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USAGE OF THE HEAT OF FLUE COMBUSTION PRODUCTS OF THE FUEL IN HEAT PUMPING INSTALLATIONS

Energy efficiency of flue gases heat from power generating boilers in heat pumping installations has been analyzed.

Key words: *evaporator, compressor, condenser, boiler, contact recovery boiler.*

Introduction

One of the efficient means of fuel economy and protection of the environment is the usage of low-temperature secondary sources of energy. In developed countries much attention is paid to creation and introduction of heat pumping installations (HPI), intended for heating, hot water supply, drying, etc. As it was mentioned in [1], positive experience of foreign countries, regarding the usage of HPI in the systems of heating and hot water supply does not function in natural-climatic conditions of Ukraine. Modern state, future trends of development and problems of heat pumping technologies application of heat supply systems are described in [2]. Introduction of HPI in heat supply systems depends first of all on the availability of the sources of low temperature heat. For natural-climatic conditions of Ukraine the usage of low-temperature sources of heat in the form of atmospheric air and surface waters is rather limited and is possible partially in southern regions. In other parts of Ukraine sewage waters, soil and ground waters can be used as low-temperature sources of heat [1, 2]. Sewage waters discharge is limited and requires purification. Proceeding from the experience [1], the expenditures for the construction of ground heat exchangers constitute up to 70 % of total capital investments in heat pumping systems. Under such conditions the search of other sources of low-temperature heat is of great importance.

In many towns and cities of Ukraine numerous boiler houses, equipped with boilers of small capacity, built in 60 – 80s of the last century function. Such boiler houses operate on natural gas, do not have tail surfaces and are characterized by relatively high, for our conditions, temperature of flue gases ($t_{fq} \geq 130\ °C$). Combustion products of natural gas (stack gases) contain increased concentration of water vapour, certain part of combustion fuel heat is spent on the formation of this vapour. Using contact recovery boilers (CRB), the temperature behind them, t_{crb}, can be reduced to the temperature, which is less than dew point temperature t_R, i.e. $t_{crb} < t_R$. This allows to use both physical and condensing components of flue gases. In [1] it is noted, that temperature of cooled gases has certain optimal value, since in case of t_{crb} reduction, the temperature of the cooling water decreases and expenses for HPI compressor drive grow. But in [1] the analysis of the changes of operation efficiency indices of combined installations, comprising boilers, CRB and HPI is not performed.

Hence, the task of the given research was to analyze the operation efficiency of boiler houses with flue gases heat utilization in contact recovery boilers and heat pumping installations.

Main result

Schematic diagram of the installation is shown in Fig. 1.

