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《Review》

Effect of Transverse Convex Curvature on Turbulent Fluid Flow in Fuel Channel

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핵연료 수로내 난류 유동에 대한 횡방향 볼록구배의 영향

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Abstract

Nuclear fuel bundles are designed such that the heat flux at a fuel pin surface should not exceed the critical heat flux (CHF) during normal operation and anticipated transients. Therefore, evaluation of the CHF for fuel bundle is demanded in an exact and reliable manner. One of the major concerns with the current application of CHF correlations is that the CHF based on circular tubes is applied to the fuel bundle subchannel analysis, mainly in terms of the hydraulic diameter with correction factors which may result in a source of possibly large uncertainties in CHF prediction. The hydraulic diameter does not recognize the local properties of fluid nor such effect as the surface curvature; the turbulence action on the convex surface is much more pronounced than that on the concave surface. Even for the tube having concave curvature, the effect of tube diameter on CHF becomes important with decreasing diameter. These facts imply that the convex curvature effect is significant and crucial to the reliable CHF prediction. This paper reviews and discusses analytical and experimental aspects of effect of transverse convex curvature in incompressible turbulent flow and heat transfer, and on CHF. Flow models to quantify this effect are briefly mentioned and future works are recommended.

요 약

핵연료 다발의 설계는 핵연료 표면에서의 열속이 임계열속에 도달되지 않도록 이루어져야 한다. 이를 위해서는 설계된 핵연료 다발에 대한 임계열속이 정확하고 신뢰성있게 평가되는 것이 매우 중요하다. 그

러나 현재 사용되고 있는 임계열속 상관식에 관한 쟁점 대상중의 하나는 원통튜브관에서 실험적으로 얻어진 임계열속 자료가 유효직경 형태의 변수와 적당한 보정인자를 사용하여 핵연료다발 부수로 분석에 사용되고 있다는 것인데, 이러한 방법은 임계열속 예측에 있어서 불확실도 요인으로 작용하고 있다. 유효직경은 유동단면적상의 국부적 유체 특성을 제대로 표현하지 못 할 뿐만 아니라 표면구배효과 등을 고려할 수 없다. 더구나 난류유동은 오목구배면에서 보다는 볼록구배면에서 더욱 두드러진다. 즉, 횡방향 볼록구배면이 오목구배면 보다 유동의 반경방향으로의 난류 형성에 영향이 훨씬 크게 나타나는데, 이는 정확한 핵연료 임계열속 평가에 있어서 볼록구배의 영향이 반드시 고려되어야 함을 암시하는 것이다. 본 논문에서는 횡방향 구배의 유동영향에 대하여 전반적으로 심도있게 고찰하고 임계열속에 대한 영향이 논의 되었으며, 이 영향을 정량화하기 위하여 고려되어야 할 유동 모델과 향후 연구 방향이 제시되었다.

1. Introduction

Nuclear fuel bundles are designed such that the heat flux at any fuel pin surface should not exceed the critical heat flux (CHF) during normal operation and anticipated transients. Therefore, evaluation of the CHF for fuel bundle is demanded in an exact and reliable manner. One of the major concerns with the current application of CHF correlations is that the CHF based on circular tubes has been used in the fuel bundle subchannel analysis, mainly in terms of the hydraulic or equivalent diameter with a multiple correction factor which may result in a source of possibly large uncertainties in CHF prediction.

It must be realized that the so-called hydraulic diameter has no physical foundation for its derivation. The hydraulic diameter does not recognize the local properties of fluid at such location as at the minimum gap where the condition is very different from the overall properties for the cross section nor such effect as the surface curvature; the turbulence action on the convex surface is much more pronounced than that on the concave surface.

While it is generally well recognized that any extrapolation of empirical correlations to beyond the conditions of their data range generally induces serious errors and, therefore, it should be employed with caution, it is somehow not fully recognized that the application of CHF correlations based on circular tube geometry to the fuel bundle subchannel which has non-circular flow channel is terribly improper.

An exhaustive literature survey on the effect of transverse surface curvature indicates that transverse curvature (TVC) effects have been observed in three types of experiments:

(i) the axisymmetric boundary layer developing on a long slender cylinder placed axially in a uniform stream[1],

(ii) the turbulent flow along the transversely convex surface of the core tube in annular space formed by two circular tubes[2, 3], and

(iii) the critical heat flux occurring on the surfaces of concentric annuli[4].

In the case (ii) above, it was noticed that while the mechanism of the flow outside the radius of maximum velocity is very similar to that occurring in circular pipe flow, this is not true for the flow inside the radius of the maximum velocity. It is now well documented that many turbulence models for the prediction of flow field are not fully adequate for the inner velocity distribution of a concentric annular flow channel, and thus the standard universal velocity is not fully adequate for the inner turbulent velocity distribution of a concentric annulus. For example, while a fixed value between 0.36 and 0.41 for the von Kármán constant, k_o, for the outer region is usually taken, the value of the constant for the inner region, k_i, must be calculated from other appropriate boundary conditions[5].

