EFFICIENCY OF DECENTRALIZED HEAT SUPPLY BASED ON TRADITIONAL HEAT GENERATORS WITH VAPOUR-COMPRESSION CONVERSION OF ENERGY FROM LOW-TEMPERATURE SOURCES

Vitaly Petrash, Vitaly Baryshev, Leonid Shevchenko, Elina Geraskina
Odessa State Academy of Civil Engineering and Architecture, Ukraine
petrant@ukr.net, baryshev@odaba.edu.ua, bgedyx7@ukr.net, poselok@te.net.ua

Abstract. The research is devoted to the solution of the actual problem of increasing the efficiency of combined systems of decentralised heat supply of public buildings and industrial enterprises with characteristic heat generators on the basis of heat-pump energy conversion of preliminary cooling of heat flows of the used heat carrier from the heat network, initial cold water and exhaust gases. The aim of the work was development, analysis and identification of conditions to improve the energy efficiency of the proposed combined system which provides increased generation capacity for decentralised heat supply. The proposed approach allowed solving the problem of improving energy and environmental performance in a two-stage scheme of heat generators with the regulation of the temperature of the used heat carrier to a rational value using the heat pump technology, consistent with the standard temperature schedule. The most important result of the analytical study of the combined decentralised heat supply system is the established generalised dependence for determining the actual conversion factor in the heat pump operation. It makes it possible to qualitatively analyse the dependence of the energy efficiency of the heat supply system on temperature changes of characteristic low-potential sources, the ratio of water flows and their distribution for municipal, domestic and industrial-technological needs. The results of the analytical study of the improved heat supply system create a basis for its development taking into account the characteristic conditions of decentralised heat supply and operating modes of municipal and industrial-technological enterprises.

Keywords: heat generating units; heat supply; vapour-compression conversion; integrated heat; actual conversion factor.

Introduction

The requirements for thermal protection of buildings [1] with modernisation of heating systems provide possibilities for appropriate reduction of the calculated capacity of heat supply systems. The maximum calculated temperature in the supply lines is 90-120 ºC and in the return lines 40-70 ºC. It is logical in combined systems to regulate the water temperature in heat networks taking into account changes in heat capacity, their physical condition, as well as seasonal fluctuations in the outdoor air temperature and time of day. Better regulation of heat supply to subscriber systems can reduce heat consumption by up to 50% [2], especially during transient periods. It is reasonable to improve thermal-hydraulic regimes of heat networks during operational and technological changes, including periodic operation of industrial enterprises and during the day. The increasing temperature of the used energy carrier contributes to a decrease in the efficiency of fuel consumption by heat generators and electricity consumption by circulation pumps with a subsequent increase in heat losses in the mains [3].

The utilisation of the available heat of the supply cold water for industrial, technological and municipal needs is becoming more and more attractive, especially in the transitional and inter-heating periods of the year. The temperature of water from open sources during these periods in the southern regions of Ukraine is in the range from 10 to 20 ºC [4]. The results of research of the proposed system of recuperation with deep utilisation of used gases heat after the heat generator provide increase of the boiler efficiency by reducing their temperature to 85 ºC [5]. Industrial plants have a number of low-temperature heat sources that can be used to improve the efficiency of combined heat supply systems.

Deep cooling of flue gases with reduction of nitrogen oxide emissions is considered actual in terms of environmental friendliness and very promising for improving the fuel efficiency [6-8]. Systems with extended capabilities of heat pumps increase the efficiency of heat supply to remote consumers at a reduced temperature schedule and economic performance of heat networks due to the utilisation of the energy potential of the used energy carrier as a low-temperature source [9] in the operation of CHPPs. The construction of combined district heating systems based on CHPPs using the possibility of heat pump technologies for remote consumers is justified by technical and economic indicators [10; 11]. The research results of the new system of flue gas heat utilisation using thermal transformer technologies are reasoned as promising for further use in municipal and industrial-technological systems [12].

The specifics of effective use of the heat pump units with electric and gas-turbine compressor drive in the operation of thermal power plants are analysed in [13]. A generalised indicator for preliminary estimating the conditions for improving the efficiency of modernised both centralised and decentralised heat supply systems with the use of heat pump technologies has been defined [14]. A heat supply scheme allowing to redistribute heat flows between separate magistral lines with different inertial capacities is proposed, which makes it possible to adjust the schedule of daily heat consumption using heat pump technologies [15].

