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Research Article

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Posted Date: January 24th, 2024

DOI: https://doi.org/10.21203/rs.3.rs-3868646/v1

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Additional Declarations: No competing interests reported.
Study on the Internal Flow State and Local Resistance of a Bend under Different Operating Conditions

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Abstract
At present, to reduce the energy consumption of buildings and thereby decrease carbon emissions, the research on the internal flow mechanism of local components represented by elbows has been paid more and more attention in heating, cooling and water supply systems. In this paper, the study delves into the mechanisms behind fluid flow pattern changes in elbows, examining factors such as the causes of fluid flow pattern changes, the dynamic mechanisms of uneven pressure distribution, and mechanical properties. Comparative analyses were conducted on the changes in local resistance coefficient, center section velocity, and wall shear stress of elbows under various conditions. The results revealed that, the local resistance coefficients of bending pipes with different pipe diameters do not completely change according to the power function law at the same inlet flow velocity. Further analysis of the mechanical properties and dynamic mechanisms in elbows with these pipe diameters indicates that the observed results are attributed to various factors, including fluid velocity and wall shear stress. It is found that the result is caused by many factors such as fluid velocity and wall shear stress. With the increase of pipe diameter, the influence of the turning section on the fluid decreases, and the curvature effect of the pipe also weakens. This study offers theoretical insights for drag reduction optimization of local components in heating, cooling, and water supply pipelines.

Keywords: elbow, local resistance coefficient, wall shear stress, turbulence intensity, CFD

Graphical abstract
1. Introduction

As an important part of building operation, the energy consumption of fluid transportation systems is mainly due to the use of power equipment (pumps or fans) to overcome the resistance loss along the pipeline caused by fluid flow and the local resistance loss generated by the flow of fluid through deformation components. Among them, elbows, as commonly used components in transportation systems, account for a large proportion of the resistance losses in pipe systems, as shown in Fig. 1. As the fluid flows through the curved section of the pipe, its flow direction and flow cross section changes, causing the separation of particles and a rapid change in the flow morphology, leading to vortices. The collision, friction, and breakage of the fluid vortices consume mechanical energy [1-3]. Reducing the local resistance of the bend can effectively reduce the energy consumption of building operation and thus reduce carbon emissions [4,5]. Therefore, studying fluid flow characteristics inside a curved pipe and analyzing its local resistance has important implications for resistance-reducing optimization and energy efficiency in building operations, thereby laying the foundation for achieving carbon neutrality goals.

Fig. 1 On site schematic diagram of curved pipes

The earliest research on curved pipes can be traced back to 1927, when Dean [6,7] utilized theoretical analysis to identify and quantify secondary flows within cross-sectional curved pipe geometry.
for the first time. The capture of Dean's vortex morphology provided insights into local resistance and internal flow patterns. Subsequently, numerous scholars conducted in-depth studies on the resistance and internal flow patterns of curved pipes using various methods. Taylor [8] systematically investigated 90° rectangular section bends using a laser Doppler anemometer and mapped important experimental results such as the time-averaged velocity distributions under two flow regimes: laminar flow and turbulent flow, and the pressure distributions along the wall in the bending segment. The former Soviet scholar Idelichik [9], through experiments and theoretical analysis, discovered that water flow in curved pipe sections fluctuates greatly in both the internal and external wall pressure and velocity due to the effect of centrifugal inertia force. At the same time, he proposed that the resistance coefficient of the elbow is not only related to Reynolds number, but also related to geometric parameters of the elbow (such as bend angle, curvature radius, and inlet / outlet area ratio). This research results provided insights for system hydraulic calculations and drag reduction optimization. Based on the existing research results, K. Sudo [10] conducted a more detailed analysis of the flow characteristics of primary and secondary flow in curved pipes and further explored the transformation of longitudinal phenomena and turbulence structures in 90° bends. With the development of computers, the application of CFD showed strong advantages in the study of complex flow fields in local components. When applying curved CFD modeling and computation, Shi [11] used a standard k-ε model to obtain the average velocity u of the curved pipe, which was compared with experiments. Before the 45° section, the two were in good agreement, but there was a significant error after the 45° section. Ding [12] utilized the Reynolds-averaged Navier-Stokes equation and the RNG k-ε model to conduct numerical simulation, and obtained the calculation results consistent with the experimental data. That also focused on the discussion of different attack angles and side slip angles of incoming flow, the flow characteristics of fluid medium and the separation phenomenon of flow field, which were rarely studied in the past. It provided a reference for revealing the mechanism of complex flow phenomena inside the elbow. Fan et al. [13] conducted a numerical simulation of a rectangular cross-section curved pipe using turbulent large eddy simulation methods, focusing on the changes in secondary flow, and found that when fluid flows through the bend, two symmetric vortex zones form on the wall surface. Based on the conclusions of previous studies, scholars delved deeper into the influence of parameters on the fluid movement in curved pipes. Felipe [14] conducted experiments on curved pipes using factors such as curvature radius, pipe wall roughness, and fluid temperature, and found that the secondary flow phenomenon in the curved section led to energy dissipation. Jiang [15] and Sun [16] conducted numerical simulations using a turbulent mathematical model to analyze the flow field of water and air in a 90-degree curved pipe, analyzing the pressure distribution, secondary flow, and changes in wall pressure coefficient. At the same time, Sun [16] also studied the effect of Reynolds number on the wall pressure coefficient. The above literature indicates that most of the existing studies on curved pipes focus on the flow characteristics of fluid flowing through the curved sections. There have been occasional studies on the wall pressure coefficient, but few studies on the causes of changes in flow patterns and the dynamic mechanisms and mechanical properties of uneven pressure distribution.

