

Article

Pulsation Analysis of Hose Pumps with Different Roller Counts Based on Two-Way FSI

Mengfan Wang ¹, Lixin Zhang ^{1,2,3,*}, Wenchun Li ^{4,*}, Xue Hu ^{2,3}, Jiawei Zhao ^{2,3}  and Xingzi Fang ⁵

¹ College of Mechanical and Electrical Engineering, Shihezi University, Shihezi 832003, China; wangmengfan@stu.shzu.edu.cn

² School of Energy and Materials, Shihezi University, Shihezi 832003, China; huxue@foxmail.com (X.H.); zhaojiawei@shzu.edu.cn (J.Z.)

³ Institute of Bingtuan Energy Development Research, Shihezi University, Shihezi 832003, China

⁴ Xinjiang Zhuanggeer Yinfen Agricultural Machinery Equipment Co., Ltd., Wujiaqu 831300, China

⁵ College of General Studies, Three Gorges Tourism Polytechnic College, Yichang 443111, China; f34936428670@163.com

* Correspondence: zhanglixinemail@163.com (L.Z.); liwenchun2024@163.com (W.L.)

Abstract: Xinjiang, as a major agricultural region, offers extensive application potential for hose pumps, given their excellent performance as fertilization devices. Analyzing the pulsation characteristics of hose pumps during operation is valuable for reducing noise and extending pump service life. To investigate the pulsation characteristics and unsteady flow of hose pumps with different roller numbers, this study adopts a bidirectional fluid-structure interaction (FSI) method and utilizes ANSYS 19.0 commercial finite element software to analyze the outlet and inlet pressure pulsations, outlet flow velocity pulsations, and the distribution of the flow field at the intermediate plane for both two-roller and three-roller pumps transporting high and low Reynolds number fluids under the same working conditions. It was observed that the three-roller pump exhibited higher outlet pressure compared with the inlet and that the pulsation intensity was lower in the three-roller pump than in the two-roller pump under the same conditions, with an analysis provided on the reasons for this phenomenon. This study offers theoretical support for the selection and further optimization of hose pump designs. To further reduce the negative effects of pulsations, it is recommended to increase the number of rollers in the design while also considering shape optimization of the pump casing or using feedback control systems to adjust and reduce pulsation intensity.



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Keywords: hose pump; different roller counts; pulsation; two-way FSI; unsteady flow

1. Introduction

Xinjiang is a major agricultural region, with a total grain output of 21.192 million tons in 2023, ranking first in China for total output growth. Large-scale farmland cultivation requires substantial amounts of fertilizer, and liquid fertilizer, as a modern type of fertilizer, has been widely adopted due to its high efficiency, low cost, and ease of application. Hose pumps, as a critical component of fertilization equipment, have significant potential for further application owing to their corrosion resistance and ability to transport highly viscous fluids. A hose pump mainly consists of key components such as the pump housing, hose, compression rollers, gearbox, and motor [1]. Its operating principle involves the motor driving the rotor, which is composed of compression rollers. These rollers exert pressure on the hose, compressing the material inside and forcing it toward the outlet. When the rollers disengage from the hose, the hose rapidly returns to its original shape

due to its elasticity, creating a vacuum that draws external fluid into the hose, thereby completing the suction and discharge process [2]. While this operating principle imparts excellent performance, it also causes periodic fluctuations in the fluid parameters inside the hose, a phenomenon known as “pulsation” [3]. The presence of “pulsation” in hose pumps leads to significant noise and vibration during operation, and in severe cases, it can even cause tearing at the pipe outlet, affecting the pump’s service life. The primary factor influencing this is the pulsation intensity. Establishing a fluid–structure interaction (FSI) model to study pulsation is a common approach [4,5]. Xiwei Pan et al. used the FSI simulation method to predict the dynamic characteristics of diaphragm pump port valves [6]; Xiaohui Duan et al. used FSI to study the time-domain and frequency-domain characteristics of transient hydraulic and radial displacement of axial flow pump units [7]. Most studies on the pulsation characteristics of hose pumps focus on analyzing the pumps under different operating conditions [1,3,8], lacking comparative analyses of hose pumps with different structures. In this study, ANSYS 19.0 software’s iterative coupling method is used for bidirectional FSI calculations, providing the first pulsation characteristic analysis of two hose pumps with different roller numbers. Additionally, the pulsation characteristics when transporting fluids with different Reynolds numbers are investigated. This research lays the foundation for better understanding the pulsation characteristics of hose pumps, enabling the rational selection of the number of rollers and providing a basis for future optimization designs.

As noted above, analyzing the flow field characteristics of hose pumps requires the use of FSI methods. FSI, as a well-established technique, has been widely applied across various domains [9–11], although it still faces certain challenges, such as long computation times and low accuracy. Jiawei Hu et al. proposed a fast FSI method based on a reduced-order model combining convolutional autoencoders (CAEs) and long short-term memory (LSTM) neural networks, which significantly reduces simulation time [12]. Zhijian Wu et al. enhanced the immersed boundary-lattice Boltzmann method (IB-LBM) by introducing an FSI method that effectively prevents particle penetration during the simulation, thereby improving computational accuracy [13]. Sun Peng-Nan et al. enhanced an FSI-SPH model to three dimensions using multi-resolution delta(+)-SPH and total Lagrangian SPH, introducing a benchmark for simulating viscous free-surface flows [14]. Yen Ting Ng et al. developed a two- and three-dimensional FSI method that accurately models fluid flow in irregular domains by discretizing the Navier–Stokes equations [15]. In this paper, ANSYS software is used to establish the FSI model; the two-way FSI method in ANSYS enables precise simulation of dynamic interactions between fluids and solids by bidirectionally exchanging forces and displacements between the fluid and structural solvers, making it suitable for high-precision, strongly coupled multiphysics simulations.

The rubber hoses used in hose pumps are hyperelastic materials, and selecting an appropriate constitutive model for the solid material is critical for improving the accuracy of FSI simulations. Majid Shahzad et al. characterized hyperelastic materials and identified a suitable strain energy function (SEF) for a specific type of rubber [16]. Yifeng Dong et al. developed a hyper-viscoelastic model to characterize the cyclic stress softening effect in rubber composites under arbitrary loading–unloading sequences with fixed strain amplitude [17]. Nurul Hassan Shah et al. introduced a new three-parameter hyperelastic strain energy function for soft and rubber-like materials, establishing a new constitutive model and validating its accuracy [18]. In this study, the widely used Mooney–Rivlin model was chosen for rubber material simulation, as it is well suited for these types of materials [19,20].

This study uses the bidirectional FSI method to compare and analyze the pulsation characteristics of two hose pumps with different roller numbers, used by our research

team, when transporting high and low Reynolds number fluids. The goal is to provide a deeper understanding of the pulsation behavior of hose pumps, to assist in the selection of appropriate pump types, and to lay the foundation for future optimization of hose pump designs. The specific content of this paper is as follows: Section 2 uses ANSYS software to establish two-way FSI models for hose pumps with different roller configurations. Section 3 presents the experimental setup to validate the accuracy of the simulation models. Section 4 analyzes the pulsation and flow field characteristics of the hose pumps. Section 5 concludes the study.

2. Establishment of the Two-Way FSI Model for the Hose Pump

2.1. Geometric Model and Finite Element Model

The research team primarily uses two types of hose pumps: two-roller and three-roller, shown in Figures 1 and 2. To investigate the pulsation characteristics of hose pumps with different roller counts, a three-dimensional bidirectional FSI model of the hose pump was established. The bidirectional FSI models of both types of hose pumps consist of three main parts: the casing, the compression rollers, and the rubber hose. Model parameters of the hose pumps are listed in Table 1. An eight-part computational domain was developed for numerical simulation, comprising the inlet, outlet, compression rollers, casing, rubber hose, contact surfaces between the casing and hose, contact surfaces between the rollers and hose, and the fluid–structure coupling surfaces, as shown in Figures 3 and 4.



Figure 1. Physical image of the two-roller hose pump.



Figure 2. Physical image of the three-roller hose pump.

Table 1. Material parameters of the hose.

