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Efficiency of Transformation of the Low-Temperature Geothermal Heat Carriers

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ABSTRACT

The main application aspects of thermal heat carriers are as follows: production of electricity, heat and cold supply. In the current report we consider two aspects: efficiency of cogeneration of electric and heat energy by the binary low-temperature setups (t = 80ºC) and improvement of efficiency of heat pump systems for multistage heating of delivery water. Results of preproject works on the low-temperature binary power plant of the 200 kW capacity with low-boiling R134a within a thermal power circuit are presented. A water heat carrier with the temperature of 80ºC is used as the heating medium. The main parameters of actuating medium R134a at the main points of the Rankine cycle, gas-dynamic characteristics of axial turbines on R134a, flow-rate and heat-transfer characteristics of power setup and performance characteristics of heat energy transformation are shown. The thermodynamic problem on energy efficiency of multistage water heating in heat supply systems and the problem of determination of performance utility of the considered technology on multistage compression of vapors of the low-boiling actuating medium were solved together at energy analysis of the reverse thermodynamic cycle of multistage compression. Transition to multiple compression always increases the coefficient of energy transformation in a vapor-compression heat pump. However, a growth of energy efficiency depending on the number of compression stages decreases. The technical-economical reasonability of the multistage process acts as a counter factor of energy efficiency. The technical-economical analysis included: determination of additional investments into the system of multistage compression; calculation of operation costs; determination of technical-economical characteristics for every closed heat power circuits and intermediate compression stage, where independent thermodynamic cycles occur; comparative analysis of efficiency for the systems of multistage compression.

1. INTRODUCTION

The low temperatures of heat carriers (< 100ºC), difficulties of dry vapor generation, problems related to vacuum formation, and corrosion of vacuum equipment, caused by air inflows, initiated the refusal to use the conventional steam-water cycle. Since some low-boiling fluorine-chlorine organic compounds are not used now because of demands for ozone safety (Montreal Protocol, Kyoto Protocol), there is a large interest to hydrocarbons, two-component low-boiling actuating media and ammonia-water mixture, whose technological implementation is called the "Kalina cycle". The low-boiling substances of these compounds were used by specific power plants: "Raft River", "Magmamax", "HiPER", etc. (USA); "Mitsishima-Plant", "Otake" (Japan); and Orkuveita Húsavíkur (Iceland). As for application of low-boiling substances in power installations, different priorities for the choice of actuating medium are set in different countries on the basis of technical feasibility and economic reasonability. We should note that all the above substances do not meet the main requirements for the actuating media of the power plants from the point of fire and explosion safety and toxicity. In Russia there is an experience of power plant operation for production of electricity on Freon R12 with the use of low-potential geothermal heat carriers with the temperature of 80ºC (Fig.1).

Another promising approach to replacement of fuel by electricity and renewable and secondary energy supply is construction of vapor-compression heat pump systems for hot water and heat supply. The above low-temperature technologies can be used for alternative energy sources. However, the problems of efficiency intensification and usage limit expansion are topical because of their specific quantity of metal and capital output ratio.

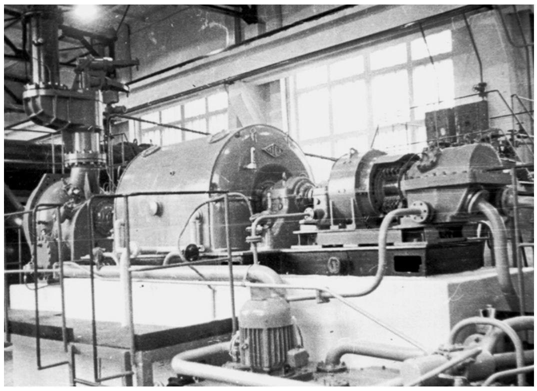

Figure 1: Setup UEF-90/0.5 with power of 750 kW (Kamchatka)

