PERGAMON

International Journal of Heat and Mass Transfer 42 (1999) 951–959

International Journal of HEAT and MASS TRANSFER

Thermodynamic optimisation of a heat exchanger

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Received 11 June 1997; in final form 16 February 1998

Abstract

The objective of this paper is to show that for the optimal design of an energy system, where there is a trade-off between exergy saving during operation and exergy use during construction of the energy system, exergy analysis and life cycle analysis should be combined. An exergy optimisation of a heat exchanger has been carried out on the basis of the life cycle analysis method in this paper. The analysis of the heat exchanger in which exergy analysis and life cycle analysis are combined gives the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.

Nomenclature

A heat transfer area
c_p heat capacity
C cumulative exergy destruction
d thickness of the tube wall
D inner diameter of the tube
I irreversibility of exergy destruction
L length of the tubes
$\dot{m}$ mass flow
M mass
Nu Nusselt number
NTU number of heat transfer units
P pressure
Pr Prandtl number
Re Reynolds' number
R diameter of the tube curving
T temperature
T_1 inner tube temperature
T_2 outer tube temperature
$\bar{u}$ mean velocity of the fluid in the tube
x recycling ratio of the material.

Greek symbols
α heat transfer coefficient
ε effectiveness
λ thermal conductivity
μ dynamic viscosity
ρ density.

Subscripts
0 environmental
1 inner tube of the heat exchanger
2 outer tube of the heat exchanger
Cu copper
h hydraulic
S steel
in inlet
LC life cycle
man manufacturing
mat material
oper operating
out outlet
pri primary
PUR polyurethane foam
sec secondary
tot total
w welding.

Superscripts
ΔP mechanical component
ΔT thermal component.

1. Introduction

Exergy analysis and life cycle analysis have been developed separately. Exergy analysis has been described

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PII: S0017-9310(98)00118-5

extensively in the books of Kotas [1] and Szargut [2]. Life cycle analysis (LCA) has been described by Consoli et al. [3] and Heijungs et al. [4]. The methodology in the LCA includes the effects of all the phases of the production, use and recycling on the environment. In this paper the methodology has been performed using only one criterion which is the minimisation of the life cycle irreversibility associated with the delivery of domestic hot water. The complete LCA involves other factors, e.g. pollution of air and water, noise, etc., which were not considered here. The concept of cumulative exergy, introduced by Szargut, uses the method of accumulation of the exergy consumption to a defined point in the life cycle analysis [2]. The cumulative exergy consumption of a product takes into account all the exergy destruction for the manufacture of the product. However, in this method the exergy destruction associated with the disposal of the product and the influence of recycling which causes a change in the exergy destruction are not taken into account. Because of its widespread use the heat exchanger has been selected as an example.

Bejan [5] studied extensively the optimisation of a heat exchanger, excluding exergy destruction associated with use of materials and cumulative exergy of 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. [6]. They took into account the cumulative exergy of the material, but did not include the irreversibility due to the pressure drops.

Tondeur and Kvaalen [7] 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. [8, 9] has shown that in the case of an optimal heat exchanger the thermal and viscous contributions to the entropy generation should be equal when the heat flux is optimised. It is shown that the ratio of thermal and viscous contribution to the entropy generation 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.

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. The inner tubes carry the cold stream and the surrounding outer tube carries the hot stream. 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 combined annular heat exchanger is helically wound as shown in Fig. 1. The influence of winding on the heat transfer has been neglected. 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:

$$\dot{I} = \dot{I}^{\Delta T} + \dot{I}^{\Delta P} \tag{1}$$

with

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

and

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

Using the heat balance for the n inner tubes

$$n \cdot \dot{m}_n c_\mathrm{p} (T_{1,\mathrm{out}} - T_{1,\mathrm{in}}) = \alpha A \Delta T \tag{2}$$