The 90° deformation member of circular pipe is not only a hot topic in theoretical research, but also attracts many researchers to conduct applied research on it due to its widespread presence in engineering. In pipeline design and hydraulic calculation, the local resistance coefficient ζ is often used to measure and determine the resistance of flow through local components, and thus the acquisition of the local resistance coefficient ζ has become the focus of most applied research. However, since the source of the local resistance coefficient is mostly based on experimental values, its results are greatly affected by
experimental conditions. The corner of the round pipe is set to 90-degree, the ratio of the radius of curvature to the diameter is R/D = 1.0 (R is the radius of curvature of the bend, D is the diameter of the round pipe). In this case, the difference between the maximum and minimum values of the local resistance coefficient was as much as 4 times [17]. He et al. [17] discussed the influencing factors and rules of local resistance in curved pipes, recommending some commonly used empirical formulas for calculating the local resistance coefficient of curved pipes. Zhao [18] analyzed the relationship between fluid flow pattern and local head loss in a 90° bend pipe. Zhang et al. [19] discussed the changing rules of local resistance coefficient and roughness of the bend pipe through numerical simulation on this basis. Cao et al. [20] conducted numerical simulations on the flow of 90° elbows with different pipe diameters, established a relationship between local resistance coefficient and pipe diameter, compared and analyzed the characteristics of velocity and pressure fields at different flow rates at the 90° elbow. Wei [21] analyzed the impact of the relative length of the middle section of a planar "Z" shaped bend on the pressure loss in the pipeline.

In this paper, by combining numerical simulation with experimental research, water flow and local resistance under different working conditions are studied by changing the diameter, inlet velocity and curvature radius of the curved pipe. The change of shear stress on the wall surface is emphatically explored, and the action mechanism of water flow phenomenon in the curved pipe is analyzed from two aspects: motion characteristics and force analysis. It provides a basis for the selection and calculation of bending pipe in engineering design and lays a theoretical foundation for the optimization of bending pipe structure.

2. Theoretical analysis of fluid flow

When the fluid flows inside the pipe, the molecules in the fluid will move relatively. This creates internal friction between fluid layers to resist relative motion. This property is known as viscosity [22]. Due to the viscosity of the fluid during fluid motion, viscous force is the tangential force exerted by a portion of fluid particles on another portion of fluid particles. The direction of internal friction generated between the layers of relatively moving fluids is generally along the tangent of the flow plane. The deformation of the fluid during flow is caused by this force, hence the term "viscous force" also refers to the shear force. The ratio between the shear force per unit area and the shearing stress per unit area is called the shearing stress. The formula for the shearing stress is:

\[ \tau = \mu \frac{\partial u}{\partial n} \]  (1)

When fluid flows in a circular tube, the flow within the tube is axisymmetric. Under uniform flow conditions, the internal wall shear stress \( \tau \) of the pipe is consistent in both the axial and circumferential directions [23,24]. Taking a 90-degree bend as the study object, after fully developing, the water flow enters the curved pipe section and changes from uniform flow to non-uniform flow due to inertial forces, resulting in significant fluctuations in shear stress. This study focuses on the analysis of wall shear stress, and relies on its distribution along the wetted perimeter to conduct relevant research on shear turbulent flow. At the same time, a comparative analysis of the wall shear stress changes in different sections is conducted to systematically analyze the entire process of wall shear stress changes in different pipe diameters as a function of fluid flow direction. The wall shear stress of different sections is the wall shear force in the region of the section where the pipe length is infinitely small. The formula for shear force is:

\[ T = \int \mu \frac{\partial u}{\partial r} dA \]  (2)
3. Methodology

3.1. Calculation method of the local resistance coefficient

The study object is a 90° elbow, with a geometric model as shown in Fig. 2. Bergman [25] found that the minimum unfolding length required was 10D for turbulent flow in a bend pipe. Ye [26] simulated the fluid velocity in multiple bends, indicating that when flowing through bends 10-12D, the flow rate was nearly uniform. Since the initial velocity at the model inlet section is generally set to a uniform distribution, a certain length is required to develop sufficiently. Therefore, the length of the upstream pipelines of the elbow is set to 20D, and the length of 10D is adopted for the downstream straight section in this study.

The model selection includes fully developed upstream and downstream straight pipelines, which account for a large proportion of the total pressure loss of the component. Therefore, for the calculation of the local resistance coefficient of the bend, the frictional resistance in the component pressure loss needs to be excluded. As shown in Fig. 2, for a pipe with an elbow, the static pressure detected at the inlet and outlet are $P_1$ and $P_2$, respectively. And the static pressure difference of the pipe is $\Delta P_{1,2} = P_1 - P_2$.

At this point, the static pressure difference obtained here is partially caused by the pressure loss due to secondary flow and boundary layer separation effects in the elbow, and partially due to the frictional effects in the bend. After removing the elbow, the static pressure difference in the straight pipe section is the frictional resistance loss of the elbow $\Delta P_{1,2,f} = P_1' - P_2'$ [27,28].

Therefore, the local resistance coefficient of elbow can be written as follows:

$$\zeta = \frac{\Delta P_{j}}{P_v} = \frac{\Delta P_{1,2} - \Delta P_{1,2,f}}{P_v} = \frac{(P_1 - P_2) - (P_1' - P_2')}{\rho \left( \frac{v^2}{2} \right)}$$

where $\zeta$ is the local resistance coefficient of the elbow, which is dimensionless; $\Delta P_{j}$ is the local resistance loss; and $P_v$ is the mean dynamic pressure.

