WAYNE STATE UNIVERSITY
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FRICITION IN MULTICYLINDER ENGINES

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Goal

Develop a friction model of a multicylinder diesel engine

Motivation

• The development of an engine simulation model requires reliable sub-models for the different processes determining engine operation.

• The accuracy of the friction sub-model has a significant importance in predicting transient behavior of the engine and the corresponding fuel consumption.

• The development of a reliable friction sub-model shall consider the separate contribution of the main friction components.
Requirements

• The friction sub-model shall be simple enough in order to avoid a large amount of calculation while assuring adequate accuracy,

• The friction sub-model shall be based on physical phenomena governing dynamics and friction of the considered components.

• The friction sub-model shall consider constructive and operating parameters of the components and reflect the influence of these parameters on the mechanical losses.
DEVELOPMENT OF THE MODEL

PISTON RING

Assumptions:
- Ring profile is defined by a second order polynomial
- Hydrodynamic lubrication (Reynolds equation applies)
- The piston ring is fully flooded
- Cavitation is not considered
- Half Sommerfeld condition is assumed, the load carrying capacity being ensured only by the leading edge of the ring profile (with respect to the direction of motion).

One-dimensional Reynolds equation

\[
\frac{d}{dx} \left( h^3 \frac{dp}{dx} \right) = 12 \eta \left( -\frac{\nu_p}{2} \frac{dh}{dx} + \frac{dh}{dt} \right)
\]

\[
\frac{dp}{dx} = 12 \eta \left( -\frac{\nu_p}{2} \frac{1}{h^2} + \frac{x}{h^3} \frac{dh}{dt} + \frac{1}{h^3} C_1 \right)
\]

\[
P = 12 \eta \left( -\frac{\nu_p}{2} \frac{1}{h^2} \int \frac{dx}{h^2} + \frac{dh}{dt} \int \frac{x}{h^3} dx + C_1 \int \frac{dx}{h^3} + C_2 \right)
\]

Profile of the oil film:

\[
h = h_0 + h_x = h_0 + c \left( \frac{x}{a} \right)^2 = h_0 + \frac{c}{a^2} x^2 = h_0 + A x^2 = h_0 \left( 1 + \frac{A}{h_0} x^2 \right)
\]

\[
I_0 = \int \frac{dx}{h} = \int \frac{dx}{h_0 + A x^2} = h_0 \int \frac{dx}{1 + (A/h_0) x^2} = \frac{1}{\sqrt{Ah_0}} arctg \sqrt{\frac{A}{h_0}} x \quad [-]
\]

\[
I_1 = \int \frac{dx}{h^2} = \frac{1}{2h_0} \left( \frac{x}{h_0 + A x^2} + I_0 \right) \quad [1/mm]
\]
\[ I_2 = \int \frac{dx}{h^3} = \frac{1}{4h_0} \left( \frac{x}{(h_0 + Ax^2)^2} + 3I_2 \right) \quad [1/mm^2] \]
\[ I_3 = \int \frac{-x}{h^3} dx = -\frac{1}{4A} \frac{1}{(h_0 + Ax^2)^2} \quad [1/mm] \]

The average pressure acting on the piston ring profile:

\[ p_m = \frac{1}{l} \int p dx = \frac{12\eta}{l} \left( \frac{-v_p}{2} \int \frac{dx}{h^2} dx + \frac{\partial h}{\partial t} \int \frac{xdx}{h^3} dx + C_1 \int \frac{dx}{h^3} dx + C_2 \int dx \right) \]

\[ I_{11} = \int \int \frac{dx}{h^3} dx = \frac{x}{2h_0 \sqrt{\frac{A}{h_0}}} \arctg \sqrt{\frac{A}{h_0}x} \quad [-] \]
\[ I_{22} = \int \int \frac{dx}{h^3} dx = -\frac{1}{4h_0} \left( \frac{1}{2A(h + Ax^2)} - 3I_{11} \right) \quad [1/mm] \]
\[ I_{33} = \int \int \frac{xdx}{h^3} dx = -\frac{1}{4A} \left( \frac{x}{2h_0(h_0 + Ax^2)} + \frac{1}{2h_0} I_0 \right) \quad [-] \]

