T2R2 東京科学大学 リサーチリポジトリ
Science Tokyo Research Repository

論文 / 著書情報
Article / Book Information

論題(和文)	高性能マイクロチャンネル熱交換器の開発のための水力等価直径の一般化
Title(English)	Generalized Hydraulic Diameter for the Development of High Performance Microchannel Heat Exchangers
著者(和文)	宇多村元昭, 木村博幸, 梶田梨奈
Authors(English)	Motoaki utamura, Hiroyuki Kimura, Rina Kajita
Citation(English)	7th International conference on Heat Transfer, Fluid Mechanics and Thermodynamics (HEFAT 2010), Paper No.169
発行日 / Pub. date	2010, 7

Powered by T2R2 (Science Tokyo Research Repository)

HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

GENERALIZED HYDRAULIC DIAMETER FOR THE DEVELOPMENT OF HIGH PERFORMANCE MICROCHANNEL HEAT EXCHANGERS

Utamura M.*[1], Kimura H.[1], Kajita R.[2]
*Author for correspondence
[1] Research Laboratory for Nuclear Reactors,
Tokyo Institute of Technology,
Tokyo, 152-8550,
Japan,
[2]Thermal Engineering & Development
Yokohama, 222-0033
Japan
E-mail:utamura@nr.titech.ac.jp

ABSTRACT

Generalized hydraulic diameter is proposed and applied to the development of high performance microchannel heat exchanger with a new shaped fins. Hydraulic diameter is one of basic concepts to characterize geometry of flow channels from the analogy of circular tube, defined as the ratio of four times of cross sectional area of the flow channel to wetted perimeter of the cross sectional area. For complex geometry such as flow channels with cross sectional area varied along flow direction with junctions and separations, the definition is no longer valid and some arbitrariness may arise by this definition. In order to solve this difficulty as well as avoid the arbitrariness, an alternative approach to redefine hydraulic diameter is attempted as the ratio of four times of fluid volume occupying the flow channel to its wetted surface area (mostly being coincident to the heat transfer area of the flow channel) and use this as the representative length for the design of heat exchangers. This is a straight-forward generalization of the conventional definition. Several applications to the development of high performance 2D flow channel with fins of variable shapes are shown and its usefulness is confirmed. Especilly it is found dimensionless groups such as Nusselt number and pressure loss coefficient derived from design channel shape is also applicable to the actual shape deformed during manufacturing process as long as newly defined hydraulic diameter is adopted. Based on this method a direct comparison of the compactness of the heat exchangers between different shape of fins is made in an rigid manner and high performance fin shape is derived and some their thermal hydraulic correlations are derived using CFD.

INTRODUCTION

High performance heat exchanger is a critical component for industrial facilities especially with waste heat recovery in the vicinity of pseudo-critical points. Hesselgreaves summarized compactness of the heat exchanger in a unified way [1]. Tsuzuki et al. developed compact heat exchanger, named MCHE with S-shaped fins [2] as shown in Fig.1. 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.

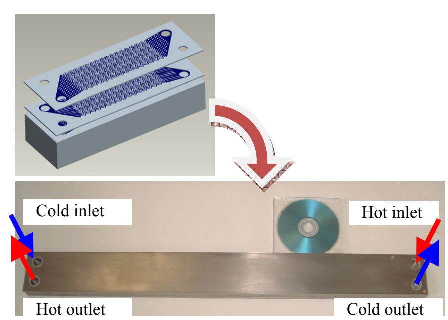

Fig. 1. Microchannel heat exchanger (MCHE).

Chemical etching is an excellent method to form a complex geometry on a plate but is not suitable for mass production because of its high cost. 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. To do this in a right way, we regard the flow channel formed by S shaped fin 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 no longer applicable. Therefore, the first objective of this study is to generalize conventional definition of hydraulic diameter applicable to various channel geometries without arbitrariness. Based on this the second one is to reveal thermal hydraulic characteristics 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 thermal hydraulic characteristics were examined keeping hydraulic diameter unchanged among them.

