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A PERFORMANCE CRITERION FOR HEAT EXCHANGERS

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ABSTRACT

A performance evaluation criterion for heat exchangers is derived for the purpose of the development of an enhanced heat transfer surface. The criterion is featured in giving heat exchange rate per unit volume as a function of geometrical parameters, fluid pumping power per unit volume and dimensionless operating variables such as Re, Nu and f (pressure drop coefficient). The performance of new plate fin heat exchanger surface is discussed with those of conventional surfaces under common pumping power condition. Using Nu and f correlations for supercritical carbon dioxide obtained from CFD, the performance is evaluated. Its heat exchange rate is 15%~50% larger than that of smooth circular tube in terms of heat exchange rate density.

INTRODUCTION

An enhanced heat transfer surface is in need for the development of engineering thermodynamic systems. One of most important applications is a regenerative heat exchanger (RHX) for advanced closed cycle gas turbines, in which extremely high temperature effectiveness typically over 95% is required in order to gain high cycle thermal efficiency. To meet the requirement, highly compact heat exchanger is required.

Webb discussed a performance evaluation criterion for exchangers as a ratio of heat conductance UA/U_rA_r and Hesselgreaves unified conventional theories in his textbook. Most of its applications, however, were limited to tube banks with turbulent promoters such as non-smooth surfaces or coiled wire packed with negligible volume (Garcia et al.) where hydraulic diameter (D_h) and nominal heat transfer area were assumed unvaried. In these, simple formula is obtained for comparison with the original reference surface.

To the contrary this paper tries to extend the criterion capable of comparing the performances of surfaces with different shapes where compactness β and D_h are different from a competing one. By introducing a new definition of D_h by Utamura, these comparison have been accomplished uniquely even for surfaces having complex geometries. Then, enhanced fin shapes to form flow channels were investigated in terms of newly developed performance criterion i.e. heat exchange rate under the same pumping power per unit exchanger volume. It was applied to plate fin surfaces compared with smooth tube banks.

NOMENCLATURE

A	[mm^2]	Heat transfer area
A_c	[mm^2]	Average free flow area along flow path
A_{cell}	[mm^2]	Projected area of a unit cell of flow channel
A_f	[mm^2]	Fluid inlet frontal area
Bi	[-]	Biot number(= $\alpha t/\lambda_w$)
C_p	[W/kgK]	Isobaric specific heat
D_h	[mm]	Hydraulic diameter
d	[mm]	Tube inner diameter
d_s	[mm]	Pitch of tube bank
f	[-]	Pressure loss coefficient(= $2\rho\Delta pD_h/L/G^2$)
G	[kg/m^{2}s]	Mass flux
H	[mm]	X channel width
H_d	[mm]	Channel depth or Fin height
H_p	[mm]	Plate thickness(=H_d+t)
L	[mm]	Length of heat exchanger
l_{ax}	[mm]	Longitudinal distance of a unit cell of flow channel
l_{tr}	[mm]	Transverse distance of a unit of flow channel
NTU	[-]	Number of transfer unit(=(T_o-T_i)/ ΔT)
Nu	[-]	Nusselt number(= $\alpha D_h/\lambda_f$)
P_0	[W]	Pumping power
P_0	[W/m^3]	Pumping power per unit volume
Δp	[Pa]	Pressure drop
Q	[W]	Heat exchange rate
q	[W/m^3]	Heat exchange rate per unit volume
Re	[-]	Reynolds number(=GD_h/μ)
St	[-]	Stanton number(= $\alpha/C_p/G=Nu/Re/Pr$)
ΔT	[K]	Average fluid temperature difference
t	[mm]	Thickness of wall
U	[W/m^{2}K]	Heat transmission coefficient
V	[mm^3]	Heat exchanger volume
V_F	[mm^3]	Fluid volume
v	[m^3/kg]	Specific volume
W	[m^3/s]	Volume flow rate
Greek		
α	[W/m^{2}K]	Heat transfer coefficient
β	[mm^{-1}]	Heat transfer area per unit volume of heat exchanger(=A/V)
ε	[-]	Two dimensional porosity(=V_F / V)
θ	[deg]	Wedge angle of X-channel fin
μ	[Pas]	Viscosity
λ	[W/mK]	Heat conductivity
ρ	[kg/m^3]	Fluid density
Subscript		
r		Reference heat transfer surface
f,F		Fluid
w		Wall
Superscript		
*		Solution

