EXERGETIC LIFE CYCLE ANALYSIS OF COMPONENTS IN A SYSTEM

The optimisation of a heat exchanger in a district heating system

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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. The two methods are often used separately, but a limited number of studies has been carried out in which they are combined in some way. An exergy optimisation of an heat exchanger has been carried out on the basis of the life cycle analysis method in this paper. The optimisation takes into account irreversibilities due to frictional pressure drops and the temperature difference between the hot and cold stream and irreversibilities due to the production of the materials and the construction of the heat exchanger. As example of this type of heat exchanger a water to water heat exchanger in a city 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 exergy analysis and life cycle analysis are combined gives the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.

1. Introduction

Exergy analysis and life cycle analysis have been developed separately. Exergy analysis has been described 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 latter one gives a detailed methodology to use in a LCA. 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 (Szargut et al. [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 cause changes in the exergy destruction are not taken into account.

The ELCA method is described by Cornelissen [5]. A comparison between the LCA and the ELCA is performed for the porcelain mug and the disposable polystyrene cup. It is shown that the life cycle irreversibility, the exergy loss during the life cycle, is the parameter for the depletion of natural resources. Furthermore, the general results of the ELCA and the LCA were similar. More research has to be performed to show in which cases this takes place. It is expected that in energy intensive processes or component operating in these processes the results of the ELCA and the LCA are similar [6]. However, the ELCA is less time consuming and gives more insight in the location of the losses. On basis of this knowledge better improvements can be suggested.

Because of its widespread use the heat exchanger has been selected as an example. Bejan [7] 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. [8]. They took into account the cumulative exergy of the material, but did not include the irreversibility due to the pressure drops.

Tondeur and Kvaalen [9] 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.

In this article the optimal design of a component, a heat exchanger, has been obtained on basis of the concepts of ELCA. This means the minimisation of the life cycle irreversibility of the heat exchanger.

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 city heating system to heat the domestic tap water. The inner tube carries 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 tube have been constructed from copper and steel, respectively. The combined annular heat exchanger is helically wound as shown in Figure 1. The influence of winding on the pressure drop has been neglected. The thermal insulation of the heat exchanger has been assumed to be perfect.

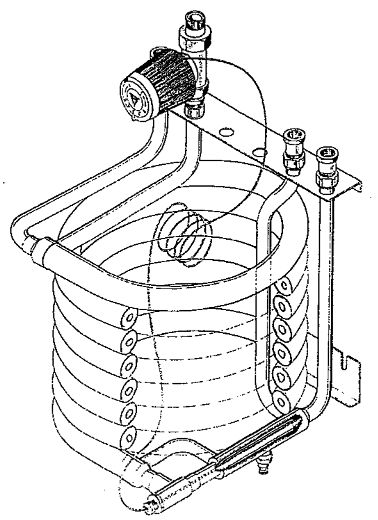


Figure 1. The analysed heat exchanger

2.1.1 Theory

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

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

and

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

The heat exchanger effectiveness is:

$$e=\frac{T_{1,in}-T_{1,out}}{T_{1,in}-T_{2,in}}=\frac{T_{2,in}-T_{2,out}}{T_{1,in}-T_{2,in}} \tag{2}$$

In a nearly ideal heat exchanger limit $N_{tu}\rangle\rangle$ and therefore $1-e\approx\frac{1}{N_{tu}}$, where $N_{tu}=\frac{\alpha A}{\dot{m}c_p}$. Neglecting the heat resistance of the tube wall we have according to Bejan [5]

$$\dot{I}^{\Delta T}=T_0\left[\frac{\dot{m}^2 c_p^2\tau^2}{\alpha_1 A}+\frac{\dot{m}^2 c_p^2\tau^2}{\alpha_2 A}\right]=T_0\cdot\dot{m}c_p\left[\frac{\tau^2}{N_{tu,1}}+\frac{\tau^2}{N_{tu,2}}\right] \tag{3}$$

with $\tau=\frac{|T_{2,in}-T_{1,in}|}{\sqrt{T_{1,in}T_{2,in}}}$.

