Modeling and simulation of dynamic performance of horizontal steam-launch system

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Abstract: Based on theory of variable-mass system thermodynamics, the dynamic mathematic models of each component of the horizontal steam-launch system were established, and by the numerical simulation of the system launching process, the thermodynamics and kinetics characteristics of the system with three valves of different flow characteristics were got. The simulation results show that the values of the peak-to-average ratios of dimensionless acceleration with the equal percentage valve, the linear valve and the quick opening valve are 1.355, 1.614 and 1.722, respectively, and the final values of the dimensionless velocities are 0.843, 0.957 and 1.0, respectively. In conclusion, the value of the dimensionless velocity with the equal percentage valve doesn’t reach the set value of 0.90 when the dimensionless displacement is 0.82, while the system with the linear valve can meet the launching requirement, as well as the fluctuation range of dimensionless acceleration is less than that of the quick opening valve. Therefore, the system with the linear valve has the best performance among the three kinds of valves.

Key words: horizontal steam-launch system; thermodynamics and kinetics characteristics; modeling and simulation

1 Introduction

The horizontal steam-launch system is a power plant which transforms the internal energy of steam into the kinetic energy of load for launch in extremely short time. Similar launching power system have been widely used in many fields, such as the ejection power plant for occupant ejection [1−3], the ejection launcher for missile launch [4−5], and the steam-power launch system for carrier-based aircraft catapult launch [6−8], etc. Some researches also have been done on the launching power system. For example, based on the analysis of working process of ejection gun of rocket ejection seat, Zhai et al [1] established the physical and mathematic models of interior ballistics of ejection gun and the interior ballistics parameters of a rocket eject seat were validated by test and calculation, the experimental results indicated that the measured chamber pressure curve was consistent with the calculation curve, namely, the models can be used to simulate the working process of the ejection process. Chen et al [5] established the ejecting interior ballistic equations of one submarine-launched cruise missile. The interior ballistic equations were simulated and computed, and the basic changing rules of interior ballistic parameters were obtained, the results indicated that gas-steam launch mode can be used to launch submarine cruise missile. Qi et al [6] established the inner launch-trajectory equations and vapor-discharge equation for deck-landing aircrafts, and then simulated the aircraft launch-off process, according to the required design indexes of inner launch-ballistic and the take-off indexes of aircraft. The simulation results demonstrated that the inner launch-ballistic dynamic parameter and the vapor-discharge regularity were able to satisfy the design indexes, and this method can provide theoretical basis for the engineering design of aircraft ejection power plant. Bai [7] presented a mathematic model for the steam launching interior trajectories and proved that the total energy into working chamber in energy equation set should be the steam enthalpies in the tan. In order to deal with the complication by variable-mass steam working and steam medium, computer programming was formed to inquire the steam tables and to find the solutions of the equation set of the interior trajectories. Moreover, Cheng et al [8] established a dynamic mathematic model of a naval steam-power aircraft launch system to study the dynamics and thermal process feature of the system and obtain the relationship between piston velocity, acceleration, displacement and time by the software of MINIS, then the influence of steam leakage was also
discussed. The numerical results indicated that the mathematic model can provide the optional method and basis for the design and test of the naval catapult in future. However, some assumptions were made for simplifying research in these works [6–8] before the process model was established. For example, the change of the state parameters of steam in the steam accumulator was neglected and only the thermal process of steam in the cylinder was studied. Moreover, the launching valve of the system was simplified as an orifice to calculate the steam flow. Therefore, the influence of control strategy and flow characteristics of control valve on the system were neglected.

Actually, the volume of the steam accumulator is not far bigger than that of the cylinder, so the pressure in the steam accumulator drops obviously and it is necessary to study the thermal process of steam in the steam accumulator. For the establishment of the thermodynamic model in the cylinder, both variation of the pressures and volumes in the cylinder may change the steam state from superheated steam into saturated steam, so it is needed to determine the state of steam in the cylinder at first. At the same time, it should be noticed that the launching valve is the key part of the system, which controls the steam flow from the steam accumulator to the cylinder by regulating the valve opening, and then determines the launching performance. Obviously, a rational choice for the flow characteristic and control strategy of the launching valve has an important effect on improving the launching performance of the system, whereas it wasn’t taken into consideration among Refs. [6–8]. Therefore, the analysis for its influence on the launching performance needs the establishment of the flow characteristic model of the launching valve. Furthermore, during the thermodynamic process of the system, the potential stages of choked flow or un-choked flow need to be determined, because of the variation of pressure differences of the launching valve.

