HIGH CAPACITY HEAT PUMPS DEVELOPMENT

Doctor of technical science, prof. I.M. Kalnin[1], Doctor of economy A.M. Mass[1], Candidate of technical science A.I. Savitsky[1]

[1]*Research and manufacturing company EKIP*

KEY WORDS

Heat pumps, capacity, temperature

The development of heat pumps with the capacity of up to 50 MW and operating on carbon dioxide (CO_2, R744) is currently in progress in Russia. The work on this project is carried out by RMC EKIP, MGUIE, NPO “Heliymash” as well as by other participants co-working on the topic “Development of technologies and equipment for the use of low-potential heat resources for the purposes of heat supply” within the framework of the federal program “Research and development on the priority directions of science and technology in 2002-2006”.
The use of R744 in refrigerators and heat pumps attracts a lot of attention throughout the globe. The main advantage is absolute non-hazardousness of this substance. Being cheap and abundant, carbon dioxide is not flammable, non-toxic, safe for the ozone layer and possesses the lowest global warming potential among all working media (Chart 1). Research works that have been carried out during the last few years, have now reached the stage of practical appliance. R744 possesses a number of unique thermodynamic characteristics the correct use of which allows for achieving of higher energy effectiveness compared to traditional refrigerants and for developing of heat pump systems of quite high singular capacity.
Due to low critical temperature of R744 heat pumps based on it combine the processes of steam-compression and gas-based machines: the low-temperature processes take place in the saturation zone and high-temperature processes in super-critical zone. In refrigerating units using such cycle the energy over-expenditure due to higher irreversible losses in heat transfer from gasiform R744 to refrigerants is practically impossible to compensate.
But the temperature conditions in heat pumps significantly differ from those in refrigerating units. The temperature of the source of low potential heat cooled down in evaporator is usually positive, therefore the boiling point of the working media is also above zero as a rule. The boiling point of the working media in the cycle basically coincides with the temperature of the low-potential heat source and lies within the limits of 0..20 ºC (fig.2). The required temperature the heat carrier must be heated up to may lie between 40 and 110 ºC which may require the difference between the inflow and outflow temperatures of the heat carrier (ΔT_w) of 15..60 K. In steam compressor heat pumps the increase of the heat carrier temperature is unambiguously connected with the increase of condensation temperature and pressure as well as with increase of difference and ratio of condensation and evaporation pressures. This leads to decrease of conversion factor. Quite a high temperature of gas after the compressor in the R744 cycle is not strictly connected to pressure. The post-compressor pressure is chosen in a rather narrow range based on the optimization calculations. A significant temperature change when cooling down the gaseous R744 in the super-critical zone allows for heating the heat-carrier to a higher delta of temperatures with the minimum energy loss. A high energy efficiency of R744 heat pumps is achieved at quite a high difference of heat carrier temperatures (ΔT_w) and rapidly rises during its increase. The value of ΔT_w has a little influence on the efficiency of Freon heat pumps because the condensation process is isothermal.
Steam compression Freon heat pumps utilize middle-pressure working media (R134a, for example) or more often low-pressure substances (R142b, for instance) depending on the required heat carrier temperature. R744 is a high pressure working medium which determines a lot of its

characteristics. A working medium opposite to R744 is water (R718). This natural and ecologically clean substance is also considered as perspective for use in heat pumps. Fig. 3 shows typical parameters of theoretical heat pump cycles with isoentropic compression and expanding for four types of working media at the following initial temperatures of: water being heated t_{w1}=40 ˚C, t_{w2}=80 ˚C; water being cooled down t_{s1}=10 ˚C; working medium boiling point t_0=5 ˚C; working medium condensation point $t_к$= 85 ˚C (except for R744). Let us point out the following particular features of R744:

- High density of the gas ρ″ and therefore higher volume heat productivity q_v (5 – 10 times higher than of Freon) stipulate for smaller required volume productivity and compressor sizes.
- Small pressures ratio in the cycle is good for effective work of compressors: higher efficiency, only one compression stage is required when using a centrifugal compressor.
- High pressure level and high density of the gasiform R744 allow for using high mass current speed at the same relative hydro-resistance as of Freon. This allows to decrease channel profiles and tube diameters.
- High mass R744 current speed in heat exchange devices allows to achieve high heat emission coefficient and to reduce the size of heat exchangers.
- A lot bigger share of working medium expansion work in the cycle compared to Freon allows for using an expander instead of throttle in order to increase the conversion factor of the heat pump.

