Experimental and numerical investigation of a pulsatile flow field in an S-shaped exhaust pipe of an automotive engine

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Abstract
A pulsatile turbulent flow within an S-shaped double bend pipe is experimentally and numerically studied to characterize the flow field in conditions resembling an automotive engine environment. Particle image velocimetry (PIV) measurements were carried out to measure streamwise and secondary flow velocities. The flows are accelerated around the inner side walls of both bends. The secondary flow, after passing through the second bend, is directed toward the inner side in the core of the cross section, and, as a result, Lyne-type vortices, which are not consistent with the second bend curvature, are formed. A numerical simulation is performed under the same condition as the experiments with computational fluid dynamics software. The numerical simulation gives qualitative results in comparison with the experimental data though there is some deviation, and shows the cause of the Lyne-type vortex formation in the second bend. After passing through the first bend, the high-speed region appearing around the inner side shifts in accordance with the Dean-type secondary flow formed in the first bend, and thus the non-uniform flow enters the second bend. In the second bend, the low-velocity region in which the centrifugal force is not strong enough to direct the flow toward the outer side, appears in the core of the cross section. Details of the Lyne-type vortex formation are discussed by considering the driving forces of the secondary flow.

Key words: Automotive engine, Pulsatile flow, Turbulent flow, Secondary flow, Particle image velocimetry, Computational fluid dynamics

1. Introduction

A close-coupled catalytic converter (CCC) has been widely used as an emission control device in automotive engines. A CCC is mounted close to an exhaust manifold to ensure the rapid warm-up of the converter during a cold engine start. The exhaust gas velocity distribution influences the performance of the CCC and tends to be complicated because of the unsteadiness of the exhaust flow, its highly turbulent nature, and the manifold geometry, including the bend pipes. Therefore, the flow condition in the manifold is important for maintaining the high performance of the catalytic converter.

Generally, a uniform flow is desirable for a high conversion efficiency and durability of the CCC. Many studies have investigated the exhaust gas velocity distribution and its uniformity in front of, or behind, a catalytic converter. Liu et al. (2003) reported the effect of mass flow rate and pulsation frequency on the velocity profiles through the converter by means of hot-wire anemometry. Persoons et al. (2006) measured the velocity distribution in the presence of reverse flow in a real exhaust manifold. Hwang et al. (2007) and Jeong (2014) carried out a computational fluid dynamics (CFD) analysis to investigate flow uniformity in exhaust manifolds. However, most studies related to pulsatile exhaust flows have not considered the detailed flow characteristics, such as curvature-related secondary flow.

Laminar flow in a circular-sectioned bend was first analyzed theoretically by Dean (1927), who showed the existence of a secondary vortical structure consisting of two symmetrical counter-rotating vortices, the so-called Dean vortices. In
turbulent flows, the instantaneous secondary motion exhibits unsteady and asymmetric behavior. Tunstall and Harvey (1968) found that the turbulent flow downstream of the sharp bend is dominated by a clockwise or anticlockwise streamwise vortex. Brücker (1998) performed particle image velocimetry (PIV) to investigate the unsteady phenomena of secondary vortices and found that the symmetry plane of the Dean vortices oscillates about the bend plane. More recent studies by Rütten et al. (2005) and Sakakibara and Machida (2012) also reported the unsteady phenomena of secondary flows under turbulent conditions.

Lyne (1970) first analyzed oscillatory laminar flow for a time-averaged velocity of zero. In that work, the additional two vortices, in the opposite rotational direction to the Dean type, were observed in the core of the bend pipe, and these vortices are known as Lyne-type vortices. Sudo et al. (1992) investigated the oscillatory laminar flow in the bend and suggested that secondary flow structures can be classified into five patterns. The Lyne-type vortices are seen under pulsatile conditions, where the time-averaged velocity is not zero, as presented in the numerical study of physiological flow by Chang and Tarbell (1985). Timité et al. (2010) reported that the secondary flow intensity increases with the appearance of the Lyne-type vortices. More recently, the complex secondary flow structures, consisting of multiple vortices in addition to the Dean and Lyne types, were investigated under physiological pulsatile conditions both through experiments and numerical simulations (Glenn et al., 2012; van Wyk et al., 2015). Whereas most studies on pulsatile bend flows were restricted to laminar cases, Kalpakli et al. (2011, 2013) measured three velocity components of the pulsatile turbulent flow in the cross section of a 90° bend with the technique of stereoscopic PIV. However, none of studies considered the secondary flow features of pulsatile exhaust flow through complex geometry, such as in an exhaust system.