Based on the complicated problems that the discharge process of the steam accumulator and the charge process of the cylinder are variable-mass process [9], the calculation for the steam flow through the launching valve needs the determination for the potential stages of choked flow or un-choked flow, the parameters of each component of the system that connected each other by the steam, as well as the state of the steam in the cylinder needs to be determined, the models of each component of the system are established, and the thermodynamics and kinetics characteristics of the system are numerically simulated. Furthermore, the performances of the system with the valves of different flow characteristics are discussed, which provides guidance for the initial design and experiment of the system.

2 Object overview

The horizontal steam-launch system consists of steam accumulator, launching valve, cylinder, piston subassembly, load and the steam pipes connecting them, which is shown in Fig. 1.

The steam accumulator is a pressure vessel storing hot water and steam for heat release. The lower part of the steam accumulator is a liquid space and the upper part is a vapor space [10–12]. At runtime, the saturated steam from the upper part of the steam accumulator flows into the cylinder in an extremely short time through the launching valve, then the steam expands in the cylinder and thrusts upon the piston subassembly, which accelerates the connected load under a linear motion as steady as possible in a stated time and reaches the set value of final velocity in a stated displacement.

During the launching process, under the pressure difference between the steam accumulator and the cylinder, the steam from the vapor space of the steam accumulator flows into the cylinder through the launching valve. At the same time, the pressure drop of the steam accumulator generates the superheated water that the temperature is higher than the corresponding saturation temperature. Simultaneously, the superheated water self-evaporates into the saturated steam instantaneously. As the steam flows into the cylinder constantly, the pressure variations of both the steam accumulator and the cylinder change the pressure difference of the launching valve, which may change the steam stage of the launching valve from the choked flow to the un-choked flow, so it is necessary to be determined.

![Fig. 1 Basic structure chart of horizontal steam-launch system: 1—Steam accumulator; 2—Steam pipe; 3—Launching valve; 4—Cylinder; 5—Piston subassembly; 6—Load](image-url)
At the same time, the pressure in the cylinder established by the steam transforms into the power constantly which makes the piston and load subassembly move ahead, so the volume of the cylinder changes all the time. As a result, the steam in the cylinder may have the state of superheated steam or saturated steam, which needs the determination of steam state by the thermodynamics model in the cylinder and IAPWS-IF97 [13] (The International Association for the Properties of Water and Steam-Industrial Formulation 1997) for the thermodynamic properties of water and steam.

It can be seen from the above analysis that each component of the system has a direct influence on the dynamic characteristics of the system. So, it is necessary to establish the dynamic mathematical models of each component to study the thermodynamics and kinetics characteristics of the system, which includes the generation and flow rules of steam, the change rules of the acceleration, the velocity and displacement of load, for the performance evaluation of the horizontal steam-launch system with the valves of different flow characteristics.

From the basic structure of the system, it can be seen that the system can be divided into four sub-modules: steam accumulator module, launching valve module, cylinder module, piston subassembly and load module. These modules are connected with some coupling parameters, for example, the calculation of steam flow in the launching valve module needs the data input of pressure in the cylinder module, on the contrary, the launching valve module should input the data of steam flow to the cylinder module for obtaining the pressure. Therefore, the parameters of each module can not be calculated independently, which need to be calculated iteratively for decoupling.

### 3 System modeling

In Fig. 1, $\tau$ is the time variable (s), $P_0$ and $T_0$ are respectively the wet steam pressure (MPa) and temperature (K) of the steam accumulator, $h_{0v}$, $\rho_0$, $V_0$ and $a$ are respectively the specific enthalpy (kJ/kg), density (kg/m$^3$), volume (m$^3$), and water filling coefficient of working fluid in the steam accumulator. Superscript “” and “” represent the liquid and vapor space parameters of wet steam, respectively. $Q$ is the instantaneous discharge steam flow (kg/s); $P_c$, $h_c$, $\rho_c$ and $V_c$ are respectively the pressure, specific enthalpy, density and volume in the cylinder; $D_p$ is the diameter of piston (mm); $m_p$ and $m_l$ are respectively the mass (kg) of piston subassembly and load; $A_p$ is the effective area (m$^2$) of cross section that steam acts on the piston subassembly; $\mu$, $F_t$ and $N$ are respectively the wall frictional coefficient, the wall friction (N) and bearing force of the cylinder.