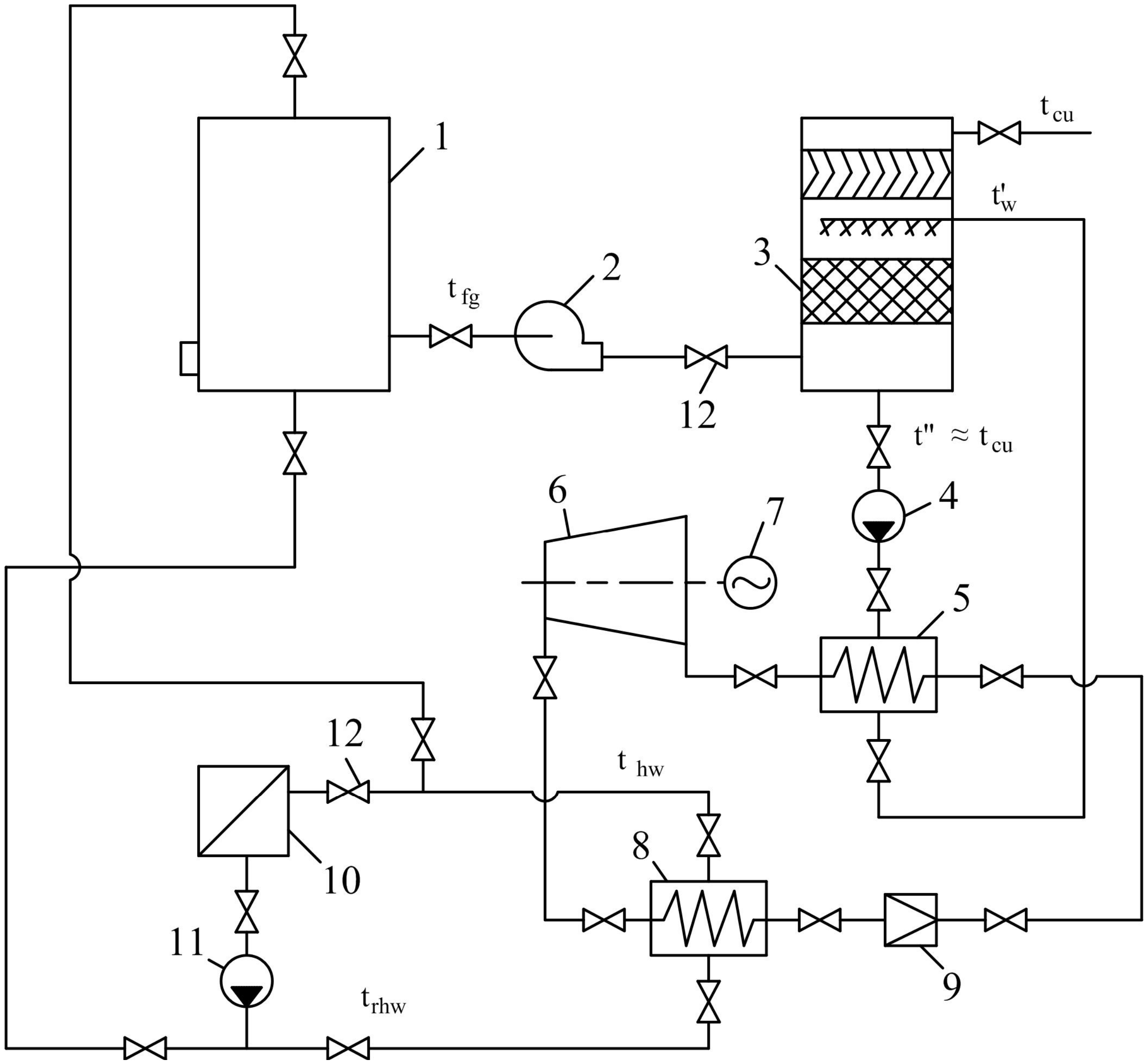

Fig. 1. Schematic diagram of utilization of flue gases heat from the boiler by means of recovery boiler and heat pumping installation: 1 – hot water boiler; 2 – smoke exhauster; 3 – contact recovery boiler; 4 – circulating pump; 5 – HPI evaporator; 6 – compressor; 7 –electric motor; 8 – HPI condenser; 9 – throttle valve; 10 – heat load; 11 – networking pump;12 – accessories

Flue gases from the boiler at the temperature t_{fq} enter contact recovery boiler and are cooled to the temperature t_{cu} by circulation water. Utilized heat with water arrives to the evaporator of HPI; where it evaporates the working medium of HPI. At the expense of the work, performed by the compressor, certain amount of thermal capacity, supplied to the consumer, is removed from HPI compressor.

For the preset thermal capacity of the boiler Q_b consumption of operating and equivalent fuel equals:

$$B_{ob} = \frac{Q_b}{Q_l^o \cdot \eta_b}; \; B_{cb} = \frac{Q_b}{Q_{lc}^o \cdot \eta_b}, \tag{1}$$

where Q_l^o and Q_{lc}^o – is combustion heat of operating and equivalent fuel, correspondingly; η_b – is efficiency factor of the boiler.

Thermal capacity, recovered in CRB at the expense of “dry” heat exchange

$$Q_d = B_{ob} \cdot Q_l^o \cdot (1-\eta_b) \cdot \psi_d = Q_{fuel}(1-\eta_b) \cdot \psi_d , \tag{2}$$

where $Q_f = B_{ob} \cdot Q_l^o = B_{cb} \cdot Q_{lc}^o$ – is thermal capacity of the fuel; $\psi_d = (t_{fg} - t_{crb})/ t_{fg}$ – is the coefficient of "dry" heat exchange recovery.

Thermal capacity, released in CRB at the expense of water vapour condensation [3]

$$Q_{wv} = B_{ob} \cdot (Q_h^o - Q_l^o) \cdot \psi_{wv} = B_{ob} \cdot Q_l^o (Q_h^o / Q_l^o - 1) \cdot \psi_{wv} = Q_f (Q_h^o / Q_l^o - 1) \cdot \psi_{wv} , \tag{3}$$

where Q_h^o – is highest fuel combustion heat, that exceeds the value of Q_l^o by the water vapour condensation heat; $\psi_{wv} = (t_R - t_{crb})/t_R$ – is the coefficient of water vapour heat recovery.

Dew point temperature t_R is calculated by [4] or [7]. Relation Q_h^o / Q_l^o for different fuels is given in [1].

