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NUMERICAL QUANTIFICATION OF CROSS-FLOW HEAT FLUX FOR TRU FUEL PIN BUNDLE

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ABSTRACT

TRU fuel has high decay heat when we fabricate a fuel bundle because it contains Minor Actinide (MA). Therefore, it is important to establish an effective and practical cooling technology during the fabrication process. In the present study, we investigate the thermal-hydraulics behavior, especially the properties of cross-flow heat flux in a fuel pin bundle. At first, we analyze thermal-hydraulics of a TRU fuel pin bundle without wrapping wire, i.e. a tube bank in the case of P/D=1.2. Secondary, we calculate a heat thermal-hydraulics of a fuel pin bundle in the case of P/D=1.1, which has not been studied enough unlike the case of P/D=1.2, to see the influence of P/D for thermal-hydraulics behavior. We found that thermal-hydraulics properties in the case of P/D=1.1 agree with the empirical correlations as in the case of P/D=1.2.

1. INTRODUCTION

A low decontaminated fuel in which a trans-uranium (TRU) is included is used for Fast Breeder Reactor (FBR) cycle system in Japan. The TRU fuel has high decay heat because it contains the minor actinide (MA). When one fabricates a TRU fuel pin bundle, we wrap a fuel pin with a thin wire to keep a distance between fuel pins and lay them transversely. Then, we make air flow into the gaps vertically across the pin bundle to remove the heat. Therefore, we have to establish an effective cooling technology to remove the heat in the fabrication process. For this purposes, we investigate the thermal-hydraulics behavior, especially the properties of cross-flow heat flux of a pin bundle. In this work, we investigate thermal-hydraulics in a TRU fuel pin bundle without wrapping wire, i.e. a tube bank for simplicity.

For a heat flux of a tube bank, we have two kinds of the empirical correlation for Nusselt number; one is according to Zukauskas (A. Zukauskas, 1972) and the other is Grimison's correlation (E. D. Grimison, 1937). Both of them are the equations between the Nusselt number and Reynolds number for a developed flow in a tube bank. We have to discuss which correlation is more appropriate for recent situation simultaneously.

As a calculation tool, we use the FLUENT ver. 6.3.26 which is commercial computational fluid dynamics code. In order to verify the applicability of FLUENT, we analyze a tube bank system using FLUENT, and then compare the results with the two correlations when the fuel pitch divided by the pin diameter (so called P/D) is 1.2. The thermal-hydraulics behavior of a tube bank when P/D=1.2 has been studied and its properties known to some extent. Therefore, it is appropriate for validation study. For sensitivity analysis, we see the dependence of the turbulence model, the near wall treatment and the mesh we used for the calculation, discussing the flow characteristics in the heated fuel pin bundle. After that, we analyze the thermal-hydraulics behavior of a fuel pin bundle in the case of P/D=1.1 to see the influence of P/D. It is very important for development of a cooling technology for TRU fuel pin bundle because a great amount of experimental data about the thermal-hydraulics properties has been accumulated unlike those in the case of P/D=1.2.

The contents of this paper are as follows. In section 2, we briefly formulate the method to obtain the Nusselt number from the FLUENT analysis and explain the empirical correlations for the present system. In section 3, results and discussions are presented. The final section is devoted to the conclusions of the present work.

2. Heat transfer calculation

We perform a numerical calculation for a tube bank system to investigate the thermal hydraulics properties by using the commercial CFD code FLUENT ver. 6.3.26. At first, we study the validity of FLUENT by analyzing a staggered tube bank of P/D=1.2. For this reason, we compare the calculational results with the empirical corelations. Then we analyze a staggered tube bank in the case of P/D=1.1 to see the dependence of P/D.

2.1 A staggered tube bank

We investigate a staggered tube bank system which corresponds to a fuel pin bundle without wrapping wire. The Simulation condition is shown in shown in Fig. 1 (P/D=1.2) and Fig. 2 (P/D=1.1). Both of the systems are consisted from two column and two rows of pins. Each tube has double structure which mimics a cladding tube. We suppose that all inner tubes have the same heat flux and fluid flows into subchannel from the bottom side. Furthermore, we impose the periodic condition both on the top and bottom side as shown by the red line in Fig. 1 and Fig. 2. Now, we ignore the gravity for simplicity.

The height is 67.15 mm, the width 77.54 mm, the tube outer diameter 32.31 mm, the tube inner diameter is 27.89 mm for the simulation condition of P/D=1.2 (See Fig. 1). The size is different from an actual TRU pin. We use larger size to make convergence of the calculation good. However, the ratio of inner and outer diameter is the same as actual fuel pin. On the other hand, the height is 61.56 mm, the width 71.08 mm, the tube outer diameter 32.31 mm, the tube inner diameter is 27.89 mm in the case of P/D=1.1 (See Fig. 2) .