Boundary conditions:

Up-stroke: \( x = 0 \implies p = 0 \)
\( x = -a_u \implies p = p_1 \)
\[ C_2 = -\frac{\partial h}{\partial t} I_3(0) \quad [1/s] \]
\[ C_1 = \frac{1}{I_2(-a_u)} \left[ \frac{-v_p}{2} I_1(-a_u) + \frac{\partial h}{\partial t} (I_3(0) - I_3(-a_u)) + \frac{p_1}{12\eta} \right] \quad [mm^2/s] \]

Equilibrium condition:

\[ p_r \left( 1 - 2 \frac{a_r}{D} \right) + \frac{T_r}{\pi D} = \frac{1}{l} \int_0^a p dx \]

Down-stroke: \( x = 0 \implies p = 0 \)
\( x = a_d \implies p = p_2 \)
\[ C_2 = -\frac{\partial h}{\partial t} I_3(0) \quad [1/s] \]
\[ C_1 = \frac{1}{I_2(a_d)} \left[ \frac{-v_p}{2} I_1(a_d) + \frac{\partial h}{\partial t} (I_3(0) - I_3(a_d)) + \frac{p_2}{12\eta} \right] \quad [mm^2/s] \]

Equilibrium condition:

\[ p_r \left( 1 - 2 \frac{a_r}{D} \right) + \frac{T_r}{\pi D} = \int_0^{a_d} p dx \]
CONSTANT LOAD: Equivalent pressure 0.15 Mpa
Oil viscosity: $5.0 \times 10^{-4}$ Mpa.s  Speed: 1200 rpm.

First Ring: Symmetric profile, Convexity (c/a): 0.1
Thickness: 2 mm
FIRST RING:
Symmetric Profile
Convexity (c/a): 0.04
Thickness: 2 mm
Tension: 40 N

Oil Viscosity:
Linear variation \((5.0+9.5) \times 10^{-8}\) MPa.s

Mean Indicated Pressure:
\(p_i = 0.83\) Mpa

Engine speed:
\(n = 1200\) rpm.
Friction Coef. v. Duty Parameter
(Ring Thickness = 1.5 mm)
The Strubeck Diagram

\[ \ln(f) = \ln(B) + m \ln(S) \]
\[ f = \exp(B) S^m \quad \text{for } S < S_{cr} \]
\[ \ln(f) = \ln(f_0) - \frac{\ln(f_0 - f_{cr})}{\ln(S_{cr})} \ln(S) \quad \text{for } S < S_{cr} \]
\[ S_{cr} \approx 1 \times 10^{-4} \]

Friction of the piston skirt

Assumptions: - the piston skirt behaves similar to a piston ring;
- secondary motion of the piston is neglected;
- the lubrication regime is always hydrodynamic;
- an average Strubeck correlation is considered.

\[ f = C \sqrt{\frac{\eta |V_p|}{F_N / L}} \]
\[ C \quad \text{a constant dependent on the piston skirt} \]
\[ F_N \quad \text{piston side thrust} \]
\[ L \quad \text{length of the piston skirt} \]
\[ V_p \quad \text{piston velocity} \]
\[ \eta \quad \text{oil viscosity} \]

The friction force:
\[ F_f = f F_N = C \sqrt{\eta |V_p| F_N L} \]
m and B vs. c/a ratio
(Ring Thickness=3.0mm)
Friction force of the piston assembly

Friction torque of the piston assembly (single cylinder engine)