with $\dot{m}_n(\dot{m}/n)$ equation (1) can be rewritten as

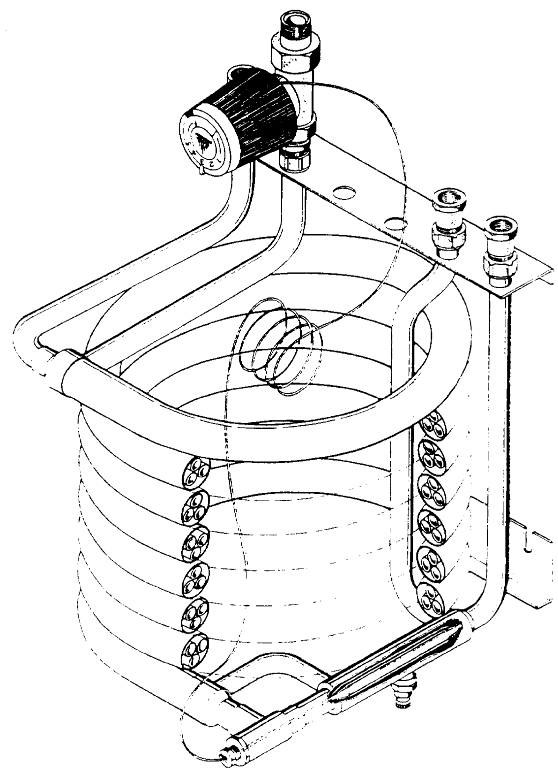

Fig. 1. The analysed heat exchanger.

$$\dot{I}^{\Delta T} = T_0 \cdot \dot{m}c_p \times \left[\ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + \ln \frac{T_{1,\text{in}} + \dfrac{\dot{m}c_p(T_{1,\text{out}} - T_{1,\text{in}})}{\alpha A}}{T_{1,\text{out}} + \dfrac{\dot{m}c_p(T_{1,\text{out}} - T_{1,\text{in}})}{\alpha A}} \right] \tag{3}$$

because $T_{2,\text{out}} = T_{1,\text{in}} + \Delta T$ and $T_{2,\text{in}} = T_{1,\text{out}} + \Delta T$ in a balanced heat exchanger.

By using the force balance we can write for the pressure drops in the tubes:

$$\Delta P_1 = P_{1,\text{in}} - P_{1,\text{out}} = 2f_1(Re) \cdot \rho \cdot \bar{u}_1^2 \frac{L}{D_1} \tag{4a}$$

and

$$\Delta P_2 = P_{2,\text{in}} - P_{2,\text{out}} = 2f_2(Re) \cdot \rho \cdot \bar{u}_2^2 \frac{L \cdot (D_2 + n \cdot (D_1 + 2d_1))}{(D_2^2 - n \cdot (D_1 + 2d_1)^2)}. \tag{4b}$$

Using

$$\bar{u}_1 = \frac{4\dot{m}_n}{\rho \pi D_1^2} \tag{5a}$$

and

$$\bar{u}_2 = \frac{4\dot{m}}{\rho\pi[D_2^2 - n \cdot (D_1 + 2d_1)^2]} \tag{5b}$$

for the mean velocities of the fluid in the tubes and substituting equation (4a) and (4b) in equation (1b) yields

$$\dot{I}^{\Delta P} = \frac{32}{\pi^2} \left[f_1(Re) \frac{n \cdot \dot{m}_n^3}{\rho^2} \frac{L}{D_1^5} + f_2(Re) \frac{\dot{m}^3}{\rho^2} \frac{L \cdot (D_2 + n \cdot (D_1 + 2d_1))}{[D_2^2 - n \cdot (D_1 + 2d_1)^2]^3} \right]. \tag{6}$$

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 according to Chapman [10]

$$Nu_i = \frac{\alpha_i D_h}{\lambda} = 0.023 \cdot Re^{0.8} \cdot Pr^n = 0.023 \cdot \left(\frac{\rho \bar{u} D_h}{\mu} \right)^{0.8} \cdot \left(\frac{c_p \mu}{\lambda} \right)^n \tag{7}$$

with $n = 0.3$ or 0.4 for cooling or heating, respectively. Experimental data of Kays and London [11] gives a similar relation.

