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INCREASING THE EFFICIENCY OF A HEAT SUPPLY SYSTEM BASED ON GEOTHERMAL WATER BY USING HEAT PUMPS*

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This article deals with a heating system which utilises heat pumps for bringing the heat from the discharged water up to a higher temperature level. There appears to be a great future ahead for the use of heat pumps with geothermal water sources, as shown by the example of the heat supply system for Petropavlovsk.

HEAT SUPPLY SYSTEMS BASED ON GEOTHERMAL OR OTHER SOURCES OF HEAT

For heat supplies based on geothermal or other heat sources, one-pipe heat supply systems can be used, with the water coming in from the source either used directly, or else subjected to some additional heating in a heating plant(1-3).

Figure 1 shows the layout of the system of the future - a system which transforms the heat of the discharged water, in conjunction with a peak heating plant.

Water from the source is passed through the purifier, when the amount G_a is pumped by the pumping station along the one-pipe heating system until, at temperature t_a, it reaches the consumer. Part of the water G_1 goes into the space heating system while the rest G_2 is used for the hot water supply. The water entering the heating system is mixed in the mixer 5 with return water, which has been previously heated in the heat pump condenser, then heated in the peak heating plant to supply water temperature t_s, after which it passes through the heating system. On completion of this part of the circuit, the return water, at temperature t_0, divides into three. One part G_3 goes into the heat pump condenser, is heated to temperature t_d and passed into the mixer 5. The second flow of return water G_4 goes into the heat pump evaporator where it is cooled to temperature t_x and discharged. The third flow enters the mixer 10, from which amount G_g at temperature t_g flows into a storage tank and to the hot water supply system.

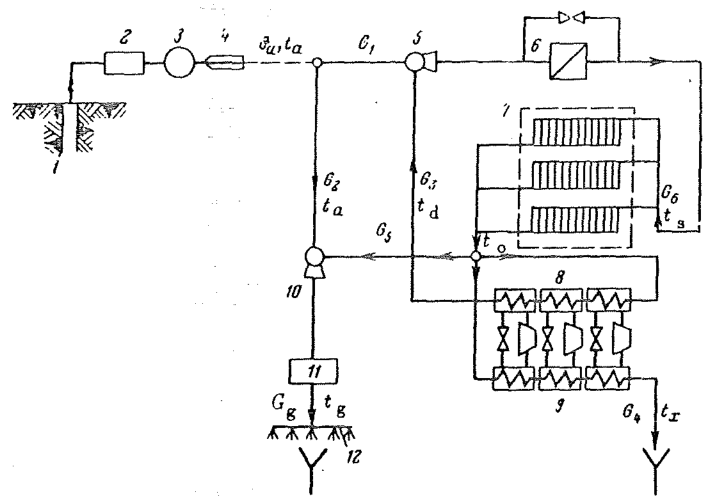

FIG. 1. Main layout of a heat supply system with heat pumps and additional (peak) heating of the water.
1 - geothermal water source; 2 - water purifying unit; 3 - pumping station; 4 - pipeline; 5, 10 - mixers; 6 - peak heating plant; 7 - space heating system; 8 - heat pump condenser; 9 - heat pump evaporator; 11 - storage tanks; 12 - hot water supply system.

To increase the heating coefficient μ, the water is heated in the condensers and cooled in the evaporators by stages, in several heat pump units.

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By jointly solving the equations of the material and heat balances of the heat supply system, we can determine the amount of water drawn from the source per 1 Gcal/h design heat capacity:

$$g_{a1}=\frac{G_{a1}}{Q_d}=\frac{m}{m+1}\left[\frac{(1-\alpha)\cdot 10^3}{t_g-5}\cdot\frac{t_g-t_0}{t_a-t_0}+\frac{\alpha\cdot 10^3}{t_{s.d}-t_{r.d}}\right]+ \frac{(1-\alpha)\cdot 10^3}{(t_g-5)(m+1)},\quad \text{t/Gcal,} \qquad (1)$$

where Q_d is the design heating capacity of the system (with the design external air temperature $t_{e.d}$), Gcal/h, equivalent to the sum of the space heating system capacity $Q_{h.d}$ and the hot water supply system capacity Q_w;

$\alpha = Q_{h.d}/Q_d$;

G_a = total flowrate of water from source, t/h;

$m = (\mu - 1)/\mu \cdot (t_d - t_o)/(t_o - t_x)$;

$t_{s.d}$ and $t_{r.d}$ are the design supply and return water temperatures (at $t_{e.d}$).

