wood; (*b*) assumptions of isotropy, negligible internal convection, and infinite-activation-energy-pyrolysis are made; (*c*) the relevant nondimensional variables (both dependent and independent) can be identified even with linear profiles; and (*d*) the data sought to be compared with our predictions come from experiments involving considerable practical complexity, we have deliberately chosen the simplest acceptable profiles. After all, the intended use of the results should always determine the degree of approximation.

In making the second comment, which is related to where the thermal penetration occurs on the insulated backface before the exposed front face commences to pyrolyze, Yeh and Chung have failed to notice the last paragraph of section 6 of our paper. Thus, their claim that we ignored this possibility is a hasty one. If our objective were to study the charring of thin sheets and films, we would have considered this issue in greater detail.

Design and Optimization of Air-Cooled Heat Exchangers[1]

F. L. Rubin.[2] The goal of the authors to develop a computer program to design a heat exchanger without iteration has been sought by many. However, most exchangers are designed with a fouling resistance. The writer is unaware of any computer program which can design a practical heat exchanger without iteration and yet include a fouling resistance.

The abstract of the aforementioned paper implies that the information presented in the main text is applicable to the design of commercial air-cooled heat exchangers. However, the basic assumptions used to develop the solutions are never met in commerical practice.

The program "take(s) into account the variation of heat transfer coefficients and differential pressure drop with temperature and/or length of flow path . . . " The assumption of uniform air flow across the tube bundle is indicated.

Air-cooled heat exchangers utilize circular fans and rectangular tube bundles. The plenum height rarely has one-half of a fan diameter between the top of the fan ring and the heat transfer surface. The ensuing maldistribution of air results in heat transfer coefficient variations, which are greater than those due to changes in physical properties.

Tube diameters available for heat exchanger construction do not have an infinite range of sizes. Only the first of the nine tabulated sizes is actually available; it is probably used in less than 0.1 percent of all exchangers.

Similarly only a discrete number of motor sizes is available for air-cooled heat exchangers. Bundle width is severely limited by a maximum value to permit shipment and a minimum to facilitate air distribution across the heat exchanger surface. These conditions preclude preliminary definition of air quantity and air side pressure drop when designing an optimum exchanger.

"The side constraints on the design variable are of a practical nature . . . " state the authors. However only equilateral triangular pitch can be considered "practical." The case study figure is for a tube layout, which does not have equilateral spacing.

Most heat exchangers (e.g., shell-and-exchangers, spiral plate exchangers, double-pipe sections, multitube sections, plate and frame exchangers, etc.) are purchased as such and auxilliary equipment is bought separately. Air-cooled exchangers are rarely sold commercially as heat transfer surfaces only. The supplier furnishes the tube bundle, the fan(s), the driver(s), the plenum, the fan ring, etc. The concept of "minimum volume" of heat transfer surface is meaningless when one designs an air-cooled exchanger with these components.

It is most unlikely that this computer program or any minor modification of it could be used commercially for the design and optimization of air-cooled exchangers.

[1] By C. P. Hedderich, M. D. Kelleher, and G. N. Vanderplaats, published in the November 1982 issue of the JOURNAL OF HEAT TRANSFER, Vol. 104, No. 4, pp. 683–689.

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Authors' Closure

In any attempt to move forward in the field of Heat Transfer, the comments and viewpoints of a practioner are always appreciated. It appears though, that some unfortunate and unwarranted implications have been drawn by Mr. Rubin. Our goal in the paper was to demonstrate the usefulness to the heat transfer designer of the optimization techniques that have been used for over a decade in structural design. Our goal is not to replace the designer with the computer—it would be foolish to try.

The following replies are offered to Mr. Rubin's specific comments:

The method proposed is *applicable* to the design of commercial air-cooled heat exchangers. It is a tool, one of many, it is hoped, that the designer will use.

Not all air-cooled heat exchangers utilize circular fans. There are many applications in the transportation industry where ram air is utilized.

Mr. Rubin's comments on the discrete sizes of components available to the designer only points out the need to extend the techniques of heat exchanger design optimization to include more advanced optimization algorithms. More to the point, the use of continuous variables is common practice early in the design process, rapidly providing a near optimum from which to discritize the actual parameters.

To reiterate: our purpose was to to eliminate the designer but to provide tools that will make the design process more efficient. As is stated in the paper's abstract, the method we have proposed "is shown to be a useful tool for heat exchanger design."

The Optimum Dimensions of Convective Pin Fins[1]

J. E. Wilkins, Jr.[2] Readers interested in this paper, or that of Sonn and Bar-Cohen [1], may wish to consult the earlier paper of Focke [2], in which the principal results for pin fins, or circular spines, with constant, triangular, and convex parabolic ($n=2$) profiles are derived. The numerical values furnished for u^*, b^*, L^*, and V^* in Table 1 of the Razelos paper are in essential agreement with those of Focke, although there are minor discrepancies (the largest is between the values 0.4400 and 0.45 for L^* for the constant profile case by Razelos and Focke, respectively). We have resolved all of these discrepancies in favor of the Razelos results.

It is also true, as claimed by Focke but not by Razelos, that the convex parabolic profile is the optimum in the class of all profiles. This result follows from a general theory [3] for optimizing circular spine profiles for arbitrary temperature-dependent heat transfer modes, when that theory is specialized to the case of convective heat transfer with constant surface heat transfer coefficient and constant thermal

[1] By P. Razelos, published in the May 1983 issue of the ASME JOURNAL OF HEAT TRANSFER, Vol. 105, No. 2, pp. 411–413.

[2] Deputy General Manager for Science and Engineering, EG&G Idaho, Inc., Idaho Falls, Idaho 83415. Assoc. Mem. ASME. Work performed under the auspices of the U.S. Department of Energy, Contract No. DE-AC07-761D01570.