Multisection heat exchangers for heat utilization of the waste gases from heat power plants

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Abstract. Improving the efficiency of fuel and energy resources in operating heat power plants is an important and relevant direction in the engineering development of various industries. A comprehensive solution to the problems arising as a result can be achieved, including the use of a step-by-step heat exchange process, which can be implemented in multisection heat exchangers for the utilization of the waste gases heat.

Keywords: heat power plant, wasteheat exchanger, step-by-step heat exchange, fuel economy

1. Introduction

Reducing energy and material costs for the widest range of heat power plants (HPP) has been a relevant and important goal of various studies [1], [2], [3]. At the same time, the perfection of energy conversion in them is largely determined by the efficiency of processes implementing compression, cooling or heating of working environments [4], [5]. The use of recuperation and utilization heat exchange equipment provides the required heat exchange, reduces the energy loss of the waste gases [6] and the cost of thermal energy [7].

Increasing the efficiency of heat exchangers for heat recuperation or utilization of technological flows requires the formation of new processes, including step-by-step heat exchange, providing the most complete and comprehensive solution of resource saving problems [6].

Heat power plants include elements for pressure (compression, expansion) and temperature (heat exchange systems and devices) changes, carrying out the necessary energy conversion (exergy) in accordance with the functional purpose.

One-step process of heat exchange between the flows of working and subsidiary environments often does not provide a given level of increase in efficiency of the plant. This is due to a number of difficulties: additional losses along the path of the utilized environment movement on increasing the temperature level of the heat-using flow [6]; for certain modes of the heat exchanger operation the need for change in the ratio of thermal equivalents A and, consequently, its main purpose (heater $A\geq1$, cooler $A<1$); reducing the possibility of regulation and rational use of environment natural cold.

The complex solution of the noted problems is reached on the way of application of step-by-step heat exchange process which can be implemented in the multisection heat exchange equipment.
2. Problem statement
Thus, the study subject in the general case is multisection wasteheat exchangers (MWE), converting the heat of technological flows of heat power plants, among which can be distinguished:
– two-section wasteheat exchangers (WE), using the waste gases temperature after the gas turbine combustion chamber to heat the compressed air in order to improve the efficiency of gas turbine plant (GTP) (recuperation) and supply heat (utilization) for heating the delivery water (Fig. 1a);
– two-section WEs for heating air in front of the compressor in order to stabilize the GTP characteristics in winter (regulation) and heat the delivery water;
– two-section WEs for heating air in front of the compressor and compressed air in front of the combustion chamber;
– three-section WEs for heating air in front of the compressor, compressed air in front of the combustion chamber and the delivery water (Fig. 1b).

![Figure 1. Multisection heat exchangers for waste gas heat utilization](image)

It is necessary to assess the efficiency of multisection heat exchangers for the utilization of the waste gases heat in general and their individual design solutions, which can be carried out according to a number of local and global criteria.

3. Theory
In general, to solve the economic problem of reasonable choice of multisection wasteheat exchanger in accordance with the global criterion, it is necessary to determine the cost of WE life cycle [4].

For example, for a two-section heat exchanger (Fig. 1a) with determining the optimum ratio of the heat load for each section of WE (Qair, Qwater) and thus the calculation mode of optimum heat exchange surface areas of WE sections (Fair, Fwater) this criterion can be represented in the form:

\[
C_{le}^{WE} = K (F_{air}, F_{water}) + \sum_{i=1}^{T_{calc}} \left( C_{WE} + C_{repair} + C_{insp} \right) / (1 + E)
\]

where \( K \) is capital costs; \( C_{WE} + C_{repair} + C_{insp} \) are annual operating costs for the drive of WE superchargers, planned repairs, inspections; \( E \) is rate of return; \( T_{calc} \) is calculation period.

As local criteria there can be used: exergy efficiency of the plant \( \eta_x \), the function of the heat efficiency, the thermal efficiency of the heat exchanger \( \eta_{UT} \), the electricity consumption connected with the pumping of the heat carriers along the heat exchanger paths \( N_{elb} \), its cost \( C_{Nelb} \), the payback period \( T_{pay} \) etc.

At the same time, providing the greatest efficiency of the studied MWE, in turn, involves the solution of the following issues: determining the composition and sequence of the stages of recuperation and utilization; finding the best combination of weight fractions of each stage, choosing the composition of heat-sensing environments, modes and ways to maintain their temperature levels.
The most important tool in this way is to represent MWE in the form of a heat exchange system (HES) and to assess the influence of various factors on the intensity of heat exchange processes in it. In general, for HPP MWE with the approach given in [8], it can be written:

$$Q_{MWE} = \sum_{j=1}^{N} (k_j F_j \Delta T_j)$$

(2)

where $Q_{MWE}$ is the amount of heat transmitted by the waste gases to a given combination of receiving heat exchanging environments; $k_j$, $F_j$, $\Delta T_j$ is heat transfer coefficient, the surface area of heat exchange, the log mean temperature difference of the MWE $j$-th section.