PERFORMANCE EVALUATION CRITERION

Basic equations

Thermal-hydraulic behavior of any heat exchanger is governed by the following equations:

$$Q = AU\Delta T = (T_o - T_i)C_p \rho W \tag{1}$$

$$\Delta p = f\frac{L}{D_h}\frac{\rho}{2}\left(\frac{W}{A_c}\right)^2 \tag{2}$$

The assumption that $Bi(\equiv \alpha t/\lambda_w)$ is infinitesimally small allows heat transmission coefficient U to be replaced by heat transfer coefficient α . Introduction of compactness $\beta \equiv A/V$ may rewrite Eq.(1) as

$$q \equiv Q/V = \beta C_p G St \Delta T \tag{3}$$

Substitution of a new definition of hydraulic diameter

$D_h = 4V_F / A$

, which is described in detail in later section and the relation $V_F = A_c L$ into Eq.(2) gives

$$p_0 \equiv P_0 / V = W\Delta p / V = f\beta G^3 / 8\rho^2 \tag{4}$$

or

$$Re \equiv \frac{GD_h}{\mu} = \frac{2D_h}{\mu}\left(\frac{\rho^2 p_0}{f\beta}\right)^{\frac{1}{3}} \tag{4'}$$

where *G* is mass flux. Elimination of *G* from Eqs.(3) and (4) yields

$$\frac{q}{\Delta T} = 2C_p \beta St\left(\frac{\rho^2 p_0}{f\beta}\right)^{\frac{1}{3}} = 2C_p \beta^{\frac{2}{3}} St\left(\frac{p_0}{f}\right)^{\frac{1}{3}}\rho^{\frac{2}{3}} \tag{5}$$

Eqs.(4)' and (5) are basic equations to characterize performance of heat exchangers.

Parameter relations under same operating conditions

Modification of Eq.(5) under the same operating conditions leads to the relation among parameter ratios

$$\frac{q}{q_r} = \left(\frac{\beta}{\beta_r}\right)^{\frac{2}{3}}\left(\frac{f}{f_r}\right)^{-\frac{1}{3}}\frac{St}{St_r}\left(\frac{p_0}{p_{0r}}\right)^{\frac{1}{3}} \tag{6}$$

where subscript r indicates competing surface.

We assume there are correlations $f = f(Re)$ and $Nu = Nu(Re, Pr)$. We compare heat duty under the same pumping power *i.e.* $p_0 / p_{0r} = 1$. Solving Eq.(4)' for Re gives

$$\frac{Re^*}{Re_r} = \frac{D_h}{D_{hr}}\left(\frac{f^*}{f_r}\right)^{-\frac{1}{3}}\left(\frac{\beta}{\beta_r}\right)^{-\frac{1}{3}} \tag{7}$$

where $Re^*, f^* = f(Re^*)$ are solutions to satisfy Eq.(7). Noting the relation $StRe = Nu/Pr$ and substituting Eq.(7) for Eq.(6), we have

$$\frac{q^*}{q_r} = \frac{\beta}{\beta_r}\frac{Nu^*}{Nu_r}\left(\frac{D_h}{D_{hr}}\right)^{-1} \tag{8}$$

$$\frac{G}{G_r} = \frac{Re^*}{Re_r}\left(\frac{D_h}{D_{hr}}\right)^{-1} \tag{9}$$

Using the relation $L = D_h NTU / 4St$ and noting *NTU* is an operating parameter because $NTU = (T_0 - T_i)/\Delta\overline{T}$, we have length ratio as

$$\frac{L}{L_r} = \frac{D_h}{D_{hr}}\left(\frac{St^*}{St_r}\right)^{-1} \tag{10}$$

Further, if fin is two dimensionally shaped i.e. the configuration of horizontal cross section does not vary along fin height and the thickness of wall to separate hot and cold fluids is infinitesimally small, then we can relate β with 2D porosity ε as

$$\beta = \frac{4\varepsilon}{D_h} \tag{11}$$

It should be noted $\varepsilon = A_c / A_f$.

