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A comparison between Miller and five-stroke cycles for enabling deeply downsized, highly boosted, spark-ignition engines with ultra expansion

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ABSTRACT

It has been well known that the engine downsizing combined with intake boosting is an effective way to improve the fuel conversion efficiency without penalizing the engine torque performance. However, the potential of engine downsizing is not yet fully explored, and the major hurdles include the knocking combustion and the pre-turbine temperature limit, owing to the aggressive intake boosting. Using the engine cycle simulation, this paper compares the effects of the Miller and five stroke cycles on the performance of the deeply downsized and highly boosted SI engine, taking the engine knock and pre-turbine temperature into consideration. In the simulation, the downsizing is implemented by reducing the combustion cylinder number from four to two, while a two stage boosting system is designed for the deeply downsized engine to ensure the wide-open-throttle (WOT) performance comparable to the original four cylinder engine. The Miller cycle is realized by varying the intake valve timing and lift, while the five stroke cycle is enabled with addition of an extra expansion cylinder between the two combustion cylinders. After calibration and validation of the engine cycle simulation models using the experimental data in the original engine, the performances of the deeply downsized engines with both the Miller and five stroke cycles are numerically studied. For the most frequently operated points on the torque-speed map, at low loads the Miller cycle exhibits superior performance over the five-stroke cycle in terms of fuel conversion efficiency, while at higher loads the thermal efficiency of the five stroke cycle, owing primarily to elimination of fuel enrichment operations, is higher than that of the Miller cycle engine. For the WOT operation, even with the two-stage boosting system, at the engine speed below 1700 rpm the deeply downsized engine with the Miller cycle fails to deliver the torque comparable to the original engine, while the targeted WOT torque can be achieved with the five stroke cycle engine at all the engine speeds but 800 rpm. The mechanism of the efficiency differences between the Miller and five stroke cycles is discussed in depth with the energy balance and influencing factor analysis on thermal efficiency.

1. Introduction

A combination of downsizing and intake boosting has been proven to be an effective way to improve the fuel conversion efficiency in spark-ignition (SI) engines without penalties of power or torque output. Lecointe et al. [1] studied the downsizing effects by comparing the performance of a 1.8 L direct injection (DI) SI engine with a turbocharging system with a 3.0 L naturally aspirated SI engine, and they demonstrated a fuel consumption benefit of more than 15% with at least the same acceleration performance. Lake et al. [2] examined the potential of reducing engine swept volume to meet the future tighter requirement of CO_2 reduction, and they proposed a promising concept, termed lean burn direction injection (LBDI), to control octane requirement while maintaining a high compression ratio (CR). Clenci et al. [3] conducted the analysis on the effects of engine downsizing on the fuel economy. In automotive SI engines, the most frequent operations are at partial loads and the load control is usually implemented by using a throttle valve to restrict airflow into cylinders, which results in pumping loss during gas exchange strokes. With keeping the engine torque output constant, a decrease in the engine displacement will lead to a shift in the typical engine operating range to higher loads, which will help reduce the pumping loss and improve the fuel conversion efficiency. To meet vehicle requirements for the maximum power output, intake boosting is generally necessary for downsized engines at high loads, causing knock a more severe problem compared

