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The Effects of Design Parameters on the Friction Characteristics in the Valve Train System

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Abstract : This paper is a report on the parametric study of the friction characteristics on the direct acting type OHC valve train system. The numerical simulation was performed by using the IV-TAP. Dynamic analysis by using the lumped mass method was previously performed to define the acting load. The friction characteristics were analyzed by using the partial asperity contact model. The effects of operating conditions and major design parameters on the total driving torque were investigated. From the analytical prediction, it is found that valve spring stiffness, surface roughness, and base circle radius are the main factors to reduce the frictional loss on the valve train system.

Key words : Valve train, asperity contact model, total torque, stiffness, roughness, frictional loss

Introduction

The most important subjects to be considered in modern engine development are high power and lower fuel consumption. Therefore new technology, for example GDI or EMV, is proposed to achieve the aim. In addition to the above efforts, other researches to reduce the friction losses have been performed. The most of engine friction losses are occurred at the valve train system and piston assembly. Specially, the severe lubricating condition in cam and tappet interface causes some troubles like excessive wear and high friction losses. Therefore the accurate understanding of friction characteristics on valve train system is essential to the reduction of friction losses, and the increase of engine durability.

As a result, several researches on the theoretical prediction and experimental measurement of the lubrication characteristics in cam and tappet interface has been performed. Staron and Willermet [1] presented the analytic model for the friction loss on the rocker arm type valve train system, and Helden *et al.* [2] investigated method to measure the friction at the cam and tappet interface. Also, Crane and Meyer [3] proposed the various approximate equations to calculate friction force on the center pivot type OHC (overhead cam) valve train system. Ji *et al.* [4] has performed the theoretical study for the effects of design parameters on the friction loss. Besides, some researches have performed to examine the effect of tappet rotation on the friction loss. [5~8]

Generally, the effect of design parameters on the valve dynamics is only considered in the valve train system design. However, to develop the low friction engine, the friction characteristics have to be investigated during the design procedure.

Therefore the aim of this paper is to establish the database for the design of low frictional valve train system by investigating the effects of design parameters and operating conditions on friction characteristics. The theoretical prediction of friction characteristics is performed by using the IV-TAP (Integrated Valve Train System Analysis Program) [9]. From this study, the main effects on the friction characteristics are found out, and the predicted results can be applied to the design of low frictional valve train system.

Basic Theory

Dynamic Model

Figure 1 shows a dynamic model of valve train system to analysis the acting force at cams and tappets interface. The valve train system is divided into 9 lumped mass elements and the HLA chamber and plunger are considered as nonlinear dampers. The valve spring is divided into five mass elements to reflect the spring surge phenomenon. The dynamic equation of above model is represented as matrix form, which is presented as follows:

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = [F] \tag{1}$$

The solutions of Eq. (1) are obtained by using the numerical integration.

Friction Model

If the acting forces at the cam and tappet interface are calculated using Eq. (1), the contact areas and maximum contact pressure are calculated through the Hertzian line contact theory. The minimum and central oil film thickness

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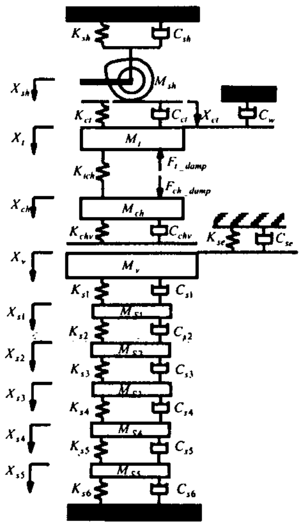

Fig. 1. Dynamic model of valve train system.

between cam and tappet under the isothermal condition are calculated by using the elasto-hydrodynamic theory, which previously presented by Dowson *et al.* [10].

$$h_{min} = 2.65R'\left(\frac{\mu_o V_e}{E'R'}\right)^{0.7}(\alpha E')^{0.54}\left(\frac{W}{E'R'l}\right)^{0.1} \quad (2)$$

$$h_{cen} = 3.06R'\left(\frac{\mu_o V_e}{E'R'}\right)^{0.69}(\alpha E')^{0.56}\left(\frac{W}{E'R'l}\right)^{0.1} \quad (3)$$