For establishing the mathematical model of each component of the system, some assumptions are made within the allowable range,

1) The change process of the steam state is regarded as a quasi-equilibrium process. The macroscopic kinetic energy of steam is not taken into account during the discharge process.

2) The steam accumulator is a rigid storage vessel, of which the internal parameters are treated as lumped parameters.

3) The heat transfer between the outside wall of the steam accumulator and the environment can be neglected for the thick thermal insulation layer and the short discharge time.

4) The initial steam state of the cylinder at $\tau=0$ is in a condition of given pressure $P_0$ and temperature $T_0$.

5) The heat transfer between the steam and the inner wall of the cylinder is neglected, as well as the influence of the initial air in the cylinder.

6) The steam in the cylinder has no leakage, and the liquid volume in the cylinder is neglected if the steam state is wet steam.

7) The piston subassembly and load are rigid bodies that can’t be distorted and doesn’t have an influence on the macroscopic change of applied point (line).

8) There is no resistance during steam flows in the pipes.

### 3.1 Thermodynamic model for steam accumulator discharge process

Adopting the space in the steam accumulator as a control volume for thermodynamic analysis, the state changing rules of the working fluid in the control volume simultaneously satisfy the conservation equations of mass, energy and volume[12].

During the discharge process, no mass flows into the control volume ($\delta m\neq 0$), So, the working fluid in the control volume satisfies the mass conservation equation:

$$Qd\tau = -d(V'_0 \cdot \rho'_0 + V''_0 \cdot \rho''_0)$$

The working fluid in the control volume also satisfies the energy conservation equation:

$$Q \cdot h'_0 d\tau = -d(u'_0 \cdot V'_0 \cdot \rho'_0 + u''_0 \cdot V''_0 \cdot \rho''_0)$$

where $u_0$ is the specific internal energy of working fluid, kJ/kg.

At the same time, the working fluid in the control volume also satisfies the volume conservation equation:

$$V_0 = V'_0/a = V''_0 + V''_0 = C_{const}$$

According to the thermodynamic property of water and steam, the characteristic parameters in liquid/vapor space of wet steam in the steam accumulator is only a
single-valued function of pressure or temperature, which can be calculated by the IAPWS for the thermodynamic properties of water and steam [12–13]:
\[
\rho_0 = \rho_0(P_0), h_0 = h_0(P_0), u'_0 = h'_0 - P_0 / \rho'_0 \quad (4)
\]
\[
\rho_0 = \rho_0(P_0), h_0 = h_0(P_0), u'_0 = h'_0 - P_0 / \rho'_0 \quad (5)
\]

3.2 Flow characteristic model of launching valve

The flow characteristic of the launching valve is the relationship between percentage of maximum flow capacity and percentage of total travel range, the typical inherent flow characteristics of the valve mainly include equal percentage, linear and quick opening.

A control valve such as the launching valve is a throttling device for changing the local resistance, which is different from a nozzle. The gas/steam flow state of a nozzle is usually divided into sonic and subsonic flows according to the pressure ratio of nozzle [14]. But with regard to a control valve, because different valves have different fluid passages and structures, the gas/steam flow state is divided into choked and non-choked flows according to the critical pressure differential ratio \( X_1 \) of valve. Especially for the compressible fluid, multiplying \( X_1 \) by the ratio of specific heat factor \( F_k \) is the critical value for determining the choked flow [15].

With regard to the steam, the non-choked flow can be determined if \( X \) is less than \( F_k \cdot X_1 \) [16] and calculated as
\[
Q = 3.16 \cdot k \sqrt{X \cdot (1000P_0) \cdot \rho_0^2 / 3600} \quad (6)
\]
where \( X \) is the pressure differential ratio of control valve, \( X=(P_c-P_t)/P_0 \); \( F_k=k/1.4 \) (the adiabatic exponent \( k \) values are 1.3 and 1.135 for the superheated steam and the saturated steam, respectively); \( X_1 \) only depends on the fluid passage and structure of control valve; \( y \) is the expansion factor, \( y=1-X(3F_k X_1) \); \( k \) is the flow coefficient that can be used to establish flow capacity.

