme provides simultaneous production of necessary amount of heat energy with different temperature and power for heating, hot water. Also the scheme allows to receive cold for ventilation system and conditioning up to food freezing. At the same time there is energy overflow in the consumer. When conditioning rooms, the warmth which is taken away from them goes for hot water supply, and when heating warmth undertakes not only from low-potential source, but also from technology drains of the consumer: ventilating channels, left-luggage office of products and other sources.

The scheme of utilization of warmth and cold will be implemented by different cascades of the thermal pump in one general multi-level scheme. Thus, at implementation of the offered thermal scheme of multifunction thermal point with separate heat supply of the consumer, both on temperature and on power.

4. Theoretical developments

As boundary conditions of mathematical model the values provided in the table 1 were chosen.

Table 1. Boundary conditions of a mathematical model.

№	Parameter	Designation and unit of measure	Lower boundary coditions	Upper boundary coditions
1	Thermal power	Q, kVt	0,5	50
2	Temperature of the low-potential heat carrier (brine, water) at the entrance	t_{n1}, °C	plus 10	plus 15
3	Temperature of the low-potential heat carrier (brine, water) at the exit	t_{n2}, °C	plus 10	plus 10
4	Temperature of the high-potential heat carrier (hot water) at the entrance	t_{v1}, °C	plus 20	plus 50
5	Temperature of the high-potential heat carrier (hot water) at the exit	t_{v2}, °C	plus 40	plus 70
6	Temperature indoors	t_0, °C	18	28
7	Temperature drop at the exit from the evaporator heat exchanger	Δt_{ev}, °C	2	8
8	Temperature drop at the exit from the condenser heat exchanger	Δt_c, °C	2	8
9	Atmospheric pressure	mm of mercury	641	816
10	Ambient temperature	t_{em}, °C	0	45

Mathematical modeling of work of multifunction thermal point was carried out in the Mathcad V.15. The description of calculation the thermal point cascade perfromed in mathematical model is included below. The calculation procedure is identical to all cascades.

Originally key points on the p, h-chart for creation the thermodynamic cycle scheme are defined. Freon evaporation temperature, is defined as difference of temperature of low potential heat carrier (brine, water) at the exit and difference of temperatures in the evaporator:

$$t_{ev} = t_{n2} + \Delta t_{ev}.$$

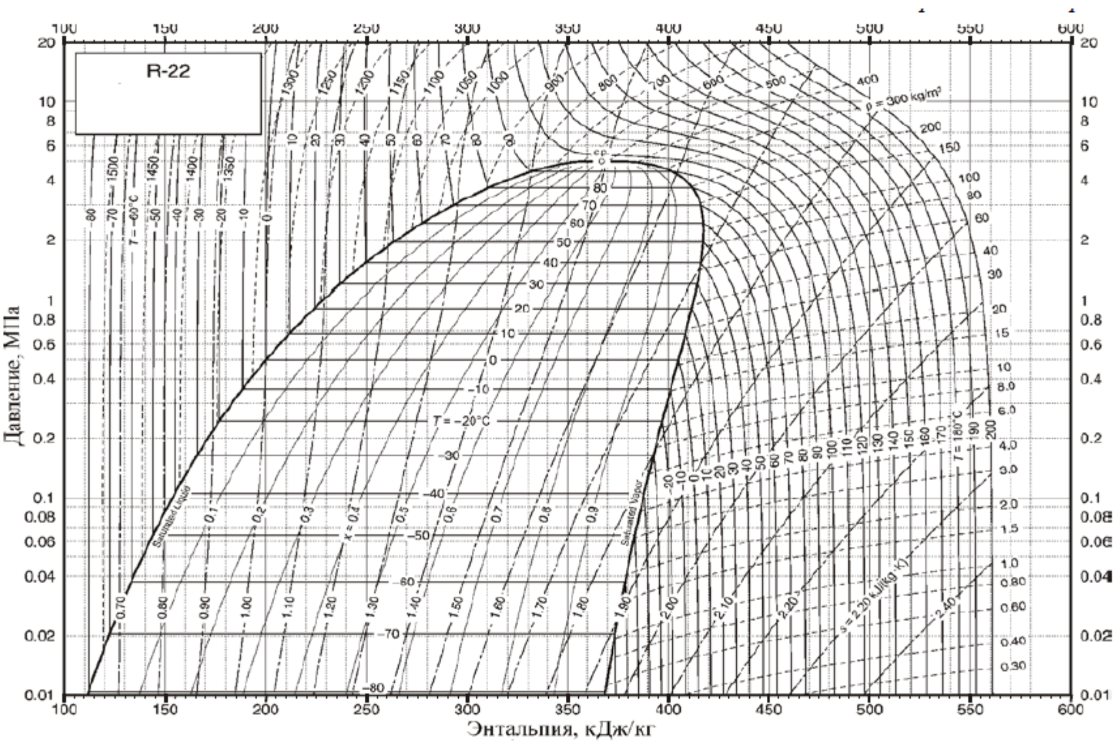
Parameters in a point 1 are determined by evaporation temperature t_k according to tables of thermodynamic properties of R22 coolant in a condition of saturation and on p,h-chart 1 – enthalpy on the right boundary curve h_{k1} and pressure p_{k1}. The point 1 is noted on *p,h*- chart (drawing 1).

