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Heat Transfer and Its Assessment

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1. Introduction

Somebody, interested in heat transfer and therefore reading one of the many books about this subject might be confronted with the following statement after the heat transfer coefficient $h = \dot{q}_{\mathrm{w}}/\Delta T$ has been introduced:

"The heat transfer coefficient h is a measure for the quality of the transfer process."

That sounds reasonable to our *somebody* though for somebody else (the authors of this chapter) there are two minor and one major concerns about this statement. They are:

1. Heat cannot be transferred since it is a process quantity.

2. The coefficient h is not a nondimensional quantity what it better should be.

3. What is the meaning of "quality"?

The major concern actually is the last one and it will be the crucial question that is raised and answered in the following. Around that question there are, however, some further aspects that should be discussed. Two of them are the first two in the above list of concerns.

The heat transfer coefficient h is typically used in single phase convective heat transfer problems. This is a wide and important field of heat transfer in general. That is why the problem of heat transfer assessment will be discussed for this kind of "conduction based heat transfer" in the following sections 2 to 4. In section 5 extensions to the overall heat transfer through a wall, heat transfer with phase change and the fundamentally different "radiation based heat transfer" will be discussed.

2. The "quality" of heat transfer

2.1 Preliminary considerations

What is commonly named *heat transfer* is a process by which energy is transferred across a certain system boundary in a particular way. This special kind of energy transfer is characterised by two crucial aspects:

– The transfer process is initiated and determined by temperature gradients in the vicinity of the system boundary.

– As a consequence of this transfer process there is a change of entropy on both sides of the system boundary. That change can be interpreted as a transfer of entropy linked to the energy transfer. It thus is always in the same direction. The strengths of both transfer processes are not in a fixed proportion, but depend on the temperature level.

According to these considerations the phrase "heat transfer" actually should be replaced by "energy transfer in the form of heat". Since, however, "heat transfer" is established worldwide

in the community we also use this phrase, but as a substitute for "energy transfer in the form of heat".

It is worth noting that from a thermodynamics point of view *heat* is one of only two ways in which energy can be transferred across a system boundary. The alternative way is *work*. This other kind of energy transfer is not caused by temperature gradients and is not accompanied by entropy changes. For further details see Moran & Shapiro (2003); Baehr & Kabelac (2009); Herwig & Kautz (2007), for example.

The energy transferred in the form of heat from a thermodynamics point of view is *internal energy* stored in the material by various mechanisms on the molecular level (translation, vibration, rotation of molecules). The macroscopic view on this internal energy can not only identify its amount (in Joule) but also its "usefulness". The thermodynamic term for that is its amount of *exergy*. Here exergy, also called *available work*, is the maximum theoretical work obtainable from the energy (here: internal energy) interacting with the environment to equilibrium.

According to this concept energy can be subdivided in two parts: exergy and anergy. Here anergy is energy which is not exergy (and thus "not useful"). If, however, energy has a certain value (its amount of exergy) a crucial question with respect to a transfer of energy (heat transfer) is that about the *devaluation of the energy in the transfer process*.

2.2 Energy devaluation in a heat transfer process

What happens to the energy in a heat transfer process can best be analysed on the background of the second law of thermodynamics. This kind of analysis which considers the entropy, its transfer as well as its generation is called second law analysis (SLA). In a heat transfer situation the entropy S is involved twofold:

– Entropy is transferred over the system boundary together with the transferred energy. In a thermodynamically reversible process entropy is transferred only and no entropy is generated. This infinitesimal transfer rate is

$$d\dot{S} = \partial \dot{Q}_{rev}/T \tag{1}$$

Here $\partial \dot{Q}_{rev}$ is an infinitesimal heat flux, T the thermodynamic temperature at which it occurs and $d\dot{S}$ the rate by which the entropy in the system is changed due to the heat transfer. Such a reversible heat transfer only occurs when there are no temperature gradients involved. Therefore (1) is either the ideal situation of a real transfer process (with ΔT as operating temperature difference) in the limit $\Delta T \to 0$ or that part of a real heat transfer process without the entropy generation due to local temperature gradients.

– Entropy is generated in the system wherever temperature gradients $\partial T/\partial n$ occur. The generation rate per volume ($'''$) is, see Bejan (1982); Herwig & Kock (2007),

$$\dot{S}_C''' = \frac{k}{T^2}\left(\frac{\partial T}{\partial n}\right)^2 \tag{2}$$

Here k is the thermal conductivity and n a coordinate in the direction of the temperature gradient vector.

Entropy generation always means *loss of exergy*. According to the so-called Gouy–Stodola theorem, see Bejan (1982), the exergy loss rate per volume due to heat conduction is

$$\dot{E}_{LC}''' = T_0 \dot{S}_C''' \tag{3}$$

with $\dot{S}_C'''$ from (2) and T_0 as the temperature of the environment.

The devaluation of energy that is transferred in the form of heat thus can be determined by integrating the local exergy loss rate (3) over the volume of the system under consideration, as will be shown in sec. 4.2.

2.3 Energetic and exergetic quality of heat transfer

Since heat transfer is caused by temperature gradients, or (integrated over a finite distance) by temperature differences ΔT there are two questions about the "quality" of a transfer process:

– How much energy can be transferred in a certain situation with ΔT as the operating temperature difference? This is called the *energetic quality* of the transfer process.

– How much is the energy devaluated in a certain transfer situation with ΔT as the operating temperature difference? This is called the *exergetic quality* of the transfer process.

Obviously two parameters are needed to answer both questions. If, however, there is only the heat transfer coefficient h or its nondimensional counterpart, the Nußelt number Nu, not all information about the heat transfer process is available as will be demonstrated hereafter.

3. Heat Transfer Assessment

3.1 The energetic quality of heat transfer

The energetic quality was introduced (sec. 2.3) as the answer to the question "How much energy can be transferred in a certain situation with ΔT as the operating temperature difference?" This amount of energy is finite (and not infinite) only because it occurs in a real process subject to losses. Finite values of the heat transfer coefficient h nevertheless are not a direct measure for these losses.

Though h, defined as

$$h \equiv \frac{\dot{q}_{\mathrm{w}}}{\Delta T} \qquad ; [h] = \mathrm{W/m^2 K} \tag{4}$$

is frequently used and widely accepted, there are several aspects of it that can be critically discussed:

– When h should characterise the energetic quality it should be a fixed value for a specific heat transfer situation. This especially referres to the strength of heat transfer, i.e. h should not depend on ΔT. There are, however, situations in which this is not the case, like for natural convection in general and for radiative heat transfer with large values of ΔT, see Herwig (1997) for details.