Then, Eq.(6) can be rewritten in more explicit form of geometrical parameters as

$$\frac{q}{q_r} = \left(\frac{\varepsilon}{\varepsilon_r}\right)^{\frac{2}{3}}\left(\frac{D_h}{D_{hr}}\right)^{-1}\left(\frac{f}{f_r}\right)^{-\frac{1}{3}}\frac{St}{St_r}\left(\frac{p_0}{p_{0r}}\right)^{\frac{1}{3}} \tag{6'}$$

,in which 1st and 2nd terms of the right hand side shows the effect of geometry, 3rd and 4th terms thermal-hydraulic characteristics and 5th pumping power respectively.

Effect of wall thickness

As the wall to separate hot and cold fluids is often pressure boundary and its thickness should be finite to secure mechanical integrity. Thus, practically wall thickness has to be included as part of heat exchanger volume. For this purpose, β, ε should be substituted by β', ε' given by

$$\beta' = \beta/(1 + t/H_d), \quad \varepsilon' = \varepsilon/(1 + t/H_d) \tag{12}$$

,where t and H_d are wall thickness and channel depth or fin height respectively.

ENHANCED HEAT TRANSFER SURFACES

S-shaped fin surface

Micro-channel heat exchanger under consideration is illustrated in Fig.1. Tsuzuki et al. developed compact heat exchanger, named MCHE with S-shaped fins. The flow channel of MCHE is created by chemical etching on metal plates and the plates are gathered to make one heat exchanger with diffusion bonding. Typical hydraulic diameter is sub-millimeter. The small hydraulic diameter increases heat transfer area and gives better heat transfer performance. The present research attempts to seek an alternative flow channel configuration with simpler geometry suitable for mass production with more compact as well as better thermal hydraulic characteristics.

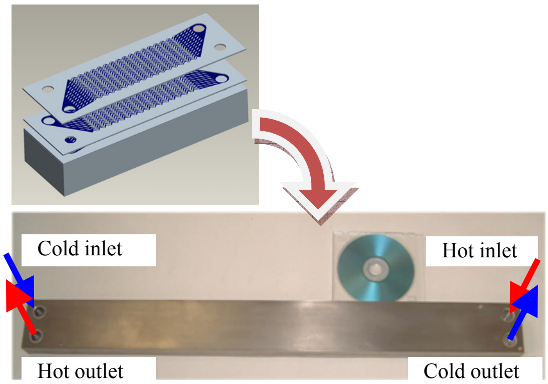
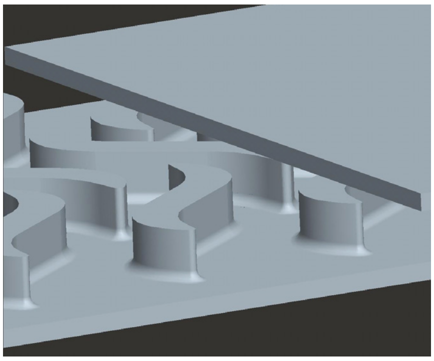


Fig. 1 Microchannel heat exchanger (MCHE).



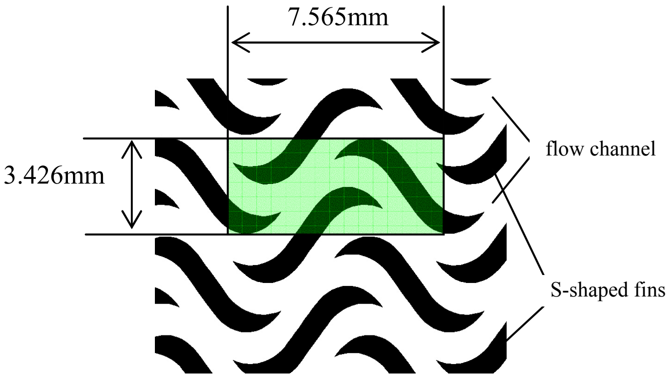


Fig. 2 S-shaped fin configuration.