Thus, the joint use of characteristic low-temperature sources on the basis of thermo-transformer technologies opens up prospects for further improvement and energy efficiency of combined heat supply systems. It is logical that reduction of primary fuel consumption on the basis of heat pump technologies and reduction of harmful impact of used gases on the environment provide increase of ecological efficiency of the corresponding systems.

Materials and methods

The aim of the work was to develop an improved system of combined heat supply for municipal and industrial-technological buildings using available energy of characteristic low-potential sources, for which it is necessary to perform an analytical study of its energy efficiency with the definition of rational parameters and regime conditions.

The objectives of the research were: – development and substantiation of a rational construction of a combined heat pump heat supply system with traditional heat generators for its energy-efficient operation; – for the proposed system to analytically establish the heat-energy interrelation of structural elements and the dependence of the actual coefficient of transformation of energy flows, allowing to perform an estimation of the conditions of its operation depending on the initial and mode parameters; – based on the results of the analytical study, to determine ways to improve the energy efficiency of the system while increasing the capacity of generated heat, taking into account the characteristic operating conditions of structural subsystems in different periods of the year.

Results and discussion

1. Structural and functional construction and operation of the proposed system

The proposed system (Fig. 1) works as follows. The spent energy carrier enters the heat exchanger 2 through the pipeline 1 for its preliminary cooling during the operational regulation of the heat supply system. After passing through the evaporator 3, a part of the used heat carrier, by means of a three-way temperature regulator 4, is directed by the circulation pump 6 through the pipeline 5 to the regenerative heat exchanger 10, and the remaining part - to the recirculation pipeline 8. The recirculation part of the waste heat carrier through the pipeline 9 goes to the pipeline 1 of the heat network.

Part of the pre-cooled general purpose cold water from the pipeline 11, which has passed through the regenerative heat exchanger 10, is directed to the heat exchanger 14 through the branch pipeline 13 by means of the three-way temperature regulator 12. This provides additional cooling of flue gases after the heat generator 15. The water heated to the appropriate temperature by cooling the flue gases passes through the pipeline 32 and enters the distribution temperature controller 31, which includes a temperature difference sensor and branches. The water is then distributed to the storage tanks 29 and 30 for a two-tiered hot water system.

2. Analytical study of the conditions for energy-efficient operation of the system

From the analysis of the combined system operation, Fig. 1, it follows that the total flow rate of initial cold water under conditions of decentralized heat supply is determined by the needs of subsystems of municipal and industrial-technological purposes with provision of two-level hot water supply, i.e. G_{tec}, G_{hw}, therefore, $G_{\text{cw}} = G_{\text{tec}} + G_{\text{hw}}$, where the total water consumption for two-level hot water supply G_{hw} is determined by its respective components $G_{\text{hw}} = G_{\text{hw1}} + G_{\text{hw2}}$. Taking into account the reasonable allocation of cold water, its components for hot water supply are defined as:

$$G_{\text{hw1}} = \beta G_{\text{cw}};\; G_{\text{hw2}} = \alpha G_{\text{cw}}, \tag{1}$$

where α, β – corresponding parts of the flow rate of the initial cold water, which enter the heat exchangers 14 and 20.

