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Low temperature Stirling engines pressurised with real gas effects

Fernando Sala*, Costante M. Invernizzi

Department of Mechanical and Industrial Engineering, University of Brescia, Via Branze 38, 25123 Brescia, Italy

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ABSTRACT

With the aim of improving the performance of Stirling engines for applications with relatively low maximum temperatures (150 °C–300 °C), working fluids that have noticeable real gas effects have been considered.

The simulation model developed for the preliminary analysis of Stirling engines takes into account the losses of mechanical power and the efficiency of the regenerator with two adimensional coefficients, on the basis of experimental data available in literature. Comparisons between the working fluids were made having fixed the value of these two parameters. The results from this simplified analysis can be taken into consideration when evaluating the potential of these Stirling engines with real gas. They revealed that an engine cycle with a maximum pressure 3–4 times greater than the critical pressure and using a fluid with a critical temperature close to the minimum temperature of the engine cycle, gives an increase in power about 2.5 times greater than the case with ideal gas and an increase in cycle efficiency of around 50 percent. Such a result means that the use of fluid mixtures as the working fluids now becomes interesting, since the critical point of mixtures can be gradually adapted to meet design requirements.

1. Introduction

To date, it is still fossil fuels that are most commonly used in the production of thermal energy and electricity. Nevertheless, recent decades have seen a growing interest in high efficiency energy conversion systems that aim to make better use of renewable energy resources. Within this context, the Stirling cycle offers interesting possibilities.

These engines have been proposed for CHP (combined heat and power) [1] and micro-CHP applications [2], for small-scale energy production from a few hundred watts [3] up to several kW [4] and for exploiting low-temperature heat sources [5] (from 110 °C up to 200 °C). Furthermore, Stirling engines are often considered as alternatives to internal combustion engines in the automotive sector [6].

Traditionally, Stirling engines have been pressurised by fluids that, under normal working conditions, have a similar behaviour to ideal gas. The most commonly used working fluids are air, nitrogen, helium and hydrogen. However, engines that are pressurised with these working fluids have two great drawbacks. The most important of these, a low specific power compared to internal combustion engines.

For example, the engine described in Ref. [7] has a power of about 35 kW at 1040 rpm with a work volume of 4814.38 cm^3 and a maximum cycle temperature of around 700 °C. Another example is the Ford-Philips 4-215 engine (built at the end of the 1980s, with a total internal volume of 3500 cm^3) [8,9], which has a useful output of 200 kW at 4500 rpm at the maximum temperature of 750 °C, with hydrogen as the working fluid and an average pressure of 200 bar.

Secondly, in order to get a good performance from these engines, the working fluid needs to be raised to very high pressure and temperature levels. Such working conditions means using materials that are highly resistant to stress and very costly.

So far, to improve the Stirling engines performance the research was focused on the development of the mechanical transmission system [10] and on the gas circuit performance [11]. In particular looking for driver mechanism to minimise mechanical losses and heat exchanger geometry to reduce pressure losses and improve heat exchange. The choice of working fluids is at present limited to helium, hydrogen and air (nitrogen).

This article analyses the effects caused by the use of working fluids under thermodynamic conditions, which, compared to the usual behaviour of an ideal gas, give rise to significant increases in useful power and in engine efficiency.

* Corresponding author. Tel.: +39 (0)303715947; fax: +39 (0)303702448.
E-mail addresses: fernando.sala@ing.unibs.it, fernandosala@hotmail.it (F. Sala).