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Nomenclature

AC	alternating current
A_f	surface area of the flame front (m^2)
ATDC	after top dead center
BDC	bottom dead center
BMEP	brake mean effective pressure (bar)
BSFC	brake specific fuel consumption (g/(kW h))
CAD	crank angle degree
CFD	computational fluid dynamics
CR	compression ratio (–)
D	dimensional
EGR	exhaust gas recirculation
EIVC	early intake valve closing
FMEP	frictional mean effective pressure (bar)
HP	high pressure
IMEP	indicated mean effective pressure (bar)
IVC	intake valve closing (CAD)
k	flow index of turbocharger (–)
KI	knock index (bar)
L_{iv}	intake valve lift (m)
LIVC	late intake valve closing
l_M^*	Taylor micro-scale length (m)
LP	low pressure
m_1	flame kernel growth multiplier
m_2	turbulent flame speed multiplier
m_3	Taylor length scale multiplier
MAPO	maximum amplitude of pressure oscillation (bar)
m_b	burned mass (kg)
MBF	mass burned fraction (%)
MBT	spark advance for maximum brake torque (CAD)
m_e	entrained mass (kg)
$\dot{m}_F$	the mass flow rate of fuel (kg/s)
$\dot{m}_K$	the mass flow rate of compressor (kg/s)
$\dot{m}_T$	the mass flow rate of turbine (kg/s)
NA	naturally aspirated
P	pressure (bar)
PFI	port fuel injection
PMEP	pumping mean effective pressure (bar)
Q	the cumulative apparent heat release (J)
Q_b	the cumulative actual heat released by combustion (J)
Q_c	the heat transfer loss (J)
Q_f	the total chemical energy produced by the fuel combustion (J)
Q_{out}	the heat took away by the exhaust gas during exhaust stroke (J)
Q_u	the energy loss due to incomplete combustion (J)
s	speed parameter of turbocharger (–)
SI	spark ignition
S_L^*	adjustable laminar flame speed (m/s)
T	temperature (K)
t	time (ms)
TDC	top dead center
t_{IVC}	moment at intake valve closing (CAD)
t_{knock}	moment at knock onset (CAD)
$\bar{u}_i$	mean velocity of intake charge through the intake valves (m/s)
u_T^*	turbulent flame speed (m/s)
V_C	the clearance volume (m^3)
V_S	the stroke volume (m^3)
V_θ	the volume at the crank angle θ (m^3)
W_e	the brake work (J)
W_m	the sum of the pumping work and frictional works (J)
WOT	wide open throttle
δ	pressure ratio at constant heat release (–)
ε_c	compression ratio (–)
ε_e	expansion ratio (–)
ϕ	equivalence ratio (–)
ϕ_w	fraction of in-cylinder heat transfer loss (%)
γ	ratio of specific heats (–)
η_b	combustion efficiency (%)
η_e	brake thermal efficiency (%)
η_e	indicated thermal efficiency (%)
η_{glh}	degree of constant volume heat release (%)
η_m	mechanical efficiency (%)
η_{th}	theoretical thermal efficiency (%)
κ	the polytropic index (–)
λ	excess air ratio (–)
θ	crank angle (CAD)
ρ_i	unburned gas densities during the intake process (kg/m^3)
ρ_u	unburned gas densities at the spark moment (kg/m^3)
τ	auto-ignition delay (ms)
τ_b^*	characteristic time (s)

with their naturally aspirated (NA) counterparts. Strategies such as reduction in geometric compression ratio (CR), retardation in spark advance and fuel enriched operation are the most typical solutions to mitigate the knock problem in downsized SI engines at high loads, but a reduced fuel economy benefit from engine downsizing has to be compromised when using these anti-knock strategies [4]. With using numerical simulation, Boretti [5] demonstrated that up to 40% thermal efficiency could be achieved in the highly-boosted SI engine with the brake mean effective pressure (BMEP) exceeding 30 bar, while pure ethanol (E100) was used as the fuel instead of gasoline in the study. From the above literatures, it is evident that while the engine downsizing combined with intake boosting is a promising way to improving the engine efficiency, the degree of the downsizing has been limited, and the possible fuel consumption reduction has not yet been fully achieved.

A strategy of shortening compression stroke relative to expansion stroke or vice versa, termed the ultra-expansion cycle here, is effective to improve the above trade-off between the power requirement and fuel efficiency benefits with downsizing and intake boosting. Since the intake charge can be cooled by the inter or after coolers before entering the cylinders in ultra-expansion cycle engines with external boosting systems, temperature and pressure of the charge at the end of compression stroke will be lower than those of the identical charge density obtained only by the piston compression in Otto cycle engines. This may help mitigate the knock problems due to the highly boosting in downsized engines. The ultra-expansion can be achieved by either Miller (sometimes called Atkinson) or five-stroke cycles. Since the relevant literature is voluminous especially for Miller cycle engines and an extensive review is beyond the scope of this research paper, here only a representative overview of the "state-of-the-art" research is intended for those who are familiar with SI engines but not have a familiarity with the ultra-expansion cycle engines, including either the Miller or five stroke engines.

1.1. Miller cycle engine

The concept of Miller engine can be traced back to Ref. [6] published in 1947, the original design employed a compression control valve on the cylinder head to release part of the charge to the exhaust port during the compression stroke to reduce the effective CR [7]. In modern SI engines, Miller cycle can be realized by either