This implies that whereas the transverse concave curvature of the surface has little effect on the radial development of flow turbulence, that of the transverse convex curvature seems to be significant. This hypothesis is different from that of Cebeci[6] who stated that the transverse-curvature effect becomes significant whenever the radius of a body in a viscous flow is of the order of magnitude as the thickness of the boundary layer. If this is true, the turbulent flow in the entrance region of flow ducts must also be affected by the transverse concave surface curvature. The available experimental results do not support this hypothesis for the concave surface curvature. Indeed, it is only with the convex curvature of wall surfaces where the transverse-curvature effect becomes significant. This must be because the freedom of movement for the "turbulent fluid lumps" in the flow is different for concave and convex surfaces.

While there are some limited number of studies on the effect of the convex transverse-curvature on fluid flow[7, 8], there seems to be very few studies made on the effect of transverse convex curvature on convective heat transfer and critical heat flux[4, 9, 10].

In this paper, analytical and experimental aspects of the effect of transverse convex curvature in incompressible turbulent flows and heat transfer, and on CHF are reviewed and discussed. And analysis models and some future works are proposed.

2. Heat Transfer Characteristics due to Transverse Concave Curvature of the Heating Surface

For single phase flow, both laminar and turbulent, the effect of transverse concave curvature on fluid flow and heat transfer has been assumed negligible when the relevant parameters are expressed in terms of such dimensionless numbers as Reynolds number, Nusselt number etc. The effect of transverse concave curvature on critical heat flux (CHF), however, has been dealt in its own way. For the tube having concave curvature, the effect of tube diameter on CHF becomes important with decreasing diameter[11-13]. The effect of the transverse concave surface curvature is generally quantified by the following equation:

$$(CHF)_D/(CHF)_{D=8} = (8/D)^n, \tag{1}$$

where n is a function of mass flow rate and thermodynamic quality, D is the diameter of the tube and the subscript $D=8$ means the reference tube of 8mm in diameter.

3. Flow Characteristics due to Transverse Convex Curvature of the Heating Surface

3.1. Turbulent Flow and Heat Transfer in a Concentric Annulus

It is seen that attempts to correlate velocity distribution in the annulus have been largely influenced by the well established results for the pipe. To study the velocity in turbulent flow in the annulus which has both concave and convex surfaces, it is first necessary to ascertain that the assumptions made in the theory are adequately fulfilled by the flow conditions in the annulus. In this connection, the turbulence characteristics in the annulus will be examined along with those for the simpler pipe flow.

Turbulence Characteristics

It appears that little has been done in the measurement of turbulence characteristics in annular flows. Data on velocity fluctuations in turbulent flow in an annulus can be found in the work of Brighton and Jones[14]. Some of data by Brighton and Jones has been compiled in a form which is more suitable for present purposes, and is shown in Fig. 1, along with some results of Laufer for pipe flow[15, 16]. We note that in the similarity theory, the ratios $\overline{u_1^2}:\overline{v_1^2}:\overline{w_1^2}:\overline{u_1v_1}:\overline{v_1w_1}:\overline{u_1w_1}$, should be constant. Here, u_1, v_1 and w_1 are the fluctuations of velocity, and u, v and w velocities (u in the direction of the mean flow).

This is clearly not so in either flow geometry, the greatest departure from the assumption occurring in