The friction factor in tubes according to the friction law of Blasius is given in Rogers and Mayhew [12] as

$$f(Re) = \frac{0.0791}{Re^{0.25}} = 0.0791 \cdot \left(\frac{\mu}{\rho \bar{u} D_h} \right)^{0.25}. \tag{8}$$

However, to include the curving of the tube the following correction factor according to Ito [13] has to be included in equation (8)

$$C = 0.962 + 0.092 \cdot Re^{0.25} \left(\frac{D}{R} \right)^{1/2} \tag{9}$$

where R is the diameter of the curved tubes. Substituting equations (5), (7) and (8) in equation (6), using $A = n \cdot \pi \cdot L \cdot D_1$ for the inner tubes and $A = n \cdot \pi \cdot L \cdot (D_1 + 2d_1)$ for the outer tubes and neglecting the heat resistance of the tube wall we obtain

$$\dot{I}^{\Delta T} = T_0 \cdot \dot{m}c_p \left[\ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + \ln \left(\frac{T_{1,\text{in}} + \Delta T}{T_{1,\text{out}} + \Delta T} \right) \right] \tag{10a}$$

with

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

and

$$\dot{I}^{\Delta P} = \frac{1.79}{\pi^{1.75}} \frac{\mu^{0.25}}{\rho^2} \left[\frac{n \cdot \dot{m}_n^{2.75} L}{D_1^{4.75}} \times \left(0.962 + 0.092 \cdot \left(4 \cdot \frac{m_n}{\pi \mu D_1} \right)^{0.25} \left(\frac{D_1}{R} \right)^{1/2} \right) + \frac{\dot{m}^{2.75} L \cdot (D_2 + n(D_1 + 2d_1))^{1.25}}{[D_2^2 - n(D_1 + 2d_1)^2]^3} \times \left(0.962 + 0.092 \cdot \left(\frac{4 \cdot m}{\pi \mu (D_2 - n \cdot (D_1 - 2d_1))} \right)^{0.25} \times \left(\frac{D_2}{R} \right)^{1/2} \right) \right]. \tag{10b}$$

2.3. Comparison with experimental results

The used formulas are compared with experimental results of the manufacturer of the heat exchanger [14], which is a heat exchanger with three inner tubes. However, no information about balanced operations are available, so the formulas have to be rewritten for the unbalanced situation.

For the unbalanced counter flow heat exchanger equation 1(a) is rewritten as

$$\dot{I}^{\Delta T} = T_0 \left[C_1 \ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + C_2 \ln \frac{T_{2,\text{out}}}{T_{2,\text{in}}} \right] \tag{11}$$

where $C_1 = \dot{m}_1 c_p$, $C_2 = \dot{m}_2 c_p$ and with help of the energy balance

$$T_{2,\text{out}} = T_{2,\text{in}} - \frac{C_1}{C_2}(T_{1,\text{out}} - T_{1,\text{in}}) \tag{12}$$

and the effectiveness

$$\varepsilon = \frac{C_1(T_{1,\mathrm{in}} - T_{1,\mathrm{out}})}{C_{\min}(T_{1,\mathrm{in}} - T_{2,\mathrm{in}})} \quad (13)$$

where $C_{\min} = C_1$, when $C_2 > C_1$ and $C_{\min} = C_2$, when $C_2 < C_1$.