![Fig. 2 Calculation diagram for curved and straight pipe sections](image)

3.2. Full-scale experiment

As shown in Fig. 3, the experimental system is consisted of water pumps, water tanks, flexible interfaces, U-bends for testing, valves, etc. In the experimental measurement, the experimental water pipe diameter $D$ is set to $\Phi 20$, the ratio of the radius of curvature to the diameter is $R/D = 1.0$. After the constant pressure water tank overflows, start the water pump and open the valve to discharge the trapped gas in the pipe. Currently, the fluid enters the pipeline through the water tank, passes through 16 static pressure measurement points, and is discharged into the water tank through the pipeline. After the flow...
rate stabilizes, record the liquid level data of each pressure measuring pipe. The first measuring point of the U-shaped pipe starts from the inlet, passes through 4D, and reaches the measuring position. Then, pressure values are measured every 5D or 10D, and pressure values are measured at 3.4D from the inlet and outlet of the curved pipe section [28].

![Fig. 3 Local resistance test bench](image)

### 3.3. Numerical method

#### 3.1.1. Model establishment and parameter settings

In numerical simulation research, structured grids are easy to achieve boundary fitting of regions, with fast generation speed and good quality. However, it is only applicable to models with regular shapes. Unstructured grids are suitable for various complex shapes, but their disadvantage is the large number of elements and slow grid generation speed [29]. During the meshing process of the model, the curved pipe that is the subject of this study is a relatively regular geometric model, so the modeling and structured meshing of the curved pipe components are performed in the Gambit software, and local densification is applied to the curved pipe sections, as shown in Fig. 4.

![Fig. 4 Schematic diagram of grid division](image)

The fluid flowing in the bend is a single-phase incompressible liquid water with constant physical properties, so this study ignores the heat exchange of the fluid. The physical properties of water are shown in Table 1. A commercial CFD software package, ANSYS Fluent 2022, is used to calculate the flow of 90° elbows with the finite volume method. In addition, the pressure difference adopts a second-order scheme. The momentum, turbulent kinetic energy, and dissipation rate all adopt a second-order upwind scheme, with the flow regime set as steady flow [27,30]. Use the COUPLED algorithm to
determine the coupling between pressure and velocity, with a convergence accuracy of $10^{-6}$. At the same time, monitor the outlet pressure of the fluid in the bend. When the iteration steps exceed 300, the residual of each physical quantity will not change. When the number of operating steps exceeds 200, the monitoring value will not change. The number of iteration steps is set to 2000, as shown in Fig. 5.

![Fig. 5 Iterative steps and monitoring](image)

(a) Iterative steps (b) Static pressure monitoring at the outlet surface

### Table 1 Physical properties of water.

<table>
<thead>
<tr>
<th>Temperature (K)</th>
<th>Density (kg/m³)</th>
<th>Dynamic viscosity (kg/m⋅s)</th>
<th>Kinematic viscosity (m²/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>298</td>
<td>998.2</td>
<td>1.003×10⁻³</td>
<td>1.005×10⁻⁶</td>
</tr>
</tbody>
</table>

The boundary layer is set according to the literature [31]. The inlet of the bend is defined as the velocity inlet, which varies from 0.5 m/s to 3.0 m/s based on standard velocity, while the outlet is defined as the outflow. The wall boundary condition is a non-slip boundary condition, and the absolute roughness of the stainless-steel pipe is set to 0.04 mm. The turbulence intensity is estimated at 5%.

### 3.1.2. Verification of turbulence models

Before conducting CFD numerical simulation, three k-ε models (Standard k-ε model, RNG k-ε model, Realizable k-ε model) are simulated under the same initial calculation conditions to select the most appropriate turbulent flow model for curved pipes. As shown in Fig. 3, comparing the static pressure values of 16 measuring points obtained from the experiment with the data of the same position measurement points in the above model, the most consistent turbulence model is selected as the model for subsequent CFD numerical simulation. The graph in Fig. 6 shows that the turbulence model closest to experimental measurements is the Realizable k-ε model, and its pressure is within the experimental error range. Due to the high similarity between Realizable k-ε turbulence models and real turbulent flows, it can simulate free flows such as jets and mixing flows, rotating uniform shear flows, internal pipe flow, side-wall separation flows, and secondary flows [30,31]. Therefore, this study selects the Realizable k-ε model for numerical simulation. The turbulence k-ε model includes the continuity equation, momentum equation, turbulent kinetic energy, and dissipation rate transport equation of incompressible fluids, which together provide three-dimensional velocity and pressure values for the flow field particles. The model is as follows:

The continuity equation is as follows:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (4)$$

The momentum equation is as follows:

$$\rho u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} - \rho u'_i u'_j \right) \quad (5)$$
The turbulent kinetic energy is as follows:

$$\frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[ (\mu + \frac{\mu_t}{\sigma_k}) \frac{\partial k}{\partial x_j} \right] - \rho u_i u_j \frac{\partial u_i}{\partial x_j} - \rho \varepsilon \tag{6}$$

The turbulent dissipation rate is as follows:

$$\frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[ (\mu + \frac{\mu_t}{\sigma_\varepsilon}) \frac{\partial \varepsilon}{\partial x_j} \right] - C_1 \frac{\varepsilon}{k} \rho u_i u_j \frac{\partial u_i}{\partial x_j} - \rho C_2 \frac{\varepsilon^2}{k} \tag{7}$$