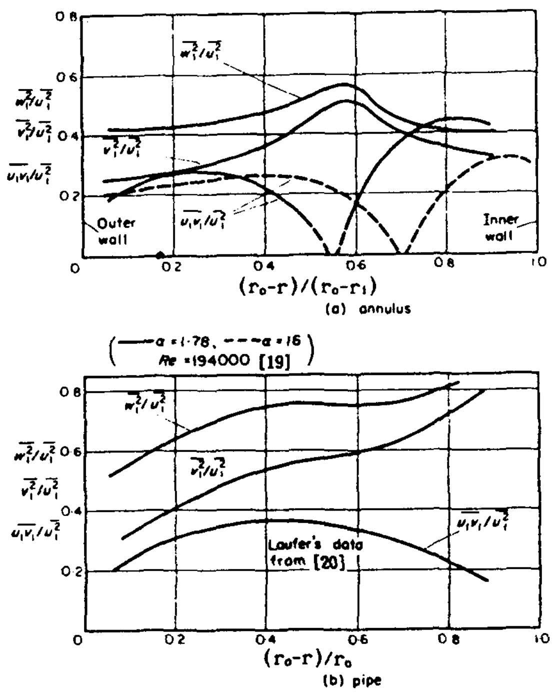


Fig. 1. Turbulance Characteristics; Pipe and Concentric Annulus

the region of zero stress and near the wall where for example the value of $\overline{u_1v_1}$ approaches zero. In these regions, therefore, we could not expect good agreement with theory in either geometry. The most interesting feature of the data for the annulus geometry is the variation in the quantity $\overline{u_1v_1}/\overline{u_1^2}$ across the flow section and its dependency on the radius ratio, $\alpha=r_o/r_i$, where subscripts o and i indicate the outer and inner radii, respectively. In the outer region $\overline{u_1v_1}/\overline{u_1^2}$ is fairly constant, whereas inside the radius of maximum velocity there is a noticeable dependency of the value of this ratio on both position and radius ratio. Brighton and Jones[14] have already observed that the pronounced curvature of the curve of $\overline{u_1v_1}/\overline{u_1^2}$ in the inner region of an annulus is accompanied by a marked departure of the velocity distribution from the law of the wall. The large variation of $\overline{u_1v_1}/\overline{u_1^2}$ in the inner region must also lead to large deviations from a prediction based on the similarity hypotheses which assumes a constant value for this ratio. It can be clearly seen by comparing Figs. 1(a) and 1(b), that the magnitude and distribution of the turbulence characteristics in the pipe and annuli are not markedly different outside the radius of maximum velocity, r_m, but inside r_m, there is poor agreement particularly as far as the ratio involving $\overline{u_1v_1}$ is concerned.

Similarity Hypothesis

The laws of the velocity distribution in turbulent flow adjacent to a solid boundary and in ducts are essentially of a semi-empirical nature. Since the ideas, concepts and theories are well known, one can proceed to derive a velocity law for turbulent flow in an annulus, using Goldstein's[17] extension of the similarity hypothesis of von Kármán. It is felt that the observations on the similarity between the turbulence parameters in pipe and annulus provide adequate justification for using similar approaches for the determination of velocity distributions. This is certainly true in the outer region; in the inner region of an annulus, this approach is questionable and less likely to yield a good correlation. Probably the simplest approach is to employ the familiar logarithmic law for zero pressure gradient that is:

$$u^+ = (1/k)\ln y^+ + B, \tag{2}$$

where $u^+=u/u_\tau$, u=velocity, u_τ=friction velocity, $y^+=yu_\tau/\nu$, y=distance from a wall, ν=kinematic viscosity, $u_\tau=\sqrt{\tau_\omega/\rho}$, τ=shear stress, ρ=density, and B and k are constants.

According to Goldstein[17], this equation might be expected to apply to pressure flows to a first approximation. In an annulus, Eq.(2) has been employed with the same constants as those used in the pipe. For example, Deissler and Taylor[18] have used k=0.36 and B=3.8 for the regions inside and outside the radius of maximum velocity, but in the light of reliable measurements[19], it is debatable whether or not a single form of the equation is valid.

A further difficulty exists in the case of turbulent annular flow. The position of zero shear is unknown and as a consequence, the wall stresses cannot be determined. It is sometimes assumed that the radii of zero shear and maximum velocity and the radius given by Lamb[20] are coincident. It would appear that an equation of the form,

$$u^{+} = f_1(\alpha)\frac{1}{k}\ln y^{+} + f_2(\alpha)B, \tag{3}$$

would be better in keeping with the pipe result, but an examination of the experimental data suggests that modification of the pipe equation is necessary only in the region inside r_m. Rothfus, Monrad and Senecal[21] have used Eq.(3) for the annulus, the "wall distance" being a complex function of the actual wall distance. With $u^{+}=u/u_{\tau 2}$ (subscript 2 indicates outside of r_m), the equation was found to be valid both inside and outside r_m and data[19] were in support of this correlation.

Two- and Three-Equation Turbulence Models

Only numerical study based on two- and three-equation models applied for the turbulent heat transfer in a concentric annulus is that of Fujii et al.[9] Three turbulence models used are k-ε, k-ε-$\overline{u_1v_1}$ and k-kL-$\overline{u_1v_1}$ and the numerical results are compared with their own heat transfer measurement. No velocity distribution was reported.

Recognizing that the previous assumption for eddy viscosity in turbulence is basically not applicable to the annular geometry because of $-\overline{\rho}\,\overline{u_1v_1} \neq \mu_t\, \partial u/\partial r$, where μ_t = total viscosity, they have expected that the popular k-ε model will not do very well for the turbulent heat transfer in a concentric annulus. The standard k-ε model proposed by Jones and Launder [22] was used. The k-ε-$\overline{u_1v_1}$ model was obtained by a slight modification of Reynolds stress model which Hanjalic and Launder[23] proposed. For the k-kL-$\overline{u_1v_1}$ model, new equations are obtained by adding the Reynolds stress term to Rotta's k-kL two-equation model modified by Kawamura[24].

Comparison Between the Theoretical Prediction and Experimental Results in Annular Flow

Figs. 2(a) and 2(b) show a wide variety of experimental data for annular flow plotted in the usual manner, the Reynolds numbers and the radius ratios being as indicated.