Friction force of the piston assembly

Friction torque of the piston assembly (four cylinder engine)
ENGINE BEARINGS

Line of centers

Reynolds Equation in cylindrical coordinates

\[
\frac{\partial}{\partial \varphi} \left( h^3 \frac{\partial p}{\partial \varphi} \right) + r^2 \frac{\partial}{\partial \psi} \left( h^3 \frac{\partial p}{\partial \psi} \right) = 12 \eta r^2 \left[ \dot{e} \cos \varphi + e \left( \dot{\beta} - \frac{\omega_s + \omega_h}{2} \right) \sin \varphi \right]
\]

Short bearing solution (Ocvirk) \( \frac{\partial p}{\partial \varphi} \ll \frac{\partial p}{\partial \psi} \)

\[
\frac{\partial}{\partial \psi} \left( h^3 \frac{\partial p}{\partial \psi} \right) = 12 \eta \left[ \dot{e} \cos \varphi + e \sin \varphi \left( \dot{\beta} - \frac{\omega_s + \omega_h}{2} \right) \right]
\]

\[
p = \frac{6\eta}{h^3} \left[ \dot{e} \cos \varphi + e \sin \varphi \left( \dot{\beta} - \frac{\omega_s + \omega_h}{2} \right) \right] y^2 + C_1 y + C_2
\]

Boundary conditions:

\[
y = 0 \Rightarrow \frac{\partial p}{\partial \psi} = 0
\]

\[
y = \pm \frac{b}{2} \Rightarrow p = 0
\]

\[
p = \frac{6\eta}{h^3} \left( y^2 - \frac{b^2}{4} \right) \left[ \dot{e} \cos \varphi + e \sin \varphi \left( \dot{\beta} - \frac{\omega_s + \omega_h}{2} \right) \right]
\]

Mobility method (Booker)

\[
\dot{\varepsilon} = \frac{\dot{\varepsilon}}{c} = \frac{F(e/c)^2}{\eta bd} \left[ \frac{(1 - \varepsilon^2)^{3/2} \cos \psi}{2 \pi (1 + 2 \varepsilon^2) (b/d)^2} \right]
\]

\[
\varepsilon \dot{\beta} = \frac{F(e/c)^2}{\eta bd} \left[ \frac{-(1 - \varepsilon^2)^{3/2} \sin \psi}{2 \pi (b/d)^2} \right] + \frac{e(\omega_s + \omega_h)}{2}
\]
Power loss of dynamically loaded bearing (Booker)

\[
P_f = \eta \pi^3 \frac{2 \pi \omega_s^2 - \omega_b^2}{\sqrt{1 - \varepsilon^2}} + \left( \frac{F \omega_s + \omega_b}{2} \right) \sin \varphi + \bar{V} \cos \beta
\]

Friction torque:

\[
M_f = \eta \pi^3 \frac{2 \pi \omega_s^2 - \omega_b^2}{\sqrt{1 - \varepsilon^2}} + \left( \frac{F \omega_s + \omega_b}{2} \right) \sin \varphi + \bar{V} \cos \beta
\]

Constant loaded bearing (Ocvirk)

\[
M_f = r F_f = r \left( \frac{2 \pi \omega_s \omega_b}{c \sqrt{1 - \varepsilon^2}} + \frac{\varphi \varepsilon}{2} F \sin \psi \right)
\]

Quasi-steady-state approximation:

\[
M_f \approx 2 \pi \eta \left( \frac{b}{c} \right) \frac{r \omega_s}{\sqrt{1 - \varepsilon^2}} + \left( \frac{\varepsilon \varphi}{2} \right) \sin \varphi
\]