By using the force balance inside the tube(s) we can write for the pressure drops:

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

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

Using $\bar{u}_1=\frac{4\dot{m}}{\rho\pi D_1^2}$ and $\bar{u}_2=\frac{4\dot{m}}{\rho\pi\left[D_2^2-(D_1+2d_1)^2\right]}$ and substituting (4a) and (4b) in (1b) yields

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

or in the rewritten form

$$\dot{I}^{\Delta P}=2\pi\mu\left[f_1(\mathrm{Re})\cdot\mathrm{Re}\,\bar{u}_1^2 L+f_2(\mathrm{Re})\cdot\mathrm{Re}\,\bar{u}_2^2\frac{L\cdot(D_1+2d_1+D_2)}{D_{h,2}}\right]$$

with $D_{h,2} = D_2 - D_1 - 2d_1$

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]

$$Nu = \frac{\alpha D_h}{\lambda} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^n = 0.023 \cdot \left(\frac{\rho \bar{u} D_h}{\mu}\right)^{0.8} \cdot \left(\frac{c_p \mu}{\lambda}\right)^n \tag{6}$$

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

$$f(\mathrm{Re}) = \frac{0.0791}{\mathrm{Re}^{0.25}} = 0.0791 \cdot \left(\frac{\mu}{\rho \bar{u} D_h}\right)^{0.25} \tag{7}$$

Substituting (6) and (7) in (5) and using $A \approx \pi \cdot L \cdot D_1$ we obtain

$$\dot{I}^{\Delta T} = T_0 \frac{14.3\,\dot{m}^{1.2}}{\pi^{0.2}} \left[\frac{c_p^{1.6} \tau^2\ \mu^{0.4}}{\lambda^{0.6}} \frac{D_1^{0.8}}{L} + \frac{c_p^{1.7} \tau^2\ \mu^{0.5}}{\lambda^{0.7}} \frac{(D_2 + D_1 + 2d_1)^{0.8} D_{h,2}}{L \cdot D_1} \right] \tag{8a}$$

with $D_{h,2} = D_2 - D_1 - 2d_1$ and

$$\dot{I}^{\Delta P} = \frac{1.79}{\pi^{1.75}} \frac{\dot{m}^{2.75}\ \mu^{0.25}}{\rho^2} \left[\frac{L}{D_1^{4.75}} + \frac{L \cdot (D_1 + 2d_1 + D_2)^{1.25}}{\left[D_2^2 - (D_1 + 2d_1)^2\right]^3} \right] \tag{8b}$$

Cumulative losses due to heat and power 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 LCA. We assume the following situation, based on the city 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 city 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 city heating water temperature. The exergetic efficiency of the pumps is assumed to be 0.7.

Hence the exergetic cost for the analysed heat exchanger of pressure rise, k_P, and heat, k_T, can be calculated to be 2.85 and 4 for the wide net and 2.85 and 2.66 for the dense net, respectively. The exergetic cost of a product is the amount of exergy which is needed for the production of one exergy unit of the product.

For the exergy destruction associated with the operation of the heat exchangers we have

$$\dot{I}_{oper} = k_T \dot{I}^{\Delta T} + k_P \dot{I}^{\Delta P} \tag{9}$$

Irreversibilities associated with the use of the material. The life cycle flow diagram of the heat exchanger is displayed in Figure 2. The exergy destruction in each process is shown in Table 1.

TABLE 1. Exergy destruction associated with the production of material and manufacturing of tubes

Process	I_S (MJ/kg)	I_{Cu} (MJ/kg)
primary process	10.5	60
Secondary process	4.4	20
Manufacturing process	5.7	15

The data listed in Table 1 is obtained from Cornelissen [5].

