Analysis of the dynamic characteristics of gas chamber in rotary hammer

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Abstract
Rotary hammer is a high-frequency impact machine with a complicated gas chamber. The design parameters of the gas chamber are dominating to impact energy output and impact efficiency of a rotary hammer. In this paper, the mathematical dynamic analysis and fluid dynamic simulations were presented to analyze the influences of the sensitive design parameters on impact energy output and impact efficiency. The dynamic model of the gas chamber consists of kinematic model of impacting piston, status parameter equations of mass-variable gas volume, flow rate equation of compensation orifices, and the gas leakage equation of rubber sealing rings. A testing frame for measuring the impact energy output was designed, and the results of dynamic analysis were in good agreement with the experimental results by the testing frame. Also a successful drilling efficiency improvement of a mass production hammer was completed by adopting the valuable suggestions from this dynamic analysis. The dynamic characteristics analysis of gas chamber in this paper can provide technical supports for hammer design and parameter optimization.

Key words: Thermodynamic, Fluid simulation, Impact efficiency, Gas chamber

1. Introduction
The rotary hammer is a type of hand-held rotary impact tool that is widely used in construction sites. It utilizes its gas volume to output alternating impact energy in order to make drills or slots in concrete, rocks, masonry, metal or other materials. The gas volume between driving piston and impacting piston is mass variable and can be treated as a gas spring with energy storage function, which is the dominating design for impact energy output and impact efficiency of the rotary hammer.

The nonlinear springs, dry friction and stress waves are important concerns for the vibration characteristics of rotary hammers, and the gas pressure directly determines the drilling efficiency for rock material (Soundranayagam, 1999). The mass of impacting piston and the impact frequency also have significant effect on the impact energy output (Zhang, 2000). The gas compensation orifice is essential to continuous impact running, and no impacts as a result of the poor design of gas compensation was found in the actual operation (Wu, 2000). The ratio of drill bit’s rotating frequency to impact frequency is essential to drilling efficiency, and the optimum ratio should be made according to the blade edge of drill bit as well as the mechanical properties of drilling material (Bilgin, 2006 and Meixner, 2013). The optimum operating condition (gas pressure and impact frequency) of rotary hammers is variable according to different rock strength (Ryu, 2015), and the drilling efficiency can be evaluated in terms of mechanical specific energy of a rock with specific strength (Hamrick, 2011 and Chen, 2014). EPTA 05/2009(10) regulations of stress wave method were implemented to analyze the dynamic characteristics of rotary hammers (Nambu, 2011 and Will, 2013), and the vibration characteristics of the rotary hammer under loading condition is directly related to the dynamic characteristics of gas chamber (Will, 2013). Most of the experimental studies focused on the machine vibration control, size optimization of mechanical mechanism or impact efficiency improvement, but rare studies have been made about the alternating impact mechanism of gas chamber and its dynamic characteristics, which are fundamental knowledge for parameter design and performance optimization of the rotary hammer.
In this paper, the mathematical dynamic analysis and the fluid dynamic simulations were made to analyze the influential laws of sensitive design parameters of gas chamber on impact energy output and impact efficiency. The dynamic characteristics of gas chamber were theoretically studied by aerodynamics theory of gas volume as well as the kinematics of pistons, also the measuring test of impact energy output was carried out with a self-prepared testing frame. Then a successful drilling efficiency improvement of a mass production hammer was completed by adopting the valuable suggestions from this dynamic characteristics analysis of gas chamber.

2. Impact mechanism of gas chamber

The impact mechanism of the rotary hammer is schematically illustrated in Fig.1. The driving piston is powered by a crank-connecting rod connected to the motor-gear system, it makes sinusoidal reciprocating motion, then the gas volume in chamber comes into a cyclic status of alternately compression-expansion. The gas pressure in chamber is then in a similar sinusoidal variation according to energy conservation theory. The axial pressure difference resulted from the variable gas pressure in chamber drives the impacting piston to make cyclic mechanical impacts on anvil, the anvil transfers its kinematic energy to drill bit in form of mechanical shock wave, the shock waves from the mechanical collision ensure the effective drilling or slotting into concrete blocks, rocks or other construction materials.

