Thermal efficiency evaluation for multisection heat exchange equipment in heat power installations

I A Yanvarev
Omsk State Technical University, 11, Mira Ave., Omsk, 644050, Russia

E-mail: iayanvarev@mail.ru

Abstract. Comprehensive approach to improving heat exchangers for heat recovery and utilization of heat power installations is achieved by applying a step-by-step heat exchange process to be implemented in multi-section heat exchange equipment. Determination of its optimal structural, design and layout parameters can be carried out taking into account various criteria, including the use of the thermal efficiency function. Expressions for the thermal efficiency function of the objects under study are obtained in the form of complex, non-uniform heat exchange systems, on the basis of which the method for evaluating their execution options is developed. Optimal sequences of heat exchange sections, heat load distribution and operating modes for this equipment are determined.

Keywords: heat power installation, multi-section heat exchanger, heat exchange system, thermal efficiency

1. Introduction
Heat power, thermal engineering, heat and other installations and productions are the largest consumers of various fuel and energy resources. A distinctive feature of these installations is the implementation of technological processes with the release of a significant heat amount. Therefore, the tasks of rational energy recovery [1], utilization of secondary thermal energy resources [2] are of particular importance. Their solution will provide a significant economic effect [3].

Within these technologies the use of more advanced compressing, cooling, or heating processes for working media [4], [5] and the corresponding modern recovery and utilization heat exchange equipment [6], [7] reduces the energy loss of waste gases [8], the cost of heat energy [9], and the required increase in thermal efficiency [10].

The implementation of recovery and utilization involves the use of different working media [8], [10]:
– air working medium (compressed air before the combustion chamber of the gas turbine plant (GTP), atmosphere before GTP compressor, air used in the oil coolers of engines and blowers in gas-pumping units);
– gas working media (flue gases from the combustion chamber or after the GTP gas turbine, steam and steam-gas flows of the regeneration systems for steam turbine (STP) and combined-cycle (CCP) plants);
– liquid working media (delivery water in the heat supply systems of the compressor station, feed water, the main condensate from STP and CCP regeneration systems, water solutions of glycols, oils).

To implement the phased process of heat exchange improves the efficiency of heat exchange equipment for cooling the compressed gas flow [6], heat recovery or utilization for technology flows [8] of heat power installations (HPI), the most complete approach to resource saving [10].
The use of multi-section gas coolers for compressor units (CU) with utilization of compression heat [9], multi-section heat exchangers for heat utilization of waste gases from heat power installations [10] determines a comprehensive approach to improving energy conversion in these installations, increasing their thermal and energy efficiency.

2. Problem statement

Thus, the study subject in general is multi-section heat exchange equipment (MHEE). It implements a step-by-step heat exchange process, in which the energy conversion (heat, cold) of technological flows is carried out in heat power installations, including:

– multi-section gas coolers (MGC) of compressor units [6], [9], designed for cooling compressed gas and utilizing compression heat after intermediate CU stages (Fig. 1a), cooling, drying the compressed gas, and utilizing the compression heat after the last compression stage (Fig. 1b);

![Figure 1. CU multi-section gas coolers for cooling compressed gas and utilizing compression heat](image)

– multi-section wasteheat exchangers (MWE) [10], using the temperature potential of the waste gases after the gas turbine for heating compressed air before the combustion chamber in order to improve the GTU efficiency (recovery) and heat supply (utilization) for heating delivery water (Fig. 2a), as well as MWE with heating atmosphere before the compressor, to stabilize the GTP characteristics in winter (regulation) (Fig. 2b).

![Figure 2. Multi-section heat exchangers for heat recovery and utilization of GTP waste gases](image)

It is necessary to evaluate the thermal efficiency of multi-section heat exchange equipment (MGC, MWE) in heat power installations for gas cooling, heat recovery and utilization of technological flows as heat exchange systems (HES) of complex structure and their individual elements.

