Fluid Flow and Heat Transfer in a Two-dimensional Miter-bend

(1st Report: Experiments and Analyses)*

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Fluid flow and heat transfer in a two-dimensional miter-bend were examined in connection with a corrugated wall channel as a means of augmenting a forced-convection heat transfer with a single-phase flow in a heat exchanger. Pressure losses, distributions of mean velocities and velocity fluctuations, streaklines, power spectra of velocity fluctuation and local Nusselt number distributions were compared with those of numerical analysis. It is shown that the effect of heat transfer augmentation is remarkable on the Reynolds number of the order of 2000, where the laminar flow is maintained in the straight channel.

Key Words: Convective Heat Transfer, Experiment, Numerical Analysis, Miter-Bend, Velocity Fluctuation, Streakline, Power Spectrum, Strouhal Number, Local Nusselt Number.

1. Introduction

The authors(1)~(5) have systematically studied a corrugated wall channel as a means of augmenting a forced-convection heat transfer with a single-phase flow in a heat exchanger. In those studies, a theoretical analysis and an experiment were made by considering a model of a parallel channel with an infinite span. However, the mechanism of the fluid flow and the heat transfer has not been sufficiently explained. Accordingly, in this paper, dealing with the miter-bend as the most fundamental element of the corrugated wall channel, we investigate it experimentally and analytically in more detail.

In this experiment, we examine the mean velocity, the intensity and the spectrum of velocity fluctuation, which were not shown in the previous papers(2)(3), together with the pressure loss, the streakline and the local Nusselt

number distribution. In this analysis, we calculate numerically the fluid flow and the heat transfer for the laminar flow and the turbulent flow by the finite difference method used in the previous papers(1)(3)(4).

Nomenclature

f : frequency of fluctuations
 H : channel width
 h : heat transfer coefficient
 k : kinetic energy of turbulence
 l : length scale of turbulence
 L_c : concave corner of channel wall
 Nu : Nusselt number
 p : pressure
 q : heat flux
 Re : Reynolds number
 R_o : convex corner of channel wall
 St : Strouhal number
 u : velocity
 x, y : coordinates in the flow direction and perpendicular to wall
 ϵ : turbulent kinetic energy dissipation rate
 ζ : pressure loss coefficient
 θ : temperature
 λ : thermal conductivity or friction factor
 ν : kinematic viscosity
 ρ : density

Subscripts

b : bulk
 m : cross-sectional mean
 max : maximum
 w : wall
 x : local
 ∞ : asymptotic value
 $-$: mean
 $'$: fluctuation

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2. Methods of Experiment and Numerical Calculation

A two-dimensional miter-bend with a 90° turn is chosen as the subject of this paper. We call the walls on the right-hand side and the left-hand side towards the downstream "right-side wall" and "left-side wall" respectively, and we define the concave and the convex corners as " L_c " and " R_v " respectively. We take a Cartesian coordinate system, where the origin is at the corner R_v , x is in the flow direction and y is perpendicular to the wall. The characteristic length for the Reynolds number Re and the Nusselt number Nu is $2H$, that is, double the width of the channel.

We improved the experimental apparatus employed in the previous papers⁽²⁾⁽³⁾. The equipment unit consists of a rectangular channel of $150\text{mm} \times 20\text{mm}$ cross section (the aspect ratio $\alpha=7.5$). The bending region was located at a distance of $2m$ downstream of the calming chamber. Air at an atmospheric pressure and temperature was used as the fluid in measuring the pressure, the velocity and the heat transfer coefficient. The wall pressure was measured by means of a displacement micro manometer through the static pressure taps attached to both side walls of the channel. The velocity was measured by means of a hot-wire anemometer with I-type and L-type probes. The signals were processed and arranged by an RMS meter and an FFT spectrum analyzer. The heat transfer coefficient was measured on a surface of constant heat flux which was produced by applying an electric current directly to a $30\mu\text{m}$ thick stainless steel foil. The heat starting point was set at a distance of 5 times H upstream of the corner R_v . The wall temperature was measured by means of a C-C thermocouple of $100\mu\text{m}$ diameter which was put on the hidden side of the stainless steel foil. The observation of the streakline was performed in another equipment unit using water as the fluid. The flow was visualized by the dye injection method with rhodamine-B as the tracer.