A close examination of the points in Fig. 2(a) (outer region) shows that, with exception of the data for $\alpha=80.72$, there is no detectable dependency of Reynolds number. The annulus, for which=80.72, shows[19] a radius of maximum velocity very much less than predicted by Lamb[20] for laminar flow. The velocity distribution outside of the radius of maximum velocity is obtained as:

$$\frac{(u_m-u)}{u_{\tau 2}} = \frac{\sqrt{3}}{k_2\sqrt{(1-\theta_m^2)}}\int_{\theta m}^{\theta}\frac{\theta}{\sqrt{(1-\theta^3)}}\,d\theta + b_2, \tag{4}$$

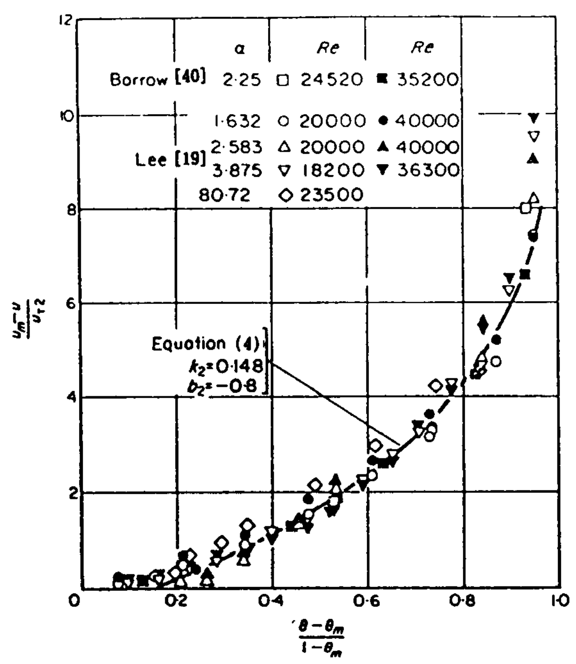
where the subscript m indicates values at r_m, $\theta = r/r_o$ and the subscript 2 indicates outside of r_m. The data for the more practical values of α are in very good agreement with the theoretical curve over the middle region, when $b_2=-0.8$ and $k_2={}^{+}0.148$.

The theoretical velocity distribution inside of the radius of maximum velocity obtained is given by:

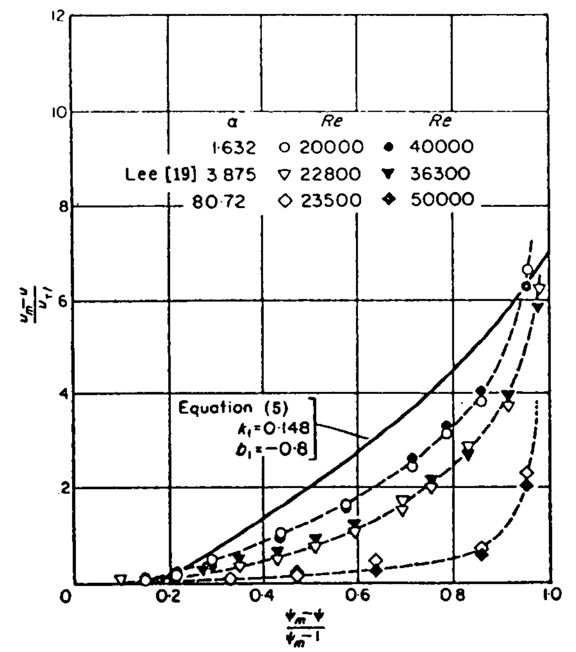
$$\frac{(u_m-u)}{u_{\tau 1}} = \frac{\sqrt{3}}{k_1\sqrt{(\psi_m^2-1)}}\int_{\psi}^{\psi m}\frac{\psi}{\sqrt{(\psi^3-1)}}\,d\psi + b_1, \tag{5}$$

where $\psi = r/r_i$ and the subscript 1 indicates inside of r_m. The corresponding values for the inner region are plotted in Fig. 2(b). Here is shown little effect of the Reynolds number but a marked dependency on the radius ratio. It is evident that it is impossible to correlate the inner velocities by a single equation but Eq. (5) with $b_1=-0.8$ and $k_1={}^{+}0.148$ is shown for comparison purposes.

A possible reason for the discrepancy between theory and experiment for the region inside r_m has already given previously. The assumption that $(\overline{u_1v_1}/\overline{u_1^2})_1$ is constant is not fulfilled there, and the value of this ratio appears to be a function of both position and radius ratio. The excellent agreement



(a) Outside Radius of Maximum Velocity



(b) Inside Radius of Maximum Velocity

Fig. 2. The Velocity Distribution in Turbulent Flow in an Annulus and its Correlation

between theory and experiment for practical radius ratios outside the radius of maximum velocity is in accordance with the constancy of $(\overline{u_1v_1}/\overline{u_1^2})_2$ in that region.

The comparisons between experiments and predictions for the heat transfer coefficient and apparent friction factor in the axial direction showed that there is little discrepancy among three turbulence models. As heat flux becomes higher, heat transfer coefficient predicted by the k-ε-$\overline{uv}$ and k-ε models are about 20 percent lower that the experimental values. On the other hand, it is reported[9] that a satisfactory agreement was obtained with the k-kL-$\overline{uv}$ model as seen in Fig. 3. These authors, however, did not discuss the effect of convex surface aspect.

