MATHEMATICAL MODELLING OF A HYBRID MICRO-COGENERATION GROUP BASED ON A FOUR STROKE DIESEL ENGINE

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Abstract: The paper presents a part of the work conducted in the first stage of a Research Grant called "Hybrid micro-cogeneration group of high efficiency equipped with an electronically assisted ORC" acronym GRUCOHYB. The hybrid micro-cogeneration group is equipped with a four stroke Diesel engine having a maximum power of 40 kW. A mathematical model of the internal combustion engine is presented. The mathematical model is developed based on the Laws of Thermodynamics and takes into account the real, irreversible processes. Based on the mathematical model a computation program was developed. The results obtained were compared with those provided by the Diesel engine manufacturer. Results show a very high correlation between the manufacturer's data and the simulation results for an engine running at 100% load. Future developments could involve using an exergetic analysis to show the ability of the ORC to generate electricity from recovered heat.

Keywords: irreversible processes, simulation, variable operation parameters, result validation

1. Introduction

The paper presents a part of the work conducted in the first stage of a Research Grant called "Hybrid micro-cogeneration group of high efficiency equipped with an electronically assisted ORC" acronym GRUCOHYB. The electricity and heat micro-generation system is equipped with a four-stroke Diesel engine with a maximum power of 40 kW. The design entails the recovery of the heat exhausted by the engine through its cooling system and of that from the exhaust gas. The recovered heat is intended to be used in order to obtain electricity through a Rankine cycle-based system using an organic substance as its working fluid (ORC), as well as to heat a thermal agent.

The main elements of the micro-generation system consist of: the internal combustion engine, thermal coupling through the engine-to-ORC system heat exchangers; the ORC system. The overall mathematical model of each subsystem of the micro-generation and global systems,

enables a simulation of the working system based on the variation of its functional and design parameters, which will result in the output of the optimum solution.

2. Mathematical model of the internal combustion engine

The engine type is overcharged Diesel. The mathematical model, based on the laws of thermodynamics [1-4], takes into account the real, irreversible processes that accompany the intake scavenging phenomenon, the free exhaust of flue gas, pressure losses at the engine cylinder intake and exhaust as well as at the turbocharger and compressor. The mathematical model takes into account the combustion process and the change in composition of the working agent as it traverses the engine work cycle.

The computation cycle of a turbocharged compression-ignited engine is illustrated in Fig. 1.
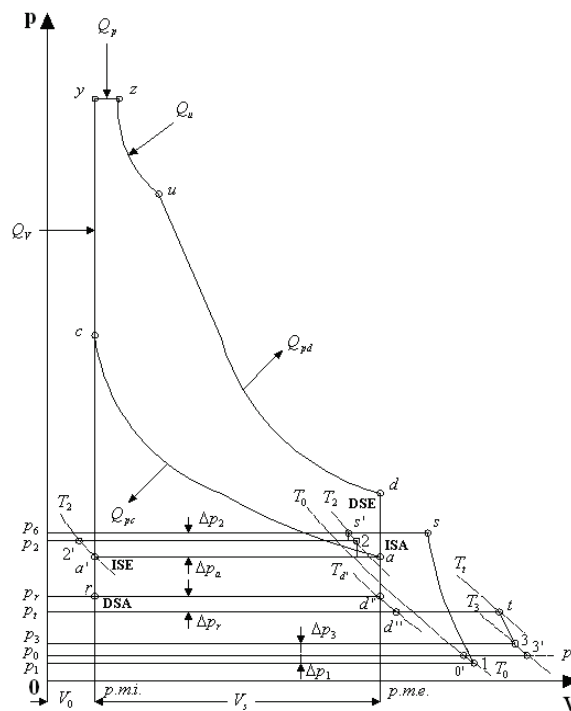

Fig. 1- Computation cycle of a turbocharged compression-ignited engine

2.1. Study of gas exchange processes

The blueprint of the gas exchange processes is illustrated in Fig. 2. Phase 2.α corresponds to the moment the exhaust valve opens and phase 2.β to the end of the forced exhaust when the m_r residual gas amount is left in the cylinder. The gas in the intake gallery carries out the cylinder run effect and finally its filling.

