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The effect of cylinder liner operating temperature on frictional loss and engine emissions in piston ring conjunction

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HIGHLIGHTS

- Cylinder liner temperature affects frictional losses.
- Optimum liner temperature improves energy efficiency and reduces emissions.
- Liner temperature is hardly affected by viscous shear of lubricant.
- This implies optimum conditions would be independent of engine speed.

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ABSTRACT

Despite extensive research into alternative methods, the internal combustion engine is expected to remain as the primary source of vehicular propulsion for the foreseeable future. There are still significant opportunities for improving fuel efficiency, thus directly reducing the harmful emissions. Consequently, mitigation of thermal and frictional losses has gradually become a priority. The piston-cylinder system accounts for the major share of all the losses as well as emissions. Therefore, the need for an integrated approach, particularly of a predictive nature is essential. This paper addresses this issue, particularly the role of cylinder liner temperature, which affects both thermal and frictional performance of the piston-cylinder system. The study focuses on the top compression ring whose critical sealing function makes it a major source of frictional power loss and a critical component in guarding against further blow-by of harmful gasses. Such an integrated approach has not hitherto been reported in literature. The study shows that the cylinder liner temperature is critical in mitigating power loss as well as reducing Hydrocarbon (HC) and Nitrogen Oxide (NOx) emissions from the compression ring – cylinder liner conjunction. The results imply the existence of an optimum range for liner working temperature, independent of engine speed (at least in the studied cases) to minimise frictional losses. Combined with the study of NOx and HC emissions, the control of liner temperature can help to mitigate frictional power loss and reduce emissions.

1. Introduction

There are three main elements when optimising the performance of Internal Combustion (IC) engines in terms of improved energy efficiency. Firstly, it is important to reduce the thermal losses, which according to Richardson [1] account for 50–60% of all the losses. Secondly, frictional and pumping losses in load bearing conjunctions such as the piston-cylinder system, valve train and engine bearings represent 15–20% of all the parasitic losses. Nearly 45% of these losses can be attributed to the cylinder system, 30–45% of which is due to the ring-pack [1]. Therefore, the parasitic losses of the cylinder system account for 4–7% of the total fuel energy [1–3]. The main reason for low efficiency of engine powertrain systems is the non-optimal management of the energy conversion chain and subsequent dissipations [4]. Mitigating the sources of energy losses would improve fuel efficiency, which is a key driver in modern engine development. Finally, reducing emissions such as HC, NOx and particulates is also essential because of the environmental health issues, subject to a growing list of mandatory regulations and directives. These three aspects are increasingly and extensively studied.

To reduce the thermal losses, ideally the cylinder wall temperature should be maintained close to that of the in-cylinder gasses.

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Nomenclature

A	contact area
A_c	surface area of combustion chamber
b	ring face (and back) width
D	bore diameter
E	Young's modulus of elasticity
E_f	Released fuel energy
E	energy
E'	composite Young's modulus of elasticity $\left(\frac{1}{E'} = \frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)$
F	force
F_2, $F_{5/2}$	statistical functions
G	ring end gap
h	film profile
h_m	minimum film thickness
h_t	convection heat transfer coefficient
I	second moment of inertia for the ring cross-section
$\hat{i}, \hat{j}$	unit vectors in x and y-directions
L	ring perimeter in the circumferential direction
l	connection rod length
m	(cylinder content) mass
m_2, m_4	the second and fourth spectral moments
$\dot{m}$	lubricant mass flow rate
N	engine speed in rpm
p	hydrodynamic pressure
P	power
Q	heat
$\dot{Q}$	heat transfer rate
R	specific gas constant
r	crank pin radius
S	compression ring face profile
s	engine stroke
T	temperature
t	time
$\underline{u}$	velocity profile
V	velocity vector
V	instantaneous volume of combustion chamber
V_c	clearance volume
V_d	displaced cylinder volume
W	carried load
x,y,z	Cartesian coordinates

Greek symbols

α	pressure-viscosity coefficient
β	thermal coefficient of expansion
γ	pressure relaxation factor
Δ	ring/liner gap due to thermoelasodyncamic deformations
ΔU	sliding speed
δ	local Hertzian/EHL deformation
ζ	asperity peak density
η	lubricant dynamic viscosity
κ	average asperity tip radius of curvature
λ	Stribeck film ratio parameter
ξ	damping coefficient
ρ	lubricant density
σ	composite standard deviation of roughness $(\sigma = \sqrt{\sigma_1^2 + \sigma_2^2})$
ς	coefficient for boundary shear of asperities
τ	shear stress
υ	Poisson ratio
τ_0	Eyring shear stress
φ	crank angle
ω	engine speed

Superscripts

c	composite
n	iteration step

Subscripts

0	ambient (atmospheric) conditions
$1,2$	liner and ring
a	asperity
avg	average
b	boundary
c	cycle
e	elastic
f	fuel
g	gas
i,j	related to each computational node
l	losses
m	minimum
t	total
v	viscous
w	wall

Abbreviations

BDC	Bottom Dead Centre
CFD	Computational Fluid Dynamics
DP	Detonation Point
EGR	Exhaust Gas Recirculation
FMEP	Friction Mean Effective Pressure
HC	Hydrocarbon (emissions)
IC	Internal Combustion
LHV	Lower Heating Value
MOFT	Minimum Oil Film Thickness
NEDC	New European Drive Cycle
NOx	Nitrogen-Oxides (emissions)
PSOR	Point Successive Over-Relaxation
TDC	Top Dead Centre

To a large extent, the cylinder wall temperature also determines the lubricant temperature in the contact of the top compression ring rather than any small rise due to the viscous shear heating of the lubricant in short transit times. This is shown by a recent analytical control volume, thermal mixing model by Morris et al. [5]. However, there are material and other economical cost limitations affecting the maximum attainable temperature of the cylinder liner. Cast iron and aluminium liners can operate with surface temperatures of 200–450 °C [6]. Materials with lower thermal conductivity can operate at higher temperatures. However, usually cost implications impose a practical limit on the use of such expensive liner materials, except for the limited case of race engines. In addition, the heat transfer from the hot cylinder wall to the incoming air-fuel mixture prevents the use of some materials for spark ignition (SI) engines [6]. Furthermore, the above set of ideals cannot be assured under cold start-up conditions or with intermittent stop-start and other transient driving conditions, which are progressively prevalent.

With regard to the emission levels, Wang and Stone [7] showed that the cylinder wall temperature is a significant parameter influencing HC and NOx emissions from IC engines. It was shown that the HC emissions tend to reduce with a higher liner temperature, whilst the contrary is usually true of NOx emissions. Therefore, an optimum liner temperature should be sought, based upon a