A NUMERICAL APPROACH TO CALCULATE CREEP IN ROLLER FOLLOWER VALVE TRAIN BASING ON FRICTION AND LUBRICATION MODELING

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ABSTRACT
A numerical approach basing on friction and lubrication analysis has been developed to determine the creep at cam/roller interface in end pivoted roller finger follower valve train. The kinematic and dynamic analysis at the geometrical mating surfaces of cam and roller follower has been carried out to predict the required instantaneous motion parameters and normal loading whereas the oil film thickness is determined using lubrication analysis. The tangential forces have been computed precisely using elastohydrodynamic and mixed lubrication concept for a complete cam cycle. At low camshaft operating speeds, the creep proves to be low whereas the creep increases significantly under the influence of high tangential loading at higher operating speeds.

Keywords: asperity interaction; cam/roller interface; creep; normal loading; oil film thickness; tangential forces.

UNE APPROCHE NUMÉRIQUE POUR CALCULER LE FLUAGE D'UN POUSSOIR À GALET DANS UN MÉCANISME DE DISTRIBUTION EN SE BASANT SUR LA MODÉLISATION DE LA FRICTION ET DE LA LUBRIFICATION

RÉSUMÉ
Une approche numérique basée sur l'analyse de la friction a été développée pour déterminer le fluage à l'interface de la came de verrouillage dont l'extrémité articulée du doigt mobile du poussoir à galet d'un mécanisme de distribution. L'analyse cinématique et dynamique à la surface d'accouplement géométrique de la came et du poussoir à galet a été produite pour prévoir les paramètres de mouvements instantanés et le chargement normal, tandis que l'épaisseur de la pellicule huileuse est déterminée en utilisant l'analyse de la lubrification. Les forces tangentielles ont été calculées de façon précise en utilisant le concept de l'élastohydrodynamique et une composition mixte lubrifiante pendant un cycle complet de la came. À une vitesse de fonctionnement peu élevée, le fluage s'avère bas, tandis que le fluage augmente significativement sous l'influence d'un chargement à une vitesse tangentielle élevée de fonctionnement.

Mots-clés : interaction des aspèritès; interface du galet de came; fluage; chargement normal; épaisseur de pellicule huileuse; forces tangentielles.

NOMENCLATURE

A	Distance between pivot of roller finger follower and the center of curvature of follower face in contact with the camshaft
$\mathbf{A}'$	Apparent area of contact
$\mathbf{A}_a$	Actual area of contact
B	Distance between pivot of roller finger follower and the center of curvature of follower face in contact with the valve
b	Hertzian contact half width
D	Distance between the cam axis of rotation and the pivot of roller finger follower
E_1	Elastic modulus for cam
E_2	Elastic modulus for roller
E'	Equivalent modulus of elasticity
F	Tangential force at cam/roller interface
F_a	Friction force due to asperity interaction
F_r	Rolling friction at cam/roller contact
F_s	Sliding friction at cam/roller contact
G	Dimensionless material parameter
h_{cen}	Central oil film thickness
J	Moment of inertia of roller
K	Spring stiffness
l_v	Valve lift
L	Roller width
M	Equivalent reciprocating mass of the valve system
M_f	Mass of roller finger follower
m	Limiting shear stress and pressure relation
P	Pressure distribution in the contact zone
$p_{\max}$	Maximum Hertzian stress
R	Equivalent radius of curvature
R_b	Cam base circle radius
r_c	Cam radius of curvature
r_1	Roller outer ring radius
T	Operating temperature
T_0	Atmospheric temperature
U	Dimensionless speed parameter
V_c	Velocity of contact point relative to cam
V_f	Velocity of contact point relative to roller follower
V_e	Mean lubricant entraining velocity
V_s	Sliding velocity of mating surfaces
W_l	Normal load at cam/roller interface
W_a	Asperity load
W_h	Hydrodynamic load
W'	Dimensionless load parameter
v_1	Poisson ratio of cam material
v_2	Poisson ratio of roller material
α	Pressure-viscosity coefficient
σ	Composite roughness height of cam/roller
β	Temperature-viscosity coefficient
γ'	Rate of change of shear stress with pressure
δ	Pre compression of spring
$\dot{\Omega}$	Roller angular acceleration
η_0	Lubricant viscosity at atmospheric temperature and pressure
τ_0	Eyring stress of lubricant

τ_L	Limiting shear stress
μ	Coefficient of friction
μ_b	Boundary coefficient of friction
μ_{brg}	Coefficient of friction in needle roller bearing
λ	Fixed angle between the lines connecting pivot to curvature centers of follower surfaces in contact with the cam and valve
χ	Fixed angle between the line connecting pivot to cam center and the direction of valve motion
ϕ'	Variable angle between the line connecting the centers of rotation of roller follower and cam and the x-axis
γ	Variable angle between the lines joining cam and roller follower center of rotation to the center of curvature of roller finger follower face in contact with the valve
ω	Camshaft speed