The majority of particular features of R744 listed above which can be considered the advantages of this refrigerant stipulate for construction of heat pumps of greater heating capacity. While the maximum heating capacity of steam compressor heat pumps nowadays is approximately 20 MWth, the R744 heat pumps may have the capacity of 50 MWth and more in one unit. Heating up the heat carrier up to t_w=80 ˚C is close to limit in Freon heat pumps mostly due to bigger pressures ratio $\pi_к$. The parameters of the R718 cycle indicate certain problems developing efficient water heat pumps compared to Freon and R744: large steam volumes, higher pressures ratio $\pi_к$, vulnerability to hydraulic resistance in the channels, the need to maintain vacuum in the system, etc.

A simulator and computer software have been developed for thermodynamic cycles parameters selection and optimizing for R744 heat pumps (with actual efficiencies of compressor, expander and drive).

Depending on operating conditions, expander and throttle schemes are used for high capacity heat pumps. Centrifugal compressors are being used.

Fig. 1 shows a general principle scheme with an appropriate thermodynamic cycle. In the result of optimization calculations the pressure of gasiform R744 at the exit from compressor (P_2) is determined by maximum conversion factor (μ_{max}). The value of $P_{2опт}$ depends on the boiling point temperature T_0 (boiling point pressure P_0), temperatures of heat carrier at entrance and exit (t_{w1}, t_{w2}), the degree of regeneration in the cycle set by the minimum temperature difference between gasiform R744 and heat carrier. A numeric experiment has shown that maximum possible regeneration is advisable for throttle schemes. And with expander schemes μ_{max} on the contrary correlates with minimum overheat of steam entering the compressor. Fig. 2 shows the field of optimum values of pressure P_2 in the actual range of boiling temperatures of R744 (t_0) and temperatures of the heated heat carrier (t_{w2}) for the expander scheme. Fig. 3 shows the values of electric conversion factor at optimum pressures P_2 in the same range of t_0 and t_{w2} temperatures. Fig. 4 contains calculation charts of dependence of electric conversion factor in high capacity heat pumps on R744 and R142b on ΔT_w=(t_{w2}-t_{w1}). An expander scheme is taken for R744 and for R142b - a scheme with throttle and liquid refrigerant over-cooler heated by heat carrier after the condenser. The following temperatures are assumed: boiling point t_0=5 ˚C; heated heat carrier t_{w2}=80 ˚C. The charts on fig. 4 demonstrate that energy efficiency of R744 and R142b heat pumps practically equals close to ΔT_w=30 К. With bigger values of ΔT_w an R744 heat

pump is more efficient energy-wise. With ΔT_w=40 K the conversion factor of an R744 heat pump is 20% higher than of an R142b heat pump.
We should also point out that even with similar energy efficiency of Freon and R744 heat pumps, the latter may turn out more preferable minding the following factors typical for Freon heat pumps:

- technical difficulties of heating up the heat carrier to over t_{w2}=60 ˚C (use of low pressure working media, high pressures ratio of refrigerant's condensation and boiling, etc.);
- large size and mass of the equipment, mainly compressors which impacts its cost and is especially significant for high capacity heat pumps;
- limits imposed on the use of Freon due to its environmental hazard (ozone layer destruction and global warming potentials)
- high cost of Freon which is times higher than the cost of R744.

With quite a high heating capacity of one unit Freon heat pumps cannot compete with R744 heat pumps in any operating conditions.
A large heat pump consists of a compressor (or compressor-expander) unit situated in the machine room and of a number of heat exchange devices located outside the building. The most common set of a compressor unit includes a compressor (and an expander), a multiplier, an electric engine drive, a lubrication system and an automatic control system. A compressor, an expander and a multiplier make up a single block with one incoming axle connected to an electric engine. The mass and the size of this unit are mostly determined by the multiplier. The compressor may be immediately driven by a steam or gas turbine which produces an extra effect by excluding three more energy conversion stages.
Also considered is construction of main heat exchange devices as compact structure based on twisted bunches of pipes with CO_2 flowing inside the pipes with the best possible hydrodynamics of the both currents that take part in heat exchange. Russia possesses manufacturing facilities to assemble the above said equipment for large heat pumps.
The main area where high capacity heat pumps can be used is central heat supply systems (CHSS) where the source of low potential heat is geothermal water or the heat discharged into the environment in the systems of cooling down technical water at heat stations or atomic power plants. At the same time chemical and heat pollution of the environment is reduced.
Fig. 5 shows a principal chart of a heat pumps system with a total heating capacity of 100 MWth used to supply heat for a town and utilizing a geothermal hot well. The hot well's temperature is gradually reduced from 80 ˚C to 45 ˚C in three heat pumps located in different districts of the town. The consumer is supplied with circulation water with the temperature of 80 ˚C (ΔT_w=35 K). The average conversion factor of a heat pump μ=4,7 (primary energy usage coefficient K_{TH}=1,5). The economy of organic fuel is … tons per year.
Usage of heat pumps systems at heat power stations and in central heat supply systems (from heat stations) can result in substantial economy of fuel and economic effect taking into account the following factors typical for CHSS:

- a large quantity of low-potential heat discharged into the atmosphere when cooling down the circulation water which can be utilized;
- significant heat and chemical pollution of the cities' air basins and water reservoirs;
- large losses of network water which requires additional fuel to refill;
- big heat losses in branchy heat supply networks;
- significant difference between the temperature of network water and the temperature used in heating devices which causes overuse of heat energy;
- unjustified high temperature of water pumped into the heating devices that is 50 to 80 °C higher than the temperature in the heated rooms;
- limited carrying capacity of the heat supply networks in the conditions of permanently growing heat demand.

The economy (substitution) of organic fuel with the help of heat pumps systems (HPS) is achieved by utilizing heat discharged from heat stations and due to lower specific fuel usage of HPS.
Chart 4 shows the comparison of consumption of equivalent fuel (29,300 kJ/kg) needed to produce 1 Gcal of heat for communal consumers. The efficiency of a boiler-house is assumed equal to 0.75. Specific usage of equivalent fuel at a heat station is assumed equal to 170 kg/Gcal. Various values of electric energy to heat conversion factor μ are taken for heat pumps systems. It is assumed that the compressor of the heat pump is driven by electric engine and the specific consumption of equivalent fuel used to produce 1 kW/hour of electric energy is equal to 0.320 kg.
Chart 4 demonstrates that with $\mu>2.2$ the HPS technology is more effective energy-wise. However in the end the advantages of each technology must be determined by economic factors, the investment payback period, for instance.
An example of such solution is a HPS with the heating capacity of 50 MW installed at a heat station and warming up circulation water from 20 to 100 °C (fig. 6). The circulating water is cooled down in the heat pump evaporator from 22 to 15 °C. 10 central heat points are equipped with 5 MW heat pump systems that warm up the network water from 40 to 80 °C. The reverse circuit water is cooled down in the evaporators of these heat pumps from 45 to 20 °C. Specific consumption of fuel used for heat production is reduced by 10%. Carrying capacity of the heat supply networks is increased by 1.6-1.8. Besides the following factors are significant too:

- peak periods at HPC are left out or significantly reduces;
- water-to-water heat exchanges at central heat points are left out;
- heat losses in the heat supply networks are reduced;
- absence of need in extending of heat supply networks can compensate the capital investments in construction of HPS;
- usage of steam turbine driven HPS compressor will additionally reduce the specific fuel consumption.

The most effective and obvious variant of using HPS immediately at heat power stations is utilizing the heat of the circulating water in order to heat the raw water fed into the system. It allows saving of at least 70% of fuel used at heat stations for this purpose which determines short payback period of the HPS installed. Heat used to warm up the feeding water takes about 10% of the heat station's heat load. Warming up the additional feeding water from 5 to 95 °C is carried out in two stages. The circulating water is cooled down from 22 to 15 °C in the evaporator of the heat pump, the conversion factor being equal to $\mu=4.5$ and the primary energy usage factor $K_{HP}=1.5$.
The ecological aspects of using heat pumps for heat supply are no less important. The pollution is reduced proportionately to the increase of fuel usage factor. The world experience shows that the use of heat pump technologies is inevitable in order to solve the problems of energy, economy and ecology. Implementation of high capacity heat pumps using carbon dioxide will work out as a powerful tool to solve these problems.

Chart 1. Comparison of features of the working media

Parameter	Symbols	Dimension	Working medium			
			R744	R134a	R142b	R718
Chemical formula	-	-	CO_2	$C_2H_2F_4$	$C_3H_3ClF_2$	H_2O
Ozone layer destruction potential	ODP	Compared to R11	0	0	0,1	0
Global warming potential	GWP	Compared to R744	1	1300	630	<1

Molecular mass	μ	kg/moll	44.1	102.03	100.5	18.02
Critical pressure	$P_{кр}$	MPa	7.38	4.06	4,.2	22.1
Critical temperature	$t_{кр}$	˚C	31.1	101.1	137.2	374.2
Normal boiling point	t_s	˚C	-78.4 (subli-mation)	-26.1	-9.8	+100