Among them, $\mu_t$, $C_\mu$ calculate according to the following equation:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

$$C_\mu = \frac{1}{A_0 + A_1 \frac{w}{U^*}}$$

$$A_0 = 4.0, A_1 = \sqrt{3} \cos \phi, \phi = \frac{1}{3} \cos^{-1}(\sqrt{3} \frac{w}{U^*})$$

$$W = \frac{E_{ij} E_{ij} E_{ki}}{(E_{ij} E_{ij})^{3/2}}, U^* = \sqrt{E_{ij} E_{ij} + \Omega_{ij} \Omega_{ij}}, \Omega_{ij} = \Omega_{ij} - 2 \varepsilon_{ijk} \omega_k$$

Where, $\sigma_k$, $\sigma_\varepsilon$ are the turbulent Prandtl numbers corresponding to turbulent kinetic energy and turbulent energy dissipation rate, respectively, and $C_1 = 1.44, C_2 = 1.92, \sigma_k = 1.0, \sigma_\varepsilon = 1.3$.  

Fig. 6 Static pressure comparison between experimental data and simulation results

3.3.3. Validating grid independence

Without affecting the accuracy and precision of the calculation results, to improve computational efficiency, a grid independence verification is conducted to determine the appropriate number of grids. In this article, taking a DN100, $R / D = 1.0$ elbow as an example, the mesh numbers of cells in the study are 473 k, 719.4 k, 1140.2 k, 1468.9 k, 1944.5 k, and 2236 k. Under the same conditions, the number of iterations for six models with different grid numbers is set to 2000. Under the different mesh numbers, the local resistance coefficient of the bend pipe is calculated. The static pressure points on the external wall surfaces are detected, at different angles of the bend pipe section. Once the difference in the local resistance coefficient between the two continuously refined mesh models is small, and the static pressure values on the external wall surfaces at different angles tends to stabilize. In this case, the solution is considered as the final solution. As shown in Fig. 7, the local resistance coefficient of curved pipes with
different mesh numbers is calculated. When the number of meshes is greater than or equal to 1611.2 k, the difference in local resistance coefficient is relatively small [27,28]. At the same time, the static pressure on the outer wall surface at the turning point is also measured. When the number of meshes is at least 1468.9 k, the static pressure on the outer wall surface from 0° to 90° is almost unchanged. When the number of meshes is equal to 1944.5 k, there is a slight fluctuation in the static pressure on the outer wall surface of the 90° bend, within the error range. Therefore, the number of grids selected is 1611.2 k for the following simulation.

Fig. 7 Grid independence verification

4. Results and discussion

4.1. Analysis of flow field characteristics inside elbow

In this study, the curved pipe assembly consisting of a 20D upstream straight pipe and a 10D downstream straight pipe is selected. It is not difficult to find from Fig. 8(a) that the state of fluid motion has become stable when the fluid enters the middle part of the upstream straight pipe (-X/D=10). The pressure fluctuation at this stage is minimal, with the fluid exhibiting axial symmetry within the pipe. The highest velocity occurs at the center, while the thin layer near the wall experiences the steepest velocity gradient. To understand the flow pattern of some fluids in the elbow, as shown in Section 2 of Fig. 8(a), the components consisting of the elbow and the upstream straight pipe 4D away from its entrance and the downstream straight pipe 2D away from its exit are intercepted for local detailed analysis.

Moreover, the pressure cloud map, velocity cloud map and streamline map of the local longitudinal section of the bend in section 2 are shown in Fig.s 8(b) and 8(c). As can be seen from these figures, when the fluid flows to the front 1D of the elbow, the fluid movement in the upstream straight pipe changes due to the influence of the downstream bend. The pressure on the inner wall surface of the pipe suddenly decreases at this moment, causing the mainstream flow to shift towards the inside due to centrifugal force. When the fluid enters the elbow from the straight pipe, the boundary conditions of the pipe change. Under
the action of inertia force, the mainstream separates from the inner wall surface. The flow structure, pressure distribution and velocity distribution also fluctuate greatly. The mainstream shifts toward the outer wall of the curved pipe, causing energy dissipation. The pressure changes on the inner and outer wall surfaces of the pipeline in this part are particularly prominent. The negative pressure is generated on the inner wall of the curved pipe due to the separation of fluid particles, resulting in a large velocity gradient there. Moreover, the flow velocity on the inner side of the elbow increases due to the deviation of the mainstream towards the inner side. Simultaneously, due to the influence of inertia force, a portion of the fluid scours the outer wall of the pipe, resulting in maximum pressure occurring at the apex of the elbow [19,20]. After passing through the turning section, the inertia force continues to exert its influence without dissipating, leading to a persistent scouring of the outer wall of the pipe by a substantial amount of fluid. Simultaneously, due to fluid separation, backflow occurs on the inner wall surface, resulting in evident secondary flow phenomena. The generation of eddy currents induces a reduction in the cross-sectional area of water flow and an increase in flow velocity, resulting in noticeable fluctuations in both pressure and velocity gradients. Consequently, the head loss experienced by the elbow primarily stems from momentum dissipation caused by changes in flow direction due to structural alterations, as well as energy losses arising from internal vortices [13,15,32].

**Fig. 8** Flow field characteristics (a) Cross-section selection for bend characteristics study (b) Pressure cloud image (c) Velocity and streamline diagram

The distribution of pressure and velocity streamlines at different angles across the elbow in Section 2 of Fig. 8(a) is presented in Table 2, providing a more detailed understanding of the generation and development of secondary flow. Each figure represents the outer wall surface at the top and the inner wall surface at the bottom.

