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Comparative Study on Cavitation Characteristics of Different Fuels Considering Thermal Effect in Injector Nozzle

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ABSTRACT

At present, facing the dual problems of world energy crisis and carbon emission pollution, the cavitation phenomenon formed in the nozzle due to the high pressure difference and whether the physical property difference of clean alternative fuels methanol and ammonia can be successfully applied and replace the traditional fuel has become the focus of current research. In this paper, the Schnerr-Sauer cavitation model considering the thermal effect of cavitation and the physical property parameters of fuel varying with temperature and pressure are embedded into the CFD software, and the cavitation characteristics of diesel, methanol and ammonia inside the nozzle are studied and compared. The results show that the influence of cavitation heat effect on diesel, methanol, and ammonia in the nozzle increases sequentially, manifested by the increase in the degree of cavitation in the nozzle; When the cavitation number is approximately 1.3 and 1.1, the fuel exhibits cavitation inception and supercavitation phenomena, respectively; The net temperature effect of fuel varies under different injection pressures, with diesel and methanol having a more significant heating effect in the nozzle flow, while ammonia has a more significant cooling effect when the injection pressure is less than 14MPa.

Keywords: cavitation, thermal effect, diesel, methanol, ammonia, nozzle

1. INTRODUCTION

Diesel engines, as important power machinery in industrial production and transportation, will continue to be an important source of power in the future. However, the emissions of diesel engines have a significant impact on the environment[1]. Faced with increasingly strict emission standards and severe energy crises, the use of diesel high-pressure common rail systems and clean alternative fuels has become increasingly important[2,3]. Low carbon fuel methanol and zero carbon fuel liquid ammonia are considered ideal clean alternative fuels for diesel due to their advantages of low emission pollution, easy storage and transportation[4,5]. In addition, during the high-pressure injection process of diesel engines, the flow inside the nozzle plays an important role in near-field atomization, thereby affecting combustion and emission characteristics. The cavitation problem caused by liquid in the process of nozzle flow is particularly significant. Cavitation refers to the process of the formation, development, and collapse of steam or gas bubbles inside the liquid when the local pressure inside the liquid decreases[6]. On the one hand, cavitation will cause cavitation erosion on the flow channel wall, block the flow channel, and cause pressure fluctuations; on the other hand, cavitation will significantly increase the spray angle, enhance the atomization effect, thereby strengthening combustion and reducing emissions[7,8]. In order to investigate the cavitation flow characteristics of fuel inside the nozzle, scholars from various countries have conducted extensive research.

The cavitation flow of fuel in the nozzle is mainly influenced by three factors: geometric structure, dynamic factors, and fuel medium factors. In 1976, Nurick[9] proposed a dimensionless parameter, cavitation number, to describe the cavitation flow state in a nozzle, and studied the proportional relationship between the flow coefficient and the square root of the cavitation number through experimental methods; He Zhixia et al.[10,11] conducted various experimental and numerical simulations on the nozzle, including the effects of nozzle geometry, temperature, and pressure on cavitation; Qiu Tao et al.[12,13] studied the influence of pressure factors on the cavitation and flow

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characteristics of the nozzle through experiments and simulations. The results showed that the flow coefficient was only related to the geometric structure when there was no cavitation, and was related to the contraction coefficient and cavitation number of the inlet section when cavitation occurred. In terms of fuel media, Payri et al.[14] compared the cavitation characteristics of n-heptane, n-decane, n-dodecane, and commercial diesel, and the results showed that fuels with low viscosity were more prone to cavitation; BalajiMohan and Ma Zhiyan et al.[15,16] used numerical simulation methods to study the cavitation characteristics of diesel and biodiesel in the nozzle. The results showed that compared to diesel, biodiesel has a higher mass flow rate and weaker cavitation degree due to its higher density and viscosity; Roberto Torelli et al.[17] selected two types of gasoline fuel and alternative fuel n-dodecane to study the effect of fuel properties on cavitation characteristics inside the nozzle. The study showed that the higher the saturated vapor pressure of the fuel, the stronger the degree of cavitation. All the above studies show that fuel cavitation in the nozzle has an important impact on the flow characteristics, but most of the studies are focused on the fuel under isothermal conditions, and the temperature change of the fuel during the flow process is rarely considered. Due to the gradual increase in injection pressure, some previously insignificant internal thermal effects of fuel injectors have become increasingly important. F.J.Salvador et al.[18] found that two opposite thermal effects occur when fuel flows through the nozzle: on the one hand, the fuel depressurizes, evaporates, and absorbs heat to cool it down; On the other hand, due to viscous friction, the fuel is heated near the nozzle wall. The City University of London team[19,20] compared isothermal flow with flow considering thermal effects, and the results showed that an increase in temperature resulted in a decrease in fuel density and viscosity, and an increase in outlet velocity. In addition, considering the thermal effect, the volume fraction of cavitation inside the nozzle is higher.