3.2. Axisymmetric Boundary Layer developing on a Long Slender Cylinder

It has been shown above that while the mechanism of the flow outside the radius of maximum velocity, in the turbulent fluid flow in a concentric annulus, is very similar to that occurring in the circular pipe flow, this is not true for the flow inside the

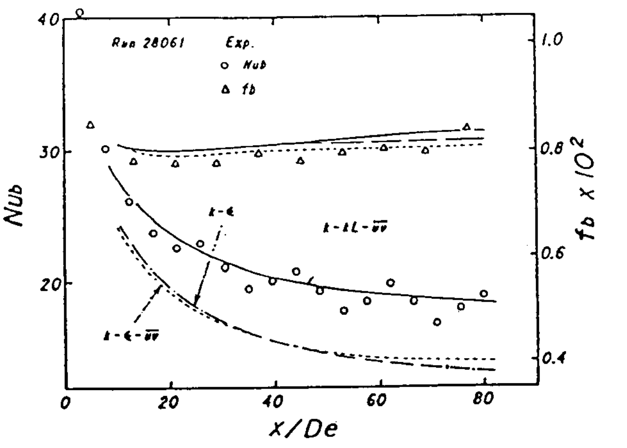


Fig. 3. Comparison of Numerical Analysis with Experiment (Re=10,500, α=0.69)[9]

radius of the maximum velocity.

Many experimental[25-27] and theoretical[28, 29] studies for the TVC effect on the fluid flow have been made for decades. However, those are limited to velocity profiles and in the experimental studies, skin friction is deduced using various indirect methods. Recently, there has been a few attempts[30, 31] to investigate turbulence structure in a thick axisymmetric boundary layer.

On the other hand, there seems to be hardly any study explicitly made on the effect of transverse convex curvature on heat transfer except that of an implicit analytical study made by Sparrow et al.[32] In this study, the expression of Deisller for the eddy diffusivity was used, retaining the same constants used for the flow in pipes and over a flat plate, and therefore, the TVC effect was not explicitly recognized.

The effect of transverse convex curvature on incompressible turbulent flows and convective heat transfer was analyzed,[33] based on a model of modified mixing length by Hornby, Mistry and Barrow.[34] The modified model is an extension of van Driest's model for turbulent flow near the wall to cater for a wider range of flow geometries by including the influence of the whole boundary of the flow, and is the only turbulence model which recognizes the flow boundary shape. The results from the analysis using this particular model for cylindrical bodies of different diameter are compared with those of flows over a flat plate.[33] The analysis was also compared against previously reported results.[27, 32, 35] For heat transfer analysis, the numerical calculations are made for the Prandtl numbers of 0.7 and 7.

The study demonstrated as shown in Figs. 4 to 6:

(i) both the friction coefficient and Stanton number increase with decreasing value of cylinder radius and their values are always greater than those for the flows over a flat plate for the ranges of parameter studied,

(ii) both the velocity and temperature profiles agree well with the measurements,

(iii) the frictional Reynolds number, $r_o^+ (=r_o u_\tau/\nu)$, and Reynolds number, $Re_{ro} (=r_o u_\infty/\nu)$, are two dominant parameters in describing TVC effect. However, the study indicates that there could be another minor parameter, such as δ/r_o or x/r_o (where δ = the boundary layer thickness, r_o = the radius of the cylinder, x = the axial distance), and

(iv) the TVC effect on heat transfer diminishes with increasing Prandtl number.

3.3. Surface Curvature Effect on Critical Heat Flux

One can clearly see the transverse surface curvature effect in the experimental results of Becker et al.[4, 36] as shown in Figs. 7 and 8. It seems that the CHF for internally heated surface of the annulus occurs at lower steam quality than the CHF for externally heat surface for the same inlet conditions.

