Numerical Analysis of a Small Pneumatic Hammer Performance Based on Evaluation Method of Friction Force

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Received 3 November 2021; Revised 9 March 2022; Accepted 23 March 2022; Published 12 April 2022

Mechanical power loss in pneumatic hammers comes from the friction between parts in relative motion, and wear is among the top hammer failure mechanisms. The friction force may become higher by compressed air leaked from the annular gaps between the cylindrical matching surfaces of the components. Hence, the Coulomb friction between piston and cylinder is about 5% to 15% of the thrust on the piston [5]. Theoretical and experimental studies on friction in hydraulic actuators with sealless pistons may be summarized as follows:

In pistons with diverging clearance, hydrodynamic lift (in other words, positive centering force) is provided for the impacting piston. This lift force is essential to center the piston in place as it travels down the cylinder. The hydrodynamic lift is generated by the pressure difference between the high-pressure fluid in the main chamber and the low-pressure fluid in the small annular gap between the piston and the cylinder. The amount of lift force generated depends on various factors such as the clearance between the piston and the cylinder, the fluid pressure, and the fluid viscosity. The lift force helps to counteract the weight of the piston and any other load it is carrying, thus preventing the piston from moving down too quickly and losing control.

1. Introduction

Pneumatic hammers have been extensively used in quarries, open pit mines, and construction sites all over the world. Chipping hammer, pick hammer, and paving breaker are common pneumatic tools for structural contractors, stone workers, and rock miners, and so on [1]. They have been historically considered as the most robust and productive tool for building and civil engineering [2]. However, they are the number one cause of acute injuries among minors due to their heavy weight and are associated with very high levels of noise and vibration levels [3]. In general, a piston driven by compressed air converts its kinetic energy to impact energy by colliding with a steel rod or drill bit, causing damage to the objects when it is in stroke [4].
piston beyond a certain velocity of the piston. In pistons with converging clearance, negative centering force is developed beyond a certain velocity, forcing the piston toward the cylinder wall [6, 7]. Belforte et al. [8] presented theoretical and experimental analysis of sealless pistons featuring a special geometry capable of eliminating friction in pneumatic cylinders and valves. Abovementioned studies hint for hammer manufacturers to focus on the friction force between the cylindrical matching surfaces of the components. However, we have seldom read the similar research data to quantify the friction force and consider it in the simulation model of pneumatic hammer.

Hu et al. [9] carried out dynamic simulation and test research of the impact mechanism on the assumption that the friction force has little effect on the impact energy and frequency. In this study, they ignored the influence of friction force and the error between the simulated and measured values (ESM) for the impact energy was 10.34%, ESM for frequency was 6.03%. Bo et al. [10] simulated and evaluated the performance of the pneumatic down-the-hole (DTH) hammer with self-propelled round bit by considering fluctuations of the front and rear chamber pressure, impact energy, acceleration, and frequency. In simulation, they indirectly obtained the impact energy from piston speed, which was determined experimentally through the analysis of the front and rear chamber pressure. ESM for the impact energy was 13.07, and ESM for impact frequency was 7.69%. Li [11, 12] and Li et al. [13] presented the research results on impact performance of hydraulic rock drill. Zhang [14] investigated the performance of reverse circulation DTH (RC-DTH) air hammer using computational fluid dynamics (CFD) with dynamic mesh method on the assumption that the friction force was sufficiently low such that it could be ignored in calculation. Xu et al. [15] and Xu [16] presented the results of numerical simulation and experimental research of hydraulic hammer with Advanced Modeling Environment for Simulation Engineering Systems (AMESim) software, ESM for the impact energy and frequency were 12.4 and 5.26%, respectively. Yin et al. [17] built a model for the impact mechanism of hydraulic rock drill with AMESim software and obtained the displacement curves of the piston and valve core, and the pressure of the piston chamber. Yang et al. [18] used the AMESim simulation model to verify the displacement and velocity curves of the piston and valve by using stress wave testing data. Results of the comparison showed that ESM for the impact energy and frequency were 9.7 4 and 1.87%, respectively.

Most of abovementioned studies have ignored friction force in both simulation and experiment to determine impact energy and frequency in pneumatic hammer, relating to numerical simulation and experimental research of fluid hammer; therefore, the error of simulation and measurement results remains. It is necessary for accurate predictions on performance to consider the friction force in a pneumatic hammer.