When the external air temperature t_e varies, the peak heating plant capacity changes in accordance with the flow control graph, while the heat pump capacity remains unchanged. Therefore with constant source water and hot water supply temperatures t_a and t_g respectively and given α the water flowrate g_{a1} = idem. This condition is also fulfilled after disconnection of the peak heating plant by regulating the heat pump capacity. The water flowrate g_{a1} (t/gcal), other conditions being equal, is in linear dependence on the proportion of the total load that is used for heating, α. The flowrate g_{a1} varies depending on the heating of the water in the heat pump $t_d - t_o$: with a decrease in the temperature difference between the incoming and outgoing water in the condenser $t_d - t_o$, G_a tends to G_g.

The heating capacity of the peak heating plant in the design conditions per 1 Gcal/h supplied to the system is equal to:

$$\beta_d=\frac{Q^d_{ph}}{Q_d}=\alpha\frac{t_{s.d}-t_{d.d}}{t_{s.d}-t_{r.d}}-\frac{1-\alpha}{m+1}\,\frac{t_a-t_{d.d}}{t_g-5}\times \frac{t_a-t_g}{t_a-t_{r.d}}-\frac{\alpha m}{m+1}\cdot\frac{t_a-t_{d.d}}{t_{s.d}-t_{r.d}} \qquad (2a)$$

where $Q^d_{p.h}$ is the design heating capacity of the peak heating plant, Gcal/h.

The temperatures $t_{s.d}$, $t_{r.d}$ and $t_{d.d}$ are determined at the design external air temperature $t_{e.d}$.

The proportion of the total annual heat output which is produced in the peak heating plant is:

$$\beta_{ann}=\frac{Q^{ann}_{ph}}{Q_d}=\int_0^{\tau_1}\left[\alpha\frac{t_s-t_d}{t_{s.d}-t_{r.d}}-\frac{1-\alpha}{m+1}\cdot\frac{t_a-t_d}{t_g-5}\times \frac{t_a-t_g}{t_a-t_{r.d}}-\frac{\alpha m}{m+1}\cdot\frac{t_a-t_d}{t_{s.d}-t_{r.d}}\right]d\tau,\quad \text{Gcal/a/Gcal/h,} \qquad (2b)$$

where τ is the duration of the assumed external air temperature t_e.

The design heating capacity of the heat pump, related to 1 Gcal/h of the heating capacity of the system, is equivalent to:

$$\gamma_d=\frac{Q^d_{hp}}{Q_d}=\frac{\alpha}{m+1}\cdot\frac{t_{d.d}-t_{r.d}}{t_{s.d}-t_{r.d}}-\frac{1-\alpha}{m+1}\times \frac{t_a-t_g}{t_a-t_{r.d}}\cdot\frac{t_{d.d}-t_{r.d}}{t_g-5}, \qquad (3a)$$

while its proportion of the total annual heat output is determined by the formula

$$\gamma_{ann}=\frac{Q^{ann}_{hp}}{Q_d}=\int_0^{\tau_2}\left[\frac{\alpha}{m+1}\cdot\frac{t_d-t_0}{t_{s.d}-t_{r.d}}- \frac{1-\alpha}{m+1}\cdot\frac{t_a-t_g}{t_a-t_0}\cdot\frac{t_d-t_0}{t_g-5}\right]d\tau,\quad \text{Gcal/a/Gcal/h.} \qquad (3b)$$

The installed capacity of the heat pump unit depends on the temperatures of boiling T_e and condensation T_h at the design conditions. For the number of heating stages k the boiling and condensing temperatures of the heat pump unit operating at the i-th stage are equivalent to:

$$\frac{T_{hi}}{T_{ei}}=\frac{T_o+i\Delta t_{hi}+\delta_{hi}}{T_o-[k-(i-1)]\,\Delta t_{ei}-\delta_{ei}}, \qquad (4)$$

where Δt_{hi} and Δt_{ei} are the heating of the water in the condenser and its cooling in the evaporator of the i-th unit, for units of the same type with identical capacities, $\Delta t_{hi} = (t_d - t_o)/k$ and $\Delta t_{ei} = (t_0 - t_x)/k$;

δ_{hi} and δ_{ei} are the recovery deficits at the end of the condenser and the cold end of the evaporator. The relative installed capacity of all the heat pump units is determined by the formula

$$N^{d}_{hp} = \frac{N^{hp}_{ins}}{Q_d} = \frac{\gamma_d \cdot 10^6}{860k}\sum_{1}^{n}\frac{1}{\mu_{di}}, \quad \text{kWh/Gcal,} \qquad (5a)$$

in which μ_{di} is the actual heating coefficient of the i-th unit.

