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Discussion

Predicting the Performance of an Evaporative Condenser[1]

W. Leidenfrost.[2] In their introduction, Peterson et al. refer to the article "Evaporative Cooling and Heat Transfer Augmentation Related to Reduced Condenser Temperature," by W. Leidenfrost and B. Korenic, published in *Heat Transfer Engineering*, Vol. 3, Nos. 3–4, pp. 38–59 (1982). The four authors criticize this article and declare it as unnecessarily overcomplicated by comparing it with their simpler approach, which can yield satisfactory results (within 30 percent). They fail to recognize that at present high-speed computers allow more complete and more complex analysis, which yields better results. These authors state further that in the referred article errors were made by assuming that the air heat transfer coefficient for dry tubes is equal to that of wet tubes and by assuming that wetted and dry tubes have identical surface areas for heat transfer with the surrounding air. Those statements miss the fact that in the criticized article two coefficients were introduced relating wet and dry heat transfer coefficients and wet and dry surface areas. Those coefficients can be approximated to be equal to unity but can easily be varied especially in the more complicated program.

All these statements made by the four authors do not do any harm and could remain unchallenged. However, they made another statement that cannot be left unanswered. They state that the Re number of a falling film defined by Leidenfrost and Korenic as the film velocity times its thickness and divided by the kinematic viscosity of the liquid in the film is not dimensionless. It is very surprising that the four authors did not recognize a widely accepted and century-old combination of the above parameters.

Such statements cannot go unchallenged, especially not when made in the JOURNAL OF HEAT TRANSFER.

Authors' Closure

While not disagreeing that the more accurate model can be solved on a computer, in our experience with testing large evaporative condensers (those with duties of 250 kW to 2 MW), the experimental data are usually rather inaccurate. Wet bulb temperatures are notoriously difficult to measure, and the measured duty of the condenser is very sensitive to changes in this value. Thus, in these situations, the types of errors introduced in the modeling, for example assuming Lewis numbers of 1, etc., are far less than those inherent in typical experimental data.

It is also very useful when testing a working condenser to have a very simple model so that data measured on site can be checked immediately for consistency, as it has been the exception to find a refrigeration plant where all the instruments are actually working correctly. Usually the experimental procedure has to be modified on site as one decides which instruments are probably accurate.

With regard to the dimensions in the paper by Leidenfrost and Korenic (1982); we have the following problems:

- Equations (35) and (36) are not dimensionally consistent if the Reynolds number is dimensionless.
- Using the definition given in equation (37), $\dot{m}_{Wvc}$ is defined as a volumetric flow rate per unit length. The subsequent equations and the Reynolds number (equation (40)) are dimensionally inconsistent when this definition is used. The symbol $\dot{m}_{Wvc}$ is not defined in the nomenclature, but is referred to equation (37) as a volumetric flow rate.

Presumably $\dot{m}$, the mass flow rate, should have been used in equation (37), but there is no obvious or simple way in which equations (35) and (36) can be made dimensionally correct.

[1]By D. Peterson, D. Glasser, D. Williams, and R. Ramsden, published in the August 1988 issue of the ASME JOURNAL OF HEAT TRANSFER, Vol. 110, No. 3, pp. 748–753.

[2]Purdue University, West Lafayette, IN.

An Extension to the Irreversibility Minimization Analysis Applied to Heat Exchangers[1]

D. P. Sekulic.[2] The authors have presented an interesting extension to the irreversibility minimization analysis applied to heat exchangers. The objective of this communication is to call attention to some limitations of the proposed methodology.

