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Performance of Series Connected Heat Exchangers with Liquid Circuit on Loop

Esa Marttila

Lappeenranta University of Technology
Department of Energy Technology

ABSTRACT

The series connected heat exchangers - configured either as an arrangement of gas-gas, gas-liquid or liquid-liquid heat exchangers- are widely used in the process industry and air-conditioning where they can be found in a variety of heat (cool) recovery, in heating and cooling applications. The design of heat exchangers follows the conventional design procedures for compact heat exchangers, and involves computation of excanger dimensions to yield a prescribed heat exchanger efficiency and pressure drops for fluid streams.

This paper describes a procedure for calculating the performance of series connected heat exchangers, when heat tranfer fluid has been looped between heat exchangers. The heat excanger system consists of heat (cool) recovery heat exchanger(s), pre-heater(s) or pre-cooler(s) and a main heat exchager to heat or to cool a main stream to the aim temperature. The heat excangers are connected with the looped fluid circuit. The connecting fluid flow rate has to be determined to achieve optimum heat or cool recovery.

The following considerations has to be almost always made when the heat exchangers is designed for the application,

- Heat transfer requiremments
- Costs
- Physical size
- Pressure drop characteristics

In the main stream the heat transfer requirements for temperatures must be met in the selection or design of any heat exchanger. The way with which the requirements are met, depends on the relative weights placed on costs, physical size or pressure-drop characterristics. Economic plays a key role in the design and selection of heat exchanger equipment, and the engineer should bear this in his mind when embarking on any new heat transfer design problem.

For the used heat exchangers the presented procedure gives the looped fluid flow rate, with which the best possible heat recovery efficiency is achieved. In the other hand the procedure gives the performances for heat exchangers to achieve the end temperature of the main stream.

List of Symbols

c_p	specific heat capacity of fluid, J/kgK
E_e	value of energy transferred by the heat recovery system annually, FIM
q_v	volume flow rate, m^3/s
q_{v1}	volume flow rate of supply air, m^3/s
q_{v2}	volume flow rate of exhaust air, m^3/s
Φ	heat flux from exhaust air to supply air, W
C_1	heat capacity rate of supply air, W/K
C_2	heat capacity rate of exhaust air, W/K
C_3	heat capacity rate of heat transfer liquid, W/K
P	electric capacity of compressor, W
R	ratio of heat capacity rates C_1/C_2
R_1	ratio of heat capacity ratesC_1/C_3
R_2	ratio of heat capacity rates C_2/C_3
T_{11}	inlet temperature of supply air in heat recovery heat exchanger, K
T_{12}	outlet temperature of supply air in heat recovery heat exchanger, K

T_{21} inlet temperature of exhaust air in heat recovery heat exchanger, K
T_{22} outlet temperature of exhaust air in heat recovery heat exchanger, K
T_{31} inlet temperature of liquid in the supply air heat exchanger, K
T_{32} outlet temperature of liquid in the supply air heat exchanger, K
T_{33} inlet temperature of liquid in the exhoust air heat exchanger, K
T_{34} outlet temperature of liquid in the exhoust air heat exchanger, K
e_h price of primary energy, FIM/MJ
f_1, f_2,f_{11},
f_{12} help functions, equations (32)-(35)
q_{v3} volume flow rate of heat tranfer liquid, m^3/s
ΔT_V change of temperature of heat tranfer liquid T_{31}-T_{34} or T_{34}-T_{31}, K
ε thermal efficiency from exhaust air into supply air
ε_1 thermal efficiency of supply air $(T_{12}-T_{11})/(T_{31}-T_{11})$
ε_2 thermal efficiency of exhaust air $(T_{21}-T_{22})/(T_{21}-T_{33})$
ε_j coefficient of performance in the refrigeration cycle
θ_0 temperature difference, T_{21}-T_{11} tai T_{11}-T_{21} , K
θ_1 temperature difference, T_{31}-T_{11} tai T_{11}-T_{31}, K
θ_2 temperature difference, T_{21}-T_{33} tai T_{33}-T_{21}, K
θ^* relative change of heat transfer liquid, $\Delta T_V/\theta_0$
θ^{**} relative change of supply air in heating $(T_{12}-T_{11})/\theta_0$ or in cooling $(T_{11}-T_{12})/\theta_0$
ϕ_1 heat capacity to the supply air, W
ϕ_2 heat capacity from exhaust air, W
ϕ_3 heat capacity from primery energy source, W
ϕ_{max} maximum possible transferrable heat flow, W
ρ density, kg/m^3

DIMENSIONING THE HEAT RECOVERY HEAT EXCHANGER IN VENTILATION

The recovery of heat (cool) in ventilation systems forms a part of the production and distribution of heat in the whole of a building. Most commonly available sources of energy for satisfying the heating need of a building are fuel, district heating, or electricity, and, as an addition to primary energy, the waste heat flow of a building. The most common waste heat flow of a building which is possible to utilize is the entalpy content of exhaust air.