We use an unstructured triangular cell, and total number of computational cell is approximately 600,000 for the P/D=1.2 system and 1,700,000 for P/D=1.1. The number of the cell of the system of P/D=1.1 is determined so that the number of the cell between the minimum gaps, which is shown in the blue lines of Fig. 1 and Fig. 2, is the same as that of P/D=1.2. This is because we consider that the velocity distribution between the minimum gaps plays important role for the recent study. A part of the mesh arrangement of P/D=1.2 is shown in the Fig. 1.We will discuss the dependence of the number of the mesh later.

We suppose that the all tubes consist from the SUS316 and fluid flown in the subchannel is air. The thermophysical properties of the each material are shown in the table 1.

We have two kinds of analytical parameters. First is the turbulent model. We use standard k-ε and RNG k-ε model. Secondary, we use three kinds of near-wall treatment, which will be explained in the next subsection in detail. We analyze at some the Reynolds numbers. In this work, the Reynolds number Re is defined by

$$Re = u_{max} D / \nu \quad . \tag{1}$$

Here D is a tube diameter, u_{max} a maximum velocity between gaps, which is the mean velocity in the minimum gap which is shown by the blue line in Fig. 1 and Fig. 2, and v is given by

$$\nu = \frac{\eta}{\rho}, \tag{2}$$

where h is viscosity and r is density.

Table 1 Thermophysical properties of matter.

matter	Density (kg/m^3)	Specific heat (J/kg/K)	Thermal conductivity (W/m/K)	Viscosity (kg/m/s)
SUS316	7970	450	13.5	
Air	1.225	1006.43	0.0242	1.7894e-05

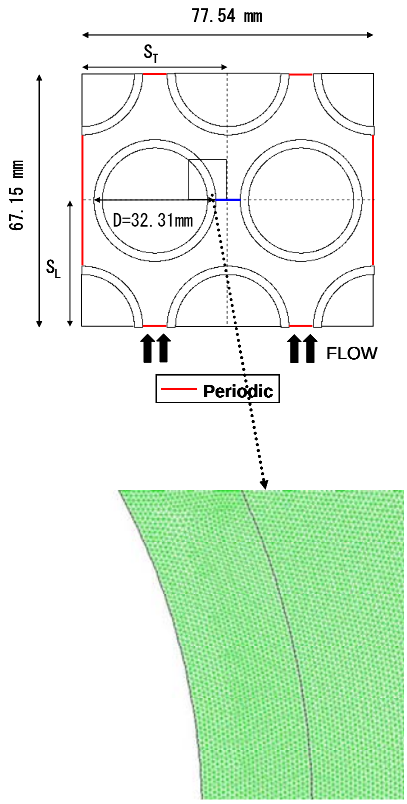


Fig. 1 A staggered tube bank and mesh (P/D=1.2).

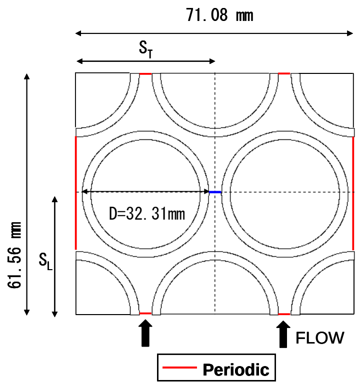


Fig. 2 A staggered tube bank (P/D=1.1).

2.2 Near-Wall Treatment

Turbulent flows are significantly affected by the presence of walls. Thus, in order to treat velocities and temperatures near the walls, we usually use a special function which is developed through numerous experiments. FLUENT provides three kinds of wall function approach, a standard wall function, a non-equilibrium wall function and an enhanced wall treatment (FLUENT ver. 6.3.26 User's Guid). In the present work, it will be found that the calculational results are very sensitive to the near-wall treatment we adopt. Therefore, we explain the formulations of those three methods here in this subsection.

2.2.1 Standard Wall Function

A flow velocity is given by

$$U^* = y^* \qquad (y^* < 11.225) \tag{3}$$

and

$$U^* = \frac{1}{\kappa}\ln(Ey^*) \quad (y^* > 11.225) . \tag{4}$$

Here U^* and y^* is defined by

$$U^* \equiv \frac{U_\mathrm{P} C_\mu{}^{1/4} k_\mathrm{P}{}^{1/2}}{\tau_\omega / \rho} \tag{5}$$

and

$$y^* \equiv \frac{\rho C_\mu{}^{1/4} k_\mathrm{P}{}^{1/2} y_\mathrm{P}}{\mu} , \tag{6}$$

respectively. The variables are defined as follows.