$T_{2,\mathrm{out}}$ and $T_{2,\mathrm{in}}$ can be replaced as shown in equations (13) and (12) to rewrite equation (11) as

$$\dot{I}^{\Delta T} = T_0\left[C_1 \ln\frac{T_{1,\mathrm{out}}}{T_{1,\mathrm{in}}} + C_2 \ln\left(1 - \frac{C_1}{C_2}\left(\frac{C_1}{\varepsilon \cdot C_{\min}} + \frac{T_1}{(T_{1,\mathrm{out}} - T_1)}\right)^{-1}\right)\right] \quad (14)$$

with

$$\varepsilon = \frac{1 - \mathrm{e}^{-NTU \cdot x}}{1 - \dfrac{C_{\min}}{C_{\max}}\mathrm{e}^{-NTU \cdot x}}$$

where $NTU = \alpha A/C_{\min}$ and $x = 1 - C_{\min}/C_{\max}$. NTU is calculated with the help of equation (7). For the pressure losses equation (10b) can be used by filling in the correct mass. The irreversibility of experimental results calculated with equation (11) is compared with the irreversibility calculated from the theory with equation (14) in Table 1 for $\dot{m}_1 = 4$ and $\dot{m}_2 = 5$ l/min. For the mechanical losses a balanced operation of $\dot{m}_1 = \dot{m}_2 = 7$ l/min is taken.

As can be seen in the case of the thermal component the experimental and thermal results match perfectly, while in the case of the mechanical losses there is a big difference. Taking into account the five bendings of the inlet and outlet tubes of the heat exchanger the theoretical values has a marginal effect. So they are left out. Further contact with the manufacturers has shown that the experimental values for the pressure drops are mean values and there are great deviations between different heat exchangers. Probably, they are the result of irrigual curvings of the inner tubes. The theoretical values are used in this article.

2.4. 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 the required heat and power is associated with exergy destruction, which has to be taken into account in the optimisation, according to the LCA. We assume the following situation, based on the district heating system in Enschede, The Netherlands. In this system a cogeneration heat and power plant, a steam and gas turbine plant (STAG), is used, which has an exergetic efficiency of 50%, when there is no useful heat production. The exergetic efficiency of the extraction type of plant stays nearly constant when a part of the steam is used for the district heating system. 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[1] (wide net) and 0.75 for a very dense distribution net[1] (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. The exergetic efficiency of the pumps is assumed to be 0.7.

Hence the exergetic cost, which is the amount of exergy which is needed for the production of one exergy unit, can be calculated for the analysed heat exchanger of pressure rise, k_{P}, and heat, k_{T}. The exergetic cost for the pressure rise (k_{p}) is $k_{\mathrm{power}} \cdot k_{\mathrm{pumps}} = 2.85$, where $k_{\mathrm{power}} = 2$ and $k_{\mathrm{pumps}} = 1.43$. These are calculated by taking the inverse of the exergetic efficiency of power generation and pumping. The exergetic cost for the heat (k_{T}) 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

$$\dot{I}_{\mathrm{oper}} = k_{\mathrm{T}}\dot{I}^{\Delta T} + k_{\mathrm{P}}\dot{I}^{\Delta P}. \quad (15)$$

2.5. Irreversibilities associated with the use of the material

The life cycle flow diagram of the heat exchanger is displayed in Fig. 2 the exergy destruction in each process is shown in Table 2.

A detailed exergy analysis of the primary anode copper process is presented by Kolenda et al. [15]. The cumu-

Table 1
Comparison of irreversibilities obtained by experimental results and theory in W

Component	Experimental	Theoretical
Thermal	1039 ± 30	1031
Mechanical	1.8 ± 0.1	1.0

[1] This is a hypothetical situation. In reality the exergy losses are higher because the peak in heat demand is supplied by auxiliary heating boilers using natural gas. Their exergetic efficiency is very poor.

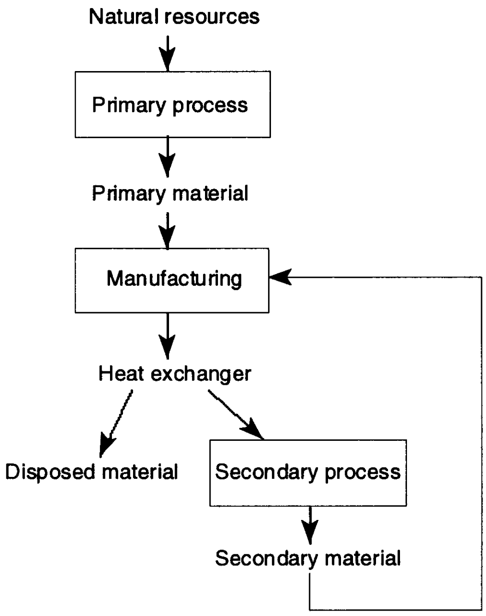

Fig. 2. Life cycle of the heat exchanger.