Chart 2. Heat pump working temperatures

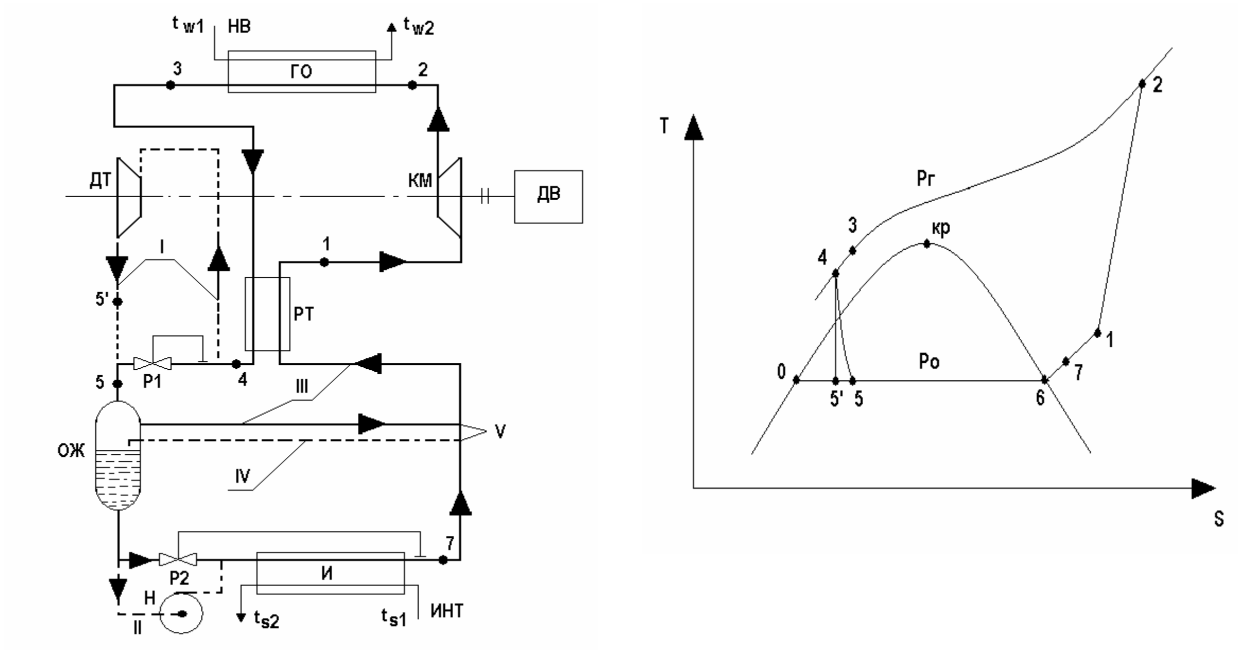
Temperature of low potential heat sources, t_s, °C:		Boiling point of the refrigerant t_0, °C
Subsoil waters	8…12	0…5
Circulation water of cooling systems	18…25	10…15
High temperature heat discharges, geothermal hot wells	>25	≤20
Circulation water temperature t_{w2}, ˚C	40…110	
Circulation water heat-up ΔT_w, K	15…60	

Chart 3. Heat pumps theoretical cycles parameters for various working media

№	Name	Symbol	Dimen-sion	Working media			
				R744	R134a	R142в	R718
1	Boiling pressure	P_0	MPa	3.97	0.35	0.17	$0.89 \cdot 10^{-3}$
2	Post-compressor pressure	$P_к$ (P_2)	MPa	12.7	2.93	1.57	0.059
3	Pressures ratio	$\pi = Pк/P_0$	-	3.2	8.36	9.03	66.3
4	Specific mass heat productivity	$q_к$	kJ/kg	162.9	180.7	215.2	3246
5	Isoentropic compression work	l_s	kJ/kg	44.6	43.2	49.7	923
6	Isoentropic conversion factor	μ_s	J/J	5.765	4.663	4.668	3.680
7	Saturated steam density at P_0	ρ''	kg/m^3	114.0	17.14	8.00	$6.79 \cdot 10^{-3}$
8	Specific volume heat productivity	$q_v = q_к \cdot \rho''$	kJ/m^3	18561	3097	1722	22.1
9	% of the above compared to R744	q_v / q_{vR744}	-	100	16.7	9.3	0.12
10	Mass speeds ratio factor at $\Delta P/P$=idem	$\underline{M} = (P_0 \cdot \rho'')^{0,5}$	-	21.3	2.45	1.18	$2.46 \cdot 10^{-3}$
11	% of the above compared to R744	$\underline{M}/\underline{M}_{R744}$	-	100	11.5	5.5	0.01
12	Expansion work to compression work ratio	$l_{s\ рс}/l_{s\ сж}$	-	0.367	0.103	0.076	0.044