3. Theory

The overall assessment of the HES effectiveness can be based on a number of local and global criteria. Economic criteria are used as global ones: cost of the life cycle; discounted income; quoted costs [4].
For the two-section heat exchanging equipment (THEE), example, for a two-section gas cooler (TGC) (Fig. 1a), or a two-section wasteheat exchanger (TWE) (Fig. 2a) taking into account the determination of the optimal heat load ratio for each section \((Q_{air}, Q_{water})\), the optimal areas of the heat exchange section surfaces \((F_{air}, F_{water})\), the cost of the life cycle can be represented as:

\[
C_{lc}^{\text{THEE}} = K(F_{air}, F_{water}) + \sum_{i=1}^{T_{calc}} \left( \frac{C_{\text{THEE}} + C_{\text{repair}} + C_{\text{insp}}}{(1+E)} \right)
\]

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

Local criteria are: exergy efficiency of the installation with utilization \(\eta_{el}\); thermal efficiency of the heat exchanger \(\eta_{HE}\); energy consumption connected with pumping heat carriers along the heat exchanger paths \(N_{ce}\); cost of consumed electricity \(C_{\text{elec}}\); payback period \(t_{\text{pay}}\), etc.

As one of the main local criteria for evaluating the efficiency of various heat exchange equipment, its thermal efficiency function \(EF\) can be used [8].

\[
EF = \left[ \frac{Z + (A + 1)}{2} + \frac{Z}{\exp(SZ) - 1} \right]^{-1}
\]

where \(Z = \sqrt{(A + 1)^2 - 4 \cdot p \cdot A} ; A = (Gc \cdot c_G)/(Gc \cdot c_G)\) is function of water equivalents; \(Gc, G_r\) is flow rates of giving and receiving media; \(c_G, c_R\) is heat capacity of “giving” and “receiving” media; \(p\) is the countercurrency index, \(S = k \cdot F/(Gc \cdot c_G)\) is number of heat transfer units; \(k\) is heat transfer coefficient; \(F\) is heat exchange surface area.

The efficiency \(\eta_{HE}\) (thermal efficiency) of an individual heat exchanger expresses the ratio between the actual amount of heat \(Q_{G} = Q_{R}\) transferred in it and the maximum possible amount of heat \(Q_{\text{max}}\) to be transferred in an ideal heat exchanger, i.e. with an ideal current scheme (counterflow) and an infinitely large heat exchange surface. The efficiency of a separate heat exchanger for the giving and receiving medium can be determined by expressions (3) and (4)

\[
\eta_{HE}^{G} = \frac{Q_G}{Q_{\text{max}}} = \frac{F_G \cdot k_G \cdot \Delta T_G^H}{(Gc)_{\text{min}} \cdot \Delta T_{\text{max}}^{HE}} = \frac{Gc_G(T_{G_{in}} - T_{G_{fin}})}{(Gc)_{\text{min}} (T_{G_{in}} - T_{R_{in}})},
\]

\[
\eta_{HE}^{R} = \frac{Q_R}{Q_{\text{max}}} = \frac{F_R \cdot k_R \cdot \Delta T_R^H}{(Gc)_{\text{min}} \cdot \Delta T_{\text{max}}^{HE}} = \frac{Gc_R(T_{R_{fin}} - T_{R_{in}})}{(Gc)_{\text{min}} (T_{G_{in}} - T_{R_{in}})},
\]

where \(\Delta T_{\text{max}}^H = T_{G_{in}} - T_{R_{in}}\) is the maximum difference between the temperatures of the giving and receiving media; \(T_{G_{in}}, T_{R_{in}}, T_{G_{fin}}, T_{R_{fin}}\) is the initial and finite temperatures of the giving and receiving media.