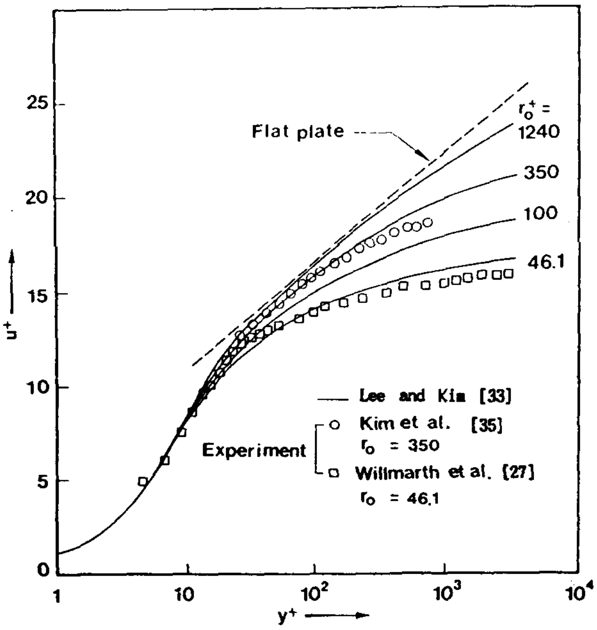


Fig. 4. Velocity Profiles, Boundary Layer over a Long Slender Cylinder

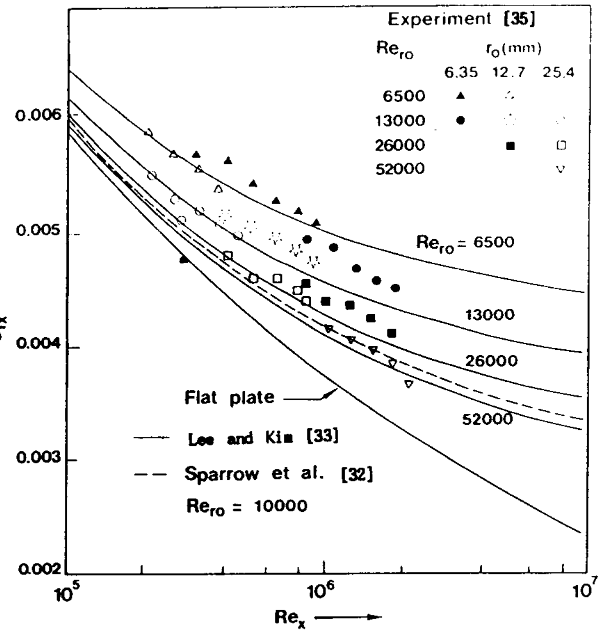


Fig. 5. Friction Factor, Boundary Layer over a Long Slender Cylinder

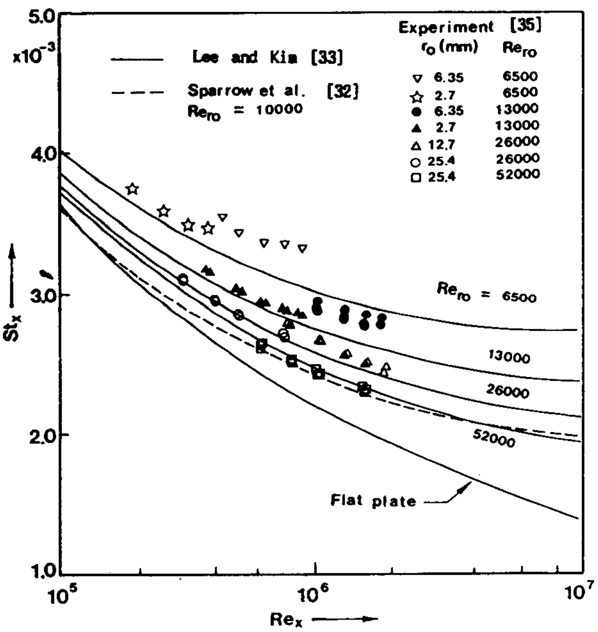


Fig. 6. Heat Transfer, Boundary Layer over a Long Slender Cylinder

For the same exit steam quality, the CHF at the inner surface is lower than the outer surface CHF and the difference seems to disappear at high pressure and mass flux, which implies that the effect of surface curvature igh pressure and mass flux is less important because at these conditions the flow can be considered to be homogeneous and the mechanism for CHF becomes different from the other conditions.

The effect of surface curvature on CHF must be related to the entrained droplets in the system. Whally[37] mentioned that the entrained droplets travel in straight line in the radial direction from the point where they are entrained to the point where they are deposited. In a concentric annulus, a droplet entrained from the inner convex surface can not be deposited at the same surface and it will necessarily hit the facing outer concave surface or remain entrained. A droplet entrained from the outer concave surface, however, can be deposited at either surface. Therefore, for the annular flow regime, the liquid film will be depleted faster for a convex heated surface.

Becker and Letzter[4] concluded that burnout in a channel does not necessarily occur on the wall, which has the highest heat flux. At 70 bar pressure and a mass velocity of 1,000 kg/m^{2}s, which are very close to BWR operating conditions, they found a ratio of 2.39 for simultaneous burnout on both walls of a concentric annulus. They suggested that in order to improve the burnout predictions for rod bundles obtained by subchannel analysis, it is necessary to incorporate a film flow model in the subchannel analysis. Jensen and Mannov[38] who have also studied bilaterally heated annuli have observed the similar trends as observed by Becker and Letzter[4] above.

4. Development of Models in Turbulent Flow for Fuel Subchannel

As stated previously, there seems to be hardly any study explicitly made on the effect of transverse con-

