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Investigation of Spacer Grid Thermal Mixing Performance Based on Hydraulic Tests

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Abstract

An evaluation method of spacer grid thermal mixing performance in rod bundles is suggested based on hydraulic tests in a single phase flow. Heat transfer correlation was derived by the analogy between momentum and heat transfer. Three of major factors, such as blockage ratio of spacer grid, convective flow swirling, and turbulent intensity, were found to be significantly influential to the spacer grid thermal mixing performance. Local heat transfer near spacer grid was predicted for the hydraulic test of 6 x 6 rod bundles with neighboring different spacer grids.

1. Introduction

The evaluation of the thermal mixing performance of the newly designed spacer grid is very important in nuclear fuel design. However, the thermal tests of the fuel assembly with the new spacer grid are costly and protractive. As a alternative or first step, hydraulic tests are performed in a purpose of evaluating the thermal mixing performance. In this study, implications concerning spacer grid thermal mixing performance based on turbulent flow experimental data were investigated. Heat transfer correlations derived by the analogy between momentum and heat transfer were applied to the 6 x 6 mixed spacer grid by which hydraulic tests were already performed[1]. Quantitative values of the local heat transfer in axial direction were obtained for different spacer grids.

2. Hydraulic tests

The tested 6 x 6 spacer grid is shown in Fig. 1. 6 x 6 spacer grid is a combined type formed by two different 3 x 6 spacer grids with different configurations and resistance. One of them at the lefthand side is a spacer grid with mixing vanes, which are not clearly seen in Fig. 1. The other one of the righthand side is a straight spacer grid. The cross-section of square housing of 81 mm in size consists of 36 rods of 9.5mm in diameter is shown with measuring locations and a coordinate system in Fig. 2. The swirling rotations due to mixing vanes are also illustrated in Fig. 2. Water is the working fluid and flows upward. The flow rate was 20 kg/sec at 25 ^{o}C resulting in an average flow velocity of 5m/sec and a Reynolds number based on the hydraulic diameter of Re=63924. The square housing is made of acrylic to allow access for laser beams to the location where the velocity measurement is to be performed. Velocity and turbulent intensity in axial and horizontal directions were measured by using LDV(laser Doppler velocimeter)[1].

3. Heat transfer characteristics

Thermal hydraulic behavior of the fluid at the spacer grid is hard to analyze in detail due to the complex geometry. In a possible way to approach this problem, the relation between heat and momentum transfer in turbulent flow can give information on the thermal behavior near spacer grid. Yao *et al.*[2] studied heat transfer augmentation by grid spacers with and without mixing vanes in rod bundles. The highest local heat transfer is observed near the spacer grid, and downstream of the spacer grid, the heat transfer augmentation rapidly decays with x/D_h exponentially. In the present study, some quantitative values of thermal mixing which significantly contributes to the heat transfer augmentation near spacer grid were evaluated as follows. Consider thermal energy balance in subchannels as schematically shown in Fig. 3 cited from Stewart *et al.*[3]. Energy conservation equation is given

$$A\frac{\partial}{\partial t}\langle\langle \rho h\rangle\rangle_v+\frac{\partial}{\partial x}\langle \rho u h\rangle_A+\{D^T{}_C\}\{\langle \rho v h\rangle_s s\}=$$

Qcv

$$\{D^T{}_r\}[P\phi H][D_r]\{T\}+\{D_W\}^T[LH][D_W]\{T\}$$

Qr Qw

$$+\frac{\partial}{\partial x}A\langle x\frac{\partial T}{\partial x}\rangle_A-\{D^T{}_C\}[\frac{SC\langle x\rangle}{L_C}][D_C]\{T\}$$

Qc

$$-\{D^T{}_C\}[m_{ij}'][D_C]\{x'\} \qquad (1)$$

Qt

where the designated terms Qcv, Qr, Qw, Qc, Qt represent Convection Heat Transfer to **V**, Heat Transfer to **V** from Rod, Heat Transfer to **V** from Wall, Conduction Heat Transfer to **V**, and Turbulent Heat Transfer to **V**, respectively. Most of detailed notations in each term are described in Ref. 3.