When θ=0°, this corresponds to the inlet section of the curved duct where a decrease in pressure is observed near the inner wall surface. The fluid's radial velocity is directed towards the inner wall, causing a shift in maximum flow velocity towards the inner side. This phenomenon can be attributed to the significant radial pressure gradient present in this region. At the same time, when the pressure value of the outer wall increases, the fluid velocity near the outer wall begins to decrease. When θ=15°, the axial pressure gradient within the entire section of the material increases further, resulting in the presence of negative pressure within the bend. The mainstream continues to lean towards the inner wall. Due to the centrifugal force, the initial formation of secondary flow can be observed near the outer side of the bend center while simultaneously. At this point, the fluid near the outer side of the bend diffuses outward, while the upper and lower fluids diffuse inward. However, a fully developed vortex has not yet been formed. From θ= 30° to 45°, the minimum and maximum pressure is still at the apex of the inner and outer bend, and the direction of the main flow is also close to the inner side. As a result of the intensifying centrifugal force, a complete secondary flow is generated in the cross-section, and the position of the vortex is symmetric about the upper and lower at the center of the section. When θ=60°, the pressure on the outer wall surface exhibits a further increase, accompanied by an onset of axial pressure increment on the inner wall surface and a simultaneous decrease in fluid velocity. At the same time, the intensity of the secondary flow reaches the maximum. The position of the vortex absorbs the low-energy fluid in the surface layer and moves to the inner side of the bend with the main flow. Then, the intensity of the secondary flow decreases. The secondary flow forces the fluid to flow through the central region of the cross section towards the outer wall, while driving it near the inner wall. Meanwhile, the fluid located on the left and right wall surfaces flows towards the inner wall under the influence of the secondary flow.
From θ=75° to θ=90°, the total pressure change in the cross section slows down as the mainstream moves towards the center, resulting in a fluid with almost zero velocity appearing on the inner wall side. Currently, the secondary flow is still permitted to flow along the wall towards the inner wall, resulting in a reduction of the secondary flow area. When the fluid passes through the bend section and enters the downstream straight pipe section, at -Y/D=0.2, the pressure change of the whole section is also severe, but it is much gentler than that at θ=30°. At this point, the maximum flow rate is shifting towards the outer edge of the curve, whereas the lower flow rate area within the curve is initiating an upward expansion, accompanied by a flow line divergence around the perimeter. Concurrently, due to the influence of inertia force, the mainstream flow undergoes redirection towards the outer wall surface upon traversing the curved section of the pipe. This results in a substantial fluid sweep along the exterior of the pipe, leading to backflow on the inner wall surface of the elbow and generating two pairs of intensified secondary flows on the adherent wall surface. The strength of these secondary flows is enhanced, aligning with previous observations reported in literature [16,32,33].

Table 2 Cloud Chart of Flow Field Characteristics at Different Angles of Bend Section

<table>
<thead>
<tr>
<th>Pressure</th>
<th>0°</th>
<th>15°</th>
<th>30°</th>
<th>45°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure</th>
<th>60°</th>
<th>75°</th>
<th>90°</th>
<th>-Y/D=0.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

4.2. Relationship between inlet velocity, pipe diameter, curvature radius and local resistance
Previous studies [9,17,19] have shown that the local resistance coefficient of 90° curved pipe mainly includes three factors: the Reynolds number Re, the ratio of curvature radius to pipe diameter R/D, and the relative roughness of pipe wall Δ/D. This study observes the changes in the local resistance coefficient of the bend while changing the inlet flow velocity, diameter, or curvature radius of the pipeline. In the final analysis, this research is still to study the influence of the Reynolds number Re and the ratio of curvature radius to pipe diameter R/D on the local resistance coefficient. Zhang et al. (2010) divided the relationship curve between local resistance coefficient and Reynolds number under different relative roughness according to the partition law of resistance coefficient along the path. It was concluded that the pipe was a smooth pipe when \( Re < \frac{60}{\Delta/D} \). When the pipeline was a smooth pipe, the local resistance coefficient \( \zeta \) was not related to relative roughness \( \Delta/D \), but only to Reynolds number Re. The local resistance coefficient decreased exponentially with the increase of Reynolds number Re, and its functional relationship was: \( \zeta = \frac{0.61}{Re^{0.12}} \). The selected pipeline for this study is a smooth pipe, so the influence of roughness on local resistance coefficient is not considered. Through model tests, He et al. determined that the Reynolds number no longer had an impact on the alteration of the local resistance coefficient when the inlet Reynolds number \( Re \geq 6 \times 10^5 \). The transportation parameters and geometric parameters are altered this time, resulting in a Reynolds number range of \( 1.6 \times 10^4 \) to \( 20 \times 10^4 \) at the inlet. Additionally, the Reynolds number remains a key factor influencing the local resistance coefficient.

As shown in Fig. 9(a), the variation of local resistance coefficient is related to both velocity and curvature radius, which is consistent with the results of the correlation between local resistance coefficient and curvature radius parameters proposed by He et al. [17] through model tests. Before entering the region of quadratic resistance law, the larger the velocity, the smaller the local resistance coefficient. Similarly, the larger the R/D, the smaller the local resistance coefficient of the bending member, and the smaller the influence of the deformation member on the fluid flow structure. Most of the existing studies are focused on the change of local resistance coefficient under the condition of large pipe diameter and high Reynolds number. But there is still a lack of research direction on small-size components. The simulated water pipe inlet velocity is set to 1.1 m/s, the ratio of the radius of curvature to the diameter is R/D = 1.0. The local resistance coefficients corresponding to different bending sizes were studied, as shown in Fig. 9(b). In this study, the curved pipes from DN15 to DN125 are examined, while the Reynolds number ranges from \( 1.6 \times 10^4 \) to \( 16 \times 10^4 \). Theoretically, the local resistance coefficient \( \zeta \) should decrease as a power function with the increase of pipe diameter. Despite the general adherence to the theoretical laws, the value of the local resistance coefficient \( \zeta \) obtains from the bending pipe of DN40, DN50, and DN65 in Fig. 9(b) shows no significant difference. This anomaly is also observed in the bends of DN100 and DN125. Hence, the bending velocity gradient and wall shear stress analysis is concentrated on the five pipe diameters of DN15, DN40, DN65, DN100 and DN125.
4.3. Vertical distribution of velocity and turbulence intensity