![Figure 1 Schematic of impact mechanism](image)

The impact procedure of gas chamber consists of two stages. The first is the stroking motion of impacting piston, in which the gas volume is mainly compressed, the higher gas pressure in chamber gives pushing force on impacting piston until reaching impacting piston’s maximum velocity at the last position of stroking stage, then the high-velocity impacting piston makes mechanical collision with anvil and the impact energy of the gas chamber is outputted in form of stress waves. The little elastic rebound of impacting piston after mechanical collision indicates the beginning of returning stage of impacting piston, meanwhile the driving piston has already in the backward motion with a higher velocity than rebound velocity, so the gas volume in chamber begins to be dilated and the decreasing gas pressure drags the impacting piston into returning motion. The lower gas pressure in chamber results in the higher acceleration of impacting piston’s returning motion, the returning velocity of impacting piston keeps on increasing until the gas pressure in chamber is equal to environmental pressure outside the chamber. When the velocity of impacting piston is higher than that of driving piston, the distance between impacting piston and driving piston begins decreasing, and the gas pressure in chamber is increasing. The impacting piston ends its increasing returning motion at the time when the gas pressure in chamber is equal to environmental atmosphere pressure. As result of kinematic inertia, the further returning motion of impacting piston compresses the gas volume in chamber and results in rapid pressure increase in chamber, and the axial pressure difference acting on impacting piston gradually reduces the returning velocity. So the gas volume in chamber functions as a gas spring that can transform the kinematic energy of moving impacting piston into internal energy stored in compressed gas. The internal energy reaches its maximum at the time of minimum distance between impacting piston and driving piston. The stored internal energy is then transformed into kinematic energy of impacting piston in stroking stage of the next impact procedure. The repeated energy transformation powered by the crank-connecting rod ensures the cyclic impact operation of rotary hammer.

Three functional orifices are designed for gas chamber to ensure the continuous impact operation and precise impact energy output, including gas compensation orifice, stroke-assuring orifice and empty-impacting orifice, as shown in Fig.1. The gas compensation orifice supplements external air from environmental atmosphere to make up the gas leakage at rubber sealing rings, in order to keep the mass balance in gas volume for each impact procedure. Stroke-assuring orifice makes the gas connection between environmental atmosphere and gas chamber in order to...
eliminate the additional gas pressure resistance on the moving impacting piston, then the returning motion of impacting piston can be successfully activated. The empty-impacting orifice with large flow section area, that functions in case of light-load or no-load conditions, makes gas connection between environmental atmosphere and another gas chamber in front of the impacting piston, then the pressure difference on impacting piston completely disappears, and the impact mechanism switches into lock mode without impact energy output.

3. Mathematical modeling
3.1 Modeling conditions

The dynamic model of gas chamber comprises kinematic model of impacting piston, status parameter equations of mass-variable gas volume, flow rate equation of orifices in different flow status and the gas leakage equation of rubber sealing rings. The integrated dynamic model is presented on the basis of the following hypothesis: (1) Ideal gas in gas chamber. (2) The thermodynamic dynamic is a quasi-static process, and the state parameters of gas volume are described in state equations of ideal gas. (3) Adiabatic passage without heat exchange with outside environment is treated in the thermodynamic model, because the period for each impact procedure is usually less than twenty milliseconds. (4) The earth gravity is considered in the dynamic model, because the rotary hammer is usually vertically supported by worker’s hands in actual operation. (5) The mechanical collision between impacting piston and anvil is regarded as rigid mechanical collision without energy dissipation, because the pistons and anvil are made of high-strength steel and the Rockwell hardness after heat treatment is over 65.

3.2 Mathematical model
3.2.1 Kinematic equation of driving piston

The driving piston is driven by the rotating crank with an eccentric radius. The kinematic equation of driving piston is given by,

$$\frac{dS_2}{dt} = R_0(1 - \cos \omega t)$$

where \( S_2 \) is displacement of driving piston, \( R_0 \) is the eccentric radius of the crank, and \( \omega \) is rotate velocity of the crank.

3.2.2. Gas leakage equation of rubber sealing rings

The rubber sealing rings, as shown in Fig.2, ensure the effective sealing for moving pistons. The rubber sealing rings endure long-time high temperature and significant mechanical wear. In condition of the pressure difference between the gas volume in chamber and the environmental atmosphere, gas leakage occurs at the sealing ring’s assembly units due to the mechanical clearances as a result of the mechanical wear and initial assembly clearance. The gas leakage model of rubber sealing rings is based on the following hypothesis (Picard, 2016): (1)The leaking flow is Laminar flow with constant viscosity coefficient. (2)The leaking flow is uncompressible. (3)The leaking flow comprises differential pressure flow and shear flow.