3.1 Heat transfer due to turbulent mixing

The lateral thermal energy exchanges at a gap are based on a fluctuating mass exchange between neighboring subchannels. The heat transport through the gap per unit length can be expressed as $q_{ij}=m_{ij}'C_p(T_i-T_j)$ where T_i, T_j and m_{ij}' are the bulk temperatures of two adjacent subchannels i and j, and the fluctuating cross-flow mixing rate per unit length between subchannels, respectively. m_{ij}' is defined as $m_{ij}'=\rho w_{eff}s$ where w_{eff} and s are the effective mean mixing velocity and rod gap spacing, respectively. If the enthalpies and velocities of the neighboring subchannels are different, an exchange of energy and momentum flux will occur.

The expressions for the net turbulent energy flux, F_h, and momentum flux, F_m, between subchannels can be given, respectively, by $F_h = \rho w_{eff} \Delta h$ and $F_m = F_t \rho w_{eff} \Delta U$, where Δh and ΔU are the average enthalpy and axial velocity difference between subchannels, and F_t is a correction factor which is related to the difference between turbulent energy and momentum transport. The turbulent cross-flow flux can be expressed as diffusive energy flux between centroids of subchannels, by introducing a mixing factor , can be given as $F_h = \varepsilon Y \rho \Delta h / \Delta y$, where ε , Y and Δy are , respectively, the reference turbulent eddy viscosity, a mixing factor and the centroid or mixing distance between subchannels. Ingesson and Hedberg[4] used the eddy viscosity at the center of a circular tube as a reference eddy viscosity, *i. e.*, $\varepsilon = \nu (f_c/8)^{0.5} Re/20$ where f_c and ν are the friction factor of a circular tube and a kinematic viscosity of a fluid, respectively. Combining the above correlations yields $Y = w_{eff} \Delta y / \varepsilon$. It is considered that all the different size eddies are contributing for turbulent cross-flow mass exchange. This consideration led to that effective mixing velocity, w_{eff} , equals to the azimuthal turbulent velocity , w'. However, since w_{eff} is not available in the present measurement, $w_{eff} = u'$ was assumed at the central region in the gap.

3.2 Heat transfer due to convective flow swirling

Following the analogy of heat and momentum transfer, convective heat transfer correlation in rod bundles with spacer grids is derived. The ratio of the pressure drops in rod bundles with and without the spacer grid is

$$\Delta P / \Delta P_o = (f + C_B D_h / L) / f = 1 + C_B D_h / (fL) \tag{2}$$

where f is the friction factor without spacer grids and C_B is the loss coefficients given by $C_B = C_v \beta^2$. Here β and C_v are a blockage ratio and a modified loss coefficient, respectively. The blockage ratio is the ratio of the projection area of spacer grid and mixing vanes to the flow area of the entire subchannel in axial direction. Marek and Rehme[5] suggested the heat transfer augmentation at straight spacer grids on smooth rods in a sigle phase flow by correlating pressure drop data and considering the analogy between heat and momentum transfer.

$$Nu / Nu_o = 1 + 5.55 \beta^2 \tag{3}$$

where Nu_o is the Nusselt number for rod bundles without spacer grids. Yao *et al.* extended the local Nusselt number correlation up to the downstream region behind the straight spacer grid without mixing vanes after suveying the related literatures.

$$(Nu / Nu_o)_{sp} = 1 + 5.55 \beta^2 e^{-0.13(x/D_h)} \tag{4}$$

for Reynolds number higher than 10^4. The spacer grid with mixing vanes generates swirling as shown in Fig. 2 , and local heat transfer behind the spacer is significantly depend on swirling. Regarding the spacer grid with mixing vanes, literatures for the local heat transfer near the spacer grid are more scarce. Thus tube flow with the swirling at the inlet was consiered to investigate the swirling effects on the local heat transfer. Swirl velocity is given as $V = UA\tan\phi$, where U is the axial component of the fluid velocity, A, a fraction of the projected area of the vanes to the flow cross-section when viewing from upstream, and ϕ, the angle of the vane with respect to the axial direction, respectively. Vector summation of each component velocity gives the magnitude of the velocty, i. e. $\tilde{v} = U(1+A^2\tan^2\phi)^{1/2}$. The decay of swirl flow in a tube has been studied by Kreith and Sonju[6]. They repesented the decay of swirl flow as $V = V_o\, e^{-0.17(x/D_h)}$ where V_0 is the swirl velocity at the tube inlet. The relationship of velocity and Nusselt number in Dittus-Boelter type equation for forced turbulent tube flow is introduced as $Nu \sim \tilde{v}^{\,0.8}$. Combining the above correlations, the heat transfer correlation for flow swirling becomes

$$(Nu/Nu_o)_{sw} = [\,1 + A^2\tan^2\phi e^{-0.034(x/D_h)}]^{\,0.4} \tag{5}$$

It is assumed that heat transfer correlation due to straight spacer grids is not coupled with that due to the flow swirling induced by mixing vanes. By multipling Eq. (4) and (5) , general correlation can be obtained.