Loading by inertia forces only:

\[
F = R \omega^2 \sqrt{\left[ m_c \cos(\theta + \beta) + m_p (\cos \theta + \Lambda \cos 2\theta) \right]^2 + \left[ m_c \sin(\theta + \beta) \right]^2}
\]

\[
F \equiv \left[ m_c \omega + m_p (\cos \theta + \Lambda \cos 2\theta) \right] R \omega^2
\]

\[
\Delta = \frac{\eta^2}{F} \left( \frac{b}{c} \right)^2 \left( \frac{b}{d} \right)^2 = \frac{\eta \omega \left( 1 + \frac{\Lambda \omega \theta}{4 F \sin^2 \theta} \right)}{b} \left( \frac{b}{c} \right)^2
\]

\[
\Delta = \frac{(1 - \varepsilon^2)^2}{\pi \varepsilon \sqrt{0.62 \varepsilon^2 + 1}}
\]

\[
F_{\text{max}} = \frac{\pi \omega^2}{4} \left( \rho_{\text{max}} - \rho_{\text{case}} \right) - \left[ m_c + m_p (1 + \Lambda) \right] R \omega^2
\]

TDC

\[
\text{if } \varepsilon < 0.95 \quad M_f = \left[ 2 \pi \eta \left( \frac{b}{c} \right) \frac{r \omega}{\sqrt{1 - \varepsilon^2}} + F_{\text{max}} \left( \frac{\varepsilon \varphi}{2} \right) \sin \psi \right] \sin(6\theta)
\]

\[
0 < \theta < \pi / 6
\]

\[
\text{if } \varepsilon > 0.95 \quad M_f = f F_{\text{max}}
\]
Polar diagram of the con-rod bearing markings every 20 degrees crank angle.

Inertia forces only (n=800-2000 rpm).

Loading of the con-rod bearing inertia forces only (N=800-2000 rpm).

Con-rod bearing load n=2000 rpm.

Con-rod bearing load n=1000 rpm.
FRICTION TORQUE (INERTIA FORCES)
800 rpm

FRICTION TORQUE (INERTIA FORCES)
2000 rpm

TOTAL FRICTION TORQUE
800 rpm, Maximum pressure: 6 Mpa

TOTAL FRICTION TORQUE
2000 rpm, Maximum pressure: 8 Mpa

FRICTION TORQUE (FOUR CYLINDER ENGINE)
800 rpm, Maximum pressure: 6 Mpa

CON-ROD BEARING

FRICTION TORQUE (FOUR CYLINDER ENGINE)
2000 rpm, Maximum pressure: 8 Mpa

CON-ROD BEARING
Figure A
The Valve Train

Figure B
The Cam-Tappet Contact
VALVE TRAIN

Kinematic functions of the valve motion

\[ h_v = h_{\text{max}} + C_2 \left( \frac{\partial}{\partial t} \right)^2 + C_{10} \left( \frac{\partial}{\partial t} \right)^{10} + C_{18} \left( \frac{\partial}{\partial t} \right)^{18} + C_{26} \left( \frac{\partial}{\partial t} \right)^{26} + C_{34} \left( \frac{\partial}{\partial t} \right)^{34} \]

Inlet valve  \( h_{\text{max}} = 12.55 \text{ mm}, \quad 0_t = 280^\circ \)
\( C_2 = 1.5; \quad C_{10} = -0.8203125; \quad C_{18} = 0.3671875; \quad C_{26} = -0.0234375; \quad C_{34} = -0.0234375 \)

\[ V_v = \frac{d h_v}{d(\partial / \partial t)} \quad d(\partial / \partial t) = \frac{\omega}{\partial_t} \frac{d h_v}{d(\partial / \partial t)} \]
\[ a_v = \frac{d V_v}{d(\partial / \partial t)} \quad d(\partial / \partial t) = \left( \frac{\omega}{\partial_t} \right)^2 \frac{d^2 h_v}{d(\partial / \partial t)^2} \]

\[ h_c, V_c; a_c \quad \text{the kinematic functions expressed at the cam profile} \]
\[ \text{according to the mechanism geometry} \]
\[ m_c \quad \text{mass of the valve train reduced at cam profile} \]

\[ F_c = (F_0 + K_s h_c) R \frac{r_1}{r_2} - m_c a_c \quad \text{Force along the tappet axis} \]