With regard to the steam, the choked flow can be determined if \( X \) is bigger than \( F_k \cdot X_1 \) and calculated as follows
\[
Q = 1.78 k \cdot X \cdot (1000P_0) \cdot \rho_0^2 / 3600 \quad (7)
\]

3.3 Thermodynamic model of cylinder

During the charge process of the cylinder, the space in the cylinder can be adopted as a control volume. When piston subassembly and load are pushed by steam to move ahead for power output, both the pressure and the volume of the control volume change all the time. Therefore, the steam in the cylinder may go through the state of superheated steam or saturated steam. The state of superheated steam is usually expressed as a function of pressure and temperature, whereas the interdependent relationship with the state parameters of pressure and temperature in the saturation steam needs another independent state parameter (usually is vapor fraction). Noticing that the state parameters of both superheated steam and saturated steam can be expressed as a function of pressure and density, the problem that vapor fraction needs to be judged in calculation can be solved well by choosing density as another independent state parameter.

During the charge process, no mass flows out the control volume \( (\delta m_c=0) \). The working fluid in the control volume satisfies the mass conservation equation:
\[
\dot{Q}d\tau = d(V_c \cdot \rho_c) \quad (8)
\]
where
\[
V_c = A_p x + V_{c0}
\]
\[
A_p = \pi D_p^2 / 4
\]
where \( V_{c0} \) is the initial clearance volume at \( r=0, x \) is the displacement of the load, m.

The working fluid in the control volume also satisfies the energy conservation equation:
\[
Q \cdot h_0 d\tau = d(u_c \cdot V_c \cdot \rho_c) + P_c dV_c \quad (11)
\]
where \( u_c \) is the specific internal energy of steam in the cylinder, kJ/kg.

The state parameters of steam in the cylinder can also be expressed as a function of pressure and density, which can be calculated by the IAPWS for the thermodynamic properties of water and steam [13]:
\[
h_c = h_c(P_c, \rho_c), u_c = h_c - P_c / \rho_c \quad (12)
\]

3.4 Kinetic model of piston subassembly and load

The piston subassembly and load are the ultimate movement components of the system, under given displacement and allowable acceleration, the load is pushed by the piston subassembly to the set value of final velocity. The force analysis of the movement components is shown in Fig. 1 during the dynamics process.

The motion equation of the piston subassembly and load in the direction of acceleration can be gotten as follows
\[
(m_t + m_p) \frac{d^2 x}{dt^2} = P_c \cdot A_p - F_i \quad (13)
\]

The equation of force balance in the vertical direction is got as
\[
N = F_i / \mu_p = (m_t + m_p)g \quad (14)
\]

Thus, the combination of Eqs. (1) to (14) is the dynamic mathematical model of the horizontal steam-launch system, which can be calculated by inputting initial steam pressure and water filling coefficient in the steam accumulator, valve control strategy, initial clearance volume in the cylinder, initial steam pressure and temperature in the cylinder, physical parameters of
piston subassembly and load. Thereby, the performances of the system with the launching valves of different flow characteristics can be calculated.

4 Results and discussion

4.1 Flow process diagram of system simulation

It can be seen that the mathematical models of the system are difficult to obtain analytical solution and need the discretization for numerical solution, so the discharge process is divided into several time steps and the performance parameters of the system in each time step are calculated. The calculation flow is shown in Fig. 2 and the main parameters used for the model are listed in Table 1.

4.2 Analysis of simulation results

Given the opening rules of launching valve with the three kinds of flow characteristics are the same uniform opening, namely, the relative opening increases linearly from zero to one under the same time of discharge process, the launching process can be numerical simulated by the given initial parameters of system (shown in Table 1). Figure 3 shows the pressure differential ratio as a function of the time during the simulation process and then the dynamic characteristics curves of the system are shown in Figs 4−9.