In this study, we experimentally and numerically investigate the pulsatile exhaust flow through an S-shaped bend pipe, which consists of two bends and straight pipes, simulating an exhaust manifold. This study aims at characterizing the streamwise and secondary flows passing through the double bend. In the experiments, an automotive engine used as a flow generator can realize the pulsatile turbulent condition close to an engine environment, and PIV was performed to visualize the flow field. The numerical simulation presented in this paper describes the secondary flow structures developing through the S-shaped pipe and its formation mechanism in more detail.

2. Methodology

2.1 Experimental set up

2.1.1 Flow rig and test section

A flow rig used in the experiments comprised a four-cylinder gasoline engine and the S-shaped test section as illustrated in Fig. 1. An inverter-driven induction motor drove the engine at 1500 rpm during wide open throttle. Room temperature exhaust air was delivered into the test section because the engine was operated without fuel and combustion. A circular-sectioned exhaust pipe, which had a hole on the pipe wall, was mounted between the exhaust manifold and the test section for the introduction of tracer particles for the PIV measurements.

The square-sectioned S-shaped pipe was fabricated from transparent acrylic to enable optical access into the test section for the PIV measurements. The test section had two 57.7° bends with a 50 mm radius of curvature and a 32 mm hydraulic diameter as shown in the drawing (Fig. 2). A square cross section was used for the test section to avoid optical refraction during the photographing of the PIV. The 200 mm straight pipe was installed between two bends. The upstream and downstream bends are denoted the first bend and the second bend, respectively. We performed two types of PIV in the test section: two-dimensional two-component (2D2C) PIV, and two-dimensional three-component (2D3C) PIV.

![Fig. 1 Schematic of the flow rig configuration.](image1)

![Fig. 2 Drawing of the S-shaped exhaust pipe. All lengths shown in the figure are in mm.](image2)
To examine the inflow conditions to the test section, a time history of the streamwise velocity was measured using a hot-wire (0251R-T5, KANOMAX). The position of the hot-wire probe was centered in the cross section located 50 mm downstream from the inlet of the test section. The data were acquired with a sampling frequency of 10 kHz. The period of an engine cycle is 0.08 s in the case of 1500 rpm, and the instantaneous data are repeated for two engine cycles in Fig. 3. The graph indicates that the velocity waveform during an engine cycle includes four large peaks corresponding to the exhaust strokes of the four cylinders.

![Instantaneous velocity (black line) at the test section inlet and Q-switch signal (blue line) of the PIV laser. Velocity data were measured with a hot-wire probe.](image)

2.1.2 2D2C PIV

Two-dimensional two-component PIV was employed to measure the flow fields with two velocity components in the planes of both the two bends. The measured sections correspond to the bend planes located at the center of the pipe as shown in Fig. 4. The flow was seeded with an olive oil mist injected upstream of the test section (see Fig. 1). The oil was atomized using a Laskin nozzle-based oil mist generator (Flowtech Research, FTR Oil Mist Generator).

The 2D2C PIV system consisted of a light source and a single camera. A schematic of the PIV set up is presented in Fig. 5. A double pulsed Q-switched Nd:YAG laser (LOTIS TII, LS-2132 PIV) was used as the light source, and a light sheet of 1-mm thickness was produced by passing the laser beam through a cylindrical lens. The seeded flow was illuminated from the side of the test section. A CCD camera (PCO AG, PCO.1600) with 1600 × 1200 pixels was situated above the test section to view tracer particles. The measurements were conducted in the first and second bends separately.

To externally trigger the PIV system by the engine operation, an ignition signal from the engine was inputted to a programmable controller (Lab Smith, LC880) connected to the light source and the CCD camera. This synchronization enables the capture of a snapshot every engine cycle: phase-locked image acquisition, and the number of images can be obtained by continuously operating the engine. The graph in Fig. 3 shows a Q-switch signal of the laser as well as the velocity data. As the laser is irradiated by Q-switching, the rising times of the Q-switch signal correspond to the measuring times of the PIV. As seen in Fig. 3, pairs of particle images were recorded at the maximum phase of the velocity, during which it is assumed that an exhaust flow of a high flow rate strongly affects catalyst performance in an automotive engine.