$\kappa = 0.42$: Karman constant
$E = 9.793$: empirical constant
$C_\mu = 0.09$
U_P : mean velocity of the fluid at point P
k_P : turbulence kinetic energy at point P
y_P : distance from point P to the wall
μ : dynamic viscosity of the fluid
ρ : density of fulid
τ_ω : wall shear stress

Similarly, the temperature is given by

$$T^* = \Pr y^* + \frac{1}{2}\rho \Pr \frac{C_\mu{}^{1/4} k_\mathrm{P}{}^{1/2}}{\dot{q}} U_\mathrm{P}^2 (y^* < y_\mathrm{T}^*) , \tag{7}$$

$$T^* = \Pr_\mathrm{t}\left[\frac{1}{k}\ln(Ey^*) + P\right] + \frac{1}{2}\rho \frac{C_\mu{}^{1/4} k_\mathrm{P}{}^{1/2}}{\dot{q}} \left\{ \Pr_\mathrm{t} U_\mathrm{P}^2 + (\Pr - \Pr_\mathrm{t}) U_\mathrm{c}^2 \right\} \cdot \quad (y^* > y_\mathrm{T}^*) \tag{8}$$

Here T^* is define by

$$T^* \equiv \frac{(T_\mathrm{W} - T_\mathrm{P}) \rho C_\mathrm{P} C_\mu{}^{1/4} k_\mathrm{P}{}^{1/2}}{\dot{q}} \tag{9}$$

and P is computed using the formulation given by:

$$P = 9.24\left[\left(\frac{\Pr}{\Pr_\mathrm{t}}\right)^{3/4} - 1\right]\left[1 + 0.28 e^{-0.007\Pr/\Pr_\mathrm{t}}\right] . \tag{10}$$

The variables are defined by the followings.

C_p : specific heat of fulid
$\dot{q}$: wall heat flux
$\mathrm{T_P}$: temperature at the cell adjacent to wall
$\mathrm{T_w}$: temperature at the wall
Pr : molecular Prandtl number
$\Pr_\mathrm{t}$: turbulent Prandtl number (0.85 at the wall)
$A = 26$: Van Driest constant
U_c : mean velocity magnitude at $y^* = y^*_\mathrm{T}$

2.2.2 Non-equilibrium Wall Functions

A flow velocity is given by:

$$\frac{\tilde{U} C_\mu^{1/4} k^{1/2}}{\tau_\mathrm{w} / \rho} = \frac{1}{k}\ln\left(E\frac{\rho C_\mu^{1/4} k^{1/2} y}{\mu}\right) . \tag{11}$$

Here

$$\tilde{U} = U - \frac{1}{2}\frac{dp}{dx}\left[\frac{y_\mathrm{v}}{\rho k \sqrt{k}}\ln\left(\frac{y}{y_\mathrm{v}}\right) + \frac{y - y_\mathrm{v}}{\rho k\sqrt{k}} + \frac{y_\mathrm{v}^2}{\mu}\right] \tag{12}$$

where $\frac{dp}{dx}$ is pressure gradient and y_v is the physical viscous sublayer thickness and is computed from

$$y_\mathrm{v} \equiv \frac{\mu y_\mathrm{v}^*}{\rho C_\mu^{1/4} k_\mathrm{P}^{1/2}} \tag{13}$$

where $y^*_\mathrm{v} = 11.225$.

As for temperature, the formulation is the same as a standard wall function.

2.2.3 Enhanced Wall Treatment (EWT)

The velocity is given by

$$u^+ = e^\Gamma u_\mathrm{lam}^+ + e^{\frac{1}{\Gamma}} u_\mathrm{turb}^+ . \tag{14}$$

Here G is given by

$$\Gamma = -\frac{a(y^+)^4}{1 + by^+} , \tag{15}$$

where

$$a = 0.01, \quad b = 5, \tag{16}$$

$$y^+ = \rho\frac{u_\tau y}{\mu} , \tag{17}$$

$$u^+ = \frac{U_\mathrm{P}}{u_\tau} , \tag{18}$$

and

$$u_\tau = \sqrt{\frac{\tau_\mathrm{w}}{\rho}} . \tag{19}$$

u^+_turb is obtained from the derivation equation,

$$\frac{du_\mathrm{turb}^+}{dy^+} = \frac{1}{ky^+}\left[S'\left(1 - \beta u^+ - \gamma\left(u^+\right)^2\right)\right]^{1/2} \tag{20}$$

where

$$S' = 1 + \alpha y^+ \quad (y^+ < y_\mathrm{s}^+), \quad 1 + \alpha y_\mathrm{s}^+ \quad (y^+ \ge y_\mathrm{s}^+) . \tag{21}$$

And

$$\alpha \equiv \frac{\mu}{\rho^2\left(u^*\right)^3}\frac{dp}{dx}, \quad \beta \equiv \frac{\sigma_\mathrm{t} q_\mathrm{w}}{\rho C_\mathrm{p} u^* T_\mathrm{w}}, \quad \gamma \equiv \frac{\sigma_\mathrm{t}\left(u^*\right)^2}{2 C_\mathrm{p} T_\mathrm{w}} . \tag{22}$$

On the other hand, u^+_lam is obtained by

$$\frac{du_\mathrm{lam}^+}{dy^+} = 1 + \alpha y^+ . \tag{23}$$

Similarly, temperature is given by

$$T^+ = e^\Gamma T_\mathrm{lam}^+ + e^{\frac{1}{\Gamma}} T_\mathrm{turb}^+ \tag{24}$$

where

$$\Gamma = -\frac{a(\Pr y^+)^4}{1 + b\Pr^3 y^+} . \tag{25}$$

Enhanced wall treatment is different from the others in the sense that enhanced wall treatment is continuous function and the others are discontinuous.