– Instead of h, its reciprocal $1/h = \Delta T/\dot{q}_{\mathrm{w}}$ would be more appropriate. Then finite values (and not zero) for h^{-1} would be due to a *resistance* which a heat flux $\dot{Q}_{\mathrm{w}} = \dot{q}_{\mathrm{w}} A$ encounters on the heat transfer area A. This is in analogy to the resistance $R = \Delta U/I$ that an electrical current I ($I = iA$; i: current density, A: cross section) encounters with a voltage ΔU. By this analogy U corresponds to T and I to $\dot{Q}_{\mathrm{w}}$.

– Since h is part of the Nußelt number

$$\mathrm{Nu} \equiv \frac{\dot{q}_{\mathrm{w}} L}{k \Delta T} = h \cdot \frac{L}{k} \qquad ; [\mathrm{Nu}] = 1 \tag{5}$$

it is often assumed to be equivalent to this nondimensional group ($[\mathrm{Nu}] = 1$). The Nußelt number, however, as a result of a systematic dimensional analysis process, is a more significant parameter. It characterises a heat transfer situation irrespective of its geometrical size L and the thermal conductivity k of the fluid involved.
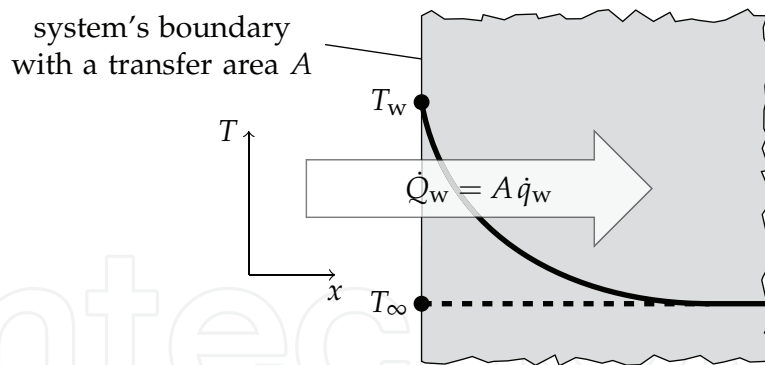
Fig. 1. One dimensional heat transfer at a system boundary ($\dot{q}_\mathrm{w} = $ const)

Whenever general statements about a certain heat transfer situation are the objective the Nußelt number is the preferred parameter.

3.2 The exergetic quality of heat transfer

The exergetic quality was introduced (sec. 2.3) as the answer to the question "how much is the energy devaluated in a certain transfer situation with ΔT as the operating temperature difference?" This devaluation is directly measured by the exergy loss rate based on the relation (3).

For a heat transfer situation as sketched in Fig. 1 which is characterised by Nu with respect to its energetic behavior a second parameter is now introduced and named *exergy loss number*. Its definition is

$$\mathrm{N}^\mathrm{E} \equiv \frac{\dot{E}_\mathrm{LC}}{\dot{E}} = \frac{T_0 \dot{S}_\mathrm{C}}{\eta_\mathrm{C} \dot{Q}_\mathrm{w}} \qquad \text{(exergy loss number)} \qquad (6)$$

with $\dot{E}_\mathrm{LC}$ and $\dot{S}_\mathrm{C}$ as integration of the local rates $\dot{E}_\mathrm{LC}'''$ and $\dot{S}_\mathrm{C}'''$ and $\dot{E}$ as the exergy fraction of the heat flux $\dot{Q}_\mathrm{w}$. This exergy fraction corresponds to the transferred heat flux $\dot{Q}_\mathrm{w}$ multiplied by the *Carnot factor*

$$\eta_\mathrm{C} = 1 - \frac{T_0}{T_\mathrm{w}} \qquad (7)$$

The crucial quantity in N^E according to (6) is the overall entropy generation rate due to heat conduction, $\dot{S}_\mathrm{C}$. In a complex temperature field it emerges through a field integration of $\dot{S}_\mathrm{C}'''$, see section 4.2 below. If, however, there is a one-dimensional heat transfer perpendicular to the system boundary as sketched in Fig. 1, the situation is different.

Then only the two temperature levels T_w and T_∞ count, and $\dot{S}_\mathrm{C}$ is the difference of the transfer rates $\dot{Q}_\mathrm{w}/T_\mathrm{w}$ and $\dot{Q}_\mathrm{w}/T_\infty$, cf. (1) for incremental parts, i.e.

$$\dot{S}_\mathrm{C} = \dot{Q}_\mathrm{w} \left(\frac{1}{T_\infty} - \frac{1}{T_\mathrm{w}} \right) \qquad (8)$$

With this $\dot{S}_\mathrm{C}$ the exergy loss number for a one-dimensional heat transfer is

$$\mathrm{N}^\mathrm{E} \equiv \frac{\dot{E}_\mathrm{LC}}{\dot{E}} = \frac{T_0}{T_\infty} \cdot \frac{T_\mathrm{w} - T_\infty}{T_\mathrm{w} - T_0} \qquad \text{(one-dimensional exergy loss number)} \qquad (9)$$

3.3 An example

As a simple example, where N^E according to (9) holds, the fully developed pipe flow with heat transfer will be considered. What usually can be found with respect to its heat transfer

cycle / fluid	Nu	$\dfrac{\dot{q}_w}{\mathrm{W/m^2}}$	$\dfrac{L}{\mathrm{m}}$	$\dfrac{k}{\mathrm{W/mK}}$	$\dfrac{T_0}{\mathrm{K}}$	$\dfrac{T_w}{\mathrm{K}}$	$\dfrac{T_w - T_\infty}{\mathrm{K}}$	N^E
SPC / water	100	10^3	0.1	0.1	300	900	10	0.006
ORC / ammonia	100	10^3	0.1	0.038	300	400	26	0.3

Table 1. Heat transfer with $Nu = 100$ in two different power cycles

performance is a Nußelt number correlation $Nu = Nu(Re)$. This, however, is only the energetic part of the performance and N^E according to (9) should also be considered.