1. INTRODUCTION

In the automotive industry, production of high performance and fuel efficient vehicles has grown in great importance to meet the customer demands and exhaust emission standards. This drive has led to many design changes in the engine components which has resulted in high loadings at the main tribological components. Increased spring stiffness, multi valves system, higher valve lift and variable valve timing has increased the loading in the valve train area. The engine valve train is often subjected to severe operating conditions due to higher contact loading varying in nature, raised operating temperatures and continuously changing lubricant entrainment velocity which may result in excessive friction and wear of the mating surfaces. In the modern passenger cars, there is a wide spread in the use of roller follower valve train (Fig. 1) due to its improved power output and better fuel economy. In previous experimental research works, Staron and Willermet [1], Sun and Rosenberg [2] and Bair et al. [3] have reported substantial reduction in the valve train power losses by using the roller follower instead of sliding tappet. However, deterioration of the mating surfaces of cam and roller operating under marginal lubrication due to relative sliding was identified by Duffy [4] and Lee et al. [5] experimentally. Gecim [6] identified the shortening of roller fatigue life under low specific film thickness values. Miyamura [7] analyzed the wear problem in roller follower mechanism. However, no work has been reported on the possible existence of creep at cam/roller interface which may influence the tribological performance of roller follower valve train substantially. The slipping behavior due to creep in the contact patch of cam and roller may raise the valve train power losses by increasing the sliding friction, hampering of lubrication conditions and deterioration of mating surfaces. Pure rolling at cam/roller contact is desirable to avoid these problems. Moreover, the lack of information about the creep would restrict to predict the oil film strength, power losses and even the deterioration of mating surfaces accurately in the valve train.

In end pivoted roller finger follower valve train, the cam/roller interaction is dominated by the rolling contact. During operation, both the mating surfaces of cam and roller will exert tangential loading to each other leading to the elastic deformation. This elastic deformation at cam/roller interface usually does not match resulting into the velocity difference and thus causing the creep which would result in deviation of kinematics as compared to the pure geometric rolling without slipping behavior of dry and un-deformed cam/roller mechanism. Since the cam/roller interaction is often subjected to high varying tangential loadings at different engine operating speeds which may increase the tendency of creep and can affect the tribological performance of the valve train substantially. Hence, a numerical approach is required to determine the instantaneous creep at cam/roller interface allowing to investigate the effects of various engine operating conditions and lubricant rheology on the occurrence of creep.

In this research paper, for the very first time, a new numerical approach basing on the friction and lubrication analysis in mixed lubrication regime has been developed and successfully employed entirely from a new perspective to determine the instantaneous creep at cam/roller interface as a function of cam angle. A

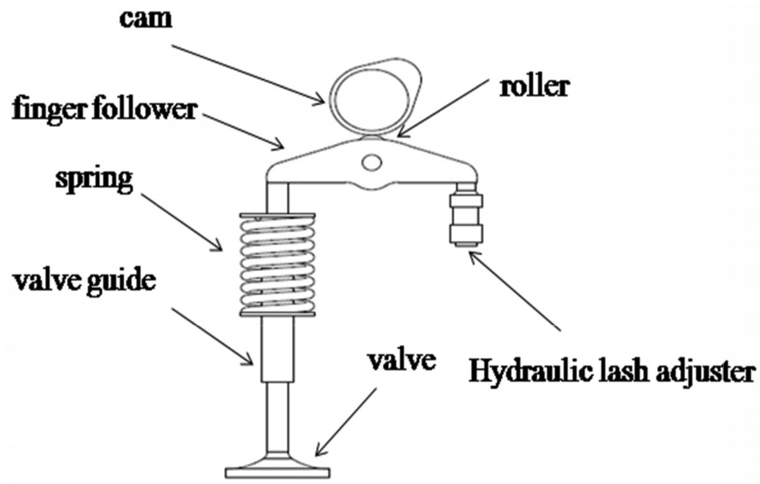

Fig. 1. End pivoted roller finger follower valve train with overhead camshaft.

simple approach adopted by Fengming [8] to determine the slip ratio in the valve train basing on the classical elastohydrodynamic lubrication theory may be considered having little relevance to the presented model. But in that approach, effects of the surface roughness, non Newtonian behavior of lubricant and variations in the lubricant viscosity due to change in temperature and pressure were not considered which simplified the modeling process but reduced the accuracy considerably whereas these tribological parameters play an important role on the valve train performance and have been considered comprehensively in the reported model. Moreover, the developed model provides an opportunity to investigate the effects of various engine operating conditions and lubricant rheology on the occurrence of creep in detail and can also help to control the conditions promoting the creep at cam/roller interface.