Gas leakage equation of rubber sealing rings is given by,

$$Q = \frac{\pi Dh^3 \Delta p\left(1 + 1.5\varepsilon^2\right)}{12\eta L} + \frac{\pi Dhv_c}{2}$$

Figure 2 Section schematic of rubber sealing ring assembly
where $D$ is the diameter of gas volume in chamber, $h$ is effective clearance for gas volume’s radius, $\Delta p$ is pressure difference between gas volume and environmental atmosphere and $\Delta p = |p_0 - p_1|$, $p_e$ is environmental atmosphere pressure and $p$ is pressure of the gas volume in chamber, $\eta$ is dynamic viscosity of leaking flow, $L$ is the flow path length at sealing ring assembly units, $\varepsilon$ is eccentric coefficient of pistons and $0 < \varepsilon < 1$, $v_r$ is the relative velocity of the moving pistons to chamber body, $v_r = dS_r/dt$ is for driving piston, and $v_r = dS_i/dt$ is for impacting piston, $S_i$ is axial displacement of impacting piston.

### 3.2.3 Resistant force equation of impacting piston

Due to the assembly contact with chamber body, the resistant force on impacting piston comprises flow viscosity resistance and mechanical contact friction.

$$f = \pi D L \left( \frac{\Delta p h}{2L} + \frac{\mu v_r}{h\sqrt{1 - \varepsilon^2}} \right) + \mu \pi D p,$$

where $\mu$ is frictional coefficient of mechanical contact between rubber sealing rings and chamber body, $l$ is the axial width of mechanical contact area, $p_\text{avg}$ is the average pressure in mechanical contact area.

### 3.2.4 Flow rate equation of compensation orifices

According to the small orifice flow theory, the gas flow rate of compensation orifice is mainly determined by flow section area, orifice’s dimensions, pressure difference and the flow status around the orifices(Usuzawa, 2014). The flow rate equation of compensation orifice is given as the following equations,

$$\Delta c = \begin{cases} 
\alpha A \sqrt{K \frac{2}{K+1} \frac{p_\text{in}}{R_g T_e}} & 0 < \frac{p_\text{out}}{p_\text{in}} \leq 0.528 \\
\alpha A \sqrt{2K \frac{p_\text{in}^2}{K-1 R_g T_e} \left( \frac{p_\text{out}}{p_\text{in}} \right)^{2-K} - \left( \frac{p_\text{out}}{p_\text{in}} \right)^{K+1}} & 0.528 < \frac{p_\text{out}}{p_\text{in}} \leq 0.9 \\
\alpha A \frac{2(p_\text{in} - p_\text{out})}{R_g T_e} & 0.9 < \frac{p_\text{out}}{p_\text{in}} < 1 \\
0 & \frac{p_\text{out}}{p_\text{in}} = 1 
\end{cases}
$$

where $\alpha$ is flow coefficient of orifice, $A$ is effective area of flow section, $K$ is specific heat ratio of gas, $R_g$ is gas constant of ideal gas and $R_g = 287.1 J/(Kg \cdot K)$. $T_e$ is the temperature of flow gas, $p_\text{in}$ is gas pressure of upstream flow, $p_\text{out}$ is gas pressure of downstream flow.

### 3.2.5 Effective flow section area of compensation orifice

The relative position of impacting piston to the compensation orifice determines the effective flow section area of compensation orifice. When the back end-face of impacting piston is at the back of the compensation orifice, the compensation orifice is fully closed, otherwise is fully open or partly open as shown in Fig.3. Sign function is used to indicate whether the compensation orifice is open or not.

$$\text{sgn}(S_i) = \begin{cases} 
1 & 0 \leq S_i < L_h + d/2 \\
0 & S_i \geq L_h + d/2 
\end{cases}$$

The effective flow section area of compensation orifice is given as the following equations,
where \( d \) is the diameter of compensation orifice cylinder, \( L_s \) is axial distance between the back end-face of impacting piston and the centerline of compensation orifice cylinder when at the beginning of returning motion, \( L_h \) is axial distance between impacting piston and driving piston when at the beginning of returning motion, that is the initial height of gas volume.

