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Design Optimisation of Heat Exchanger based on Exergy Analysis

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Abstract—*The analysis has been carried out for the optimal design of a heat exchanger in a district heating system. The optimization takes into account irreversibilities due to frictional pressure drops and the temperature difference between the hot and cold stream and to irreversibilities due to the production of the materials and the construction of the heat exchanger. As an example of this type of heat exchanger a water to water heat exchanger in a district heating system has been selected. The influence of the configuration of the heating system, including the energy conversion, on the optimisation of the heat exchangers has been shown. The analysis of the heat exchanger in which the concepts of the ELCA are used gives the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.*

1. INTRODUCTION

Because of its widespread use the heat exchanger has been selected as an example. Bejan [1] studied extensively the optimisation of a heat exchanger, excluding exergy destruction associated with use of materials and the generation of heat and power. His approach uses the concept of entropy generation minimisation. An extension to his approach to include material use has been made by Aceves-Saborio et al [2]. They took into account the irreversibility associated with the use of the materials, but did not include the irreversibility due to the pressure drops.

Tondeur and Kvaalen [3] have shown that in the case of heat exchangers or separation devices involving a given heat transfer and achieving a specified transfer duty, the total entropy produced is minimal when the local rate of entropy production is uniformly distributed along space variables and time. De Oliveira et al. [4-5] have shown that in the case of an optimal heat exchanger the thermal and viscous 96 contributions to the entropy generation should be equal when the heat flux is optimised. The ratio of thermal and viscous contribution to the exergy destruction is between one and three when the Reynolds number or hydraulic diameter is optimised. However it is not shown that this is the case when both, Reynolds number and hydraulic diameter, are together optimised.

Lozano and Valero [6] have developed a theory to allocate exergetic and monetary cost. In this theory they define a matrix containing all irreversibilities, including irreversibilities associated with the building of installations and the disposal of waste materials. No extensive examples of this theory, including irreversibilities associated with the building of installations and the disposal of waste materials, are available.

In this paper the optimisation of the heat exchanger has been performed using the criterion of the minimisation of the life cycle irreversibility for the delivery of domestic hot water. The methodology is based on the minimisation of the life cycle irreversibility. It uses the marginal life cycle irreversibilities of the different components.

2. OPTIMISATION OF A HEAT EXCHANGER

2.1 The heat exchanger

The heat exchanger analysed is a balanced counter flow heat exchanger, which is used in a district heating system to heat the domestic tap water. An equal mass flow of the hot and cold water has been assumed. The inner and outer tubes have been constructed from copper and steel, respectively. The thermal insulation of the heat exchanger has been assumed to be perfect.

The following formula can be derived for the irreversibility in the heat exchanger due to the stream to stream heat transfer and pressure drops:

$$\dot{I} = \dot{I}^{\Delta T} + \dot{I}^{\Delta P}$$

$$\dot{I}^{\Delta T} = T_0\left[\dot{m}c_P \ln\frac{T_{1,out}}{T_{1,in}} + \dot{m}c_P \ln\frac{T_{2,out}}{T_{2,in}}\right] \tag{1}$$

$$\dot{I}^{\Delta P} = \left[\frac{\dot{m}}{\rho}\left(P_{1,in} - P_{1,out}\right) + \frac{\dot{m}}{\rho}\left(P_{2,in} - P_{2,out}\right)\right] \tag{2}$$

$$\dot{I}^{\Delta T} = T_0 \cdot \dot{m} c_P \left[\ln \frac{T_{1,out}}{T_{1,in}} + \ln \frac{T_{1,in} + \Delta T}{T_{1,out} + \Delta T} \right] \ldots \quad (3)$$

Because $T_{2,out} = T_{1,in} + \Delta T$ and $T_{2,in} = T_{1,out} + \Delta T$ in a balanced heat exchanger. With the heat balance of the n inner tubes

$$n \cdot \dot{m}_n c_P \left(T_{1,out} - T_{1,in}\right) = \alpha A \Delta T \quad \ldots\ldots\ldots \quad (4)$$