In summary, existing studies have shown that cavitation has a significant impact on the flow and heat transfer characteristics inside the nozzle, and there are many factors that affect cavitation characteristics. However, most studies only focus on the cavitation characteristics of diesel under isothermal conditions, lacking research on the cavitation characteristics of diesel, methanol, and ammonia that takes into account thermal effects. Taking the single-hole nozzle of a diesel engine as the research object, this paper adopts CFD three-dimensional numerical simulation to consider the relationship between the physical properties of the fuel and the change of temperature and pressure, and considers the influence of temperature on the source term of mass transport in the original cavitation model. By solving the continuity equation, momentum equation and mass transport equation, coupled with the energy equation considering the influence of latent heat of vaporization, the research is carried out under different injection pressures. Effects of diesel, methanol and ammonia on cavitation characteristics considering thermal effects.

2. NUMERICAL METHOD

2.1 Governing equation

The cavitation flow in the nozzle is a typical multiphase flow. For cavitation flow considering thermal effects, in addition to the continuity equation and momentum equation, there is also an energy equation considering the influence of vaporization latent heat, which is as follows:

(1) Continuity equation:

$$\frac{\partial}{\partial t}(\rho_m) + \nabla \cdot (\rho_m \vec{v}_m) = 0 \tag{2-1}$$

Where t is time, s; ρ_m is the mixture density, kg/m^3; $\rho_m = \alpha_l \rho_l + (1-\alpha_l)\rho_v$, Subscript m, l, v represent the mixed phase, liquid phase and gas phase respectively, α is the volume fraction; $\vec{v}_m$ is the velocity vector, m/s.

(2) Momentum equation:

$$\frac{\partial}{\partial t}(\rho_m \vec{v}_m) + \nabla \cdot (\rho_m \vec{v}_m \vec{v}_m) = -\nabla p + \nabla \cdot [\mu_m(\nabla \vec{v}_m + \nabla \vec{v}_m^T)] + \vec{F} \tag{2-2}$$

Where p is static pressure, Pa; μ_m is the mixture viscosity, pa·s; $\vec{F}$ is a volume force.

(3) Energy equation:

$$\frac{\partial}{\partial t}\sum_{k=1}^{n} \alpha_k \rho_k E_k + \nabla \cdot \sum_{k=1}^{n} (\alpha_k \vec{v}_k (\rho_k E_k + p)) = \nabla \cdot (k_{eff} \nabla T) + S_E \tag{2-3}$$

Where E_k is the total fluid energy of the control body, j; k_{eff} is the effective thermal conductivity, W/(m·K); T is the temperature, K; S_E is the energy source term, is the change of flow field energy caused by cavitation, and the magnitude is equal to the product of the mass exchange capacity between vapor and liquid and the latent heat of vaporization.

2.2 Cavitation model

The Schnerr-Sauer cavitation model with good compatibility and reliability was selected for this numerical calculation[21]. This model is derived from the growth equation of a single bubble, which is derived from the Rayleigh Plesset equation, as shown in equations 2-4:

$$R_b \frac{D^2 R_b}{Dt^2} + \frac{3}{2}\left(\frac{DR_b}{Dt}\right)^2 = -\left(\frac{P_b - P}{\rho_l}\right) - \frac{4\mu_l}{R_b}\frac{DR_b}{Dt} - \frac{2\sigma}{\rho_l R_b} \quad (2\text{-}4)$$

Among them, R_b is the radius of the bubble, σ is the surface tension of the liquid, and P_b is the surface pressure of the bubble.

Ignoring the second-order term and surface tension, simplify the equation to equations 2-5:

$$\frac{DR_b}{Dt} = \sqrt{\frac{2}{3}\frac{P_b - P}{\rho_l}} \quad (2\text{-}5)$$

Schnerr and Sauer deduced that the net mass exchange rate from the liquid phase to the gas phase is:

$$\frac{\partial}{\partial t}(\alpha\rho_v) + \nabla\cdot(\alpha\rho_v\vec{v}) = \frac{\rho_v\rho_l}{\rho}\frac{D\alpha}{Dt} \quad (2\text{-}6)$$

The net mass source term is shown in equations 2-7:

$$R = \frac{\rho_v\rho_l}{\rho}\frac{d\alpha}{dt} \quad (2\text{-}7)$$

According to formula 2-8, Schnerr and Sauer established the relationship between the volume fraction α of the vapor phase and the bubble density number n_b:

$$\alpha = \frac{n_b\frac{4}{3}\pi R_b{}^3}{1 + n_b\frac{4}{3}\pi R_b{}^3} \quad (2\text{-}8)$$

