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# Vaporization Characteristics of the Alcohol-Fueled Spark Ignition Engine

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Alcohol fuels have been considered for use as automotive fuel since two energy crises in the 1970's, but they have a defect of high latent heat of vaporization. Therefore, in order to improve vaporization of methanol, the authors have made the fuel vaporizing device with which to heat the mixture and eliminate the fuel film flow. This paper is a study on the characteristics of vaporization and engine performance according to the change of heating water temperature by means of the fuel vaporizing device. The study shows that as the vaporization of mixture improves, the mixture of methanol becomes homogenized and the fuel film flow decreases, which results in the increase of vaporization rate. And the increase of the vaporization rate improves the engine performance of the alcohol-fueled spark ignition engine.

**Key Words:** Vaporization Rate, Droplet, Liquid Film, Flow of Vapor, Fuel Vaporizing Device, Equivalence Ratio

## Nomenclature

$A_a$ : Area per unit length
$A_m$ : Area per unit length mesh
$C_p$ : Specific heat at constant pressure
$D$ : Diameter of inlet pipe
$d$ : Diameter of a mesh inlet pipe
$G_f$ : Mixture flow of vapor
$G_s$ : Mixture flow of droplet
$G_w$ : Mixture flow of liquid
$h$ : Heat transfer coefficient
$K$ : Thermal conductivity
$L_a$ : Length of the FVD
$M$ : Mass
$N$ : Engine speed
$Pr$ : Prandtl number
$Q$ : Heat transfer rate
$Q_h$ : Heat quantity of mixture
$R_{af}$ : Air fuel ratio
$Re$ : Reynolds number
$T$ : Temperature
$T_b$ : Bulk temperature
$T_w$ : Wall temperature
$V_d$ : Displacement volume
$V$ : Velocity
$X_e$ : Vaporization rate
$\gamma$ : Specific weight
$\delta$ : Interval of a mesh
$\eta_v$ : Volumetric efficiency
$\theta$ : Crank angle
$\rho_a$ : Density of air
$\phi$ : Equivalence ratio
$\rho$ : Density

**Subscripts**

$a$ : Air
$f$ : Fuel
$i$ : $i$'th component

## 1. Introduction

Methanol is regarded by many experts as the alternative automotive fuel. The reasons are that methanol can be available abundantly extracted from natural gas or through coal processing, that the fuel cost will become competitive with those of current petroleum fuels, that methanol liquid at

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normal temperature can be handled with ease for its storage and transportation, and that its applied technology is relatively easy to develop(Harrington et al., 1974).

For the application of methanol as automotive fuel, basic application and development have been made extensively since the oil crisis, and such technology as the neat methanol spark ignition engine system has been upgraded for practical application.

Use of methanol in these engine has many advantages high octane quality without lead additives, power increase, energy economy, lean misfire limit extension, and reduction of exhaust emissions with such as nitro oxide, carbon monoxide and unburned hydrocarbon(Schweikert, 1976 ; Hagen, 1977). However, fueling with methanol has been found to increase the exhaust gas concentration of aldehydes 3 to 5 times, when compared to operation with gasoline.

And methanol has a defect that latent heat of vaporization is higher than that of gasoline, it could be supposed that the inducted methanol is not sufficiently vaporized(Baines et al., 1982 ; Bechtold et al., 1980), in which case the liquided methanol can play a role of solvent, wash off the lubricant film from the cylinder wall, leak out of the piston ring gap, and mix with the lubricant oil, decreasing its viscosity and thereby causing high lubrication loss(Katoh et al., 1986 ; Samaga et al., 1986). Due to its high liquid film flow, methanol engine requires additional heating. Therefore, in order to improve vaporization of methanol, the authors have made the fuel vaporizing device which can eliminate the fuel film flow and heat the mixture(Mun et al., 1986 ; Han et al., 1993).

This paper represents a study on the characteristics of vaporization and engine performance in methanol-fueled spark ignition engine.

## 2. Design of the Fuel Vaporizing Device

As shown in Fig. 1, the mixture behavior in intake manifold can be devided into the flow of vapor $G_f$, droplet $G_s$ and liquid film $G_w$. The

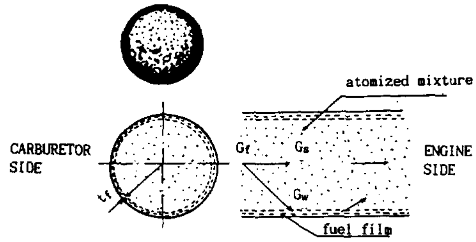


**Fig. 1** Mixture behavior in intake manifold

mixture flows within inlet pipe are vaporized or condensated according to air temperature before the mixture enters the combustion chamber.