To be specific, it is assumed that a heat transfer situation with $Nu = 100$, $\dot{q}_w = 10^3\,\mathrm{W/m^2}$ and $L = 0.1\,\mathrm{m}$ occurs in two different power cycles. One is a steam power cycle (SPC) with water as the working fluid and a temperature level for heat transfer $T_w = 900\,\mathrm{K}$. The alternative is an ORC cycle with ammonia (NH_3) as working fluid and a temperature level $T_w = 400\,\mathrm{K}$. When in both cycles $Nu = 100$ with the same values for $\dot{q}_w$ and L holds the temperature difference ΔT is larger by a factor 2.6 for ammonia compared to water. This is due to the different values of thermal conductivity of water (at $T = 900\,\mathrm{K}$ and $p = 250\,\mathrm{bar}$) and ammonia (at $T = 400\,\mathrm{K}$ and $p = 25\,\mathrm{bar}$), assuming typical values for the temperature and pressure levels in both cycles.

Table 1 collects all data of this example including the exergy loss number according to (9). For the ORC-cycle this number is 0.3 and this is 50 times that for the SPC-cycle. Note, that an amount of 0.6 % and 30 % less exergy after the heat transfer in a power cycle means: that amount of available work is lost for a conversion into mechanical energy at the turbine of the cycle.

Here the devaluation of the transferred energy obviously is an important aspect of the process. This devaluation cannot be quantified by the Nußelt number though its finite (and not infinite) value is due to the fact that losses occur in a real transfer process. The exact amount of the losses is given by the exergy loss number N^E.

4. Complex convective heat transfer problems

Heat transfer often occurs as *convective heat transfer*, i.e. influenced and supported by a fluid flow. Especially when the flow is turbulent there is a strong impact on the heat transfer performance. This is due to the strong effect turbulent fluctuations have on the transport of internal energy. As a modeling strategy a so-called turbulent or effective thermal conductivity can be defined which often is a magnitude larger than the molecular conductivity k, see Munson et al. (2009); Herwig (2006) for details.

Increasing the flow rate almost always increases the heat transfer intensity. This is reflected by the increasing Nußelt numbers $Nu(Re)$. There is, however, a prize to pay: it turns out that not only the exergy losses due to the conduction of heat, $\dot{S}_C$ in (6), have to be accounted for, but that also the exergy losses due to the dissipation of mechanical energy in the fluid flow must be considered. Only when both losses are examined and accounted for together, one can answer the question whether an increase in the flow rate is beneficial for the transfer process as a whole.

4.1 Fluid flow assessment

Before the heat transfer process as a whole is considered we want to address the losses in a flow field. Again these losses are exergy losses accompanied by entropy generation. The

common way to characterise a flow with respect to its losses is to define a *friction factor* for a flow in pipes and channels, for example. For external flows it would be a *drag coefficient*. Both parameters are finite values (and not zero) due to the fact that losses occur. Again the question arises, whether these parameters are an immediate measure of the exergy losses.

It turns out, that the friction factor f, which will be considered in the following, strictly speaking is a parameter that assesses the *energetic quality* of the flow. It is the answer to the question "How much mechanical energy (measured by the total head of the flow) can be transferred along the pipe or channel in a certain situation with Δp as the operating pressure difference?"

In analogy to the heat transfer process the second question is "How much is the mechanical energy (measured by the total head) devaluated in a certain situation with Δp as the operating pressure difference?" This is about the *exergetic quality* of the flow

Again in analogy to the heat transfer process we define two assessment parameters:

– Head loss coefficient K

$$\mathrm{K} = f\frac{L}{D_\mathrm{h}} \equiv \frac{2\,\varphi}{u_\mathrm{m}^2} \tag{10}$$

with φ as specific dissipation of mechanical energy and u_m as the mean velocity in the cross section of the pipe or channel. This parameter is frequently used in fluid mechanics. By introducing K instead of f alone, a pipe or channel (of length L and with a hydraulic diameter D_h) is treated as a conduit component like bends, trijunction, diffusers etc.

– Exergy loss coefficient K^E

$$\mathrm{K}^\mathrm{E} \equiv \frac{T_0}{T_\infty}\mathrm{K} = \frac{\dot{E}_\mathrm{LD}}{\dot{m}\,u_\mathrm{m}^2/2} \tag{11}$$

with T_0 as the temperature of the environment and T_∞ as that temperature level on which the flow occurs. The exergy loss rate due to dissipation $\dot{E}_\mathrm{LD}$ is the integrated local value $\dot{E}_\mathrm{LD}''' = T_0\dot{S}_\mathrm{D}'''$ with $\dot{S}_\mathrm{D}'''$ defined later, cf. (3) for the heat transfer counter parts.

The background of both definitions can again be analysed by looking at the entropy and its generation in the flow field. Here the specific dissipation φ and the entropy generation due to dissipation, $\dot{S}_\mathrm{D}$, are closely related, though not the same. The relation is, see Herwig et al. (2010), for example

$$\dot{m}\,\varphi = T_\infty\dot{S}_\mathrm{D} \tag{12}$$

Note, that $T_0\dot{S}_\mathrm{D}$ and not $T_\infty\dot{S}_\mathrm{D}$ is the exergy loss rate according to the Gouy–Stodola theorem, cf. (3). That is why K is the energetic, but not the exergetic assessment parameter.

For simple flow geometries such as straight pipes, bends, etc. (10) and (12) can be combined with

$$\dot{S}_\mathrm{D} = \mathrm{K}\frac{u_\mathrm{m}^2\,\dot{m}}{2\,T_\infty} \tag{13}$$

as the entropy generation rate based on emperically determined correlations for K-values.

For complex flow situations the entropy generation rate due to the dissipation of mechanical energy

$$\dot{S}_\mathrm{D} = \int \dot{S}_\mathrm{D}'''\mathrm{d}V \tag{14}$$

and thus the dissipation rate $\dot{m}\,\varphi$ according to (12) can be determined by accounting for the local entropy generation rate per volume ($'''$)

$$\dot{S}_D''' = \frac{\mu}{T}\left(2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] \right.$$
$$\left. + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2\right) \quad (15)$$

For details of the derivation see Herwig & Kock (2007); Herwig et al. (2010). When the flow is non-isothermal, T in (15) is different at different locations. As long as temperature variations are small compared to T this effect is also small and can be neglected as a first approximation. Then a unique temperature T_∞ appears in (12).