To do this in a right way, we regard the flow channel formed by S shaped fin with unit cell shown in Fig.2 as a network of flow conduits rather than plate fin exchanger. As the network has bifurcations and confluences along flow path repetitively and the flow area varies along flow length, conventional definition of hydraulic diameter is hard to be applied. To overcome this difficulty, this study tries to generalize conventional definition of hydraulic diameter applicable to various channel geometries without arbitrariness.

Generalized hydraulic diameter

It is necessary to derive an expression for hydraulic diameter applicable to a network of conduit with variable free flow area along flow direction.

To this end, following assumptions and definitions were adopted.

1) Heat transfer area A is defined as the sum of surface area where fluid is attached (wetted surface). Therefore, the surface of fin side wall is counted to the heat transfer area. This is appropriate because the top and bottom parts of fins are attached to heat exchanger wall and thus flow channel is more likely to a network of conduits having separations and junctions
2) Average free flow area of heat exchanger A_c is defined as the free volume V_F divided by the nominal flow path length L, typically the axial length of the heat exchanger.
3) As the flow field is formed by the conduit network in which fin pattern is repetitive in both axial and transverse directions, then hydraulic diameter D_h is redefined by

$$D_h = \frac{4V_F}{A} . \qquad (13)$$

It is important to note that Eq.(2) can give a unique value for any flow channel configuration regardless of flow direction and thus there is no arbitrariness present. Obviously, it reduces to usual definition of D_h in the case that cross sectional area is unchanged along flow path. Therefore, Eq.(13) may be regarded as the generalization of the definition of conventional hydraulic diameter to allow variable geometry along the flow path.
In the case of the present XCHE, using the relation $A_c = V_F /L$ by assumption 2), Eq.(13) reduces to $4A_c L/ A$.

X-channel surface

Based on new definition of hydraulic diameter that is used as representative length of heat transfer surfaces, we will reveal thermal hydraulic characteristics and performance of what we call X-channel heat exchanger (XCHE) using commercial CFD code FLUENT.

The flow channel of XCHE can be formed in a simpler way by engraving trenches with same width and depth at an angle inclined to the longitudinal direction and also in the same way in an opposite angle. Several derivatives of XCHE were created and their performances were compared with smooth tube banks or S-shaped fin.

In order to find a new heat transfer surface easy to manufacture with no deterioration of thermo-hydraulic performance compared to S-shaped fin MCHE, X-channel configuration is investigated. Flow channel may be formed on a plate by engraving straight lines in an interval with an inclination angle to the longitudinal direction of the heat exchanger and in the same way also with an opposite angle. Resultant geometry is a staggered array of a group of diamond shaped fins and X shaped flow channels having branches along flow path as illustrated in Fig.3, in which unit cell is displayed.

Important geometrical parameters in a unit cell of a flow channel are expressed by use of three basic geometric parameter *H*, channel depth H_d and wedge angle θ which are shown in Fig.4

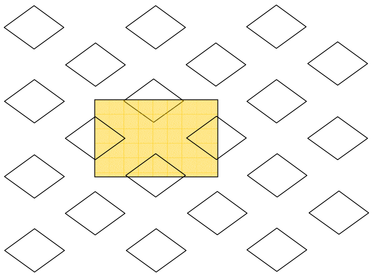


Fig.3 Plan view of X-channel surface

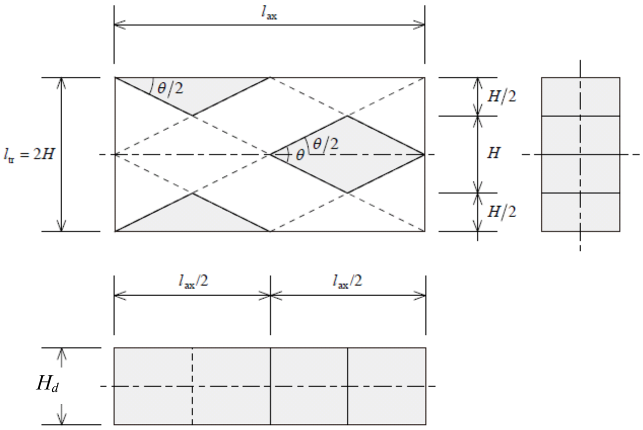


Fig.4 Geometry of unit cell of X-channel surface

$$A=\frac{2H^2(3\cos(\theta/2)+2(H_d/H)}{\sin(\theta/2)}$$

$$V_F=\frac{3H^2H_d}{\tan(\theta/2)}$$

$$D_h\equiv\frac{4V}{A}=\frac{6H_d\cos(\theta/2)}{3\cos(\theta/2)+2(H_d/H)}\qquad(14)$$