With $\dot{m}_n = \frac{\dot{m}}{n}$, ΔT in equation (4) can be substituted. By using the force balance one can express the pressure drops in the tubes as:

$$\Delta P_1 = P_{1,in} - P_{1,out} = 2 f_1(\mathrm{Re}) \cdot \rho \cdot \bar{u}_1^{\,2} \frac{L}{D_1} \quad (5)$$

$$\Delta P_2 = P_{2,in} - P_{2,out} = 2 f_2(\mathrm{Re}) \cdot \rho \cdot \bar{u}_2^{2} \frac{L \cdot \left(D_2 + n \cdot \left(D_1 + 2d_1\right)\right)}{\left(D_2^2 - n \cdot \left(D_1 + 2d_1\right)^2\right)}$$

(6)

Using $$\bar{u}_1 = \frac{4\dot{m}_n}{\rho \pi D_1^2} \quad (7)$$

And

$$\bar{u}_2 = \frac{4\dot{m}}{\rho \pi \left[D_2^2 - n \cdot \left(D_1 + 2d_1\right)^2\right]} \quad (8)$$

For the mean velocities of the fluid in the tubes and substituting (5) and (6) in (2) yields

$$\dot{I}^{\Delta P} = \frac{32}{\pi^2} \left[f_1(\mathrm{Re}) \frac{n \cdot \dot{m}_n^{\,3}}{\rho^2} \frac{L}{D_1^5} + f_2(\mathrm{Re}) \frac{\dot{m}^3}{\rho^2} \frac{L \cdot \left(D_2 + n \cdot \left(D_1 + 2d_1\right)\right)}{\left[D_2^2 - n \cdot \left(D_1 + 2d_1\right)^2\right]^3} \right] \quad (9)$$

2.2 Turbulent region

In the turbulent flow region in tubes and annular spaces with a limited temperature difference of 5 K for liquids between the bulk fluid and pipe surface temperature we have for the heat transfer coefficient according to Chapman [7]

$$Nu = \frac{\alpha D_h}{\lambda} = 0.023 \, \mathrm{Re}^{0.8} \, \mathrm{Pr}^n = 0.023 \cdot \left(\frac{\rho \bar{u} D_h}{\mu}\right)^{0.8} \cdot \left(\frac{c_P \mu}{\lambda}\right)^n \quad (10)$$

With n = 0.3 or 0.4 for cooling or heating, respectively. Experimental data of Kays and London [8] gives a similar relation. The friction factor in tubes according to the friction law of Blasius is given in Rogers and Mayhew [9] as

$$f(\mathrm{Re}) = \frac{0.0791}{\mathrm{Re}^{0.25}} = 0.0791 \cdot \left(\frac{\mu}{\rho \bar{u} D_h}\right) \quad (11)$$

However, to include the curving of the tube the following correction factor according to Ito [7] has to be included in (9)

$$C = 0.962 + 0.0962 \, \mathrm{Re}^{0.25} \left(\frac{D}{R}\right)^{0.5} \quad (12)$$

Where R is the diameter of the curve. Substituting (11) and (12) in (10), using A = πLD1 for the inner tubes and A = πL(D1 + 2d1) for the outer tubes and neglecting the heat resistance of the tube yields _

with

$$\dot{I}^{\Delta T} = T_0 \cdot \dot{m} c_P \left[\ln \frac{T_{1,out}}{T_{1,in}} + \ln \frac{T_{1,in} + \Delta T}{T_{1,out} + \Delta T} \right] \quad (13)$$

With

$$\Delta T = \frac{\left(T_{1,out} - T_{1,in}\right)}{0.023 \cdot n \cdot 4^{0.8} \cdot \pi^{0.2}} \left(\begin{array}{l} \mu^{0.4} \left(\frac{c_P}{\lambda}\right)^{0.6} \frac{\dot{m}}{\dot{m}_n^{0.8}} \frac{D_1^{0.8}}{L} + \\ \mu^{0.5} \left(\frac{c_P}{\lambda}\right)^{0.7} \frac{m^{0.2} \left(D_2^2 - n \cdot \left(D_1 + 2d_1\right)^2\right)}{L \cdot \left(D_1 + 2 \cdot d_1\right) \cdot \left(D_2 + n \cdot \left(D_1 + 2 \cdot d_1\right)\right)^{0.2}} \end{array} \right) \quad (14)$$