Power Frequency (Hz)	Number of Rollers	Roller Diameter (mm)	Rotor Radius (mm)	Hose Inner Diameter (mm)	Hose Outer Diameter (mm)
65	2/3	69	121	25	29

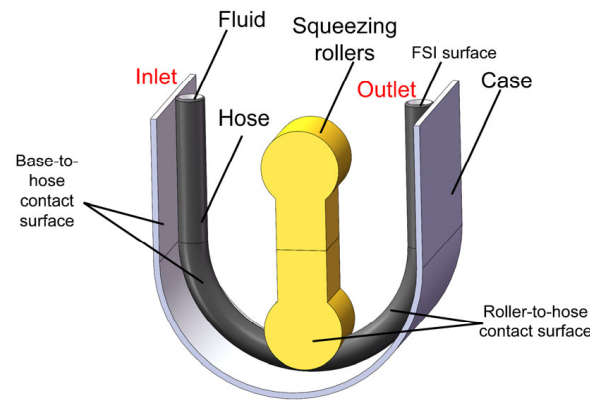


Figure 3. Three-dimensional finite element model of the two-roller hose pump.

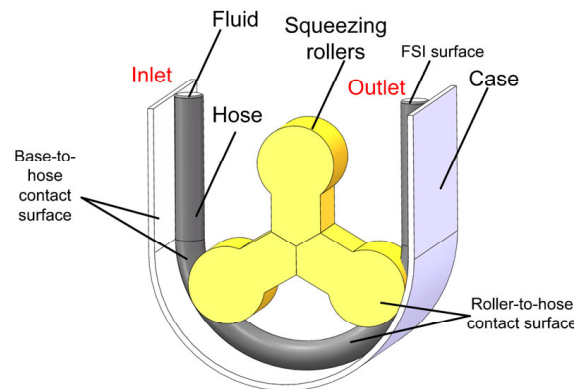


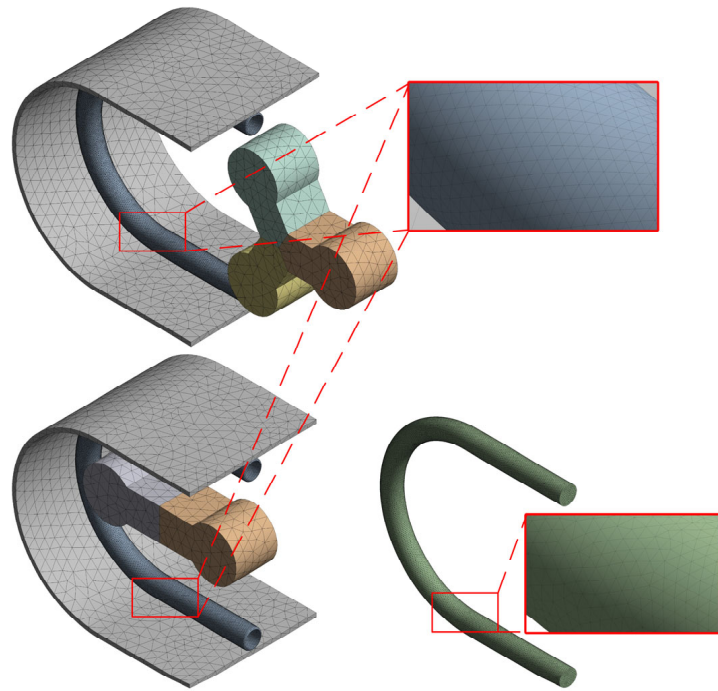
Figure 4. Three-dimensional finite element model of the three-roller hose pump.

2.2. Establishment of the Two-Way FSI Model

This study employs a two-way FSI approach, which refers to bidirectional data exchange; that is, fluid analysis results are transferred to solid structure analysis, while results from the solid structure analysis (such as displacement, velocity, and acceleration) are fed back to the fluid analysis. Such analyses are commonly used when the density ratios between fluid and solid media are relatively similar or in situations involving high speeds and pressures where solid deformation is significant and has a pronounced effect on fluid flow. This design utilizes the ANSYS Workbench platform, employing both the Transient Structural and Fluent modules to facilitate data transfer through the fluid–structure interaction surface, thereby achieving two-way FSI.

2.3. Mesh Generation and Mesh Independence Analysis

The extrusion of rubber hoses in peristaltic pumps is a highly complex nonlinear problem involving geometric, material, and contact nonlinearities. Achieving convergence in the simulation is challenging due to the significant deformations and interactions between the rollers and the hose. To address this, the mesh for the fluid and solid components is separated, with fine mesh sizes of 2 mm applied to the hose and fluid regions, ensuring accurate resolution of these critical areas. In contrast, a coarser mesh of 15 mm is used for less critical parts. High-order tetrahedral elements (Solid187) are chosen to balance computational efficiency and precision. The overall mesh distribution is shown in Figure 5. Additionally, dynamic mesh techniques such as smoothing and remeshing are implemented to handle the evolving fluid boundaries and shapes throughout the simulation.



(a) Solid Domain Mesh Distribution (b) Fluid Domain Mesh Distribution

Figure 5. Overall mesh distribution of two types of hose pumps.

For the two-roller pump, the total number of mesh elements in the fluid domain is set to 3.45×10^5 , while the solid domain has 1.13×10^5 elements. For the three-roller pump, the fluid domain maintains the same total mesh count as the two-roller pump, and the solid domain mesh count is set to 1.16×10^5 . In the FSI simulation, the time steps for the fluid and solid domains do not necessarily need to be identical. For the two-roller pump, the time step for the fluid domain is set to 0.002 s, while the solid domain uses 0.001 s. For the three-roller pump, the time steps for both domains are set equally at 0.002 s.

To ensure mesh independence, a mesh sensitivity analysis was conducted. Using the two-roller pump as an example, a monitoring point was placed at the outlet (0.044, −0.113, −0.090), with flow velocity used as the reference parameter for assessing mesh independence. The analysis compared mesh counts of 1.03×10^5 , 1.75×10^5 , 3.45×10^5 , 8.05×10^5 , and 2.71×10^6 in the fluid domain. The results showed that when the mesh count exceeded 3.45×10^5 , the flow velocity at the monitoring point stabilized, as shown in Figure 6.

Additionally, the time step sensitivity was analyzed under this mesh resolution. The original settings were confirmed to be optimal: increasing the time step caused negative mesh volumes, while reducing the time step significantly increased computational time. Therefore, the final mesh size and time step settings were determined to satisfy mesh independence requirements. This indicates that the chosen mesh division and time step settings achieve an effective balance between computational accuracy and efficiency.

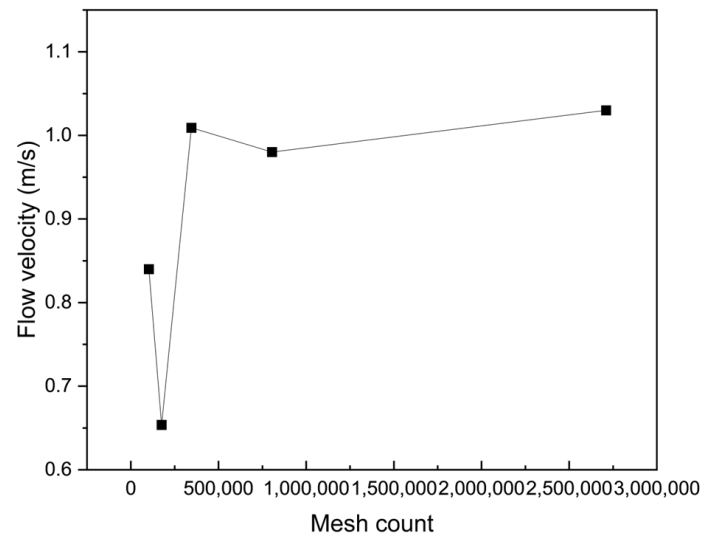


Figure 6. Variation of flow velocity at monitoring points with mesh size.

2.4. Simulation Materials and Physical Model Selection

Since the hose pump used in this study is primarily intended for field fertilization, water and liquid fertilizer are selected as the fluid materials. The Reynolds number of water within the pipe ranges from 6000 to 15,000, while the Reynolds number of liquid fertilizer ranges from 10 to 30. The parameters for the two fluids at 25 °C are shown in Table 2.