2. VALVE TRAIN FRICTION AND LUBRICATION MODEL

In this article, the model is based on the kinematic and dynamic analysis to predict the motion and normal loading parameters. The effects of asperity interactions, non Newtonian behavior of lubricant and variations in lubricant viscosity due to change in temperature and pressure have been considered comprehensively in the development of model. The lubrication analysis has been carried out to predict the central oil thickness. The tangential forces acting at cam/roller interface were determined and used to calculate the instantaneous creep at various camshaft operating speeds. The basic geometry of the system is shown in Fig. 2. The cam rotates about its centre O, Q is the centre of rotation for the roller and U is the pivot point of roller finger follower system. Angles χ and λ and distances $A = UQ$, $B = UV$ and $D = UO$ are fixed in the system. P is the contact point of cam and roller.

2.1. Kinematic Analysis

The kinematic analysis provides an information about the velocity of contact point relative to cam V_c and the radius of curvature of cam surface r_c at the contact point P and variable angle γ. These parameters can be determined by using the previous work of Dyson [9] carried out on the sliding finger follower. The contact point velocity relative to roller follower V_f is determined with the help of torque balance analysis at the roller follower center.The sliding velocity V_s between the cam and roller surfaces and the lubricant mean entrainment velocity V_e is required to predict the oil film thickness and friction due to shear of lubricant at

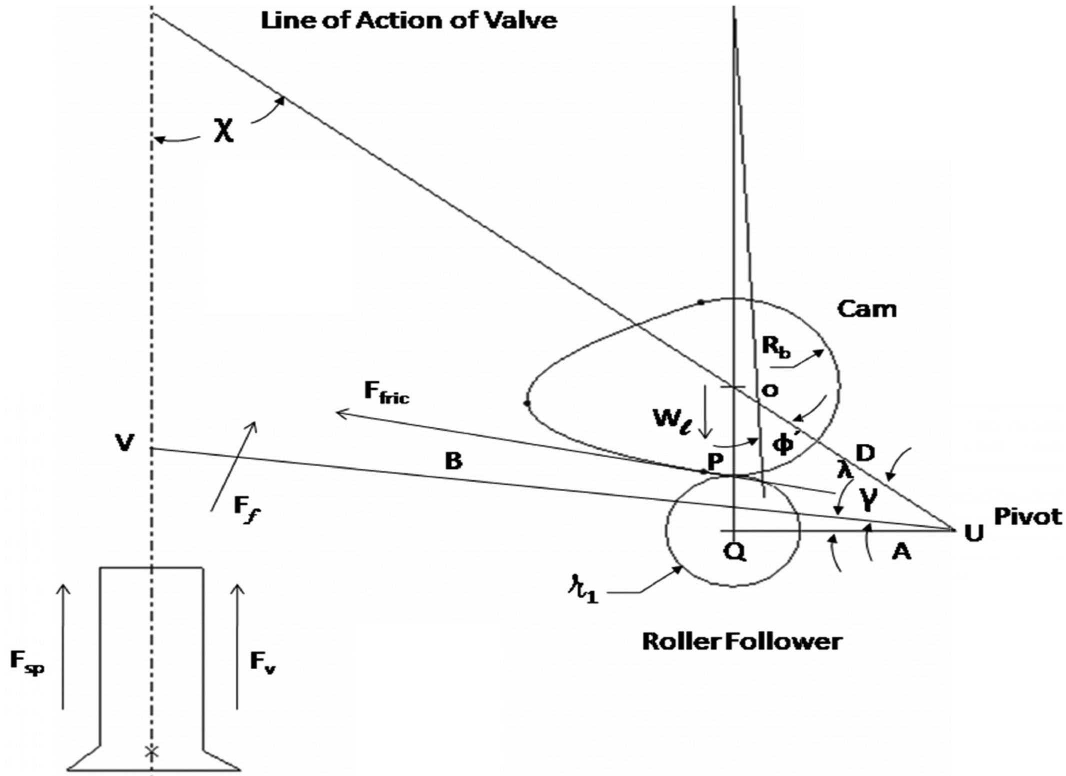

Fig. 2. Basic geometry and forces acting in end pivoted roller finger follower valve train.