2. COMBINED POWER AND HEAT SUPPLY

The possibility of application of the Russian experience on development and use of the first power setup (UEF-90/0.5), using the low-potential heat carriers (70-130ºC) for electricity production by the total technological cycle on the low-boiling ozone safe actuating medium R134a was considered. In this connection investigation result on power characteristics of the 200-kW setup with R134a in the thermal-power circuit are of the practical interest. Water of 80ºC was used as the heating medium. Waste vapor was cooled by water from the service water system with the temperature of 4ºC for winter and 18ºC for summer periods. The efficiency of Carnot cycle is about η_k = 18%. The Freon power plant uses the Rankine cycle, which is performed by the low-boiling actuating medium within a closed thermal-power circuit. According to the scheme, shown in Fig. 2, liquid Freon is fed to vapor generator 6 by feed pump 5, where Freon vapor with the given parameters is generated by the delivered heat. This vapor is sent to turbine 1, where it is expanded to the final pressure and rotates generator rotor 2, producing electricity. Waste Freon vapors from the turbine are sent to condenser 3, where they are cooled by service water (or by the low temperature of ambient air).

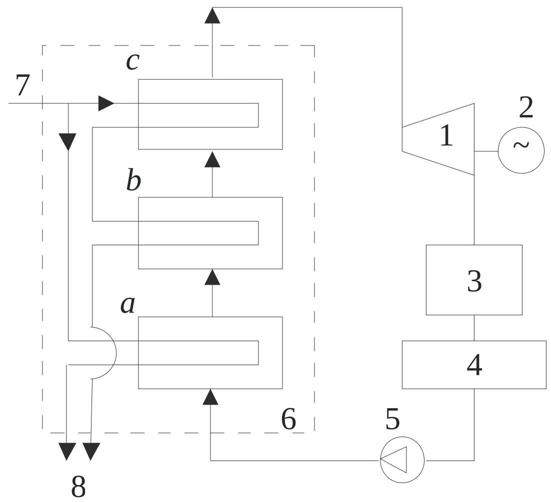

Figure 2: Technological scheme of a binary power setup

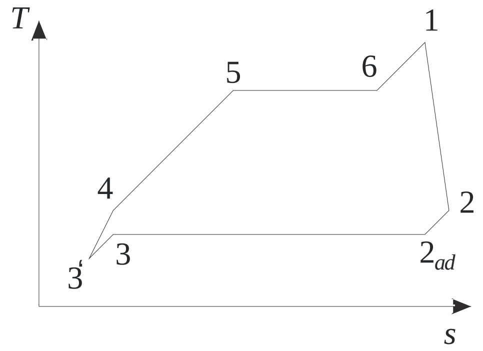

Figure 3: Thermal-power cycle in the s-T diagram

Liquid Freon is sent by feed pump 5 to the Freon vapor generator. After the Freon vapor generator cooled heating carrier 8 is sent to the system of heat supply. The thermal-power cycle is shown in Fig. 3 as the s-T diagram, obtained in the thermal power circuit of the technological scheme. Parameters at the main points of this cycle are shown in Table 1.

2.1. Gas-Dynamic Characteristics of Turbine

The axial single- and three-stage turbines are considered. The given frequency of turbine rotor rotation was n = 3000 rpm, and the vapor pressure in front of the nozzle array was 16 bar, what was 5 % lower than the pressure in front of the turbine stop valve. At this, the initial temperature of R134a vapor was 63.3ºC. The angle of vapor outflow from the nozzle arrays was the same for all stages, and it was α_1 = 11^{o}. The ratio of the circumferential velocity to the velocity, equivalent to available energy, was taken $x_a = U/C_a = 0.47$. The flow rate were set constant μ = 0.95; velocity coefficients within the nozzle array were ψ = 0.9. The effect of turbulence and velocity field nonuniformity was taken into account via the constant coefficient 0.98. The overlapping of nozzle arrays and rotor blades were taken Δ = 0.0035 m. The calculations considered total usage of kinetic energy of the leaving flow in the next stages. The subsonic conditions of Freon vapor outflow from the turbine nozzle array were considered. The gas-dynamic characteristics, obtained during the predesign estimate of the main heat losses, which determine the efficiency and operability of turbine blades, are shown in Table 2. Heat losses for friction, ventilation and leakages through the gaps between the diaphragm and clutch hub can be determined during the design study of Freon turbine. According to preliminary calculations of relative losses, they will vary from 0.03 to 0.035, what allows an estimate of the inner relative efficiency at the level of 0.74÷0.78 for the single- and three-stage turbines, respectively.