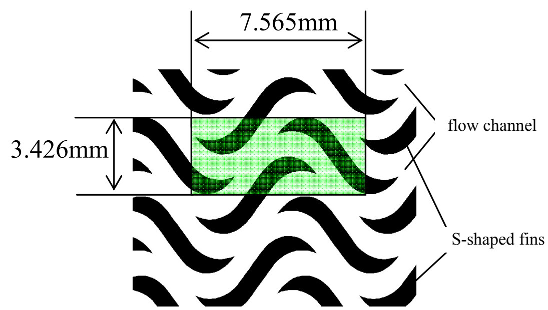

Fig. 2. S-shaped fin configuration.

NOMENCLATURE

A	[m^2]	Heat transfer area
A_c	[m^2]	Average cross sectional flow area to nominal flow direction
A_{cell}	[m^2]	Projected area of a unit cell of flow channel
$D_{h\%}$	[m]	Hydraulic diamer
G	[kg/m^{2}s]	Mass flux
H	[m]	X channel width
H_d	[m]	Channel depth
H_p	[m]	Plate thickness
L	[m]	Length of heat exchanger
L_{ax}	[m]	Longitudinal distance of a unit cell of flow channel
l_{tr}	[m]	Transverse distance of a unit of flow channel
Q	[W/m^3]	Heat exchange rate per unit volume
ΔT	[K]	Average fluid temperature difference
U	[W/m^{2}K]	Heat transmission coefficient, Heat transfer coefficient
v	[m^3/kg]	Specific volume
Greek		
α	[-]	Heat transfer area to unit cell area ratio($=A/A_{cell}$)
β	[mm^{-1}]	Heat transfer area per unit volume of heat exchanger
π	[-]	Two dimensional void fraction of a plate
λ	[W/m/K]	Heat conductivity

TERMAL-HYDRAULIC PARAMETERS

To compare several fin configurations, hydraulic diameter and mass velocity between them are set equal so that Reynolds number is common to all

$$Re = GD_h / \mu \tag{1}$$

To do this rigorously, it is necessary to derive an expression for hydraulic diameter applicable to a network of conduit with variable cross sectional area along flow.

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. Therefore, side wall area of fins 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) Free flow area of heat exchanger A_c is defined as the free volume V divided by the principal 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 here is defined here by

$$D_h = \frac{4V}{A}\,. \tag{2}$$

It is important to note that Eq.(2) can give a unique value for any flow channel configuration regardless of flow direction. It reduces to usual definition of D_h in the case that cross sectional area is unchanged along flow path. In the case of the present XCHE, using the relation $A_c = V/L$ by assumption 2), Eq.(2) reduces to the usual form of $4A_c / a$ where a is average wetted perimeter defined by $a=A/L$. Therefore, Eq.(2) may be regarded as the generalization of the definition of conventional hydraulic diameter to allow variable geometry along the flow path.

COMPACTNESS OF XCHE

In order to find a new flow channel configuration 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