cam/roller contact. These velocities are calculated as

$$V_s = V_c - V_f, \tag{1}$$

$$V_e = (V_c + V_f)/2. \tag{2}$$

2.2. Dynamic Analysis

The normal loading at cam/roller interface can be determined by considering the forces acting in the system. These forces have been shown in Fig. 2 which includes the inertial force of valve F_v, roller finger follower inertial force F_f, friction force F_{fric} and the spring force F_{sp}. Dynamic deflections and damping may be neglected by assuming the valve train as rigid. The normal loading at cam/roller interface is slightly affected by the friction force and can be neglected. Considering the above mentioned forces and their moments about the pivot point U by taking clock wise moments as negative, the normal loading can be calculated as under

$$W_l = [M\omega^2 d^2 l_v/d\phi'^2 + k(l_v+\delta)]B\cos(\pi/2-\gamma-\chi) + M_f[B^2\omega^2 d^2\gamma/d\phi'^2]/A\sin(\pi-\gamma-\lambda-\chi) \tag{3}$$

2.3. Hertzian Stress

After determination of the contact load at cam/roller interface, the contact stresses and the dimensions of the contact area are calculated using the theory of elastic contact established by Hertz [10] which will be required in the prediction of sliding friction. The maximum Hertzian stress $P_{\max}$ for the non conformal line contact at the cam/roller interface can be expressed as

$$P_{\max} = 2W_l/\pi bL, \tag{4}$$

where b is the contact half width and is given by

$$b = \sqrt{8W_l R/\pi L E'}, \tag{5}$$

where the equivalent radius of curvature R and equivalent elastic modulus E' is given as

$$1/R = 1/r_c + 1/r_1, \tag{6}$$

$$2/E' = (1-v_1^2)/E_1 + (1-v_2^2)/E_2. \tag{7}$$

The pressure distribution in the contact zone of cam and roller is computed as

$$p = p_{\max}(1 - x^2/b^2)^{1/2} \tag{8}$$

2.4. Load Carried by the Asperities

In the next step, the load carried by the asperities W_a will be calculated which would be required in the prediction of oil film thickness and the boundary friction. The surface roughness plays an important role in the tribological behavior of cam/roller pair. Oil film thickness especially at low engine speeds is usually less than the surface roughness indicating mixed and even boundary lubrication modes at the cam/roller interface [6, 11]. The asperity interaction model basing on the Gaussian distribution developed by Greenwood and Tripp [12] has been adopted in the present study. The load carried by the asperities at cam/roller contact may be estimated as

$$W_a = \frac{8\sqrt{2}}{15}\pi(\eta\beta\sigma)^2\, E'\sqrt{\frac{\sigma}{\beta}}AF_{5/2}\left(\frac{h}{\sigma}\right). \tag{9}$$

Greenwood and Tripp suggested that the product of asperity density, asperity radius of curvature and composite roughness height $\eta\beta\sigma$ is usually within a range of 0.03–0.05 and the value of 0.04 is taken for this study. The ratio σ/β is in the range of 0.0001–0.01 and the value of 0.001 is opted. Actual area of contact A_a and the apparent area of contact A' which is required to compute the load carried by the asperities can be calculated as under

$$A_a = \pi^2(\eta\beta\sigma)^2 AF_2\left(\frac{h}{\sigma}\right), \tag{10}$$

$$A' = 2bL. \tag{11}$$

The Gaussian distribution function $F_n(H)$ is given as

$$F_n(H) = 1/\sqrt{2\pi}\int_H^\infty (s-H)^n e^{-s^2/2}\, d_s, \tag{12}$$

where H is equal to h/σ.

2.5. Lubrication Analysis

The roller finger follower valve train though enjoys healthy lubrication conditions even then full separation of cam and roller surfaces around the cam lift cycle is not guaranteed due to severe operating conditions. After determination of load at cam/roller contact and load carried by the asperities, the central oil film thickness would be calculated as it is required in determining the friction due to shear of lubricant. The oil film thickness can be anticipated using Dowson and Toyoda [13] formula as follows:

$$\frac{h_{\text{cen}}}{R} = 3.06U^{0.69}\, G^{0.56}\, W'^{-0.10}. \tag{13}$$

The parameters used in Eq. (13) are determined as

$$U = \frac{\eta_0 V_e}{RE'}, \quad G = \alpha E', \quad W' = \frac{W_h}{LRE'},$$

where $W_h = W_l - W_a$

2.6. Frictional Forces

The frictional forces at cam/roller contact are composed of shear of the lubricant (hydrodynamic part) and the asperity contact (boundary part) for contact in mixed lubrication regime.The predicted frictional forces will be utilized to calculate the coefficient of friction, tangential forces and the creep at cam/roller interface. The hydrodynamic friction due to the shear of the lubricant at the cam and roller contact may be divided into sliding and rolling friction. The sliding friction force at cam/roller interface has been calculated by considering that the central oil film thickness separates the two mating surfaces. During operation, the non Newtonian effects of the lubricant may come into play strongly. For this purpose, the rheological approach developed by Evans and Johnson [14] has been adopted in this study. Thus the shear stress (τ) at cam and roller interface is determined by using the following relationship:

$$\tau = \begin{cases} \frac{\eta V_s}{h_{\text{cen}}} & (\tau \leq \tau_0) \\ \tau_0 + \gamma' p & (\tau_0 < \tau < \tau_L) \\ \tau_L & (\tau \geq \tau_L) \end{cases} \tag{14}$$