Equation (15) can be immediately used for the determination of $\dot{S}_D'''$ when the flow is laminar or for a turbulent flow in a DNS-approach. In a RANS approach $\dot{S}_D'''$ is split into $\dot{S}_{\overline{D}}'''$ and $\dot{S}_{D'}'''$ like all other variables with

$$\dot{S}_D''' = \dot{S}_{\overline{D}}''' + \dot{S}_{D'}''' \quad (16)$$

and

$$\dot{S}_{\overline{D}}''' = \frac{\mu}{\overline{T}}\left(2\left[\left(\frac{\partial \overline{u}}{\partial x}\right)^2 + \left(\frac{\partial \overline{v}}{\partial y}\right)^2 + \left(\frac{\partial \overline{w}}{\partial z}\right)^2\right] \right.$$
$$\left. + \left(\frac{\partial \overline{u}}{\partial y} + \frac{\partial \overline{v}}{\partial x}\right)^2 + \left(\frac{\partial \overline{u}}{\partial z} + \frac{\partial \overline{w}}{\partial x}\right)^2 + \left(\frac{\partial \overline{v}}{\partial z} + \frac{\partial \overline{w}}{\partial y}\right)^2\right) \quad (17)$$

$$\dot{S}_{D'}''' = \frac{\mu}{\overline{T}}\left(2\left[\overline{\left(\frac{\partial u'}{\partial x}\right)^2} + \overline{\left(\frac{\partial v'}{\partial y}\right)^2} + \overline{\left(\frac{\partial w'}{\partial z}\right)^2}\right] \right.$$
$$\left. + \overline{\left(\frac{\partial u'}{\partial y} + \frac{\partial v'}{\partial x}\right)^2} + \overline{\left(\frac{\partial u'}{\partial z} + \frac{\partial w'}{\partial x}\right)^2} + \overline{\left(\frac{\partial v'}{\partial z} + \frac{\partial w'}{\partial y}\right)^2}\right) \quad (18)$$

Only $\dot{S}_{\overline{D}}'''$ can be determined directly once a CFD-solution of the flow field exists. The fluctuating part $\dot{S}_{D'}'''$ must be subject to turbulence modeling. For example, $\dot{S}_{D'}'''$ can be linked to the turbulent dissipation rate ε which is known when a k–ε-model is used (but which is also part of almost all other models) by

$$\dot{S}_{D'}''' = \frac{\varrho\,\varepsilon}{\overline{T}} \quad (19)$$

For details, again see Herwig & Kock (2007).

This approach can for example be used to determine the friction factor of a pipe with a special roughness type, called Loewenherz-thread roughness, by integrating the entropy generation rate as shown in Fig. 2, see Herwig et al. (2008) for details. The dark lines are numerical results based on (17)-(19). They compare very well with experimental results from Schiller (1923) and show that the classical Moody chart is not even a moderately good approximation for this kind of rough pipes.

Once the head loss coefficient K is known it is an easy though important step to find K^E according to (11). Only K^E is a direct measure for the devaluation of the transferred energy in terms of lost available work (or exergy).
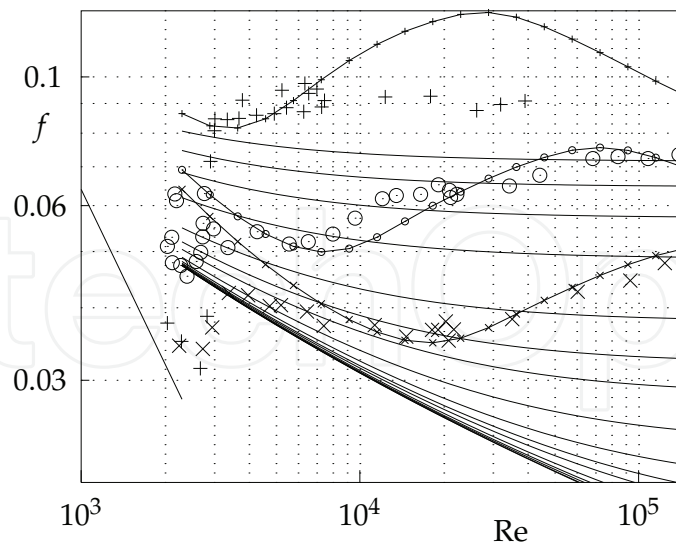
Fig. 2. Friction factor for a pipe with a Loewenherz-thread with different roughness heights; dark lines: numerical integration; large symbols: experiments; faint lines: Moody chart

As an example, a pipe with a certain head loss coefficient K is part of a power cycle. It is operated on different temperature levels when it is in a steam power cycle (SPC, temperature $T_\infty = 900\,\text{K}$) or in an organic Rankine cycle (ORC, temperature $T_\infty = 400\,\text{K}$). With $T_0 = 300\,\text{K}$ according to (11)

– $K^E = 0.33 \cdot K$ for an SPC

– $K^E = 0.75 \cdot K$ for an ORC

The losses of exergy (available work) in an ORC-cycle are more than twice that for the same flow situation (and head loss coefficient K) in an SPC-cycle.

4.2 Convective heat transfer assessment

For a complete assessment of a convective heat transfer situation the *energetic* part is accounted for by the Nußelt number (with no need to look at the head loss coefficient). The *exergetic* part, however, requires the combined consideration of losses in the temperature and the flow field in order to determine the overall reduction of available work caused by the (convective) heat transfer.

This, however, is straight forward within the SLA-analysis. An *overall exergy loss number* N^E now is defined as

$$N^E \equiv \frac{\dot{E}_{LC} + \dot{E}_{LD}}{\dot{E}} = \frac{T_0(\dot{S}_C + \dot{S}_D)}{\eta_C \dot{Q}_w} \tag{20}$$

It referres the overall exergy loss $\dot{E}_{LC} + \dot{E}_{LD}$ in a convective heat transfer situation to the exergy fraction of the transferred energy, $\eta_C \dot{Q}_w$.

With the help of (20) it can be decided whether the increase in the Nußelt number by a certain technique to improve the heat transfer, like adding turbulence promoters, roughening of the wall or simply increasing the flow rate, is beneficial from the perspective of exergy conservation. When N^E is decreased, more available work is left after the transfer than without the change made in order to improve the heat transfer.