Chart 4. Comparison of equivalent fuel usage

Boiler-house	Heat station	Heat pumps system			
		μ=2,2	μ=3	μ=4	μ =6
190	170	170	124	93	62



a *b*

Fig. 1 An R744 heat pump
a – general principal scheme; b – general thermodynamic cycle

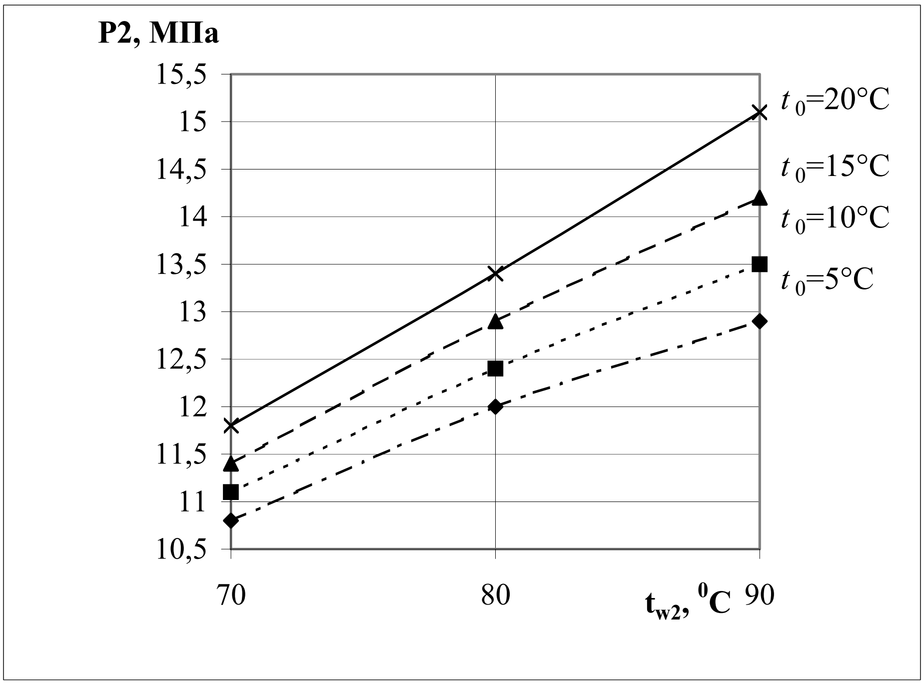


Fig. 2.Optimum pressures field P_{2opt} for R744 heat pumps $\Delta t_w = t_{w2} - t_{w1} = 40$ °C

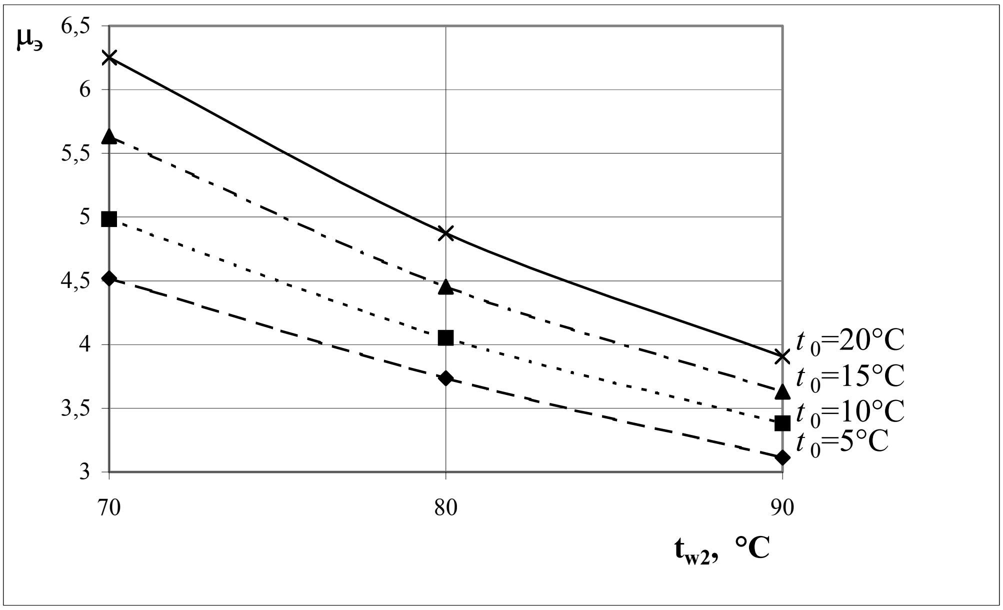


Fig. 3 Field of achievable values of electric conversion factors of R744 heat pumps at optimum values of pressure P_2. $\Delta t_w = 40\ ^0C$