Under moderate contact pressure, Baraus equation can be used to determine the lubricant viscosity (η) as under

$$\eta_p = \eta_o \exp^{\alpha p}. \tag{15}$$

However, the lubricant viscosity changes exponentially with temperature and pressure. Considering the combined effect of temperature and pressure, the lubricant viscosity can be obtained by the following expression as:

$$\eta_{T,p} = \eta_o \exp^{[\alpha TP - \beta(T - T_o)]}. \tag{16}$$

The sliding friction force can be calculated by using the relationship mentioned in equation (14) and substituting the lubricant viscosity as mentioned in equation (16) and integrating it over the whole Hertzian region as

$$F_s = \int_{-b}^{+b} \frac{\eta_o V_s \exp^{[\alpha TP - \beta(T - T_o)]}}{h_{\text{cen}}} (Ldx). \tag{17}$$

Substituting the pressure distribution from equation (8) into equation (17) will result in

$$F_s = \frac{\eta_o V_s L \exp^{[\alpha T P_{\max} - \beta(T - T_o)]}}{h_{\text{cen}}} \int_{-b}^{+b} e^{(1 - x^2/b^2)^{1/2}}\, dx. \tag{18}$$

The rolling friction contributes significantly in the overall frictional force present at cam/roller interface since the sliding friction reduces considerably due to small sliding area and speed. The rolling friction force is calculated [15] as

$$F_r = 4.318(GU)^{0.658}\, W^{0.0126}\, R/\alpha. \tag{19}$$

The friction force due to asperity contact F_a can be computed as

$$F_a = \mu_b W_a. \tag{20}$$

The summation of friction forces due to the shear of lubricant and asperity interactions gives the total friction forces acting at the cam/roller contact and can be calculated as follows:

$$F_{\text{fric}} = F_a + (1 - A_a/A')F_s + F_r. \tag{21}$$

Coefficient of friction μ is obtained by dividing total friction force with the normal load. The total tangential force F acting at cam/roller interface can be determined by considering the combined effect of frictional forces at cam/roller interface, the frictional forces of needle roller bearing and inertial force of the roller.

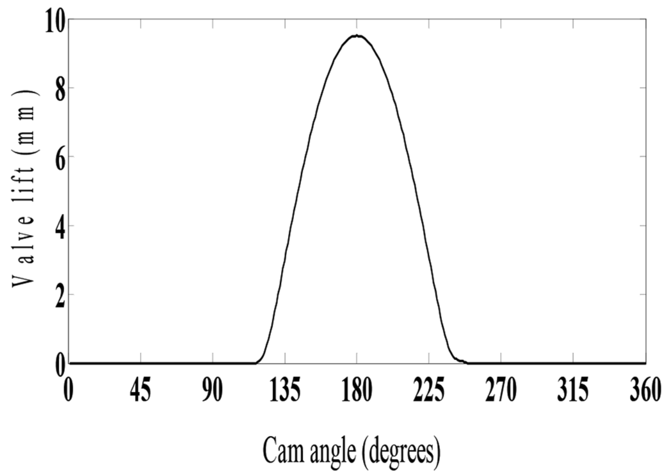

Fig. 3. Valve lift curve vs cam angle.

2.7. Determination of Roller Speed

Many researchers [6, 16] assumed that the slip does not exist between the cam and roller. Such condition of pure rolling indicates that the mating surfaces of cam and roller possess equal surface velocities i.e. $V_c = V_f$ and only stick zone exists at cam/roller junction which simplifies the problem by yielding the sliding velocity as $V_s = 0$ and thus the sliding friction force is eliminated. However, some slippage at cam/roller interface was observed experimentally by Duffy [4] and Bair and Winer [17] which dictates as $V_c \neq V_f$. The friction force at cam/roller contact, inertia of roller and friction force of needle roller bearing are the main forces which will govern the rotation of the roller. The torque balance around the center of the roller bearing results in

$$T_{\text{fric}} = T_{\text{brg}} + J \times \dot{\Omega}. \tag{22}$$

Equation (22) can be used to determine the roller rotational speed which will be utilized to determine the velocity of contact point relative to roller surface.