1 The objective function [equation (12)] cannot be used to optimize a cocurrent flow heat exchanger. The objective function for this arrangement (the curve Ns versus N_{tu}) does not have an extremum for an arbitrary finite thermal size (N_{tu}) (Sekulic, 1986; Bejan, 1988). In fact, from the relation

$$\frac{\partial}{\partial N_{tu}}(Ns_{\Delta T})=\frac{\partial}{\partial \epsilon}(Ns_{\Delta T})\frac{\partial \epsilon}{\partial N_{tu}}$$

it is clear that for cocurrent flow the maximum of $Ns_{\Delta T}$ occurs only when $N_{tu}\rightarrow\infty$; in other words

$$\frac{\partial}{\partial \epsilon}(Ns_{\Delta T})\neq 0 \text{ for } N_{tu}>0, \text{ and } \frac{\partial}{\partial N_{tu}}(\epsilon)=0 \text{ for } N_{tu}\rightarrow\infty$$

In conclusion, the cocurrent flow arrangement does not enjoy the objective function optimum mentioned in the paper (for any choice of ω, N_{tu}, τ, or γ parameters). In addition, the above conclusion holds for several other flow arrangements, but for different reasons.

[1]By S. Aceves-Saborio, J. Ranasinghe, and G. M. Reistad, published in the February 1989 issue of the ASME JOURNAL OF HEAT TRANSFER, Vol. 111, No. 1, pp. 29–36.

[2]Senior Fulbright Scholar, Department of Mechanical Engineering and Materials Sciences, School of Engineering, Duke University, Durham, NC 27706; also Assoc. Professor, Department of Mechanical Engineering, University of Novi Sad, 21000 Novi Sad, Yugoslavia.

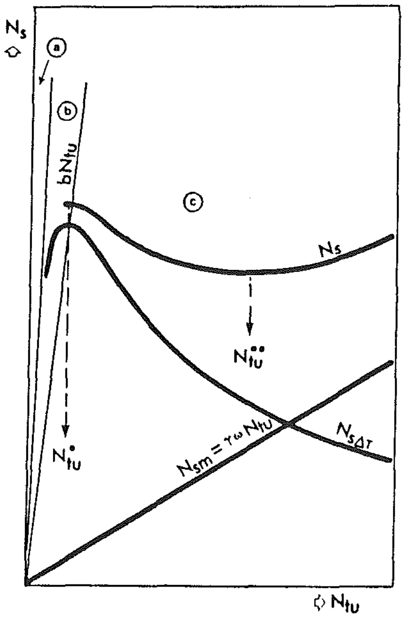

Fig. 1 Three possible orientations of the Ns_m line on the Ns-N_{tu} plane

The functional dependence $Ns_{\Delta T}$ versus N_{tu} is very weak for a number of flow arrangements in the $N_{tu} > N^*_{tu}$ region [note: N^*_{tu} at $Ns_{\Delta T} = Ns_{\Delta T,\max}$; $\epsilon^* = \epsilon(N^*_{tu}) = 1/(1+\omega)$], i.e., in this region the irreversibility decreases very slowly as N_{tu} increases. In some cases, even a local minimum is present (Sekulic, 1986). This behavior is exhibited, for example, by the crossflow arrangement with both fluids mixed, or with one fluid mixed and the other unmixed. Similar characteristics are found in some two-pass shell-and-tube heat exchanger flow arrangements.

In conclusion, the authors' statement

> "This objective function [equation (12), D.S.] can be used to optimize any heat exchanger if the ϵ-N_{tu} relationship is known and the assumptions made during the development of this equation are satisfied."

is not generally valid.

2 Formal limitation of the use of the proposed objective function is implicitly recognized for small values of ω in the paper. The limitation is the consequence of the inappropriate scaling of the heat exchanger irreversibility rate. Specifically, for $\omega = 0$ and the accepted Ns definition, both irreversibility contributions [$Ns_{\Delta T}$ and Ns_m, given by equations (11) and (9)] become zero. However, both components, "heat transfer" and "material" irreversibility rates, exist! But, even then, there is no optimum design – the same conclusion as reached in the paper.

It would be more appropriate to use the Ns definition of Sarangi and Chowdhury (1982) [$N_s = \dot{I}/(T_o C_{\min})$] or the definition used by Sekulic (1986) (entropy generation divided by the maximum entropy generation).