Temperature of condensation of Freon is defined as the sum of temperature of highly potential heat carrier (hot water) at the exit t_{v2} and a difference of temperatures in the condenser:

$$t_к = t_{v2} + \Delta t_к.$$

In a point the 3rd enthalpy h_3 and pressure p_3 are determined by condensation temperature $t_к$ ccording to tables of thermodynamic properties or on p,h- chart. Then on p,h-chart the point 3 is noted.

On p,h-chart on crossing of the line of constant entropy of S_1, passing through a point 1, and the line of an isobar p_k, passing through a point 3 the point 2a is defined. Then the enthalpy in this point is determined by the chart h_{2a}.



Drawing 1. Chart of thermodynamic properties Freon R-22.

The adiabatic efficiency of the compressor is the ratio of the specific internal compression work the ideal adiabatic cycle to the specific compression work of the actual cycle:

$$\eta_a = k\frac{l_{comp./a}}{l_{comp.c}};$$

$$l_{сж.а} = c_{pf} \cdot (273 + t_{norm});$$

$$l_{сж.ц} = c_{pf} \cdot (273 + t_c);$$

$$\eta_a = k \cdot \frac{273 + t_{norm}}{273 + t_c}.$$

The enthalpy of Freon after compression with allowance for losses is defined as the sum of the enthalpy:

$$h_2 = h_1 + \frac{h_{2a} - h_1}{\eta_a}.$$

By the value of the enthalpy h_2 and pressure p_3 the diagram marks point 2 and determines the

IOP Conf. Series: Materials Science and Engineering 463 (2018) 022012 doi:10.1088/1757-899X/463/2/022012

temperature at this point t_2.

By the value of the enthalpy $h_3 = h_4$ and pressure p_k the diagram marks the point 4.

Specific heat loads in the nodes of the heat pump are defined as the difference in enthalpy at critical points:

$$q_c = h_1 - h_4;$$
$$q_{ev} = h_2 - h_3;$$
$$l_{comp} = h_2 - h_1.$$

Correctness of calculation is defined by check of thermal balance:

$$q_c = q_{ev} + l_{comp}.$$

Thermal loading of the thermal pump is equal to thermal loading of the condenser:

$$q_p = q_c.$$

The energy consumed by the electric motor W is the ratio of the energy expended on compression, referred to the efficiency of the electric motor:

$$W = \frac{l_{comp}}{\eta_{em} \cdot \eta_{el}},$$

η_{em} – the electromechanical efficiency of the compressor is 0.9 for reciprocating compressors and 0.95 for scroll compressors;

η_{el} – The efficiency of the electric motor during the cyclic work is 0,8.

Indicators of power efficiency of the heat pump:

- the heat coefficient of performance is the ratio of the heat received relative to the energy expended on its compression:

$$\mu = \frac{q_c}{l_{comp}}.$$

- the electric coefficient of performance is the heat coefficient of performance taking into account the efficiency of the compressor and electric motor design:

$$\mu_{el} = \eta_{em} \cdot \eta_{el} \cdot \mu.$$

- specific consumption of primary energy – this is the ratio of the energy of the fuel used to generate heat to the amount of heat produced by the heat pump:

$$PE = \frac{1}{\eta_{em} \cdot \eta_{el} \cdot \eta_{pp} \cdot \eta_{ps} \cdot \mu},$$

η_{pp}– efficiency of power plant, 0,4;

η_{ps}– efficiency of power supply systems, 0,95.

As value PE more than 1, heating with use of the thermal pump is more favorable, than at combustion of the natural fuel applied to electricity generation.