The annual electricity consumption for driving the heat pump units is

$$N^{ann}_{hp} = \frac{\gamma^{ann}_{hp} \cdot 10^6}{860\,k}\sum_{1}^{n}\frac{1}{\mu^{aa}_{di}}, \quad \text{kWh/a/Gcal/h,} \qquad (5b)$$

where μ^{aa}_{di} is the average annual heating coefficient of the i-th unit.

The annual fuel consumption (t coal equiv./a) for the heat output in the peak heating plant and the heat pump per 1 Gcal/h of the design heating capacity of the system is equivalent to:

$$b^{ann}_{1} = 0.143\,\frac{\beta_{ann}}{\eta_K} + 10^3\,N^{ann}_{hp} b_e, \quad \text{t/a/Gcal/h,} \qquad (6)$$

where b_e is the specific fuel consumption per 1 kWh in the condensing station.

In one-pipe heating systems without heat pumps the flowrate of the water from the source is determined by the formula

$$g_{a2} = \frac{(1-\alpha)10^3}{t_g - 5} + n\,\frac{\alpha \cdot 10^3}{t_{s.d} - t_{r.d}}, \quad \text{t/Gcal.} \qquad (7)$$

The proportion of water discharged from the system n is the ratio of the amount discharged from the water system G_4 to the amount of water circulating in the space heating system G_0. This value is determined by the simultaneous solution of the equations for the material and heat balances:

$$n = \frac{t_s - t_0}{t_K - t_0}\cdot\frac{t_c - t_0}{t_a - t_0} + \frac{1-\alpha}{\alpha}\,\frac{t_{s.d} - t_{r.d}}{t_g - 5}\,\frac{t_g - t_0}{t_K - t_0}\times$$
$$\times\frac{t_c - t_0}{t_a - t_0} - \frac{1-\alpha}{\alpha}\,\frac{t_{s.d} - t_{r.d}}{t_g - 5}, \qquad (8)$$

where t_c is the temperature of the water prior to the peak heating plant;

t_k is the temperature of the water after the heating plant.

The proportion of the total installed capacity covered by the peak heating plant is equal to:

$$\beta_{d2} = \alpha - (1-\alpha)\,\frac{t_a - t_g}{t_g - 5} - n\alpha\,\frac{t_a - t_{r.d}}{t_{s.d} - t_{r.d}} \qquad (9)$$

The annual specific heat output in the peak heating plant is:

$$\beta_{ann} = \frac{Q^{ann}_{ph}}{Q_d} = \int_{0}^{\tau_0}\left[\alpha\,\frac{t_s - t_0}{t_{s.d} - t_{r.d}} - (1-\alpha)\,\frac{t_a - t_g}{t_g - 5} - \right.$$
$$\left. - n\alpha\,\frac{t_a - t_0}{t_{s.d} - t_{r.d}}\right]d\tau, \quad \text{Gcal/a/Gcal/h.} \qquad (10)$$

The annual specific fuel consumption in the peak heating plant is

$$b_2 = 0.143\,\frac{\beta_{ann}}{\eta_K}, \quad \text{t coal eq/a/Gcal/h.} \qquad (11)$$

One-pipe systems can operate both with and without water being discharged. When operating without discharge the proportion n is independent of time and equals zero and, in accordance with formula (10), $G_a = G_g$ for the whole of the heating season.

These conditions, and the layout for a one-pipe system of heating, were put forward by V.B.Pakshver for a remote heating system[2,3]. The advantage of a one-pipe heating system based on low-temperature heat sources is that existing radiator systems can be incorporated into it, operating on the 95°/70° flow control graph. The flowrate of the source water is reduced in this case by increasing the fuel consumption. The temperature of the source water affects the efficiency of the system; when $t_a = t_g$ the hot water requirement is met by water from the source, while the heating system water is supplied by the peak heating plant; with lower values of t_a fuel is supplied not only for the heating system but also partly for the hot water supply system.

When the heating network is operating in accordance with the flow control graph, the temperature of the water t_g in the hot water supply network is reduced at the end of the heating season. It is therefore desirable that the design temperatures for the heating network graphs should be as high as possible, not less than 150°/70°, so that the temperature of the water t_g supplied to the hot water system would be reduced for a comparatively short time. It is not always possible to use high design heating network temperatures in heat supply systems based on geothermal heat sources. Thus, for instance, in a number of cases of systems based on geothermal heat supplies the heating of the mineralised geothermal water to temperatures exceeding t_k = 100°C has caused serious operational difficulties.