Taking into account the value of the water equivalent function, the thermal efficiency of the heat exchanger (separate section) can be determined using the thermal efficiency function using by expressions (5) and (6).
At $A=(G_c c_g)/(G_r c_r) < 1$ in cooler mode minimum $(G c)_{\text{min}} = G_c c_g$, therefore

$$\eta_{\text{HE}}^G = \frac{(T_{G_{\text{in}}} - T_{G_{\text{fin}}})}{(T_{G_{\text{in}}} - T_{R_{\text{in}}})} = EF = \frac{(T_{R_{\text{fin}}} - T_{R_{\text{in}}})}{(T_{G_{\text{in}}} - T_{R_{\text{in}}})} \cdot \frac{1}{A}. \quad (5)$$

If $A=(G_c c_g)/(G_r c_r) > 1$ in heater mode minimum $(G c)_{\text{min}} = G_r c_r$, therefore

$$\eta_{\text{HE}}^R = \frac{(T_{R_{\text{fin}}} - T_{R_{\text{in}}})}{(T_{G_{\text{in}}} - T_{R_{\text{in}}})} = P = EF \cdot A = \frac{(T_{G_{\text{in}}} - T_{G_{\text{fin}}})}{(T_{G_{\text{in}}} - T_{R_{\text{in}}})} \cdot A. \quad (6)$$

The finite temperatures of the media for the heat exchanger (separate section) are determined by expressions (7) and (8).

$$T_{G_{\text{fin}}} = T_{G_{\text{in}}} - (T_{G_{\text{in}}} - T_{R_{\text{in}}}) \cdot EF \quad (7)$$

$$T_{R_{\text{fin}}} = T_{R_{\text{in}}} + (T_{G_{\text{in}}} - T_{R_{\text{in}}}) \cdot A \cdot EF \quad (8)$$

Then for the efficiency (thermal efficiency) of multi-section heat exchange equipment (MGC, MHEU), where the main is the only flow of the giving medium $\eta_{\text{MHEE}}$ at $A < 1$ can be written:

$$\eta_{\text{MHEE}} = \frac{Q_{\text{MHEE}}}{Q_{\text{max}}} = \frac{\sum_{j=1}^{N} k_j F_j \Delta T_j}{(G c)_{\text{min}} \cdot \Delta T_{\text{MHEE}}^\text{max}} = \frac{\sum_{j=1}^{N} G_c c_G (T_{G_{\text{in}}} - T_{G_{\text{fin}} j})}{G_c c_G (T_{G_{\text{in}}} - T_{R_{\text{in}} \text{min}})} = \frac{T_{G_{\text{in}}} - T_{G_{\text{fin}}}}{T_{G_{\text{in}}} - T_{R_{\text{in}} \text{min}}}, \quad (9)$$

where $k_j$, $F_j$, $\Delta T_j$ is heat transfer coefficient, heat exchange surface area, log mean temperature difference of the MHEE $j$-th section; $(G c)_{\text{min}}$ is the minimum product of medium mass flow and heat capacity (for one of the media); $\Delta T_{\text{MHEE}}^\text{max} = T_{G_{\text{in}}} - T_{R_{\text{in}} \text{min}}$ is the maximum difference between the temperatures of the giving and receiving media; $T_{G_{\text{in}}} = T_{G_{\text{in}} \text{1}}$; $T_{G_{\text{fin}}} = T_{G_{\text{fin}} N}$; $N$ is the number of sections.

For an ideal scheme, where each subsequent receiving flow has a lower initial temperature along the path of the giving medium (for $A_j \leq 1$), the MHEE efficiency is determined by expression (9) under the condition $T_{R_{\text{in}} \text{min}} = T_{R_{\text{in}} N}$, and for an arbitrary scheme (an arbitrary sequence of receiving flows along the giving medium path) under the condition $T_{R_{\text{in}} \text{min}} = T_{R_{\text{in}} R}$ ($1 \leq R \leq N$).

