Energy Conversion and Management 101 (2015) 181–202

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Optimal piston motion for maximum net output work of Daniel cam engines with low heat rejection

Viorel Badescu *

Candida Oancea Institute, Polytechnic University of Bucharest, Spl. Independentei 313, Bucharest 060042, Romania
Romanian Academy, Calea Victoriei 125, Bucharest, Romania

ARTICLE INFO

Article history:
Received 8 December 2014
Accepted 7 May 2015

Keywords:
Compression ignition engine
Daniel cam engine
Low heat rejection
Cooling system
Optimal control

ABSTRACT

Compression ignition engines based on classical tapper-crank systems cannot provide optimal piston motion. Cam engines are more appropriate for this purpose. In this paper the piston motion of a Daniel cam engine is optimized. Piston acceleration is taken as a control. The objective is to maximize the net output work during the compression and power strokes. A major research effort has been allocated in the last two decades for the development of low heat rejection engines. A thermally insulated cylinder is considered and a realistic model taking into account the cooling system is developed. The sinusoidal approximation of piston motion in the classical tapper-crank system overestimates the engine efficiency. The exact description of the piston motion in tapper-crank system is used here as a reference. The radiation process has negligible effects during the optimization. The approach with no constraint on piston acceleration is a reasonable approximation. The net output work is much larger (by 12–13%) for the optimized system than for the classical tapper-crank system, for similar thickness of cylinder walls and thermal insulation. Low heat rejection measures are not of significant importance for optimized cam engines. The optimized cam is smaller for a cylinder without thermal insulation than for an insulated cylinder (by up to 8%, depending on the local polar radius). The auto-ignition moment is not a parameter of significant importance for optimized cam engines. However, for given cylinder wall and insulation materials there is an optimum auto-ignition moment which maximizes the net output work. The optimum auto-ignition moment does not depend on wall and insulation materials, neither on wall and insulation thicknesses. The optimized cam size increases by up to 32% on the local polar radius, when the auto-ignition moment increases from 7 ms to 9 ms.

© 2015 Elsevier Ltd. All rights reserved.

1. Introduction

Compression ignition engines (CIE) are used now in rail, road and sea transportation and for a large number of stationary applications. They involve both turbocharged and naturally aspirated design solutions. Usage of advanced electronic injection controls significantly reduced in the last decades the noise levels and soot and increased the thermal efficiency of CIEs [1,2].

Optimal motion of CIE pistons has been studied in many papers [3–8]. Most authors take into consideration the finite combustion rate and piston friction. Different laws for the heat transfer between the working fluid and the cylinder wall have been considered; most authors adopted the Newton's law [5] but Dulong–Petit law [8], radiative and convective–radiative laws have been also investigated [6,9]. The optimization has been performed by using several objective functions such as maximum efficiency [4], maximum work output [8] and entropy generation minimization [3]. Both unconstrained and constrained optimization has been treated [10]. The constraints include: fixed total cycle time and fuel consumed per cycle [9], fixed compression ratio and fixed power output [7], among others. Generally, the results show that that optimizing the piston motion could improve engine efficiency by nearly 10% [7]. More details may be found in [3–11] and references therein.

In most reciprocating engines the piston is moved by means of a tapper-crank system. Other ways of piston operation are also possible, such as in the free piston engines [12,13] or in the unconventional engine described in [14]. Previous optimization studies [3–8] found that the classical tapper-crank systems cannot provide optimal piston motion. They have only two geometric parameters and provide limited control on piston movement. Cams are more appropriate for this purpose [3,10]. In principle, the cam geometry can be designed to obtain any desired piston motion. Well

* Corresponding author at: Candida Oancea Institute, Polytechnic University of Bucharest, Spl. Independentei 313, Bucharest 060042, Romania. Tel.: +40 21 402 9339; fax: +40 21 318 1019.

E-mail address: badescu@theta.termo.pub.ro

http://dx.doi.org/10.1016/j.enconman.2015.05.020
0196-8904/© 2015 Elsevier Ltd. All rights reserved.