In this article, we presented a novel approach to quantitatively determine the friction force and consider it the simulation model of a small pneumatic hammer. The friction force was experimentally determined at different inlet pressures, and AMESim software was used to analyze the hammer performance by considering the friction force. This research can provide practical guidance for predicting the service life of piston and finding a low friction piston of hydraulic hammer, as well as pneumatic hammer.

2. Description of the Small Pneumatic Hammer

Figure 1 shows the structure of a novel small pneumatic hammer. Its features are as follows: (i) its structure was designed to easily change piston and valve during experimental work; (ii) it is easy to arrange sensors to measure not only pressures in front and rear chambers but also the movement of piston.

As shown in Figure 1, the pneumatic hammer has only an impact mechanism and mainly consists of a piston, a control valve, control ports, outlets, and an impact cylinder. The movement of the piston can be divided into two phases as follows: the backhaul phase and the stroke phase, and each phase experiences air intake, air expansion, air compression, and air exhaust stages (Figure 2). As depicted in Figure 2 and Table 1, when compressed air flows into the front or rear chamber of piston via control port 3 or 4 opened by a valve, reciprocating motion of piston begins. Meanwhile, the valve is stable due to applying high pressure to the one side of valve. At the initial stage (a), compressed air flows into the rear chamber of piston. At the stage (b) \((l_2 - l_p < l < l_1)\), the piston accelerates until the outlet in the front chamber is closed. With opening the outlet in the rear chamber after the outlet in the front chamber closed, compressed air vents from the piston rear chamber into the atmosphere, while the air in the front chamber begins to compress due to the piston motion (stage (c)). The piston impacts shank and the piston lies in the starting position of return stroke while the pressure in the rear chamber gets lower than that in the front chamber (stage (d)). Then, the valve quickly moves toward the left direction due to the pressure difference at two sides of valve to close up the port 3 and open the port 4. At the same time, the compressed air flows into the front chamber of piston (stage (e)). When the outlet in the front chamber is opened in the backhaul phase, the returning acceleration of piston is ended (stage (f)). At the end of stage (f), stage (a) will become active again and a new work cycle begins.

Table 1 lists piston logic to the conditions in each stage.

3. Method

3.1. Numerical Simulation. The model assumptions are as follows:

(1) The gas is perfect, and the viscosity is unaffected by the pressure.

According to the study [18], if fluid pressure is below 20 MPa, viscosity is expected to only slightly change. In the current study, the pre-setting pressure of the relief valve is below 0.7 MPa. Therefore, the viscosity is assumed unaffected by the pressure.
In this study, the friction force of impact cylinder is constant under the given inlet pressure. The friction force fluctuates with piston velocity, pressure difference between chambers, and so on. In this study, since we focus on evaluating the hammer performance, the friction fluctuation has little effect on the research purpose; therefore, the influence of friction fluctuation is ignored.

The impact piston and control valve are the absolute rigid bodies without distortion [18].

Table 1: Piston logic and conditions at each stage.

| Stages | Piston displacement (L) | Velocity (v) | Air condition (\(m_f, m_f', P_1, P_2\)) | Collision |
|--------|------------------------|--------------|----------------------------------------|-----------|
| a      | \(0 < L < l_2 - l_p\)  | \(v \geq 0\) | \(m_f = 0, m_f' > 0, P_1 = P_0\)       | No        |
| b      | \(l_2 - l_p < L < l_1\) | \(v > 0\)    | \(m_f = 0, m_f' > 0, P_2 > P_1 > P_0\) | No        |
| c      | \(l_1 < L\)            | \(v > 0\)    | \(m_f = 0, m_f' = 0, P_2 < P_1 < P_0\) | No        |
| d      | \(l_1 < L\)            | \(v \leq 0\) | \(m_f > 0, m_f' = 0, P_2 = P_0\)       | Yes       |
| e      | \(l_2 - l_p < L < l_1\) | \(v < 0\)    | \(m_f = 0, m_f' > 0, P_2 > P_1 > P_0\) | No        |
| f      | \(0 < L < l_2 - l_p\)  | \(v < 0\)    | \(m_f > 0, m_f' = 0, P_2 < P_1\)       | No        |

Note. \( L, l_p, l_1, l_2 \) represent certain key locations in the piston where the air conditions change at each stage; \( m_f \) represents the instantaneous air mass flow rate passing through the front chamber; and \( m_f' \) represents the instantaneous air mass flow rate passing through the rear chamber, as shown in Figure 1.