Then, taking into account $Q_{MWE} = \sum Q_j^U$ for the maximum possible increase in the exergy efficiency of the HPP from the use of the waste gases heat, we obtain

$$\Delta \eta_e^U = \frac{\sum_{j=1}^{N} Q_j^U \cdot \bar{\tau}_e}{N_{HPP}} = \frac{\sum_{j=1}^{N} (k_j F_j \Delta T_j) \cdot \bar{\tau}_e}{N_{HPP}}$$

(3)

where $Q_j^U$ is the amount of the waste gases heat used in the MWE $j$-th section, implementing a phased heat exchange process; $N_{HPP}$ is power consumption of compressors, fans and pumps of HPP; $\bar{\tau}_e$ is the average exergy flow temperature of the $j$-th section, using the waste gases heat.

Thus, $Q_{MWE}$ and $\Delta \eta_e^U$ are represented as functions of {$k_j$, $F_j$, $\Delta T_j$}, which determines the need to assess their impact on the intensity of heat transfer processes in MWE. Or taking into account the structural hierarchy of the object:

– at the system level – the study of the effect on the efficiency of heat exchange relationships between individual sections (apparatus) and the distribution of heat load between them {$F_j$};

– at the hardware level – the study of heat exchange processes for various schemes of environments interaction in heat exchangers (sections) when their temperature {$\Delta T_j$} and flow rate change, as well as the effect of changes in thermal efficiency in individual sections on the work of MWE as a whole;

– at the element level – the study of the change effect in heat transfer conditions {$k_j$} on the intensity of heat exchange in the heat exchanger paths, including the use of new heat transfer surfaces.

In similar way to [8] at the system level we estimate the possibility of increasing the HPP exergy efficiency due to step-by-step heat exchange in the MWE in terms of reducing the exergy losses. Consequently, the losses of exergy from irreversibility due to the finite temperature difference can be defined as:

$$D_T = \sum \delta q \cdot \Delta \tau_e$$

(4)

where $\delta q$ is the element of heat transferred on the site; $\Delta \tau_e = \tau_e^G - \tau_e^R$ is the difference between the exergy temperatures of the elementary site for the "$G$"– giving and "$R$" – receiving heat exchanging
environments; \( \tau_e = 1 - \frac{T_{am}}{T} \) is exergy temperature function; \( T_{am}, T \) is ambient temperature and the current temperature of the heat carrier.

In a one-step process of WE heat exchange for counter-current circuit and an infinitely large heat transfer coefficient \( k \to \infty \) exergy loss \( D_e \), using the heat of the giving environment flow (waste gases) from \( T_{G \text{ in}} \) to \( T_{G \text{ fin}} \) can be defined as the area of the figure bounded by lines 1-2 and 2-3, marked on the \( q, \tau_e \) diagram (Fig. 2).

The implementation of sequential, step-by-step use of the waste gases heat initially in the air section (reducing the temperature of the giving environment to the value \( T_{G \text{ fin a}} \) – lines 1-4 and 4-6, and then in the water section (reducing the temperature of the giving environment to the value \( T_{G \text{ fin w}} = T_{G \text{ fin}} \) – lines 4-2 and 2-5 determines the possibility of a tangible reduction in the total exergy losses \( D_e < D_{e a} + D_{e w} \).

**Figure 2.** Exergy losses from irreversibility due to finite temperature difference for one-step and two-step heat transfer process in WE

Also, as it can be seen from the diagram, the use of various heat-using flows in the air and water sections as receiving environments can significantly increase their exergy temperature at the output of each section \( \tau_e(T_{R \text{ fin a}}) > \tau_e(T_{R \text{ fin w}}) > \tau_e(T_{R \text{ fin}}) \).

The heat efficiency of the multisection wasteheat exchanger \( \eta_{MWE} \) expresses the ratio between the actual transferred amount of heat \( Q_{MWE} \) and the maximum possible amount of heat \( Q_{MAX} \) that can be transferred in an ideal heat exchanger.

\[
\eta_{MWE} = \frac{Q_{MWE}}{Q_{MAX}} = \frac{\sum_{j=1}^{N} k_j F_j \Delta T_j}{(Gc)_{MIN} \cdot \Delta T_{MAX}}
\] (5)

where \((Gc)_{MIN}\) is the minimum of the mass flow rate for the heat capacity of the heat exchanging environment, \( \Delta T_{MAX} \) is the maximum difference between the initial and final temperature of the environments.
When using a part of the waste gases heat (recuperation, utilization), less fuel is consumed in the heat power plant. As a result, its efficiency changes $\eta_{HPP}$:

$$\eta_{HPP} = \frac{Q^U}{Q_{LOW}^F \cdot G_F - Q_{WG} \cdot \eta_{MWE}}$$

(6)

where $Q_{LOW}^F$ is the lowest heat of fuel combustion; $G_F$ is fuel consumption for the operation of the plant without regenerative use of heat; $Q_{WG}$ is the amount of the waste gases heat.