Extent of pressure increase in the compressor is the ratio of the pressure in the condenser and the evaporator:

$$\varepsilon = \frac{p_3}{p_1}.$$

We will apply a method of exergic balances to thermodynamic assessment of efficiency of a heatpump cycle. The exergy is the maximum work which can be made upon reversible transition of any thermodynamic system from a state with the set parameters in equilibrium state with the environment.

Exergic calculation of the scheme is made:

- average logarithmic temperature of the cold heat carrier:

$$T_{av.n} = \frac{(t_{n1} - t_{n2})}{\ln\left(\frac{t_{n1} + 273}{t_{n2} + 273}\right)};$$

- exergic temperature of the low-potential heat carrier:

$$\tau_n = \frac{T_{av.n} - (t_0 + 273)}{T_{av.n}};$$

- the exergy given by the low-potential heat carrier in the evaporator:

$$e_n = \tau_n \cdot q_{ev};$$

- average logarithmic temperature of the hot heat carrier:

$$T_{av.v} = \frac{(t_{v1} - t_{v2})}{\ln\left(\frac{t_{v1} + 273}{t_{v2} + 273}\right)};$$

- exergic temperature of the high-potential heat carrier:

$$\tau_v = \frac{T_{av.v} - (t_0 + 273)}{T_{av.v}};$$

- the exergy received by the high-potential heat carrier in the condenser:

$$e_{vk} = \tau_v \cdot q_{cond};$$

- exergy of the electric power consumed by the electric motor:

$$e_{el} = \frac{l_{comp}}{\eta_{em} \cdot \eta_{el}};$$

- exergy heat pump efficiency:

$$e_{out} = e_{vk};$$

$$e_{in} = |e_n| \cdot e_{el};$$

$$\eta_{el} = \frac{e_{out}}{e_{in}}.$$

For definition of an expense the heat carrier and losses we make the subsequent calculation of a contour:

$$G_{car} = \frac{Q_p}{q_p}.$$

Full load of heat pump units:

- in the compressor:

$$N = W \cdot G_{car} ;$$

- in the evaporator:

$$Q_{ev} = q_{ev} \cdot G_{car};$$

- in the condenser:

$$Q_{cond} = q_{cond} \cdot G_{car} m;$$

Specific exergic losses in the compressor:

- external exergic losses in the compressor and electric motor, caused by mechanical friction:

$$\Delta e_{km}^{ex} = \left(W - l_{comp}\right);$$

- internal exergic losses in the compressor, caused by the irreversibility of the process the refrigerant compression of (the entropy S_1,S_2, S_3, S_4, is determined by the p,h-chart):

$$\Delta e_{km}^{in} = T_0 \cdot (S_2 - S_1).$$

Exergetic losses in heat exchangers are determined by the difference in the exergy of the refrigerant, according to the formula $\Delta h - T_0 \Delta S$, and exergy, fed or taken away from the coolant, equal τ_q. Thus, having determined entropy by tables of properties of freon in a condition of saturation or by the p,h-chart, we receive:

- exergy losses in the evaporator:

$$\Delta e_{ev} = e_n - [q_{ev} - T_0 \cdot (S_1 - S_4)];$$

- exergic losses in the condenser:

$$\Delta e_{cond} = [q_{cond} - T_0 \cdot (S_2 - S_3)] - e_{vk}.$$

The enthalpy of Freon during throttling doesn’t change, so the exergy losses in the throttle are equal:

$$\Delta e_d = T_0 \cdot (S_4 - S_3).$$

The sum of exergy losses in the heat pump:

$$\sum \Delta e = \Delta e_{km}^{ex} + \Delta e_{km}^{in} + \Delta e_{ev} + \Delta e_{cond} + \Delta e_d.$$

Checking the calculation is made by the equality of the resulting exergy losses and the difference in exergy at the input and output of the heat pump:

$$\sum \Delta e = (e_n + e_{el}) - e_v.$$

5. Conclusion

The given mathematical modeling allows to receive output characteristics of multifunction thermal point at input of the initial parameters meeting boundary conditions of mathematical model. It will help to make the decision on power of the installed system for the specific consumer in certain environmental conditions.

Acknowledgments

The article was published with financial support by the Ministry of Education and Science of the Russian Federation within the framework of the Federal Target Program "Research and development in the priority directions of the scientific-technological complex of Russia for 2014-2020" (№14.577.21.0228 Agreement on "Development and experimental approbation of technical solutions for the creation of a multifunctional heat point based on a cascade heat pump that provides for the joint generation of thermal energy and cold for heating, hot water, air conditioning and ventilation by transforming low-potential energy and domestic heat". The unique identifier of the applied scientific research and experimental developments (of the project) is RFMEFI57716X0228).