Table 2. Fluid parameters used.

	Density (kg/m ³)	Dynamic Viscosity (kg/(m·s))
Water	998.2	0.001
Liquid Fertilizer	1300	0.7

These two fluids represent high Reynolds number and low Reynolds number flows, respectively. By performing a comparative analysis using these fluids, the study provides better practical guidance and demonstrates significant real-world applicability.

During the FSI analysis, the deformation and rebound of the hose due to extrusion create suction and pressure zones, inducing fluid flow within the hose. To accurately simulate the actual fluid flow in the hose, the realizable k - ϵ turbulence model is employed to model the fluid dynamics within the hose [21]. The realizable k - ϵ turbulence model consists of two equations: the k equation, which describes the transfer of turbulent kinetic energy, and the ϵ equation, which describes the transfer of turbulent dissipation rate. Standard wall functions were also employed for wall settings.

The turbulence model is described by the transport equations of turbulent kinetic energy and its dissipation rate [22].

$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \epsilon - Y_M \quad (1)$$

$$\rho \frac{D\epsilon}{Dt} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} \right] + \rho C_1 S \epsilon - \rho C_2 \frac{\epsilon^2}{k + \sqrt{\nu \epsilon}} + C_{1\epsilon} \frac{\epsilon}{k} C_{3\epsilon} G_b \quad (2)$$

where turbulent viscosity factor $\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$.

$$C_1 = \max \left[0.43, \frac{\eta}{\eta + 5} \right], \eta = S \frac{k}{\varepsilon}, S = \sqrt{2S_{ij}S_{ij}}, S_{ij} = \frac{1}{2} \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right)$$

During the extrusion process, the hose's volume remains nearly constant, so it is categorized as a hyperelastic material. Its mechanical behavior is described using an appropriate strain energy function. The rubber hose is modeled using the commonly applied two-parameter Mooney–Rivlin hyperelastic model [20].

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) \quad (3)$$

where $\begin{cases} I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \\ I_2 = 1/\lambda_1^2 + 1/\lambda_2^2 + 1/\lambda_3^2 \end{cases}$, $\lambda_i (i = 1, 2, 3)$ is the elongation ratio in the i direction.

C_{10} and C_{01} are the material constants in the Mooney–Rivlin model, which can be obtained through uniaxial tensile tests, as shown in Table 3.

Table 3. Hose material parameters.

Density (kg/m ³)	Material Constant C_{10} (Pa)	Material Constant C_{01} (Pa)
1100	5.2×10^7	1.3×10^7

2.5. Boundary Condition Setup

The motion boundary conditions for the FSI model of the hose pump are divided into two parts: one for the fluid domain and one for the solid domain. For the fluid domain, the inlet velocity and outlet pressure are set as boundary conditions. The inlet velocity is specified, while the outlet pressure is controlled. For the solid domain, the rotor is initially displaced slightly to prevent contact with the hose. After the simulation begins, the rotor is moved to a fixed position. To ensure convergence of the simulation, the rotor is gradually accelerated to the desired rotational speed before performing the steady-state rotational speed simulation. Directly accelerating the rotor to the set speed would likely cause difficulties in convergence.

The model has two main contact areas: one between the rollers and the hose and the other between the hose and the casing. The contact between the hose and the casing is set up using bonded face-to-face contact. For the roller-to-hose contact, contact surface elements are defined using TARGE170 and CONTA174. Standard frictionless contact is applied between the hose and the compression rollers, and the augmented Lagrangian method is chosen for the contact algorithm. The hose does not have self-contact. In the solid region, a sparse matrix direct solver (SPAR) is used to ensure solution convergence, with large deformation enabled.

2.6. Observation Surface Settings

To analyze the flow field and pulsation characteristics within the hose, observation surfaces were set up on the hoses of both types of hose pumps at distances of 24 mm from the outlet and inlet positions, as well as at the midpoint. Pressure contour plots of the cross-sections were generated, as shown in Figure 7.

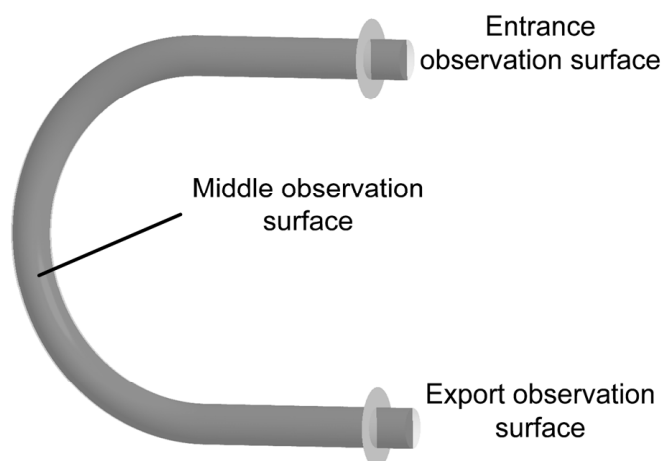


Figure 7. Observation surface schematic.

3. Experimental Setup

3.1. Description of the Test Bench

The experimental testing was conducted on an open test bench system. The system comprises actuators (two types of hose pumps), a programmable logic controller (PLC) (Xiankong, China) control cabinet, pressure sensors (0–5 V), an electromagnetic flow meter, level valves, control valves, and a computer. The specific parameters of the main experimental instruments are shown in Table 4.

Table 4. Specific parameters of experimental instruments.

Device Name	Detailed Parameters
PLC Control Cabinet	Composed mainly of PLC, touchscreen, frequency converter, relay, AC contactor (Xiankong, China), etc. The PLC input terminals are connected to a knob switch, and the output terminals are connected to the frequency converter. The frequency converter adjusts the hose pump speed by controlling the output frequency, which ranges from 0 to 400 Hz. The output analog signal range is 0–10 V.
Flowmeter (China)	Intelligent electromagnetic flowmeter, capable of collecting flow signals via AD/DA signal conversion and real-time display. The accuracy is $\pm 0.5\%$ R, and the current output range is 4–20 mA.
Pressure Sensor (Sienno, China)	SCYG310 high-frequency dynamic pressure sensor (Sienno, China) with a range of 0–1 MPa, power supply voltage of ± 15 V DC, output voltage range of 0–5 V, frequency response ≤ 3 kHz, and accuracy of 0.5%.
Data Acquisition Card (Xiankong, China)	12 differential input channels, capable of detecting electrical signals of ± 10 V and ± 5 V. The ADC resolution is 16 bits, and it can be automatically calibrated via software. The highest sampling rate reaches 250 KS/s, with a gain error of $\pm 0.25\%$ FSR.

The PLC control cabinet adjusts rotor speed, the pressure sensors measure outlet pressure pulsations, and the electromagnetic flow meter records the average outlet flow rate. A valve is installed at the end of the outlet pipeline to regulate pressure by adjusting the valve opening, while a level valve maintains stable inlet pressure. The inner and outer diameters of both inlet and outlet pipes are 25 mm and 32 mm, respectively. With water as the test material, the testing system and test pump are shown in Figure 8. The pump was started following all standard procedures and operated until suitable working conditions were achieved.