As shown in Fig.4, the origin of the displacement coordination is located at the initial returning position of impacting piston, and the positive vector follows the returning velocity of impacting piston.

### 3.2.6 Volume equation of gas in chamber

The gas volume in chamber is variable due to the displacement difference between driving piston and impacting piston, the volume equation of gas in chamber is given by,

\[
V = \pi D^2 \left( L_0 + S_2 - S_1 \right)/4
\]

\((7)\

### 3.2.7 Pressure equation of gas in chamber

According to the first law of gas thermodynamics and energy conservation theory, the pressure equation of gas in chamber can be deduced as follows,

\[
\frac{dU}{dt} + h_e \frac{dm}{dt} + p \frac{dV}{dt}
\]

\((8)\

where \( U \) is thermodynamic energy of the gas in chamber, \( h_i, h_e \) are the gas specific enthalpies of inflow and
outflow respectively, \( m_i \) and \( m_e \) are the inflow mass and outflow mass respectively. The inflow and outflow occur at both compensation orifice and sealing ring assembly units.

It is known that \( U = mc_vT_e \), it can be deduced as follows,

\[
\frac{dU}{dt} = mc_v \frac{dT_e}{dt} + c_v \frac{dm}{dt}
\]

(9)

where \( c_v \) is the specific heat capacity at constant volume.

It is known that \( pV = mR_gT_e \), it can be deduced as follows,

\[
\frac{dV}{dt} = \frac{R_g}{p} \left( T_e \frac{dm}{dt} + m \frac{dT_e}{dt} \right) - \frac{V}{p} \frac{dp}{dt}
\]

(10)

The temperature differential for outflow from gas chamber is nearly equal to zero, that is

\[
\frac{dT_e}{dt} = 0
\]

(11)

From Meyer formula,

\[
\begin{align*}
\frac{c_p}{c_v} &= \frac{KR_g}{K - 1} \\
c_v &= \frac{R_g}{K - 1}
\end{align*}
\]

(12)

where \( c_p \) is specific heat capacity at constant pressure, and \( c_p = 1 \) for ideal gas.

From eq. (8), (9), (10), (11) and (12), the pressure equation of gas in chamber is deduced by,

\[
\frac{dp}{dt} = \frac{K - 1}{V} \left( h_i \frac{dm_i}{dt} - h_e \frac{dm_e}{dt} \right) - \frac{Kp}{V} \frac{dV}{dt}
\]

(13)

3.2.8 Temperature equation of gas in chamber

According to adiabatic process equation, the temperature equation of gas in chamber is given by,

\[
p^{1-k}T^K = \text{const}
\]

(14)

The gas temperature in chamber can be deduced form eq.(13) and (14),

\[
\frac{dT}{dt} = \frac{T(K - 1)^2}{pKV} \left( h_i \frac{dm_i}{dt} - h_e \frac{dm_e}{dt} \right) - \frac{T(K - 1)}{V} \frac{dV}{dt}
\]

(15)

where \( T \) is the temperature of gas in chamber.

3.2.9 Kinematic differential equation of impact piston

The kinematic differential equation of impacting piston is given by,

\[
m_i \frac{d^2S_1}{dt^2} = \frac{\pi D^2(p_0 - p)}{4} - f - m_i g
\]

(16)

where \( m_i \) is the mass of impacting piston.

3.2.10 Impact energy output

The kinematic energy of moving impacting piston represents the impact energy output from gas chamber, and should reach the maximum before the mechanical collision with anvil.

\[
W = 0.5m_i \left( \frac{dS_1}{dt} \right)^2
\]

(17)

where \( W \) is the impact energy output of gas chamber.

4. Result and discussion

4.1 Mathematical modeling results
Fourth-order Runge-Kutta method was used in Matlab code to solve the mathematical dynamic model of gas chamber in section 3. The detailed parameters of DRT-DW518 rotary hammer are listed in Table 1. The curves of the status parameters in gas chamber, such as gas pressure, mass, inflow or outflow, were plotted to deeply understand the impact mechanism in gas chamber, as well as to analyze the influential laws of sensitive design parameters of gas chamber on impact energy output and impact efficiency.