$$l_{ax}=\frac{2H}{\tan(\theta/2)}$$

$$A_{cell}\equiv l_{ax}l_{tr}=\frac{4H^2}{\tan(\theta/2)}$$

$$A_c\equiv\frac{V_F}{l_{ax}}=\frac{3}{2}HH_d\qquad(15)$$

$$\varepsilon\equiv\frac{V_F}{H_dA_{cell}}=0.75\qquad(16)$$

$$\beta=A/V=(3+\frac{2(H_d/H)}{\cos(\theta/2)})/2H_d\qquad(17)$$

Calculation results for D_h(Eq.(14)) and β (Eq.(17)) are shown as a function of angle θ. D_h value same as S-shaped fin is attained at the angle 58degC where β of X-fin is larger . than that of S-shaped one.

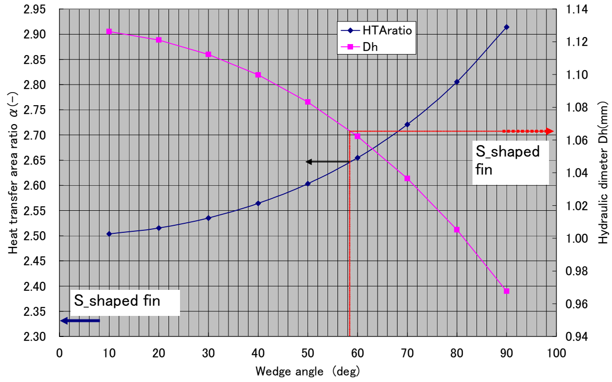


Fig.5 Hydraulic diameter and heat transfer area ratio of X-shaped channel as function of wedge angle θ

CFD SIMULATION

CFD was incorporated to investigate heat transfer coefficient and pressure drop coefficient of X channel. Fig.6 shows three dimensional numerical model. 16 rows of unit cell were prepared as computational domain to determine thermal characteristics of fully developed flow. Hot side carbon dioxide flow of a regenerative heat exchanger involved in supercritical CO_2 gas turbine that requires very high temperature effectiveness and thus compact heat exchanger is indispensable to put it into practice.

About boundary conditions, periodic condition was applied to the both side walls. In order to obtain heat transfer coefficient, temperatures on wall surface were fixed. The inlet fluid temperature was set 2K higher than the wall temperature. Spatially homogeneous velocity profile was applied to inlet and free discharge boundary condition at exit boundary of the computational domain.

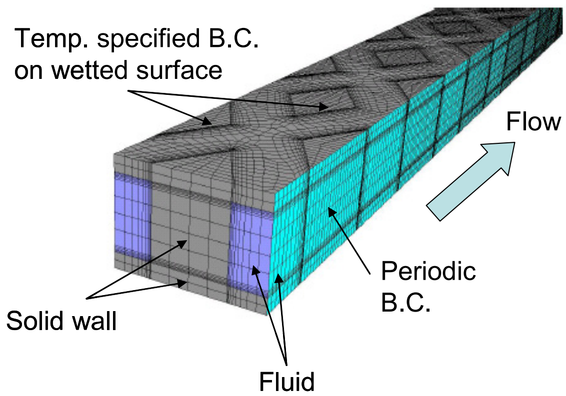


Fig.6 Computational mesh and boundary conditions

Simulation calculations were carried out using 3D-CFD code, FLUENT. (ANSYS Inc., 2007) Governing equations considered are the continuity, momentum and energy equations. Turbulence was taken into account by the ReNormalization Group (RNG) k-ε model and wall function. All equations were solved using second-order upwind discretization for convection.