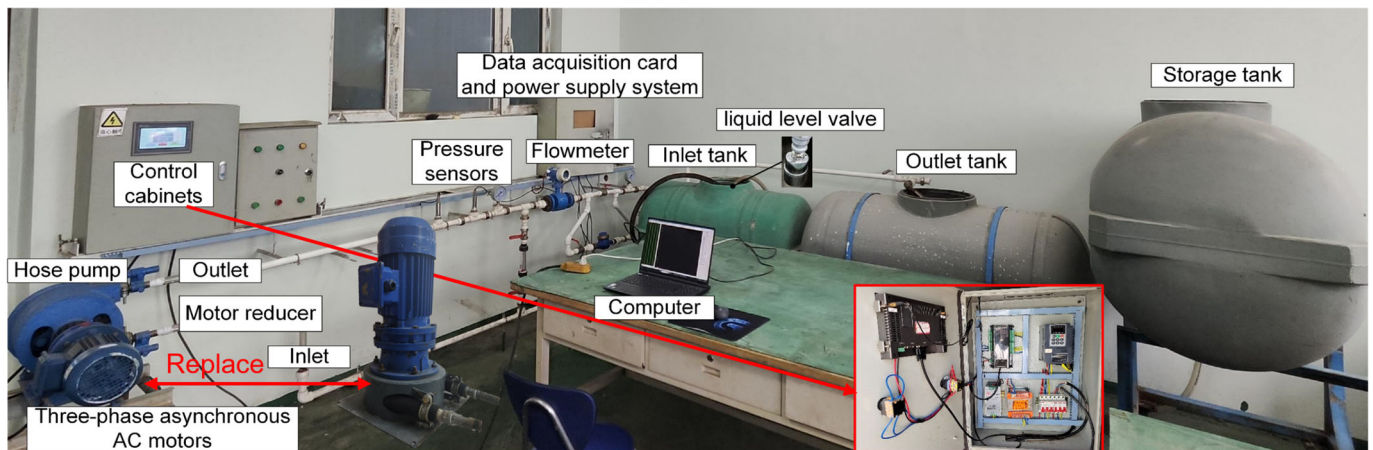


Figure 8. Hose pump testing system.

3.2. Experimental Results Validation

The hose pump for field fertilization can achieve multiple rotational speeds. The hose pump used by the research group operates at three frequencies: 35 Hz, 50 Hz, and 65 Hz. The power frequency of 65 Hz corresponds to the high-speed stage of the hose pump, 50 Hz corresponds to the medium-speed stage, and 35 Hz corresponds to the low-speed stage of the hose pump. The formula for calculating the rotor speed of the hose pump is

$$n = \frac{2\pi f(1-s)}{ip} \quad (4)$$

where f is the power frequency, s is the slip rate, with $s = 7\%$, i is the reduction ratio of the gearbox, with $i = 29$, and p is the number of pole pairs, with $p = 2$. Taking $f = 65$ Hz, 50 Hz, and 35 Hz, the calculated speeds are $n = 6.549$ rad/s, 5.037 rad/s, and 3.526 rad/s.

In the actual assembly process of the hose pump, the pressing rollers completely compress the hose; however, the simulation must adhere to energy conservation principles to achieve good convergence, resulting in a 4.5 mm gap on the inner wall of the hose. Flow leakage can have a certain impact on the variation of the flow field within the hose. During the pressure loading process, the leakage increases gradually with the rise in maximum pressure, but the flow tends to stabilize at high speeds. The inlet flow velocity can be calculated using the flow velocity formula for the hose pump.

$$v = C_d n r \quad (5)$$

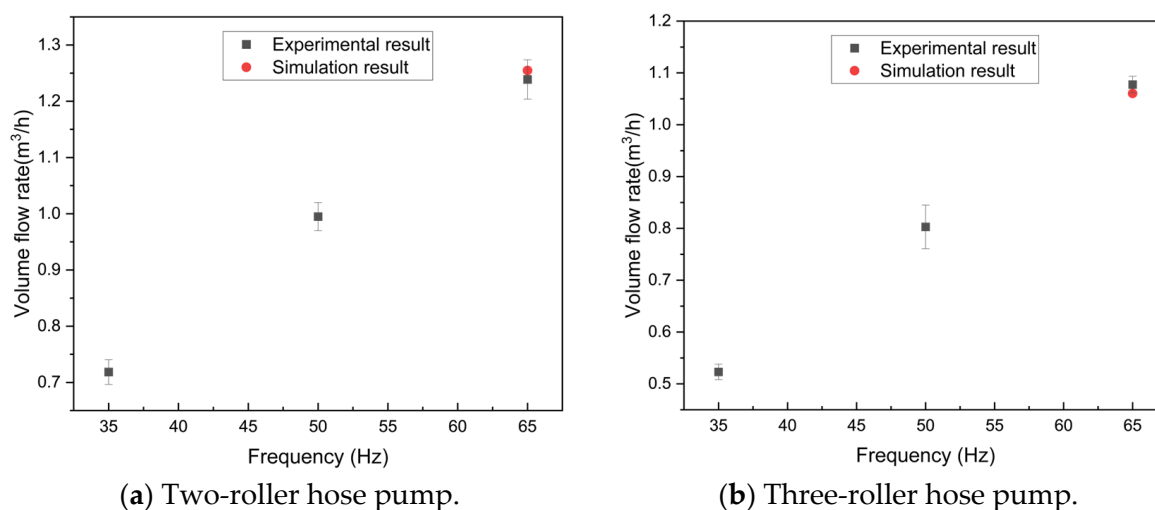
where C_d is the correction factor due to hose deformation, typically ranging from 0.7 to 0.9. This factor is closely related to the number of rollers in the pump. n is the rotor speed, in units of rad/s, and r is the rotor radius, set to 0.121 m.

Through experiments, at a power frequency of 65 Hz, the fluid intake side velocities of the two hose pumps are as follows: two-roller hose pump: 0.7 m/s; three-roller hose pump: 0.6 m/s; at a power frequency of 50 Hz, the velocities are as follows: two-roller hose pump: 0.55 m/s; three-roller hose pump: 0.45 m/s; and at a power frequency of 35 Hz, the velocities are as follows: two-roller hose pump: 0.4 m/s, three-roller hose pump: 0.3 m/s. The experimental results are shown in Table 5. After calculation, the deformation correction factor for the two-roller pump is 0.9, and for the three-roller pump, it is 0.74.

Table 5. Experimental results of volumetric flow rate for hose pumps.

Pump Type	Power Frequency (Hz)	First Test (m ³ /h)	Second Test (m ³ /h)	Third Test (m ³ /h)	Fourth Test (m ³ /h)	Fifth Test (m ³ /h)
Two-Roller Pump	65 Hz	1.2047	1.1896	1.2564	1.1915	1.3520
	50 Hz	1.0056	0.9116	1.0079	1.0535	0.9964
	35 Hz	0.7691	0.7506	0.6888	0.7246	0.6598
Three-Roller Pump	65 Hz	1.0482	1.0501	1.0677	1.1056	1.1166
	50 Hz	0.8813	0.8145	0.7497	0.8772	0.6923
	35 Hz	0.5239	0.5155	0.5492	0.5488	0.4781

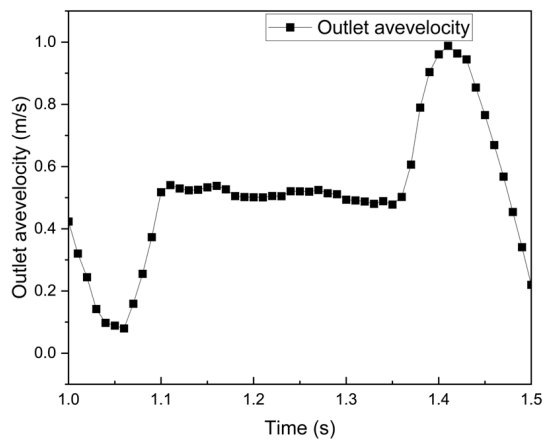
The simulation in this study uses a power frequency of 65 Hz and an outlet pressure of 0 MPa as boundary conditions. According to Formula 5, the inlet velocities are calculated to be 0.71 m/s and 0.6 m/s, respectively. The fluid material was specified as water. A comparison of the experimental results with the simulation results for the volume flow of the hose pumps is shown in Figure 9, with the error between the experimental and simulated results maintained within 5%, confirming the accuracy of the model.

**Figure 9.** Flow rate curves of experimental and simulation results.

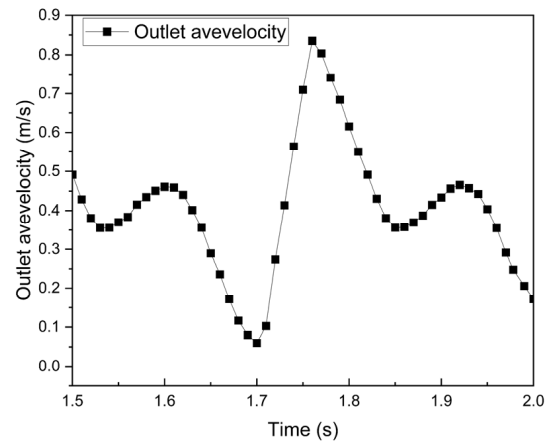
4. Results and Analysis

In field fertilization applications, a power frequency of 65 Hz is typically selected to achieve rapid fertilization. Therefore, under the conditions of a rotor speed of 6.549 rad/s and an outlet pressure of 0 MPa, this study compares the average outlet and inlet pressures of two-roller and three-roller hose pumps when transporting water and liquid fertilizer. Here, water represents a high Reynolds number fluid, and liquid fertilizer represents a low Reynolds number fluid.