Figure 3 shows the simulation model of the proposed pneumatic hammer.

3.1.1. The Impact Piston and Shank Model. We can achieve the physical function of the piston chamber by selecting components from the Pneumatic Component Design (PCD) library. The elastic contact component between piston and shank enables to attain the collision between two bodies. When the contact occurs, a contact force acts on both bodies.
This consists of a spring force and a damping force. The model enables to simulate piston stroke acceleration, shock pause, shock acceleration, and braking motion while the piston returns.

3.1.2. The Control Valve Model. The control valve is a two-position three-way reversing valve. The control valve action is associated with the movement of the piston. When the piston opens the outlet in the front chamber, the left side of the control valve becomes pressurized, and the valve moved to the right side. When the piston opens the outlet in the rear chamber, the pressure at the left side of the control valve decreases rapidly, and the valve moves to the left side. In AMESim modeling, we can achieve the function of the control valve as a two-position three-way pneumatic valve sketched by selecting the pneumatic conical poppet with sharp edge seat.

3.1.3. Parameter Settings. Mass of the piston is 1.12 kg; diameter of the piston is 41 mm; piston length \( l_p \) is 35 mm; piston rod diameter is 28 mm; diameter of the control port is 8 mm; and diameter of the outlet is 5 mm.

Distances between the right side of the cover and the central line of outlets are \( l_1 = 30 \text{ mm} \) and \( l_2 = 55 \text{ mm} \), respectively (Figure 1). The clearance on diameter is 0.05 mm and the eccentricity between piston and cylinder is 1. The mass of the control valve is 0.005 kg; valve port flow coefficient is 0.62; the valve opening size is 1 mm; moving distance of the valve is 2 mm; the inlet pressure is 4 bar. In the simulation mode, the sampling time is set to 0.5 s; the sampling interval is 10 \( \mu \)s.

3.1.4. Simulation Analysis. Figures 4–6 are curves of displacement, velocity, and acceleration of the impact piston.

The movement of the impact piston is in steady state after 0.05 s. The initial position of the piston is 10 mm and its stroke is about 42 mm (Figure 4). The stroke velocity reaches a minimum of 2.05 m/s, which exceeds the steady-state value of 2.83 m/s. Then, the stroke velocity gradually increases to become steady (Figure 5). The instability of the impact piston results from the initial pressure inside the pipe and chambers at the beginning of simulation. The piston
acceleration curve has negative peaks in returning stroke due to collision of the piston and the shank (Figure 6). The values show that the impact force acts on the piston during the collision.

Figure 7 illustrates the variation pressures in the front and rear chambers of the pneumatic hammer. The rear chamber pressure is up to 3.282 bar and the front chamber pressure is up to 2.489 bar.

Figure 8 shows that the conversion of the motion of the impact piston corresponds to the inflection point of the pressure curves of the rear and front chambers.

When the piston displacement reaches 30 mm during the forward movement, the outlet in the rear chamber is opened and the pressure in the rear chamber of piston gets lower. Meanwhile, when the piston displacement reaches 30 mm during the backward movement, the outlet in the rear chamber is closed and the rear chamber pressure gets higher. In the same way, the piston opens or closes the outlet in the front chamber when the piston displacement reaches 20 mm (Figure 8). All of the simulation results described above are obtained when friction force is ignored; therefore, the current results can be compared with those published in the previous literature.

3.2. Experimental Method

3.2.1. Experimental Setup. Figure 9 shows the experimental setup and its layout of the pneumatic hammer.

As can be shown in Figure 9, the experimental setup consists of two pressure sensors and an acceleration sensor, and a multichannel waveform recorder. The pressure sensors are piezoelectric, which is manufactured by KISTLER® in Winterthur, Switzerland, range: 0–10 bar, 140 kHz natural frequency, and 0.8% FS accuracy. The acceleration sensor is the internal electric piezoelectric, which is manufactured by Boston in MA, USA, range: 10−6–106 m/s², 0–50 kHz natural frequency, and 0.5% FS accuracy. These sensors can simultaneously measure pressures and accelerations by multichannel waveform recorder.

The experimental setup can be used for two purposes; (I) Evaluation of friction force between piston and cylinder; (II) Evaluation of the pneumatic hammer performance.

