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Research Paper

Design considerations for an Ericsson engine equipped with high-performance gas-to-gas compact heat exchanger: A numerical study

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HIGHLIGHTS

- Multi-passing improves gas energy exploitation but increases gas-side pressure drop.
- Multi-passing requires compensation in flow areas to reduce gas-side pressure drop.
- Larger heaters increase heat flow and engine power but decrease thermal efficiency.
- Performance improves at higher loads but gas-side pressure drop increases.

ARTICLE INFO

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Ericsson engine
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Heat exchanger
Heat transfer coefficient
Pressure drop
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ABSTRACT

A new simulation model is presented for the simulation of the open cycle Ericsson engine operation on a crank-angle basis. The model simulates the compressor and expander operation as well as their interactions with a compact gas-to-gas heat exchanger selected from the literature based on its high heat-transfer-area-to-volume ratio. Hot gases passing through the hot side of the exchanger are the heat source that heats the compressed air, which is considered the working gas. The analysis considers both sides of the heat exchanger. Factors such as pressure drops in the air and gas streams, free-flow areas, heat transfer coefficients, heat exchanger effectiveness and heat transfer rate are considered to determine the engine performance. These factors guide the threefold numerical investigation presented herein. The effects of heater geometry and configuration (single- or multi-passing) are first analyzed and elucidate the importance of gas-side pressure drop. Next, the effect of the heat exchanger size is considered that highlights the importance of heat transfer areas and heater pressure fluctuations that affect the work produced. Finally the effect of heat source temperature on the engine performance is determined. All trends observed are analyzed and explained on a crank-angle basis, to elucidate the essential variables that link the overall engine performance to the physical phenomena evolving in the devices.

1. Introduction

The ever-increasing energy demands for power and heat, as well as environmental concerns have lead current research to seek alternative solutions for power generation, improve existing power producing engines, and efficiently exploit wasted heat from power plants and combustion processes [1]. Internal combustion engines have dominated for many years due to their high power density, their simple construction, and the experience gained following years of research and development. However, the need for cleaner and more efficient engines places stringent demands on the engine design and manufacturing process. Often these demands lead to the introduction of additional control systems or to the refinement of existing ones, and increase the manufacturing and service costs. Furthermore, recent research has shown that even with current technological advancements the emissions from an appreciable number of heavy- and light-duty diesel engine vehicles exceed certification limits [2].

In view of the energy and environmental concerns, efforts have been made to introduce new concepts for power production. Towards this goal, small scale or micro combined heat and power (micro-CHP) units have been proposed for domestic use [1,3–8], referring to power production of the order of 10 kW. A number of commercial CHP units exist and a comparative experimental study has been performed comparing the performance of natural-gas-fueled Stirling and internal combustion engines for domestic small-scale production of heat and power [3].

Stirling engines belong to the broader classification of external

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Nomenclature

A_f	fin area (m^2)
A_{ff}	free flow area (m^2)
A_{fr}	frontal area (m^2)
A_{ref}	reference area for estimation of mass flow through valves (m^2)
A_t	total heat transfer area (m^2)
C_a	air-side heat capacity rate (W/K)
C_d	discharge coefficient (–)
C_g	gas-side heat capacity rate (W/K)
$C_{\max}$	maximum heat capacity rate (W/K)
$C_{\min}$	minimum heat capacity rate (W/K)
c_p	specific heat (J/(kg K))
C_r	$C_{\min}/C_{\max}$ (–)
D_h	hydraulic diameter (m)
h	heat transfer coefficient (W/(m^2 K)) or specific enthalpy (J/kg)
L_a	total air-side flow path length (m)
l_a	unit block air-side flow path length (m)
l_g	unit block gas-side flow path length (m)
L_v	valve lift (m)
$\dot{m}_v$	mass flow rate through valves (kg/s)
$\dot{m}_a^*$	air-side characteristic mass flow rate (kg/s)
$\dot{m}_g$	gas-side mass flow rate (kg/s)
n_e	engine speed (1/s)
N_b	number of heat exchanger unit blocks in compact form (–)
N_p	number of passes in multi-pass heat exchanger arrangement (–)
Q	heat (J)
$\dot{Q}$	heat rate (W)
$\bar{\dot{Q}}_h$	average heat input rate to the heater (W)
p	pressure (Pa)
R	universal gas constant 8314 (J/(kmol·K))
t	time (s)
T	temperature (K)
U	internal energy (J) or overall heat transfer coefficient (W/(m^2 K))
UA_a	air-side component of UA
UA_g	gas-side component of UA
V	volume (m^3)
v	specific volume (m^3/kg)
W	work (J)
w_{he}	heat exchanger unit block width (m)
W_{he}	heat exchanger width (m)

Greek

γ	specific heat ratio (–)
$\delta\theta$	open valve duration (degrees CA)
ε	effectiveness of multi-pass heat exchanger (–)
ε_1	effectiveness of single-pass heat exchanger (–)
η_f	fin efficiency (–)
η_o	overall surface efficiency (–)
$\eta_{th,i}$	indicated thermal efficiency (–)
θ	crank angle (degrees after TDC)
θ_o	crank angle at valve opening (degrees after TDC)
λ	connecting rod length to crank radius ratio (–)
μ	dynamic viscosity (kg/m s^2)
σ	free flow area to frontal area ratio (–)

Subscripts

a	air
c	compressor
cl	clearance (volume)
d	downstream
e	expander
h	heater
he	heat exchanger
i	indicated or inlet
g	gas
m	mean
max	maximum
min	minimum
sw	swept (volume)
u	upstream
x	indicates air "a" or gas "g", compressor "c" or expander "e"

Dimensionless numbers

f	friction factor
j	Colburn factor
Pr	Prandtl number
Re	Reynolds number
St	Stanton number

Abbreviations

aTDC	after top dead center
BDC	bottom dead center
C	compressor
CA	crank angle
CHP	combined heat and power
CR	compression ratio
E	expander
EHVE	externally heated valve engine
EOI	exhaust valve open interval
EVC	exhaust valve closing timing
EVO	exhaust valve opening timing
H	heater
IOI	inlet valve open interval
IVC	inlet valve closing timing
IVO	inlet valve opening timing
MP	multipass
NTU	number of transfer units
PPF19.86	plain plate-fin surface 19.86
SP	single pass
TDC	top dead center

combustion engines, which provide heat source flexibility. Thus, various fuels may be used, if combustion is selected to provide heat, or renewable sources may be incorporated, such as solar energy [9]. Another external combustion engine, which has not been extensively studied, is the Ericsson engine operating on air. The air-standard thermodynamic Ericsson cycle comprises of four processes: isothermal compression, isobaric heat addition, isothermal expansion, and isobaric cooling. In practice, the Ericsson engine resembles an open- or closed-circuit gas turbine, with the compressor and turbine replaced by their reciprocating counterparts. It is probably for this reason that various terms have been given to these configurations: for the closed circuit the term "externally heated valve engine" (EHVE) has been used [10]; the terms "volumetric hot-air Joule engine" [11,12], "open Joule cycle Ericsson engine" [13], "open Joule cycle reciprocating Ericsson engine" [14], and "reciprocating Joule cycle engine" [15] have been used to describe the open circuit configuration. In its simplest form the open