Table 2
Exergy destruction associated with the production of material and manufacturing of tubes

Process	I_S (MJ kg^{-1})	I_{Cu} (MJ kg^{-1})
Primary process	10.5	60
Secondary process	4.4	20
Manufacturing Process	5.7	15

lative irreversibility of the flash smelting process, one of the main copper production processes, is shown to be 45.8 MJ kg^{-1} anode copper. However, the exergy destruction consists of the exergy dissipation in the process and the exergy of materials, which are assumed to be discharged and dissipated into the environment, like slag and combustion gasses. So the total cumulative exergy destruction is 55.6 MJ per kg primary anode copper.

No recent publication about the exergy destruction of the production of primary steel, the production of copper out of anode copper, the production of secondary copper and manufacturing of steel and copper tubes is available, so instead of the exergy destruction the energy consumption has been taken. To use the energy consumption instead of the exergy destruction due to lack of sufficient data, the following conditions have to be fulfilled. Most inputs ought to be raw materials, because the exergy and enthalpy content are then roughly 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^{-1} copper according to Boustead and Hancock [16]. The exergy increase is almost zero and we assume the enthalpy values of the inputs to be equal to the exergy values, so this value is taken for the exergy destruction. The cumulative energy destruction for primary copper becomes $55.6+4.5 = 60.1$ MJ kg^{-1} ($C_{pri,Cu}$).

The energy use for the production of primary steel slabs is 16.9 MJ kg^{-1} [2] according to Worrell et al. [17]. 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 from the iron oxide to the steel. The exergy increase is estimated to be 6.4 MJ kg^{-1}, so the exergy destruction is 10.5 MJ kg^{-1} ($C_{pri,S}$). The cumulative exergy destruction of secondary steel is calculated from Wall [18] 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 [16] give two totally different values for the energy consumption of the secondary copper production, namely 7.2 and 48.8 MJ kg^{-1}. The exergy destruction of this process is taken to be 20 MJ kg^{-1} ($C_{sec,Cu}$).

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^{-1} according to Worell et al. [17]. 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 bend the steel has been neglected. The exergy destruction associated with welding of steel tubes has been estimated to be 0.260 MJ per meter ($C_{w,S}$) 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^{-1} ($C_{man,Cu}$) according to Alvarado-Grandi et al. [19]. The exergy destruction is assumed to equal the energy consumption, because the exergy of the material is hardly changed during the manufacturing.

The heat exchanger is insulated by surrounding the outer tube by polyurethane foam (PUR) with a thickness of 0.025 m. The cumulative energy consumption for the

[2] The energy use is based on the steel factory of Hoogovens IJmuiden in the Netherlands, which can be considered to be one of the most energy efficient steel factories in the world.

production of PUR with a density of 30 kg m^{-3} is 98 MJ kg^{-1} and the exergy content is estimated to be 27 MJ kg^{-1} on the basis of the lower heating value according to Kindler and Nikles [20]. So the cumulative exergy destruction is 71 MJ kg^{-1} (C_{PUR}). 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 PUR-foam.