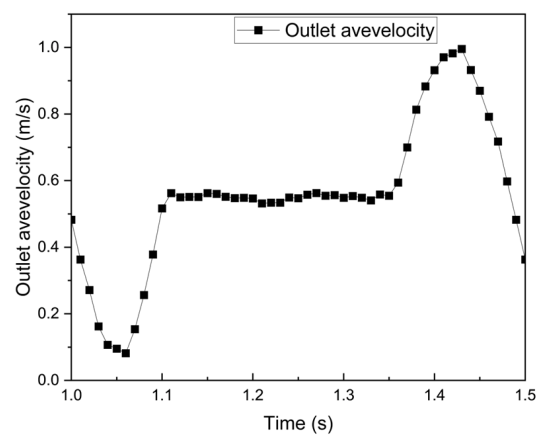
The simulations were performed on a dual Intel Xeon Gold 6230R processor system (52 cores) with DDR4 192 GB memory, with each model taking approximately 168 h to complete. The results are shown in Figures 10 and 11, where time is plotted on the horizontal axis, and average flow velocity and average pressure on the observation surface are plotted on the vertical axis. A time-average flow velocity and time-average pressure reference coordinate system is established to analyze the pulsation characteristics of hose pumps with different roller counts when delivering high and low Reynolds number fluids under identical rotor speed and outlet pressure conditions.



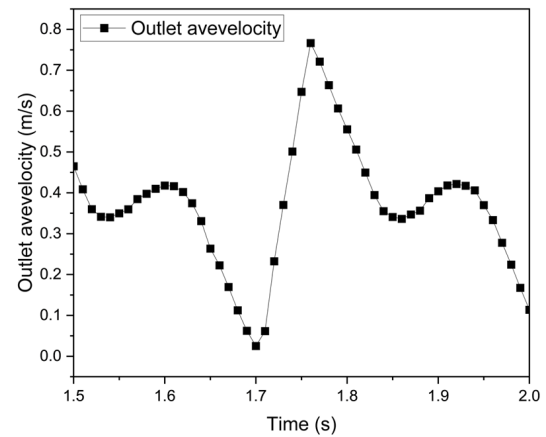
(a) Two-roller hose pump transporting water.



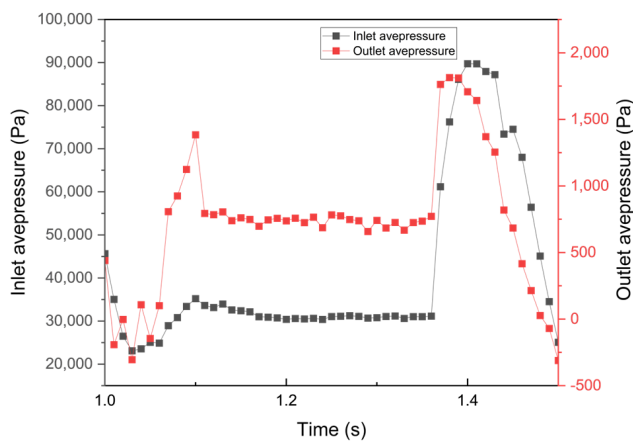
(b) Three-roller hose pump transporting water.



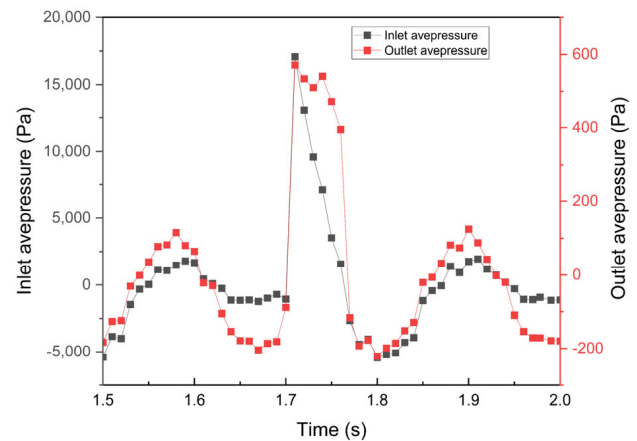
(c) Two-roller hose pump transporting liquid fertilizer.



(d) Three-roller hose pump transporting liquid fertilizer.

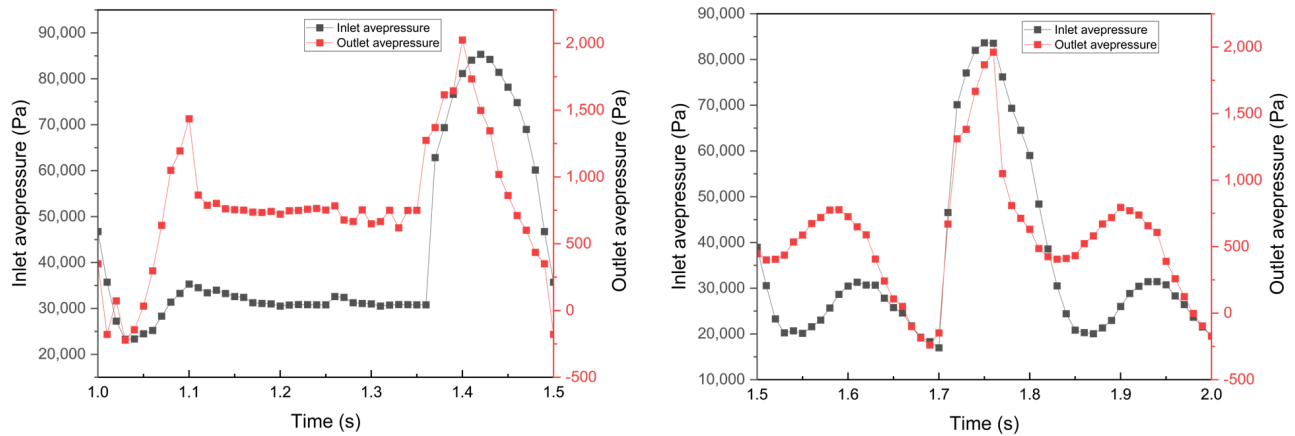
Figure 10. Variation of average outlet flow velocity for two types of hose pumps.

(a) Pressure during water transport by two-roller pump.



(b) Pressure during water transport by three-roller pump.

Figure 11. Cont.



(c) Pressure during liquid fertilizer transport by two-roller pump.

(d) Pressure during liquid fertilizer transport by three-roller pump.

Figure 11. Variation of average outlet and inlet pressure for two types of hose pumps.

It is evident that under the same rotor speed and outlet pressure conditions, the pulsation frequency of the three-roller hose pump is higher than that of the two-roller hose pump. However, pulsation frequency is not the primary cause of the negative effects of pulsation. The following discussion focuses on the magnitude of pulsation intensity.

To analyze pulsation intensity more intuitively, the maximum–minimum difference method is employed to calculate the pulsation intensity, as shown in Equation (6):

$$I = N_{\max} - N_{\min} \quad (6)$$

where I represents the pulsation intensity. N represents the value of flow velocity or pressure.

From Figure 10, it can be observed that, under the same rotor speed and outlet pressure conditions, the outlet flow velocity of the two-roller pump is significantly higher than that of the three-roller pump within one rotational cycle. The maximum average outlet flow velocities of the two-roller pump for water and liquid fertilizer are 0.988 m/s and 0.995 m/s, respectively, whereas those for the three-roller pump are 0.835 m/s and 0.766 m/s, respectively. According to Equation 6, the flow velocity pulsation intensities of the two-roller pump for high and low Reynolds number fluids are 0.908 m/s and 0.913 m/s, respectively, while those of the three-roller pump are 0.775 m/s and 0.741 m/s, respectively. The higher pulsation intensity of the two-roller pump is due to its larger internal chamber, resulting in greater flow and higher velocity, which in turn leads to increased pulsation intensity.