The Semi Implicit Method Pressure Linked Equation (SIMPLE) algorithm was used to resolve coupling of velocity and pressure.Thermodynamic properties of CO_2 (density, viscosity, specific heat, and thermal conductivity) were calculated using PROPATH, a database for thermo-physical properties of fluids (PROPATH group, 1990). The fluid is assumed to be in local equilibrium for thermodynamic and transport properties. Free convection, viscous dissipation, and compressive work are neglected in this simulation. The property values of steel initially given in FLUENT are used as metal plate properties.

Mesh size, as generated by GAMBIT, was varied from the wall to the central flow region as shown in Fig.7. In the vicinity of wall surfaces, the minimum mesh size was 0.015 mm. Such fine mesh units were allocated into five rows from the fin wall surfaces. The mesh size was about 0.2 mm for regions that were distant from wall surfaces.

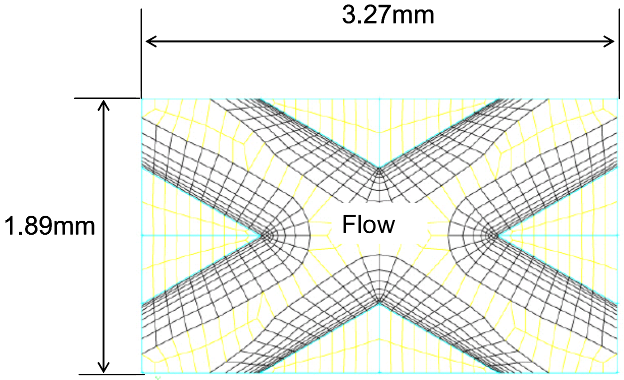


Fig.7 Computational mesh of unit cell of X channel

Fig.8 shows calculated velocity contour map in the case of X channel at Re=7000. Symmetry flow pattern is observed and flow develops very shortly from the entry point at left end of calculation domain.

Thermal hydraulic characteristics

The inlet temperature and pressure were 633.15 K and 8MPa. With flow rate varied heat exchange rate and pressure drop were obtained and heat transfer coefficient *Nu* as well as pressure loss coefficient *f* were derived in non-dimensional form that were defined by

$$\alpha = Q/(A\Delta T_{LMTD}) \tag{18}$$

$$Nu = \alpha D_h / \lambda_f$$

$$f = 2\rho D_h \Delta p/(LG^2) \tag{19}$$

X channel showed the large value of pressure drop coefficient f. To mitigate it, a surface with four corners of every fin rounded. Its calculation mesh and results are shown in Figs.9 and 10 in which the legend Xw0.944 denotes fin with sharp edge and fin height H_p =0.944mm and Xd those with rounded corners. The case of Pr =0.767 corresponding to thermodynamic state of P=8MPa,T=280degC of carbon dioxide are shown. The effect of rounded corner is significant. Shorter fins are seen to have better thermal hydraulic characteristics.