$$\begin{aligned}\dot{I}_{\mathrm{man}} &= \frac{M_{\mathrm{Cu}}C_{\mathrm{Cu}}+M_{\mathrm{Fe}}C_{\mathrm{S}}+LC_{\mathrm{w,S}}+M_{\mathrm{PUR}}C_{\mathrm{PUR}}}{t} \\ &= \frac{\pi L}{t}[(D_1+0.5d_1)\,d_1\rho_{\mathrm{Cu}}(C_{\mathrm{man,Cu}}+x_{\mathrm{Cu}}C_{\mathrm{sec,Cu}} \\ &\quad +(1-x_{\mathrm{Cu}})C_{\mathrm{pri,Cu}})]+\frac{\pi L}{t}[(D_2+0.5d_2)d_2\rho_{\mathrm{S}}(C_{\mathrm{man,S}} \\ &\quad +x_{\mathrm{S}}C_{\mathrm{sec,S}}+(1-x_{\mathrm{S}})C_{\mathrm{pri,S}})+C_{\mathrm{w,S}}] \\ &\quad +\frac{1}{t}[(L\cdot(D_2+2\cdot d_2+0.025)\cdot\pi)\cdot 0.025\rho_{\mathrm{PUR}}C_{\mathrm{PUR}}]\end{aligned} \tag{16}$$

in 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 calculation of the amount of material the mean diameters of the tubes has been taken.

3. Results

From the above considerations we obtain an expression for the total life cycle irreversibility, which has to be minimised.

$$\dot{I}_{\mathrm{LC}} = \dot{I}_{\mathrm{oper}}+\dot{I}_{\mathrm{man}} \tag{17}$$

where $\dot{I}_{\mathrm{oper}}$ and $\dot{I}_{\mathrm{man}}$ are stated in (15) and (16), respectively.

The following operating parameters have been assumed for the heat exchanger. The mass flow of the district heating water as the domestic water is 0.1 kg s^{-1}. 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 flow 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 min a day on full load for 10 years. The mean temperature of the inlet and outlet streams is used for the heat capacity, viscosity[3] and thermal conductivity of water. The recycling ratio is set to be 0.9 for the copper and steel parts of the tube.

3.1. Reference configuration

As a reference situation the SP 15 N is taken [14]. This domestic water heat exchanger is commercially available. It has three inner tubes with an inner diameter of $8.4\cdot10^{-3}$ m and an outer diameter of $2.77\cdot10^{-2}$ m. The length is 6.4 m. The wall thickness of the inner and outer tube are $0.8\cdot10^{-3}$ and $2\cdot10^{-3}$ m, respectively. As shown in Table 3, the life cycle irreversibility of the heat exchanger in full operation is $6.65\cdot10^3$ W for the wide net and $4.44\cdot10^3$ W for the dense net. Resulting from a ΔT, ΔP_1 and ΔP_2 of 28.2 K, 0.049 bar and 0.014 bar, where ΔT is the temperature difference between the hot and cold stream, which is constant, because we have a balanced counter flow heat exchanger.

3.2. Results of optimisation

The minimisation of $\dot{I}_{\mathrm{LC}}$ for the four variables, D_1, D_2, L and n is given in Table 3 for the wide and dense net. The value of the number of inner tubes is limited to require the condition of turbulent region operation.

The minimum value of life cycle irreversibility per heat exchanger in the wide net of the district heating system is $7.24\cdot10^2$ W. The optimum geometrical parameters were found to be $D_1 = 2.88\cdot10^{-3}$ m, $D_2 = 2.15\cdot10^{-2}$ m, $L = 25.5$ m and $n = 15$. n has been limited to 15, because otherwise the turbulent flow regime would be left. The ΔT, ΔP_1 and ΔP_2 of this optimised heat exchanger are 1.42 K, 1.8 and 2.1 bar, respectively.

The minimum life cycle irreversibility of the dense net is $5.90\cdot10^2$ W for $D_1 = 2.88\cdot10^{-3}$ m, $D_2 = 2.15\cdot10^{-2}$ m, $L = 20.8$ m and $n = 15$. The ΔT, ΔP_1 and ΔP_2 of the optimised heat exchanger are 1.75 K, 1.5 and 1.7 bar, respectively.

