We would hope this discussion might make some additions in this regard.

In the entrance to the mixing section where the annular flow pattern can be assumed, the authors find that their model predictions are strongly dependent on the vapor condensing rate and the transfer of momentum via the interfacial shear stress. To achieve agreement with their data, arbitrary values of $\bar{h}$ and $f_{SV}$ were used to calculate the interfacial heat transfer rate and shear stress. As the authors point out, the values assumed can be generally supported on the basis of similar experience in the literature. However, physical models for predicting the vapor condensation and interface shear stress have been proposed. To predict the condensation rate, a model based on the analogy with Prandtl mixing length theory for free turbulent jets yielded a liquid jet Stanton number, St, equal to a constant, 0.012, which must be empirically determined. As shown in [21], the interface shear stress in the presence of condensation is augmented by a term which is proportional to the vapor condensation rate and the free stream vapor velocity,

$$\tau_{SV} = f_{SV} \frac{1}{2} \rho_L (V^2 - V_0^2) + V_0 \frac{M_v}{2\pi r}$$

In the paper by the discussers, it was found that with this formulation, the variation of the magnitude of $f_{SV}$ from 0.04 to 0.4 made little difference in the annular flow model predictions. Satisfactory comparison of the model to data similar to Fig. 15 and to data of [17] could be made. In view of the complex nature of the transport mechanisms in the condensing ejector as illustrated in Fig. 2, the annular flow model should not be taken literally. On the basis of our experience, this model appears most useful in the regime I identified by the authors as the high liquid inlet velocity regime. It is felt, however, that a somewhat more general identification of regime I as well as the others identified by the authors should be based on an energy criteria referred to the inlet flow rate of the vapor times its latent heat and the inlet liquid flow rate times its subcooling. If the ratio $R$ is defined as $R = W_{LV}/(\Delta h_{in} = \bar{h}_{LV, in})$ where $\Delta h_{in}$ is evaluated at the inlet pressure, the following tabulation can be made from Eq. 4.

<table>
<thead>
<tr>
<th>Table</th>
<th>Inlet thermal energy ratio for data of Fig. 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Run</td>
<td>A</td>
</tr>
<tr>
<td>$R$</td>
<td>0.29</td>
</tr>
</tbody>
</table>

It could be inferred from the authors' discussion that Runs H and I, $R \geq 1$, would fall into regime III while Runs D to G might be regime II, and runs A to C, $R < 1$, would be characteristics of region I. This approach can accommodate both the effects of variations in inlet liquid velocity as well as variations in liquid inlet subcooling on the disintegration of the annular flow pattern as was found in [17]. Certainly, the annular flow pattern breaks down rapidly in these geometries when $R \geq 1$. It is also interesting that the rapid deterioration of the pressure performance of the device as shown in Fig. (14) can also be related to conditions where the inlet energy ratio $R = 1$.

### Discussion

M. A. Grohms and J. H. Linehan

The authors are to be complimented for their work. In this device a variety of complex two-phase flow problems must be sorted out, and the authors make reasonable attempts to do this.
Authors' Closure

The authors are grateful to Drs. Grolmes and Linehan for an excellent discussion of the paper. We agree with them that the method they used to calculate the interfacial shear stress is a step beyond that used in our analysis. As they point out, their method requires that one furnish only a value for $h$ rather than values for both $h$ and $f_{L}P_{V}$. They did not indicate in their paper (discusser's reference [4]) whether they obtained good agreement between their calculated profiles of vapor Mach number and the experimentally determined values shown in Fig. 16. Certainly such agreement for both pressure and Mach number would be a strong argument for using a particular analytical model. The heat capacity criterion which they propose for predicting jet condenser overall performance seems consistent with the data reported here and helps to explain many of the phenomena which have been observed.

Despite the simplifications offered by their analysis, the problem of acquiring a fundamental understanding of the heat transfer processes which govern the interfacial condensation rate still remains. Until a method is developed for evaluating $h$ independently, the mixing section analysis cannot be considered a complete design tool.