And

$$\dot{I}^{\Delta P} = \frac{1.79}{\pi^{1.75}} \frac{\mu^{0.25}}{\rho^2} \left[\begin{array}{l} \frac{n \cdot \dot{m}_n^{2.75}}{D_1^{4.75}} \left(0.962 + 0.092 \left(\frac{4 \cdot m_n}{\pi \mu D_1}\right)^{0.25} \left(\frac{D_1}{R}\right)^{0.5}\right) + \\ \frac{\dot{m}^{2.75} L \cdot \left(D_2 + n\left(D_1 + 2d_1\right)\right)^{1.25}}{\left[D_2^2 - n\left(D_1 + 2d_1\right)^2\right]^3} \left(0.0962 + 0.092 \left(\frac{4 \cdot m}{\pi \mu \left(D_2 - n \cdot \left(D_1 - 2d_1\right)\right)}\right)^{0.25} \left(\frac{D_2}{R}\right)^{0.5}\right) \end{array} \right]$$

(15)

2.3 Cumulative losses due to power and heat generation

Irreversibilities in heat exchangers are associated with external exergy destruction, for example, to overcome the frictional losses in the heat exchanger a pressure difference is needed. The generation of heat and power is associated with exergy destruction, which has to be taken into account in the optimisation, according to the life cycle approach. We assume the following situation, based on district heating. In this system a combined heat and power plant, consisting of a steam and gas turbine and a provision to extract steam, is used, which has an exergetic efficiency of 50%, when there is no useful heat production. The exergetic efficiency of such a plant stays nearly constant when part of the steam is used for the district heating system. The operator is free to a certain extent to provide exergy in the form of either electricity or heat. Irreversibilities associated with the building of the combined heat and power plant and the transport of the fuel, natural gas, are neglected. The exergetic efficiency of the heat transport to the houses is estimated to be 0.5 for a widespread distribution net (wide net) and 0.75 for a very dense distribution net (dense net). The exergy destruction in the wide net is greater because of more power needed to overcome the frictional pressure drops and more heat transfer to the environment. A great part of the exergy destruction in the heat transport takes place because of the temperature difference in the heat exchanger between the main transport tube and the local distribution net. It has been assumed that the exergy destruction associated with the heat transport is independent of the district heating water temperature to ease the calculations. The exergetic efficiency of the pumps is assumed to be 0.7. Hence the exergetic cost, which is the amount of exergy which is used for the production of one exergy unit, can be calculated for the analysed heat exchanger of pressure rise, kP, and heat, kT. The exergetic cost for the pressure rise (kp) is kpower·kpumps = 2.85, where kpower = 2 and kpumps = 1.43. These are calculated by taken the inverse of the exergetic efficiency of power generation and pumping. The exergetic cost for the heat (kT) can be calculated to be 4 for the wide net and 2.66 for the dense net. For the exergy destruction associated with the operation of the heat exchangers we have

$$\dot{I}_{\text{oper}} = k_{\text{T}} \dot{I}^{\,\Delta\text{T}} + k_{\text{P}} \dot{I}^{\,\Delta\text{P}} \tag{16}$$

2.4 Irreversibilities associated with the use of the material

A detailed exergy analysis of the primary anode copper process is presented by Kolenda et al [10]. The cumulative irreversibility of the flash smelting process, one of the main copper production processes, is shown to be 45.8 MJ per kg anode copper. Including the exergy destruction due to discharge and dissipation of materials into the environment, like slag and combustion gasses, gives a total cumulative exergy destruction of 55.6 MJ per kg primary anode copper.