By substituting equations 2-8 into the previous equations, the expressions for mass transfer rate and bubble radius can be obtained:

$$R = \frac{\rho_v\rho_l}{\rho}\alpha(1-\alpha)\frac{3}{R_b}\sqrt{\frac{2}{3}\frac{P_v - P}{\rho_l}} \quad (2\text{-}9)$$

$$R_b = \left(\frac{\alpha}{1-\alpha}\frac{3}{4\pi}\frac{1}{n}\right)^{\frac{1}{3}} \quad (2\text{-}10)$$

Pv is the saturated vapor pressure of the liquid, and the mass transfer rates of evaporation and condensation can be obtained by bringing equations 2-5 and 2-10 into equations 2-9, combined with the influence of temperature change in the cavitation flow on the mass transfer process:

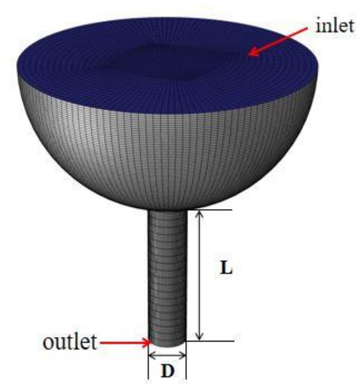
$$P_v \geq P \quad R_e = \frac{\rho_v\rho_l}{\rho}\alpha(1-\alpha)\frac{3}{R_b}\sqrt{\frac{2}{3}\frac{P_v(T) - P}{\rho_l}} \quad (2\text{-}11)$$

$$P_v \leq P \quad R_c = \frac{\rho_v\rho_l}{\rho}\alpha(1-\alpha)\frac{3}{R_b}\sqrt{\frac{2}{3}\frac{P - P_v(T)}{\rho_l}} \quad (2\text{-}12)$$

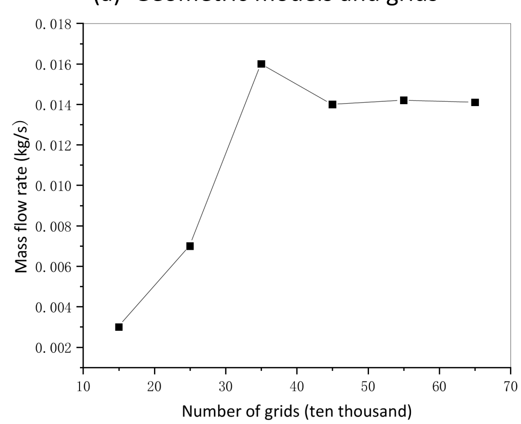
Where R_e and R_c are evaporation and condensation rates respectively, and $P_v(T)$ is the saturated vapor pressure corresponding to the local temperature T of the distant flow field.

2.3 Geometric model and mesh division

The geometric model used in this article refers to the nozzle in M. Ghiji[22]. Due to the main research on the flow of fuel in the diesel engine nozzle, and considering the computational performance and simulation efficiency of the computer, the nozzle geometry structure was appropriately simplified and set without affecting the accuracy of numerical simulation. The simplified three-dimensional model and grid diagram are shown in Figure 1(a), with a nozzle diameter D of 0.25mm and a nozzle length L of 1mm. ICEM software is used to divide the whole hexahedral mesh, and the mesh near the nozzle wall and the cross-sectional area of the flow channel are encrypted. To avoid the impact of grid partitioning on the calculation results, it is necessary to perform grid independence verification on the model. This article conducted grid independence verification under an inlet pressure of 100MPa, and the monitored physical quantity was the mass flow rate of the nozzle. The results are shown in Figure 1(b). As the number of grids increases, the mass flow rate remains basically unchanged. In order to balance calculation speed and accuracy, the final number of grids selected was 45 ten thousand.



(a) Geometric models and grids



(b) Grid independence verification

Fig. 1 Geometric models, grids and grid independence verification

2.4 Calculation settings

Table 1 shows the boundary conditions and detailed parameter settings for numerical simulation. The solver uses a pressure based solver, the gradient discretization format uses the Least Squares Cell Based format with better accuracy, the discretization of gas phase volume fraction, momentum, energy, turbulent kinetic energy, and turbulent dissipation rate is solved using the Second Order Upwind scheme to obtain more accurate results.