2.3 Numerically Derivation method for Nusselt number

In this subsection, we will explain how to obtain the Nusselt number from the analysis using FLUENT. We can obtain T_{w}, T_{inflow} and T_{outflow} using FLUENT computation. Here, we define T_{w} as averaged temperature on the surface of the tube. And T_{inflow} is a temperature of fluid before flowing into a tube bank, T_{outflow} a temperature of fluid after flowing out of a tube bank. Here in this analysis, we use an averaged values defined by

$$T_{\mathrm{inflow}} = \frac{\int_{\mathrm{inflow}} GT}{\int_{\mathrm{inflow}} G}, \tag{26}$$

$$T_{\mathrm{outflow}} = \frac{\int_{\mathrm{outflow}} GT}{\int_{\mathrm{outflow}} G}, \tag{27}$$

respectively. Here G expresses mass flow rate and T temperature. The integral are performed along the lines which are expressed by the red line for inflow and blue line for outflow in Fig. 3, respectively. In addition, we define T_{m} as

$$T_{\mathrm{m}} = \frac{T_{\mathrm{inflow}} + T_{\mathrm{outflow}}}{2}. \tag{28}$$

Now, we can obtain the Nusselt number Nu_{m} of the present system by using

$$Nu_{\mathrm{m}} = \frac{h_{\mathrm{m}} D}{k} = \frac{\phi D}{(T_{\mathrm{w}} - T_{\mathrm{m}})k}. \tag{29}$$

Here h_{m} is an average heat transfer coefficient, D a tube diameter, k a heat transfer coefficient of fluid, ϕ a heat flux.

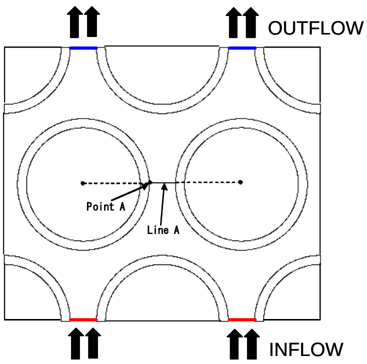


Fig. 3 Integral region and minimum gap.

2.4 Correlations

2.3.1 Zukauskas's Correlation

Zukauskas derived the empirical correlations by fitting numerous experimental data for various fluids (A. Zukauskas, 1972). In each Reynolds number region, Nusselt number is given by

$$\frac{Nu_m}{Pr^{0.36}\left(Pr/Pr_w\right)^{0.25}} = \begin{cases} 0.90\,Re^{0.4}; \quad 10 < Re < 10^2 \\ 0.51\,Re^{0.5}; \quad 10^2 < Re < 10^3 \\ 0.35\left(S_T/S_L\right)^{0.2} Re^{0.60}; \\ 10^3 < Re < 2\times10^5,\ S_T/S_L < 2 \\ 0.40\,Re^{0.60}; \\ 10^3 < Re < 2\times10^5,\ S_T/S_L > 2 \\ 0.022\,Re^{0.84}; \quad 2\times10^5 < Re < 2\times10^6. \end{cases} \tag{30}$$

Here S_{T} and S_{L} is a distance between pins in the horizontal and vertical direction, respectively as shown in the Fig. 1 and Fig. 2. Pr and Pr_w are Prantdl numbers. Pr is calculated based on the an arithmetic mean value of temperatures of fluid flowing into and flowing out of a tube bank. Pr_w is the value based on the temperature on the wall of a tube. In the present analysis, $Pr = 0.74$ and $Pr / Pr_w = 1$.

2.3.2 Grimison's Correlation

Grimison obtained the empirical correlations by fitting experimental data which fluid is air unlike Zukauskas (E. D. Grimison, 1937). In that sense, Grimison's equation may be more appropriate for the recent study. Grimison consider the influence of S_{T}/D and S_L/D more detail. They give the empirical correlation as follows.

$$Nu_m = C\,Re^n, \qquad 2000 < Re < 40000 \tag{31}$$

where C and n is given in the table. 2. In the present analysis, we use $C = 0.518$, $n = 0.556$. These values are not exact value for recent system where $S_T/D = 1.2$ and $S_L/D = 1.0$. Nevertheless we adopt the coefficients where $S_T/D = 1.25$ and $S_L/D = 1.25$ as it is relatively near to the recent case.