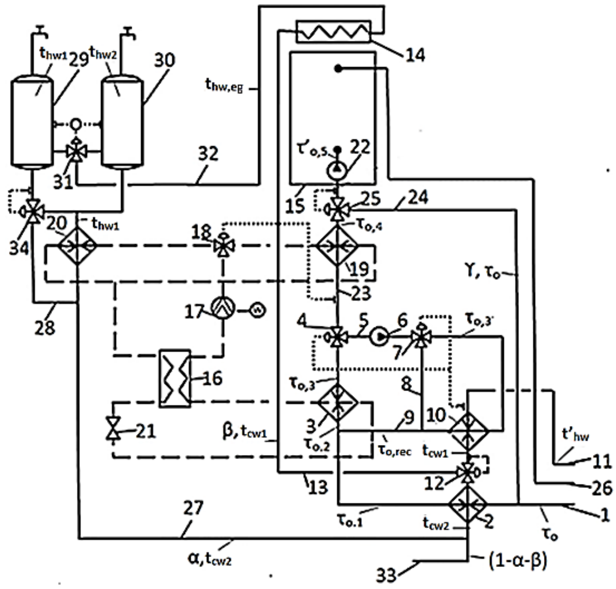

Fig. 1. **Schematic diagram of the device of a thermo-transformer system for decentralised heat supply based on integrated aftercooling energy of the used heat carrier, initial cold water and flue gases:** 1 – heat network return pipeline; 2 – recuperative heat exchanger; 3 – heat pump evaporator; 4 – three-way temperature regulator; 5 – recirculation pipeline; 6 – circulation pump; 7 – three-way temperature regulator; 8 – branch section of the recirculation pipeline; 9 – general recirculation pipeline; 10 – heat exchanger; 11 – initial cold water pipeline; 12 – three-way temperature regulator; 13 – pipeline to the exhaust gas aftercooling heat exchanger; 14 – exhaust gas heat extraction heat exchanger; 15 – heat generator; 16 – heat pump regenerative heat exchanger; 17 – heat pump compressor with external drive; 18 – three-way temperature regulator; 20, 19 – first and second parts of the condenser connected in parallel; 21 – throttle valve; 22 – circulation pump; 23 – pipeline of the exhaust heat carrier from the network; 24 – bypass pipeline; 25 -– three-way temperature regulator; 26 – main pipeline of the heat network; 27, 28 – distribution pipelines for two-level hot water supply; 29, 30 – accumulator tanks for two-level hot water supply; 31 – two-way temperature regulator with the temperature difference sensor for hot water distribution; 32, 33 – pipelines for hot water supply and heat-technological purposes; 34 – three-way temperature regulator

In the recuperative heat exchanger 10, the recirculation part of the exhaust energy carrier absorbs the cooling heat of the initial cold water with the total flow rate G_{cw} for municipal and industrial-technological purposes. For this process, a relation in this form is valid:

$$c_p G_{cw}(t'_{cw} - t_{cw1}) = c_p \rho (1-\gamma) G_{wec}\left(\tau_{0,rec} - \tau_{0,3}\right), \tag{2}$$

where: G_{wec} – flow rate of waste heat carrier from the heat network;

c_p – isobaric specific heat capacity of water;

$\tau_{0,rec}$ – temperature of recirculating the used heat carrier after the heat exchanger 10;

$\tau_{0,3}$ – temperature of the used heat carrier after evaporator 3, which according to [16] on heat-technological requirements for HPU is recommended to take equal to 5 ºC;

γ – part of the total flow rate of the used heat carrier flowing through the bypass pipeline (position 24);

ρ – recirculation coefficient of pre-cooled exhaust energy carrier through the heat exchanger 10.

From the relation (2) the dependence of the recirculation coefficient of the pre-cooled used heat carrier through the heat exchanger 10 is determined as follows:

$$\rho = \frac{G_{\mathrm{cw}}}{G_{\mathrm{wec}}} \frac{(t'_{\mathrm{cw}} - t_{\mathrm{cw1}})}{(1 - \gamma)\left(\tau_{0,\mathrm{rec}} - \tau_{0,3}\right)}, \tag{3}$$

where t'_{cw} – temperature of initial cold water, which value increases from 5°C under design conditions to 10°C by the end of the heating period;

t_{cw1} – temperature of initial cold water after the heat exchanger 10, which in the heating period without recirculation is equal to $t_{\mathrm{cw1}} = t'_{\mathrm{cw}}$.

On the basis of equality of heat flows of the pre-cooled used heat carrier and its recirculation part in the heat exchanger 10 with the corresponding temperatures $\tau_{0,1}$ and $\tau_{0,\mathrm{rec}}$, the temperature of their mixture $\tau_{0,2}$ at the inlet to the evaporator 3 of HPU is determined, which after the appropriate transformation takes the form:

$$\tau_{0,2} = \frac{\left(\tau_{0,1} + \rho\tau_{0,\mathrm{rec}}\right)}{1 + \rho}, \tag{4}$$

where $\tau_{0,1}$ – temperature of the used heat carrier from the heat network after the heat exchanger 2, which in the design conditions of the heating period in the absence of recirculation (ρ = 0), according to the heat technology requirements of HPU [7] at the inlet to the evaporator is assumed to be $\tau_{0,1} = \tau_{0,2} = 25$ ºC.