6. References

[1] Iovanov M 2011 The matematical model of clarifying processes for industrial wastewaters *Fiabilitate şi Durabilitate* Vol. 2 (8) pp. 85-90.
[2] D Malindžák, J Šaderová, D Vitko, M Malindžáková, A Gazda 2015 The methodology and model for in-process inventories calculation in the conditions of metallurgy production *Metalurgija* Vol. 54(1) pp. 227-230.
[3] Editörden 2004 Matematical Analysis of Classic Cascade Cooling Cycle Using in Liquefaction of Natural Gas *Uludağ University Journal of The Faculty of Engineering* vol. 9 (1).
[4] Adhiguna R T, Sutrisno, Sugiyono, Ridwan T 2018 Model Matematika Pengisian Gabah dari Malai Primer dan Sekunder *J. Dytnersky.Moscow*, Chemistry pp. 23-30.
[5] Papin V V, Efimov N N, Bezuglov R V 2016 Micro Energy Complex Based on Wet-Steam Turbine *Procedia Engineering* Vol. 150 pp. 324-329.
[6] A V Zolin, V V Chugunkov 2015 Modeling the Thermal Rocket Fuel Preparation Processes in the Launch Complex Fueling System *Aèrokosmičeskij Naučnyj Žurnal* vol. 1 pp. 27-38.
[7] Oelert H H, Hemmer E A 1970 *Neue Ergebnisse der Ermittlung der Structur von Steinkohlen durch spektroskpische Metoden*. «Erdöl und Kohle, Erdgas Petrochem» Bd. 23 pp. 87-91.
[8] Solovyova E G, Kondratenkov A N 2013 Independent power supply system of a building in the second climate zone *Vestnik MGSU* vol. 10 pp. 208-215.
[9] Trusov Artur, Shilkina Svetlana 2016 The effectiveness of the implementation and automation of heat pump on public buildings *MATEC Web of Conferences* p. 192.
[10] Vasilyev G P, Peskov N V, Gornov V F, Kolesova M V , Yurchenkol I A 2015 The method for Determining the Effectiveness of Low-Grade Geothermal Heat Usage under the Conditions of the Russian Climate *MATEC Web of Conferences* vol.12 pp.12-34.
[11] Vasilyev G P, Peskov N V, Gornov V F, Kolesova M V, Yurchenko I A 2015 Zoning of the Territory of Russia According to the Efficiency of the Use of Low-Grade Ground Heat for Heating *MATEC Web of Conferences* vol.8 pp. 56-76.
[12] Luis A Correa, Mohammad Mehboudi 2016 Testing a Quantum Heat Pump with a Two-Level Spin *Entropy* Vol. 18 (4) p.132.
[13] Sit B M 2010 Two-phase ejector of carbon dioxide heat pump calculus *Problems of the Regional Energetics* vol. 3(14) pp.62-70.

[14] Nihar R Sahoo, Uma S Pal, Dipika Sahoo, Lalit M. Bal 2015 Impact of Pretreatment and Drying Methods on Quality Attributes of Onion Shreds *Food Technology and Biotechnology* vol. 53 (1) pp. 57-65.
[15] A I Akaev, A I Ninalalov 2015 Method of calculating the optimal heat emission geothermal wells *Vestnik Dagestanskogo Gosudarstvennogo Tehničeskogo Universiteta: Tehničeskie Nauki* vol. 36(1) pp. 53-62.
[16] Francisco Javier Fernández, María Belén Folgueras, Inés Suárez 2017 Study and Optimization of Design Parameters in Water Loop Heat Pump Systems for Office Buildings in the Iberian Peninsula *Energies* vol. 10(121).
[17] Wang Fang, Fan Xiao-Wei, Chen Jie, Lian Zhi-Wei 2014 Energy and exergy analysis of heat pump using R744/R32 refrigerant mixture *Thermal Science* vol. 18(5) pp. 1649-1654.
[18] Papin V V, Efimov N N, Bezuglov R V 2016 *The determination the rotor surfacing time of the vertical microturbine with the axial gas-dynamic bearings*. Vol. 150 pp. 294-299.
[19] Y L Lee, S J Park, I S Hwang 2014 Study on Energy Efficiency Improvement of a Heat Pump Dryer during the Early Stage of Drying Frozen Hot Peppers *Advances in Mechanical Engineering*.
[20] Cui Bo-wen 2016 A New Hybrid Bathroom System Based on Energy Saving Concept *MATEC Web of Conferences* vol. 12 pp.12-18.