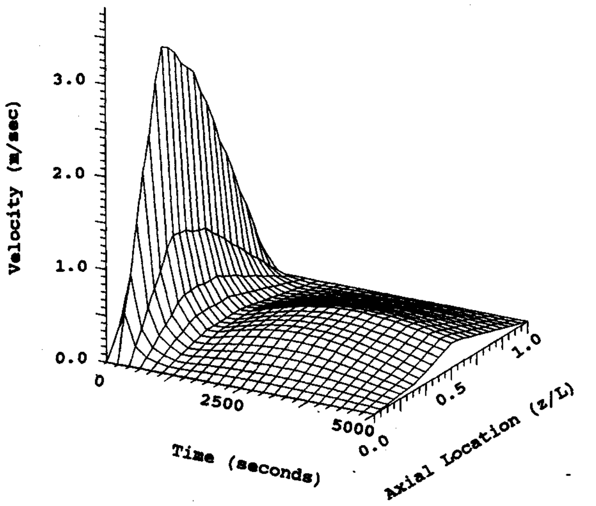

Fig. 4 Vapor velocity as a function of time and location for the heat-pipe transient.

transient, the external condenser temperature is shown to drift higher than the experimental values. This may be a result of incorrectly modeling the convection process from the condenser surface to the environment.

Figure 4 shows how the vapor velocity varies both axially and with time during the transient. The vapor velocities at the ends of the pipe were always 0 m/s as would be expected. It is interesting to note the larger velocities early in the transient. This is a result of the initially small vapor density. As the heat pipe warms up, and the vapor density increases, the vapor velocities decrease. As the heat pipe is cooled (time >1200 s) the velocity is seen to increase and then eventually decrease. Once again, this is a result of vapor density variations combined with the change in power throughput. Figure 4 shows the axial variation in velocity as well. It can be seen that the velocity always increases along the evaporator and decreases along the condenser section of the heat pipe. Along the adiabatic section of the heat pipe, the velocity is seen to decrease while the heat pipe is warming up, and increase while the heat pipe is cooling down. Velocity variations in the adiabatic section are a result of the thermal mass of this part of the heat pipe. It is felt that the model does an excellent job of modeling the heat-pipe vapor flow. The model provides the researcher with more information than can be verified experimentally.

Conclusions

This paper presents a simple way of modeling the transient vapor behavior found in heat pipes. For the model, the vapor is assumed to be quasisteady, compressible in time, incompressible in space, and one-dimensional. The model is shown to do an excellent job of modeling a heat-pipe transient. The model would be excellent for coupling with developmental code for studying more complicated heat-pipe phenomena.

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Technical Comments

Comment on "Natural Convection from Isothermal Plates Embedded in Thermally Stratified Porous Media"

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IN their presentation of the history, the authors failed to mention that the first and only time that the problem of natural convection from a vertical surface to a linearly stratified porous medium was treated in the literature was in Ref. 1. This problem is precisely the one for which Lai et al. report concrete results.

It is true that they mentioned my "integral solutions" once, in the tenth to the last line of the paper, but this does not change the impression that is conveyed in their *Introduction* regarding the history of the problem. Furthermore, my work is included in but not credited on their Fig. 2.[2] (see Fig. 1).

I take this opportunity to report an original and much simpler means of estimating the effect of stratification on the total heat transfer rate. To accomplish this, I reproduce the graph in which my 1984 solution is reported. This graph answers the question: given the *maximum* temperature difference $\Delta T = (T_0 - T_{\infty,0})$, the wall height H, and the stratification rate γ (or b, dimensionless, defined on the figure), what is the overall heat transfer rate? In this figure, the overall Nusselt number and the Rayleigh number are based on the maximum temperature difference, namely $Nu_{0-H} = \overline{q''}H/(k\Delta T)$ and $Ra_H = g\beta KH\Delta T/(\alpha\nu)$.