For the boundary conditions of the countercurrent scheme of operation of modern plate heat exchangers 10 with the highest efficiency of the heat exchange process, the temperature of cold water of general purpose t_{cw1} in engineering practice is recommended [4; 9] to be taken, in general, with a limit value of 5ºC.

At the same time, the temperature of the recirculation part of the used heat carrier after the heat exchanger 10 $\tau_{0,\mathrm{rec}}$ exceeds the temperature t_{cw1} by 5 ºC. Therefore, in general, after appropriate transformations, the dependence for determination t_{cw1} takes the form:

$$t_{\mathrm{cw1}} = t'_{\mathrm{cw}} - \frac{G_{\mathrm{wec}}}{G_{\mathrm{cw}}} \rho(1-\gamma)\left(\tau_{0,\mathrm{rec}} - \tau_{0,3}\right). \tag{5}$$

As a result of the heat exchange process of the initial cold water in the heat exchanger 2 with the waste heat transfer medium with the flow rate $(1-\gamma)G_{\mathrm{wec}}$, its temperature rises to t_{cw2}, therefore:

$$t_{\mathrm{cw2}} = t_{\mathrm{cw1}} + \frac{G_{\mathrm{wec}}}{G_{\mathrm{cw}}} \frac{(1-\gamma)}{(1-\beta)} \left(\tau_0 - \tau_{0,1}\right). \tag{6}$$

Special calculations have established that during the heating period its change is in the range of $t_{\mathrm{cw2}} = 50 \div 10$ ºC in real conditions of the water flow ratio in the heat exchanger 2 at $G_{\mathrm{cw}}/G_{\mathrm{wec}} \leq 1$.

Based on the previously established values t_{cw1} and t_{cw2} according to (5) and (6), a generalized dependence for determining the pre-cooling temperature of the used heat carrier $\tau_{0,1}$ after the heat exchanger 2 follows:

$$\tau_{0,1} = \tau_0 - \frac{(1-\beta)G_{\mathrm{cw}}}{(1-\gamma)G_{\mathrm{wec}}} (t_{\mathrm{cw2}} - t_{\mathrm{cw1}}). \tag{7}$$

It should be noted that for the design conditions of the heating period in the absence of recirculation (ρ = 0) should be provided temperature according to the thermal requirements for reliable operation of vapour-compression HPU.

From the equality of heat flows of utilized heat of exhaust gases and heated water for hot water supply follows the dependence for determining the appropriate temperature of its heating $t_{\mathrm{hw,eg}}$ in the heat exchanger 14, which is presented in the following form:

$$t_{\mathrm{hw,eg}} = t_{\mathrm{cw1}} + \frac{\mathrm{c_{p,eg}} G_{\mathrm{eg}} \left(t_{\mathrm{eg,b}} - t_{\mathrm{eg,e}}\right)}{\mathrm{c_p} \beta G_{\mathrm{cw}}}, \tag{8}$$

where $\mathrm{c_{p,eg}}$ – isobaric specific heat capacity of flue gases;

$t_{eg,b}$ and $t_{eg,e}$ – respectively, the initial and final temperature of flue gases.

On the basis of dependence (8) the conditions of heat utilization and rational ratio of heated water and flue gas flow rates with available potential in the reconstructed heat generating plants are determined.

The heat of cooling the used heat carrier and its recirculation part in the evaporator 3 is determined on the basis of the previously determined temperatures $\tau_{0,2}$ and $\tau_{0,3}$ in an appropriate form:

$$Q_v = c_p(1+\rho)(1-\gamma)G_{wec}(\tau_{0,2}-\tau_{0,3}). \tag{9}$$

The temperature of the cooled used heat carrier $\tau_{0,3}$ after evaporator 3 is generally determined on the basis of dependence (9) as:

$$\tau_{0,3} = \tau_{0,2} - \frac{Q_v}{c_p(1-\gamma)(1+\rho)G_{wec}}. \tag{10}$$

Note that the used heat carrier with the flow rate $(1-\gamma)G_{wec}$ and set temperature $\tau_{0,3}$ enters condenser 19 for subsequent reheating. As a result, the dependence for determining its temperature $\tau_{0,4}$ after condenser 19, taking into account the dependence $\tau_{0,3}$ according to (10), takes the following form:

$$\tau_{0,4} = \tau_{0,2} - \frac{Q_v}{c_p(1-\gamma)(1+\rho)G_{wec}} + \frac{Q_{c2}}{c_p(1-\gamma)G_{wec}}. \tag{11}$$