\( F_0 \) Spring prestressing force, \( K_s \) Spring constant,
\( r_1 / r_2 \) transformation ratio of the rocker arm

\[ u_1 = (R_b + h_c) \omega_c \quad v_2 = \omega_c e \quad \Rightarrow a_2 = \frac{dv_2}{dt} = \dot{e} \omega_c \]
\[ u_2 = 0 \]

\[ s = u_1 + \dot{e} = (R_b + h_c) \omega_c + \dot{e} \quad \text{Velocity of contact point along the cam surface} \]
\[ R_c = \frac{\dot{s}}{\dot{y}} = \frac{ds}{dt} \frac{dc}{dy} = \frac{ds}{dy} \quad \text{Radius of curvature} \]

\[ R_c = \frac{\dot{s}}{\omega_c} \quad \text{Flat tappet} \]

\[ R_c = R_h + h_c + \frac{\dot{c}}{\omega_c} = R_h + h_c + \frac{a_2}{\omega_c^2} \]

EHT lubrication \quad \[ H = 2.65U^{0.7}G^{0.54}W^{0.13} \quad \text{Non dimensional film thickness} \]

(Dowson and Toyoda)

\[ h = HR_c \quad \text{EHD film thickness} \]

\[ U = \frac{\eta_0 (u_1 + u_2)}{2E'Kc} \quad \text{Non-dimensional velocity} \]

\[ \eta_0 \quad \text{Viscosity at bulk lubricant temperature and ambient pressure} \]

\[ \frac{E'}{E} \quad \text{Composite modulus of elasticity (207 Gpa)} \]

\[ G = \alpha E \quad \text{Material parameter} \]

\[ \alpha \quad \text{pressure-viscosity coefficient of lubricant (2.2 \times 10^{-8} \text{ m}^2 /N)} \]

\[ W = \frac{F_c}{b} \quad \text{Non-dimensional load} \]

\[ b \quad \text{cam width} \]

\[ \sigma_1, \sigma_2 \quad \text{RMS surface roughness of cam and tappet} \]

\[ \sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \quad \text{Composite surface roughness} \]

\[ \lambda = \frac{HR_c}{\sigma} \quad \text{Film thickness parameter} \]

\[ \lambda > 1 \quad \text{EHD lubrication} \]

\[ \lambda = 0 \quad \text{Boundary lubrication} \]

\[ 0 < \lambda < 1 \quad \text{Mixed lubrication} \]

\[ F_f = F_b + F_c \quad \text{Friction force} \]

\[ F_b = f F_c (1-\lambda) \quad \text{for } \lambda < 1 \quad \text{Boundary friction component} \]

\[ F_c = 2b\eta(u_1 + u_2)/h \quad \text{for } \lambda > 1 \quad \text{Viscous friction component} \]

\[ M_f = F_f (R_h + h_c) \quad \text{Frictional torque} \]

\[ M_c = -(F_c + M_f) \quad \text{Total torque on the camshaft} \]
VALVE LIFT
n = 2000 rpm

FORCES ACTING ON THE CAM PROFILE
n = 2000 rpm

FRICITION TORQUE (CAMSHAFT)
n = 2000 rpm

TOTAL TORQUE (CAMSHAFT)
n = 2000 rpm

INTAKE VALVE
TORQUE REQUIRED TO DRIVE THE INLET + EXHAUST VALVES

SINGLE CYLINDER

TORQUE REQUIRED TO DRIVE THE CAMSHAFT

FOUR CYLINDER ENGINE
Conclusions

• Friction models for the most important components of engine friction: piston assembly, bearings and valve train have been developed.

• The models are based on physical phenomena governing lubrication in each particular case.

• The models consider geometry and dimensions of the rubbing parts and the influence of engine dynamics.

• A complete validation of the proposed models requires extensive experimental investigation and the development of new experimental techniques.
Future Work

• Experimental investigation of engine friction on single cylinder and multicylinder engines with a different number of cylinders.

• Development of new experimental techniques for measuring friction forces and friction torque on individual components and the whole engine.

• Improve accuracy of the friction model of each important component of engine friction.