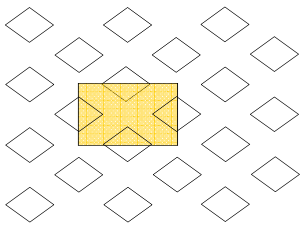

Fig.3 Plan view of X-channel flow configuration

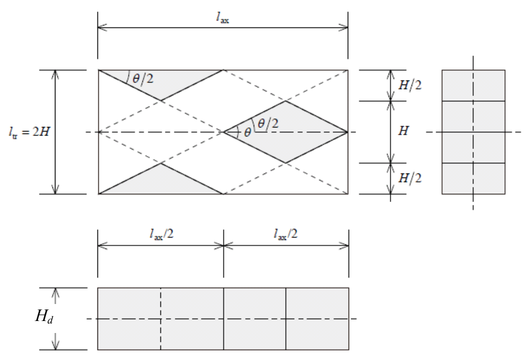

Fig.4 Unit cell of X-shaped channel

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

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

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

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

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

$$A_c \equiv \frac{V}{l_{ax}} = \frac{3}{2} H H_d \tag{4}$$

$$\pi \equiv \frac{V}{H_d A_{cell}} = 0.75 \tag{5}$$

$$\alpha = A / A_{cell} = (3 + \frac{2(H_d/H)}{\cos(\theta/2)})/2 \tag{6}$$

$$\beta = \alpha / H_p \tag{7}$$

It should be noted that average free flow area A_c and two dimensional void fraction π are independent of wedge angle θ. Heat transfer area ratio α is an index of compactness of a heat exchanger more compact with larger α. Here, α depends on θ because side wall area depends on θ. It is not less than 2.5 for every θ and larger than 2.33 of S-shaped fin. Fig.5 shows D_h and α as function of θ in the case that H_d is equal to H. With wedge angle D_h decreases and α increases. D_h and channel depth H_d of S-shaped fin are 1.067mm and 0.94mm respectively and the angle to give equivalent D_h is 58deg. For simplicity, θ is selected as 60deg in the following discussion. If one takes plate thickness H_p as 1.5mm, then compactness of heat exchanger β would become 1.75mm and 1.56mm for X channel and S fin respectively those of which are more than the doubled of the minimum criterion of 0.7mm regarded as compact heat exchanger. Heat conductance per unit volume $U\beta$ is a measure of heat exchanger performance since heat exchange rate per unit volume Q is expressed by

$$Q = U\beta\Delta T \tag{8}$$

Geometrical parameter values of unit cell of X channel are summarized in Table 1 compared with S shaped fin.

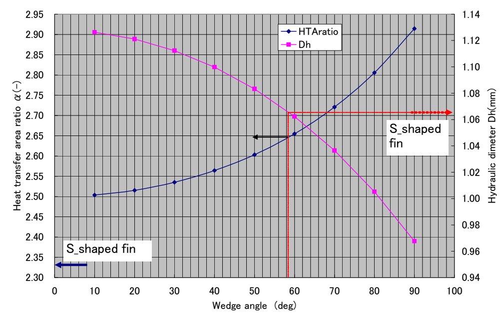

Fig.5 Hydraulic diameter and heat transfer area ratio of X-shaped channel as function of wedge angle θ

Table 1 X channel geometrical parameter values

Variables	Unit	X-channel	S-shaped
H_p	mm	1.5	1.5
D_h	mm	1.062	1.067
l_{ax}	mm	3.271	7.565
l_{tr}	mm	1.889	3.426
H	mm	0.94	0.94
H_d	mm	0.94	0.94
A	mm^2	16.25	60.48
α	–	2.63	2.33
β	mm^{-1}	1.75	1.56

Derivative channel configurations
Derivative fin shapes other than diamond (X channel) were examined. Fig.6 illustrates fin with half of ellipsoid is attached instead of triangle at the latter half part of the fin, named wedge-ellipsoid (WE). Application of the same D_h condition gives length ratio a/a' is equal to 0.780 and α is 2.535 being 0.964 times of the value of X channel. In the case of flow channel configuration where flow is reversed in Fig.6, we call EW and let a shape made of a complete ellipsoid named EE in which α is 2.441. All fins X, WE, EW and EE have α values larger than S shaped fin.

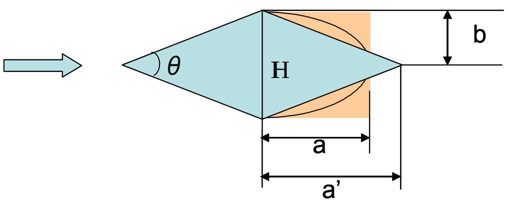

Fig.6 Wedge ellipsoid fin

CFD SIMULATION
CFD was incorporated to investigate heat transfer coefficient and pressure drop coefficient of X channel. Fig.7 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 [3] that requires very high temperature effectiveness and thus compact heat exchanger is indispensable to put it into practice. was taken as example.
About boundary conditions, periodic condition was applied to the both side walls. The bottom plane is symmetrical plane. And the temperature of top wall was 2°C higher than the fluid inlet temperature all the way along the channel to outlet. Spatially homogeneous velocity profile was applied to inlet and free discharge boundary condition at exit boundary of the computational domain.