$$Nu/Nu_o = [\,1 + 5.55\beta^2 e^{-0.13(x/D_h)}]\,[\,1 + A^2\tan^2\phi e^{-0.034(x/D_h)}]^{\,0.4} \tag{6}$$

for the downstream region in a single phase flow. This equation is valid for the vane angles less than 45 deg. In this work, swirl factor is defined as

$$Sw_{(x/D_h)} = \frac{1}{L}\int \frac{|V|}{U}\,dz \tag{7}$$

where L : length of integral path, V : lateral velocity at measurement location, and U : axial velocity at measurement location. The mgnitude of swirl factor is assumed to be a qualitative indicator of the spacer design on DNB(departure from nucleate boiling) performance[7]. Swirl factor was evaluated from the measured horizontal velocitiy data. By this swirl factor, Eq. (6) can be written as

$$Nu/Nu_o = [\,1 + 5.55\beta^2 e^{-0.13(x/D_h)}]\,[\,1 + Sw^2_{(x/D_h)}]^{\,0.4} \tag{8}$$

4. Results and discussions

In Fig. 4, local mixing factors, Y, at Paths 3~5 were plotted. Mixing factors show the highest value near the spacer grid and they decrease rapidly up to about x/D_h = 15. In Fig. 5, local Nusselt numbers predicted by Yao *et al.* for the blockage ratio of 0.328 and swirl factors of 0.57 and 1.0 are plotted. The heat transfer predictions for the present case were performed by using the experimental data[1]. In calculations at Paths 3 and 5, Eq. (8) was used instead of Eq. (6), since swirl factor was directly obtained from Eq. (7) using the experimental results. In 6 x 6 rod bunles, two different types of spacer grids are combined in different blockage ratios, 0.3(lefthand spacer grid with mixing vanes in Fig. 1) and 0.214(righthand straight spacer grid in Fig. 1). The predicted local Nusselt numbers for the present case are plotted and compared with those by Yao *et al.* in Fig. 5. It is concluded that the coolant thermal efficiency of the mixing vaned spacer grid is higher than that of the straight spacer grid in the present tested model.

References

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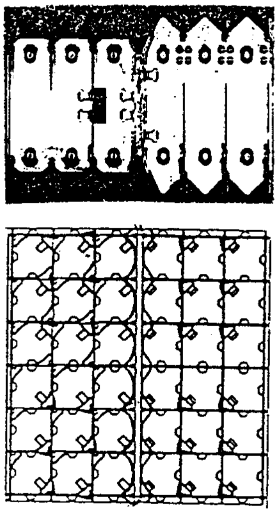