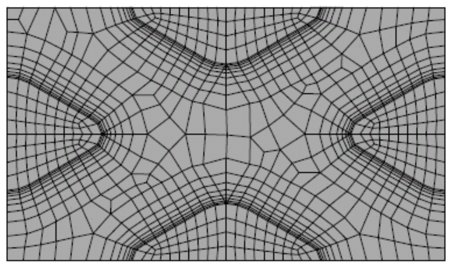


Fig.9 Calculation mesh of deformed X channel

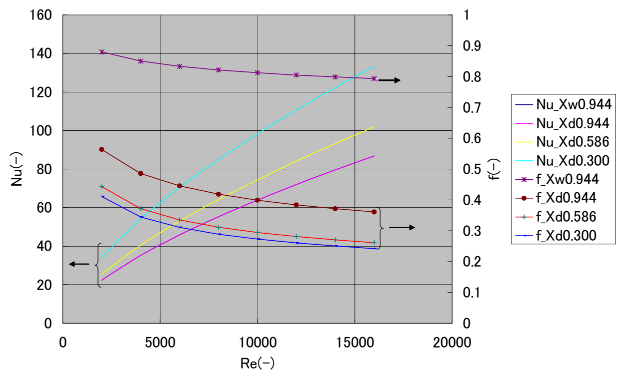


Fig.10 Thermal-hydraulic characteristics of X-channel fins

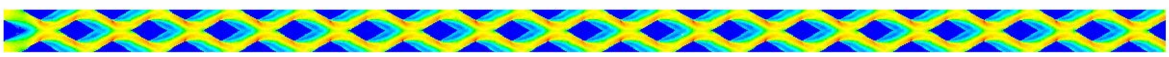


Fig.8 Contour map of velocity in an entire computational domain using X channel

PERFORMANCE EVALUATION

X-channel fin with rounded corners and reduced height

Thermal- hydraulic characteristics for carbon dioxide are shown in Fig.11 in comparison with S-shaped fin as well as smooth tube. X and S fins have much larger Nu and f values than smooth tube.

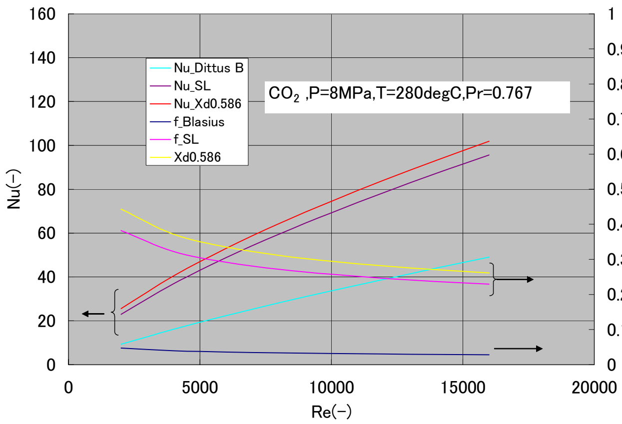


Fig.11 Thermal hydraulic characteristics of X-channel compared with smooth tube and S-shaped fin

Taking smooth tube as a reference surface, and Re_r =10000 under thermodynamic conditions P=8MPa, T=280degC of carbon dioxide leading to Pr=0.767, several parameters were calculated using Eqs.(7)~(10) under equal pumping power condition. Tube bank was assumed to be arranged in a staggered array so that their unit cell has the same free flow area and hydraulic diameter for hot and cold side of banks same as S-shaped fin. This requirement gives distance d_s between adjacent tube centers $\pi^{1/2}3^{-1/4}d$. It is illustrated in Fig.12 Tube thickness was not considered.

Results are listed in Table 1.

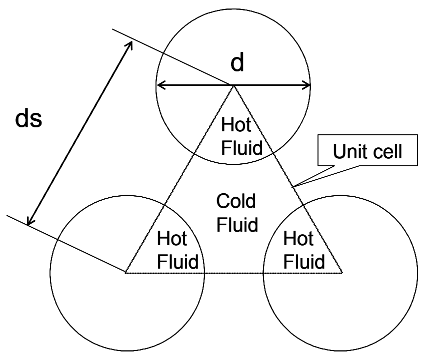


Fig.12 Staggered array tube bank

Table 1 Results of performance evaluation with smooth tube bank as a reference

Surface	Dh (mm)	β (mm^-1)	$(\beta/D_h)/(\beta/D_h)_r$	Nu/Nu_r	q/qr	G/Gr	V/Vr	L/Lr
Smooth tube	1.067	3.75	1	1	1	1	1	1
SL_fin	1.067	2.48	0.66	1.35	0.895	0.542	1.12	0.401
Xd 0.586	0.823	3.7	1.28	1.07	1.37	0.435	0.73	0.242

Compared with smooth tube bank, X-channel fin can provide 37% larger exchange rate with 1/4 shorter length of exchanger. Similarly having shorter length, however, S-shaped has 10% less exchange rate. Mass flux is found likely to be reduced for enhanced surfaces. Taking S-shaped fin as a reference, performance parameter ratios were plotted in Fig.13 as a function of Re of the reference operating condition. Weak dependences on Re were confirmed. V/Vr was obtained as a reciprocal of q/qr that implies volume required to achieve given heat duty under the condition that pumping power is proportional to the volume. Therefore, both the volume and the power was reduced to about 65% compared with the S-shaped.