| $R_e$ | $\omega$ | $D$ | $L_0$ | $d$ | $m_i$ | $h$ | $L$ | $l$ | $p_c$ | $\mu$ | $\varepsilon$ |
|------|---------|------|------|-----|------|-----|-----|-----|-------|-------|--------|
| mm   | rad/s   | mm   | mm   | mm  | Kg   | mm  | mm  | mm  | MPa   |       |        |
| 12.5 | 314.89  | 24.8 | 21.7 | 29.97 | 1.0  | 0.084 | 0.07 | 6.52 | 1.33 | 0.041 | 0.035  | 0.014  |

As shown in Fig.5, the acceleration of impacting piston is zero at the beginning of returning motion, because there is no pressure difference applied on impacting piston. The initial returning velocity of impacting piston is nearly zero, as shown in Fig.6. Then the distance between impacting piston and driving piston (shown in Fig.7) is increasing, which results in negative gas pressure in chamber. The negative pressure is continuously decreasing regardless of the gas inflow through the compensation orifice, as shown in Fig.8. The pressure difference drives the impacting piston into an increasing returning motion with a lower acceleration. At the critical time when impacting piston has equal returning velocity to driving piston, the distance between impacting piston and driving piston reaches its maximum in returning stage, which means maximum gas volume and minimum gas pressure in chamber (shown in Fig.8). Then the returning velocity of impacting piston, which is higher than that of driving piston, begins to decrease the gas volume and increase the gas pressure in chamber. When the increasing gas pressure is equal to environmental atmosphere pressure, the pressure difference on impacting piston disappears. Hence, the impacting piston reaches the maximum returning velocity and ends its increasing motion. It can be seen from Fig.5 that the increasing returning motion of impacting piston lasts for as long as nearly fourteen milliseconds, and the actual period of a complete impact procedure is about twenty milliseconds. Then the impacting piston continues returning motion with a decreasing velocity, the gas volume in chamber turns into compression status and reaches its maximum at the minimum distance between impacting piston and driving piston. The dramatically increasing pressure can reduce the impacting piston from the maximum returning velocity to zero at its returning displacement limit in about 2 milliseconds. Then the maximum gas pressure in chamber drives impacting piston into reverse direction motion with the highest stroking acceleration, the stroking velocity will instantly increase up to as high as 9.87 m/s in about 2.3 milliseconds.

As shown in Fig.6, significant velocity variation occurs during the time from 14 millisecond to 18 millisecond, the impacting piston switches its maximum returning velocity to maximum stroking velocity. The pressure fluctuation in gas chamber functions as a variable stiffness of a gas spring that transforms the internal energy stored in gas volume into kinematic energy of impacting piston and the heat energy for compressed gas in chamber.

Figure 5 Acceleration of impacting piston vs. time      Figure 6 Velocity of impacting piston vs. time
As shown in Fig. 9 and Fig. 10, due to the highest pressure of the gas volume, the impacting piston switches into increasing stroking motion with the highest acceleration. And over eighty percent of internal energy stored in compressed gas has been transformed into kinematic energy of impacting piston before the full opening of compensation orifice. As shown in Fig. 12, the gas outflow through compensation orifice can significantly reduce the gas pressure in chamber when the compensation orifice is fully opened. In the remaining motion after full opening of compensation orifice, the impacting piston will keep on nearly uniform velocity as shown in Fig. 10, this is because the gas pressure in chamber is a little higher than environmental atmosphere and can hardly make any additional pressure difference on impacting piston.

For a most efficient gas chamber, the impacting piston should has gain its maximum stroking velocity a little time ahead or just at the time of mechanical collision with anvil. The time period from maximum stroking velocity to mechanical collision with anvil should be as short as possible to prevent the unnecessary energy dissipation caused by mechanical friction, and more kinematic energy of impacting piston can be outputted into anvil. From the above analysis, impact efficiency can be effectively improved by increasing the compression pressure in gas chamber, extending the time for accelerating impacting piston during stroking and improving lubrication condition.

The effective flow section of the compensation orifice is determined by the relative position of impacting piston to chamber body (shown in Fig. 13), and the compensation orifice is open to environmental atmosphere for gas inflow.
during returning motion and gas outflow during stroking motion. The time of gas outflow is shorter than that of inflow, but the outflow rate is much larger than inflow rate (shown in Fig.14), which can be explained by the higher pressure in chamber during stroking as shown in Fig.12.