Total thermal power, recovered in CRB

$$Q_{crb} = Q_d + Q_{wv} = Q_f \left[(1-\eta_b) \cdot \psi_d + (Q_h^o / Q_l^o - 1) \cdot \psi_{wv}\right] = Q_f \cdot A . \tag{4}$$

This power is directed into HPI evaporator, power of which equals

$$Q_{ev} = Q_{crb} \cdot \eta_{he} = Q_f \cdot A \cdot \eta_{he} = Q_f \cdot A_1 , \tag{5}$$

where η_{he} –is efficiency factor of heat exchanger .

Thermal capacity of HPI condenser

$$Q_{cd} = \frac{Q_{ev} \cdot \varphi}{(\varphi - 1)} = \frac{Q_f \cdot A_1 \cdot \varphi}{(\varphi - 1)} = Q_f \cdot C \cdot \varphi , \tag{6}$$

where φ – is thermal (heating)coefficient of HPI, that depends on the temperatures in the evaporator T_{ev}, in the condenser конденсаторі T_{cd} and efficiency factor of the compressor .

The value φ is determined either as a result of construction of operation process of HPI on P-h diagram or by the relations in [5].

Necessary power of HPI compressor electric drive.

$$N = \frac{Q_{cd}}{\varphi \cdot \eta_{em}} = \frac{Q_f \cdot C \cdot \varphi}{\varphi \cdot \eta_{em}} = Q_f \cdot C_1 , \tag{7}$$

where $C_1 = C/\eta_{em}$; η_{em} – is electromechanical efficiency factor of electric drive.

Equivalent consumption of conventional fuel per electric drive of the compressor

$$B_{ec} = \frac{N}{Q_{lc}^o \cdot \eta_{ps} \cdot \eta_{en}} = \frac{B_{cb} \cdot C_1}{\eta_{ps} \cdot \eta_{en}} , \tag{8}$$

where η_{ps} *and* η_{en} – mean value of the efficiency factors of electric stations and electric networks in energy system, correspondingly, which are determined by the data, taken from statistical year books of Ukraine [6].

Total consumption of equivalent fuel in the suggested installation

$$B_{gc} = B_{cb} + B_{ec} = B_{cb}\left[1 + \frac{C_1}{\eta_{ps} \cdot \eta_{en}}\right]. \tag{9}$$

Total thermal capacity, generated in the installation

$$Q_g = Q_b + Q_{cd} = B_{cb} \cdot Q_{lc}^o (\eta_b + C \cdot \varphi) . \tag{10}$$

Specific consumption of equivalent fuel for generation of thermal capacity, kg/GJ

$$b_c = \frac{B_{gc} \cdot 10^3}{Q_g} = \frac{10^3}{Q_{lc}^o}\left[\frac{1 + C_1/(\eta_{ps} \cdot \eta_{en})}{\eta_b + C\varphi}\right]. \tag{11}$$

If aggregate thermal capacity is generated in the boiler specific consumption of equivalent fuel would be, kg/GJ

$$b_c^b = \frac{10^3}{Q_{lc}^o \cdot \eta_b}. \tag{12}$$

The difference between specific consumptions of equivalent fuel, kg/GJ

$$\Delta b_c = b_c^b - b_c = \frac{10^3}{Q_{lc}^o}\left[\frac{1}{\eta_b} - \frac{1 + C_1/(\eta_{ps} \cdot \eta_{en})}{\eta_b + C\varphi}\right]. \tag{13}$$

The saving of equivalent fuel at the expense of utilization of flue gases heat in CRB and HPI, t/h

$$\Delta B_c = \Delta b_c \cdot Q_{cd} \cdot 3.6 \cdot 10^{-3}. \tag{14}$$

Heating water consumption, kg/sec:

– in hot water boiler

$$G_{wb} = \frac{Q_b \cdot 10^3}{Cp_b (t_{dhw} - t_{rhw})}, \tag{15}$$

– in the condenser of HPI

$$G_{wb} = \frac{Q_{cd} \cdot 10^3}{Cp_{cd} (t_{wb} - t_{rhw})}, \tag{16}$$

where t_{dhw} and t_{rhw} – is the temperature of direct and return heating water, correspondingly; t_{cw} – is the temperature of the water at the output of the condenser; Cp_b and Cp_{cd} – is isobar heat capacity of water for certain temperatures.