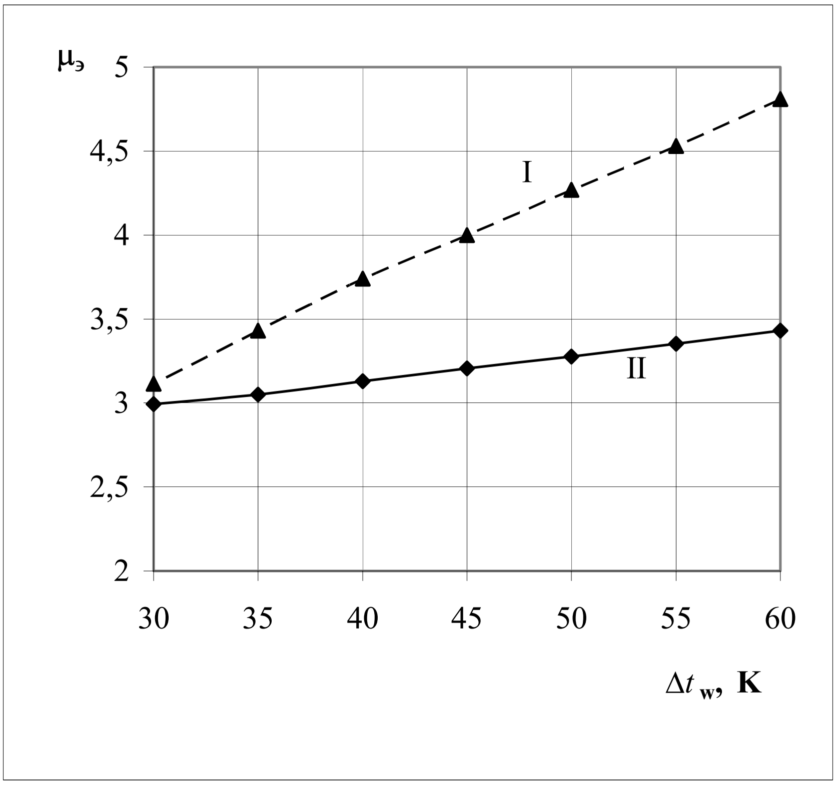


Fig.4. Heat pump electric conversion factor dependence on the difference of temperatures $\Delta t_w=(t_{w2}–t_{w1})$, t_{w2}=80 °C, t_0=5 °C.
I– ТН на R744; II– ТН на R142b