We see that for optimal geometrical parameters 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

Table 3
Components of life cycle irreversibility in W

Component	Wide net (SP 15 N)	Dense net (SP 15 N)	Wide net	Dense net
Thermal	6604	4392	363	296
Mechanical	2	2	112	91
Manufacture	47	47	249	203
Total	6653	4441	724	590

[3] The value of the viscosity is strongly temperature dependent. The viscosity at 15, 70 and 80°C is $1.14\cdot10^{-3}$ Pa s, $0.406\cdot10^{-3}$ Pa s and $0.357\cdot10^{-3}$ Pa s, respectively. So the assumption of the mean temperature will cause a deviation from the real situation.

distribution net. The components of the life cycle irreversibility in the heat exchanger are displayed in Table 3.

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 both heat exchangers with length of 6.4 meters, respectively.

3.3. Discussion

The effect of the tube diameters and the length on life cycle irreversibility for the optimal geometrical parameters is shown in Fig. 3. The cross-sections of flow areas of the inner and outer passages are set fixed in the ratio of 1 : 0.764 to get a three-dimensional figure. This ratio has been obtained for the optimisation of the wide net with $n = 15$.

In Fig. 4 it can be seen that the life cycle irreversibility rate rises for smaller tube length and inner tube diameter. At zero length or at zero tube diameter the life cycle irreversibility rate becomes of course infinite. When the length or inner diameters are reduced to save material costs it is shown that there is an optimal relation between these two geometrical parameters to do so.

A decrease of the number of inner tubes gives the following optimal geometrical parameters for the minimal irreversibility as can be seen in Table 4.

An increase of the operating time of the heat exchanger leads to optimal geometrical parameters with greater diameters and a longer heat exchanger as shown in Table 5. It can be seen that an increase of the operating time leads to a relatively small increase of the tube diameters D_1, D_2 and a greater increase of the length. The life cycle irreversibility rate decreases steadily by an increase of the operating time. There are no significant differences between the wide and dense net for the relation operating time and optimal geometrical parameters.

De Oliveira et al. [8] gives for the exergetic optimisation a ratio of 3 : 3.5 for the exergy destruction due to the pressure drop to the exergy destruction due to the tem-

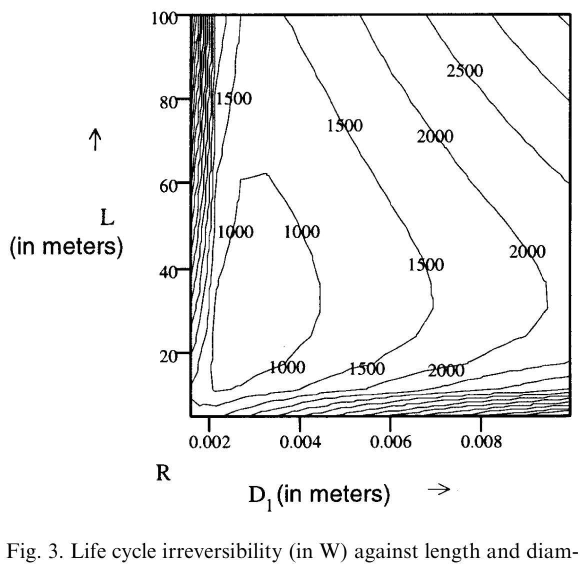

Fig. 3. Life cycle irreversibility (in W) against length and diameter of the heat exchanger.

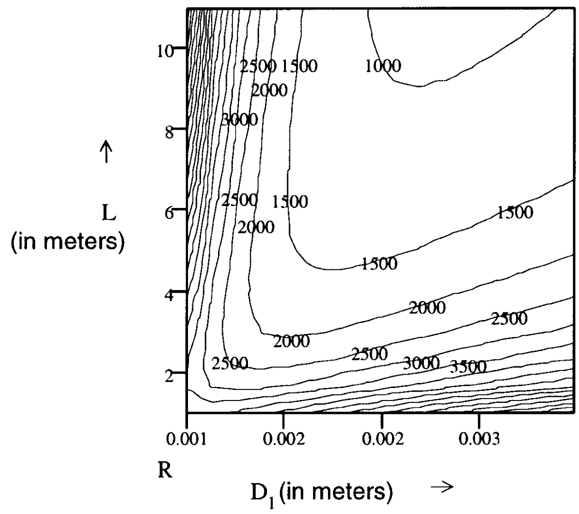

Fig. 4. Life cycle irreversibility (in W) against length and diameter of the heat exchanger in more detail.