Table. 1 Boundary conditions and simulation parameter

parameter	unit	value
saturated vapor pressure	MPa	Pv(T)
injection pressure	MPa	5~100
back pressure	MPa	3
turbulence model		Realizable k-ε
cavitation Model		Schnerr-Sauer model
bubble density number		10^{13}
residual		$<10^{-6}$

The compressibility and thermal effects of fuel must be considered when flowing under high temperature and pressure. The physical properties of methanol and liquid ammonia (such as density, viscosity, thermal conductivity, specific heat capacity, and saturated vapor pressure) are specified as functions of temperature and pressure for the data generated by REFPROP 10.0. Due to the lack of a comprehensive diesel physical property database, diesel physical property parameters were derived from the physical property fitting formula of diesel in European winter proposed by F.J.Salvador[23] et al. For example, the fitting equations of fuel density ρ and viscosity μ are as follows:

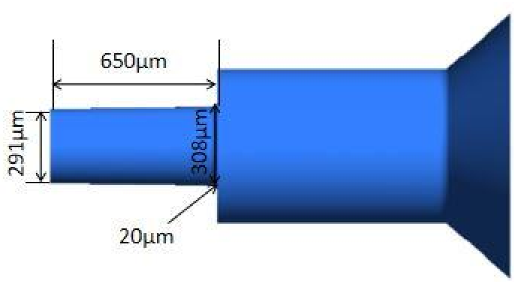
$$\rho = 826.5 - 1.0217(T-298) + 1.251\cdot 10^{-3}(T-298)^2 + 0.6035(p-0.1) - 8.265\cdot 10^{-4}(p-0.1)^2 + 1.441\cdot 10^{-3}(T-298)(p-0.1) \quad (2\text{-}13)$$

$$\mu = 3.2158\cdot e^{[-0.0263(T-298)]}\cdot[(-1.48+5.86\cdot[3.2158\cdot e^{[-0.0263(T-298)]}]^{0.181})\cdot(\frac{p-0.1}{1000})]10^{-2} \quad (2\text{-}14)$$

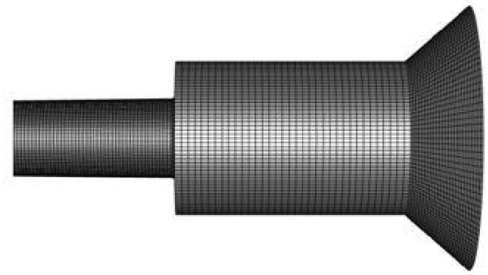
Where p and T are fuel pressure and temperature.

2.5 Model validation

To verify the accuracy of the cavitation heat effect model and parameter settings, this study compared and validated the experimental data of the injector inlet orifice (OZ orifice) in F.J.Salvador[23]. As shown in Figure 2 (a), the length of OZ hole is 650 μm. Inlet diameter is 308 μm and outlet diameter is 291 μm. The radius of the inlet chamfer is 20 μm. The OZ orifice calculation grid model divided by ICEM software is shown in Figure 2(b). During the division process, key dimensions of the OZ orifice were locally encrypted and a near wall boundary layer was established to ensure the accuracy of the calculation.



(a) Geometric model



(b) Grid model

Fig. 2 Geometry model and mesh model of OZ throttle hole

For model calibration, the same boundary conditions as the F.J.Salvador experiment were used. Figure 3 shows the comparison curve between the calculated and experimental results of fuel mass flow under different pressure drops. From the figure, it can be seen that the calculation results of the non isothermal compressible model (considering thermal effects) are closer to the experimental results than the calculation results of the isothermal incompressible model, and the maximum error between the calculation results of the non isothermal compressible model and the experimental data is less than 5%, while the maximum error between the calculation results of the isothermal incompressible model and the experimental results is 8%. Figure 4 shows a comparison between the calculated results of the average temperature at the outlet of the OZ hole and the experimental data. The calculated results of the average temperature at the outlet of the OZ hole are consistent with the trend of the experimental measurement curve and the values are close, with a maximum relative error of less than 5%, which is within an acceptable error range. Therefore, the simulation model used in this study can be accurately used to study the non isothermal compressibility flow calculation of fuel in the diesel engine nozzle.