For the mixed lubrication analysis, we make assumption as follows. The friction between cam and tappet is composed of viscous friction due to hydrodynamic and boundary friction by asperity contact. According to the asperity contact theory, which presented by Greenwood and Tripp [11], the asperity contact load through the elastically deformed asperity is defined as:

$$W_b = \frac{8\sqrt{2}}{15}\pi(\eta\beta\sigma)^2\sqrt{\frac{\sigma}{\beta}}AF_{5/2}\left(\frac{h}{\sigma}\right) \quad (4)$$

In Eq. (4), the upper limit of the asperity contact load, W_b is the acting force between cam and tappet, W. If we assume the Gaussian distribution of surface roughness, the function $F_{5/2}(h/\sigma)$ in Eq. (4) is defined as follows:

$$F_n(H) = \frac{1}{\sqrt{2\pi}}\int_H^{\infty}(s-H)^n e^{-\frac{s^2}{2}}ds \quad (5)$$

The asperity contact area, A_c, is written by:

$$A_c(H) = \pi^2(\mu\beta\sigma)^2 F_2(H) \quad (6)$$

In Eq. (4), (5), and (6), the central oil film thickness between cam and tappet is adopted as the nominal oil film thickness, h. Therefore, the friction force by boundary lubrication is induced by shear of thin film and asperity contact. In this case, the shear stress and friction force are defined as follows

$$\tau_b = \tau_o + \gamma p_b \quad (7)$$

$$F_b = \tau_o A_c + \gamma W_b \quad (8)$$

However, the shear stress by hydrodynamic lubrication is induced by sliding movement of cam and tappet, which is defined as:

$$\tau_h = \mu(p, T)\frac{V_s}{h_{cen}} \quad (9)$$

$$\mu = \mu_0 exp(\alpha_1 \Delta T - \alpha_2 p_h)$$

In Eq. (9), μ is the viscosity of working oil, which is the function of pressure and surface instance temperature.If the oil film is very thin, the shear stress becomes extremely high, so, the shear stress can not be calculated by using the Eq. (9). In this case, the shear stress can be calculated by using the limit shear stress theory, which can be written:

$$\tau_h = \tau_o + mp_h \quad (10)$$

The total friction force is the sum of viscous and boundary friction force, and that is written by:

$$F = (\tau_b + \tau_h)A \quad (11)$$

Also, the driving torque in camshaft is sum of torque due to the normal load and friction torque, and that can be written by:

$$T_t = T_f + T_w \quad (12)$$

If the cam profile is symmetry and dynamic characteristics are excluded, the driving torque means frictional torque.

The analysis for the valve dynamics and friction are performed by using the IV-TAP.

Results and Discussion

Figure 2 shows the interactive data input windows, and Figure 3 shows the examples of dynamic and friction analysis results by using the IV-TAP.

Following figures represent the various effects of design parameters using a standard data in Table 1. and 20% variation of that data.

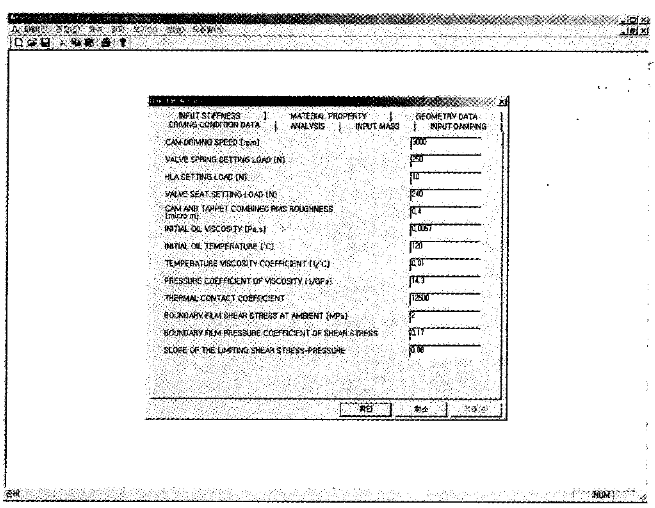

Fig. 2. Example of input window in IV-TAP.