For two-section waste heat exchanger (Fig. 1a) determining the optimal ratio of the heat load for each section can be carried out taking into account the dominant parameter – the temperature of the compressed air after the air section $t^{opt}_{air}$. Then the optimization problem for the WE structural and compositional parameters – to determine the optimal heat exchange surface areas of its air and water sections $F = \{F_{air}, F_{water}\}$ can be formulated in the following way:

$$\varphi^* = \text{opt}_U \varphi\left(t^{opt}_{air}, F, t_c, p_c, t_a, t_w\right)$$

$$U = \{F_{air}, F_{water}\} = \{t^{opt}_{air}\}$$

(7)

where $t_c, p_c$ is the temperature and pressure of compressed air entering the WE from the compressor, $t_a$ is the air temperature, $t_w$ is the temperature of the delivery water entering the WE water section.

4. Experiment results

Numerical studies of two-section WEs (Fig. 1a) were carried out for gas turbine unit GTK-10-4.

For the regenerating (air) section, the calculations were carried out in the compressed air temperature range of 350...507°C, corresponding to the degree of regeneration from 0.5 to 1 (1 is the theoretical limit, 0.5 is the efficiency limit). The calculation results are shown in Fig. 3.

![Figure 3](image)

**Figure 3.** Evaluation of the effect of the heated compressed air temperature on the GTU power loss (a) and the growth of heat capacity, taking into account the corresponding increment of the heat exchange surface (b)

The air temperature rise at the outlet of the regenerating (air) section is directly related to the heat exchange surface increment. This leads to the hydraulic resistance increase along the GTU gas-air
path. Thus, there are additional power costs to overcome the resistance. The results of numerical studies stated in the graph (Fig. 3a) show that a significant increase in power losses is achieved at temperature of 470°C and above. The relative power losses in the range of compressed air temperatures 350...507°C are 1.5...55%.

Despite a further increment in the heat exchange surface ΔF, the temperature rise of the heated air at the outlet of the regenerating section results in a decrease in the growth of the heat capacity ΔQ (Fig. 3b). Thus, the temperature rise t_air above 460°C due to further increase in the surface area is impractical, due to the significantly outrunning growth of the necessary costs.

5. Results and discussion

Figure 4 reflects changes in fuel economy φ while maintaining the GTU power at a given level – φ, taking into account power losses – φ₁, as well as the growth rate function of relative fuel economy taking into account power losses to overcome hydraulic resistances – ϕ₂.

According to the data obtained, the optimum temperature \( t_{\text{air}}^{\text{opt}} \) of the air heated in the air section for a given gas turbine unit (GTK-10-4) is 460°C.

![Figure 4. Relative fuel economy: \( \phi \) – while maintaining GTU power at a given level; \( \phi_1 \) – with power loss; \( \phi_2 \) is the growth rate function of relative fuel economy taking into account power losses](image)

It is possible to define an optimum ratio of thermal loads and the areas of heat-exchange surfaces of MWE two sections for the obtained optimum compressed air temperature after the regenerating (air) section in accordance with the data presented in Fig. 3, Fig. 4 and taking into account the requirements to the heated delivery water. The air section accounts for 75...85% of the heat load and the surface of the two-section wasteheat exchanger.

6. Conclusion

A comprehensive solution to the problems of increasing the efficiency of heat exchangers for the utilization of the waste gases heat of heat power plants can be achieved by applying a step-by-step heat exchange process implemented in multisection heat exchange equipment.

The efficiency assessment of multisection wasteheat exchangers in general and their individual design solutions can be carried out according to a number of local and global criteria, including: life cycle cost; exergy efficiency; heat efficiency function; thermal efficiency; electricity consumption connected with the pumping of heat carriers along the heat exchanger paths; the cost of the consumed electricity; payback period, etc.
The analysis of the regenerating (air) section of the two-section WE (Fig. 1a) for gas turbine unit GTK-10-4, conducted in the compressed air temperature range of 350...507°C, showed that the temperature rise \( t_{air} \) above 460°C due to an increase in the surface area is impractical, because of the significantly outrunning growth of the necessary costs.

The optimal ratio of heat loads and the areas of the heat exchange surfaces of MWE two sections is 75...85% for the regenerating section and 15...25% for water one.

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