The low rate of sound propagation in the medium of R134a at the outlet of the nozzle array (160…167 m/s) at Ma < 1 limits temperature difference on the nozzle array, what increases the reactivity degree (35…57%) and requires an increase in the number of turbine stages. With a rise of the number of turbine stages, the efficiency increases because the outlet velocity of intermediate stages is also used. However, this complication of turbine design increases its cost.

Determination of separate components of energy losses and final point of expansion before Freon inflow into the condenser allows more accurate calculation for Freon vapor condensation at heat removal.

2.2. Flow Rate and Heat Transfer Characteristics of Power Setup

The elaborated mathematic model enables to calculate Freon power plants of any industrial capacity. The prerequisite of invariable electric capacity taken from electric generator has been assumed as the main methodological principle at mathematic modeling of the technological scheme of Freon power plant. This principle is due to the fact that the said technological scheme is designed for standard industrial equipment of serial home production [1].

It is shown in our previous investigations that the share of heat transfer equipment at the low-temperature Freon power plant made up about 50% of all investments to facilities. Heat transfer intensification in the equipment units and original technological solutions allow reduction in the main component of lump-sum costs. One of possible design approaches, reducing specific quantity of metal and capital output ratio, is an increase temperature difference in heat exchangers with parallel directions of heating carrier to each of heat transfer zones: heating and boiling with overheating. With a rise of mean logarithmic difference in a heater, the flow rate of the heating carrier and the temperature at vapor generator outlet increase. In this case the temperature of the heating carrier at the vapor generator outlet is 65ºC (in contrast to 57.7ºC for the version of successive cooling of a primary heat carrier in every considered zone). Therefore, it is reasonable to send a water heat carrier, cooled in the Freon vapor generator, to the system of heat and hot water supply. Thus, we obtain Freon power plant with combined production of electricity and heat. The flow rate and heat transfer characteristics for each heat exchanger are shown in Table 3. At operation in summer at condensation temperature of R134a t_k = 26ºC the electric power of a binary plant is 163.4 kW.

2.3. Economic Indicators

Under the conditions of developing economics, when predictive technical-economical estimates are impeded, the answer to the question about commercial efficiency should be sought not only in the field of technical solutions, but also from the point of economics. Thus, with a rise of costs of primary energy carriers the electricity and heat prices should also increase. It is evident that the maximal efficiency, characterized by fuel economy, can be achieved at long use of the systems for heat and hot water supply. The number of hours of installed power use is set 7000 h/y. At combined production of electricity and heat for the heat supply system, the prices for electricity and heat for all consumers are estimated as 0.04 USD/kW•h and 0.1 USD/kW•h, respectively. Annual results of product sales make up 699 thous. USD, the lump-sum costs of the considered setup make up 282 thousand USD. Under these conditions the payback period is less than one year.

3. ENHANCEMENT OF EFFICIENCY OF THE HEAT PUMP SYSTEMS FOR HEAT SUPPLY

According to review of the foreign and Russian publications, now the efficiency of heat pump systems for heat supply is increased in two aspects: 1) development of a compressor with two-stage compression [2] and 2) design solutions of multistage compression on the basis of available compression equipment. The diagram illustrating replacement of one heat pump (HP) by the HP assemblage, operating within the varying temperature ranges, was presented in the paper of Yu. Petin [3]. This idea seemed to be very original, and investigations were started because implementation of this idea was based on Russian traditional vapor-compression systems engineering, saving time for investigation and development of multistage compressors. The work was financially supported by the Russian Foundation for Basic Research (Project No. 06-08-00447-a).

3.1. Efficiency of Multistage Compression

The principle technological scheme of water heating in the heat supply system by means of the vapor-compression heat pumps is shown in Fig. 4. This technology is characterized by the counter-current flows of low-potential heat carrier and delivery water. The reverse thermodynamic Rankine cycle is implemented in this scheme. Mathematical modeling was used for grounding of staged heating of water in the heat supply system. The feature of this method is based on the combined solution to the thermodynamic problem about energy efficiency of the staged heating of water for the heat supply system and the problem on determination of technical-economical reasonability of the considered technology of staged compression of vapors of the low-boiling actuating medium.