Commercial software Koncerto (Seika Digital Image Corp.) was used for image acquisition and post-processing. The time interval between image pairs (Δt) was 5 μs so that the displacement of the particles was approximately one quarter of an interrogation window size. The image pairs were correlated using a multi-pass correlation algorithm with a 16 × 16 pixel interrogation window with 50 % overlap. In each of the planes, the 1000 instantaneous velocity fields were calculated to obtain the phase-averaged velocity fields.

2.1.3 2D3C PIV (Stereoscopic PIV)

Stereoscopic PIV is classified as a 2D3C measurement capable of obtaining three velocity components in a plane by stereoscopic photography using two cameras. We measured three velocity components in the pipe cross section located 25.2 mm downstream from the second bend exit for observation of the secondary and streamwise flows developing through the double bend (Fig. 4). The method for seeding the flow was the same as that in the 2D2C PIV.

The stereoscopic PIV system consisted of the light source common to the 2D2C PIV and two CCD cameras (JAI, AM-200GE) with 1600 × 1200 pixels. As presented in the stereoscopic PIV set-up of Fig. 5, the test section was rotated by 90° when mounting it to the engine to optimize the arrangement of the PIV equipment. This modification does not
affect the flow condition in the test section because the velocity flowing into the test section is spatially uniform. A light sheet of 2-mm thickness was produced to illuminate the seeded flow from the underside of the test section. The two CCD cameras were situated on both sides of the test section. A lens mounted to each of the cameras was adjusted so that the positional relation of object, lens, and image planes could satisfy the Scheimpflug condition. Both the laser and cameras were externally triggered by a delay pulse generator (Flowtech Research Inc., FtrVSD2000) to which the engine ignition signal was inputted for the phase-locked measurement. The 1000 image pairs were acquired at each maximum phase to obtain the phase-averaged velocity field, as with the 2D2C PIV.

The FtrPIV software (Flowtech Research Inc.) used in the stereoscopic PIV is based on a direct cross-correlation algorithm for the particle-image correlation. A pair of particle images with a $65 \times 65$ pixel interrogation window and 50% overlap were correlated within a range of an $80 \times 80$ pixel search window. The time interval $\Delta t$ was set as 5 $\mu$s.

![Fig. 4 Measurement planes highlighted by gray regions for 2D2C PIV (left) and 2D3C PIV (right).](image)

![Fig. 5 Arrangement of 2D2C PIV (left) and 2D3C PIV (right) equipment. The left figure is a schematic during the measurement in the second bend.](image)

### 2.2 Numerical simulation

CONVERGE™ (Convergent Science) CFD software was used for numerical analysis under the same conditions as the experiments. As shown in Fig. 6, the computational domain consisted of the S-shaped pipe (yellow), circular-sectioned exhaust pipes (green) placed at both the upstream and downstream sides of the test section, and a duct (blue) mounted for discharging the exhaust flow seeded with atomized oil in the experiments. The whole computational domain is discretized with rectangular grids at run time, and a cut-cell Cartesian grid generation method is used on the boundary. Figure 7 shows the computational grid generated in the present calculation. The base grid size was 4 mm and refined to 1 mm in the S-shaped pipe where an accurate result is required.

The three-dimensional governing equations for compressible flow are solved in finite volume form on a collocated grid. The equations are evolved in time using the method of implicit time discretization. The MUSCL (Monotonic Upstream-Centered Scheme for Conservation Laws) scheme is used to achieve third-order spatial accuracy for the
convection term. The PISO (Pressure Implicit with Splitting of Operator) algorithm is applied for the pressure-velocity coupling. In this study, the RNG (Re-Normalization Group) k-ε turbulence model is used. The law-of-the-wall provides the momentum boundary conditions at the wall because the RNG k-ε model does not resolve the region in the vicinity of the wall. The specific law-of-the-wall used for this purpose is described by