The most important tool to evaluate the thermal efficiency of multi-section heat exchange equipment for heat power installations is the representation of MGC or MHEU in the form of a heat exchange system (HES). Possible options for the structural organization of MGC CU and MHEU GTU (two-section, three-section), shown in Fig. 1 and Fig. 2, assume their implementation in the form of an elementary cross-heat exchange system (CHES), summarized in Fig. 3.
For CHES using the method of mathematical induction, expressions of the system (10) were obtained to determine the final temperatures of media and the function of thermal efficiency

\[ EF_{CHES} = \{P_{ij}^G, P_{ij}^R\}. \]

\[
\begin{align*}
T_{G,fin} &= T_{G,in} - \sum_{j=n}^{k} (T_{G,in} - T_{R,in,j}) \cdot P_{jk}^G, \\
T_{R,fin,i} &= T_{R,in,i} + \sum_{j=n}^{i} (T_{G,in} - T_{R,in,j}) \cdot P_{ji}^R, \\
& \quad \text{for } i = n, k.
\end{align*}
\]

\[
\begin{align*}
P_{jk}^G &= EF_j \prod_{m=j+1}^{k} (1 - EF_m), \\
P_{ji}^R &= \begin{cases} EF_j \prod_{m=j+1}^{i-1} (1 - EF_m) \cdot A_i \cdot EF_i, & \text{for } j < i, \\
A_i \cdot EF_i, & \text{for } j = i.
\end{cases}
\end{align*}
\]

where \( P_{jk}^G \) is function of the thermal efficiency for CHES on the giving medium; \( P_{ji}^R \) is the function of the thermal efficiency for CHES on receiving medium; \( EF_j, A_j \) is the function of the thermal efficiency and water equivalents for the \( i \)-th element; \( j, i, m \) are the elements numbers of HES structural scheme.

Then for a two-section gas cooler (Fig. 1a) or a two-section wasteheat exchanger (Fig. 2a), for \( n=1, k=2 \), we get:

\[
\begin{align*}
T_{G,fin} &= T_{G,in} - \sum_{j=1}^{2} (T_{G,in} - T_{R,in,j}) \cdot P_{j2}^G, \\
T_{R,fin,i} &= T_{R,in,i} + \sum_{j=n}^{i} (T_{G,in} - T_{R,in,j}) \cdot P_{ji}^R, \\
& \quad \text{for } i = 1, 2, \\
P_{12}^G &= EF_1 \cdot (1 - EF_2), \quad P_{22}^G = EF_2, \\
P_{1i}^R &= \begin{cases} -EF_1 \cdot A_i \cdot EF_i, & \text{for } j < i, \\
A_i \cdot EF_i, & \text{for } j = i.
\end{cases}
\end{align*}
\]

Figure 3. Elementary cross heat exchange system
For two-section heat-exchange equipment (TGC, TWE), where the only flow of the giving medium is main, the determination of optimal structure $S$ (sequence of heat exchanger sections) and the ratio of the heat load between the sections can be carried out with regard to the dominant parameter – the temperature of compressed gas (compressed air) after an air section $t_{air}$. Then the optimization problem for TGC and TWE structural and compositional parameters – to determine the optimal heat exchange surface areas of its regenerating and water sections $F = \{F_{air}, F_{water}\}$ can be formulated in the following way:

$$
\psi^* = \text{opt } \psi(S, F_{air}, p_c, t_{c}, t_{air}, t_{w}) \\
U = \{S, F_{air}, F_{water}\} = \{S, t_{air}\}
$$

(12)

where $t_{c}, p_c$ is the temperature and pressure of compressed air entering the THEE from the compressor, $t_{a}$ is the air temperature, $t_{w}$ is the temperature of the delivery water entering the THEE water section.

The following can be used as an efficiency criterion $\psi$ MHEE efficiency $\eta_{MHEE}$; MHEE thermal efficiency function in the form $EF_{CHES}$ defined by expressions (9), (11).

4. Experiment results
The studies of the two-section gas cooler (Fig. 1а) were carried out on an experimental installation including the compressor (capacity of 0.5 m$^3$/min), the heat exchange unit (HEU) consisting of an air tube-ribbed (0.42 m$^2$) and water shell-and-tube (0.45 m$^2$) sections, and the receiver. A special switch was available to provide different cooling sequences for compressed air (initially in the air section and then in the water one, or conversely). After passing both heat exchangers, the compressed air was sent to the receiver, and from it through the shut-off valve to the atmosphere. In order to expand the range of compressed air temperatures, a heater was installed before the switch at the entrance to the first heat exchanger. The initial compressed air and water temperatures for HEU had constant values 120°C and 19°C, respectively. The air temperature at the entrance to the air section took different values: -12°C; -4°C; +2°C; +12°C; +17°C; +23°C.