Table 2 Grimison coefficient.

S_T/D	1.25		1.5		2.0		3.0	
	C	n	C	n	C	n	C	n
0.600							0.213	0.636
0.900					0.446	0.571	0.401	0.581
1.000			0.497	0.558				
1.125					0.478	0.565	0.518	0.560
1.250	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.500	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2.000	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.000	0.310	0.592	0.356	0.580	0.440	0.562	0.421	0.574

2.5 Numerical Analysis condition

We suppose that each heat flux has 400 W/m^2 which corresponds to the actual TRU fuel pin, and bulk temperature of fluid is 300 K. In addition, a convergence condition for the energy is set to 10^{-10} and for others 10^{-6}. Pressure-velocity coupling is SIMPLEC.

3. Results and Discussions

3.1 Results and Discussions in the case of P/D=1.2

We have calculated the velocity and temperature distributions of a staggered tube bank when *P/D*=1.2. First, we investigate the energy conservation to ensure the validity of the present analysis. Then, we show the result for the velocity and temperature distributions, and obtain the Nusselt number using the method explained in the previous section.

3.1.1 Global balance

If the energy of the present system conserves, the following equations are kept.

$$A_{in}\phi_{in} = A_{out}\phi_{out} = C_p G(T_{outflow} - T_{inflow}) \qquad (32)$$

where A_{in} is a surface area of a inner tube, A_{out} a surface area of a outer tube, ϕ_{in} heat flux of a inner tube, ϕ_{in} heat flux of a outer tube, G mass flow rate and C_p specific heat.

When we set $A_{in}\phi_{in}$ as a standard, we define the quantity Δ_1, Δ_2 to see how the energy conservation is kept as follows.

$$\Delta_1 = \left| \frac{(A_{out}\phi_{out} - A_{in}\phi_{in})}{A_{in}\phi_{in}} \right| \times 100 \qquad (33)$$

$$\Delta_2 = \left| \frac{C_p G(T_{outflow} - T_{inflow}) - A_{in}\phi_{in}}{A_{in}\phi_{in}} \right| \times 100 \qquad (34)$$

Table 3 show the results when we calculated by various near-wall treatment such as EWT, standard wall function and non-equilibrium wall function, and turbulence models such as standard k-ε and RNG k-ε model in the case of *Re*=8,000. These result shows that the energy conservation is almost kept. It is sufficient to see the gross properties of recent system. As for all the other case, the energy conservation is also kept although we do not show them here.

Table 3. The degree of energy conservation

Near–wall treatment	turbulence model	Δ_1 (%)	Δ_2 (%)
EWT	k-□	1.31E-4	7.08E-2
EWT	RNG	1.11E-3	6.35E-2
Standard	k-□	6.35E-5	1.01E-1
Standard	RNG	3.18E-2	8.34E-2
Non-equilibrium	k-□	1.85E-3	1.43E-1
Non-equilibrium	RNG	2.25E-3	1.36E-1

3.1.1 Velocity and temperature distributions

Fig. 4 shows the average temperature of the fluid defined by Eq. (28) as a function of Reynolds number. The blue line represents the surface temperature calculated in a enhanced wall treatment and standard k-ε turbulence model, the pink line in a enhanced wall treatment and RNG k-ε turbulence model, the yellow line in a standard wall function and standard k-ε turbulence model, the light blue line in a standard wall function and RNG k-ε turbulence model, the purple line in a non-equilibrium wall function and standard k-ε turbulence model and the brown line in a non-equilibrium wall function and RNG k-ε turbulence model. All lines start from about 303 K and decrease as Reynolds number increases. These results show that the temperature of fluid depends on neither the near wall treatment nor turbulence model.

Fig. 5 shows the surface temperature of tube. The definitions of all lines are the same as in Fig. 4. Fig. 5 shows that the surface temperature depends on the near wall treatment strongly. Each lines starts from different temperature and decrease as Reynolds number increases. The blue and pink lines which calculated by EWT remains maximum value in all Reynolds number region among the three near wall treatment results whichever Reynolds number changes, and the lines calculated by standard wall function remains secondary large value, and the lines by non-equilibrium function minimum. On the other hand, the turbulence model affect to the results less than near wall treatment.