When a one-pipe system is operating with water discharge ($n > 0$) and flow control of the heat output the larger proportion of the discharge n_{max} will take place with intermediate water temperatures in the network, and where $t_k = t_a = t_c$:

$$n_{max} = \frac{t_s - t_o}{t_a - t_o} - \frac{1-\alpha}{\alpha}\,\frac{t_{s.d} - t_{h.d}}{t_g - 5}\cdot\frac{t_a - t_g}{t_a - t_o}. \qquad (12)$$

In the region of temperatures $t_k > t_a$, n decreases with an increase in t_k and reaches its smallest values at the design temperature $t_{k.d}$ corresponding to $t_{e.d}$. If the design graph for the heating network is selected such that with the design temperature $t_{k.d}$, n_d returns to zero, then in design conditions $G_a = G_g$, and with temperatures $t_k < t_{k.d}$, $G_a > G_g$. In this case a one-pipe system will operate under the conditions put forward by Dyuskin[1,2]. For a system of this type, the design supply water temperature $t_{k.d}$ is determined from the condition:

$$t_{k.d} = \frac{\alpha}{1-\alpha}(t_g - 5) + t_g. \qquad (13)$$

The design flowrate of water from the source g_{a2} for this system is determined from formula (7), with n_{max}.

In one-pipe systems the heat supply system is insufficiently effective on account of discharge of water at temperature t_o, while in systems without discharge a decrease in the water flowrate is brought about by increasing the fuel consumption.

Systems which make the fullest use of the source heat are those with heat pumps, since here water is discharged at a temperature t_x = 5 - 10°C. The effectiveness of the use of the source heat increases with a decrease in the return water temperature $t_{r.d}$. For systems with heat pumps it is therefore best to use panel or convector heating. Panel heating systems ensure that the surroundings are cooled in summer.

In geothermal heat supply systems where the drilling of boreholes and construction of pipelines accounts for a considerable portion of the expenditure, a considerable improvement can be made in the technical-economic characteristics by using heat pumps, as shown below by the example of Petropavlovsk.

THE GEOTHERMAL HEAT SUPPLY SYSTEM OF PETROPAVLOVSK, BASED ON HEAT FROM THE GEOTHERMAL WATERS OF THE PARATUNSK REGION

The town is supplied with heat from the geothermal waters of the Paratunsk sources from a distance (as regards pipelines) of about 60 km. The design heating capacity of the heat supply system is Q_d = 200 Gcal/h. There is a guaranteed hot water supply for the industrial settlements and the living accommodation, and the thermal loads are therefore distributed thus: $Q_{h.d}$ = 133 Gcal/h for heating and Q_g = 62 Gcal/h for the hot water supply (α = 0.69). The geothermal water temperature t_a = 75°C, t_g = 65°C.

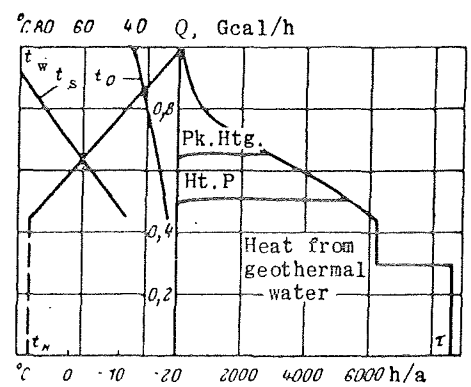

FIG. 2. Annual graph of heat output for heat supply systems with heat pumps (α = 0.69).

In these conditions the most suitable system is one that entails the use of heat pumps, which transform the heat of the discharged water to a higher temperature level, together with an additional heating plant (as in Fig. 1).

The advantages of systems with heat pumps can be shown by comparing these with other heat supply systems:

one-pipe system without discharge of water[2,3];

one-pipe system with discharge of water[1,2];

heat supply system using peak heating and heat pumps.

One-pipe heating systems both with and without discharge employ the 95°/70° flow control graph. In the system without discharge the water flowrate is constant for a period of a year, and $G_a = G_g$. In the system with discharge the highest design water flowrate, determined by (7) and (12), is $G_{a.d} = 1.6\ G_g$. The design supply water temperature in the mains $t_{k.d}$, determined from (13), is 193°C. The 150°/70° graph was used for network distribution without discharge.