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Nomenclature

Symbols

Be	Beale number, $(=\dot{W}/P_{max}V_{sw}f)$
c_p, c_v	mass heat capacity, kJ/kgK
f	frequency, Hz
h	heat transfer coefficient, W/m^2 K
M_M	molar mass, g/mol
M	mass, kg
$\dot{m}$	mass flow, kg/s
n	revolution per minute, rpm
P	pressure, bar
Pr	Prandtl number
Q	thermal energy, J/cycle
$\dot{Q}$	thermal power, kW
R	gas constant, J/kgK
Re	Reynolds number
s	specific entropy, J/kgK
S	entropy, J/K
S_{irr}	entropy produced by irreversible processes
t	time, s
T	temperature, °C
v	flux velocity, m/s
V	volume, m^3
V_{cl}	total clearance volumes, m^3
V_{clc}, V_{cle}	compression and expansion clearance volumes, m^3
V_{sw}	total swept volumes, m^3
V_{swc}, V_{swe}	compression and expansion swept volumes, m^3
W	engine work, J/cycle
$\dot{W}$	mechanical power, kW
α_p	isobaric coefficient of thermal expansion
ε	regenerator efficiency parameter $(\min(\varepsilon_K, \varepsilon_H))$
ϕ	phase angle
η	fluid dynamic coefficient
η_{cy}	cycle efficiency
η_{Carnot}	Carnot cycle efficiency $(=1-T_{min}/T_{max})$
κ	thermal conductivity, W/mK
κ_T	isothermal compressibility coefficient, W/mK
ρ	density, kg/m^3
ω	acentric factor
ω_s	angular velocity, rad/s $(=2\pi N/60)$
μ	dynamic viscosity, Pa• s
ρ	density, kg/m^3
τ	temperature ratio, $(=T_{max}/T_{min})$
$\dot{X}$	time derivative of X, $(=dX/dt)$

Subscripts

cr	critical point of the fluid
max	maximum
min	minimum
r	reduced condition $(T_r = T/T_{cr}, P_r = /P/P_{cr})$
sat	saturation point of the fluid
C	compressor
E	expansor
R	regenerator
H	heater
K	cooler
T	total
D	dead
CK	physical quantity on the interface between Compressor and Cooler
KR	physical quantity on the interface between Cooler and Regenerator
RH	physical quantity on the interface between Regenerator and Heater
HE	physical quantity on the interface between Heater and Expansor

Acronyms

CHP	combined heat and power
EOS	equation of state

J.F. Malone in Ref. [12] was the first to propose using working fluids under conditions that differed from those of an ideal gas in Stirling engines. Malone designed an engine pressurised by water under super-critical conditions. Subsequently, his ideas were studied by other authors [13–15].

The purpose of this work is to improve and expand upon the analysis proposed in Ref. [15], optimising the corrective coefficients of the numeric model on the basis of experimental data drawn from literature and extend the thermodynamic analysis to different working fluids, both pure and mixed.

In Ref. [15] it was also show that an engine cycle with real gas effects can increase both power and efficiency of the Stirling engine. Mixing two substances of different critical temperatures yields an infinite number of fluids with different critical point [16] and enables the exploitation of real gas effects in all operating condition. For this reason their study, in Stirling engine application, has particular interest.

2. The engine model

We took the so-called Adiabatic Model as our model [8]. The model was then modified by applying two corrective coefficients, which were introduced to account for the typical power and heat losses of these engines [15].

In the calculation model, created by using the commercial program aspenONE®, the engine is subdivided into five volumes: the compression volume (Compressor), the cooling volume (Cooler), the regenerative volume (Regenerator), the heating volume (Heater) and the expansion volume (Expander). As shown in Fig. 1 the heat exchangers are included between the two pistons.

The model is based upon the following hypotheses:

- the compression, the expansion and the regenerator volumes are adiabatic

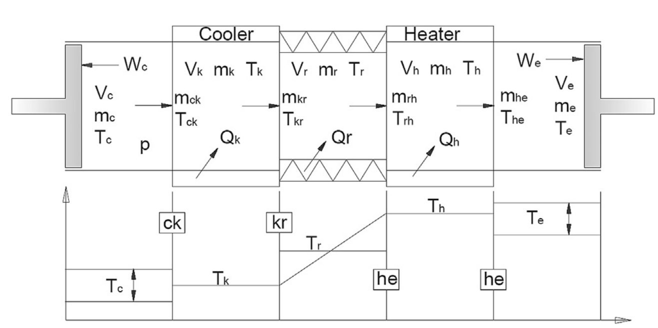

Fig. 1. Schematic Adiabatic Model [8].

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