Estimation of the exergy destruction is ever constant endeavor in the heat exchanger research, so due to lack of sufficient data the energy consumption instead of the exergy destruction has been taken. To use the energy consumption instead of the exergy destruction, the following conditions have to be fulfilled. Most inputs ought to be fossil fuels, because the exergy and enthalpy content are then roughly equal or electricity, because exergy and energy are then equal. The exergy increase of the product during the processing has to be subtracted from the energy use to get the exergy destruction. The cumulative energy use of the production of solid copper out of anode copper is around 4.5 MJ/kg copper according to Boustead and Hancock [11]. The exergy increase is almost zero and we assume the enthalpy values of the inputs to equal the exergy values, so this value is taken for the exergy destruction. The cumulative exergy destruction for primary copper becomes 55.6 + 4.5 = 60.1 MJ/kg (Cpri,Cu). The energy use for the production of primary steel slabs is 16.9 MJ/kg8 according to Worrell (1992). Because most inputs are raw materials of which the exergy content roughly equals the energy content the cumulative exergy use is estimated to be equal to the energy use. The exergy destruction is the exergy input reduced by the exergy increase due to the change of iron oxide to steel. The exergy increase is estimated to be 6.4 MJ/kg, so the exergy destruction is 10.5 MJ/kg (Cpri,S). The cumulative exergy destruction of secondary steel is calculated from Wall [12] by assuming the efficiency of the electricity production to be 0.5. The exergy destruction associated with the production of the alloying materials and lime has been neglected. Boustead and Hancock [11] give two totally different values for the energy consumption of the secondary copper production, namely 7.2 and 48.8 MJ/kg. The exergy destruction of this process is taken to be 20 MJ/kg (Csec,Cu).

The manufacturing process of the steel tubes includes hot and cold rolling and the bending of the tube. The energy consumption of hot and cold rolling is 5.3 MJ/kg according to Worell [13]. The exergy destruction is assumed to be equal to the energy consumption, because the exergy of the material is hardly changed. The exergy destruction of the bending of steel has been estimated by assuming that metal is heated to 900°C by a gas heater. The exergy destruction associated with the force required to bent the steel has been neglected. The exergy destruction associated with welding of steel tubes has been estimated to be 0.260 MJ per meter (Cw,S) according to the Dutch steel maker, Hoogovens IJmuiden. The energy consumption associated with the manufacturing of the copper tubes, which includes the welding of copper tubes, is 14.7 MJ/kg (Cman,Cu) according to Alvarado-Grandi [14]. The exergy destruction is assumed to be equal to the energy consumption, because the exergy of the material is hardly changed during the manufacturing. The exergy destruction of the production of each material is shown in Table 1.

Table 1: Exergy destruction associated with the production of material and manufacturing of tubes

Process	IS (MJ/kg)	ICu (MJ/kg)
Primary process	10.5	60
Secondary process	4.4	20
Manufacturing process	5.7	15

The heat exchanger outer tube is insulated by polyurethane (PUR) foam with a thickness of 0.025 m. The cumulative energy consumption for the production of PUR with a density of 30 kg/m^3 is 98 MJ/kg according to Kindler et al [15]. The exergy content is estimated to be 27 MJ/kg on basis of the lower heating value according to Kindler et al [15]. So the cumulative exergy destruction is 71 MJ/kg (CPUR). No recycling of the PUR has been assumed.

The exergy destruction associated with the manufacture of the heat exchangers is due to the production of copper tube and steel tube, welding and the production of the PURfoam.

Which t is the operating time of the heat exchanger during its life cycle and x is the recycling ratio which is the proportion of secondary material, i.e. material which is recycled. For the calculations of the amount of material the mean diameters of the tubes has been taken.

$$\dot{I}_{man} = \frac{M_{Cu}C_{Cu} + M_S C_S + LC_{w,S} + M_{PUR}C_{PUR}}{t} =$$
$$\frac{\pi L}{t}\left[(D_1 + 0.5d_1)d_1\rho_{Cu}\left(C_{man,Cu} + x_{Cu}C_{\sec,Cu} + (1 - x_{Cu})C_{pri,Cu}\right)\right]$$
$$+\frac{\pi L}{t}\left[(D_2 + 0.5d_2)d_2\rho_S\left(C_{man,S} + x_S C_{\sec,S} + (1 - x_s)C_{pri,S}\right) + C_{w,S}\right]$$
$$+\frac{1}{t}\left[(L\cdot(D_2 + 2\cdot d_2 + 0.025)\cdot\pi)\cdot 0.025\rho_{PUR}C_{PUR}\right] \tag{17}$$