Since a device with a small number N^E obviously is more efficient than one with a larger N^E we introduce an *overall efficiency factor* as

$$\eta^E \equiv 1 - N^E = \frac{\dot{E} - (\dot{E}_{LC} + \dot{E}_{LD})}{\dot{E}} \tag{21}$$

When η^E with N^E should be applied to a complex convective heat transfer situation, $\dot{S}_C$ cannot be determined like in (9) which only holds for a one-dimensional case. Then $\dot{S}_C$ is found from

$$\dot{S}_C = \int \dot{S}_C''' dV \tag{22}$$

with $\dot{S}_C'''$ in Cartesian coordinates, cf. (2),

$$\dot{S}_C''' = \frac{k}{T^2} \left(\left(\frac{\partial T}{\partial x} \right)^2 + \left(\frac{\partial T}{\partial y} \right)^2 + \left(\frac{\partial T}{\partial z} \right)^2 \right) \tag{23}$$

or after the turbulence splitting according to the RANS-approach

$$\dot{S}_C''' = \dot{S}_{\overline{C}}''' + \dot{S}_{C'}''' \tag{24}$$

with

$$\dot{S}_{\overline{C}}''' = \frac{k}{\overline{T}^2} \left(\left(\frac{\partial \overline{T}}{\partial x} \right)^2 + \left(\frac{\partial \overline{T}}{\partial y} \right)^2 + \left(\frac{\partial \overline{T}}{\partial z} \right)^2 \right) \tag{25}$$

$$\dot{S}_{C'}''' = \frac{k}{\overline{T}^2} \left(\overline{\left(\frac{\partial T'}{\partial x} \right)^2} + \overline{\left(\frac{\partial T'}{\partial y} \right)^2} + \overline{\left(\frac{\partial T'}{\partial z} \right)^2} \right) \tag{26}$$

analogous to the determination of $\dot{S}_D$ in (14)-(18). Like $\dot{S}_{D'}'''$ in (19), $\dot{S}_{C'}'''$ has to be modeled, for example by (α, α_t: molecular and turbulent thermal diffusivities)

$$\dot{S}_{C'}''' = \frac{\alpha_t}{\alpha} \dot{S}_{\overline{C}}''' \tag{27}$$

described in Herwig & Kock (2007).

4.3 An example

As an example a counter flow plate heat exchanger is analysed with respect to its heat transfer performance. For that purpose a special 2D-geometry is chosen which corresponds to the geometric situation in a stack of sinusodially formed plates shown in Fig. 3. One element of the cold part is "cut out" as the numerical solution domain, assuming periodic boundary conditions (Fig. 3 b). The boundary conditions are non-slip for the flow and temperature boundary conditions for the heat transfer. Basically a temperature rise $T_2 - T_1$ is set which is assumed to be linearly distributed between the cross sections ① and ②. With $T_2 - T_1$ set, the overall heat flux into the solution domain is also prescribed. The questions to be answered now are

– What is the Nußelt number $Nu = Nu(Re)$

– What is the overall exergy loss rate $\dot{E}_L = \dot{E}_{LD} + \dot{E}_{LC}$ and thus the efficiency factor η^E?

– Is there a maximum of η^E with respect to the Reynolds number?

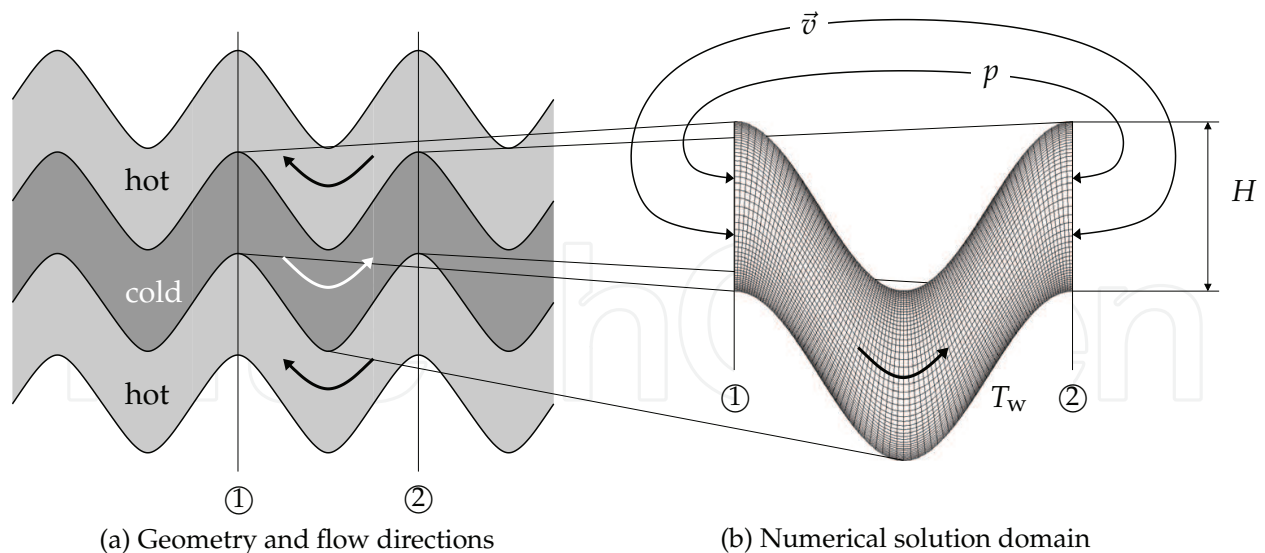(a) Geometry and flow directions   (b) Numerical solution domain

Fig. 3. 2D-sinusodial plate arrangement in a plate heat exchanger

The flow part of the entropy generation $\dot{E}_{LD}/\dot{E}$ will increase when the Reynolds number gets higher whereas the heat transfer part $\dot{E}_{LC}/\dot{E}$ will decrease due to the favorable influence of a stronger convection. The overall effect $\dot{E}_L/\dot{E}$ is expected to show a minimum which corresponds to a maximum in η^E.

Details of the numerical approach, using the k–ω-SST turbulence model will not be given here (since we want to concentrate on the assessment strategy), they can be found in Redecker (2010). Figure 4 shows the qualitative distribution of the time averaged velocity $\vec{\bar{v}} = \bar{u}, \bar{v}$, temperature $\overline{T}$ and the local entropy generation rate $\dot{S}''' = \dot{S}'''_D + \dot{S}'''_C$.

The overall heat transfer performance is shown in Fig. 5. The energetic part in terms of Nu(Re) is shown in Fig. 5(a), the exergetic part in terms of η^E in Fig. 5(b). The Reynolds number Re = 1995, see Fig. 5(b), turns out to be the optimal Reynolds number with respect to the loss of available work, since η^E has its maximum for this parameter value.

