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The impact of heat exchanger fouling on the optimum operation and maintenance of the Stirling engine

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Abstract

This paper focuses on the effect of heat exchanger fouling on the performance of the Stirling engine in combined heat and power (CHP) application. Fouling results from using biomass fuels and affects the heat exchanger that transfers heat into the engine. This heat exchanger is referred to as the heater. The heat exchanger that recovers heat from the flue gases is also affected by fouling. To determine the performance of the Stirling engine, a commercial Stirling analysis tool is applied together with models that have been developed for the heat transfer in the heater, regenerator and cooler of the engine. The Stirling engine model uses constant temperatures for the heat addition and rejection, with the theory of displacement engine as a basis. The fouling in the heat exchanger is taken into account by using a fouling factor that corresponds with the degradation in the total heat transfer coefficient. The Stirling engine model together with the model for heat exchanger fouling makes it possible to estimate the effect of fouling on the performance of the Stirling engine. A cost model is developed for the engine to translate changes in performance into economy in CHP operation. In the studied application, the Stirling engine is operated by the heat demand. Together with the selected control method, performance and cost models compose a tool for the simulation and optimization of the system. The use of the models to determine the optimal cleaning interval of the heat exchanger surfaces is considered.

Keywords: Stirling engine; Combined heat and power; Fouling; Cost models; Operation and maintenance

1. Introduction

Distributed energy systems are becoming increasingly popular in supplying for local energy needs. One contributing factor for the popularity is the possibility to bring electricity generation close to the end-user. If heat is also required, cogeneration technologies can be implemented to gain high energy conversion efficiencies. The Stirling engine provides one option for combined heat and power (CHP) production especially under the 30 kWe scale. The engine uses external firing, which allows the use of solid fuels unlike with diesel engines, for instance. Compared to gaseous and liquid fuels, solid fuels have high ash contents, which contributes to the increased slagging and fouling of the heat transfer surfaces. The objective of this study is to model the effect of fouling on the performance of the Stirling engine when using solid biomass fuels. In addition, consideration is given for the use of the models to determine the optimal cleaning interval in CHP operation. The work is a part of a larger research project that investigates the applicability of the biofuel-operated Stirling engine for small-scale distributed CHP systems.

2. Stirling engine model

In this paper, simple isothermal models are considered while modelling the α-type Stirling engine. The α-type engine contains one work piston and one displacer piston in separate cylinders. These cylinders are coupled together through three series-connected heat exchangers: the heater,

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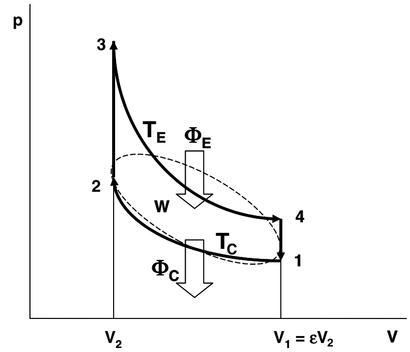

Fig. 1. The ideal Stirling cycle (continuous line) and the real cycle (dashed line) in the Stirling engine.

regenerator, and cooler. In the isothermal model, the heat from the flue gas is transferred to the cycle working gas during the isothermal expansion at a constant working gas temperature and at a mean pressure. An ideal Stirling cycle consists of the following sub processes (Fig. 1) [1]:

- The working gas is compressed to higher pressure (between points 1 and 2) by the work piston. The compression takes place at the constant lower temperature T_C. The cooler power Φ_C thus exits the cycle from the colder space. Heat is rejected and work is supplied.
- The displacer piston displaces the gas from the colder space to the hotter space. The gas is heated to the higher temperature T_E in the regenerator (between 2 and 3). The heating takes place at a constant volume since the work piston is not moving. No external heat is supplied to the system.
- The gas is expanded at the constant temperature T_E and thus the heater power Φ_E is imported to the cycle (between 3 and 4). Isothermal expansion takes place in the hotter space. The work piston moves down. Heat is supplied and work produced.
- The gas is displaced (by displacer) to the colder space and temperature T_C (between 4 and 1). The cooling process takes place in the regenerator at a constant volume. The work piston is not moving. The gas is cooled and heat is stored into the regenerator.

The area defined by the cycle on the pV diagram represents the work done. It is the difference of the supplied and rejected heats. The ratio of the work produced, W, to the heat supplied, Q_E, is defined as thermal efficiency η_{th}. In the ideal Stirling cycle, thermal efficiency depends only on the maximum and minimum temperature of the cycle, Eq. (1).

$$\eta_{th} = \frac{W}{Q_E} = \frac{Q_E - Q_C}{Q_E} = 1 - \frac{T_C}{T_E} \tag{1}$$

The above equation demonstrates the strong thermodynamic potential of the Stirling engine: the thermal efficiency of the ideal Stirling cycles equals to the Carnot efficiency, the highest possible value for a heat engine operating between two temperature levels [2]. A real cycle is marked with the dashed line in Fig. 1.

The classical analysis of the operation of real Stirling engines is that of Schmidt [3]. The theory provides for harmonic motion of the reciprocating elements, but retains the major assumptions of isothermal compression and expansion and perfect regeneration. It thus remains highly idealized, but is certainly more realistic than the ideal Stirling cycle [4]. In a practical engine, aerodynamic flow losses in the heat exchangers cause a decrease in the area of the pV diagram and the net cycle output (indicated work). Analyses of the Schmidt type are of limited value, since the predictions must be modified by a multiplier in the range 0.3–0.4, in order to establish realistic values.

In this study, a commercial Stirling analysis program was adopted. The Snap2002 [5] computes the power output of the given configuration. The basic procedure follows Martini's [6] isothermal procedure. The code takes also into account the friction losses for the heater, regenerator and cooler, and estimates the following losses: mechanical friction loss, reheat loss (heat that must be added due to imperfect regenerator operation), shuttle conduction loss, static conduction losses, as well as pumping and temperature swing losses. The shuttling and pumping losses are hot cap or displacer gap losses. The temperature swing losses are due to the temperature of the regenerator matrix swinging when the gas is blown through it on each cycle.

3. Reference process

A reference process [7] was selected to test the modelling tools and to study the effect of fouling on a biomass Stirling engine. The specifications for the α-type engine (Fig. 2) are presented in Table 1.

The goal was to model the steady state operation of the Stirling engine inside the dashed line in Fig. 2, which is heated by the flue gas of a biomass furnace. The fuel is bark.

In Fig. 2, hot flue gas from the combustion chamber enters the heater of the Stirling engine and heat is transferred into the engine. In the engine, the heat is converted into work and lower temperature-level heat which is rejected in the cooler. The codes for the preliminary design of the heater, cooler and regenerator were written to give the right input values for the reference process configuration

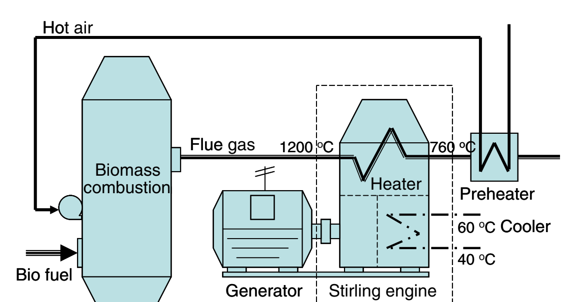

Fig. 2. Configuration of the reference engine.