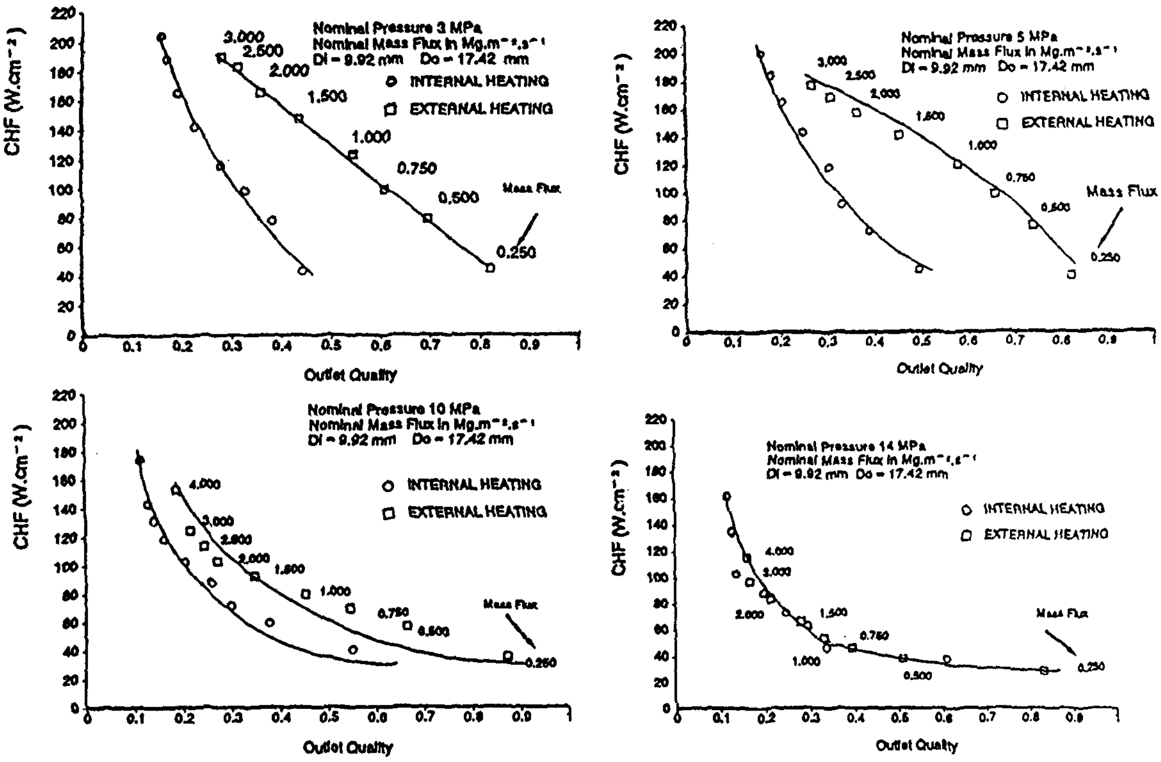


Fig. 7. Surface Curvature Effect on CHF in Annuli[36]

vex curvature on heat transfer except that of an implicit analytical study made by Sparrow et al.[32] A turbulence model of modified mixing length by Hornby, Mistry and Barrow[34] seems to be the only one which explicitly account for the effect of transverse convex curvature on incompressible turbulent flows and convective heat transfer.

Other turbulence models to account for the effect of the transverse surface curvature effect in a concentric annulus are those of Fujii et al.[9], although these models did not explicitly recognize the transverse curvature effect. They have not reported any velocity distribution and it seems that, at low heat flux, the results with these two models are about the same as those obtained with the ever popular k-ε model which is basically not applicable to the inner region of a concentric annulus because of $-\rho\overline{u_1 v_1} \neq \mu_t \partial u/\partial r$.

As stated in the section 3.1 above, the good agreement obtained by the k-kL-$\overline{u_1 v_1}$ model with the experimental results at extremely high heat flux of Fujii et al.[9] still requires further attention. The two- and three-equation turbulence models would not recognize the transverse surface curvature effect and it can not be always expected that the models and experimental results have a good agreement. There is accordingly a need for further detailed measurements of the nature and structure of turbulent flow in the annulus geometry to develop a new turbulence model which explicitly recognizes the transverse surface curvature effect.

For the flow near the wall, various composite mixing-length and eddy-viscosity formulae have been suggested by a number of workers on the basis of

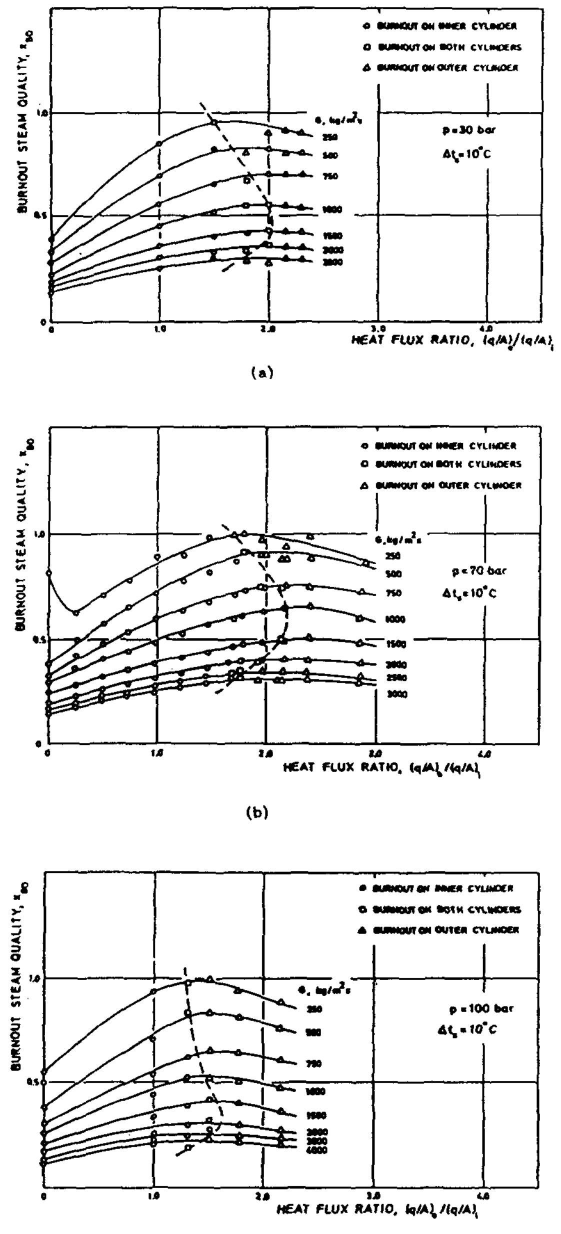


Fig. 8. Burnout Steam Quality versus Outer to Inner Heat Flux Ratio[4]