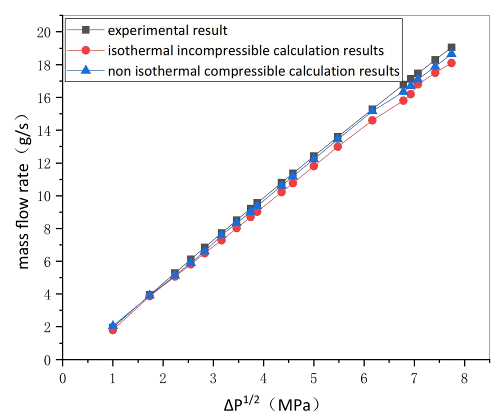


Fig. 3 Comparison of fuel mass flow simulation results with experimental results

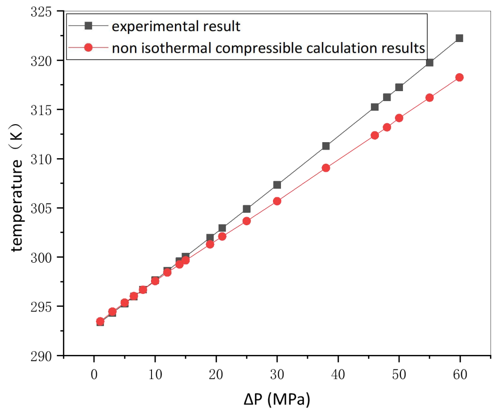
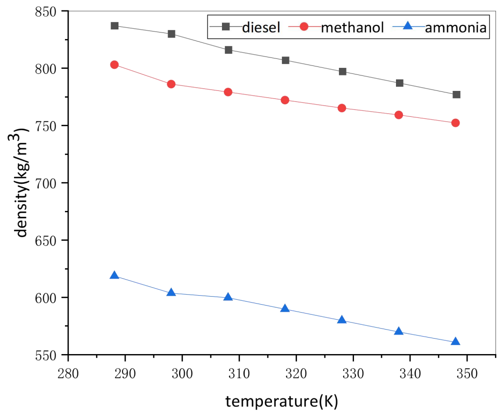


Fig. 4 Comparison of calculated fuel temperature at OZ outlet with experimental results

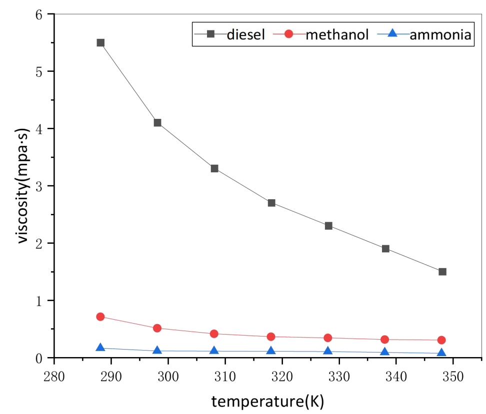
3. RESULTS AND DISCUSSION

3.1 Effect of temperature on physical properties of different fuels

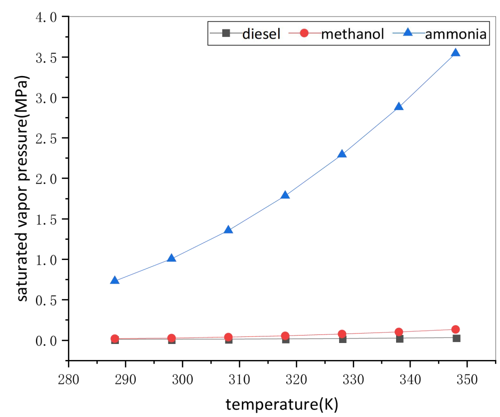
In order to better compare and analyze the cavitation characteristics of diesel, methanol and liquid nitrogen under the thermal effect model, the density, viscosity and saturated vapor pressure curves of the three fuels with temperature are shown in Figure 5. It can be seen from the figure that the change trend and gradient of physical properties of different fuels at different operating temperatures are different. The density and viscosity of the three fuels decrease with the increase of temperature, while the saturated vapor pressure increases with the increase of temperature. At the same temperature, the density and viscosity of the three fuels are diesel>methanol>ammonia, and the saturation vapor pressure is the opposite. Compared with diesel and methanol, ammonia has lower density and viscosity, and higher saturation vapor pressure. Especially, the saturation vapor pressure of ammonia is more sensitive to temperature changes. The higher the temperature, the greater the gradient of saturation vapor pressure increase. At room temperature, the saturation vapor pressure of ammonia is about 500 times that of diesel. In addition, at the same temperature, the density and saturated vapor pressure values of diesel and methanol are close, and the viscosity of methanol is slightly higher than that of ammonia.