\[
 u^* = \begin{cases} 
 \frac{1}{\kappa} \ln(Ey^+) & y^+ > 11.2 \\
 y^+ & y^+ \leq 11.2 
\end{cases}, \tag{1}
\]

and

\[
 y^+ = \frac{y \mu u_z}{\mu}, \tag{3}
\]

where the von Karman constant \( \kappa \) is 0.4187, the coefficient \( E \) is 9.8, \( \rho \) is the fluid density, \( \eta_j \) is the unit vector, \( \sigma_{ij} \) is the stress tensor, \( y \) is the distance from the wall, and \( \mu \) is the viscosity. The tangential relative velocity \( U_{i,tan} \) and the sheer speed \( u_t \) are given as

\[
 U_{i,tan} = U_j - (U_j \eta_j) \eta_i, \tag{4}
\]

and

\[
 u_t = c_{\mu}^{1/4} k^{1/2}, \tag{5}
\]

respectively, where \( U_j \) is the fluid velocity, \( c_{\mu} (0.0845) \) is the model constant from the RNG k-ε model, and \( k \) is the turbulent kinetic energy. In CONVERGE, the law-of-the-wall determines the tangential components of the stress tensor at the wall.

The velocity data obtained from the hot-wire experiment are used as the inlet boundary condition. The inlet boundary is at the position where the hole for introducing tracer particles was installed on the pipe wall, and where the hot-wire probe was inserted into the pipe center to obtain the inlet boundary condition. The instantaneous velocity data were phase-averaged over 100 engine cycle. The velocity waveform shown in Fig. 8 is applied as a uniform flow at the inlet boundary. The waveform of the inlet boundary condition is clearly different from the graph presented in Fig. 3 as a result of the different sampling points. The velocity data shown in Fig. 3 was acquired in the test section, which has a hydraulic diameter of 32 mm, while that shown in Fig. 8 was collected in a 50-mm diameter circular section. Therefore, the magnitude of the velocity in Fig. 8 is smaller than that in Fig. 3. The outlet boundary condition is set at an ambient pressure of 0.1 MPa.

![Fig. 6 Computational domain used in the numerical simulation. The calculation domain consists of the S-shaped pipe (yellow), a circular-sectioned pipe (green), and an exhaust duct (blue).](image-url)
2.3 Non-dimensional flow parameters

A number of studies dealing with unsteady pulsatile flows in pipes have characterized flows using the Reynolds number based on time-averaged velocity, which is given by

\[
Re = \frac{W_m D}{\nu},
\]

where \( W_m \) is time-averaged streamwise velocity, \( D \) is the hydraulic diameter of a pipe, and \( \nu \) is the kinematic viscosity of the fluid.

Dean (1928) introduced the important parameter that characterized bend flows generally for laminar cases in the course of his theoretical work, which is today known as the Dean number,

\[
De = \frac{Re}{\sqrt{2R_c}},
\]

where \( R_c \) is the radius of curvature of a pipe bend. The Dean number is the combination of the Reynolds number and a curvature ratio \( D/2R_c \), and interpreted as the ratio between centrifugal and viscous forces.

When considering oscillatory or pulsatile flows, the Womersley number (Womersley, 1955) is a widely used flow parameter given by

\[
\alpha = \frac{D}{2} \sqrt{\frac{\omega}{\nu}},
\]

where \( \omega \) is the angular frequency of the pulsation. The Womersley number can be interpreted as the interaction between the inertial effect due to the pulsation and the viscous effect. A large value of the Womersley number implies that the boundary layer has had insufficient time to develop over the small pulsation period.

The above flow parameters in this study are determined based on the hot-wire result measured in the test section. When calculating the Womersley number, the pulsation period is regarded as 0.02 s, a quarter of the period per engine cycle, since the velocity waveform during an engine cycle includes four peaks from the four cylinders. The calculated flow parameters are as follows: \( Re = 48000, De = 27200, \) and \( \alpha = 70.9. \)
3. Results and Discussion