Nomenclature

Latin letters

A	cylinder's wall surface area (m^2)
A	crank angle (°)
A'	crank angle set to zero at top-dead-center ($A' = A - 180°$) (°)
a	empirical coefficient entering Eq. (A7) (dimensionless)
a	piston acceleration (m/s^2)
b	empirical coefficient entering Eq. (A7) (dimensionless)
C	heat capacity (J/(mol K))
c	empirical coefficient entering Eq. (A8) (dimensionless)
D	constant entering Eq. (B4d) (m)
d	cylinder bore (m)
h	heating function (W)
h_c	convection heat transfer coefficient between metal wall and cooling fluid ($W/(m^2 K)$)
k	heat transfer coefficient between charge and cylinder walls ($W/(m^2 K)$)
k_{ins}	thermal conductivity of thermal insulation (W/(m K))
k_w	thermal conductivity of metal wall (W/(m K))
l	tapper length (m)
l_1	coefficient entering Table 2 (fractional units not given)
l_2	coefficient entering Table 2 (fractional units not given)
M	constant entering Eq. (16) (J)
m_1	coefficient entering Table 2 (fractional units not given)
m_2	coefficient entering Table 2 (fractional units not given)
N	mole number (mol)
Q_c	molar heat of the charge (J/mol)
q_c	heat flux transferred from charge towards the cylinder walls (W/m^2)
Re	Reynolds number
R	ideal gas constant (J/(mol K))
R_l	thermal resistance per unit length (m K/W)
r	cylinder inner radius (m)
r	crank length (m)
S	thickness (m)
T	charge temperature (K)
T_{wg}	charge temperature at the inner surface of the cylinder wall (K)
t	time (s)
t_b	combustion duration (s)
t_{tot}	time for compression and power stroke (s)
t_z	auto-ignition moment (s)
U	internal energy (J)
V	cylinder volume (m^3)
v	piston speed (m/s)
$\bar{v}$	average piston speed (m/s)
W	net output work (J)
w	rate of mechanical energy (W)
x	distance between piston and fire deck (m)
y	position (m)
Z	associate variable (J)

Greek letters

α	coefficient entering Eq. (A10) (N s/m)
χ	dimensionless speed
η	thermal efficiency
κ	thermal conductivity of the charge (W/(m K))
μ	viscosity of the charge (kg/(m s))
ϑ	dimensionless charge temperature
ρ	charge density (kg/m^3)
σ	Stefan–Boltzmann constant ($W/(m^2 K^4)$)
τ	dimensionless time
ξ	dimensionless distance
ω	dimensionless acceleration
ξ	reaction coordinate (extent of combustion) (dimensionless)
ζ	dimensionless associated variable defined by Eq. (16)

Subscripts

0	minimum distance, reference
A	Annand
c	convection
f	final, friction
i	initial
ins	thermal insulation
l	per length unit
$lost$	lost
max	maximum
min	minimum
N	Newton
opt	optimum
r	radiation
w	metal wall

Superscripts

A	Annand
N	Newton

Abbreviations

CIE	compression ignition engine
LHR	low heat rejection
NLP	nonlinear problem
OCP	optimal control problem
PSZ	plasma spray zirconia
SN	silicon nitride
SQP	sequential quadratic programming

designed cams may accommodate in practice a much larger class of velocity and acceleration profiles for piston motion than tapper-crank systems can do. In cam engines, the pistons deliver their force to a cam that is then caused to rotate. The output work of the engine is driven by this cam. Advantages of the cam engines are very good dynamic combustion and balance, lower internal friction, cleaner exhaust, less fuel consumption, longer life, and more power per unit mass [15,16]. Hydraulic cam motors, particularly the swashplate form, are widely and successfully used but internal combustion cam engines such as Daniel engine [17], Michel engine [18], Fairchild–Caminez engine [19] or Marchetti engine [20] are almost unknown [16,21]. The Daniel cam engine has the simplest design and its optimal piston motion is studied in this paper.

Charge temperature inside the CIE cylinder is of the order of 2500 K, under normal operation conditions. The temperature of all components in contact with the charge, such as the piston, cylinder head and valves, depend on charge temperature and other parameters such as the heat convection coefficient. Melting points of advanced materials are still far below the combustion temperature. For instance, the melting point of nickel-based super alloys and coated carbon–carbon composites is 1500 K and 1800 K, respectively [22]. Material constraints necessitate active CIE cooling. Notice that overcooling is to be avoided since it causes high

Download English Version:

https://daneshyari.com/en/article/7162468

Download Persian Version:

https://daneshyari.com/article/7162468

Daneshyari.com