In order to understand the reason the surface temperature depend on the near wall treatment, let us discuss the absolute value of velocity and temperature distribution of the present system. Fig. 6 and Fig. 7 represents the absolute value of velocity and temperature distribution for each cases when *Re*=8,000. In both figures, for the expression A/B, A means near-wall treatment we use and B turbulence model. EWT, St and Non-eq express enhanced wall treatment, Standard wall function and Non-equilibrium wall function, respectively. On the other hand, k-ε and RNG express standard k-ε and RNG k-ε model, respectively. We can see that the absolute value of velocity and temperature distribution is different in all cases. To make quantitative discussion, we show Fig.8 which represents the absolute value of velocity and temperature distribution along line A (See Fig. 3) as a function of a distance from the surface of the tube when *Re*=8,000. In Fig. 8, EWT, St and Non-eq express enhanced wall treatment, Standard wall function and Non-equilibrium wall function, respectively. In all calculations, we use standard k-ε model. We can immediately see that the velocity and temperature distribution is totally different in each near-wall treatment. As for the absolute value of velocity, all lines are symmetric about 0.003 m, which ensure the accuracy of recent calculation. The black line, which is calculated in EWT, starts from 0 m/s, raising at first, then turns to decrease at 0.001. In contrast, both the red and blue lines, which are calculated in standard and non-equilibrium wall function respectively, increase as the distance increase and take the maximum value at the center of the line A although the rate of increase is different in each case. As for the temperature, all lines decrease as the distance increase and become minimum at about 0.003 m. The black line starts from about 307.5 K, the blue line 305 K and the red line 303 K. Namely, each line starts from different temperature, meanwhile, all lines take almost the same minimum value. The absolute value of the velocity at the first calculation point which is nearest to the tube surface along line A are 0.24 m/s for the black line, 0.26 m/s for the blue line and 0.34 m/s for red line, respectively. It means that in case of EWT, the air flows the slowest at the point nearest the tube surface. For this reason, the temperature in the case of EWT takes the highest value between three cases because a slow flow takes less heat from a tube.

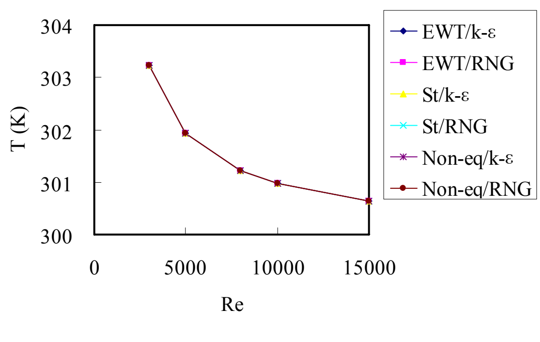


Fig. 4 The average temperature of fluid T_m (P/D=1.2).

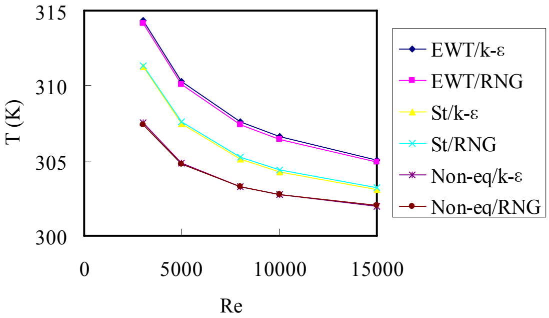


Fig. 5 The temperature of tube surface T_w (P/D=1.2).

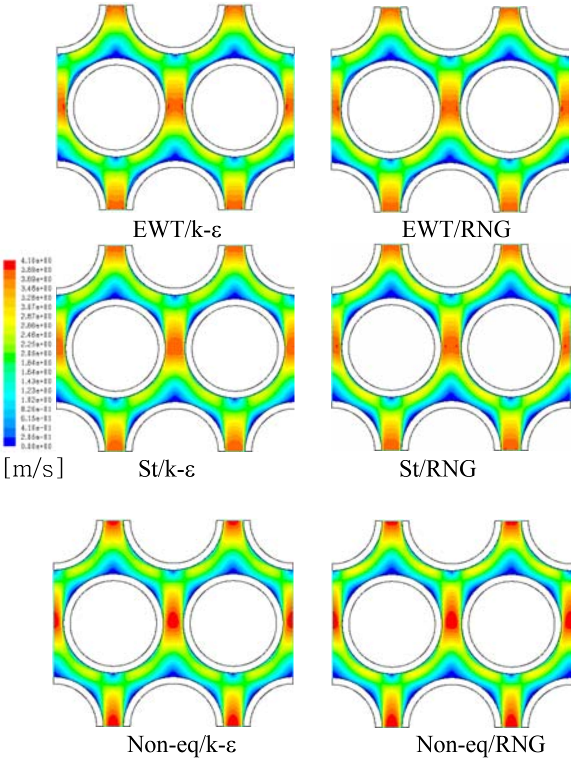
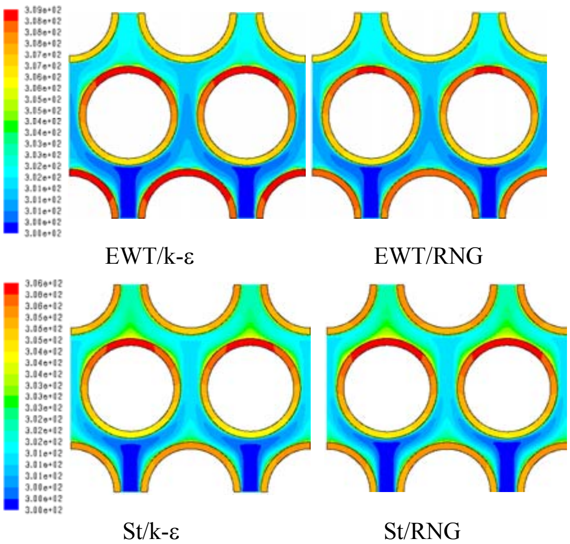


Fig. 6 The distribution of magnitude of the velocity.