2.8. Creep

The mating surfaces of cam and roller with friction will transmit tangential loading to each other and will result in dissimilar elastic deformation of surfaces at the interface. After determining the required tribological parameters as computed above, the instantaneous creep at cam/roller contact can be calculated using the Carter relation [18, 19] as follows:

$$\chi = \mu b \left(\frac{1}{r_c} + \frac{1}{r_1} \right) \left(1 - (1 - F/\mu W_l)^{1/2} \right) \tag{23}$$

3. RESULTS AND DISCUSSION

A computer program in MATLAB was developed for this research study. The required input parameters presented in Table 1 and the valve lift data shown in Fig. 3 have been utilized in the model to determine the contact loading, oil film thickness, tangential forces and the creep at cam/roller contact.

Figure 4 shows the variations in normal loading at cam/roller interface on 300, 1200 and 2000 rpm. At lower cam speeds, the normal loading is relatively less and is mostly affected by the spring stiffness. However,at higher cam rotational frequencies, the normal loading increases substantially and is mainly governed by the inertia of various reciprocating masses of valve train components.

Table 1. Input data of the valve train for creep analysis.

A	20.80 mm
B	35.50 mm
D	31.50 mm
λ	–3.5 deg
χ	37.5 deg
Radius of cam base circle	18 mm
Roller outer radius	8.49 mm
Roller width	10.7 mm
Roller moment of inertia	6.385×10^{-7} kg-m^2
Coefficient of friction of needle roller bearing	0.0015
Elastic modulus of cam	170 GPa
Elastic modulus of roller	204 GPa
Poisson ratio of cam material	0.28
Poisson ratio of roller material	0.29
Spring stiffness	25000 N/m
Pre compression of spring	8.4 mm
Mass of roller finger follower	0.0418 kg
Mass of valve, spring and retainers	0.0432 kg
Dynamic viscosity of lubricant	9.865×10^{-3} Pa·s
Pressure-viscosity coefficient of lubricant	0.2×10^{-7} m^2/N
Temperature-viscosity coefficient of lubricant	0.03 1/°C
Limiting shear stress-pressure relation	0.17
Boundary coefficient of friction	0.2
Eyring stress of the lubricant	8×10^6 Pa
Operating temperature	60°C
Rate of change of shear stress with pressure	0.08

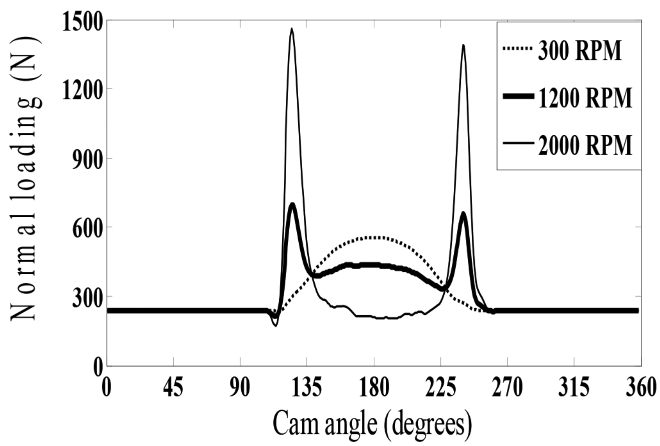

Fig. 4. Normal loading variations at cam/roller contact at 300, 1200 and 2000 rpm.

The variation of oil film thickness at cam/roller contact at 300, 1200 and 2000 rpm has been shown in Fig. 5. At low speeds, the lubricant entrainment velocity is small resulting into thin oil film especially in the cam nose area. At this stage, the asperity interactions may come into play strongly leading to the boundary lubrication and may increase the frictional forces accordingly. However, at higher cam rotational speeds, healthier oil film may be available at the cam/roller junction due to high lubricant entrainment velocity.

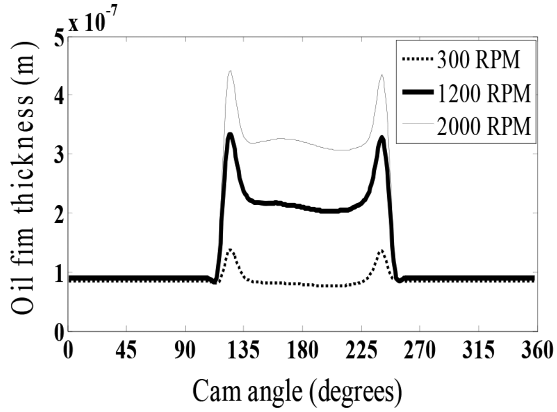

Fig. 5. Variations of oil film thickness at cam/roller contact at 300, 1200 and 2000 rpm.