empirical correlation of experimental observation. As previously discussed, Hornby, Mistry and Barrow[34] extended the van Driest's turbulence model which made use of Stokes Second Problem which is concerned with laminar flow near an oscillating plate. The solution of the momentum equation is known from the analogous heat conduction problem.

The van Driest's model is basically for a flat plate and does not recognize the influence of the duct shape. To obtain a new model which takes cognizance of the duct shape, Hornby, Mistry & Barrow obtained the velocity distribution solving the laminar momentum equation with the whole duct oscillating. Once the fluid velocity was obtained, the damping factor for turbulent flow in ducts of any shape was acquired, applying van Driest's analogy. The resulting damping factor is function of both wall distance and duct geometry.

To illustrate the new model, Hornby, Mistry & Barrow gave three examples[34]: plane parallel flow, pipe flow and annular flow. Applying this method to the external flow along a cylindrical body, Lee and Kim [33] have recently obtained a new expression as the mixing length distribution. The turbulence model needs, further, to be applied in the analysis of fluid flow and heat transfer in the inner region of concentric annuli having the same radius ratio, but with different core diameters to identify the effect of TVC and to be incorporated into two- and three-equation models of turbulence.

It has been generally accepted that, in a concentric annulus which is the simplest geometry for nuclear fuel bundles, the pressure drop for the fully developed flow along the channel is only a function of Reynolds number with a physically meaningful dimension and the radius ratio of the outer and inter tubes; i.e., $f=f(Re_{De}, \alpha)$ where f = friction factor, Re_{De} = Reynolds number based on the equivalent diameter, De, and the radius ratio of annuli, α. For the heat transfer, it is $Nu=Nu(Re_{De}, \alpha, Pr)$, where Nu = Nusselt number and Pr = Prandtl number. However, recent studies have shown[33, 35] that this is not quite correct. Both the functions f and Nu must also include the Reynolds number based on the diameter of the inner tube, Re_{Di}, to account for the effect of surface convex curvature on both momentum and heat transfer. Therefore, it can be deduced that CHF must also be affected by the convex curvature of the inner tube, D_i.

The functional relationships for both the momentum and heat transfer in a fully developed turbulent flow in concentric annuli are believed to take the forms of $f = f(Re_{De}, \alpha, Re_{Di})$ for the momentum transfer and $Nu = Nu(Re_{De}, \alpha, Pr, Re_{Di})$ for the heat transfer.

At the moment, there seems to exist no theoretical model to predict CHF in tubes or in subchannel which accounts for the transverse surface curvature of heating surface, nor expected that there will be one soon.

5. Concluding Remarks

It has been demonstrated through the results for the concentric annulus that the effect of surface curvature on fluid flow and heat transfer including CHF is very closely related to the design of thermo-hydraulics and the safety analysis of nuclear fuel assemblies which are basically an assembly of fuel elements having surfaces with transverse convex surface curvature.

It has been shown that while the effect of the transverse concave surface curvature on fluid flow is negligible, that of transverse convex curvature is significant.

The good agreement obtained by the k-kL- $\overline{u_1 v_1}$ model with the experimental results of Fujii et al. still requires further consideration. The two- and three-equation turbulence models would not recognize the transverse surface curvature effect and it can not be expected that there could be any good agreement between the models and experimental results. There is accordingly a need for further detailed measurements of the nature and structure of turbulent flow in the annulus geometry.

Most of the current CHF correlations are based on circular tubes and are applied to the fuel bundle subchannel, mainly in terms of the hydraulic diameter with a multiple correction factor and this would result in a source of possibly large uncertainties in CHF prediction.

It is seen that the transverse surface curvature influences the values of CHF from the experiments in concentric annuli and that the CHF for a concave surface (the outer surface) is significantly higher than that for a convex surface (the inner core). This implies that the tube-based CHF correlations should not be applied to the flow outside of tube bundles or in subchannel whose heated surfaces are convex.
Since there exists no workable theoretical model nor the experimental results to quantify the effect of the convex curvature on fluid flow, heat transfer and especially on CHF, any experimental verification of the CHF of nuclear fuel bundles must come from the fuel-bundle simulators of one-to-one bundle lattice geometry.

It must be stressed that with the full scale electrically-heated multi-rod fuel-bundle simulators of the exact lattice geometry, the condition of surface curvature effect is satisfied. However, the condition of thermal conjugation[41, 42] is still not fulfilled if the direct electrical heating is used in the full scale simulator experiments. It must be noted that the effects of heating surface curvature and conjugation are not easily separable.

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