The utilization of the entalpy content of exhaust air for heating the supply air depends on the costs of utilization and the savings obtained through utilization. The object in the dimensioning of a heat recovery unit is at first to find the heat exchanger size with whicht it is possible to achieve the economically best final results.

The factors which determine the dimensioning of a heat recovery unit are: the saving of primary energy through heat recovery, the investment costs of carrying out the heat recovery system, and the operating costs of heat recovery and, in addition, the investment savings achieved through the carrying out of a heat recovery system.

Value of Recovered Entalpy Content of Exhaust Air

The entalpy content recovered from exhaust air can be got with the aid of the recuperation ratio (ε) of a heat recovery heat exchanger. The restriction of recuperation ratio is possible in connection with protection against ice formation and overheating. Ventilation systems usually operate between certain times (from t_{1j} to t_{2j}, j = 1 to 365) of a day on certain days of week. The value of the entalpy content of exhaust air recovered then can be calculated from the following equation (1) (Marttila 1996).

$$E_e = \epsilon q_{V1} c_{p1} \rho e_h \sum_{j=1}^{365} \int_{t_{1j}}^{t_{2j}} \langle T_{21} - T_{11} \rangle_j \, dt \tag{1}$$

When only temperature difference is a function of time in Equation (1), the so-called degree-hour amount can be calculated as in integral on the basis of maximum temperature difference on operation time (Marttila 88)

$$\bar{\theta} t = \sum_{j=1}^{365} \int_{t_{1j}}^{t_{2j}} \langle T_{21} - T_{11} \rangle_j \, dt \tag{2}$$

The recuperation ratio of a heat recovery heat exchanger is defined as the ratio between the transferred heat flow and the largest possible transferrable heat flow (Kays and London 84).

$$\Phi_{max} = (q_{vi} c_{pi} \rho_i)_{min} (T_{21} - T_{11}) \tag{3}$$

$$\varepsilon = \frac{\Phi_c}{\Phi_{max}} = \frac{\Phi_h}{\Phi_{max}} \tag{4}$$

Series Connected Heat Exchangers with Liquid Circuit on Loop

In common application of the heat (cool) recovery (Figure 1) the thermal efficiency ε can be defined as in the equation (5) (Kays ja London 1984).

$$\epsilon = \frac{\epsilon_1 \epsilon_2}{\frac{\epsilon_1 R_1}{R_2} + \epsilon_2 - \epsilon_1 \epsilon_2 R_1} \tag{5}$$

Next the thermal efficiency will be defined when the heat and cool coils in the figure 1 has been replaced with the plate heat exchaners as in the figure 2.

In the air conditioning system the supply air has to be cooled or to be heated to the certain temperature T_{12}. The heating or cooling capacity can be written as in the equation (6).

$$\phi_1 = C_1 (T_{12} - T_{11}) \tag{6}$$

The heating or cooling capacity from the exhaust air can be written as in the equation (7).

$$\phi_2 = C_2 (T_{21} - T_{22}) \tag{7}$$

The heating or cooling capacity from the primary source (fuel, district heating, electricity) can be written as in the equation (8).

$$\phi_3 = C_3 (T_{31} - T_{34}) \tag{8}$$

Now the heating capacity can be written as in the equation (9).

$$\phi_1 = \phi_2 + \phi_3 \tag{9}$$

The parameters for the thermal efficiency ε are defined as ε_1, ε_2,.θ_o, θ_1, θ_2, θ^{**} and θ^* (equations (10) - (15).

$$\epsilon_1 = \frac{T_{12} - T_{11}}{T_{31} - T_{11}} \tag{10}$$

$$\theta_o = T_{21} - T_{11} \tag{11}$$

$$\theta_1 = T_{31} - T_{11} \tag{12}$$

$$\theta_2 = T_{21} - T_{33} \tag{13}$$

$$\theta^{**} = \frac{T_{12} - T_{11}}{\theta_o} \tag{14}$$

$$\theta^{*} = \frac{T_{31} - T_{34}}{\theta_o} \tag{15}$$

Now the thermal efficieny can be written as in the equation (16).

$$\epsilon = \theta^{**} - \frac{1}{R_1}\theta^{*} \tag{16}$$

The paramter θ^* can be written as in the equation (17).

$$\theta^{*} = \theta^{**} R_1 \left(1 + \frac{1}{R}\frac{\epsilon_2}{\epsilon_1} - \epsilon_2 R_2 \right) - R_2 \epsilon_2 \tag{17}$$

Now the thermal efficiency can be written as in the equation (18) (Marttila, 1996).

$$\epsilon = \theta^{**} \left(\epsilon_2 R_2 - \frac{1}{R}\frac{\epsilon_2}{\epsilon_1} \right) + \frac{1}{R}\epsilon_2 \tag{18}$$

Heat Exchanger network design in Cooling (Figure 3)

Next the situation are dealed , where the condensation and vaporization heat exchaners are coupled in series with the heat recovery heat exchangers.