The numerical calculation was performed for a steady laminar flow ($Re=u_m 2H/\nu < 4000$) and for a turbulent flow ($Re=36300$, using k - k_l and k - ϵ models) by the finite difference method in terms of vorticity and stream function, which had been used in the previous papers⁽¹⁾⁽³⁾⁽⁴⁾, where u_m is the cross-sectional mean velocity and ν is the kinematic viscosity.

3. Experimental and Calculated Results and Discussions

3.1 Pressure loss

The friction factor for a fully developed flow at the upstream of bending region and the pressure loss coefficient due to bending are defined as λ and $\zeta = \Delta p / (1/2) \rho u_m^2$, respectively, where Δp is the net pressure loss due to bending, that is, the total pressure loss minus the

pressure loss in the straight channel with the same flow length. The relationship between λ and Re and that between ζ and Re are shown in Figs.1 and 2 with symbols $\circ$ and solid lines. The dashed lines in Fig.1 indicate the friction factors calculated by the formula of Jones⁽⁶⁾ for the laminar and the turbulent flow.

3.2 Mean velocity and intensity of velocity fluctuation

The mean velocity $\bar{u}/u_m$ and the intensity of velocity fluctuation $k/(1/2)u_m^2$ are shown in Figs.3 and 4, where (a) and (b) are examples for the laminar flow ($Re=1820$) and the turbulent flow ($Re=36300$), respectively. The value of k was calculated from the equation $k=(3/2)u'^2$ on the assumption of isotropic turbulence. Here u and u' are the components of the mean velocity and the velocity fluctuation in the flow direction. The output voltage of the hot-wire anemometer is naturally converted into the velocity. As a natural consequence, the value of the velocity is positive even in the separation regions at the downstream of the corner R_v (to the point about 3 times H from R_v) and in the neighborhood of the corner L_c , where the flow changes its direction. Moreover, the intensity of velocity fluctuation is not corrected even in the case where the mean velocity and the velocity fluctuation are in the same order. Therefore, some inaccuracy cannot be avoided in the measurement in such regions, but we think that the general tendency of the flow is almost independent of such inaccuracy. The velocity distribution in the direction of

Fig. 1 Friction factor in the straight channel

Fig. 2 Relationship between ζ and Re

channel width was measured changing the position of L-type probe at the center of channel width, so there is a difference in the profile of the measured value at the downstream of the corner R_v (to the point about 4 times H from R_v). However, since the velocity changes very much in the

Fig. 3 Distribution of $\bar{u}/u_m$

Fig. 4 Contour of $k/(1/2)u_m^2$

Fig. 5 Photographs of streakline

Fig. 6 Spectrum of velocity fluctuations ($Re=1820$)

Fig. 7 Relationship between St and Re

direction of channel width, the effect of such a difference on the mean velocity and the intensity of velocity fluctuation can be neglected. The comparison of Fig.3(a) with Fig.3(b) makes it clear that the effects of bending appear more remarkably at $Re=1820$ than at $Re=36300$. Especially, at $Re=1820$, the flow-separation region is large and the flow-acceleration is remarkable. Moreover, the effect of bending extends to fairly far downstream, that is, the velocity distribution is symmetric but still trapezoidal at a distance of about 9 times H downstream of the corner R_v , and it barely returns to a parabolic curve at a distance of about 80 times H downstream of R_v . Figures 4(a) and (b) show the growth process of the turbulence in the shear layer where the velocity gradient is steep. The positions where the intensity of velocity fluctuation has a maximum are at the distances of $3.5H$ and $2.5H$ downstream of the corner R_v for $Re=1820$ and $Re=36300$, respectively. The production of the turbulence is suppressed at the flow-acceleration region. The intensity of velocity fluctuation is made uniform in the direction of channel width in the flow-recovery region downstream of R_v , and there occur maxima near the walls of channel farther downstream of R_v with a decrease of the value in the center of channel width. The experimental result is generally similar to the calculated results for the laminar and the turbulent flow⁽¹⁾⁽⁴⁾. But, the former differs somewhat from the latter in the following points: there exists a turbulence at the downstream of the corner R_v as shown in Fig.4(a), the position where the intensity of turbulence has a maximum deviates from the analytical one, and the intensity of turbulence near the impinging point is fairly small.