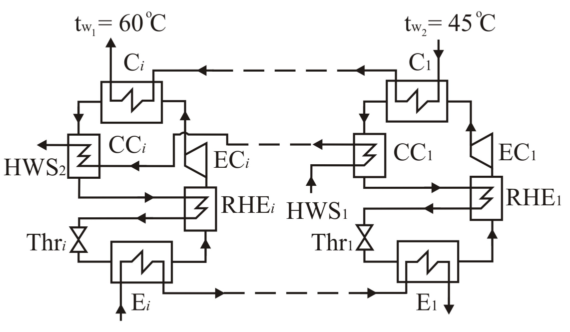

Figure 4: Principle technological scheme of multistage heating of delivery water in the heat supply system

The high energy efficiency of the vapor-compression heat pump system (HPS), whose criterion is the coefficient of energy transformation, is reached at minor difference between the temperature of low-potential heat source and temperature of the heated medium. Beneficial use of the hot Freon heat after its output from the HPS condenser before throttling for water heating in the system of hot water supply (HWS) and heat regeneration in the cycle promotes enlargement of the energy transformation coefficient via reduction of irreversible losses at throttling. However, during comparative analysis it was decided to neglect these processes. While analyzing the energy of reverse thermodynamic cycle of multistage compression in the heat supply system, the following parameters were kept constant for all considered situations: the temperatures of direct and return water in the system (60 and 45ºC, respectively), and the temperature of the cooled low-potential heat carrier behind the heat pump system (10ºC). The number of compression stages was varied at unknown initial temperature of the heat carrier from the low-potential source. Coefficient of energy transformation φ, determined as $\varphi = q_{heat}/q_{com}$, where q_{heat} is heat loading for heating, q_{com} is specific heat of compression of vapors of a low-boiling substance in the compressor, was taken as the efficiency criterion at multistage compression of the actuating medium in the closed thermal-power circuit (*R134a*). The calculation algorithm was developed for the multistage vapor-compression HPS, consisting of several calculation blocks, based on the stage of compression. Calculations were carried out for one, two, three and four stages of compression with consideration of limitations on the temperature of return water in the heat supply system. According to comparative thermodynamic analysis of results, the transition to double-stage compression increases the by 37.5% and reduces the work of compression by 26.2%. A shift of the final point of liquid Freon throttling towards the range of low vapor contents (from x_1= 0.454 for single-stage compression to x_2= 0.382 for double-stage compression) decreases the degree of process reversibility, characterized by entropy of the final point of throttling (S_{thr1}= 1.3442 kJ/kg·K and S_{thr2}= 1.298 kJ/kg·K); this initiates involvement of a higher amount of low-potential heat into the process of energy transformation ($q_{o2\text{-}x}$ = 119.21 kJ/kg > q_{o1} = 105.72 kJ/kg). At this, the temperature of compressed vapors behind the compressor decreased. The effect of a change in intermediate temperatures (pressures) of condensation at intermediate stages was estimated. At three-stage compression of Freon vapors in the compressor the following stages of Freon condensation were considered: 50-57-65°C. It is interesting that the range of water heating temperatures decreased. With this purpose four-stage and six-stage compression was considered. Results are shown in Table 4. Transition to multistage compression always increases the coefficient of energy transformation in the vapor-compression heat pump because irreversible losses are reduced at liquid Freon throttling, and this allows involvement of a higher amount of low-potential heat into the process of energy transformation. The temperature (pressure) difference between evaporation and condensation decreases at every stage of heating and, hence, the work on compression of low-boiling substance vapors reduces. However, an increase in energy efficiency depending on the number of compression stages decreases. Reduction in the temperature range of staged water heating increases energy efficiency of the heat pump system.