The heat exchanger is located in an insulated box made of polyurethane foam (PUR) with a thickness of 0.10 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. (1980). The exergy content is estimated to be 27 MJ/kg on basis of the lower heating value according to Kindler et al. [13]. So the cumulative exergy destruction is 71 MJ/kg (C_{PUR}). The box containing the heat exchanger has two outer sides of 0.70 meter. The height of the box is calculated from the length of the heat exchanger. The heat exchanger has been helical wound in

[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.

three coaxial tubular layers. The mean diameter of the tube windings is 0.45 meter. No recycling of the PUR has been assumed.

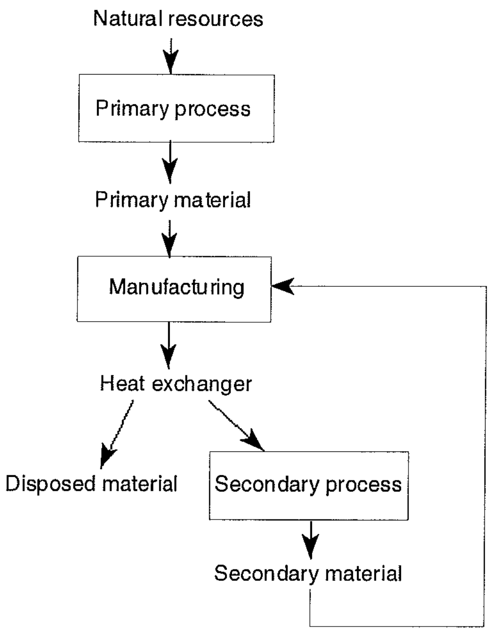


Figure 2. Life cycle of the heat exchanger

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 box.

$$\dot{I}_{man} = \frac{M_{Cu}C_{Cu} + M_{Fe}\,C_S + LC_{w,S} + M_{PUR}C_{PUR}}{t} =$$
$$\frac{\pi L}{t}\left[(D_1 + 0.5d_1)d_1\rho_{Cu}(C_{man,Cu} + x_{Cu}C_{\sec,Cu} + (1 - x_{Cu})C_{pri,Cu})\right]$$
$$+\frac{\pi L}{t}\left[(D_2 + 0.5d_2)d_2\,\rho_S(C_{man,S} + x_S C_{\sec,S} + (1 - x_S)C_{pri,S}) + C_{w,S}\right]$$
$$+\frac{1}{t}\left[\left(4\cdot\frac{L\cdot(0.7-0.1)}{3\cdot\pi\cdot 0.45}\cdot(D_2 + 2\cdot d2) + 2\cdot 0.49\right)\cdot 0.10\,\rho_{PUR}C_{PUR}\right] \qquad (10)$$

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.

3. Results

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

$$\dot{I}_{LC} = \dot{I}_{oper} + \dot{I}_{man} \tag{11}$$

Where $\dot{I}_{oper}$ and $\dot{I}_{man}$ are stated in (9) and (10), respectively.

The following operating parameters have been assumed for the heat exchanger. The incoming temperature of the cold domestic tap water is 15°C. The domestic tap water is heated to 65°C. The temperature of the incoming city 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 mass flow of the city heat water as the domestic water is 0.1 kg/s. The mean temperature of the inlet and outlet streams is used for the heat capacity, viscosity[2] and thermal conductivity of water. The wall thickness of the inner and outer tube are 0.8 and 2 mm, respectively. 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 a domestic water heat exchanger is taken with the hypothetical fixed length of 30 meter. The optimised inner and outer tube diameter are $8.56 \cdot 10^{-3}$ m and $1.29 \cdot 10^{-2}$ m for the wide net and $9.04 \cdot 10^{-3}$ m and $1.36 \cdot 10^{-2}$ m for the dense net. The life cycle irreversibility of the heat exchanger in full operation is $1.82 \cdot 10^{3}$ W for the wide net and $1.34 \cdot 10^{3}$ W for the dense net. Resulting from a ΔT, ΔP_1 and ΔP_2 of 5.24 K, 1.34 bar and 7.57 bar for the wide net and 5.57 K, 1.03 and 5.48 bar for the dense net, respectively. Where ΔT is the temperature difference between the hot and cold stream, which is constant, because we have a balanced counter flow heat exchanger. $T_{1,out}$, which can be calculated by $T_{2,\,in} - \Delta T$, is set to 65°C by use of iterative calculations.