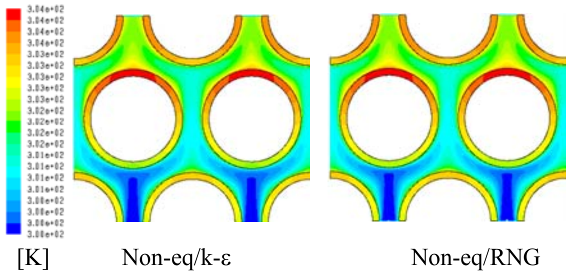


Fig. 7 The distribution of temperature.

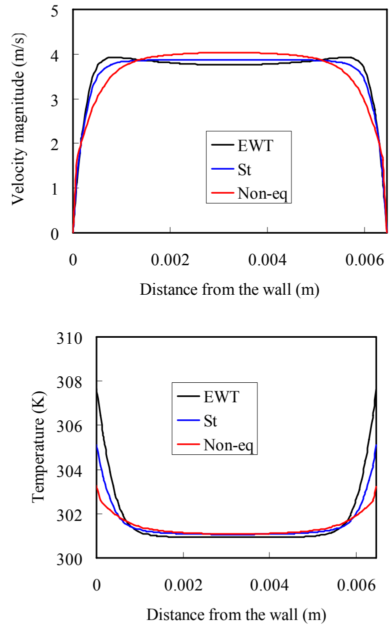


Fig. 8 The magnitude of the velocity and temperature along the Line A (See Fig. 3).

3.1.2 Nusselt number

Fig. 9 shows the Nusselt number as a function of Reynolds number calculated in various near wall treatments and turbulence models when *Re*=3,000, 5,000, 8,000, 10,000 and 15,000, and Zukauskas and Grimison empirical correlations for comparison. The black points, which calculated in EWT and standard k-ε model, increase as Reynolds number along the Zukauskas's correlation extremely nearly. Similarly, the red points which calculated in EWT and RNG k-ε model increase as Reynolds number along the Grimison's correlations very nearly. On the other hand, in all the other cases, which calculated in standard wall and non-equilibrium function, and standard and RNG k-ε models, Nusselt numbers take larger values, especially the results calculated in non-equilibrium wall function take about the two time larger than those in EWT. By A/B, A means near-wall treatment we use and B turbulent model like Fig. 4 and Fig. 5. As shown in Eq. (29), Nusselt number depends on $T_{\rm m}$ and $T_{\rm w}$. And we already know the fact that $T_{\rm w}$ depends on near wall treatment and not so much on turbulence model while $T_{\rm m}$ take the same value whichever near wall treatment and turbulence model we adopt. Therefore, Nusselt number depends on T_w , namely, near wall treatment. Thus, these results shown in Fig. 9 are understandable.

These results show that the calculation using FLUENT agrees with the empirical correlations if we use EWT as a near-wall treatment. Thereby, we could have verified that the FLUENT code can be used for the thermal-hydraulics analysis of an infinite staggered tube bank in the case of P/D=1.2.

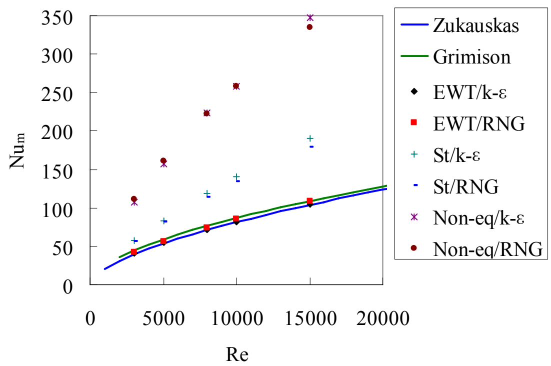


Fig. 9 The Nusselt number (P/D=1.2).

3.1.3 Mesh dependence

In order to see the convergence of the recent analysis, we calculate Nusselt number when the total mesh number is about 150,000, 600,000 and 2,500,000. All cases are calculated in EWT and standard k-ε model in the case that Re is 3,000. Fig. 10 shows the results. The Nusselt number decreases as total mesh number increases. We should note that the rate of the decrease is gradually small. It means that the Nusselt number approaches to the constant value. Here in this study, we choose the mesh whose total number is about 600,000 because we consider the time costs.