Fueling with methanol had been found to increase the liquid flow and the lubrication loss, compared to operation with gasoline. Therefore, in order to cope with the defect, the authors have designed the FVD which can eliminate the fuel film flow and improve mixture. Calculation of the length of the FVD is shown as follows. Heating value necessary for complete vaporization is given by

$$Q=G_f[(C_{pa}R_{af}+C_{pf})(T_f-T_w)+L_f] \quad (1)$$

where, $G_f=G_a/R_{af}=\eta_v \rho_a V_d N/120R_{af}$,

and the heat transfer rate for inlet inner surface area can be determined by

$$Q=h\ A(T_w-T_b), \quad (2)$$

where, $h=0.023\ Re^{0.8}Pr^{0.4}K/D$,

and from the Eqs. (1) and (2), length of the FVD given by

$$L=\frac{\eta_v \rho_a V_d N[(C_{pa}R_{af}+C_{pf})(T_f-T_w)+L_f]}{120R_{af}\{\pi D h(T_w-T_b)\}}. \quad (3)$$

Liquid flow rate considered, the length of the FVD can be determined by

$$L_a=\gamma L, \quad (4)$$

and in order to reduce the length of the FVD, the increase of heat transfer area are required.

The length of the FVD which mesh(stainless steel mesh, #100) is installed can be determined by

$$L_m=\frac{A_a}{A_m}L_a, \quad (5)$$

where, $A_a$ : area per unit length, $\pi\ D$

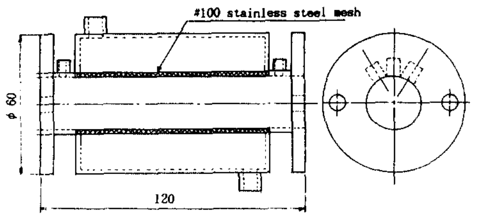


**Fig. 2** Specified diagram of fuel vaporizing device

$A_m$ : area per unit length mesh,
$(2\pi^2 dDN_m)/(d+\delta)$.

As shown in Fig. 2, the FVD is designed ; the length of the FVD is 80mm, the thickness of mesh which is installed inside the FVD is 2mm.Heat source of the FVD is the water heated in the water tank by electric resistance device and the heating water is recirculated by water pump.

## 3. Experimental Apparatus and Method

The engine used in this test is a water cooled single cylinder engine. The engine specification is given in Table 1.

Fig. 3 shows various components of the experimental apparatus. The engine is coupled to a *D. C* dynamometer. To analyze combustion characteristics, pressure transducer is attached to combustion chamber, and the data obtained thereby are input into combustion analyzer. To test the performance of emission in each operating condition, exhaust gas analyzer was used.

Main objective of this investigation is to determine the effect of FVD and the characteristics of vaporization. The temperature as well as the velocity are measured in the entrance and exit of the FVD.

As shown in Fig. 4, the temperature and velocity of mixture is measured with the travelling device composed of the thermocouple and pitot tube moving in it. The measure points which are vertically inclined 30°are 11 with 2mm in interval.

Due to the small difference of static pressure as a function of height of inlet pipe, the total pressure of mixture is measured only with the pitot tube and the static pressure is measured at one point which is positioned in top of inlet pipe.

**Table 1** Specification of test engine used

| Item | Engine |
|---|---|
| Type | 4 stroke cycle, horizontal type |
| No. of cylinder | 1 cylinder |
| Cooling system | water cooled |
| Method of ignition | battery |
| Bore × stroke | 85 × 85 mm |
| Displacement volume | 482 cc |
| Compression ratio | 4.8 |
| Length of con. rod | 150 mm |