(a) Density



(b) Viscosity



(c) Saturated vapor pressure

Fig. 5 The curve of physical properties of diesel, methanol and ammonia with temperature

3.2 Cavitating heat effect characteristics of different fuels

The dimensionless number currently used to describe the degree of cavitation in the nozzle is the cavitation number defined by Nurick[9], with the expression K = (Pin-Pv)/(Pin-Pb), where Pin is the injection pressure, Pv is the saturated vapor pressure of the liquid, Pb is the back pressure of the nozzle, and in this article, it is the outlet pressure of the nozzle. The smaller the K value, the more severe the cavitation degree. Figures 6 and 7 show the vapor volume fraction and temperature cloud maps of the three fuels at the nozzle under isothermal incompressible and considering thermal effects models, with an injection pressure of 14MPa and a fuel inlet temperature of 298K. From Figure 6, it can be seen that both the isothermal incompressible model and the model considering thermal effects, the gas phase volume fraction of ammonia in the nozzle is greater than that of methanol and diesel, and the gas phase proportion difference between diesel and ammonia is significant. This is because under the same operating conditions, the saturated vapor pressure of ammonia is much higher than that of diesel oil, while the viscosity is lower than that of diesel, and the degree of cavitation of methanol is between the two. After considering the thermal effect of cavitation, the cavitation intensity of the three fuels in the nozzle increases. However, diesel and methanol have a smaller change in the gas phase proportion in the nozzle after considering the thermal effect, while ammonia has a larger change in the gas phase proportion in the nozzle after considering the thermal effect, manifested as a longer cavitation length and thicker thickness, changing from the cavitation development state in the isothermal incompressible model to the supercavitation state considering the thermal effect model.

In addition, from Figure 7, it can be seen that due to the wall friction heating effect and the cooling effect generated by fuel pressure reduction and vaporization heat absorption, after considering the thermal effect, the temperature changes of the three fuels in the nozzle are significant. The maximum temperature drop of ammonia in the nozzle is about 8K, while diesel and methanol are about 0.5 and 1K, respectively. This is because the vaporization latent heat value of ammonia is greater than that of both. The maximum temperature rise of the three fuels in the nozzle is about 7, 6 and 2K, respectively, because the density and viscosity of diesel, methanol and ammonia are successively reduced, so the energy dissipation loss is successively reduced. To sum up, in the calculation of cavitation flow of different fuels in jet holes, especially for alternative fuels methanol and ammonia, temperature has a great influence on the characteristics of cavitation thermal effect, so the non-isothermal compressible model of fuel is used for calculation in subsequent studies.

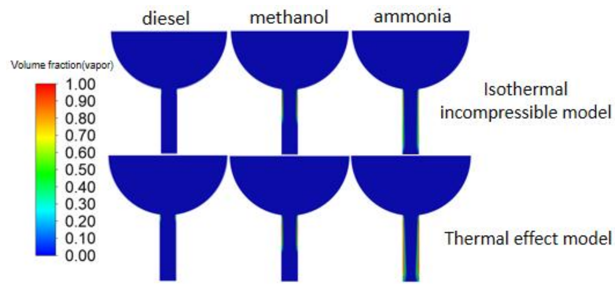


Fig. 6 Gas phase volume fraction distribution of fuel in nozzle under different models

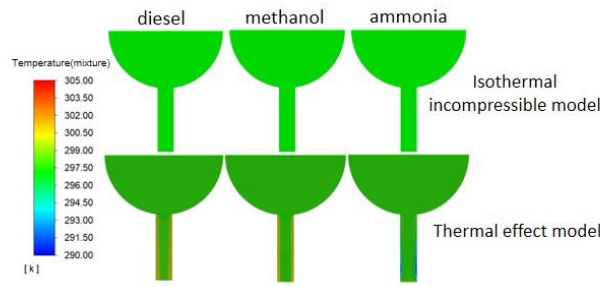


Fig. 7 Temperature distribution of fuel in the nozzle under different models

3.3 *Effect of injection pressure on cavitation characteristics of different fuels*

TFigure 8 shows the gas phase volume fraction cloud plots of three fuels in the nozzle under different injection pressures (5-50MPa). From the figure, it can be seen that as the injection pressure increases, the fuel sequentially undergoes single-phase flow, cavitation initiation, development, and supercavitation states, and the cavitation generation region is the same. When the fuel develops to supercavitation state, the proportion of gas phase in the nozzle does not increase with the increase of injection pressure. As the saturated vapor pressure of diesel oil, methanol and ammonia increases and the viscosity decreases in turn, ammonia cavitation occurs earlier and develops faster than methanol and diesel. When the injection pressure is 9MPa and 14MPa, the cavitation initiation and supercavitation state are reached respectively, while diesel only reaches the cavitation initiation state when the ammonia reaches the supercavitation state. The cavitation evolution process of methanol is between the two.