3.2.2. Friction Force Experiment. The experimental test system was used for determining the friction force due to the contact between piston and cylinder at different inlet pressures.

The friction force due to the contact of the piston with the cylinder can be calculated based on the measured values of acceleration and pressure as follows:

\[
f = \begin{cases} 
P_1 A_1 - P_2 A_2 - ma(v > 0), \\
P_2 A_2 - P_1 A_1 - ma(v < 0), 
\end{cases}
\]

where \(P_1\) and \(P_2\) are the instantaneous pressure in the piston rear and front chambers, respectively, \(P_a\); \(A_1\) and \(A_2\) are the effective working areas of piston in the rear and front chambers, respectively, \(m^2\); \(a\) is the instantaneous acceleration of piston except for the shock acceleration due to collision, \(m/s^2\), and \(m\) is the mass of piston, kg.

As known by above equation (1), \(v > 0\) denotes the stroke phase and \(v < 0\) denotes the backhaul phase of piston. Therefore, the mean value of the friction force between the piston and the cylinder can be expressed as follows:

\[
\bar{f} = \frac{1}{T} \int_0^T f dt = \frac{\sum_{i=1}^{n} f_i}{n},
\]

where \(T\) denotes the measuring time, \(i\) denotes the number of measured values, \(n\) is the total number of measured values \((n = T/\Delta t)\), and \(\Delta t\) denotes the sampling time.

For each inlet pressure, three measurements are tested. In the same way, the mean values of the friction force can be determined by equation (2).

3.2.3. Experiment on Impact Performance. Impact energy and frequency were measured and interpreted to evaluate impact performance of the pneumatic hammer. These measurements could be performed simultaneously using the test bench. The impact energy is obtained indirectly from

![Figure 7: Pressure curves in the rear and front chambers.](image)

![Figure 8: Curve between piston displacement and pressure.](image)
piston speed. Many researchers obtained the impact velocity through the analysis of pressures in front and rear chambers of piston [10–12]. They ignored friction force on the assumption that it has little effect on the research purpose. However, we can take into account friction force that is determined experimentally as above described.

The action force acting on the impact piston in the stroke phase can be obtained as follows:

\[ F = P_1 A_1 - P_2 A_2 - f, \quad (3) \]

where \( F \) denotes the action force acting on the impact piston in the stroke phase, N; and \( f \) denotes the friction force, N.

Based on the law of momentum change, the action force \( F \) acting on the impact piston integrated:

\[ \int_{a_1}^{b_1} F \, dt = mv. \quad (4) \]

Substituting equation (3) into (4), following relation can be obtained:

\[ \int_{a_1}^{b_1} (P_1 A_1 - P_2 A_2 - f) \, dt = mv. \quad (5) \]

Thus, the speed of each moment can be written as follows:

\[ v = \frac{A_1}{m} \int_{a_1}^{b_1} P_1 \, dt - A_2 \int_{a_1}^{b_1} P_2 \, dt - \bar{f} (b_i - a_i). \quad (6) \]

Since \( \bar{f} \) is determined experimentally by equation (2), the impact energy is obtained when the velocity is maximum, that is, \( E = 0.5 \, mv^2 \). The impact frequency, air consumption, and impact efficiency are determined as presented by literature [10–12].

\subsection{3.2.4. Uncertainty Analysis in Experimental Measurements}

The method requires initial estimates of the model parameters \((A_1, A_2, m, P_1, P_2 \text{ and } a)\) in equations (1) and (2). However, only three unknown parameters, \( P_1, P_2, \) and \( a \) are measured on the test facility because the piston dimensions \((A_1 \text{ and } A_2)\) and the piston mass \((m)\) can be accurately measured independently. Friction force computed using the measured values of \( P_1, P_2, \) and \( a \) will differ from the indirectly measured friction force.

The theory of errors propagation has been applied on the data measured on the experimental setup in order to define the uncertainty ranges of the three main working parameters, that is, rear chamber pressure of piston \((P_1)\), front chamber pressure of piston \((P_2)\), and piston acceleration \((a)\).