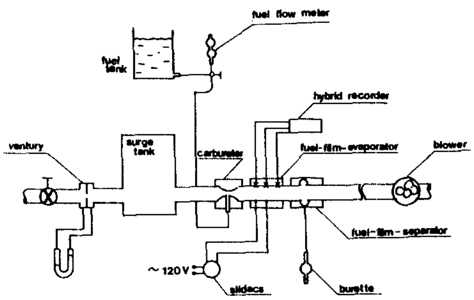


**Fig. 3** Schematic diagram of experimental apparatus

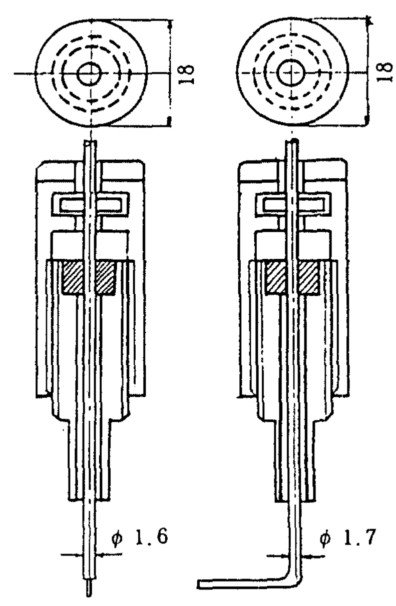


**Fig. 4** Travelling device of pitot tube and thermocouple

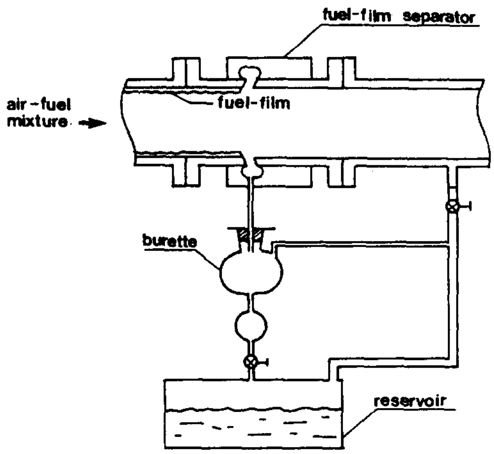


**Fig. 5** Schematic diagram of fuel film seperator

The quantity of liquid flow is measured with seperator device. The schematics of fuel film seperator is shown in Fig. 5. Throughout testing, engine speed is 1800 rpm in which a best torqe can be achieved and spark is timed at the MBT(minimum spark advance for best torque). The cooling water temperature is kept at 80±2℃ to eliminate their dominant influence.

Air flow is determined on the difference pressure in the entrance and exit of orifice(round type, inner diameter 12mm) with the *U* manometer. Fuel flow is controlled by the needle valve in carburetor. Equivalence ratio is defined as the actual air fuel ratio to stoichiometric air fuel ratio. The operation variable of test engine are the heating water temperature and equivalance ratio. The heating water temperature depends on the electric resistance device in water tank.

## 4. Characteristics of Vaporization

The velocity of mixture is shown in Fig. 6 as a function of inlet height at the heating water temperature 80℃. Due to the effect of throttle valve, velocity of the FVD entrance is represented the minimum level in inlet pipe center. Maximum value of the FVD exit is achieved at center of inlet pipe. It represents general turbulent velocity profile of tube and the velocity profile does not depend on increase of the heating water temperature.

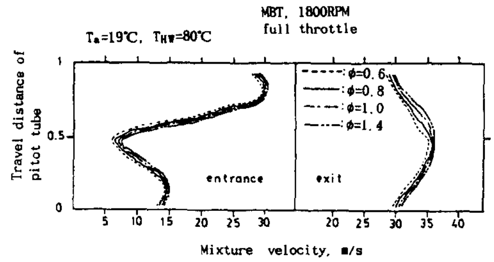


**Fig. 6** Velocity profile with equivalence ratio in the fuel vaporizing device

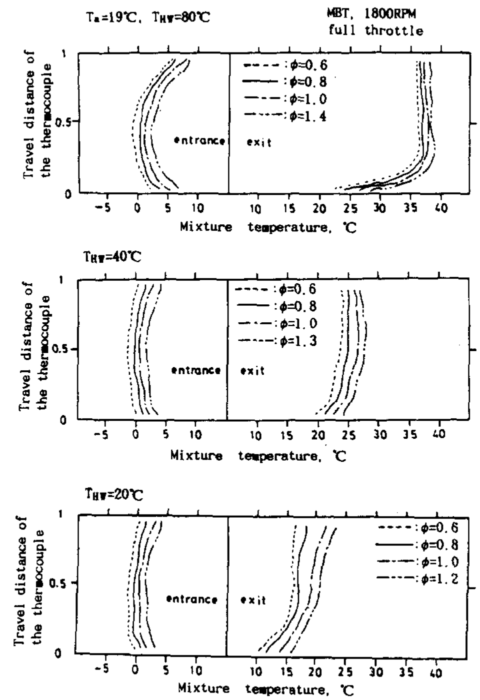


**Fig. 7** Temperature profile with equivalence ratio and water temperature in the fuel vaporizing device

The temperature profile of mixture is shown in Fig. 7 as a function of inlet height and heating water temperature. According to increase of the heating water temperature, the temperature value of mixture is found to increase.