Cooling demand to the supply air can be written as the equation (19).

$$\Phi_1 = C_1 \, (T_{11} - T_{12}) \tag{19}$$

Heating capacity to the exhaust air consists of the cooling demand of the supply air and the electric capacity of the compressor. The heating capacity to the exhaust air can be written as the equation (20).

$$\Phi_2 = C_2 \, (T_{22} - T_{21}) \tag{20}$$

Electric capacity of the compressor can be written as the equation (21).

$$P = \frac{C_3 \, (T_{34} - T_{31})}{\epsilon_j} \tag{21}$$

Heating capacity to the exhaust air can be written as the equation (22)

$$\Phi_2 = \Phi_1 + P \tag{22}$$

The thermal efficiencies of the heat exchangers are defined as in the equations (23) and (24).

$$\epsilon_1 = \frac{T_{11} - T_{12}}{T_{11} - T_{34}} \tag{23}$$

$$\epsilon_2 = \frac{T_{22} - T_{21}}{T_{33} - T_{21}} \tag{24}$$

Caculation parameters θ_o, θ_1, θ_2, θ^{**} and θ^* are defined as in the equations (25) - (29).

$$\theta_o = T_{11} - T_{21} \tag{25}$$

$$\theta_1 = T_{11} - T_{31} \tag{26}$$

$$\theta_2 = T_{33} - T_{21} \tag{27}$$

$$\theta^{**} = \frac{T_{11} - T_{12}}{\theta_o} \tag{28}$$

$$\theta^{*} = \frac{T_{34} - T_{31}}{\theta_o} = \frac{\Delta T_v}{\theta_o} \tag{29}$$

With the aid of these parameters the temperature differences ΔT_v ja θ_2 can be written as in the equations (30) and (31).

$$\Delta T_v = R_1 \epsilon_1 \theta_1 + [\ \Delta T_v (1 + \frac{1}{\epsilon_j}) - R_2\ \epsilon_2\ \theta_2] \tag{30}$$

$$\theta_2 = \Delta T_v\ (1 + \frac{1}{\epsilon_j}) - \theta_1\ (1 - R_1\ \epsilon_1) + \theta_o \tag{31}$$

The funqtions f_1, f_2, f_{11} ja f_{12} have been defined as the equations (32) - (35).

$$f_2 = \frac{1}{\epsilon_j}\ (\ 1 - \epsilon_2\ R_2) - \epsilon_2\ R_2 \tag{32}$$

$$f_{11} = R_2 \, \epsilon_2 \tag{33}$$

$$f_{12} = \theta^{**} \frac{R_2}{\epsilon_1} \left(R \, \epsilon_1 + \epsilon_2 - \epsilon_1 \, \epsilon_2 \, R_1 \right) \tag{34}$$

$$f_1 = f_{11} - f_{12} \tag{35}$$

With the aid of equations (32) - (35) the temperature cahnge of the liquid flow (q_{v3}) can be defined as the equation (36).

$$\theta^* = \frac{\Delta T_v}{\theta_o} = \frac{f_1}{f_2} \tag{36}$$

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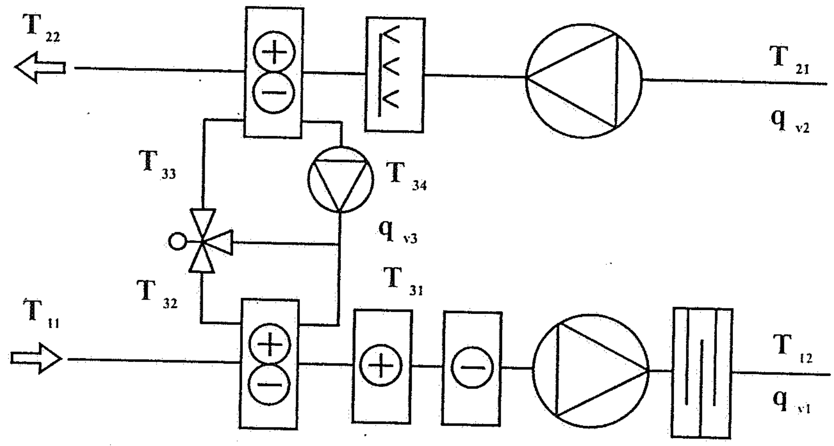

Figure 1 The indirect heat excanger system when primary energy has been carried for the system to the heating (cooling) coils.