When this heat exchanger element is analysed in the "conventional way" without recourse to the second law of thermodynamics, one would for example apply an often used *thermo-hydraulic performance parameter*

$$\eta \equiv \left(\frac{St}{St_0}\right)\left(\frac{f}{f_0}\right)^{-\frac{1}{3}} \tag{28}$$

introduced in Gee & Webb (1980) with St = Nu/RePr and $f = K \cdot D_h/L$. This parameter η is used to compare a certain convective heat transfer device (with St_0, f_0) to modified versions (with St, f) and then to decide which of the modifications are beneficial. There is, however, no clear physical interpretation for η, except for the presumption that $\eta > 1$ corresponds to an improved situation.

Applied to the present example, η according to Fig. 6 results without an indication of an optimum within the Reynolds number range shown here. With the combination of St and f as two very different parameters in the definition of η there is no clear physical meaning to be recognised.
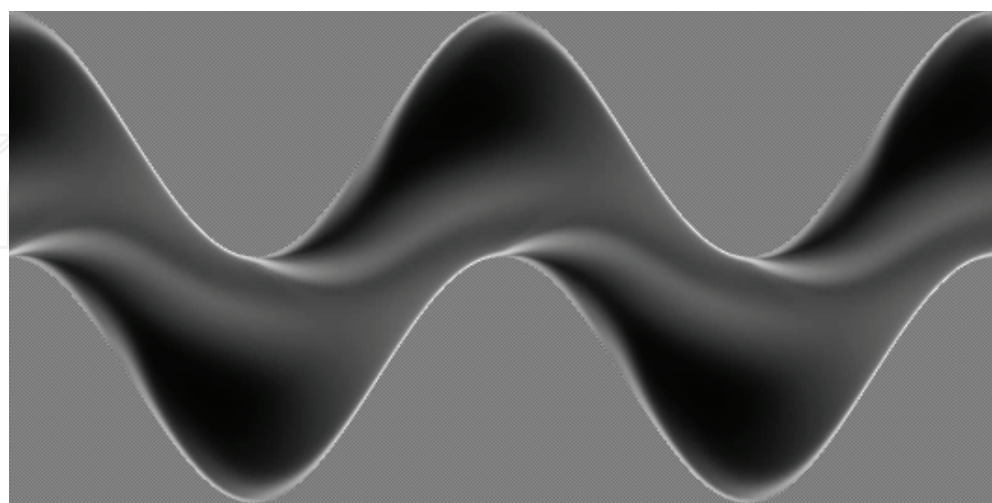
We therefore suggest to use η^E (or N^E) in addition to Nu in order to get a complete assessment of convective heat transfer situations.
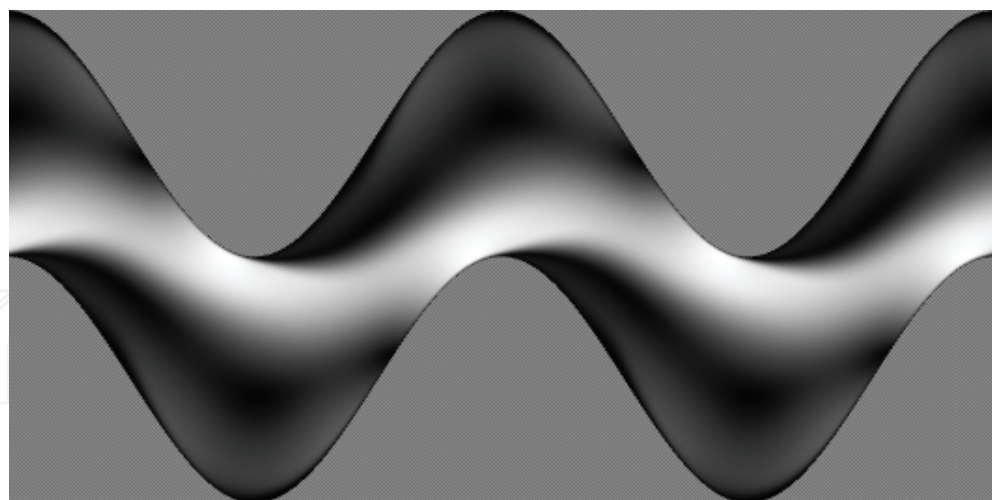
(a) $\left|\vec{\bar{v}}\right|$
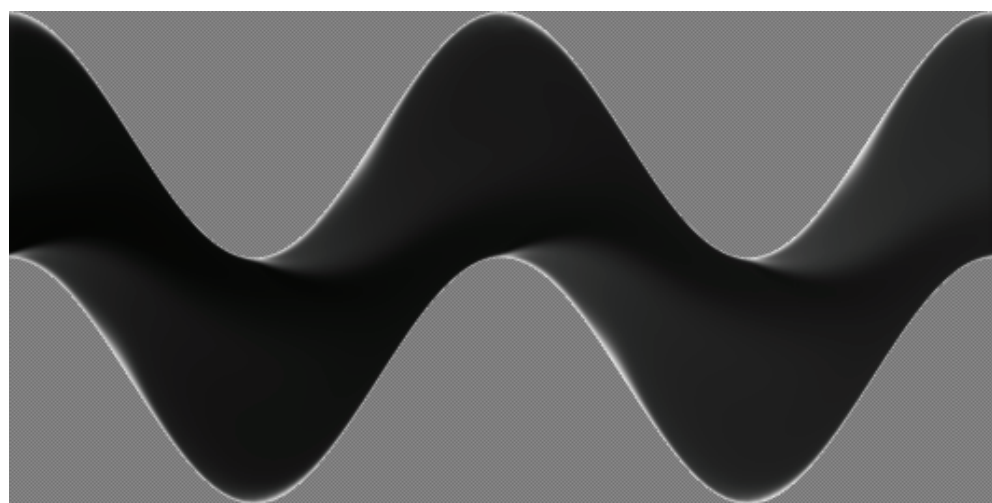
(b) $\overline{T}$