As described in Sec. 4.1, the fluid fluctuation causes by the separation phenomenon forms a very complex flow structure in the bend section and the downstream straight section. Figs 10(a)-10(c) show the complex fluid movement in the curved pipes. Fig. 10(a) shows a number of different sections for flow visualization, which the complex flow in the curved pipes can be visualized more intuitively. As shown in Fig. 10(b), the inlet velocity of the curved pipe is set to \( v = 1.1 \text{m/s} \), and \( R/D = 1.0 \) to study the velocity vector field of the longitudinal profile. Obviously, the velocity changes of the fluid in the turning section are the same as the research results in Table II. At the 0° section, the mainstream velocity has undergone obvious distortion. At the same time, the mainstream overall shifts inward, with the maximum velocity occurring inside the pipe. This change trend continues until the 30° section. After this, due to the action of inertial force, a large amount of fluid rushes to the outer wall, so that the outer speed of the bend begins to increase and the inner speed decreases. There is an obvious low speed zone within about 25% of the inner part of the 90° section. After entering the downstream straight pipe, the fluid moves faster near the outside of the pipe, and a very low speed area appears on the inside of the bend due to the separation of flow particles. The reverse pressure gradient inside the pipeline causes a sharp decrease in the velocity of this part of the fluid, which made the downstream flow at the outlet very unstable and complex, and clearly appeared secondary flow [11,33].

To further investigate the impact of local component structural changes on fluid dynamics, the fluid velocity at the various angles in central regions of the elbow is captured and analyzed in detail, and compares with the velocity change in the central section when the upstream straight pipe is fully developed (- X/D=10), as shown in Fig. 10(c). Fig. 10(c) shows the mean velocity profile normalized with inlet velocity for different pipe diameters. The negative and positive \( r_i/r \) values represent the positions inside and outside the center section of the round pipe section, respectively.

This study is found that the fluid is axisymmetric flow when the state of fluid motion has become stable (- X/D=10). The fluid near the wall is affected by the wall and has a large velocity gradient. The fluid at the center is free from the influence of the boundary layer, with a zero-velocity gradient and the maximum velocity. It is evident that, for a given level of roughness, the maximum velocity of the center of fully developed fluid gradually decreases with increasing diameter. Additionally, the influence range of the boundary layer on fluid velocity weakens as well, indicating a diminishing effect of roughness on wall velocity with larger diameters. From the \( \theta = 0^\circ \), the position where the maximum velocity occurs shifts to the inside of the pipe, and the distance of inward migration gradually increases with the increase of pipe diameter. This means that with the increase of pipe diameter, the influence range of the elbow on the changes of the fluid movement in the straight pipe before entering the region increases. From the \( \theta = 0^\circ \) to \( \theta = 45^\circ \), the change of fluid velocity is not obvious, and the mainstream is biased to the inside of the bend. From the \( \theta = 60^\circ \), the low flow rate area first appears inside the bend of the small diameter, and the fluid in the bend of the large diameter does not fluctuate much except the deviation of the main flow position. From the \( \theta = 75^\circ \), the low flow rate area begins to appear in the inner side of the large diameter bend, and the range of this area increases gradually as the diameter decreases. From the \( \theta = 90^\circ \), the low flow velocity area inside the bend expands further, and the ratio of the low flow velocity area is opposite to the change of pipe diameter. Therefore, it can be inferred that the influence of the deformation area of the local component on the change of fluid motion becomes smaller, with the increases of the pipe
diameters. At the same time, the fluid will recover its fully developed shape through the deceleration and acceleration of the outer and inner sides of the bend [32,34,35].

Fig. 10 Complex flow structure in a bend (a) Flow section studied (b) Velocity vector field of the center section (c) Velocity curves at different positions in a bend

On the premise of consistent inlet velocity, the vortex intensity of the fluid in the elbow increases, with strong turbulent pulsation, and the mechanical energy consumed also increases. The turbulence intensity [38,39] vertically distributed at different positions in the elbow is analyzed based on Fig. 10(a), as shown in Fig. 11. As can be seen from Fig. 11(a), when the fluid is fully developed in the upstream straight pipe segment (-X/D=10), the turbulence intensity of the fluid is symmetrically distributed around the center. At this time, the flow field is disturbed by the pipe wall, and momentum exchange occurs near the boundary layer, resulting in turbulent kinetic energy. However, because the fluid is fully developed inside the pipe, there is no high average shear layer, and the turbulence intensity is almost 0. The turbulent energy at the boundary layer generally decreases with the increase of the pipe diameter, and the influence of the fluid on the boundary layer also decreases with the increase of the pipe diameter. When the fluid enters the turning section of the elbow, the area of momentum exchange between fluids expands under the influence of deformed components. In the turning section of 0-30°, the mainstream is affected by centrifugal force and shifts to the inner wall as a whole, and the turbulence intensity reaches its peak at the inner wall surface. In the turning section of 30°-90°, inertial force begins to dominate, and the mainstream in the elbow moves to the outer wall, and the turbulence intensity reaches its peak near the outer wall surface. Since the turning section of 60°, turbulence can generate new fluctuations near the inner wall of the elbow, as shown in Section 4.1. At this time, the second eddy current appears near the inner wall surface of the elbow. After the fluid leaves the turning section and enters the downstream
straight pipe, the influence of inertial force further expands. Meanwhile, the second eddy current gradually expands with the appearance of separation phenomenon, and the turbulent motion on the inner side of the elbow gradually strengthens, as shown in Fig. 11(b). With the increase of pipe diameter, the influence of the pipe wall on the flow field decreases, the turbulent motion gradually weakens, and the mechanical energy consumed relatively decreases.