3. RESULTS

From the above considerations an expression was obtained for the total life cycle irreversibility, which has to be minimised. Where I_{oper} and I_{man} are stated in (16) and (17), respectively.

$$\dot{I}_{LC} = \dot{I}_{oper} + \dot{I}_{man} \tag{18}$$

The following operating parameters have been assumed for the heat exchanger. The mass flow of the district heating water as the domestic tap water is 0.1 kg/s. The incoming temperature of the cold domestic tap water is 15°C. The domestic tap water is heated to 65°C. So, the exergy increase of the domestic tap water is 1628 W. The temperature of the incoming district heating water is variable. The environmental temperature, T_0, is 25°C. The operating time for the heat exchanger is 30 minutes a day on full load for 10 years. The mean temperature of the inlet and outlet streams is used for the heat capacity, viscosity and thermal conductivity of water. The recycling ratio (x) is set to be 0.9 for the copper and steel parts of the tube.

3.1 Results of optimization

The minimisation of I_{LC} for the 4 variables, D_1, D_2, L and n is given in Table 2 for the wide and dense net. The number of inner tubes is limited to require the condition of turbulent operation. The value of the minimum life cycle irreversibility for the heat exchanger in the wide net of the district heating system is 7.24·102 W. The optimum geometrical parameters were found to be D_1 = 2.88·10-3 m, D_2 = 2.15·10-2 m, L = 25.5 m and n =15. The number of inner tubes (n) has been limited to 15, because otherwise there is no turbulent flow anymore. The T, P_1 and P_2 of this optimised heat exchanger are 1.42 K, 1.8 bar and 2.1 bar, respectively. Where T is the temperature difference between the hot and cold stream, which is constant, because we have a balanced counter flow heat exchanger. The dense net life cycle irreversibility is 5.90·102 W for D_1 = 2.88·10-3 m, D_2 = 2.15·10-2 m, L = 20.8 m and n = 15. The T, P_1 and P_2 of the optimised heat exchanger are 1.75 K, 1.5 bar and 1.7 bar, respectively.

The optimal geometrical parameters of the tube diameters are independent of the efficiency of the distribution net, the dense or wide net. However, the length of the tube is strongly dependent on the type of distribution net. The components of the life cycle irreversibility in the heat exchanger are displayed in Table 2.

From the table 2 it can seen that the irreversibility due to the thermal component is reduced by increasing the irreversibilities due to the mechanical and manufacture component. The contribution of the use of copper, steel and PUR-foam to the irreversibility associated with the manufacture is 67%, 20% and 13% for the wide and dense net optimisation and 47%, 33% and 20% for the both heat exchangers with a length of 6.4 meter, respectively.

Table 2: Components of life cycle irreversibility in watts

Component	Widenet (KVM)	Densenet (KVM)	Widenet (minimized)	Densenet (minimized)
Thermal	6604	4392	363	296
mechanical	2	2	112	91
manufacture	47	47	249	203
Total	6653	4441	724	590

4. CONCLUSIONS

With the use of the concepts of an ELCA the optimal design of a heat exchanger can be obtained. For all systems where there is a trade-off between exergy saving during operation and exergy destruction during construction of the system the minimisation of life cycle irreversibility should be used to get the true optimum from the point of view of conservation of exergy reservoir of natural resources. In the case under study the optimal design parameters of the heat exchangers are obtained under the specified conditions. The dense net, which is a more energy efficient heat supply system than the wide net, has the same inner tubes diameters as the wide net while the length of the heat exchanger is smaller for the former than for the latter. The decrease of the number of inner tubes leads to a relatively small increase of the irreversibility. The increase of the operating time leads to a slight increase in the inner tubes diameters and a greater increase in the length of the heat exchanger. In the optimised situation the life cycle irreversibility is more uniformly distributed between the component irreversibilities than in the fixed length of 6.4 meters situation. In general we can conclude that the thermodynamic optimisation of the design parameters of a subsystem is dependent of the thermodynamic efficiency of the whole system and that the different components of the life cycle irreversibility of heat exchangers are more uniformly distributed when there are fewer restrictions on the design parameters for the optimisation.

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