The friction force values have been calculated as functions of \( P_1, P_2, \) and \( a \), they directly measured, so uncorrelated, variables

\[ f = f (P_1, P_2, a). \quad (7) \]

Therefore, the expression of the absolute uncertainty in the friction estimation is

\[ \Delta f = \sqrt{\left( \frac{\partial f}{\partial P_1} \Delta P_1 \right)^2 + \left( \frac{\partial f}{\partial P_2} \Delta P_2 \right)^2 + \left( \frac{\partial f}{\partial a} \Delta a \right)^2}, \quad (8) \]

where \( \Delta P_1, \Delta P_2, \) and \( \Delta a \) are the absolute uncertainties in the measurements of pressures and acceleration, respectively.

Pressure sensors used during the tests feature a declared accuracy equal to 0.8% of the read value, so \( \Delta P_1 \), and \( \Delta P_2 \) varies for each different measurement point, while the accelerometer accuracy is reported as a fraction of the instrument full scale range, then \( \Delta a \) is equal to 0.5%. Starting from equation (8), the expression of the relative error in the definition of the friction force is

\[ e_f = \sqrt{e_{f1}^2 + e_{f2}^2 + e_{fa}^2}, \quad (9) \]

where \( e_{f1}, e_{f2}, \) and \( e_{fa} \) represent the relative uncertainties in the measurements of pressures and acceleration, respectively. The results achieved from equation (9), when applied
on the data obtained from the different measurement points, vary in a narrow range, so a constant value can be assumed for $e_p$, equal to 1.2%. It also may be related to external factors such as the lubrication condition, coupling status of acceleration sensor, surface quality, and so on. The application of all the equations described above, together with other simple calculations, led to determine the relative uncertainty on the impact energy, equal to 2.41% of the calculated values.

4. Results and Discussion

4.1. Validation of the Simulation Model. In order to verify the model results with AMESim software, we compared pressure and acceleration curves obtained by preliminary simulation results with those tested in the experimental equipment shown in Figure 9. Preliminary simulations have been run under the same condition (at 4 bar) tested in the lab. Figure 10 shows the simulated and measured pressure curves of impact cylinder chambers at 4 bar of the inlet pressure. Figure 11 shows the simulated and measured acceleration curves of impact piston at 4 bar.

As shown in Figure 10, it can be seen that the model results are really close to the experimental data on the pressures in the front and rear chambers. However, there is a significant gap between model and experimental data on the acceleration of the piston, as shown in Figure 11. It indicates the fact that the model does not consider the presence of mechanical friction due to the contact of the piston with the cylinder.

4.2. Friction Force. The mean values of the friction and its relative errors are listed in Table 2.

According to the test, the static friction force was about 0.8 N when there was pressure supply under the horizontal installation and lubricated condition. However, it increases from 10.27 to 16.7 N due to increase in inlet pressure at 4–7 bar. The experimental results illustrate that the friction force relates to the inlet pressure and is 10–20 times of the static friction force. The higher friction force is due to the piston of the proposed hammer with the converging clearance, forcing the piston toward the cylinder wall. Increase in friction force leads to a more severe wear and results in a more amount of air leakage from the annular gaps between the cylindrical matching surfaces of the components. It also indicates that mechanical power loss in the pneumatic hammer is about 10% of impact energy.

When the inlet pressure is 4 bar, the mean value of the friction force is 10.27 N. Therefore, it could be taken into account for the simulation model. Figure 12(a)–12(d) shows the difference between the current approach and the existing one at 4 bar.

Piston stroke and impact velocity decrease simultaneously because of the friction force between piston and cylinder (Figures 12(a) and 12(b)). Therefore, while impact energy will be significantly influenced by friction force, impact frequency will be slightly influenced by friction force. ESM for the impact frequency was 1.87–7.69% due to ignoring the influence of friction force in previous studies [9–13]. The difference between the piston acceleration curves is essentially due to fact that the previous model does not consider the presence of mechanical friction (Figure 12(c)). Especially, the friction force has also an influence on the pressure variation in the front and rear chambers of piston (Figure 12(d)). Therefore, the previous experimental approaches, that have ever obtained the impact energy indirectly through the analysis of pressure curves in previous papers [9–13], might make errors in measurement due to the influence of friction forces. However, the current experimental approach enables to predict with good accuracy the performance of the pneumatic hammer.

4.3. Impact Performance. We compared the current and preliminary simulation results to the experimental data for the impact energy, and frequency of pneumatic hammer at different inlet pressures. Overall comparison results show that the friction has little influence on the impact frequency, but it affects significantly the impact energy of pneumatic
hammer. It relates to the fact that the impact energy is proportional to the square of the impact velocity (i.e., $E \propto v^2$).