3.1 Experimental results

Figure 9 shows the velocity fields obtained from 2D2C PIV in the bend planes. In the figure, vectors and contour maps denote the in-plane velocity and the velocity magnitude, respectively. Plotting vectors are skipped at a constant interval and thus the number of vectors displayed in the figure is smaller than that of acquired data. The velocity data were smoothed by means of phase averaging and displayed on the right side of Fig. 9. The phase-averaged velocity fields are calculated over a large number of instantaneous data taken at the same crank angle (the maximum velocity phase in this study) as given by

\[
\bar{u}(\theta) = \frac{1}{N} \sum_{i=1}^{N} u(\theta, i),
\]

where \( u(\theta, i) \) is an instantaneous velocity component at a crank angle \( \theta \) for a particular engine cycle \( i \). In this study, the number of cycles for averaging \( N \) is equal to 1000 as stated above. It is clearly observed that the high-speed region appears around the inner wall side of the first bend. Sudo et al. (1998) explained the flow acceleration phenomena around the inner side in terms of the pressure distribution by measuring the pressure on the bend pipe wall for turbulent conditions, to show that the pressure becomes lower around the inner side wall. Hence, a favorable pressure gradient exists beginning at the bend entrance and continues to the midsection of the bend. Even for our conditions, the pressure becomes lower around the inner side, as seen below in the numerical results presented on the right-hand side of Fig. 15, which accelerates the flow along the inner side, which has also been observed in previous studies under relatively high Reynolds number conditions; for example, the experimental studies by Tunstall and Harvey (1968) (\( Re = 40000 \)) and Mazhar et al. (2016) (\( Re = 40000 \) and 70000). Leaving the first bend, the high-speed fluid tended to shift slightly toward
the outer side. This phenomenon has been reported in many studies on bend flows under various conditions: steady, unsteady, laminar, and turbulent flows. The shift of the high-speed region is attributed to the secondary flow arising by the effect of the centrifugal force, which is directed outward in a bend. Evidently, the high-speed flow shifted along the direction of the centrifugal force. Details concerning the secondary flow structures will be discussed later. As a result of the reversed curvature of the second bend, the high-speed region appeared around the inner side before entering the second bend. The velocity characteristics in the second bend were similar to that in the first bend in terms of the occurrence of the flow acceleration around the inner side.

Figure 10 shows the instantaneous and phase-averaged velocity fields obtained from the stereoscopic PIV in the cross section located downstream from the second bend exit. In the figure, vectors and contour maps denote the in-plane and streamwise velocity components, respectively. The picture is drawn such that the cross section is viewed from the downstream of the test section. Computing flow velocity vectors around the left \((x < -14 \text{ mm})\) and right \((x > 14 \text{ mm})\) side walls is difficult because of the wall surface reflection of the laser light. Therefore, the grid point for the image correlation was not specified in that region during the PIV analysis. The instantaneous results showed the asymmetric vortical structure due to turbulent fluctuations. The vortical structure was formed symmetrically with respect to the line of \(x = 0 \text{ mm}\) as evident from the phase averaging for the elimination of velocity fluctuations. As seen from the phase-averaged velocity field, the secondary flow direction on the line of \(x = 0 \text{ mm}\) was from the outer side to the inner side, and the flow along the lines of \(x = \pm 10 \text{ mm}\) is in the opposite direction. Thus, a pair of counter-rotating symmetric vortices was formed over a wide area of the cross section. The circulations of these vortices are in the direction of Lyne-type secondary flow, which is frequently observed under oscillatory or pulsatile conditions. However, the generation of the Lyne-type vortices in this study is not because of the pulsation, but rather a result of the change in the flow field by the presence of the first bend. The similar vortical structure was observed by Ng et al. (2008), who investigated steady turbulent flows in square-sectioned S-shaped ducts through the experiments. For steady flows in circular-sectioned S-shaped pipes, the vortices in the opposite circulating direction to the Dean-type were also observed in the second bends under both laminar and turbulent conditions (Niazmard and Jaghargh, 2010; Mazhar et al., 2016). They explained that the Dean-type vortices formed in the first bend still remain in the core of the cross section even in the second bend. However, we reason below that the vortical structure in the second bend results from the non-uniformity of the streamwise velocity distribution at the entrance of the second bend. The detailed explanation for the secondary flow formation in the second bend will be given later. The secondary flow was directed toward the inner side around the core of the cross section, and for this reason, the high-speed streamwise flow depicted with a contour map remained slightly shifted toward the inner side, as also seen from the streamwise velocity profile of Fig. 12.