3 In their Fig. 4, the authors show that three limiting cases exist, depending on the actual value of capacity rate ratio and parameter γ. However, several other important effects are also present.

Regarding the slope of the Ns_m line on the Ns-N_{tu} diagram (Fig. 1 in this note), there are three important ranges ($\omega > 0$) to consider:

$$(a) \quad \gamma \geq \frac{(1-\tau)^2}{\tau}$$

$$(b) \quad \frac{(1-\tau)^2}{\tau} > \gamma \geq \frac{1}{\omega} b$$

$$(c) \quad \gamma < \frac{1}{\omega} b$$

The optimum (Ns minimum) exists only in case (c), when the slope of the Ns_m line is considerably smaller than in case (b). This is the only necessary condition for the existence of the optimum (see comment No.1).

In the case of a countercurrent heat exchanger this condition is

$$\gamma << \frac{\omega - 1}{\omega \ln \omega} \left[(\omega+1) \ln \frac{\omega\tau + 1}{\omega + 1} - \omega \ln \tau \right]$$

In conclusion, when γ and the flow arrangement are given (fixed), the existence of the optimum depends on both ω and τ.

References

Bejan, A., 1988, *Advanced Engineering Thermodynamics*, Wiley, New York.

Sarangi, S., and Chowdhury, K., 1982, "On the Generation of Entropy in a Counterflow Heat Exchanger," *Cryogenics*, Vol. 22, pp. 63–65.

Sekulic, D. P., 1986, "Entropy Generation in a Heat Exchanger," *Heat Transfer Eng.*, Vol. 7, pp. 83–88.

Authors' Closure

We want to thank Dr. Sekulic for the interest that he has shown in our work. We also recognize his important contributions in the field of heat exchanger analysis. Our response to his comments follows.

The main purpose of our paper is to develop and show the use of the material exergy method to establish nonchanging design guideposts, and we feel that the paper accomplishes this purpose. Dr. Sekulic's comments refer exclusively to the examples used to explain the method. These examples, although an important part of the paper, were used for illustrative purposes only, due to their algebraic simplicity. The examples and the optimization procedure used were never intended to be perfectly general. In the next paragraphs, we respond to the specific comments.

1 We agree with Dr. Sekulic that our wording of the statement, "This objective function can be used to optimize any heat exchanger," may cause some confusion. What we had in mind in writing the statement was that the objective function can be used in the optimization procedure for any heat exchanger, although there may be the possibility of the optimization procedure yielding no optimum heat exchanger. Notice, however, the following two aspects:

(a) Objective function (12) is still perfectly valid for any heat exchanger when all the assumptions used in its development are satisfied. The existence or nonexistence of an optimum heat exchanger is a characteristic of the heat exchanger being analyzed, and not a limitation of the method, as Dr. Sekulic indicates.

(b) We did not overlook the fact that there may be no optimum heat exchanger, even in the most favorable case of a counterflow heat exchanger. We indicate this possibility in Fig. 4 and the discussion associated with it.

2 We again agree with Dr. Sekulic that equation (12) cannot be applied for a condenser or evaporator ($\omega = 0$). However, as stated above, the formulation used was never intended to be perfectly general. Rather, it was desired to have an equation easy enough to manipulate that still preserved the characteristics associated with more general heat exchanger models.

The definition used for $N_s = I/T_0 C_{\max}$ was chosen over $N_s = I/T_0 C_{\min}$ due to its more extended use in the heat exchanger literature. The concept of writing the nondimensional equation as the ratio between the irreversibility rate and the maximum irreversibility rate, as proposed by Dr. Sekulic, was not considered due to the difficulty in implementing this method when the pressure drop irreversibility is not negligible.

3 We regard the three ranges of values for gamma obtained by Dr. Sekulic as a valuable contribution. However, this decomposition in ranges was not shown in our paper due to our feeling that, essentially, all practical heat exchanger designs are going to fall in region (C) of his diagram. Notice also that the minimum values of ω for each gamma required to obtain an optimum design are indicated in Fig. 5 as the point at which the curves start.