3.2. Economic Efficiency

The counter factor, effecting energy efficiency, is technical-economical reasonability of the multistage heating of water in the heat supply system. Under the conditions of developing economy, when predictive technical-economical estimates are impeded, the answer to the question about commercial efficiency should be sought not only in the field of technical solutions, but also from the point of economics. Information from the closed joint-stock company “Energia”, which is the leading Russian producer of the heat pump equipment, was used for technical-economical calculations. The review of technical characteristics and technical-economical indices of the Russian heat pump systems and compression equipment allowed us to determine a rise in the cost of multistage HPS via the total volume of compressed vapors, which equals the volume of vapors, compressed by a single-stage system (Table 5). At this, there is an uncertainty field, related to management of the main equipment of heat pumps (first value) and rise in the cost of construction; water passages, heat network and contingencies (second value). This technical-economical analysis included: estimate of additional investments into the system of multistage compression; calculation of operation costs; determination of technical-economical induces of every closed thermal-power circuit; intermediate compression stages, where independent thermodynamic cycles occur; and comparative analysis of efficiencies of the systems of multistage compression. The system with a vapor-compression heat pump NT-3000 is considered as the basic version of the heat pump system for heat supply with single-stage compression. The number of hours in the heating season is assumed 5450 h/y. It is shown that despite a rise in costs of two HPS by 30% with total volumetric heat productivity of one system, the cost of heat energy, for instance, at two-stage compression decreases by 16.4% because of reduction in expenses for electricity among the total production of heat and cold from 71.3 to 62.1%. A relative change in the cost of heat energy in the system of staged heating of water by heat pump systems is shown in Table 6. Dynamics of a change in the cost of heat energy by means of staged heating of delivery water proves economic reasonability of multistage compression of vapors of the low-boiling actuating medium in a closed thermal-power circuit. According to result analysis, economic efficiency increases within the alteration range of compression stage number. At this, the maximal increase in economic efficiency is observed at two-stage compression of vapors of the low-boiling actuating medium.

4. CONCLUSION

Construction of power plants on R134a is actual and grounded by developments and experimental checks; this option should be considered thoroughly. The scheme solutions allow an increase in plant efficiency and achieve good ecological characteristics. Sometimes these plants allow the refuse to apply high-potential organic fuel. Multistage compression of vapors of the low-boiling substance with staged heating of delivery water increases energy-saving properties of the vapor-compression heat pump system. The system of multistage heating of water in the heat supply system by means of vapor-compression multistage heat pumps if energy efficient and economically reasonable. The reverse Rankine cycle with multistage supply of heat from a low-potential source in evaporators and multistage heat removal in condensers at variable temperatures is some kind of approximation to more efficient Lorenz cycle.

NOMENCLATURE

η_k – efficience of Carnot cycle;

1 – turbine;

2 – generator;

3 – condenser;

4 – linear receir;

5 – feed pump;

6 – vapor generator;

7 – inlet of heating coolant;

8 – to system of heat supply;

a, b, c – heat exchange zones in freon vapor generator;

Ma – Mach number;

ν – viscosity;

λ – heat conductivity;

α_1 – angle of vapor outflow the nozzle;

U – circumferential velocity;

C_a –velocity, equivalent to available energy;

μ – flow rate coefficients;

ψ – velocity coefficients within the nozzle;

Δ – overlapping of nozzle;

t_k – temperature of Freon condensation in summer;

φ – coefficient of energy transformation in a heat hump;

E – evaporator;

RHE – regenerative heat exchanger;

EC – electrically driven compressor;

C - condenser;

CC – condensate cooler;

Thr – throttling valve;

T_{lps1}, tlps2 – temperature of low-potential heat source at the inlet and outlet, respectively;

HWS – hot water supply;

Indices: 1 – for the first stage, *i* – for the i^{th} stage of multistage compression;

t_{w1} – temperature of direct delivery water;

t_{w2} – temperature of return delivery water;

HPS – heat pump system;

NT-3000 – heat pump of 3-MW capacity;

Other symbols are explained in the text and in tables.