The bulk temperature of the FVD entrance and exit can be determined by

$$T_b = \frac{\sum_{i=1}^{n} \rho_i C_p V_i T_i A_i}{\sum_{i=1}^{n} \rho_i C_p V_i A_i}, \tag{6}$$

and, $\Delta T_b$ is defined as a difference of bulk temperature in the FVD entrance and exit. Heat quantity of mixture which passes the FVD can be determined by

$$Q_h = M_a C_{pa} \Delta T_b + M_f C_{pf} \Delta T_b, \qquad (7)$$

and $T_b$ and $Q_h$ are shown in Fig. 8 as a function of the heating water temperature. They are shown to increase as the heating water temperature increases.

Liquid film flow as a function of equivalence ratio and the heating water temperature is shown in Fig. 9. As the heating water temperature increases, liquid flow decreases. The test shows that the increase of the heating water temperature can homogenize the mixture. Vaporization rate ($X_e$) can be determined by

$$X_e = \frac{G_f - G_w}{G_f} \times 100. \qquad (8)$$

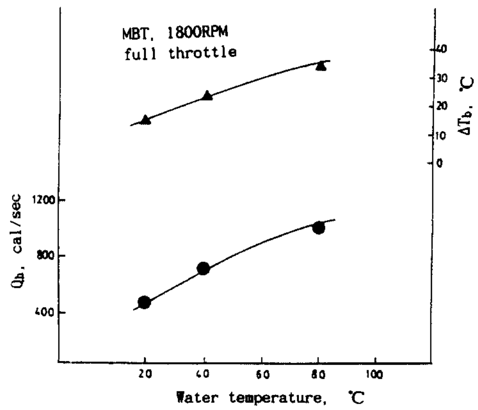


**Fig. 8** Heat flow and temperature rise with water temperature in the fuel vaporizing device

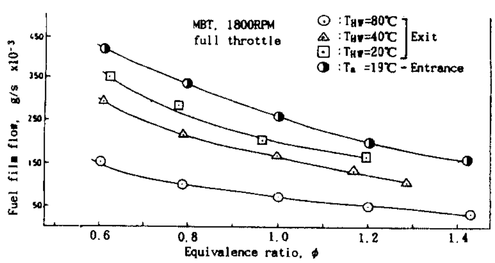


**Fig. 9** Change of fuel film flow with equivalence ratio and water temperature in the fuel vaporizing device

The trend of liquid flow rate and vaporization rate is shown in Fig. 10 as a function of the heating water temperature and equivalence ratio.

The vaporization rate increases with increase of the heating water temperature and equivalence ratio. At the heating water temperature 80℃, vaporization rate is nearly 95%. And according to increase of the heating water temperature, the lean misfiring limit can be extended.

Output of power as a function of equivalence ratio and heating water temperature is shown in Fig. 11. Maximum value of output power is achieved in the equivalence ratio 0.8 to 0.9. And it shows that output power is reduced in the rich

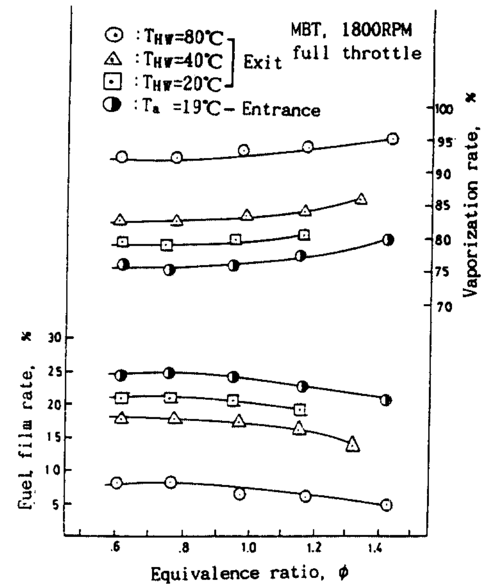


**Fig. 10** Vaporization rate and fuel film rate with equivalence ratio and water temperature in the fuel vaporizing device