To classify the turbulent characteristics of the flow, we consider the turbulent kinetic energy as calculated from the stereoscopic PIV data based on phase averaging similar to many previous studies on engine flows (for example, Druault et al., 2005; Müller et al., 2010). An instantaneous velocity component \(u(\theta, i)\) is decomposed into a phase-averaged part \(\bar{u}(\theta)\) and a fluctuating part \(u'(\theta, i)\) given by

\[
u(\theta, i) = \bar{u}(\theta) + u'(\theta, i) .\tag{10}\]

where \(u'(\theta, i)\) includes a cycle-to-cycle variation as well as a turbulent fluctuation component. The turbulent kinetic energy \(k\) is calculated from the root-mean-square (rms) velocity fluctuations \((u_{rms}, v_{rms}, w_{rms})\) and is defined as

\[
k = \frac{1}{2}\left\{u_{rms}^2 + v_{rms}^2 + w_{rms}^2\right\} , \tag{11}\]

\[
u_{rms} = \left\{\frac{1}{N} \sum_{i=1}^{N} u'(\theta, i)^2 \right\}^{1/2} .\tag{12}\]

Figure 11 shows the contour map of the turbulent kinetic energy calculated according to Eq. (11), where it is clear that
the higher $k$ region appears around the inner side. As seen in the phase-averaged flow field of Fig. 10 (b), the secondary flow impinges on the inner side wall, where the $k$ value becomes higher. The streamwise velocity is also an important factor for explaining the spatial characteristics of the velocity fluctuations, where it is observed that the streamwise velocity decreases in the vicinity of the inner side, as also seen from the velocity profile depicted as dots in Fig. 12. This gradient of the streamwise velocity thus generates the velocity fluctuations.

3.2 Numerical results
3.2.1 CFD validation

To validate the numerical simulation conducted in this study, the numerical results are compared with the above mentioned experimental results. Figure 12 shows the comparison between the results from the numerical simulation and the experimental data obtained from 2D2C and 2D3C PIV in terms of the streamwise velocity distribution downstream from the second bend exit. The data along the direction orthogonal to the pipe axis, equivalent to the line of $x = 0$ mm in Fig. 10, are plotted in Fig. 12. The numerical result is in broad agreement with the experimental data, though there is some deviation around the side walls.

The velocity field obtained through the numerical simulation for the pipe cross section at the same position as the
measured section of the 2D3C PIV is displayed in Fig. 13. As with the PIV result in Fig. 10, the Lyne-type vortices are observed in the core of the cross section. Although the center of the Lyne-type vortices in the numerical result is located at a lower region (the outer side) compared with the experimental result shown in Fig. 10, the numerical simulation does predict the large-scale vortical structure. Therefore, the numerical result will be used to discuss the secondary flow formation in the subsequent section. In the numerical results, the other vortices, which were not observed in the PIV measurements, can be recognized in the four corners of the cross section where the PIV post processing was not carried out. These vortices circulate in the direction of Dean-type secondary flow, and are consistent with the curvature of the second bend.

3.2.2 Secondary vortical structures

Figure 14 shows the velocity fields obtained through the numerical simulation at the three different phases from the results shown in Fig. 13, where each of the phases is indicated in the graph of Fig. 8. We see that the vortical structures at all the phases resemble each other in terms of the appearance of the Lyne-type vortices as observed at the maximum phase. The Lyne-type vortices persist after passing through the second bend regardless of the phase. Consequently, we focus on the maximum phase at which the experiments were conducted, which enables a more detailed description of the secondary vortical structures developing through the first and second bends of the S-shaped pipe using the results of the numerical simulation. We then provide the formation mechanism of the Lyne-type vortices in the second bend by considering the driving forces of secondary flows.
The secondary and streamwise flows in the pipe cross sections located at the entrance and exit of each bend are displayed in Fig. 15 for the maximum velocity phase indicated by the red cross in Fig. 8. At the first bend entrance, the secondary flow is completely directed toward the inner side wall in the entire cross section. This type of secondary flow is observed in the case that a straight pipe is mounted upstream from a bend entrance as reported by Bovendeerd et al. (1987), and is caused by the pressure gradient in the radial direction of curvature. As displayed on the right side of Fig. 15, the higher pressure region appears around the outer side, and for this reason, the secondary flow is accelerated toward the inner side. As the fluid enters the bend, the centrifugal force is strong enough to reverse the flow direction toward the inner side. Thus, at the first bend exit, the secondary flow direction is from the inner side to the outer side over a wide area of the cross section, while the flows close to the left and right side walls are in opposite directions. The circulation resulting from this difference in the flow direction is similar to a Dean-type vortex. At the second bend entrance, the secondary flow is directed toward the inner side due to the pressure gradient effect as in the first bend. The flow in the direction of the pressure gradient still remains at the second bend exit, and the secondary flow contains the Lyne-type vortices as observed in the PIV results (Fig. 10).