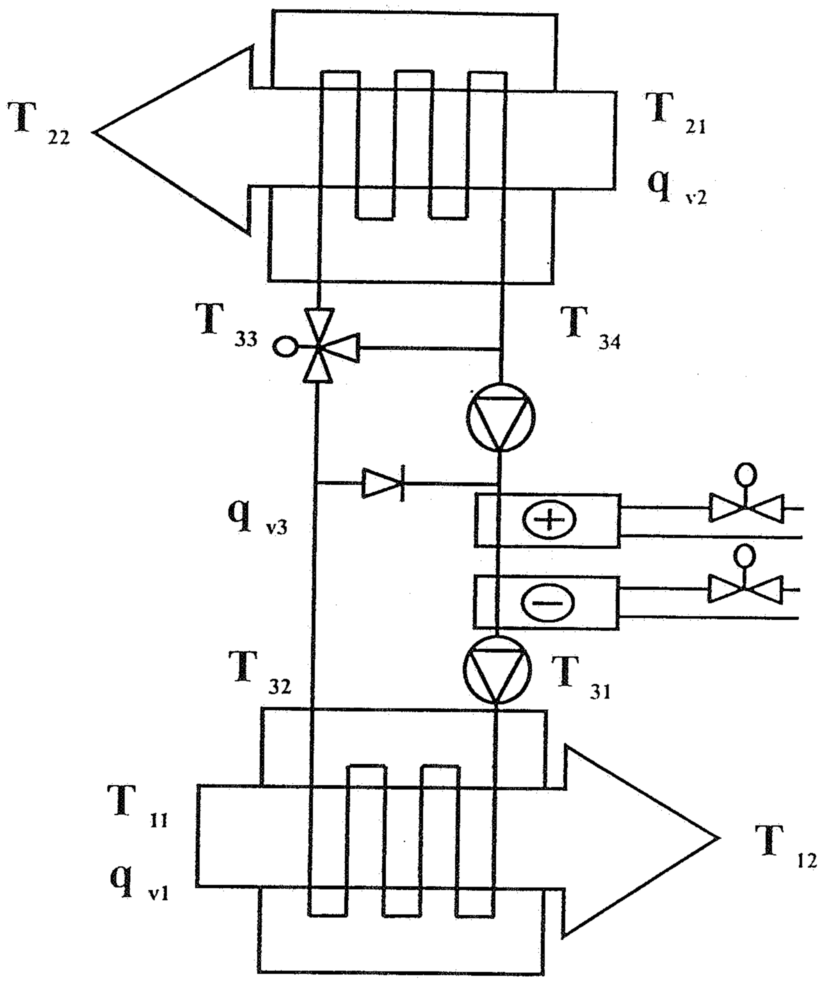

Figure 2. The indirect heat excanger system when primary energy has been carried for the system to the plate heat exchangers.

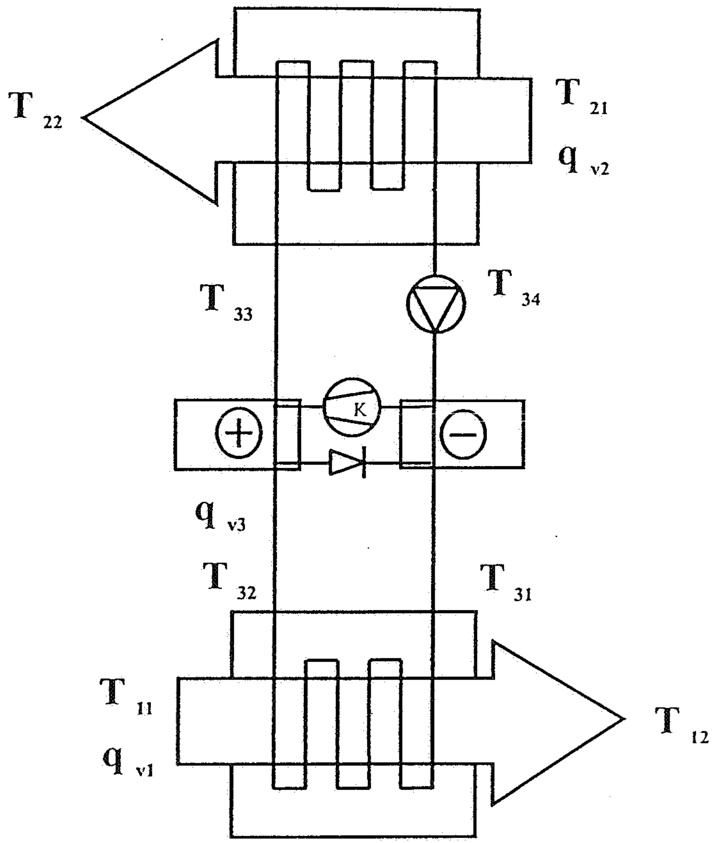

Figure 3. The indirect heat excanger system when condensation and vaporization heat exchangers are coupled in series with heat recovery heat exchangers.