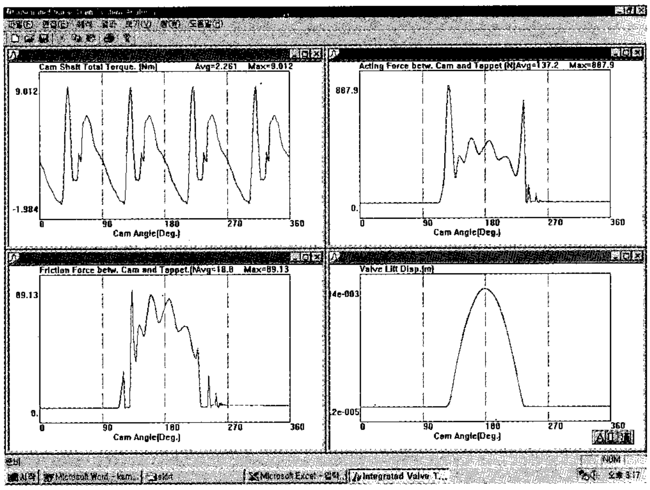

Fig. 3. Example of calculation results of IV-TAP.

Table 1. Standard input data

cam width	11 mm
base circle radius	18 mm
spring stiff	34,600 N/m
set load	250 N
value mass	36.8 g
viscosity	0.0054 Pa · s
RMS roughness	0.4 micro m

Figure 4 and 5 show the effects of cam width and radius of cam base circle on the driving torque. At first, the total torque is decreased according to camshaft rotational speed up. In the mixed lubrication region, the increase of speed reduces the friction force. The reduction of cam width increases boundary friction in total friction due to the reduction of oil film thickness, but viscous friction term is decreased. Therefore the total driving torque is decreased according to the cam width reduction, but the effect is very small. Contrary to the cam width, the base circle radius has important influence on the reduction of driving torque.

The effect of valve spring stiffness is displayed in Fig. 6. The stiffness of valve spring is dominant factor to affect valve

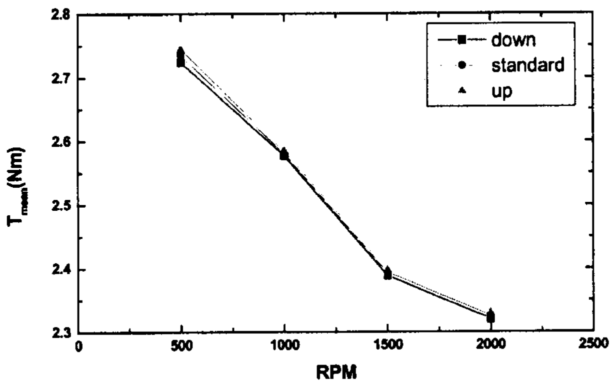

Fig. 4. The effects of cam width on the total driving torque.

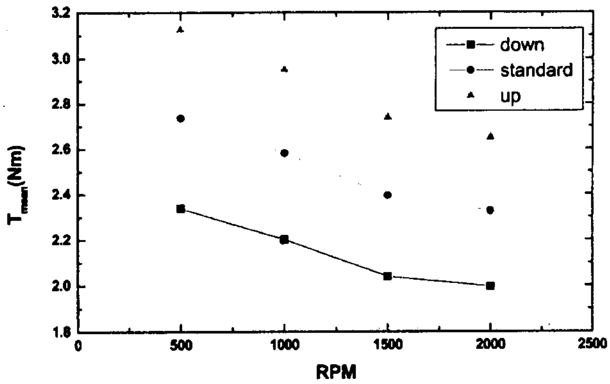

Fig. 5. The effects of base circle radius on the total driving torque.

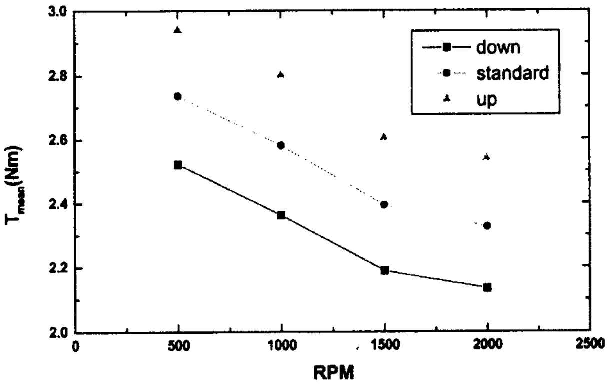

Fig. 6. The effects of valve spring stiffness on the total driving torque.

dynamics and acting force at the cam and tappet interface. In spite of obtaining the stable valve behavior, an increase in spring stiffness causes the friction loss to increase. The effect of valve spring stiffness on the driving torque is very significant.