Figure 13 shows results of the comparison between measure and simulations of impact energy at 4–7 bar. Figure 14 shows results of the comparison between measure and simulations of impact frequency at 4–7 bar.

As shown in Figure 13, it indicates that the difference between the current approach and previous one increases from 13% to 17% due to increase in inlet pressure at 4–7 bar. While there were substantial differences (more than 13%) in impact energy between the current study and previous ones, there were little differences (about 2%) in impact frequency (Figure 14). Little differences in impact frequency were due to a combination of simultaneous decreases in piston stroke and impact velocity influenced by friction force, as shown in Figures 12(a) and 12(b). Difference between the current approach and previous one is as follows (Table 3).

In Table 3, according to comparing several studies with previous approach preliminarily simulated in this article, the lower values for impact energy reported in previous articles [9,10,16] may be due to experimental method used in those studies. They obtained the impact velocity through the analysis of pressures in front and rear chambers of piston. Therefore, the experimental results themselves might have errors due to the influence of friction forces. The reason is explained by Figure 12(d) as mentioned above. The lower values reported in Yang’s paper [18] may be due to errors of stress wave testing method used in that study. It is not known why previous articles generate higher values of ESM for impact frequency than the previous approach simulated.
in this article. Perhaps, it may be that the mechanism of bit impact with the rock in the real operations differs from those in laboratory condition.

Figure 13 and Table 3 indicate that the current approach can significantly eliminate the ESM for the impact energy reported in those previous papers. It also indicates that mechanical power loss in the pneumatic hammer is more than 10% of impact energy. Therefore, it is important for a high-performance and a significantly longer service life of pneumatic hammers to reduce the friction force between parts in relative motion. Reducing friction and preventing wear require lubrication enhancement, which may be accomplished by proper surface texture design [8]. Unfortunately, we were unable to investigate the sealless piston with proper surface texture design. However, we believe that our method could be used to determine the friction force quantitatively and find a new low friction piston of the hammer.

A limitation of our experimental testing study was that the measurements were not conducted completely according to ISO guidelines [19]. The ISO guidelines call for having workers do drilling rather than using an automated experimental testing system. The advantage of having workers do drilling is that they can apply varied grip and push forces according to their familiarity with the tool, thereby producing a wide range of friction forces similar to those that workers might be exposed to in the real world. However, the variability is also a disadvantage and may introduce worker bias, for example, workers may apply different push forces for a pneumatic hammer than another test. With a test bench, the applied force magnitude and direction are the same for all tests. Although the friction forces obtained in the standardized condition of the test bench may not be fully representative of real exposures, it is widely accepted that the lower the friction force of the tool in a test bench study, the lower the exposure will be in real operations. Another limitation for the pressure or acceleration sampling is the collection inside chambers of cylinder. Therefore, measurements are difficult to be conducted in field test. However, this condition is suitable for laboratory equipment.

5. Conclusion

In this article, we have proposed a novel approach to determine the impact performance of a small pneumatic hammer considering the friction force.

(I) A numerical analysis for evaluating the performance of pneumatic hammer without the friction force was carried out. Using AMEsim model for pneumatic hammer, the displacement, velocity, and acceleration curves of impact piston, as well as the characteristic curves for pressure variation in the piston chambers were obtained. The result of numerical analysis showed that the proposed model has fidelity and feasibility to predict ideally the impact energy and frequency of pneumatic hammer.

(II) The experimental setup for evaluating friction force of the pneumatic hammer was introduced. Friction force increased from 10.27 to 16.7 N due to increase in inlet pressure at 4–7 bar and mechanical power loss in the pneumatic hammer was more than 10% of impact energy. Therefore, it can be concluded that the friction force between parts in relative motion should be considered significantly in evaluating the impact energy of fluid hammer.

(III) A novel approach to consider the friction force for evaluating performance of pneumatic hammer was reported. This approach can predict not only the impact performance with good accuracy but also service life of the pneumatic hammer.

In the future, we will carry out further study on the piston design with proper surface texture to reduce friction and prevent wear.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest regarding the publication of this article.
Acknowledgments

The authors gratefully acknowledge the financial support from the National Science and Technical Development Foundation of DPR Korea (Grant nos. 02017548–455).

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