For the average outlet pressure, as shown in Figure 11, the two-roller pump exhibits significantly higher outlet pressure than the three-roller pump. The maximum average outlet pressures for the two-roller pump when transporting water and liquid fertilizer are 1813.516 Pa and 2024.228 Pa, respectively, compared with 571.813 Pa and 1961.310 Pa for the three-roller pump. According to the Navier–Stokes equations, areas with higher pressure gradients naturally exhibit higher velocities, indicating a clear correlation between outlet pressure and flow velocity. Based on Equation (6), the outlet pressure pulsation intensities of the two-roller pump for high and low Reynolds number fluids are 2118.606 Pa and 2245.519 Pa, respectively, while those of the three-roller pump are 793.048 Pa and 2201.112 Pa, respectively. In terms of pressure, the three-roller pump exhibits lower pulsation intensity compared with the two-roller pump.

Regarding the average inlet pressure, the trend is similar to that of the outlet pressure. Generally, the inlet pressure of both pumps is higher than the outlet pressure. However, an exception is observed in the three-roller pump when transporting water, where the outlet pressure exceeds the inlet pressure between 1.60 and 1.70 s, a phenomenon not observed in the two-roller pump.

To observe the overall flow field within the hose and understand the pulsation of pressure and flow velocity throughout the entire flow area, analyses were conducted on the pressure and flow velocity contour plots at the midpoint. For the two-roller pump, contour plots were generated at 1.08 s, 1.22 s, and 1.38 s, while for the three-roller pump, contour plots were generated at 1.53 s, 1.73 s, and 1.83 s, as shown in Figures 12–19.

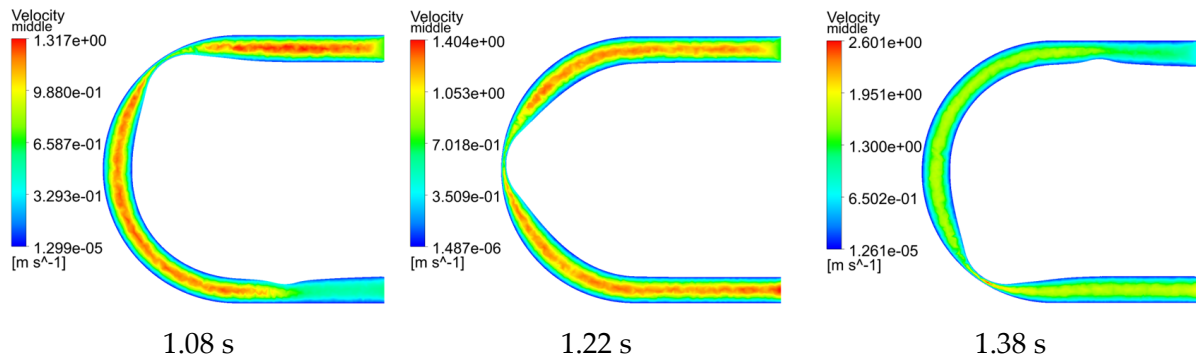


Figure 12. Mid-plane velocity contour map of the two-roller hose pump transporting water.

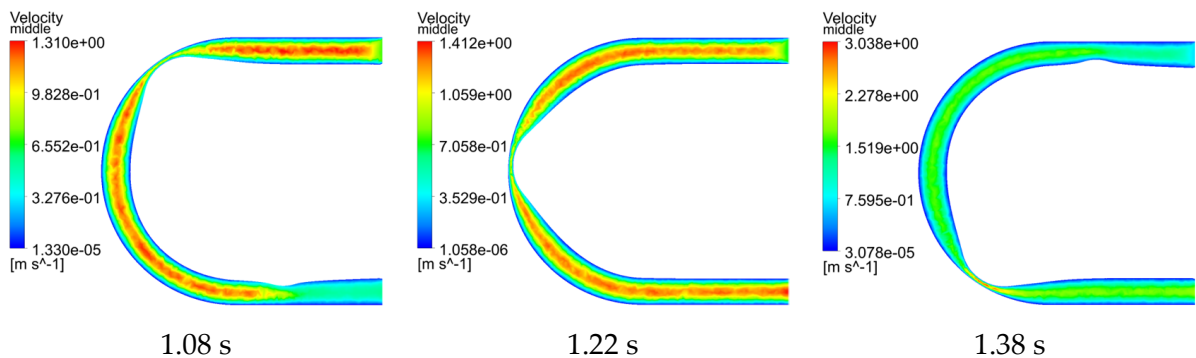


Figure 13. Mid-plane velocity contour map of the two-roller hose pump transporting liquid fertilizer.

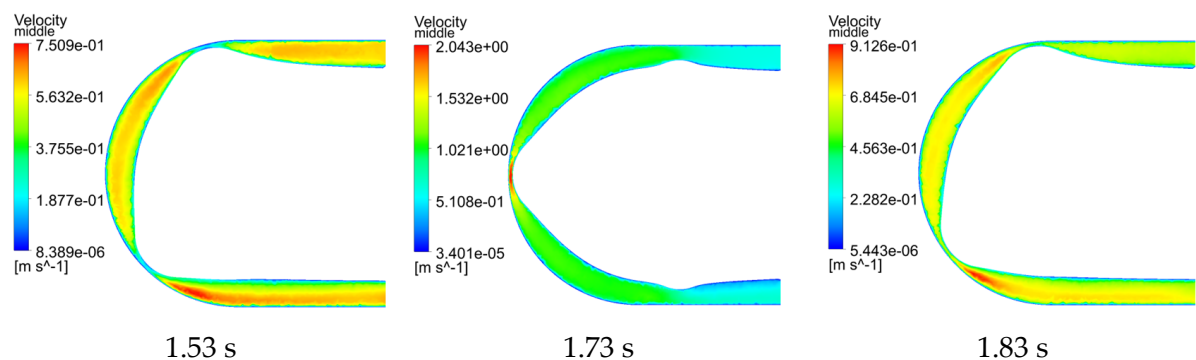


Figure 14. Mid-plane velocity contour map of the three-roller hose pump transporting water.

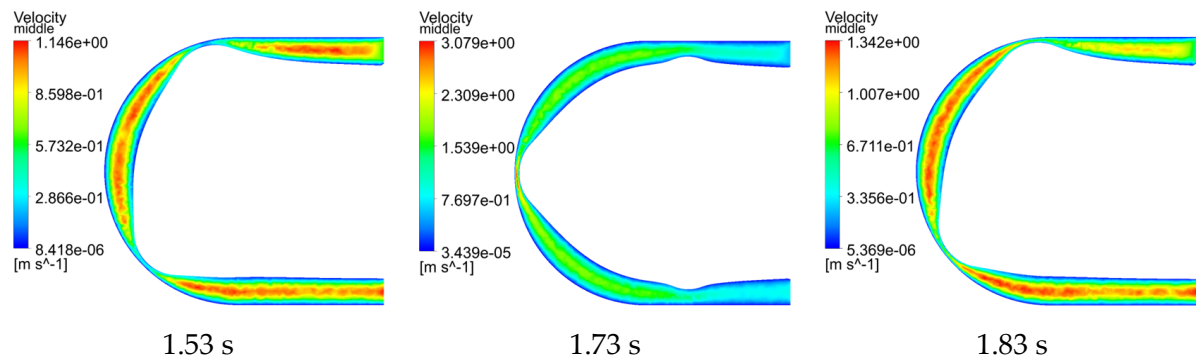


Figure 15. Mid-plane velocity contour map of the three-roller hose pump transporting liquid fertilizer.