\[ F_r = \frac{W^2}{r} - \frac{1}{\rho} \frac{\partial P}{\partial r} \]  

(13)
A large value of $F_r$ implies that the secondary flow tends to be directed toward the outer side and consists of Dean-type vortices. Figure 16 shows the calculated centrifugal force, pressure gradient, and $F_r$ profiles along the radial direction in the central plane. In the both bends, the centrifugal force decreases when approaching the outer side due to the deceleration of the streamwise flow, as seen in the bottom right figure of Fig. 16. The pressure gradient profiles denote the same tendency as seen for the centrifugal force. The $F_r$ profiles in both bends exhibit peaks around the inner side (at the radial distance of approximately 8 mm) as a result of the relatively larger centrifugal force compared with the pressure gradient. The $F_r$ profile in the second bend is smaller compared with the first bend and, furthermore, becomes negative around the outer side. This small $F_r$ is not strong enough to reverse the secondary flow direction induced at the second bend entrance, and as a result, the secondary flow remains directed toward the inner side in the center of the second bend exit.

The small $F_r$ in the central plane of the second bend is attributed to the state of the streamwise velocity before entering the second bend. Figure 17 depicts the streamwise velocity as a contour map in the straight pipe between the two bends. As the fluid passes through the straight pipe, the high-speed region appearing on the inner side of the first bend shifts toward the outer side, which corresponds to the inner side of the second bend, along the Dean-type secondary flow formed in the first bend (Fig. 15). At the second bend entrance, the relative low-speed region appears in the cross section, and the non-uniform flow enters the second bend. This leads to a small $F_r$ in the low-speed region, which induces the secondary flow directed toward the inner side and the Lyne-type vortices.

![Fig. 16](image)

Centrifugal force (top left), pressure gradient (top right), and $F_r$ (bottom left) profiles in the first and second bends. Calculated values are plotted along the radial direction, the positions of which are indicated on the bottom right figure depicting the streamwise velocity distribution as well. In the graph, $F_r = 0$ is indicated as the dashed line.
4. Conclusions

The pulsatile turbulent flow in the S-shaped double bend pipe was investigated both through experimentation and numerical simulation. The 2D2C and 2D3C PIV were carried out to measure the streamwise and secondary velocities at the phase of maximum velocity. The numerical simulation with the RNG k-ε turbulence model revealed the formation mechanism of the characteristic secondary vortical structures after passing through the second bend. The main results are summarized as follows:

(1) 2D2C PIV measurements showed that the flow is accelerated on the inner side wall of the bend.
(2) 2D3C PIV measurements showed that the secondary flow at the downstream section from the second bend exit consists of Lyne-type vortices, which is not consistent with the second bend curvature.
(3) The Lyne-type vortices, which result from the secondary flow directed toward the inner side, are attributed to the fact that the Dean-type vortices formed in the first bend make the streamwise velocity non-uniform in the straight pipe before entering the second bend. This was confirmed by considering the radially directed force $F_r$, which accelerates the fluid toward the outer side in the bend.

To control flow fields after passing through a bend, streamwise flow characteristics must be considered, as the streamwise velocity condition affects the secondary vortical structure. The knowledge of the pulsatile turbulent flow in the double bend pipe, presented in this paper, serves as a design guideline for an exhaust system of an automotive engine. Exhaust flow fields become complicated under the effects of unsteadiness as well as pipe geometry. Our future work will focus on the temporal characteristics of the pulsatile turbulent flow.

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