Figure 3 shows that, at the initial discharge process, the pressure differential ratios with the three kinds of valves drop sharply, namely, the steam flow changes rapidly from choked flow into non-choked flow. With the charge process going on, the pressure differential ratios with the linear and quick opening valve increase slowly and are less than the critical value (the critical value of $F_K X_T$ is 0.567 5), so the steam flow is at the stage of non-choked flow. The pressure differential ratio with the equal percentage valve increases firstly but then drops gradually, which causes the result that the steam flow isn’t always at the stage of non-choked flow. When $t'$ is among 0.31 and 0.66, the steam flow is at the stage of choked flow.

Figure 4 shows the dimensionless pressure and gross steam discharge (discharge flow accumulation) profiles in the steam accumulator as a function of the dimensionless time with three different valves, where the same final pressure ($P_0' = 0.915$ at $t' = 1.0$) with three kinds of valves is achieved by selecting different $k_v$ accordingly. These results reflect that, there is a reciprocal one-to-one correspondence between the steam pressure and gross steam discharge in the steam accumulator. At the earlier stage, the pressure drop in the steam accumulator with the quick opening valve is very large, followed by the linear valve and the smallest with the equal percentage valve. On the contrary, at the later stage, the pressure drop with the quick opening valve is very small, followed by the linear valve and the largest with the equal percentage valve. There are two reasons

![Fig. 2 Flow chart for simulation](image-url)
Table 1 Main parameters used for model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial water filling coefficient, $a/%$</td>
<td>45</td>
</tr>
<tr>
<td>Critical pressure differential ratio, $X_T$</td>
<td>0.7</td>
</tr>
<tr>
<td>Adjustable ratio of launching valve, $R$</td>
<td>30</td>
</tr>
<tr>
<td>Initial cylinder clearance volume, $V_{c0}/m^3$</td>
<td>0.75</td>
</tr>
<tr>
<td>Initial cylinder clearance temperature, $T_{c0}/K$</td>
<td>553</td>
</tr>
<tr>
<td>Piston diameter, $D_P/mm$</td>
<td>450</td>
</tr>
<tr>
<td>Mass of piston subassembly, $m_P/kg$</td>
<td>500</td>
</tr>
<tr>
<td>Mass of load, $m_f/kg$</td>
<td>20 000</td>
</tr>
<tr>
<td>Frictional coefficient of cylinder wall, $\mu_p$</td>
<td>0.02</td>
</tr>
</tbody>
</table>

that lead this result: one reason is the inherent flow characteristic of three different valves. The changing rate of flow at the small opening of quick opening valve is very large, which brings the large pressure drop at the early stage. Whereas the changing rate of flow is very small for equal percentage valve when the opening is small, so the pressure drop is small at the early stage. Whereas the flow changes fast when the opening is large, the pressure also drops fast at the later stage. The flow characteristic of the linear valve is between the quick opening valve and the equal percentage valve, so the changing rule of steam pressure in the steam accumulator is also between the both. The other reason is the back pressure of the valve (the pressure in the cylinder), the dimensionless pressure profiles in the cylinder as a function of the dimensionless time is shown in Fig. 5.

Figure 5 shows the dimensionless pressure profiles in the cylinder as a function of the dimensionless time. These results indicate that, at the initial discharge process, the pressure in the cylinder increases sharply with the quick opening valve, the highest dimensionless pressure is 0.941, followed by the linear valve, which is 0.838, and the smallest value is 0.604 with the equal percentage valve. With the discharge process going on, the pressure in the cylinder with the three valves all drop gradually, the result can be explained that the influence of pressure increase led by the discharge flow accumulation is less than that of the pressure drop led by the volume extension in the cylinder. At the later stage, though the pressure in the cylinder are still drop with the quick opening and linear valves, the pressure with the equal percentage valve begins to increase. The reason is as follows, the changing rate of flow with the equal percentage valve is relative larger than those of the quick opening and linear valves at the same opening. Therefore, the influence of pressure increase is larger than the influence of pressure drop in the cylinder with the former. When the discharge process is over, the pressure in the cylinder with the quick opening valve is the smallest which values 0.396 (dimensionless pressure), followed by the linear valve, which values 0.433, and the max value is 0.577 with the equal percentage valve.
Figure 6 shows the dimensionless load acceleration profiles as a function of the dimensionless time. It can be seen that, there is a reciprocal one-to-one correspondence between the load acceleration and the pressure in the cylinder. The value of the max dimensionless acceleration $a_{\text{max}}^*$ with the quick opening valve is 1.0, the value of the peak-to-average ratio of dimensionless acceleration (the ratio of the max value to the average value) is 1.722. The value of $a_{\text{max}}^*$ with the linear valve is 0.896 and that of peak-to-average ratio is 1.614. The value of $a_{\text{max}}^*$ with the equal percentage valve is 0.681 and that of peak-to-average ratio is 1.355, which is the closest to 1, that means the smallest and the largest fluctuation ranges of accelerations are the equal percentage and the quick opening valves, respectively.