It is known that the energy efficiency of vapor-compression thermo transformers is determined by the actual conversion efficiency in the form of the energy flux ratio according to the dependence:

$$\phi = \frac{Q_c}{W_c}, \tag{12}$$

where Q_c – heat capacity of the condenser with total heat flux Q_{c1} and Q_{c2}in condensers 19, 20;
W_c – thermal equivalent of the drive power in the compressor operation with external heat pump drive, which can be represented as follows $W_c = Q_{c1} + Q_{c2} - Q_v$.

Consequently, the used heat carrier with temperature $\tau_{0,4}$ after the condenser 19 with the bypass flow rate γG_{wec} and temperature τ_0, enters the heat generator 15 with the temperature $\tau_{0,5}$, which is determined according to the dependence:

$$\tau_{0,5} = \gamma\tau_0 + (1-\gamma)\tau_{0,4}. \tag{13}$$

The corresponding dependences for determining the generated heat fluxes Q_{c1} and Q_{c2}in condensers 20 and 19, taking into account the mode parameters α, γ, β, relation $G = \frac{G_{cw}}{G_{wec}}$ and dependences for the temperature t_{cw2} (6), take the following form:

$$Q_{c1} = c_p\alpha G_{cw}\left(t_{hw} - t_{cw1} - \overline{G}\frac{(1-\gamma)}{(1-\beta)}(\tau_0 - \tau_{0,1})\right). \tag{14}$$

$$Q_{c2} = c_p(1-\gamma)G_{wec}(\tau_{0,4}-\tau_{0,3}). \tag{15}$$

Taking into account the above equations for determining the heat flows Q_v, Q_{c1}, Q_{c2} the dependence of the actual conversion factor $\phi$of HPU in the operation of the improved decentralized heat supply system after the appropriate transformations takes the form:

$$\varphi = \left(1 - \frac{(1-\gamma)(1+\rho)(\tau_{0,2}-\tau_{0,3})}{\alpha\overline{G}\left(t_{hw} - t'_{cw} - \frac{1}{\overline{G}}\rho(1-\gamma)(\tau_{0,rec}-\tau_{0,3}) - \frac{1}{\overline{G}}\frac{(1-\gamma)}{(1-\beta)}(\tau_0 - \tau_{0,1})\right) + (1-\gamma)(\tau_{0,4}-\tau_{0,3})}\right)^{-1}. \tag{16}$$

For graphical presentation of the research results, the temperature of initial cold water $t_{cw}^{'}$ was determined depending on the change in the outdoor air temperature during the heating period of the year [5], and the current temperature of the used heat carrier τ_0 was determined according to the dependence of operational regulation of decentralised heat supply systems in its design range $(\tau_r - \tau_0)$ = (95 – 70) ºC. The justified values of parameters were considered as regime conditions:

$$\overline{G}=0.1 \div 1.5,\ \alpha = 0.1 \div 0.15,\ \beta = 0.05,\ \gamma = 0.0.$$

Fig. 2 shows the dependence of the actual conversion coefficient φ at the change of ambient temperature t_a' during the heating period and the corresponding change of the used heat carrier temperature τ_o according to the generally accepted schedule of operational regulation of local heating systems.

The increase of the conversion coefficient φ in the heating period is noted in conditions of the outdoor air temperature t_a' decreasing with the temperature τ_0 of the used heat carrier from the heat network increasing at partial utilisation of its energy potential as a low-potential source.

At the same time, it is characteristic that the decrease in the consumption of cold water for the general purpose G_{cw} is positively reflected in the objective reduction of its consumption during this period for heat-technological use when the outside air temperature decreases.