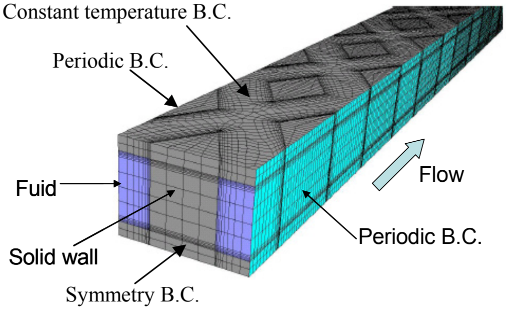

Fig.7 Computational model

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) [4]. 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.8. At 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.

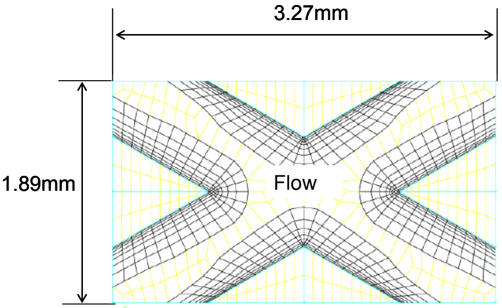

Fig.8 Computational mesh of unit cell of X channel

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 *U* as well as pressure loss coefficient were derived by

$$U = Q/(A\Delta T_{LMTD})$$

$$Nu = UD_h/\lambda \qquad (9)$$

$$f = 2D_h\Delta P/(LvG^2) \qquad (10)$$

Fig.9 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. Figs.10 and 11 show Nu and f as function of Re for various channel shapes in comparison of S shaped fin. X-,WE-,EW-fins have all higher Nu values than S shaped fin and EE at the sake of larger pressure drop coefficient. EE-fin showed most enhanced thermal performance. X channel showed

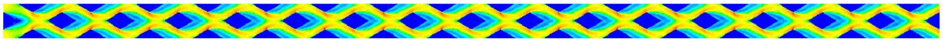

Fig.9 Contour map of velocity in an entire computational domain using X channel

the largest value of pressure drop coefficient f because a pair of vortices were produced just behind the fin. Table 2 compares basic aspect of compactness β and thermal aspect of compactness Nu, and their product βNu normalized by the value of S-fin. For X fin β and Nu become largest and smallest respectively while opposite for EE fin. βNu is much the same among X and its derivatives having a value of 1.21~1.24 over S-fin. From Fig.11 the existence of sharp edge tends to increase pressure drop. To reduce this, four edges of diamond shape of X- fin are made rounded as shown in Fig.12. The result of this deformed X channel is shown in Fig.13. Significant reduction of f was observed being almost the same magnitude of that of S shaped fin while Nu value maintained same as original X channel. From this observation one can estimate enhancement of compactness of heat exchanger from S shaped fin. From Table1 relative increase of β is 13% while from Fig.13 Nu number increases 9.5% around Re=8000, thus the sum of both i.e. relative increase of heat conductance $U\beta$ in Eq.(8) gives 22.5%.