(c) $\dot{S}'''$

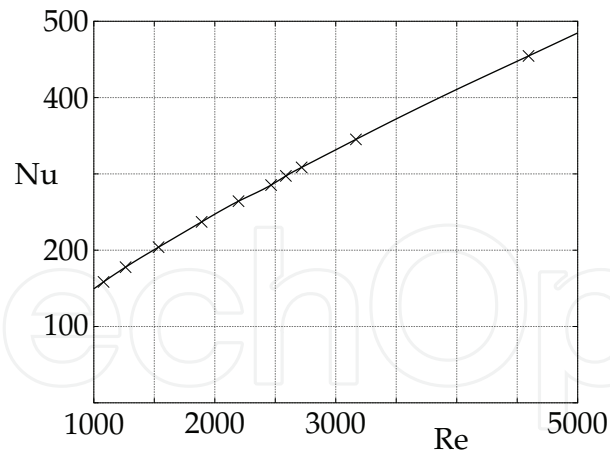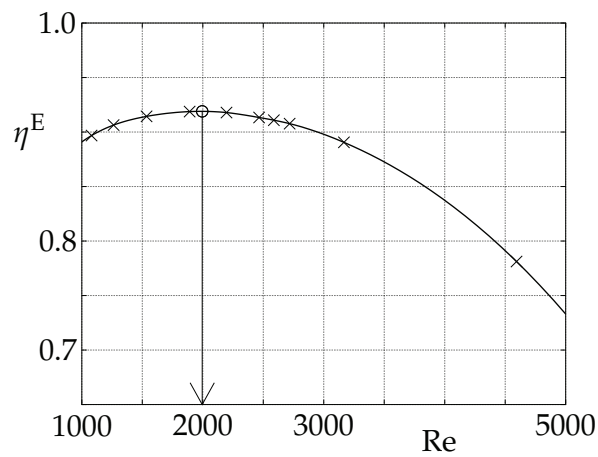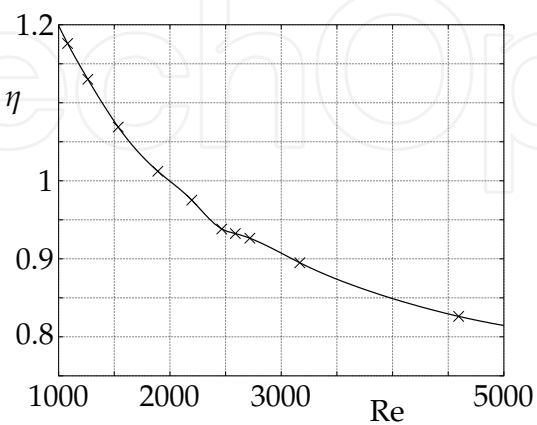Fig. 4. Numerical results in twice the solution domain, see Fig. 3(b); Re = 1995; light: high values, dark: low values

(a) Energetic assessment by $\mathrm{Nu}(\mathrm{Re})$

(b) Exergetic assessment by $\eta^{\mathrm{E}}(\mathrm{Re})$

Fig. 5. Heat transfer performance of a plate heat exchanger element

Fig. 6. Heat transfer performance of a plate heat exchanger element, see Fig. 5 for comparison; η: thermo-hydraulic performance parameter (28)
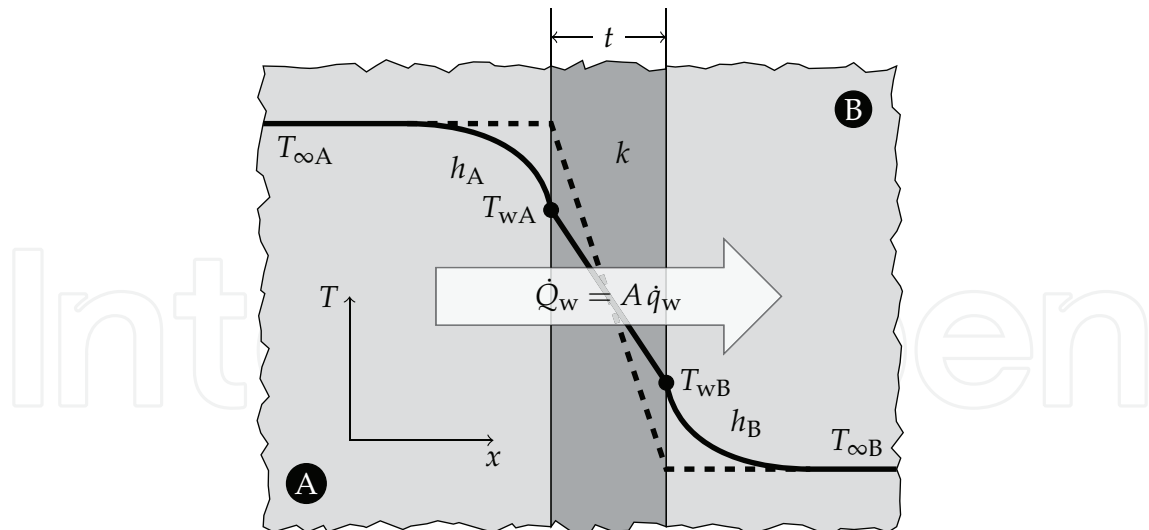
Fig. 7. Overall heat transfer through a wall of thickness t with a thermal conductivity k

5. Further heat transfer problems and their assessment

So far the general idea of a combined energetic and exergetic assessment of heat transfer situations has been discussed for single phase convective heat transfer processes. Some further aspects will be addressed in the following subsections.

5.1 Overall heat transfer through a wall

In Fig. 7 the overall heat transfer through a wall between two systems A and B is sketched. In terms of heat transfer resistances the overall resistance is the sum of two convective and one conductive parts, i.e. $h_A^{-1} + k\Delta T_w/t + h_B^{-1}$ with $\Delta T_w = T_{wA} - T_{wB}$. Since these details are not of interest an overall heat transfer coefficient

$$U = \frac{\dot{q}_w}{\Delta T_\infty} \qquad \text{with } \Delta T_\infty = T_{\infty A} - T_{\infty B} \tag{29}$$

is introduced, see for example Incropera et al. (2006). This is the *energetic* aspect of the assessment.