![Diagram of turbulence intensity](image)

(a) Turbulence intensity at fully developed and curved pipe sections

(b) Turbulence intensity in straight pipe after leaving the turning section

**Fig. 11** Vertical distribution of turbulence intensity at different positions inside the elbow

4.4. Wall shear stress analysis

In recent years, some scholars [12,14] have studied and analyzed the flow characteristics and energy loss inside the bend pipe through the wall pressure coefficient. However, the research on the dynamic mechanism of the bend resistance, such as the wall shear stress, is still in a blank area of study. Therefore,
the standard wall function combined with Realizable k-ε turbulence model is used in this study to solve the wall shear stress [36,37] of each node in the wall region. The monitoring point is shown in Fig. 12(a). The inner wall vertex of the bend center is the center point, and the distance from the bend wall surface to the center point is defined as $\chi_w$. The counterclockwise is positive, and the clockwise is negative. Wet circumference is $\chi$. The $\chi_w / \chi$ is called relative wall distance. Based on Fig. 12(a), the wall shear stress of different sections at the fully developed section (-X/D=10) and the elbow sections (0°-90°) is analyzed, as shown in Figs. 12(b) to 12(f). Under the different pipe diameters, the wall shear stress is symmetrically distributed along the middle vertical line. The wall shear stress changes according to the different truncated circular surface, and its value is greatly affected by pipe diameter. In the analysis of center surface velocity changes in Section 4.3, the mainstream at the 0° section has a slight inward shift trend, and the overall change is not obvious. However, there is a significant fluctuation in the wall shear stress of the bend at the 0° section. At this point, the inner wall shear stress value of the bend is higher than the fixed value of the shear stress measured at full development, and the outer wall shear stress value is lower than this fixed value. At the 45°-60° sections, the wall shear stress exhibits opposite changes, with the value on the inner side of the wall being lower than the fixed value and the value on the outer side of the wall being higher than the fixed value.

**Fig. 12** (a) Measurement points for the wall shear stress, (b)-(f) The wall shear stress of different sections.
From Fig. 12, the wall shear stress of the curved pipe circular section is symmetrically distributed about the inner and outer wall vertices ($\chi_w/\chi=0, 1/2$). Therefore, only the numerical values of the wall shear stress changes along the fluid flow at these five points (0-1/2) will be studied later shown in Fig. 13. From Fig. 13, the wall shear stress is uniformly distributed in the upstream straight pipe, and its value is relatively small. A small portion of the upstream straight pipe segment is cut at $-X/D=10$. After verification, it is found that the wall shear force at the cut pipe is equal to the pressure loss of the straight pipe.

As shown in Figs. 13(a) to 13(e), before entering the elbow, the distribution of wall shear stress inside the pipe begins to fluctuate at a distance of 0.6D-1.0D from the elbow inlet. In the distance before entering the elbow, a large pressure gradient begins to appear near the inner wall, which increases the velocity of this part and makes the mainstream wholly shift to the inner side. Before entering the elbow, the wall shear stress fluctuation becomes more obvious with the increase of the velocity. Currently, the maximum value of the wall shear stress at this position is generated on the inner wall and gradually decreases along the inner wall towards the outer wall. After entering the bend, as shown in Fig. 14, the average wall shear stress of the section is still increasing gradually, but its distribution at different locations of the same circular section is more uneven. Since then, the difference of wall shear stress in different pipe diameters is not limited to the numerical value, but also the wall shear stress fluctuation in different positions of the same section has obvious difference. Affected by the deformation of the component, the position of the maximum wall shear stress in cross-section is shifted. At the initial stage of entering the elbow, the wall shear stress on the inner side of the elbow ($\chi_w/\chi=0$) is relatively large, and its change is basically consistent with the variation law of the mainstream, which also indicates that the flow pattern and velocity distribution of water flow have a direct impact influence on the wall shear stress [33]. But for different pipe diameters, the positions corresponding to this stage are also completely different.