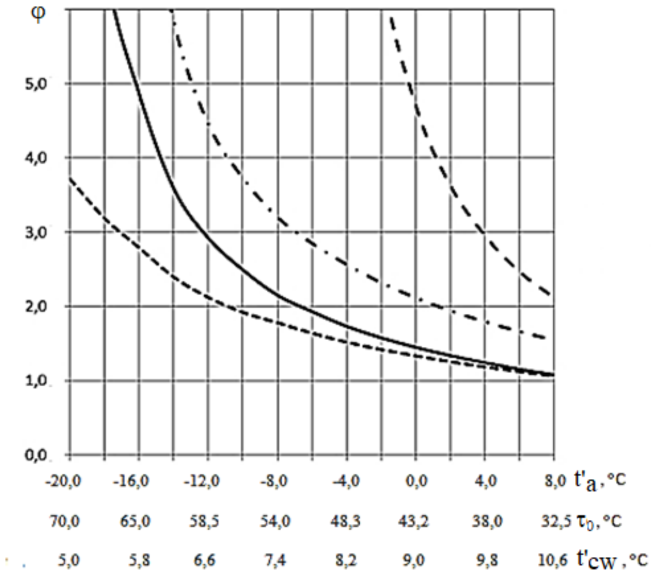

Fig. 2. **Dependence of the actual conversion coefficient φ on the change of the ambient temperature t_a', ºC in the process of operational regulation of the combined heat supply system with HPU at β=0.05, γ=0.0**
– – – – – at $\tau_{0,4} = 45$ ºC, $\alpha = 0.1$, G_{cw}/G_{wec}=0.25; – · – · – – at $\tau_{0,4} = 55$ ºC, α = 0.1, G_{cw}/G_{wec}= 0.25;
——— – at $\tau_{0,4} = 45$ ºC, $\alpha = 0.1$, G_{cw}/G_{wec} = 1.0; -------- – at $\tau_{0,4} = 55$ ºC, $\alpha = 0.1$, G_{cw}/G_{wec} = 1.0

It follows from the graphs that determination of the intermediate temperature in the two-stage scheme of heat generation is of a search character in order to increase the energy efficiency of HPU operation and to increase the total capacity of the generated heat.

Thus, it follows from the structural and functional construction of the combined system and the results of the analytical study that the withdrawn heat flux Q_v in the process of cooling the used heat carrier in the evaporator 3 to the temperature $\tau_{0,3} = 5$ ºC is transferred by the condenser at the inlet to the heat generator with an increase in its power determined by the value of the actual conversion coefficient φ of energy flows in HPU.

In this case, according to the proposed two-stage scheme, the total capacity of generated heat is increased during the heating period with the range of operating temperatures of the heat carrier in the heat supply system operation extended up to $\left(\tau_r - \tau_{0,3}\right)$ = (95 – 5) ºC. For the established capacity of the generated heat, the total flow rate of the circulating heat carrier and the corresponding energy costs are reduced.

Conclusions

1. For the developed combined heat supply system with a two-stage heat generation scheme, a generalised dependence for determining the actual conversion coefficient of HPU based on the integrated potential of aftercooling of the used heat carrier, initial cold water and flue gases has been established. It allows qualitative analysis of the heat supply system efficiency depending on the energy potential of low-temperature sources, as well as the ratio of water flows of structural subsystems, taking into account the influence of distribution of initial cold water for municipal and industrial-technological purposes.
2. It is established that the value of the actual conversion coefficient significantly depends on the value of the intermediate temperature of the used heat carrier in the two-stage scheme after the condenser at the inlet to the heat generator. Its lower temperatures $\tau_{0,4}$ provide a significant increase in the conversion coefficient at increasing water consumption for domestic hot water supply in the heating period.
3. The results of the study established that the search value of the intermediate temperature of vapour-compression afterheating of the used heat carrier at the inlet to the heat generator should be based on its complex optimisation, taking into account the actual temperature difference between the condensation of the working body during heating of the subscriber coolant and its boiling in the HPU evaporator.
4. The results of the analytical study of the improved heat supply system create a basis for its engineering development with regulation of the justified intermediate temperature of the heat carrier in the two-stage heat generation scheme.

Author contributions

Conceptualization, P.V. and B.V.; methodology, P.V. and B.V.; formal analysis, P.V., B.V., S.L.; investigation, P.V. and B.V., writing – original draft preparation, S.L. and G.E.; writing – review and editing, P.V., B.V., G.E.; project administration, P.V. All authors have read and agreed to the published version of the manuscript.

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