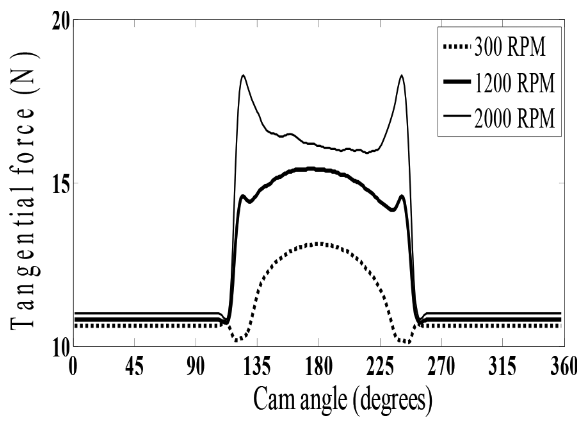

Fig. 6. Tangential force variations at cam/roller contact at 300, 1200 and 2000 rpm.

Figure 6 shows the variations of tangential forces at cam/roller contact at 300, 1200 and 2000 rpm. At low camshaft operating speed of 300 rpm, the value of tangential force is relatively less and mainly exists in the cam nose area. The boundary lubrication is anticipated due to the low lubricant entrainment velocity. The friction force would have relatively larger value around the cam nose area due to dominance of boundary friction depending on the asperity interactions. At this stage, the contributions of frictional forces in the needle roller bearing and roller inertia force would be relatively less in the overall tangential forces at cam/roller interface. However, at 1200 and 2000 rpm, the tangential forces at cam/roller interface increases substantially. Increased oil film thickness may be present at cam/roller contact due to high lubricant entrainment velocity. There would be a continuous pumping of oil at cam/roller contact which may be swept into the pressurized contact zone. The shear stress will be produced due to shearing of oil resulting into increased rolling friction force resisting the rotation of cam and roller. In addition, increased surface velocities of cam and roller may produce relatively high sliding friction due to shearing of oil. At this stage, the tendency of the roller slippage at cam surface may also be increased since the operating speeds are high and roller inertia would also play an important role. At such high speed, friction forces in needle roller bearing and roller inertia force can also play an important role in the increase of tangential force at cam/roller contact in the valve train.

In the engine valve train, the acceleration of follower is negative around the cam nose area. As the camshaft speed is increased,the acceleration of the follower becomes more negative resulting into further

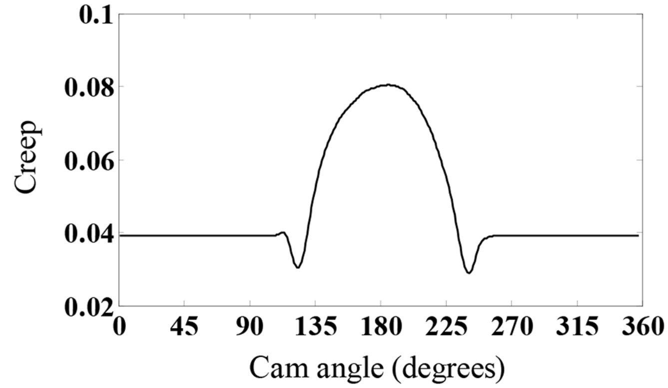

Fig. 7. Creep variation at cam/roller contact at 300 rpm.

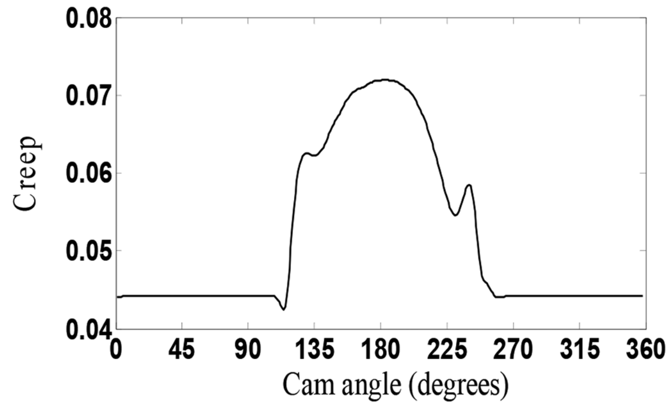

Fig. 8. Creep variation at cam/roller contact at 1200 rpm.

reduction of loading in the cam nose area [20, 21]. This trend of decrease in loading in the cam nose area with an increase in engine speed will reduce the creep at cam/roller interface accordingly in the cam nose area.