Figure 7 shows the effect of initial valve spring setting load on the driving torque. The decrease of setting load reduces the acting load at the cam and tappet interface. Therefore, the friction force is decreased. Form the Figs. 6 and 7, reduction of valve spring stiffness and setting load have advantage on the low friction, but they have demerit on the valve dynamics.

The effect of valve mass is displayed in Fig. 8. The increasing of valve mass is effective on the stability of valve dynamics, but it causes the slightly fall in driving torque. The valve mass reduction is effective at the high speed operating condition because the dynamic force induced by inertia force is dominant at high speed.

Figure 9 shows the effect of viscosity variation on the driving torque. The low-viscosity oil increases the driving torque by the increasing of friction torque. The results in Fig. 9 coincide with mixed lubrication characteristics in *Stribeck* diagram. The low-viscosity oil is effective to reduce the frictional loss in engine bearing, which are operated in hydrodynamic lubrication regime. However, in mixed lubrication regime like as valve train system, the low-viscosity

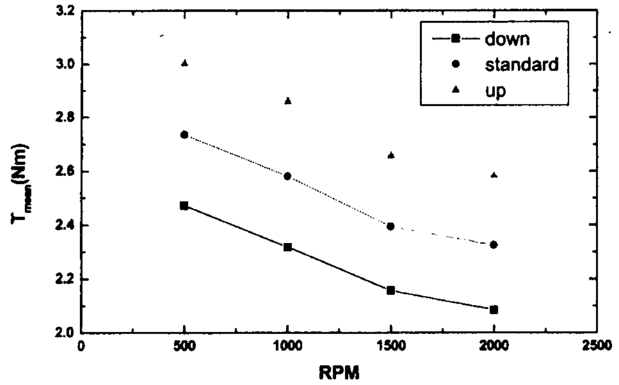

Fig. 7. The effects of valve spring setting load on the total driving torque.

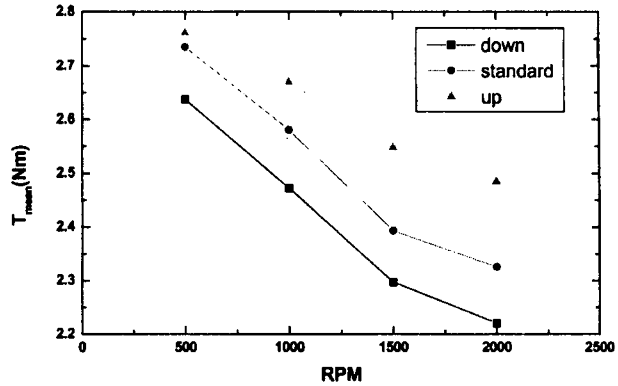

Fig. 10. The effects of surface roughness on the total driving torque.

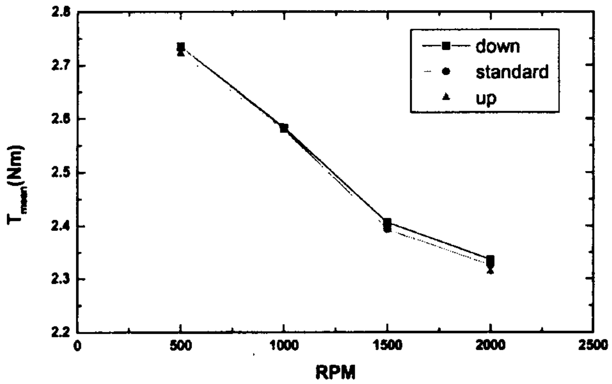

Fig. 8. The effects of valve mass on the total driving torque.