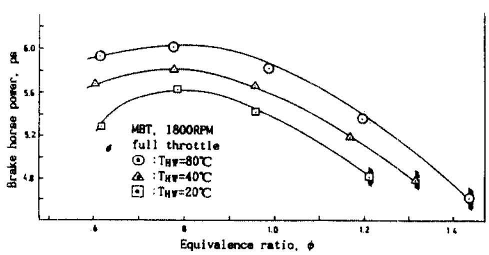


**Fig. 11** Change of brake horse power with equivalence ratio and water temperature in the fuel vaporizing device

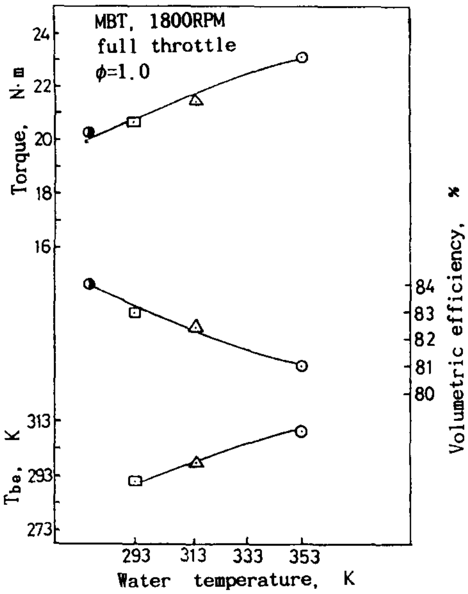


**Fig. 12** Comparisoon of bulk temperature and volumetric efficiency and torque with water temperature in the fuel vaporizing device

or lean mixture region. As the heating water temperature increases, the lean misfire limit and output power increase largely.

Comparison of bulk temperature, volumetric efficiency and torque dependent on the change of water temperature in the fuel vaporizing device is shown in Fig. 12. The volumetric efficiency and torque are shown to increase with heating of mixture in a spark ignition engine. In the methanol engine with the FVD in spite of the decrease of volumetric efficiency, the trend of increase of engine torque seems to result from homogenized mixture from which liquid flow is excluded. Maximum of pressure and heat release are shown in Fig. 13 as a function of the heating water temperature. As the heating water temperature increases, maximum of pressure and heat release increase.

Fig. 14 shows the mass burned rate as a function of the heating water temperature. Here, combustion duration is defined as crank angle required to burn from 10 to 90% of the mixture. The combustion duration in the heating water temperature 80℃ is shorter than that in the heating water temperature 20℃. Additionally the

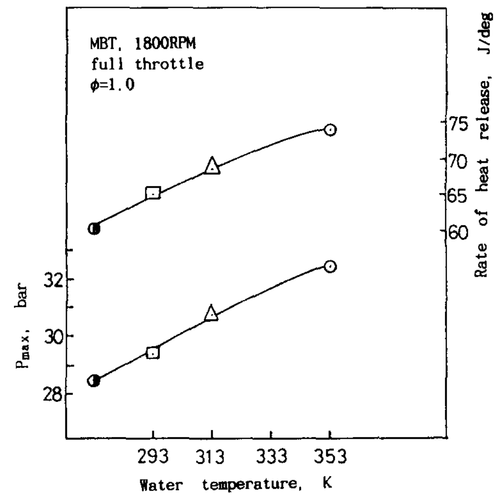


**Fig. 13** Comparisoon of maximum pressure and maximum heat release with water temperature in the fuel vaporizing device

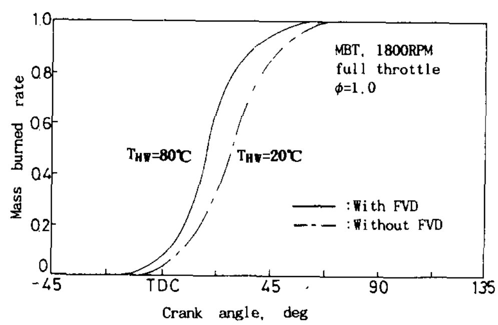


**Fig. 14** Mass burning process with the fuel vaporizing device and without the fuel vaporizing device

increase of vaporization rate with the heating water temperature makes mixture homogenized, the combustion duration shortened, and torque increase.

## 5. Conclusions

The mixture of methanol-fueled spark ignition engine with the fuel vaporizing device is homogenized as the heating water temperature increases. According to increase of heat quantity of mixture, lean misfire limit is extended remarkably. As liquid fuel flow is eliminated, mixture is supposed to be homogenized.

The improvement of mixture of exit side has a high effect on the results of the reduced liquid flow, the increased vaporization rate, and mesh effect which increses heat transfer area.

Due to the improved mixture with low liquid fuel flow, engine torque increases dependending on increase of the heating water temperature.

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