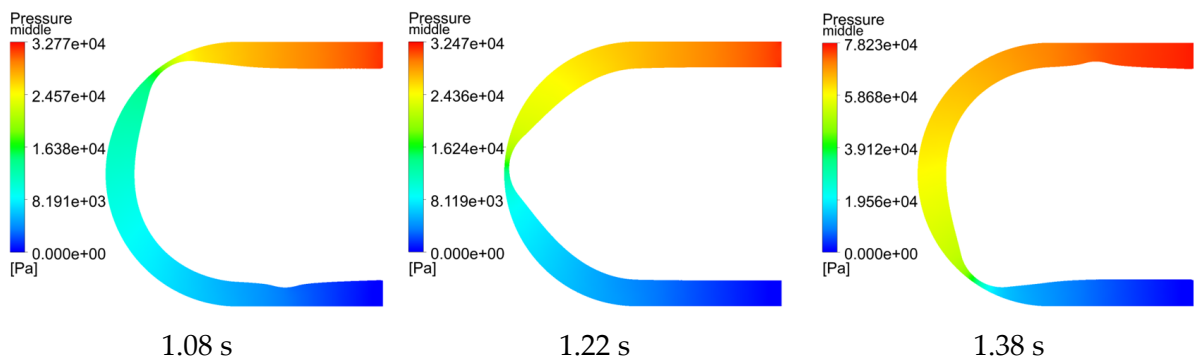


Figure 16. Mid-plane pressure contour map of the two-roller hose pump transporting water.

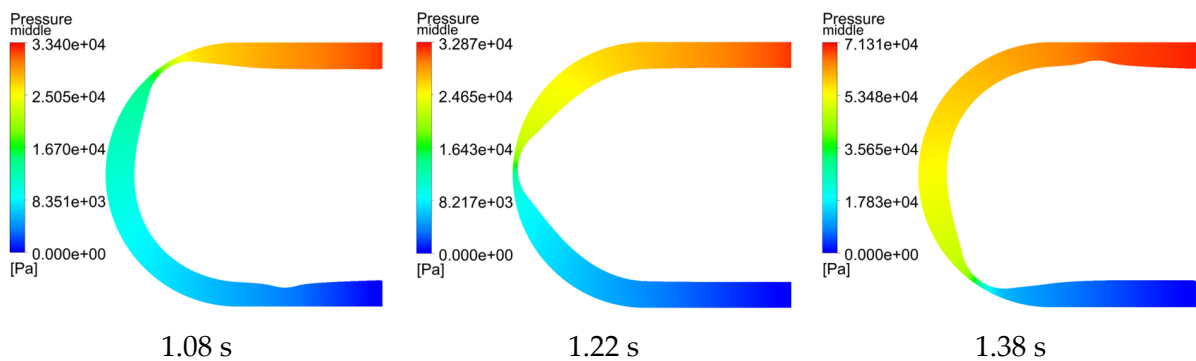


Figure 17. Mid-plane pressure contour map of the two-roller hose pump transporting liquid fertilizer.

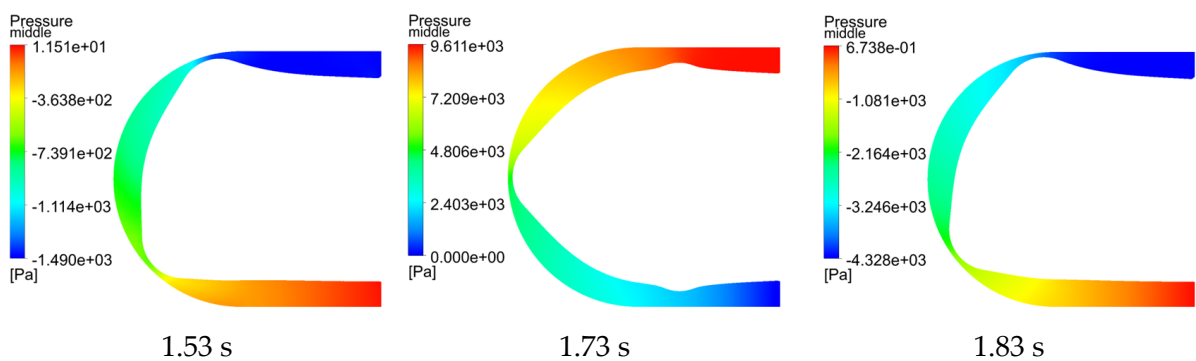


Figure 18. Mid-plane pressure contour map of the three-roller hose pump transporting water.

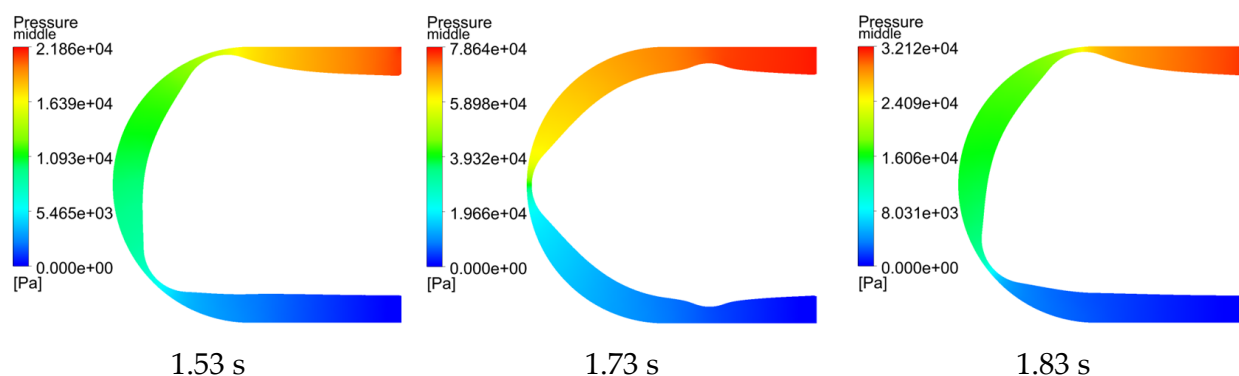


Figure 19. Mid-plane pressure contour map of the three-roller hose pump transporting liquid fertilizer.

From the intermediate surface pressure contour maps, it can be observed that, compared with the two-roller pump, the three-roller pump exhibits a situation where the inlet pressure is higher than the outlet pressure at 1.53 s and 1.83 s when transporting water. This occurrence is not observed under other operating conditions, which corresponds to the analysis in the previous section. The highest pressure points in the flow field for both pumps, when transporting high and low Reynolds number fluids, occur at the moment when the rollers enter the compressed hose. When transporting high Reynolds number fluids, the overall flow field pressure of the two-roller pump is significantly higher than that of the three-roller pump, which also aligns with previous analysis. However, when transporting low Reynolds number fluids, the difference in flow field pressure between the two pumps is minimal.

From the intermediate surface velocity contour maps, it can be seen that the flow velocity in the two-roller pump is generally higher than in the three-roller pump. The maximum velocity for both pumps occurs when the rollers enter the compressed hose. Therefore, under identical operating conditions, the pulsation intensity of the three-roller pump is noticeably lower than that of the two-roller pump.

To further analyze the pulsation characteristics of the two-roller and three-roller pumps, turbulence vortex dissipation is introduced to analyze the energy dissipation of the flow field when transporting water, as shown in Figures 20–23. The energy dissipation contour maps indicate that the two-roller pump has significantly higher energy dissipation compared with the three-roller pump when transporting water, reflecting its higher turbulence intensity, which in turn causes more vibration and noise, enhancing its pulsation. When transporting liquid fertilizer, the energy dissipation of the two-roller pump is similar to that of the three-roller pump. Moreover, the energy dissipation in both pumps is higher when transporting liquid fertilizer than when transporting water, which may be due to transient flow separation and reattachment occurring when the fluid passes through the contraction and diffusion sections of the hose. Due to the high viscosity of liquid fertilizer, small-scale vortices generated during this process are quickly smoothed out by viscous forces, resulting in higher energy consumption.