At the 30° section of DN15 and DN40 elbows, the wall shear stress at the top of the inner wall is lower than the value of the wall shear stress in the middle of the inner wall ($\chi_w/\chi=1/8$), which means that the mainstream is affected by the inertial force and has begun to shift toward the outer wall. However, for DN65, DN100, and DN125 elbows, the influence of centrifugal force on the fluid is still dominant, and the shear stress on the convex side wall of the bend is still the maximum section. In the subsequent elbow, the shear stress of the inner wall surface continues to decrease and becomes the minimum value of the section at a certain moment. At the same time, the wall shear stress of the center and the outer shows an upward trend. At the exit of the bend ($\theta=90^\circ$), the wall shear stress at the concave side ($\chi_w/\chi=1/2$) of the small diameter elbow has exceeded the value at the inner middle of the bend. For the bends of DN100 and DN125, the phenomenon that the wall shear stress on the concave side ($\chi_w/\chi=1/2$) of the bend exceeds the inner middle position of the bend has to be observed at the downstream straight pipe section. After the water flow enters the downstream straight pipe, a large amount of fluid scour the outside of the pipe wall under the further action of the inertial force, and the shear stress value of the wall near it reaches the maximum. Therefore, the fluid particles are separated, and secondary flow occurs in the inner side of the bend, where the wall shear stress has a large fluctuation. Then, the flow velocity distribution tends to be uniform and the shear stress on the wall decreases accordingly by adjusting the acceleration and deceleration of the flow on the inner and outer walls in the downstream straight pipe section. As the research continues, it is not difficult to find that with the increase of pipe diameter, the change of wall shear stress appears to delay, and it can be inferred that the curvature effect of bending...
pipe is inversely proportional to the size of pipe diameter. At the same time, this is like the result obtained from the velocity normalization analysis at the bend center in the previous section. Then, the water flow in the downstream straight pipe continuously adjusts the acceleration and deceleration of the flow on the internal and external walls, making the flow velocity distribution more uniform and the wall shear stress correspondingly reduced. As the research continues, it is not difficult to find that, with the increase of pipe diameter, the overall change in wall shear stress is delayed. And it can be inferred that the curvature effect of bending pipe is inversely proportional to the size of pipe diameter. Meanwhile, this is similar to the results obtained from the velocity normalization analysis at the center of the bend in the Sec. 4.3.
The CFD-Post post-processing software is used to read the wall shear stress of different sections, and the data of the average wall shear stress in Fig. 14 is obtained. As can be seen from the Fig. 14, the larger the pipe diameter, the smaller the value and the gentler the curve, which more intuitively verifies the above reasoning. However, this change law does not apply to the bending of DN100 and DN125. The average wall shear stress values of the two specifications are almost the same, and in the range of $\theta=45^\circ$ to $-Y/D=0.4$, the bending wall shear stress value of DN125 once exceeded that of DN100. Similarly, at $-Y/D=0.8$, the curves of DN40 and DN65 with respect to the average wall shear stress of the section have a meeting point, and then the ground bend of DN65 achieves a reverse overpass. These differences are one of the reasons for the abnormal change of the local resistance coefficient of these bent pipes.
5. Conclusions

In this study, the accuracy of Realizable k-ε model of 90° bend pipe is verified by combining experimental and numerical simulation methods. The flow state of turbulence in the bend is subsequently analyzed through velocity and pressure, with calculations made for changes in local resistance coefficient under varying transport and geometric parameters. At the same time, the velocity fluctuation of the central surface and the distribution of wall shear stress under steady flow are further studied. The following conclusions are drawn:

(1) Numerical simulation is conducted when only the pipe diameter is changed, and the local resistance coefficient of the pipe is calculated. It is not difficult to find that the overall local resistance coefficient decreases with the increase of pipe diameter, but the local resistance coefficient in the two intervals of DN40-DN65 and DN100-DN125 does not decrease. The movement of the internal flow state of the pipe and its force mechanism are analyzed. It is found that the occurrence of this situation is the result of the multi-party effect of the radial velocity fluctuation of the fluid and the wall shear stress.

(2) With the increase in pipe diameter, the relative area of the low-speed zone at the outlet of the bend section is continuously decreasing, and the overall value of the pipe wall shear stress is also decreasing. The larger the diameter of the pipe, the smaller the relative wet circumference, and the smaller the curvature effect of the pipe. But with the change of pipe diameter, there is a significant difference in the center velocity and wall shear stress at the same position. For small diameter bends, at the 30° of the elbow, the inertia force has become dominant, causing the mainstream begins to shift upwards. Currently, large diameter bends are still dominated by centrifugal force, and the shear stress value on the inner wall is still the highest. As the diameter of the pipe increases, there is a delayed effect on the force acting on the curved pipe section.

(3) The presence of the component’s deformation zone leads to change in the wall shear stress and velocity prior to the fluid’s entry into the transition section. However, variations in the pipe diameter exert minimal influence on the wall shear stress of the fluid prior to its entry into the curved segment. When the inlet velocity is v=1.1m/s, the fluid begins to fluctuate at 0.6~1.0D before entering the elbow. with these fluctuations emerging from a distance of 0.6D to 1.0D from the entrance into the curved section. It is hypothesized that the influence of the deformed component on the straight segment does not experience substantial variations with the expansion of the pipe diameter.
The above conclusions have certain reference value for the current research. In order to better explain why the local resistance coefficients of DN40-DN65 and DN100-DN125 are constant, additional studies on the viscous dissipation and separation region of the bend pipe are needed, which will be left for the future work.

Acknowledgements This work was supported by Applied Basic Research Programs of Shanxi (NO.202103021224091), and the Applied Basic Research Programs of Shanxi Province (Grant No. 202203021212246). The authors would like to thank all the students and staff who were involved in data collection.

Author contributions Jie Wang: Writing – original draft, Writing – review & editing, Methodology, Validation, Investigation, Formal analysis. Chongfang Song: Conceptualization, Writing – review & editing, Supervision, Formal analysis, Project administration, Funding acquisition. Wuxuan Pan: Conceptualization, Supervision, Writing – review & editing. Yajing Yan: Conceptualization, Software. Ke Zhao: Formal analysis, Software.

Availability of data and materials The authors declare that all of the material is owned by them.

Compliance with Ethical Standards
Conflict of interest No competing interests.
Research involving Human Participants and/or Animals Not applicable.
Consent for publication All authors gave their final approval for publication.

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