In the system with heat pumps, the water is additionally heated to 95°C, is heated in the condenser of the heat pump to 55°C and cooled in the evaporator to 5°C. Heating and cooling of the water take place in three stages, in three heat pump turbo-compressor units. The design temperature for the return water in panel heating systems is 40°C. Figure 2 shows the annual heat output for this system, while Table 1 gives the design heating capacities and the annual proportions of the total heat output for the peak heating plant and the heat pumps.

TABLE 1.

	Installed capacity, kW	Installed heating capacity		Annual heat output		Fuel consumption, t/a
		Gcal/h	%	Gcal/h	%	
Peak heating plant	–	71.2	35.5	66 000	6.8	15 200
Heat pump	7 800	28.2	14.1	120 000	12.3	13 200
Geothermal water	–	100.6	50.4	794 000	80.9	–

We will look at the effect which the ratio of the thermal loads for the heating and hot water supply systems has on the geothermal water flowrate g_a. Figure 3 shows the nature of the variation of g_a depending on α for the geothermal heat supply systems considered. The flowrate g_a in systems without discharge and with heat pumps depends linearly on α. In the system with water discharge the dependence $g_{a.d} = f(\alpha)$ is non-linear, since in

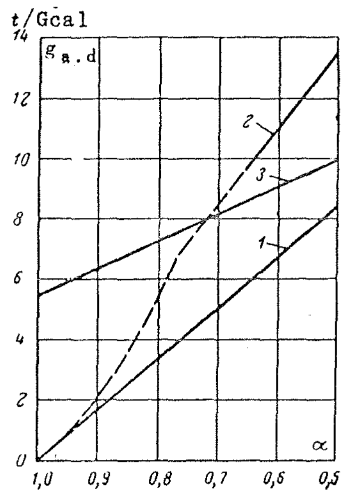

FIG. 3. Design flowrates of source water in various one-pipe heating systems depending on the proportion of space heating α in the total thermal load.
1 – system without water discharge[2,3];
2 – system with water discharge (in design conditions $G_a = G_g$)[1,2]; 3 – system with heat pumps.

accordance with equation (13) $t_{k.d} = f(\alpha)$. Starting from $\alpha = 0.69$ the design water flowrate $g_{a.d}$ in this system exceeds that in a system with heat pumps, since, with a decrease in α, G_g and the ratio $G_{a.d}/G_g$ increase and $t_{k.d}$ decreases. With $\alpha < 0.69$ the flowrate $g_{a.d}$ is less than that in a system with heat pumps, since $t_{k.d}$ and the water temperature t_k increase at the end of the heating period. With $t'_k > t_a$ ($\alpha \approx 0.77$) the design flowrate $g_{a.d}$ decreases more rapidly, the nearer it approaches G_g. It is not, however, an advantage to use this kind of system in this region of thermal loads, on account of the unnaturally high water temperatures t_k in the network (cut-off line in Fig. 4). In a one-pipe system without discharge[2,3] the water flowrate corresponds to the flow to the hot water supply system G_g.

The technical and economic characteristics of the various systems are determined by the capital and operational costs. Capital investments in the geothermal heating system are made up of capital expenditure $K = \Sigma K_i$: cost of boreholes K_{bo}, pipelines $K_{p.1}$, peak heating plant $K_{p.h}$, fuel K_f, heat pumps $K_{h.p}$, condensing station K_e, heating equipment $K_{h.e}$, pumping station K_p. The operational costs comprise the total expenditure for depreciation deductions, cost of mining and transporting fuel, staff wages and other expenditure (30% of depreciation costs and wages costs) – $S = \Sigma S_i$. Comparison of the heating systems is made according to the predicted costs: $PZ = 0.125\ K + S$.

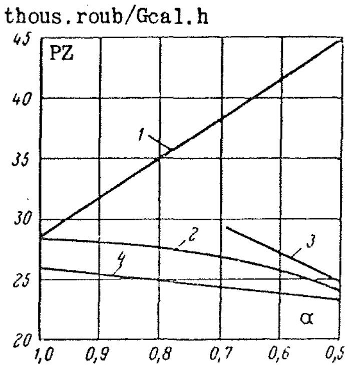

FIG. 4. Theoretical costs depending on the proportion of space heating α in the total thermal load.
1 – heat supply from heating plant;
2 – one-pipe system without water discharge[2,3]; 3 – one-pipe system with water discharge, with design conditions $G_a = G_g$[1,2]; 4 – with heat transformed by heat pumps.