3.3 Streakline

The photographs of the streaklines visualized by the dye injection method⁽¹⁰⁾ are shown in Fig.5. A critical value of the Reynolds number for the transition in the straight channel (the flow between two parallel walls) is $3000 \sim 4000$, while the flow in the miter-bend is unsteady in the region of $Re > 460$. Namely, the vortices forced by the Kelvin-Helmholtz shear-layer instability are shed in a regular street in the shear layer at the downstream of the corner R_v , and the flow is disturbed and fluctuated unsteadily by these vortices. In this paper, the above mentioned behavior is termed "the flow unsteadiness". Thus, the state of fluid flow at the downstream of the bending region changes, with an increase of the Reynolds number, in the order of steady flow, unsteady flow, and turbulent flow in which the existence of a vortex street can be hardly recognized. The intensity of the velocity fluctuation described in the previous section naturally contains not only ordinary turbulent components but also relatively low frequency components due to the unsteady flow. Comparing Fig.5(b) with Fig.4(a), the intensity of

the velocity fluctuation becomes high with the growth of vortex street shed from the corner R_v , and then it reaches a maximum at the position where the vortex street attains its full growth. Farther downstream, with the decay of the vortices, the intensity becomes uniform in the direction of the channel width, and then it asymptotically approaches a very small value. The Strouhal numbers $St=f2H/u_w$, which are calculated with the vortex shedding frequency f measured on the basis of the streaklines, are indicated with the symbol Δ in Fig.7 as described later.

3.4 Spectrum of velocity fluctuation

Examples of the power-spectrum of the velocity fluctuation $P(f)$ at $Re=1820$ are shown in Figs.6(a), (b), (c) and (d), corresponding to the positions $x/H=0, 1.0, 2.0$ and 4.0 with a fixed value of $y/H=0.65$, respectively. As shown in Fig.6(a), the spectral distribution at the position where the flow issued from the neighborhood of the corner R_v is reached, has a distinguished value at $f=39$ Hz ($St=2.4$) corresponding to the vortex shedding as mentioned above. Farther downstream of R_v , the spectra are generally larger, and they monotonously decrease with an increase of f in the same way as a turbulent flow. A similar tendency is shown at the other values of Re . The Strouhal number St derived from the spectral distribution at the position $x/H=1.5$ and $y/H=0.65$ is shown in Fig.7. The Strouhal number St increases with an increase of Re in the region of $Re < 2000$. The distinguished value of the frequency can be hardly recognized in the region of $2000 < Re < 4000$, since the spectral distribution has not a steep peak and it is leveled. At $Re=36300$, the components of high frequency decrease in the spectral distribution at the position $x/H=0.5$ and $y/H=0.65 \sim 0.85$ corresponding to the flow-acceleration region, and the spectral distribution in the shear layer at the immediate downstream of the corner R_v has a maximum at $f=1$ kHz ($St=2.8$). Farther downstream, the components of high frequency increase and the overall value of the spectrum becomes gradually large.