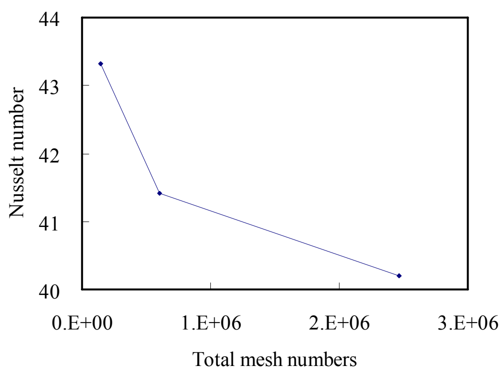


Fig. 10 The total number of dependence for Nusselt number when Re=3,000 calculated in EWT and standard k- model.

3.2 Results and Discussions in the case of P/D=1.1

In this subsection, we apply the FLUENT code to an infinite tube bank in the case of P/D=1.1, which corresponds to an actual TRU fuel pin bundle.

3.2.1 Nusselt number

We calculate the Nusselt number in the same way as the subsection 3.1 in the case of P/D=1.1. We set mesh division so that the number of the mesh division between minimum gap is the same as in the case of P/D=1.2 because the properties of the thermal-hydraulics behavior there is considered to be important for this analysis. Fig. 11 show the resultant Nusselt numbers calculated in EWT, and standard k-ε and RNG k-ε turbulent models when Re=5,000, 8,000, 10,000 and 15,000, and Zukauskas's and Grimison's correlations for comparison. All definition of points and lines are same as Fig. 9. These two correlating equations are exactly the same as those in the case of P/D=1.2 in the present system. As in the case of P/D=1.2, the results are located near the correlating equations, especially Grimison's one, although those are located in the upper ward compared with those of P/D=1.2. Quantitatively, these results are about 1.05~1.10 times larger than those of P/D=1.2. We would like to discuss this fact in detail.

We define the different Reynolds number which the characteristic length is the minimum gap as to understand this fact. Then, the Reynolds number becomes smaller if the characteristic speed remains same. In consequence, Nusselt number becomes larger.

Nevertheless, we conclude that Fig. 11 the recent analysis agree with the correlating equations although they are not exactly same. As the correlating equations are developed by fitting the experimental data in logarithmic scale, and therefore, the differences such as shown in Fig. 11 make little sense.

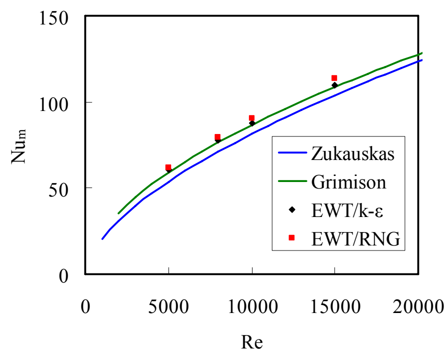


Fig. 11 The Nusselt number (P/D=1.1).

4. Conclusion

We have studied the thermal-hydraulics behavior of an infinite staggered tube bank in the case of P/D=1.1 and 1.2. First, the validation of FLUET has been confirmed by investigating the thermal-hydraulics properties of a tube bank of P/D=1.2. As a result, we have found that the Nusselt numbers derived based on the numerical simulation agree with the empirical correlation in linear scale when we adopt EWT as near wall treatment although empirical correlation has been made in logarithmic scale. Therefore we have concluded that the validation of FLUENT has been ensured. Also, results are very sensitive to the near wall treatment. Second, we have studied a tube bank in the case of P/D=1.1. And we have found that the Nusselt numbers agree with the correlations in linear scale as in the case of P/D=1.2. This fact shows that empirical correlations are available for a tube bank system of P/D=1.1 although those correlations are not developed using the data of dense tube bank system such as P/D=1.1.

In this work, we have investigated a tube bank which corresponds to a TRU fuel pin bundle without wrapping wire. It is idealized system because the purpose of recent system of to see the thermal-hydraulics behavior of dense tube bank. Therefore, we have to analyze a TRU fuel pin bundle with wrapping wire and see that influence to develop further the cooling technology. In addition, it may be important to consider gravity for more practical calculation. Those works are now in progress.

ACKNOWLEDGEMENTS

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NOMENCLATURE

P	a fuel pitch	[m]
D	a tube diameter	[m]
Re	Reynolds number based on a tube diameter and a maximum gap velocity	
v	viscosity coefficient	[m^2/s]
η	viscosity	[kg/m/s]
ρ	density	[kg/m^3]
A_{in}	surface area of a inner tube	[m^2]
A_{out}	surface area of a outer tube	[m^2]
ϕ_{in}	heat flux of a inner tube	[W/m^2]
ϕ_{out}	heat flux of a outer tube	[W/m^2]
C_p	specific heat	[J/kg/K]
G	mass flow rate	[kg/s]

Subscripts

m	*mean value*
w	*the value at the wall*

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