3.2. RESULTS OF OPTIMISATION

The minimisation of $\dot{I}_{LC}$ for the 3 variables, D_1, D_2 and L gives the minimum value of life cycle irreversibility per heat exchanger in the wide net of the city heating system of $1.06 \cdot 10^{3}$ W. The optimum geometrical parameters were found to be $D_1 = 1.07 \cdot 10^{-2}$ m, $D_2 = 1.69 \cdot 10^{-2}$ m and L = 105.8 m. The ΔT, ΔP_1

[2] The value of the viscosity is strongly temperature dependent. The viscosity at 15°C, 70 °C and 80°C is $1.14 \cdot 10^{-3}$ Pa·s, $0.406 \cdot 10^{-3}$ Pa·s and $0.357 \cdot 10^{-3}$ Pa·s, respectively. So the assumption of the mean temperature will cause a deviation from the real situation.

and ΔP_2 of this optimised heat exchanger are 1.95 K, 1.62 bar and 3.65 bar, respectively.

The dense net life cycle irreversibility is $8.76 \cdot 10^2$ W for $D_1 = 1.07 \cdot 10^{-2}$ m, $D_2 = 1.69 \cdot 10^{-2}$ m and L = 87.0 m. The ΔT, ΔP_1 and ΔP_2 of the optimised heat exchanger are 2.93 K, 1.34 bar and 3.00 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 distribution net. The components of the life cycle irreversibility in the heat exchanger are displayed in Table 2.

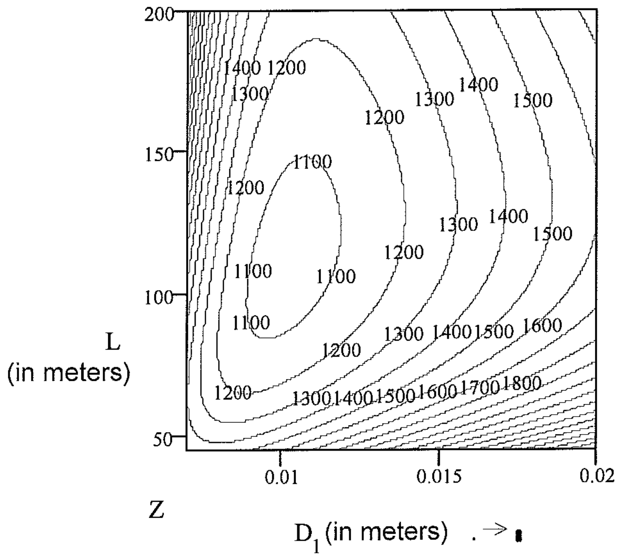
TABLE 2. Components of life cycle irreversibility in Watts

Component	Wide net (L=30m)	dense net (L=30m)	Wide net	dense net
Thermal	1451	1033	514	422
Mechanical	255	186	151	124
Manufacture	114	118	395	330
Total	1820	1337	1060	876

The contribution of the use of copper, steel and PUR-foam to the irreversibility associated with the manufacture is 39%, 42% and 19% for the wide and dense net optimisation and 32%, 33% and 35% for the both heat exchangers with a length of 30 meter, 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 Figure 3. The cross sections of flow areas of the inner and outer passages are set fixed in the ratio of 1: 0.863 to get a 3 dimensional figure. This ratio has been obtained for the optimisation of the wide net. In Figure 3 can be seen that the life cycle irreversibility rate rise 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. The effects of the diameters and the length on $\dot{I}_{LC}$ is greatest at their smallest values. The optimal inner tube diameter varies from around $0.8 \cdot 10^{-2}$ m for smaller lengths to $1.1 \cdot 10^{-2}$ for longer lengths of the tube.