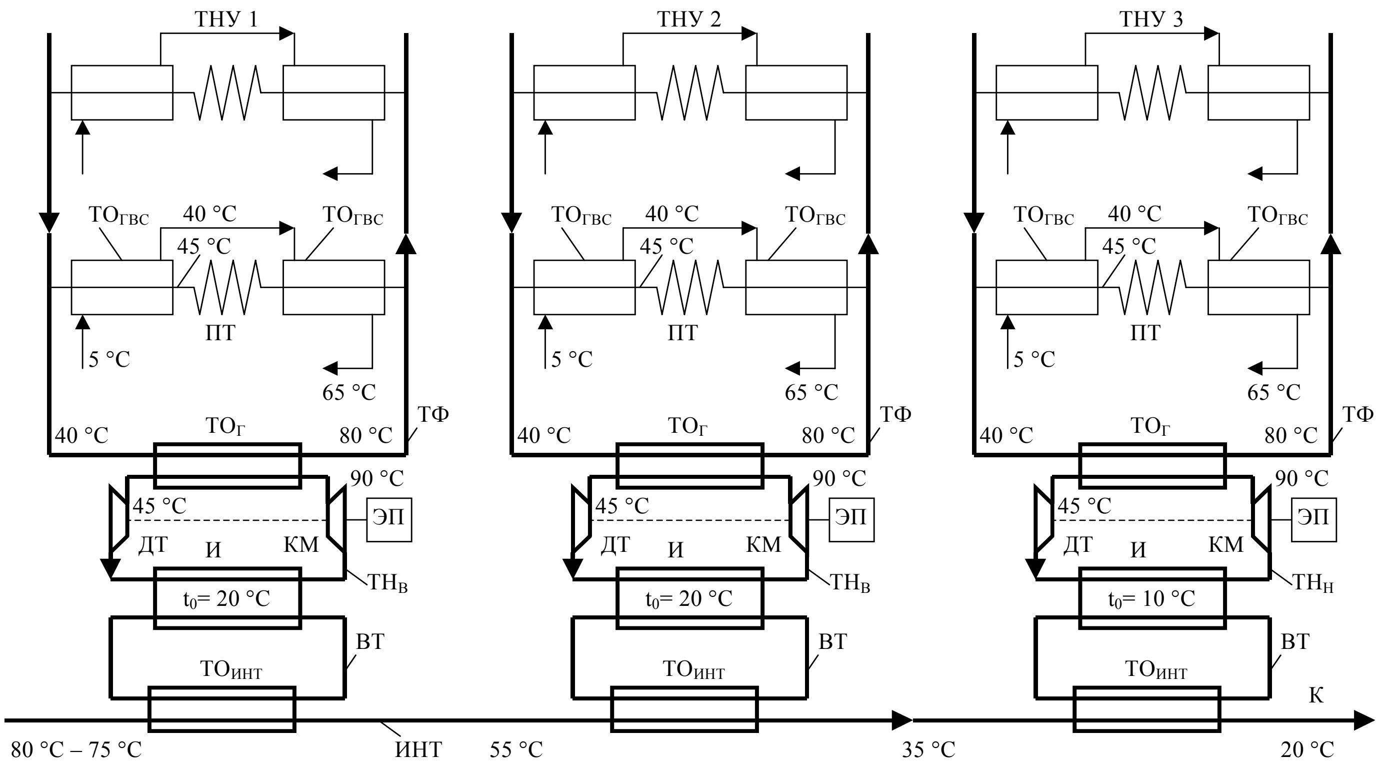


Fig. 5. Principal scheme of HPS for central heat supply of a town with a hot well used as the source of low potential heat and CO_2 used as a working medium.

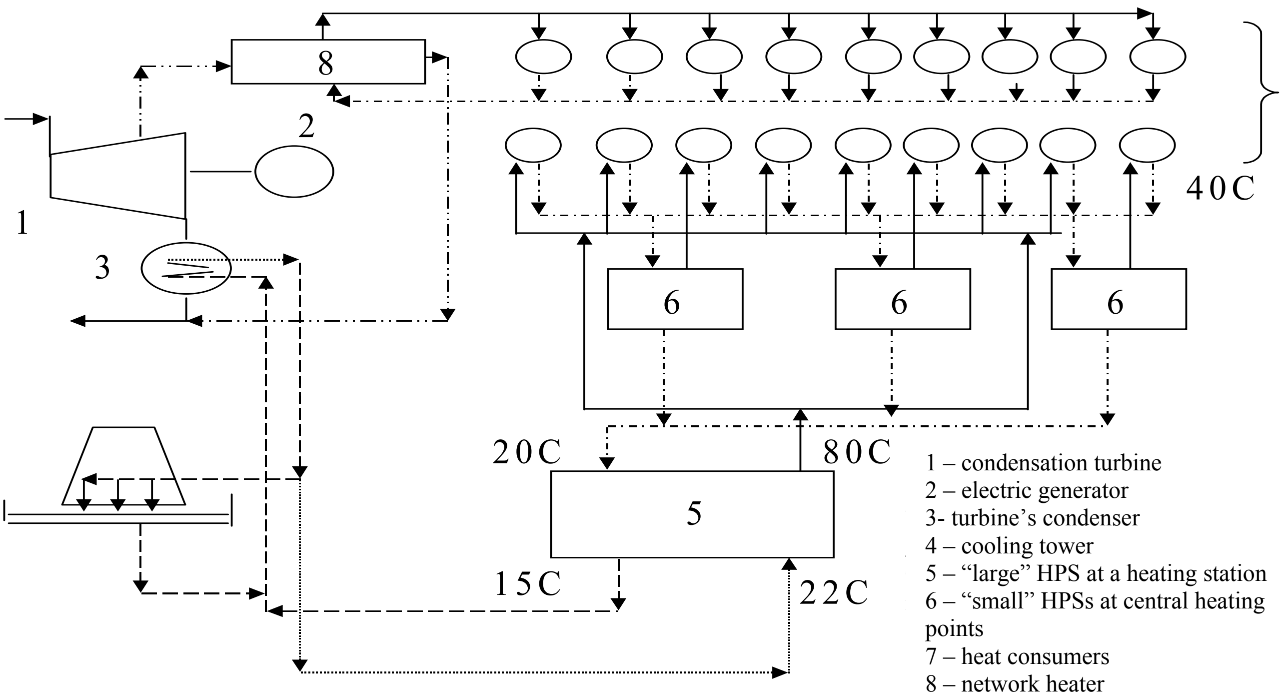


Fig.6 Thermal scheme of heating up feed circulation water with the help of a heat pump.