The results of experimental studies for the thermal TGC efficiency are shown in Fig. 4. Figure 4a shows the process of changing the thermal HES efficiency and the efficiency of individual sections for sequential cooling of compressed air initially in the air and then in the water section (air $\rightarrow$ water). In Figure 4b there is a reverse order (water $\rightarrow$ air).

![Figure 4. TGC efficiency for the sequence air $\rightarrow$ water (a) and water $\rightarrow$ air (b)](image-url)
The dependences of the HES thermal efficiency function for both sequences are shown in Fig. 5.

Figure 5. Function of thermal efficiency for two-section gas cooler with different structure

Numerical studies for two-section wasteheat exchanger (Fig. 2a) were carried out for gas turbine unit GTK-10-4. The calculations were made in the temperature range of the compressed air heated in the air section \( t_{air} = 350...507°C \). For the water section \( t_{w} = 70°C \), \( t_{water} = 115°C \), \( G_{w} = 10.56 \) kg/s. The results of the calculations are shown in Fig. 6a.

Figure 6. Temperature effects of the compressed air heated in the air section on the thermal TWE efficiency (a) and GTP power loss (b)

Figure 6b shows the dependence of GTP power losses on the temperature of compressed air and \( EF_{HES} \).

5. Results and discussion

As follows from the given data, the dependences of the thermal efficiency function allow one to determine the most effective ranges of air temperatures for HES of each sequence. HES with a sequence (air \( \rightarrow \) water) is more effective at air temperatures of more than 12°C, provided that the water section is switched off below -4°C. At air temperatures below 12°C, according to the \( EF_{HES} \), HES with a sequence (water \( \rightarrow \) air) is more effective.

The air temperature increase at the outlet from air TWE section is directly related to the growth in its thermal efficiency \( EF \) over the entire range \( t_{air} \) of 350...507°C and the area of the heat exchange surface \( F \) (Fig. 6a). The latter leads to an increase in hydraulic resistances along the GTU gas-air path and additional power costs to overcome them. Because of further increase in the surface area \( F \), the compressed air temperature \( t_{air} \) rise above 460°C is impractical, due to the significantly outrunning growth of the associated energy costs (Fig. 6b).
It is possible to define the optimum ratio of thermal loads and the areas of heat-exchange surfaces for TWE for the obtained optimum compressed air temperature after the air section in accordance with the data presented in Fig. 6 and taking into account the requirements to the heated delivery water. The rational thermal load (surface) of the air section is 75 ... 85% of the two-section wasteheat exchanger load (surface).

6. Conclusion
For a two-section gas cooler using the thermal efficiency function $EF_{HES}$, it was determined the optimal structure $S$ (sequence of sections), and in particular, the rational ranges of atmospheric air temperature values for each sequence. At atmosphere temperatures below $12^\circ C$, according to $EF_{HES}$, HES with the sequence (water$\rightarrow$air) is more effective.

The thermal efficiency analysis $EF$ of air TWE section for gas turbine GTK-10-4 carried out in the temperature range of compressed air $350...507^\circ C$ showed the growth over the entire range, and taking into account additional local criteria (GTP power loss) allowed determining the rational temperature of the compressed air after the air section ($t_{w}=460^\circ C$) and optimal ratio of heat loads (heat exchange surface areas) for MWE air and water sections (75...85% to 15...25%).

Efficiency evaluation of HPI multi-section heat exchange equipment providing a comprehensive solution to the problems of improving energy conversion can be carried out according to a number of local and global criteria. One of the most important local criteria is the thermal efficiency function. The proposed method of using this criterion for multi-section heat exchange equipment, along with economic, energy and exergy criteria, allows for a better evaluation of various MGC and MWE versions.

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