The slipping behavior due to the creep can play an important role in power losses and lubrication conditions at cam/roller junction in the end pivoted roller finger follower valve train system.The creep has been calculated at 300, 1200 and 2000 rpm. At low camshaft speed of 300 rpm, the creep proves to be low and exists mainly in the cam nose area as shown in Fig. 7. At this stage the tangential force is relatively less and would be governed by the asperity interactions due to boundary lubrication. Gross slippage at cam/roller contact is unlikely as the traction force is strongly supported by the metallic contact. The overall behavior of shear traction and pressure will be altered in the mating region of cam and roller due to presence of asperities resulting into relatively high localized pressures and may increase the tendency of subsurface stresses. At 1200 rpm, the creep would be present both in the cam nose as well as flank areas as shown in Fig. 8. At this stage, the contact loading under the influence of inertia of various reciprocating masses of the valve train would also be high whereas the greater oil film may exist at cam/roller junction due to increased lubricant entrainment velocity. The tangential loading would also be relatively high. However, the overall creep value remains low at this speed also.

The variation of creep at cam/roller junction at 2000 rpm is shown in Fig. 9. High tangential loadings at cam/roller interface may be leading to dissimilar elastic deformation of mating surfaces causing difference in

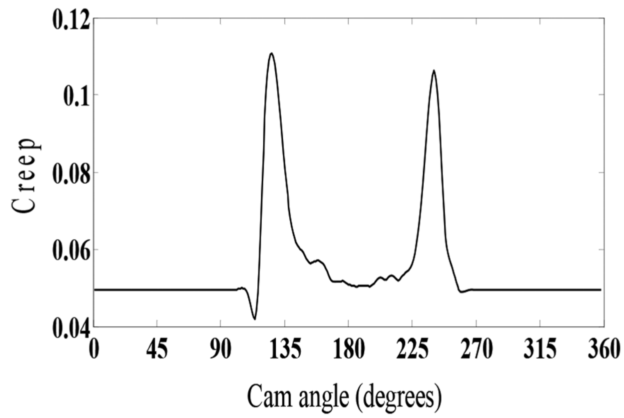

Fig. 9. Creep variation at cam/roller contact at 2000 rpm.

velocity and thus resulting into high value of creep. At such high speed, the slippage behavior of cam/roller mechanism would increase under the influence of raised creep which may result into high valve train power losses and can hamper the lubrication conditions at the cam/roller interface in end pivoted roller finger follower valve train.

4. CONCLUSIONS

1. A comprehensive numerical approach basing on the friction and lubrication analysis for an end pivoted roller finger follower valve train system has been developed incorporating surface roughness, non Newtonian effects of lubricant and the viscosity variations considering both the temperature and pressure to determine the instantaneous creep at cam/roller interface with greater accuracy.

2. The developed model provides an opportunity to investigate the effects of various engine operating conditions and lubricant rheology in detail on the occurrence of creep at cam/roller contact in the engine valve train.

3. The required kinematic parameters, normal loading, Hertzian stresses, oil film thickness, frictional forces, tangential forces and the instantaneous creep at cam/roller interface can be determined at various engine operating speeds precisely by using the developed model.

4. The camshaft operating speed has a direct impact on the contact loading, oil film thickness, friction forces, roller inertia force, needle roller bearing friction force, tangential forces and the creep at cam/roller interface.

5. At low camshaft speeds, the contact loading is relatively less and is mainly governed by the valve spring force but increases substantially under high operating speeds once the inertial loads of various reciprocating masses of the valve train come into play.

6. The greater oil film may be present at cam/roller junction at higher engine speeds due to high lubricant entrainment velocity whereas boundary lubrication is anticipated under low engine speeds.

7. At low engine speeds, the relatively low tangential force occurs mainly in the cam nose area and is governed by the asperity interactions. However, at higher operating speeds, the tangential force increases substantially.

8. The amount of creep proves to be less at low engine speed and occurs mainly in the cam nose area whereas at relatively high speeds, creep exists in the cam nose as well as in the flank areas.

9. The creep increases substantially at very high engine operating speeds especially in the cam flank areas under the influence of higher loading and may affect the valve train tribological performance significantly.

10. As the engine speed increases, the acceleration of roller finger follower becomes more negative in the cam nose area resulting into a reduction of loading at cam/roller contact causing subsequent drop of the creep in cam nose area.

11. The dynamic deflections and damping were neglected by assuming the valve train as rigid while predicting the contact loading at cam/roller interface. The same may be taken into account as a part of future work to further improve the model.

12. The developed model may prove to be a very useful tool in the identification and control of various engine operating conditions promoting the creep at cam/roller interface in the roller follower valve train system.

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