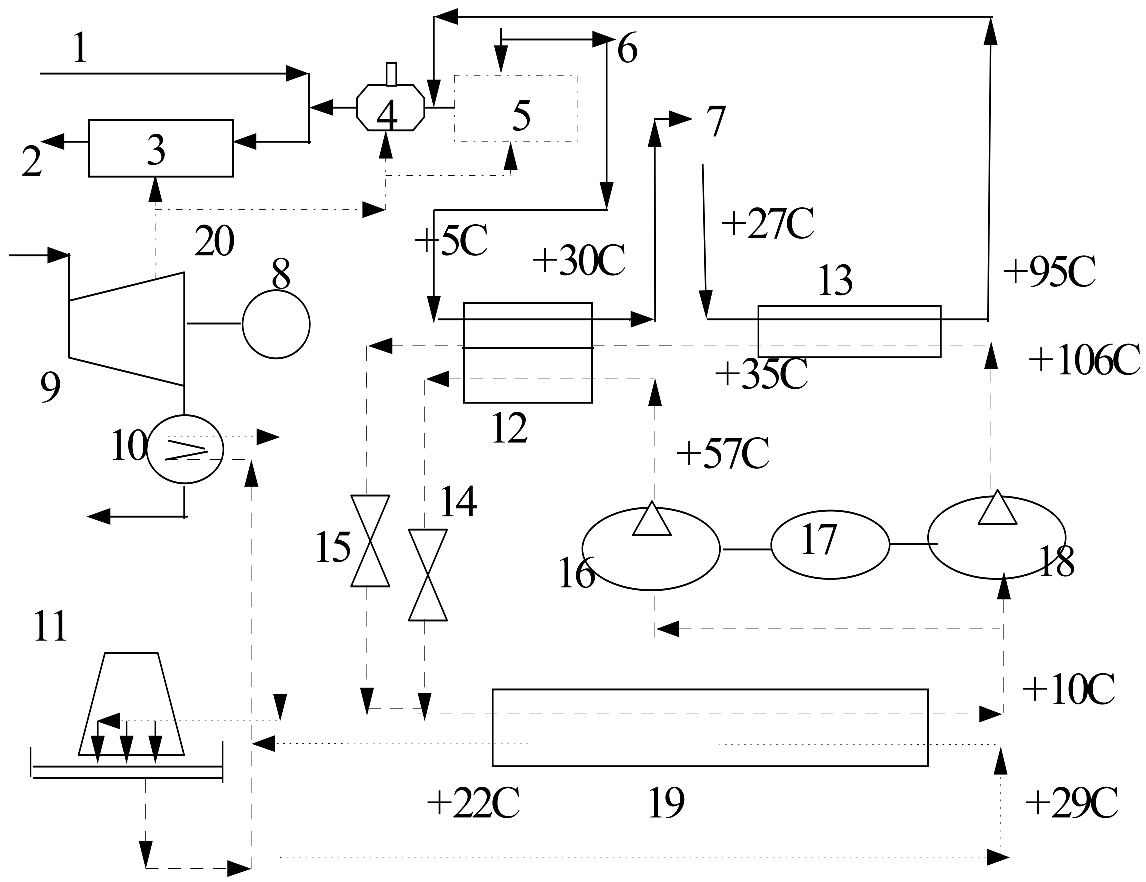


Fig. 7 Principal scheme of HPS for heating up raw feed water at a heat station. 1,2 – reverse and direct circuit network water; 3 – network heater; 4 – deaerator of feed network water; 5 – heaters system for raw and chemically purified network water (replaceable system); 6 – raw water supply line; 7 - chemical water purufucation; 8 – electric generator of thermalclamping turbine; 9 - thermalclamping turbine; 10 - condenser; 11 – cooling tower; 12,13- heat exchangers-gas coolers of the heat pump; 14,15 – heat pump throttles; 16,18 – heat pump compressors; 17- heat pump's turbo-(electric) drive; 19- heat pump evaporator; 20 – steam removal 1.2 атм.