By integrating the differential equation for the first law of thermodynamics between states α and β we obtain the T_e exhaust gas temperature calculation equation.

$$\frac{T_e}{T_d} = \frac{1}{n_e} \cdot \frac{1+(\dfrac{\varepsilon-1}{\varepsilon}n_e-1)\dfrac{p_r}{p_d}}{1-\dfrac{\rho_r}{\varepsilon}(\dfrac{p_r}{p_d})^{1/k_e}}$$

(1)

Where n_e is exhaust process adiabatic coefficient.

The study of the run and intake processes takes place by integrating the energy ratio differential equation between the moment the exhaust valve opens (DSA) (phase β) and the time it closes (ISA) (phase γ).
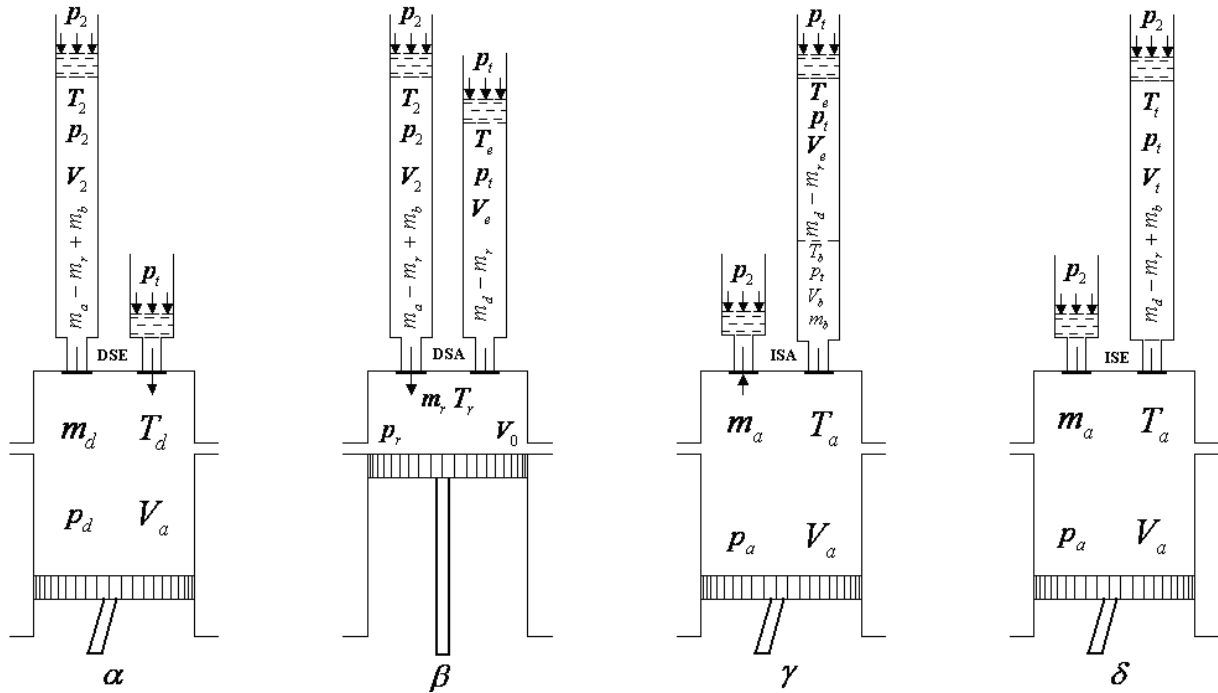

Fig. 2- Blueprint of the gas exchange processes

The work exchanged by the cylinder-pipe system and the exterior during the run and intake processes is expressed by the relation:

$$L_{\beta-\wp} = p_a \cdot V_a \left\{ \frac{\varepsilon-1}{\varepsilon} + (1 - \frac{T_a}{T_r} \cdot \frac{p_r}{p_a} \cdot \frac{1}{\varepsilon}) \left[(\beta-1)\frac{T_b}{T_a} - \beta\frac{T_2}{T_a} \right] \right\} \tag{2}$$