The *exergetic* aspect can again be accounted for by the exergy loss number N^E according to (6) but now with

$$\dot{S}_C = \dot{Q}_w \left(\frac{1}{T_{\infty B}} - \frac{1}{T_{\infty A}} \right) \tag{30}$$

so that

$$N^E \equiv \frac{\dot{E}_{LC}}{\dot{E}} = \frac{T_0}{T_{\infty B}} \cdot \frac{T_{\infty A} - T_{\infty B}}{T_{\infty A} - T_0} \tag{31}$$

Here the whole process is assumed to be conductive in nature by replacing the real temperature distribution by one which is decreasing in the wall only (broken line in Fig. 7). Then the entropy generation occurs in the wall only. This simplyfied model is appropriate since it encounters the same devaluation of the transferred energy as the real case. The only thing that counts is the temperature drop from $T_{\infty A}$ to $T_{\infty B}$.
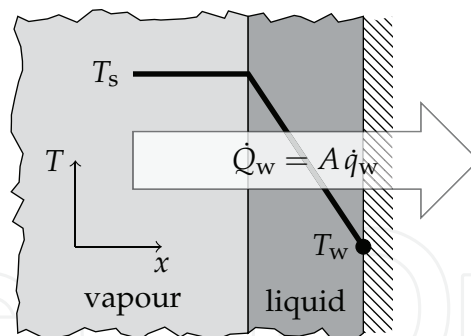
Fig. 8. Heat transfer at a vertical wall with film condensation at a position z away from the leading edge

5.2 Heat transfer with phase change

When close to a heated or cooled wall boiling or condensation occurs the heat transfer mechanism is still "conduction based" since the energy enters or leaves the system by conduction over the system boundary.

In Fig. 8 the situation is sketched for film condensation as an example, showing the local parameters at a certain position z from the leading edge of a vertical wall.

The energetic assessment is by the local heat transfer coefficient

$$h = \frac{\dot{q}_w}{\Delta T_s} \qquad \text{with } \Delta T_s = T_s - T_w \tag{32}$$

Here ΔT_s is the subcooling of the wall (beyond the saturation temperature T_s at the prevailing pressure). The exergetic assessment is possible again by the exergy loss number according to (6), now with

$$\dot{S}_C = \dot{Q}_w \left(\frac{1}{T_w} - \frac{1}{T_s} \right) \tag{33}$$

so that

$$N^E \equiv \frac{\dot{E}_{LC}}{\dot{E}} = \frac{T_0}{T_w} \cdot \frac{T_s - T_w}{T_s - T_0} \tag{34}$$

This simple model assumes that there is no overheating in the vapour phase. An extension to account for this effect would be straight forward, however.

5.3 Radiation based heat transfer

When heat transfer is by radiation the situation is very different compared to the "conduction based" heat transfer processes discussed so far. Energy transport occurs in an electromagnetic field and no longer by molecular interaction of adjacent molecules. For an effective non-zero transport of energy in the form of heat again temperature differences of the two surfaces between which this transport occurs is necessary. When these temperatures are T_{wA} and T_{wB} in a special situation, the overall heat transfer density is (σ: Stefan–Boltzmann constant)

$$\dot{q}_w = F_{AB}\, \sigma \left(T_{wB}^4 - T_{wA}^4 \right) \tag{35}$$

The special situation behind (35) is, that both surfaces are "black surfaces", i.e. they are ideal radiators, see Incropera et al. (2006); Herwig & Moschallski (2009) for more details. Only then

both surfaces interact alone with their geometrical orientation towards each other accounted for by the view factor F_{AB}. When $(T_{wB} - T_{wA})/T_{wA} \ll 1$ one approximately gets

$$T_{wB}^4 - T_{wA}^4 = 4T_{wA}^3(T_{wB} - T_{wA}) + \dots \tag{36}$$

so that again a heat transfer coefficient

$$h = \frac{\dot{q}_w}{\Delta T} = F_{AB}\,\sigma\,4T_{wA}^3 \qquad \text{with } \Delta T = T_{wB} - T_{wA} \tag{37}$$

can be introduced for the energetic assessment.

The exergetic assessment, however, is not easily achieved. The entropy and also the entropy generation depend on various properties of the radiation, like the degree of polarisation and, more general, the deviation from the ideal black body radiation. Also, since radiation heat transfer is a mutual process between two surfaces which both are absorbers and emitters of radiation at the same time also the radiation properties of the counter surface have to be taken into account properly, see Kabelac (1994) for details. Therefore exergy loss numbers N^E according to (6) would only hold in very special situations which are of very limited interest for practical applications.

In this field more research and modelling with relevance for practical applications is needed.

6. Conclusion

The assessment of heat transfer processes is a crucial issue in the development of improved heat transfer devices. Especially nowadays where people are more and more concerned about the efficient use of energy the devaluation of the energy in a heat transfer process is of great importance. To account for this aspect of heat transfer one has to consider the second law of thermodynamics as has been shown in this chapter.

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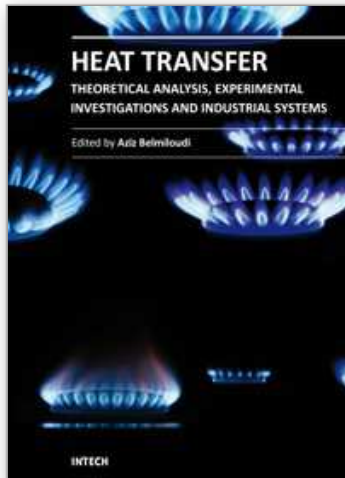
Heat Transfer - Theoretical Analysis, Experimental Investigations and Industrial Systems
Edited by Prof. Aziz Belmiloudi

Over the past few decades there has been a prolific increase in research and development in area of heat transfer, heat exchangers and their associated technologies. This book is a collection of current research in the above mentioned areas and discusses experimental, theoretical and calculation approaches and industrial utilizations with modern ideas and methods to study heat transfer for single and multiphase systems. The topics considered include various basic concepts of heat transfer, the fundamental modes of heat transfer (namely conduction, convection and radiation), thermophysical properties, condensation, boiling, freezing, innovative experiments, measurement analysis, theoretical models and simulations, with many real-world problems and important modern applications. The book is divided in four sections : "Heat Transfer in Micro Systems", "Boiling, Freezing and Condensation Heat Transfer", "Heat Transfer and its Assessment", "Heat Transfer Calculations", and each section discusses a wide variety of techniques, methods and applications in accordance with the subjects. The combination of theoretical and experimental investigations with many important practical applications of current interest will make this book of interest to researchers, scientists, engineers and graduate students, who make use of experimental and theoretical investigations, assessment and enhancement techniques in this multidisciplinary field as well as to researchers in mathematical modelling, computer simulations and information sciences, who make use of experimental and theoretical investigations as a means of critical assessment of models and results derived from advanced numerical simulations and improvement of the developed models and numerical methods.

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