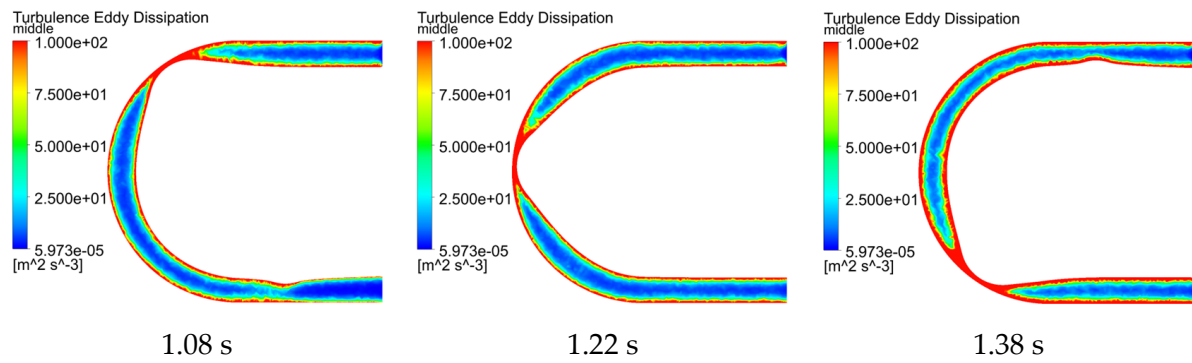


Figure 20. Contour map of turbulent energy dissipation in the intermediate surface of a two-roller hose pump when transporting water.

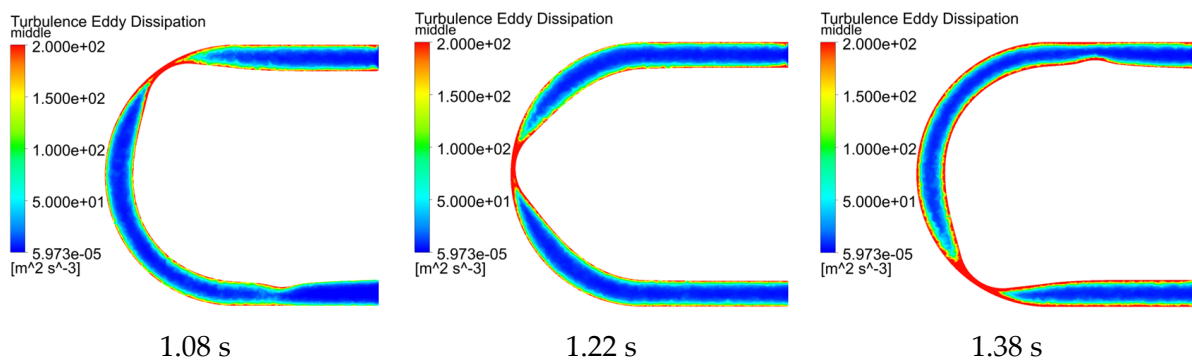


Figure 21. Contour map of turbulent energy dissipation in the intermediate surface of a two-roller hose pump when transporting liquid fertilizer.

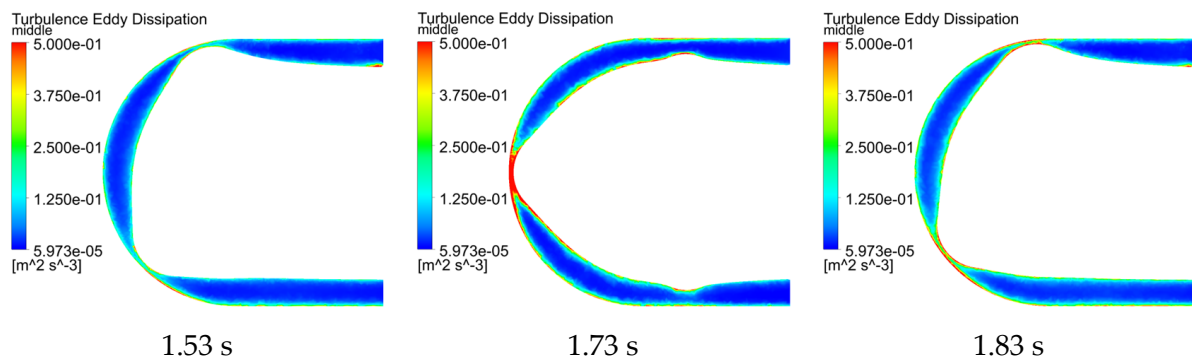


Figure 22. Contour map of turbulent energy dissipation in the intermediate surface of a three-roller hose pump when transporting water.

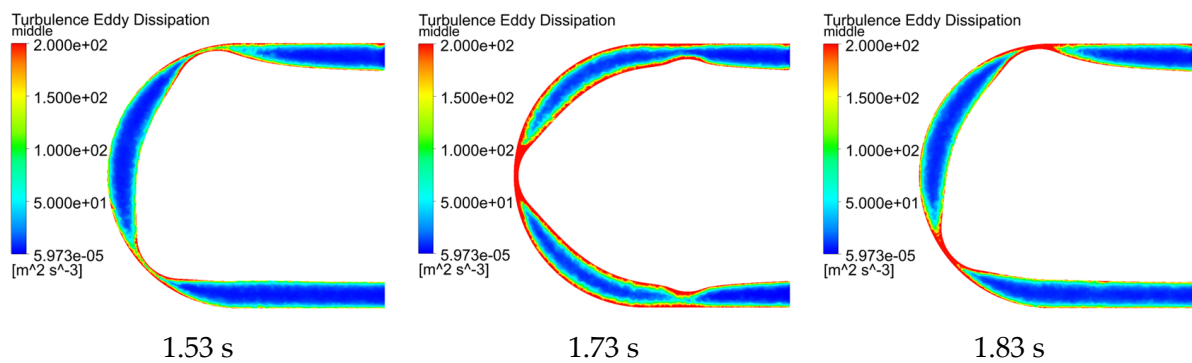


Figure 23. Contour map of turbulent energy dissipation in the intermediate surface of a three-roller hose pump when transporting liquid fertilizer.

5. Conclusions

A bidirectional fluid–structure interaction model was established, and a test bench was set up for verification. The pulsation characteristics of hose pumps with different numbers of rollers were analyzed. The comparison results are shown in Table 6, and the following conclusions can be drawn:

Table 6. Comparison of Results.

Hose Pump Type	Fluid Transported	Flow Pulsation (m/s)	Pressure Pulsation (Pa)	Pulsation Frequency (Hz)
Two-Roller Hose Pump	High Reynolds Number	0.908	1813.516	2.085
	Low Reynolds Number	0.913	2024.228	2.085
Three-Roller Hose Pump	High Reynolds Number	0.775	793.048	3.127
	Low Reynolds Number	0.741	2201.112	3.127

The outlet flow velocity of the two-roller pump is significantly higher than that of the three-roller pump throughout the entire rotation cycle. The flow pulsation intensity of the two-roller pump is also higher, primarily due to its larger chamber space and greater flow rate. When transporting high Reynolds number fluids, the overall flow field pressure of the two-roller pump is noticeably higher than that of the three-roller pump. However, under low Reynolds number conditions, the pressure difference between the two pumps is minimal. The pressure pulsation intensity of the two-roller pump is also significantly higher than that of the three-roller pump when transporting high Reynolds number fluids, while the difference is minimal when transporting low Reynolds number fluids.

When transporting high Reynolds number fluids, the three-roller hose pump exhibits a situation where the average outlet pressure exceeds the average inlet pressure, which is not observed in the two-roller pump. This increases the risk of backflow in the three-roller pump.

In terms of turbulent energy dissipation, the two-roller pump shows higher turbulence intensity and energy dissipation when transporting high Reynolds number fluids, indicating more noticeable vibrations and noise. When transporting low Reynolds number fluids, the energy dissipation of both pumps is similar and higher than that observed during the transportation of high Reynolds number fluids. This is related to the transient separation and vortex dissipation of the flow field caused by the high viscosity of low Reynolds number fluids.

To reduce the pulsation intensity of hose pumps and mitigate the associated risks, it is recommended to increase the number of rollers. Additionally, pulsation can be reduced through shape optimization or by using feedback control systems. For shape optimization, increasing the shell angle can reduce the fluid acceleration time in the hose, thereby decreasing pulsation. For feedback control systems, using a PID controller can adjust the fluid flow rate or mechanical motion in real-time by monitoring pressure changes, ultimately reducing pulsation.

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Conflicts of Interest: Author Wenchun Li was employed by the company Xinjiang Zhuangeer Yinfen Agricultural Machinery Equipment Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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