Capital investment in the heat supply system was determined taking into account the specific standards applicable to the particular conditions of Kamchatka.

With the mean depth of a borehole in the Paratunsk of 600 m, the mean output 12 litres/s and the cost per metre of drilling 110 roub/m, $K_{bo} = 5.5\ G_a$ thous.roub/Gcal/h. Capital investment in the condensing stations, heat pumps and pumping stations was determined according to the cost of 1 kW installed capacity. Capital investment per 1 kW was taken as: condensing station – 350 roub/kW, heat pumps 80 roub/kW and pumping station 25 roub/kW installed capacity. The specific fuel consumption in the condensing station was taken as $b_{cs} = 0.345$ kg coal equiv./kWh, the mean heating coefficient with three-stage heating μ, depending on the conditions, 4.4 - 5.6, and pump efficiency $\eta_p = 0.8$. Specific capital investments in a one-pipe line about 60 km long was determined approximately, taking into account variations in the water flowrate, from the relation $K_{p.1} = 17.0\ g_a^{0.38}$ thous.roub/Gcal/h.

In the variants compared the cost of supplying heat, from investment in the heating plant to the mining and transport of the fuel (Sakhalinsk coal with a calorific value of 5800 kcal/kg) was taken as 57.9 roub/t fuel, the net cost of mining 12.9 roub/t and transport 5.44 roub/t. Specific capital investments on the construction of the heating plant for Kamchatka amounted to 30 000 roub/Gcal/h.

Table 2 shows the results of technical-economic calculations for various types of heating systems with the same heating capacity Q_d = 200 Gcal/h. For these conditions the most effective system is that using geothermal heat and heat pumps.

TABLE 2. Technical-economic characteristics of the Petropavlovsk heat supply systems.

	Heat supply systems			
	From heating plant	One-pipe without discharge[2,3]	One-pipe with discharge[1,2]	With heat pump
Design geothermal water flowrate, litres/s	–	280	470	470
Design heating capacity Gcal/h	200	200	200	200
Annual heat output, Gcal/h	960 000	960 000	960 000	960 000
Capital investments, mill.roub.	19.2	15.77	18.3	16.2
Operational costs, mill.roub.	5.25	3.28	3.38	2.60
Fuel consumption, t/a	222 000	92 000	87 200	35 000
Saving in capital investment, mill.roub	–	3.43	0.9	3.0
Saving in operational costs, mill.roub.	–	1.98	1.87	2.65
Saving in fuel, t/a	–	130 000	135 000	187 000
Theoretical costs, mill.roub/a.	7.65	5.25	5.88	4.62

Figure 4 shows the variation in the specific theoretical costs (for Q_d = 1 Gcal/h) depending on the ratio of the thermal loads for the heating and hot water supplies.

With an increase in the proportion of the theoretical costs accounted for by the hot water supply, the theoretical costs for the heating system from the heating plant increases, since the annual output of heat and the fuel consumption is greater. For heating systems with heat pumps the theoretical costs drop in proportion to a decrease in the proportion of the heating load α, since there is a reduction in the proportion of heat put out from the heating plant and the heat pumps, and an increase in the direct use made of the heat from the geothermal water.

The theoretical costs of a one-pipe system without water discharge[2,3] decrease with an increase in the proportion of the hot water supply in the total thermal load. The theoretical costs for the system with water discharge[1,2] are shown in Fig. 4 in the region of values of α where operation of this system is still possible. In practice, in one-pipe systems of heating both with and without water discharge, the space heating system is based on the heating plant which uses fuel, while the hot water supply uses geothermal water. Both one-pipe systems are economically effective only with high hot water supply loads. For the conditions we have looked at, one-pipe systems are advantageous when α is 0.55 and below.

For heat supply systems with heat pumps the theoretical costs fall with a decrease in the proportion of the heating load α, since there is a reduction in the amount of heat produced by the heating plants and the heat pumps, and there is an increase in the direct use of geothermal water. Supplying heat with heat pumps enables full use to be made of the heat from the source on account of the effective transformation of the heat from the water when it is heated in stages.

SUMMARY

1. The fullest use of heat sources is made by systems based on geothermal or other low-temperature sources where heat pumps are used to bring the source heat up to a higher temperature level.

2. Heat supply systems with heat pumps used in conjunction with peak heating are characterised by small flowrates of water from the source with the hot water supply in practice occupying a comparatively small proportion of the total thermal load (with α = 100 – 65%).

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