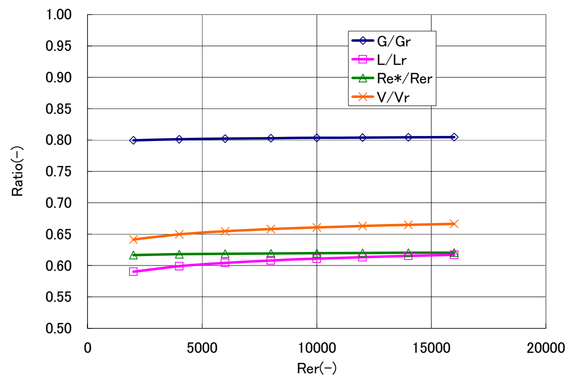


Fig.13 Relative performance of X-channel with S-shaped as a reference

Effect of fin height

Fig.14 shows the effect of fin height. Shorter the fin height, the larger the exchange rate density.

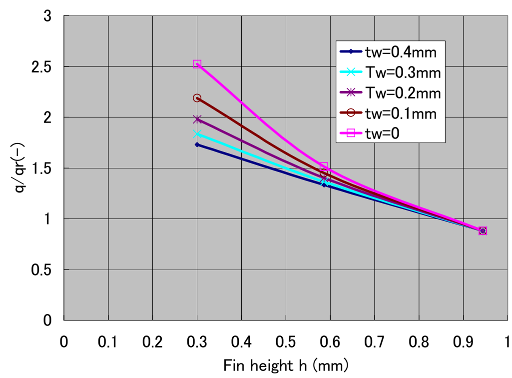


Fig.14 Effect of wall thickness

Effect of wall thickness becomes more significant as fin height becomes shorter. It is because β' (Eq.(12)) is reduced while *U* is not affected since Bi was assumed to be infinitesimally small. Therefore the results may be applicable in the case solid material is made of copper

or aluminum. If it is stainless steel, the assumption Bi is small enough is no longer valid and U>α, then wall resistance should be taken into account.
T=0.4mm is reasonable value for wall thickness to withstand pressure boundary. In this case, the performance is reduced by 20% at 0.586mm of fin height. It is also shown that a full fin height will give worse performance.

Fig.15 exhibits geometrical parameter change with the thickness. Re_r was selected 10000 at reference exchangers. The lower the fin height, the better the performance will appear in X-channel heat exchanger.

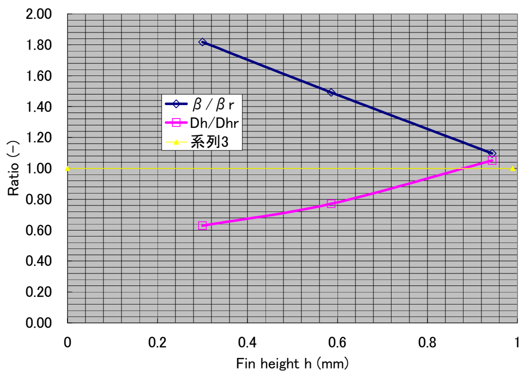


Fig.15 Geometry parameter change with thickness

CONCLUSION

Performance evaluation criterion was derived based on generalized hydraulic diameter and applied to the development of a new plate type of enhanced heat transfer surface. The criterion is capable of comparing performance of surfaces among different geometries. Newly developed X-channel fin has 37% larger heat exchange capacity over smooth tube under the same pumping power and the reference Re_r=10000 conditions.

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REFERENCES

Webb, L.R and Kim, N-H, Principles of Enhanced Heat Transfer, 2nd edition, *Taylor & Francis Group*, (2005)
Hesselgreaves, J. E., Compact Heat Exchangers, *Pergamon*, (2001)
García,A., Pedro G. Vicente and Antonio Viedma, Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers. *International Journal of Heat and Mass Transfer* Vol. 48, Issues 21-22, October 2005, Pages 4640-4651.
Utamura. M, Kimura, K. and Kajita, R.:” Generalized hydraulic diameter for the development of high performance microchannel heat exchangers”, *HEFAT2010,* (2010),Turkey
Tsuzuki, N. Kato, Y. and Ishizuka, T., High performance printed circuit heat exchanger, *HEAT-SET2005,* (2005), France.