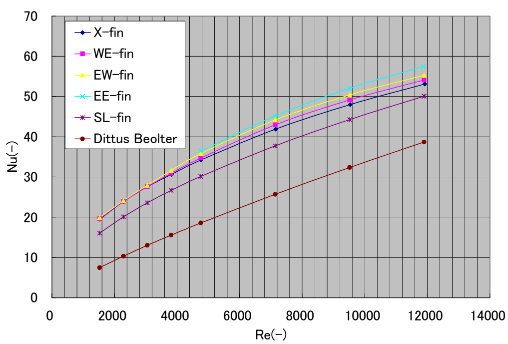

Fig.10 Nu of X channel compared with S shaped fin

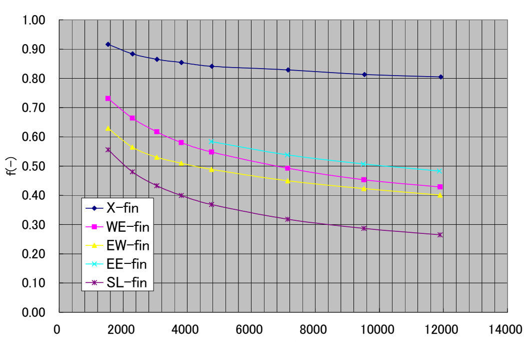

Fig.11 Pressure drop coefficient

Table 2 Compactness of basic and thermal aspects

Re=10000,D_h=1.067mm

Fin type	β (mm^{-1})	Nu(-)	(β Nu)/(β Nu)$_s$
S	1.56	44	1
X	1.75	48	1.22
WE	1.70	49	1.21
EW	1.70	50	1.24
EE	1.63	52	1.23

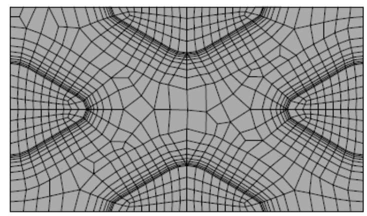

Fig.12 Calculation mesh of deformed X channel

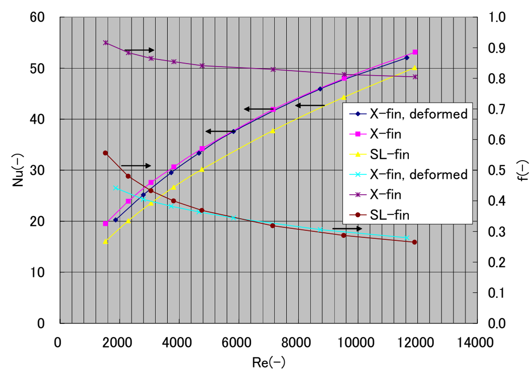

Fig.13 Thermal hydraulic characteristics of original and deformed shape X-channels

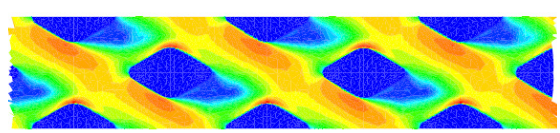

Fig.14 Flow pattern in deformed X channel

To find the reason why pressure drop significantly decreases, flow pattern of deformed X channel was examined. Fig.14 shows velocity contour in the middle part of the domain. It is seen most flow develops along X channel in parallel in one direction. Onset of this oblique flow was found dependent on channel depth. Theoretically this flow may develop in the other direction i.e. anticlockwise rotation by 60deg.

CONCLUSION

New formulation of hydraulic diameter was proposed that can be applied to a network of flow channels with a variety of geometry. It succeeds in avoiding arbitrariness present in specifying cross sectional area in flow path and thus calculating hydraulic diameter in a unique manner. Based upon the formulation several fin patterns engraved on a plate were created with the hydraulic diameter kept to a specified value and their thermal hydraulic characteristics were calculated numerically and directly compared under the same Reynolds number. Bilateral X channel engraved on a plate was investigated. It was found repetitive nature of combined confluence and bifurcation of flow generally enhances heat transfer at the sake of pressure drop increase. Also specific heat transfer area contributes to even better thermal performance than that of S shaped fin. It was also found deformed X channel with blunt edge significantly reduced pressure drop down to the level of S shaped fin. It was due to it that flow pattern changes from symmetry to asymmetry. It is expected the onset of the flow along X channel contributes to the reduction of pressure drop. Further investigation is necessary to find geometrical condition to emerge oblique channel flow.

REFERENCES

[1] Tsuzuki, N. Kato, Y. and Ishizuka, T., High performance printed circuit heat exchanger, HEAT-SET2005, (2005), France.

[2] Hesselgreaves, J. E., Compact heat exchanger, Pergamon, 2001

[3] Utamura, M., Nikitin, K., Characteristics of a closed cycle gas turbine with supercritical CO_2 as working fluid, Proc. HEFAT2008 paper MU1, (2008).

[4] PROPATH group, PROPATH: a program package for thermo-physical properties of fluids, version 10.2, PROPATH group, Japan(1990)