4.2 Fluid dynamic simulation

The gas chamber was modeled in exact accordance with DRT-DW518 rotary hammer in section 4.1. As shown in Fig.15, volume extension of the orifice was made in order to get stable flow around the orifices, the interface between gas volume and chamber body was treated as attach interface, the surfaces of the gas volume extension were environmental pressure and other walls were adiabatic. 174060 hexahedral elements were made within the gas volume by plane mesh extruding method (PMEM) in Pro-star software. More refined elements around the compensation orifice and attach interface were meshed to guarantee the mesh-independency, the size of the refined elements is 1/8 of the orifice diameter. Adaptive moving grid technology (AMGT) was employed in this transient fluid dynamic simulation. The gas in chamber is assumed to be ideal gas. The solving code is SIMPLE algorithm of STAR-CD codes, the relaxation factor is 0.1 and the solving time step is 0.001 millisecond. The velocity equation of driving piston and the kinematic differential equation of impacting piston were programmed in a MACRO file and integrated into STAR-CD codes.

After the fluid dynamic simulation, four typical gas statuses in chamber, which are at 2 millisecond, 10 millisecond, 17 millisecond and 18.5 millisecond, were selected to analyze the gas velocity, pressure and temperature distribution in chamber. Flow velocity in the chamber is shown in Fig.16. The gas inflow through both compensation orifice and stroke-assuring orifice is activated by the negative gas pressure in chamber. The increasing distance between impacting piston and driving piston leads to increasing inflow rate, and the inflow rate reaches its maximum just at the time of the closing of compensation orifice. The outflow reaches its maximum as high as 278 m/s at 17.2 millisecond when the compensation orifice begins to open. At 18.5 millisecond, the outflow velocity is as low as about 2.8 m/s. As the continues of the stroking motion of impacting piston, there is no outflow from gas chamber because the gas pressure in chamber is equal to the environmental atmosphere pressure.
As shown in Fig.17, the gas pressure in chamber is relatively uniform. The negative pressure at 2 millisecond plays an important role in activating the returning motion of impacting piston. The highest pressure in chamber at the end of returning motion indicates the maximum internal energy stored in gas volume. The disappearance of positive gas pressure in chamber, at the time of just 1.3 milliseconds ahead of the mechanical collision between impacting piston and anvil, means the end of accelerating stroking motion.

Nonuniform temperature distribution in chamber is shown in Fig.18. At 10 millisecond, the average temperature in chamber is about 11 degrees lower than environmental temperature, and great temperature gradient is found around the compensation orifice. At 17 millisecond, the average temperature increases up to 103 degrees higher than environmental temperature and the temperature gradient is approximately eliminated. At 18.5 millisecond, the average temperature is about 5 degrees higher than environmental temperature, and the temperature distribution is similar to that at 2
millisecond.

The velocity of impacting piston deduced from fluid dynamic simulation is shown in Fig.19. The velocity curve indicates a slight fluctuation in the five impact cycles, and the fluctuation range for the maximum stroking velocities is below 4.7 percent.

4.3 Experimental results and analysis

Velocity measuring method was used to trace the changing velocity of impacting piston in impact cycles. As shown in Fig.20, the self-prepared testing frame comprises of signal detecting unit with a PR-M/F laser optical sensors made by Keyence company, data processing system and steel frames. The sampling frequency of the laser optical sensor is 52600Hz. A DRT-DW518 rotary hammer drill was installed on the testing frame, and had to make a loading operation for 15 minutes before data recording in order to get accurate and reliable data. 23 sets of measured maximum stroking velocity were sampled to calculate the average stroking velocity. The average of maximum impact energy output of DRT-DW518 is 10.2 Joules and the impact frequency is 50.13Hz.

A slight data fluctuation in maximum stroking velocities can be found in the velocity curve in Fig.21, the fluctuation range is within 6%, which is probably caused by the elastic deformation of testing frames under impacting load or the data acquisition error of laser optical sensor.
As shown in Fig.22 (a), the velocity curve form experimental test is not as smooth as velocity curves from fluid dynamic simulation and mathematical model, and a little lower during the returning motion. The special characteristics of the experimental curve can be explained by the variable frictional conditions at contact surface between rubber sealing rings and chamber body during the low-velocity returning motion. On the whole, the three curves share the same shape and trends, also indicate reasonable consistency with each other. The maximum impact energy values of DRT-DW51 deduced from the maximum stroking velocity are shown in Fig.22 (b), which demonstrates reliability or high accuracy of the mathematical model and the fluid dynamic simulation when evaluating the impact energy output of rotary hammers.