Figures 7 and 8 respectively show the dimensionless load velocity $w^*$ and displacement $x^*$ profiles as a function of the dimensionless time. It can be seen that the changing rule of the load velocity with three control valves are nearly linear, and the displacement as a function of time are all approximate to conic. The values of velocity and displacement with the quick opening valve are the largest all the time, followed by the linear valve, and the smallest are that with the equal percentage valve.

To further analyze the effect on the thermodynamics and kinetics characteristics of the system, with the different flow characteristics of launching valve, the dimensionless load velocity profiles as a function of the dimensionless displacement, shown in Fig. 9, is used to judge whether three kinds of launching valves can all meet the system launching requirement that the load must reach the set value of final velocity in a stated displacement ($w^*$ and $x^*$ are set to the values of 0.90 and 0.82 according to the system design, respectively) when the charge process is finished. Apparently, the velocity with the quick opening valve is always the largest, followed by the linear valve, and the smallest is the equal percentage valve. Under the same discharge time ($t^*$ is from 0 to 1), the valve opening rule (the relative opening increases linearly from zero to one) and the final pressure ($P_0^*=0.915$) in the steam accumulator, the value of the dimensionless displacement $x^*$ with the quick opening valve
valve is 1.0 and that of the dimensionless velocity \( w^* \) is 1.0. The value of \( x^* \) is 0.926 with the linear valve and that of \( w^* \) is 0.957, while the value of \( x^* \) is 0.756 with the equal percentage valve and that of \( w^* \) is 0.843. Therefore, under the set value of \( x^* (x^*=0.82) \), the values of \( w^* \) are 0.922 and 0.910 respectively with the quick opening valve and linear valve, which both can meet the system launching requirement. Whereas the maximum value of \( x^* \) is 0.756 with the equal percentage valve that doesn’t reach the set value when the charge process is finished. However, the load will still move ahead by the inertia force and the velocity will decrease gradually without the thrust force of steam, so \( w^* \) must be less than the value of final velocity (\( w^* = 0.90 \)) when \( x^* = 0.82 \).

Finally, by comparing the analysis results of Fig. 6 and Fig. 9, it can be concluded that all the three valves have their advantage and disadvantage. The load can meet the system launch requirement with the quick opening valve, but its disadvantage is that the fluctuation range of acceleration is the largest, which can’t keep the load moving as steady as possible. Whereas the fluctuation range of the load acceleration with the equal percentage valve is the smallest, but it can’t meet the system launch requirement. Considering the system with the linear valve can both meet the system launching requirement and have a moderate fluctuation range of the load acceleration, it’s better to choose the linear valve for improving the thermodynamics and kinetics characteristics of the system.

5 Conclusions

1) The value of the peak-to-average ratio of dimensionless acceleration with the equal percentage valve is the smallest which values 1.355, followed by the linear valve, which values 1.614, and the maximum value is 1.722 with the quick opening valve.

2) The final values of the dimensionless velocity with the equal percentage valve, the linear valve and the quick opening valve are 0.843, 0.957 and 1.0, respectively. It indicates that the value of the dimensionless velocity with the equal percentage valve doesn’t reach the set value of 0.90 when the dimensionless displacement is 0.82, while the system with the linear valve and the quick opening valve can meet the launching requirement.

3) Among the three kinds of launching valves, the system with the linear valve can both meet the system launching requirement and have a moderate fluctuation range of the load acceleration, so it is better to choose the linear valve and more researches on the control strategies of the linear valve are needed for further improving the launching performance of the system.

References


(Edited by DENG Lü-xiang)