Figures 9-11 show the variation curves of cavitation number, outlet velocity, and mass flow rate of three fuels with injection pressure. It can be seen from FIG. 9 that the cavitation number of different fuels gradually decreases with the increase of injection pressure; the lower the cavitation number, the stronger the cavitation degree; and the change rate of cavitation number gradually decreases with the increase of inlet pressure, reflecting that the increase of inlet pressure, reflecting that the increase of cavitation intensity becomes more and more gradual with the increase of inlet pressure. When the cavitation number is about 1.3 and 1.1, the fuel reached the initial cavitation and supercavitation states respectively. Due to the high saturated vapor pressure of ammonia, the cavitation number is smaller than that of diesel and methanol under the same injection pressure, and the cavitation number of methanol is slightly lower than that of diesel. Figures 10 and 11 show the variation curves of the outlet velocity and mass flow rate of different fuels with injection pressure. The outlet velocity and mass flow rate of the three fuels show an upward trend with the increase of injection pressure. Under the same operating conditions, the outlet velocity of ammonia is greater than that of methanol and diesel, but the mass flow rate of ammonia is lower than that of methanol and diesel. This is because the density and viscosity of ammonia are lower than that of methanol and diesel, and the mass flow rate of methanol is approximately equal to that of diesel.

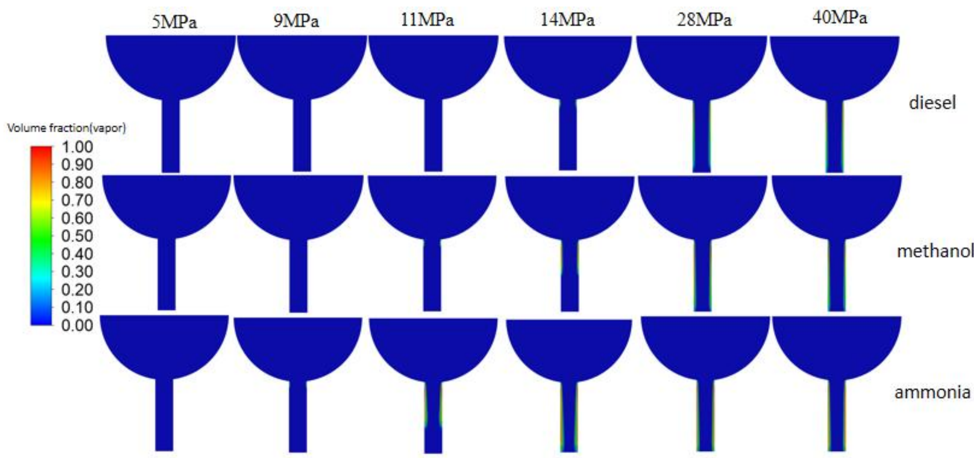


Fig. 8 Gas phase volume fraction distribution of three fuels under different injection pressures