3.5 Local Nusselt number

Examples of the distributions in the flow direction of the local Nusselt number Nu_x are shown in Figs.8 and 9, where the symbols $\circ$ and Δ indicate the experimental results, and the solid lines indicate the analytical results. The Nusselt number is defined as follows:

$$Nu_x = h2H/\lambda \\ = (q/\lambda)2H / (\theta_w - \theta_b)$$

where h is the thermal conductivity of the fluid, q is the wall heat flux, and θ_w and θ_b are the wall and the bulk temperature⁽²⁾⁽³⁾. The experimental and the analytical results differ considerably from each other. This difference is caused by the flow unsteadiness and the transition to the turbulent flow at the downstream of bending region for the low

Reynolds number such as $Re=1820$, while it is caused by the inadequate parts of the analytical model of the turbulence for the high Reynolds number such as $Re=36300$. The former will be examined in detail by the numerical calculation of the unsteady solution of Navier-Stokes equation in the continued paper⁽¹¹⁾. The latter requires more complicated turbulent models and advanced techniques such as large-eddy-simulation. The effect of the Reynolds number on the local Nusselt number distribution is shown in Fig.10. The heat transfer is remarkably augmented at the position where the main flow impinges and the thermal boundary layer is thin, and at the position where the turbulent energy is large and the heat transport effect is excellent. These results correspond well to the fluid flow states as shown in Figs.3 and 4. Especially, there are two peaks of Nu_x at the left-hand side wall: the front and the rear peaks are due to the flow impingement and the turbulent energy, respectively. It should be noted that the calculated result of the unsteady solution in the continued paper⁽¹¹⁾ reproduces two such peaks which were not indicated in the present calculation.

3.6 Maximum and asymptotic value of local Nusselt number

The relationships between Nu_{max} or Nu_{∞} and Re are shown in Fig.11, where Nu_{max} is the largest maximum at each side of wall and Nu_{∞} is the minimum at the

downstream of bending region in the range of this measurement. The dashed line (A) corresponds to the analytical result $Nu_{\infty}=7.95$ for the laminar flow⁽⁷⁾, which has been reduced using the definition of this paper. The dashed lines (B) and (C) correspond to the experimental results for the turbulent flow by Novotny et al.⁽⁸⁾ and Dittus-Boelter's formula⁽⁹⁾, respectively. The value of Nu_{max} is generally more than twice as large as Nu_{∞} . Especially, the heat transfer augmentation is remarkable for the Reynolds number of the order of 2000, where the laminar flow is maintained in the straight channel.

4. Conclusions

We examined experimentally and analytically the fluid flow and the heat transfer in a two-dimensional miter-bend as the most fundamental element of a corrugated wall channel. The results are summarized as follows:

(1) The relationship between pressure loss coefficient due to bending and the Reynolds number, the distributions of the mean velocities and the velocity fluctuations, the streaklines, the power spectra of the velocity fluctuation are shown, and then the fluid flow characteristics in the miter-bend are made clear. Especially, the flow is unsteady in the region of $Re>460$. The vortices forced by the Kelvin-Helmholtz shear-layer

(a) right-hand side wall

(b) left-hand side wall

Fig. 8 Distribution of local Nusselt numbers ($Re=1820$)

(a) right-hand side wall

(b) left-hand side wall

Fig. 9 Distribution of local Nusselt numbers ($Re=36300$)

(a) right-hand side wall

(b) left-hand side wall

Fig. 10 Effects of Reynolds number on the local Nusselt number

instability are shed in a regular street in the shear layer at the downstream of the corner R_v and the flow is disturbed and fluctuated unsteadily by these vortices.

(2) The local Nusselt number distributions in the flow direction are shown, and the relationships between the maximum Nu_{max} or the asymptotic value Nu_{∞} of the local Nusselt number and the Reynolds number Re are made clear. The experimental results differ considerably from the analytical results. This is caused by the flow unsteadiness and the transition to the turbulent flow at the downstream of bending region or by the inadequate parts of the analytical model of the turbulence. The heat transfer is remarkably augmented at the position where the main flow impinges and the thermal boundary layer is thin, and at the position where the turbulent energy is large and the heat transport effect is excellent. The value of Nu_{max} is generally more than twice as large as Nu_{∞} . Especially, the heat transfer augmentation is remarkable for the Reynolds number of the order of 2000, where the laminar flow is maintained in the straight channel.

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Fig. 11 Relationship between Nu_{max} or Nu_{∞} and Re

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