The analysis carried out in this work is only applicable to the situation when both the heat and the power are supplied from a cogeneration plant. If the heat is provided by a source which produces only heat from combustion of fossil fuel the exergy saved in the heat exchanger will be lost in the heater. There is no similar trade-off between exergy saving during operation and exergy use during construction of the heat exchanger as in the case when heat and power are generated separately. The saving in exergy resulting from the use of a longer heat exchanger requiring lower heating water temperature can not in this case be utilised elsewhere e.g. for generating more electricity.

It is possible that a heat exchanger with more than one heated water tube, arranged in parallel inside the larger outer tube carrying the heating, may lead to a more efficient heat exchanger process. However, this matter must be left for a further investigation.

The Reynolds numbers of the inner and outer flow are $19.8 \cdot 10^3$ and $7.15 \cdot 10^3$ for the optimal configuration of the heat exchanger, respectively, when the mean temperature of the inlet and outlet streams is taken for the viscosity of the water. For the fixed length optimisation the Reynolds numbers are between $7.72 \cdot 10^3$ and $24.4 \cdot 10^3$ for the inner and outer flow, so the condition for the turbulent flow region is fulfilled.

Heat transfer inside the heat exchanger box, which leads to exergy destruction, is neglected. If this is taken into account an insulation directly wound around the tubes would probably be more effective than the assumed box. However, if this is the case the analysis shown would not changed drastically, because the exergy use during construction would still increase when the tubes become longer.

4. Conclusions

With the Exergetic Life Cycle Analysis, 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 reservoirs 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 whilst the length of the heat exchanger is smaller for the former than for the latter. 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. In the optimised situation the life cycle irreversibility is more uniformly distributed between the component irreversibilities than in the fixed length of 30 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 less restrictions on the design parameters for the optimisation.

5. Nomenclature

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

Greek letters

α = heat transfer coefficient
e = effectiveness
ρ = density
λ = coefficient of thermal conduction
μ = dynamic viscosity

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

6. References

1. Kotas, T.J., (1995) *The Exergy Method of Thermal Plant Analysis*, Krieger Publishing Company, Melbourne, Florida, USA .
2. Szargut, J., Morris, D.R. and Steward, F.R., (1988) *Exergy analysis of thermal, chemical and metallurgical processes*, Hemispere Publishing Corporation.
3. Consoli, F., et al., (1993), Guidelines for Life Cycle Assessment of Products: A 'Code of Practice', SETAC, Brussels, Belgium.
4. Heijungs, R. et al., (1992) *Environmental Life cycle Analysis of Products*, Center for environmental studies, University of Leiden, The Netherlands.
5. Bejan, A., (1982) *Entropy generation through heat and flow*, John Wiley & Sons.
6. Aceves-Saborio, S., Ranasinghe, J. and Reistad, G.M., (1989) *ASME Journal of Heat Transfer* **111**, 29 - 36.
7. Tondeur, D. and Kvaalen, E., (1987) *Industrial & Engineering Chemistry Research* **26**, 50 – 56.
8. Cornelissen, R.L., (1997) *Thermodynamics and sustainable development, the use of exergy analysis and reduction of irreversibility*, Ph.D. thesis, University of Twente.
9. Cornelissen, R.L., Bourdri, J.C. and Kalf, M.A., (1998) The value of the ELCA above the LCA, KEMA/University of Twente/CCS, in Dutch, in press.
10. Chapman, A.J., (1974) *Heat transfer*, Macmillan, New York .
11. Kays, W.M. and London, A.L., *Compact Heat Exchangers*, third edition, McGraw-Hill Book Company .
12. Rogers, G.F.C. and Mayhew, Y.R., *Engineering Thermodynamics, Work and Heat Transfer*, Longmans, London (1967).
13. Kindler H. and Nikles A., (1980) *Kunststoffe* **70**, 12 .