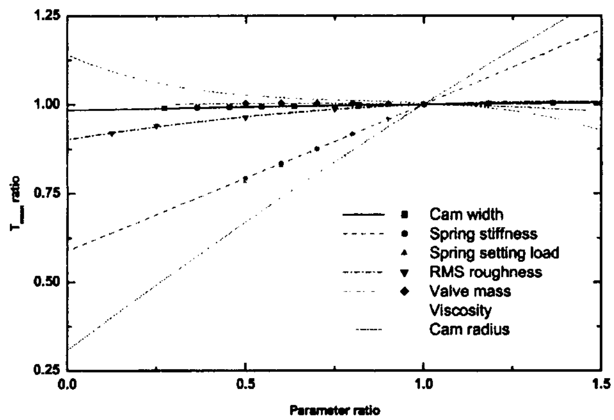

Fig. 11. The effect of design parameters on the mean driving torque.

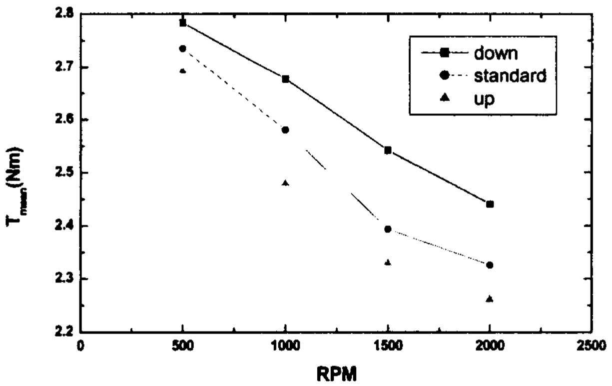

Fig. 9. The effects of oil viscosity on the total driving torque.

oil increases the frictional power loss.

Figure 10 shows the effects of surface roughness *rms* height on the mean driving torque. The boundary friction due to the asperity contact is decreased according to the reduction of roughness height. Therefore total driving torque is decreased due to the reduction of frictional torque.

Figure 11 summarizes the effects of operating conditions and design parameters on the mean driving torque. From the result, it is found that spring stiffness, set load and base circle radius are the main factor to affect on the mean driving torque, but the effect of reciprocating mass and cam width are not significant.

Conclusions

The effects of design parameters and operating conditions on the frictional characteristics in HLA type OHC valve train system is investigated by using the IV-TAP. The following conclusions are derived.

1. The valve train system shows mixed lubrication characteristics.
2. The equivalent mass, and cam width marginally affect on the frictional driving torque.
3. The valve spring stiffness and initial setting load of valve spring are important design factors to reduce the frictional loss.
4. The low-viscosity oil increases friction loss of valve train system.
5. It is thought that the above tool is very useful to design and verify the low friction valve train system.

Nomenclature

A	: Contact area	m^2
A_c	: Asperity contact area	m^2
b	: Half width of contact rectangle	m
C	: Damping coefficient	Ns/m
E	: Reduced elastic modulus	Pa

F	: Total friction force	N
F_b	: Asperity contact friction force	N
h	: Nominal oil film thickness	m
h_{cen}	: Central oil film thickness	m
H_{min}	: Minimum oil film thickness	m
K	: Spring coefficient	N/m
l	: Length of contact rectangle	m
M	: Mass of each element	Kg
m	: Coefficient of lubricant-limiting shear stress-pressure relation	
p_b	: Asperity load per unit area	N/m^2
p_h	: Hydrodynamic load per unit area	N/m^2
p_{max}	: Maximum Hertzian stress	Pa
R'	: Reduced radius of curvature	m
T_f	: Torque due to friction force	Nm
T_t	: Total torque	Nm
T_w	: Torque due to normal force	Nm
V_e	: Entrain velocity	m/s
V_s	: Sliding velocity	m/s
W	: Total contact load	N
W_b	: Asperity contact load	N
α	: Pressure coefficient of viscosity	m^2/N
α_1	: Temperature coefficient of viscosity	1/°C
α_2	: Pressure coefficient of viscosity	1/GPa
β	: Radius of asperity tip	m
γ	: Pressure coefficient of boundary shear strength	
η	: Number of asperity per unit area	$1/m^2$
θ	: Rotational angle of cam	rad
μ	: Lubricant viscosity	Pa· s
μ_o	: Lubricant viscosity at ambient viscosity	Pa· s
σ	: Standard deviation of asperity height	m
τ_o	: Boundary shear strength of ambient pressure	N/m^2
τ_b	: Boundary shear stress	N/m^{232}
τ_h	: Hydrodynamic shear stress	N/m^2
ω	: Angular velocity	rad/s

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