Table 4
Optimal geometrical parameters and life cycle irreversibility for the wide net for different number of tubes

n (tubes)	$\dot{I}_{LC}$ (W)	D_1 (10^{-3} m)	D_2 (10^{-2} m)	L (m)
1	1059	10.5	1.69	90.6
2	943	7.58	1.75	67.4
3	887	6.25	1.81	56.6
5	825	4.90	1.89	44.0
10	757	3.50	2.04	31.6
15	724	2.88	2.16	25.6

Table 5
Optimal geometrical parameters and life cycle irreversibility for the wide net for different times

t (years)	$\dot{I}_{LC}$ (W)	D_1 (10^{-3} m)	D_2 (10^{-2} m)	L (m)
5	920	2.53	1.97	18.0
10	724	2.88	2.16	25.6
15	630	3.10	2.28	36.1
20	570	3.28	2.38	36.7

perature degradation. This ratio is 3.3 for the dense net optimisation, which has about the same exergetic cost factor for the heat and the pressure rise.

The number of inner tubes is limited to stay in the turbulent flow regime. The inner tubes leaves the turbulent flow regime above $n = 20$. The outer tube leaves the turbulent at $n = 15$, which is set as the upper limit. However, a correlation could be made to increase the allowable number of inner tubes to 20 by decreasing the outer tubes diameter to stay in the turbulent flow regime.

The entrance and exit pressure drops are neglected in this study. In the case of the wide net optimisation the entrance pressure drop is calculated to be lower than $6 \cdot 10^{-3}$ bar.

No heat transfer from the heat exchanger to the environment has been assumed. For the wide net optimisation the heat transfer to the environment during operation has been calculated to be 0.42% of the total amount of heat exchanged, when for the heat exchanger box an outside and a mean inside temperature of 20 and 40°C are assumed, respectively. So the neglect of the heat transfer from the heat exchanger box to the environment during operation is justified. However, if you assume a constant temperature of 40°C for all times the heat transfer to the environment becomes 20.2%. But, if the heat exchanger is located in a closed space, the temperature will increase, which decreases the heat transfer, so more attention should be devoted to this subject. A constant temperature in the heat exchanger at all times is desirable because of comfort conditions. The coefficient of thermal conduction is 0.024 W m^{-1} K^{-1} for the PUR and the heat transfer coefficient for the dry air is taken to be 10 W m^{-2} K^{-1}.

The transformation of mechanical energy into thermal energy has been neglected in the model. As can be calculated for the optimised situation for the wide net, which results in 15 inner tubes, this transformation would cause a temperature increase of 0.043 K for the cold side and 0.050 K for the hot side. Assuming a temperature increase of 0.045 K due to this transformation gives an associated exergy increase of 2.7 W, this would decrease the irreversibility due to the heat transfer with 11.0 W as the effect of the decrease of required thermal energy.

4. Conclusions

With the combination of exergy analysis and life cycle analysis the optimal design of a heat exchanger can be obtained. For all energy systems where there is a trade-off between exergy saving during operation and exergy use during construction of the energy system this method should be adopted to get the true optimum from the point of view of conservation of exergy 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 tube diameters as the wide net whilst the length of the heat exchanger is smaller for the former than for the latter. The number of inner tubes is maximal for both nets under the given conditions. A decrease of the number of inner tubes leads to an increase of the irreversibility. The dense net has lower life cycle irreversibility due to the manufacture of the heat exchanger compared to the wide net, because less exergy is saved by the same increase of exergy use due to the manufacture. 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 on 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 less restrictions on the design parameters for the optimisation.

Acknowledgement

We would like to thank Dr T. J. Kotas of Queen Mary and Westfield College, University of London, U.K., for his help and advice on this paper.

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