Fig. 1 6x6 Spacer Grid

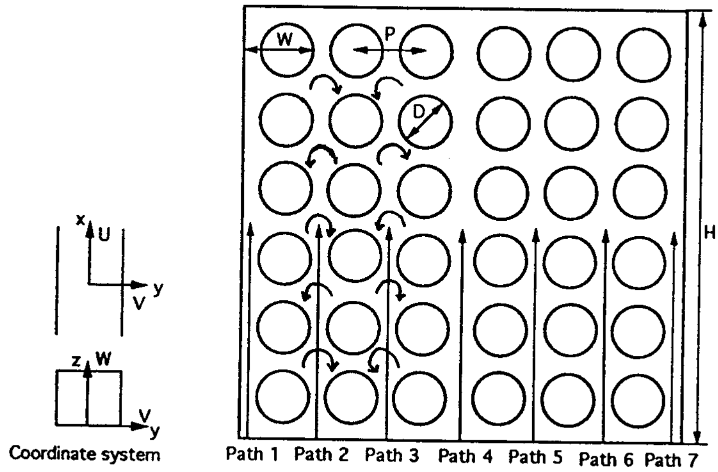

Fig. 2 Test Section showing Measuring Paths and Swirling Rotation

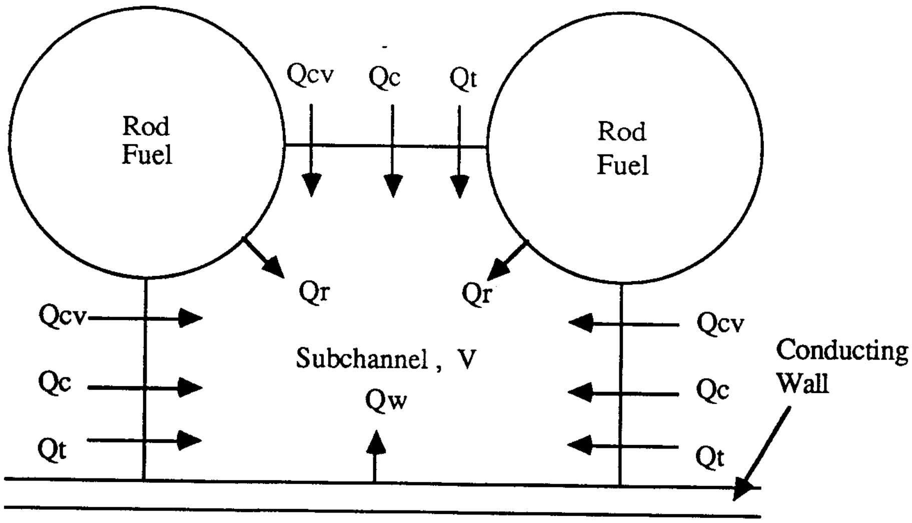

Fig. 3 Schematic Diagram of Energy Balance

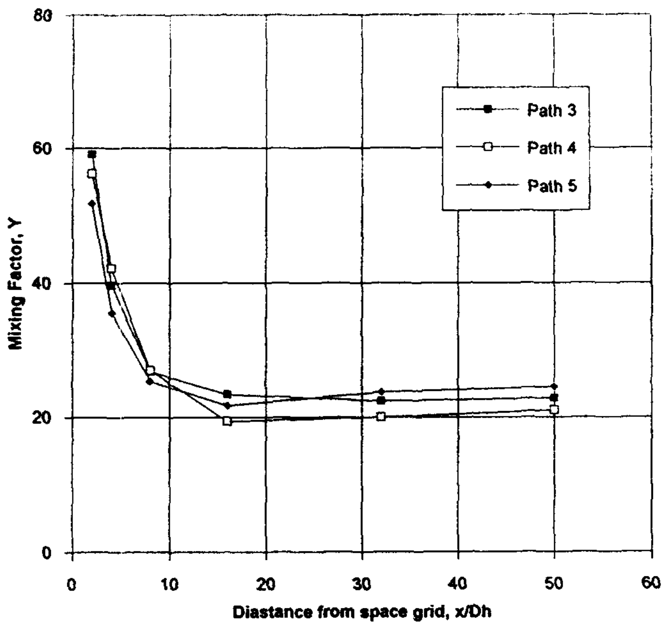

Fig. 4 Mixing Factors at Paths 3~5

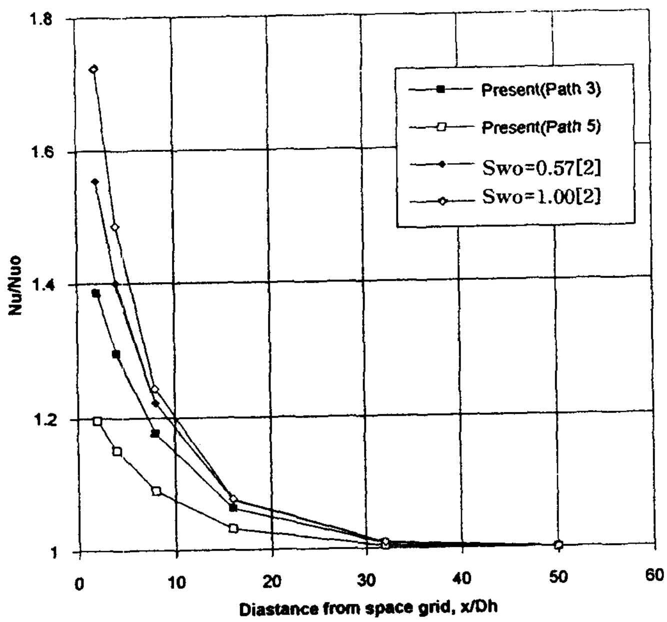

Fig. 5 Local Nusselt Numbers in Axial Locations