5. Application

Two rotary hammers with equal input power for gas chamber were selected in a drilling efficiency test. Hammer I is a market sales machine with high performance reputation. Hammer II is a newly developed machine in trial production. Hammer I is the design benchmark of hammer II. Both hammers share the same design parameters except the gas chamber. It was found in the drilling efficiency test that the drilling efficiency of hammer II is about 24% lower than Hammer I. The detailed test procedures are as follows: as shown in Fig.23, making hole drilling into the C30 concrete block with SDS-Plus S5L drill bits manufactured by BOSCH company. Two types of drill bits, diameter of 6 mm and 8 mm, were sequently employed to drill holes of 50mm depth. The C30 concrete block was prepared according to GB50010-2010 Code for Design of Concrete Structure Issued by Ministry of Housing and Urban-rural Development of People’s Republic of China. Both hammers were powered by a DC regulated power supply and vertically supported under 40N from worker’s hands on hammer body. 12 sets of time duration for each hole drilling were recorded and averaged for drilling efficiency evaluation.
The optimization study on design parameters of gas chamber in hammer II was made to improve the energy conversion efficiency of gas chamber. Considering time saving and convenient modification work during machining or grinding process of validating samples, two main design parameters were chosen in this optimization application, including the diameter of gas volume in chamber and the mass of impacting piston.

The velocity curves of different mass in Fig.24 show that higher mass results in smaller acceleration according to Newton’s law, the smaller returning displacement of impacting piston leads to less gas compression in chamber, the insufficient internal energy stored in compressed gas can not provide satisfactory impact energy output. The original mass of impacting piston in hammer II is 93.6 grams, its accelerating motion is not yet finished at the time of mechanical collision with anvil.

The velocity curves of different gas volume diameters in Fig.25 show that the maximum stroking velocity of impacting piston increases with the diameters of gas volume in chamber. Higher diameter increases the gas mass in chamber, more internal energy can be stored in gas volume, then more impact energy output is guaranteed. The original diameter of gas volume in hammer II is 24 mm. If the diameter increases up to 26.5 mm, the impacting piston can reach its maximum stroking velocity at the time just 0.84 milliseconds before the mechanical collision with anvil, which is a preferable matching design.

Three optimized designs were evaluated in the physical validation test, which are hammer III with lighter impacting piston of 83.6 grams, hammer IV with lighter impacting piston of 73.6 grams, and hammer V with lighter impacting piston of 83.6 grams and larger diameter of 25.5 mm. The assembling components of hammer V are shown in Fig.26.

The five hammers were sequently tested on a same concrete block, the test procedure was same to the former drilling efficiency test. As shown in Fig.27, the average drilling time illustrates that hammer V is the best among three optimized designs, and a little more efficient than Hammer I. But the vibration acceleration at handle structure is 8.3% higher than Hammer I, which is probably caused by higher rebound force from drill bit because of the increased impact power output.
6. Conclusions

The design parameters of gas chamber in rotary hammer are very important to impact energy output and impact efficiency. The gas volume in chamber functions as a gas spring that can make the energy transformation between the kinematic energy of impact mechanism and the internal energy stored in gas volume. The mathematical dynamic model and fluid dynamic simulations were made to analyze the influences of sensitive design parameters of gas chamber on impact energy output and impact efficiency. It was found that the most sensitive parameters are mass of impacting piston, diameter of gas volume in chamber, compensation orifice dimensions and its axial position along gas volume, and the rotation velocity of crank. A test frame for measuring impact energy output was designed and manufactured, and a DRT-DW51 hammer was chosen in the experimental test on the testing frame. The impact energy values of impacting piston obtained from mathematical model, fluid dynamic simulation and experimental test indicated good consistency with each other, which demonstrated the reliability or high accuracy of the mathematical model and the fluid dynamic simulation. A new-designed rotary hammer in trial production with lower drilling efficiency was parametrically improved by the optimization of two parameters, which were mass of impacting piston and diameter of gas volume. The optimized hammer V with lighter impacting piston of 83.6 grams and larger diameter of 25.5 mm was verified to be optimal design, and a little more efficient than benchmark tool. The dynamic characteristics analysis of gas chamber can provide powerful technical supports for hammer design and parameter optimization, and also reduce the cost for product development.

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