The temperature of heating water in power and heat supply system, ºC

$$t_{hw} = \frac{G_{wb} \cdot Cp_b \cdot t_{dhw} + G_{cd} \cdot Cp_{cd} \cdot t_{cd}}{G_{wb} + G_{cd}}. \tag{17}$$

By means of variational calculations the operation efficiency of the combined installation with CRB and HPI, based on hot water boiler ПТВМ-30, rated for 34.8 MW, temperature of flue gases 135 °C and operation temperature of heating network 100/50 °C was investigated. Operating fuel – natural gas with the heat of combustion $Q_l^o = 33.4$ MJ/m^3 and relation $Q_h^o / Q_l^o = 1.136$. Efficiency factor of the boiler – 0.915. Efficiency factors of electric stations and electric network were 0.34 and 0.9, correspondingly [6]. Working medium of HPI-R717 (ammonia), efficiency factor of HPI compressor – 0.84. The value of under heating in the condenser and evaporator of HPI – 3 °C.

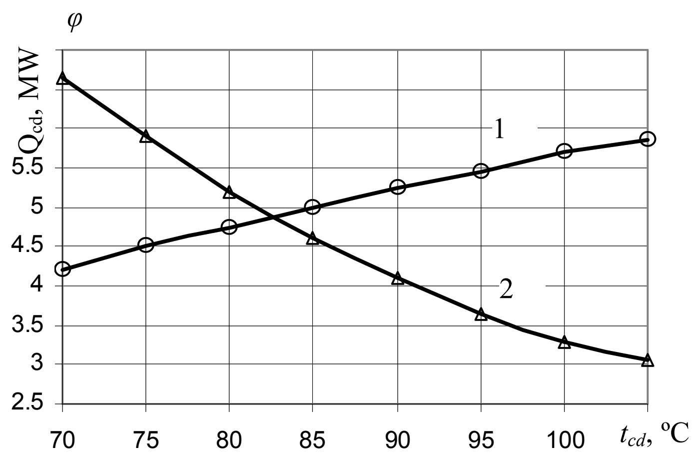

Fig. 2. Character of HPI condenser capacity and heating coefficient change

Fig. 2. shows the characteristic changes of HPI condenser power (curve 1) and heating coefficient φ (curve 2), depending on the temperature of cooling medium in the condenser for $t_R = 65\,°C$, $t_{crb} = 42\,°C$, $t_{ev} = 30\,°C$.

For the given conditions, the greatest impact the temperature t_{crb} exercises on heating coefficient, decreasing its value 2.2 times. The power of recovered heat in HPI increases only 44 %.

Dependences of HPI compressor power and relative power of heating pump are given in Fig. 3. Here non-dimensional quantity N^* characterizes relative increase of heating pump capacity, as a result of additional pumping of heating water across HPI condenser.

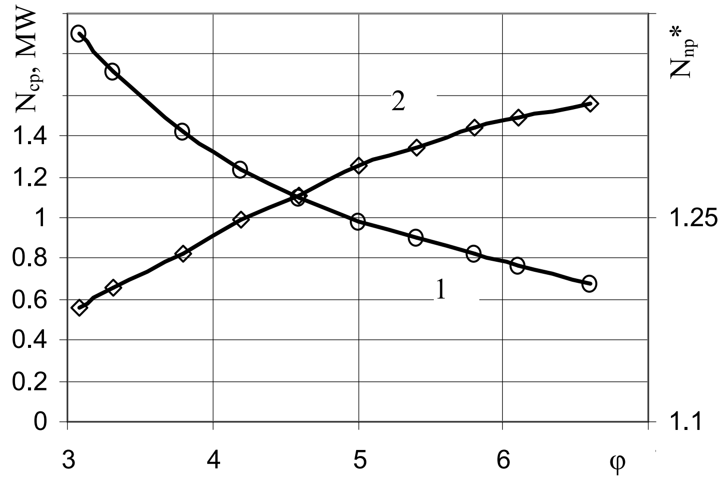

Fig. 3. Current values of HPI compressor capacity (curve 1) and relative power of the heating pump (curve 2)

It is seen in Fig. 3, that within the given range of φ change, HPI compressor capacity decreases almost three tines, and power of heating pump increases 33 %.

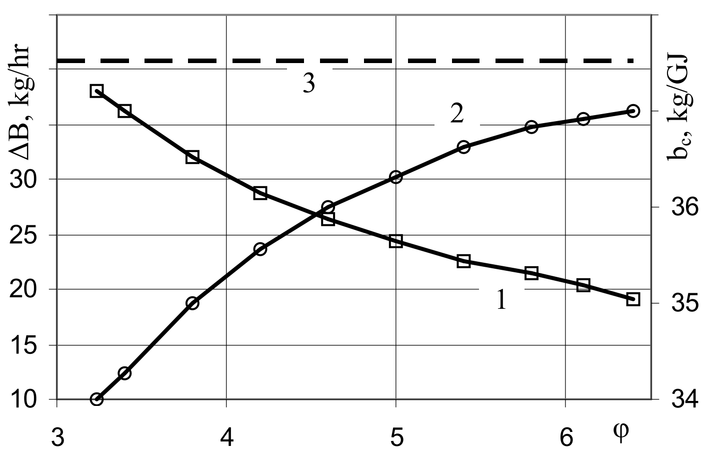

Fig. 4. Dependences of specific consumption of equivalent fuel for heat generation (curve 1) and value of equivalent fuel saving ΔB, кg/h (curve 2)