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TABLE 1. Main characteristics of actuating medium R134a at common cycle point

Parameter/cycle point	1	2_{ad}	2	3	$3'$	4	5	6
Pressure, bar	16.817	4.43	4.43	4.43	4.43	16.817	16.817	16,817
Temperature, ^{o}C	65	12	16.4	12	11.5	12.5	60	60
Enthalpy, kJ/kg	432.98	405.42	409.6	216.32	215.77	217.01	287.5	426,62
Entropy, kJ/(kg·K)	1.7212	1.7212	1.7360	1.0580	1.0508	1.0576	1.2943	1,7024
Density, kg/m^3	83.827	21.583	21.130	1253.9	1259.15	1252.15	1052.8	87,379
Heat conductivity $\lambda \cdot 10^3$, W/(m·K)	18.45	12.66	13.21	86.30	87.30	86.10	67.10	17,70
Viscosity, $\nu \cdot 10^6$, m^2/s	0.1777	0.5467	0.5689	0.1939	0.1966	0.1931	0.1273	0,1659
Latent heat of vaporization *r*, kJ/kg			189.10				139.12	
Surface tension σ, MN/m							3.98	

TABLE 2. Gas-dynamic characteristics of axial turbines on Freon R134a

Characteristics	Single-stage turbine	3-stage turbine		
		1st	2nd	3rd
Outflow velocity from nozzle C_1, m/s	143.3	75.05	87,59	106,2
Circular velocity U, m/s	108	52.8	60	75,34
Average stage diameter $D_{ave.}$, m	0.688	0.34	0,382	0,48
Speed of sound C_s , m/s	165.7	161.1	167.4	159.8
Reaction degree	0.568	0.506	0.498	0.529
Mach number	0.91	0.49	0.55	0.7
Relative circular velocity x_1	0.7537	0.7	0.685	0.7
Relative vapor velocity at the inlet to working grate W_1, m/s	42.6	25.5	30.9	36.1
Angle of inflow to the working grate β_1, °	39.93	34.1	32.8	34.1
Height of working blade $l_л$, mm	21.6	0.0230	0.0261	25.8
Angle of relative velocity of vapor outflow from the working grate β_2 , °	7.8	10.69	11.2	12.7
Velocity of vapor outflow from the working grate C_2, m/s	55.5	25.5	28.2	38.6
Angle of direction of the vector of outlet velocity α_2, °	15.7	20.65	21.45	24.52
Specific energy losses:				
- in nozzles ξ_n	0.0379	0.046	0.050	0.044
-on working blades ξ_b	0.1146	0.111	0.1126	0.11
- with outlet velocity ξ_{ov}	0.058			0.06
Efficiency of air-gas channel η_{ac}	0.773	0.823		

TABLE 3. Flow rate and heat transfer characteristics of the power plant

Characteristics	Heater	Zone of boiling	Zone of overheating	Condenser
Freon flow rate in circuit D, kg/s	10.3 (10.9)*			
Heat load Q_i, kW	726	1433	65,5	2022.8
Temperature difference Δt_{log}, K	33.7 (18.25)*	-	17.1	5.1
Heat transfer coefficient K, W/(m^2· K)	614.7	1787.8	115.6	1363.5
Heat transfer surface F, m^2	35 (64.7)*	80.2	7.3	293.8
Hot water flow rate G , kg/s	35.3 (23.8)*			
* Values for successive cooling of the heating carrier in the zones of vapor-generator.				

TABLE 4. The effect of staged heating of delivery water on coefficient of energy transformation in a heat pump

No.	Characteristics	Number of compression stages in heat pump				
		1	2	3	4	6
1.	Water heating Δt ,^{o}	15	5/10	5/7/8	5/5/5/5	5/3/3/3/3/3
2.	Coefficient of energy transformation φ	3.457	4.755	5.01	5.374	5.75
3.	By the stages of water heating	3.457	6.03/ 3.879	6.03/4.76/ 4.42	6.03/5.01/ 5.0/5.38	6.03/4.9/5.42/ 5.86/6.2/6.52

TABLE 5. Relative rise in cost of multistage HPS

Characteristics of HPS	Number of compression stages				
	1	2	3	4	6
Relative cost	1.0	1.30-1.35	1.44-1.58	1.67-1.88	1.80-1.89

TABLE 6. A change in the cost of heat energy depending on the number of compression stages

Economic indices	Number of compression stages in HPS				
	1	2	3	4	6
Relative cost of heat energy in the heat pump system	1	0.836	0.823	0.812	0.804