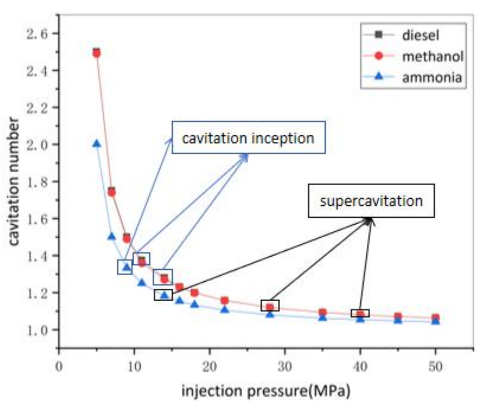


Fig. 9 Curve of cavitation number changing with injection pressure

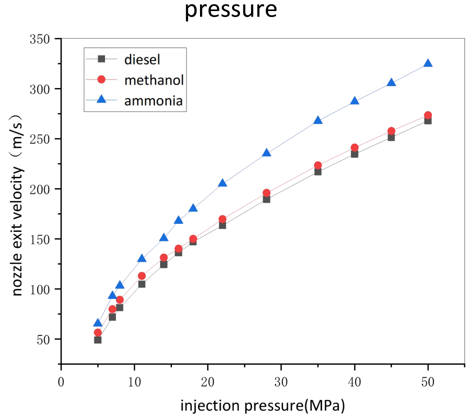


Fig.10 Curve of nozzle exit velocity changing with injection pressure

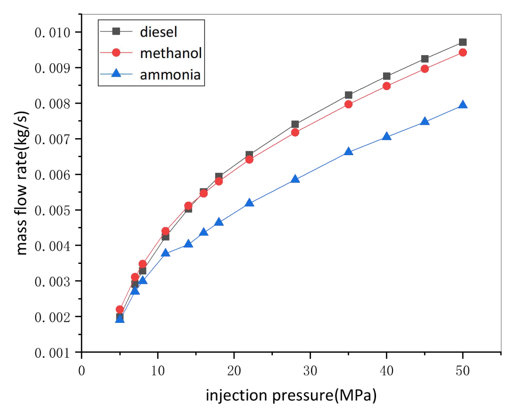


Fig.11 Curve of mass flow rate changing with injection pressure

Figure 12 shows the curves of the minimum and maximum temperatures of diesel, methanol, and ammonia in the nozzle as a function of injection pressure. It can be seen from the graph that there are irreversible frictional heating effects (highest temperature in the nozzle) and cooling effects of vaporization and heat absorption (lowest temperature in the nozzle) in the nozzle for the three fuels. As the injection pressure increases, the maximum and minimum temperatures of diesel and methanol in the nozzle gradually increase and decrease, and the maximum temperature rise is greater than the maximum temperature drop at the same pressure. Therefore, the heating effect of diesel and methanol in the nozzle is more significant than the cooling effect. When the injection pressure of ammonia is less than 14MPa, the maximum temperature drop in the nozzle is greater than the maximum temperature rise, and the cooling effect of ammonia in the nozzle is more significant, When the injection pressure is greater than 14MPa, the highest and lowest temperatures of ammonia gradually increase, and the heating effect of ammonia is more significant at this time. When the injection pressure is 50MPa, the maximum temperature rise of diesel, methanol and ammonia in the nozzle is 28, 22 and 12K, respectively, which is because the viscosity of diesel, methanol and ammonia decreases in turn, so the energy dissipation loss decreases in turn, and because the latent heat value of vaporization increases in turn, the cooling effect of the three fuels in the flow process is enhanced in turn.

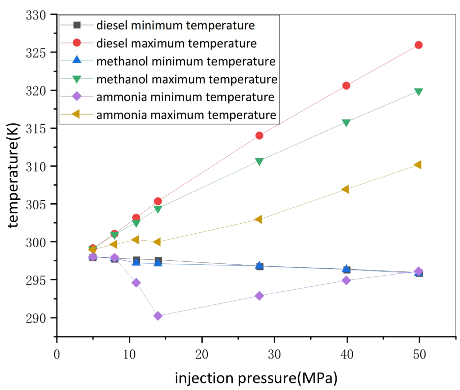


Fig.12 The curve of minimum and maximum fuel temperature in the nozzle with injection pressure

4. CONCLUSION

The use of low-carbon fuel methanol and zero carbon fuel ammonia is one of the important ways to alleviate energy crisis and reduce carbon emissions. This paper takes single-hole nozzle of diesel engine as the research object. The Schnerr-Sauer cavitation model considering the thermal effect of cavitation and the physical property parameters of fuel varying with temperature and pressure were embedded into the CFD software. At the same time, the energy equation considering the effect of latent heat of vaporization was coupled to solve. The influence of diesel, methanol and ammonia considering the thermal effect on cavitation characteristics was studied under different injection pressures. The main conclusions are as follows:

(1) The gradient and trend of changes in physical properties of different fuels vary at different temperatures, with the density and viscosity of the fuel decreasing with increasing temperature, and the saturated vapor pressure increasing with increasing temperature. Compared with diesel, ammonia has a lower density and viscosity, and a higher saturated vapor pressure. Especially, the saturated vapor pressure of ammonia is more sensitive to temperature changes, and the physical properties of methanol are between the two.

(2) Regardless of whether it is an isothermal incompressible model or a model considering thermal effects, the degree of cavitation in the nozzle of the three fuels under the same operating conditions is ammonia>methanol>diesel, and there is a significant difference in the degree of cavitation between ammonia and diesel. Considering the thermal effect model, the influence of diesel, methanol and ammonia on the cavitation thermal effect in the nozzle is enhanced successively, which shows that the cavitation length becomes longer, the thickness becomes thicker, and the cooling effect in the nozzle is enhanced.

(3) As the injection pressure increases, the three fuels undergo single-phase flow, cavitation initiation, development, and supercavitation states in the nozzle, and cavitation initiation and supercavitation phenomena occur when the cavitation number is approximately 1.3 and 1.1, respectively. Ammonia fuel with low density and viscosity and high saturated vapor pressure appears cavitation earlier and changes rapidly, followed by methanol and diesel fuel.

(4) The net temperature effect of fuel varies under different injection pressures. As the injection pressure increases, the highest temperatures of the three fuels in the nozzle show an overall upward trend. The lowest temperatures of diesel and methanol gradually decrease, and their maximum temperature drop is lower than the maximum temperature rise under the same pressure. Therefore, the heating effect of diesel and methanol in the nozzle flow is more significant. However, when the injection pressure of ammonia is less than 14MPa, the maximum temperature drop in the nozzle is greater than the maximum temperature rise, At this point, the cooling effect of ammonia is more significant, and the heating effect is more significant when the injection pressure is greater than 14MPa.

DECLARATION OF INTEREST STATEMENT

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. All authors read and approved the final manuscript.

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