Fig. 4. shows current values of specific consumption of equivalent fuel and saving of equivalent fuel in the suggested installation. Here dash line 3 characterizes specific consumption of equivalent fuel in the boiler. The value of specific consumption of equivalent fuel, which characterizes the efficiency of fuel usage due to utilization of flue gases heat in contact recovery boiler and HPI, decreases by 7.2 %. In the given range of φ change the saving of equivalent fuel can be within 10 – 40 кg/hr.

Calculations showed, that the temperature of the heating water, that is the mixture of water from the boiler and HPI condenser, decreases with the decrease of φ and is 99 °C if $\varphi = 6.1$ and 94 °C if $\varphi = 3.2$. As a result of calculations performed it was established that for each value of the dew point temperature there exists certain interval of optimal values of gases temperatures behind the contact recovery boiler. The Table contains, as an example, main operation indices of the installation for $t_R = 55$ °C and cooling medium temperature in the condenser $t_{cd} = 80$ °C .

Indices	Temperature of gases behind the contract recovery boiler, ºC				
	23	28	33	38	43
Thermal capacity as a result of "dry" heat exchange in CU, MW	2.68	2.56	2.44	2.32	2.2
Thermal capacity as a result of water vapour condensation in CU, MW	2.98	2.52	2.05	1.58	1.12
Thermal capacity of HPI evaporator, MW	5.55	4.98	4.49	3.83	3.25
Heating coefficient	3.43	3.62	3.95	4.22	5.25
HPI compressor capacity, MW	2.28	1.9	1.488	1.19	0.764
Thermal capacity of HPI condenser, MW	7.83	6.88	5.976	5.02	4.01
Specific fuel rate of equivalent fuel, кg/GJ	36.4	36.2	35.49	35.9	36.3

Value of specific consumption of equivalent fuel b_c characterizes the efficiency of fuel usage in the installation [3, 4]. It is seen from the Table, that for $t_R = 55$ °C optimal values of t_{crb} are within the limits of 32.5 – 33.5 °C . The dependence of optimal values of gases cooling temperature in contact recovery boiler on dew point temperature is given in Fig. 5.

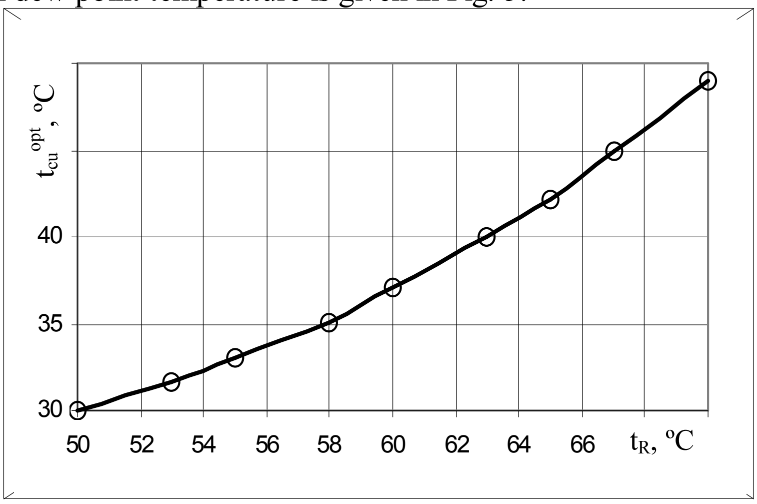

Fig. 5. Values of optimal cooling temperature of flue gases in contact recovery boiler

The obtained results are necessary preconditions for the usage of flue gases heat in boiler houses.

Conclusions

1. The relations for determination of operation indices of the installation with the recovery of fuel flue combustion products heat have been obtained.
2. It is revealed that the application of heat contact recovery boilers together with heat pumping installations enables to increase the efficiency of fuel usage by 7.2% and save equivalent fuel consumption up to 40 кg/hr.
3. Application of heat pumping installations stipulates the increase of heating pumps power from 10 to 30%.
4. Optimal temperatures for cooling the combustion products in contact recovery boilers of heat are defined.

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