The filled circle drawn for the case of no stratification ($b = 0$) represents Cheng and Minkowycz' exact solution,[3] which

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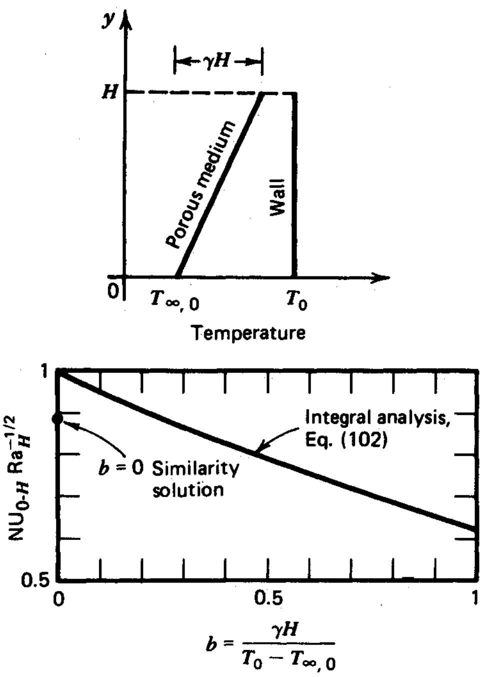

Fig. 1 Heat transfer from a vertical isothermal wall to a linearly stratified fluid-saturated porous medium.

in this notation reads

$$Nu_{0-H} = 0.888\ Ra_H^{1/2} \tag{1}$$

This shows that the integral solution overestimates by 13% the overall heat transfer rate when $b = 0$. In other words, this degree of accuracy of the integral solution was noted by me in 1984: it is not Lai et al's discovery.

The new and simpler solution is based on the observation that Eq. (1) holds approximately even when stratification is present, provided the Nusselt and Rayleigh groups are based on the *H-averaged* temperature difference

$$\overline{\Delta T} = \Delta T([1 - b/2]) \tag{2}$$

Therefore, in place of Eq. (1), we write approximately

$$\frac{\overline{q''}H}{k\overline{\Delta T}} \cong 0.888 \left(\frac{g\beta KH\overline{\Delta T}}{\alpha\nu}\right)^{1/2} \tag{3}$$

and, after reverting to the ΔT-based notation (Nu_{0-H}, Ra_H) defined earlier, we obtain approximately

$$Nu_{0-H} \cong 0.888([1 - b/2])^{3/2}\ Ra_H^{1/2} \tag{4}$$

On the attached figure, Eq. (4) would generate a curve that falls under the solid curve of the integral solution

b	0	0.25	0.5	0.75	1
$Nu_{0-H}Ra_H^{-1/2}$	0.888	0.73	0.58	0.44	0.31

However, Eq. (4) has the advantage that it is exact in the limit $b \rightarrow 0$. It would be interesting to compare Eq. (4) with the $Nu_{0-H}Ra_H^{-1/2} = f(b)$ curve that would result from the local nonsimilarity analysis employed by Lai et al. (note that Lai et al. did not report the overall Nusselt number).

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Reply by the Authors to A. Bejan

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WE thank Dr. Bejan for his comments on our paper, and we offer the following in reply.

As stated in the introduction, the focus of the paper[1] is to extend the earlier study of Nakayama and Koyama[2] to include a more realistic temperature stratification that may be encountered in applications. Our presentation of the history of the problem may be incomplete, and we appreciate Dr. Bejan's reference to his earlier study.[3] However, we have given his work credit in the comparison of our results with his integral solution. More importantly, we do not claim the discovery of the discrepancy found in the solutions we present. It is well known that the accuracy of an integral solution is largely dependent on the approximate solution form and that accuracy can be greatly improved if a better functional form can be adopted. This is exactly the conclusion of our paper.

Although our results did not include the overall heat transfer rate, the calculation of this value is straightforward. Following the notation used by Bejan, we have:

$$Q = \int_0^H - k\left.\frac{\partial T}{\partial x}\right|_{x=0} \mathrm{d}y$$

$$= k(T_0 - T_{\infty,0})Ra_H^{1/2}b^{-1/2}\int_0^b -\Theta'(\xi, 0)\xi^{1/2}\,\mathrm{d}\xi \tag{1}$$

The result is most informative if the overall heat transfer rates thus obtained are compared to those of a plate in an isothermal environment; that is, no thermal stratification. The

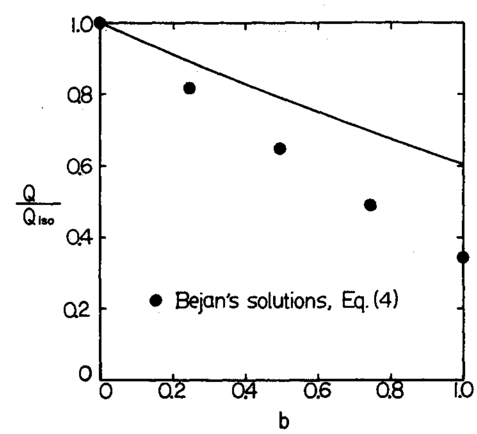

Fig. 1 Ratio of overall heat transfer rates, Eq. (3), as a function of the stratification parameter, *b*

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