The T_b temperature of the mixture of air and flue gas that took part in the run process and were exhausted from the cylinder becomes:

$$\frac{T_b}{T_a} = \frac{n_a \left[\beta(1-g_r)\frac{T_2}{T_a} - 1 \right] + \frac{1}{\varepsilon}\left(n_a + \frac{k_a-1}{k_e-1} \cdot \frac{p_r}{p_a} - 1 \right)}{n_a(\beta-1)(1-g_r)} \tag{3}$$

2.1.1. Determining gas temperature at the turbine intake

This temperature is the result of an isobar mixing of gases having the temperatures Te and Tb (Fig.2.γ); noticing that enthalpy is an additive measurement, we get:

$$H_t = H_b + H_e \tag{4}$$

and continuing (Fig.2.δ):

$$(m_d - m_r + m_b)c_{p_g} T_t = m_b c_{p_a} T_b + (m_d - m_r)c_{p_g} T_e \tag{5}$$

The energy balance ration (5) enables the calculation of temperature T_t upstream of the turbine based on the run coefficient β:

$$T_t = \frac{\beta-1}{\beta+d} \cdot \frac{k_a}{k_e} \cdot \frac{k_e-1}{k_a-1} T_b + \frac{1+d}{\beta+d} T_e \tag{6}$$

2.1.2. Determining gas temperature at the turbine intake

Taking into account the adiabatic efficiency of the turbine, the equation of the T_3 gas temperature at the turbine exhaust becomes:

$$T_3 = \left\{ 1 - \eta_T \left[1 - \left(\frac{p_3}{p_t} \right)^{\frac{k_T - 1}{k_T}} \right] \right\} T_t \qquad (7)$$

2.2. Study of the burn process

The burn efficiency enables the correlation of the heat released by burning the fuel to the engine design parameters.

$$\eta_{ar} = \frac{Q_g}{Q_{cb_{in}}} = \frac{Q_{cb} - \left| Q_{P_{ar}} \right|}{Q_{cb}} = 1 - \frac{\left| Q_{P_{ar}} \right|}{Q_{cb}} \qquad (8)$$

Where Q_{cb} is the heat created by burning the fuel, Q_g the heat received by the flue gas and $Q_{P_{ar}}$ is the heat lost through the burn chamber walls.

The heat developed by burning the fuel is calculated like this:

$$Q_{cb} = m_{cb} \cdot H_i \qquad (9)$$

Where m_{cb} is the amount of fuel burned in the cylinder during each cycle and H_i is the lower calorific power of the fuel.

The amount of fuel consumed during a cycle is calculated by the formula:

$$m_{cb} = \frac{m_{aer}}{\alpha \cdot L_0} = \frac{m_{aer}}{m_{ad}} \cdot \frac{m_{ad}}{\alpha \cdot L_0} \qquad (10)$$

where L_0 is the minimum air required to burn one kg of fuel and α the minimum burn air coefficient. The thermal load of the engine displacement unit:

$$q_{cb} = \frac{Q_{cb}}{V_s} = d \cdot \eta_v \cdot \rho_0 \cdot H_i \quad \left[\frac{J}{m^3} \right] \qquad (11)$$

Forcefully filling the cylinder by overcharging acts on its volumetric thermal load, which increases the higher the overcharge pressure p_s is, and the cooling degree τ_R of the air in the cooler is higher.

3. Simulating engine behavior based on variations of operating parameters

Based on the mathematical model presented before, a computation program has been developed and solved using the EES [5] software. The program simulates the engine behavior that is based on variations of the operating parameters.

The simulation aim was to gather information pertaining $\dot{m}_{cb}$ fuel consumption, the temperature of the flue gas exhausted from the engine T_3 (at the turbine exhaust) and the engine η_{eM} efficiency when modifying the α air excess coefficient, the overcharge pressure p_s and the maximum pressure p_z of the operating cycle. The functional and design characteristics of the four-stroke overcharged Yanmar [6] Diesel engine are listed in Table 1.

**Functional and design characteristics of the Diesel engine**

Characteristic	Measurement Unit	Value
Power	kW	37,7
Rotation speed	rot/min	1500
Cylinder diameter, D	mm	98
Piston stroke, S	mm	110
Capacity, V_s	dm^3	3,319
Compression ratio, ε	-	18,1

The influence of the change of the air excess coefficient on the engine performances is illustrated in Fig. 3 to Fig. 5.
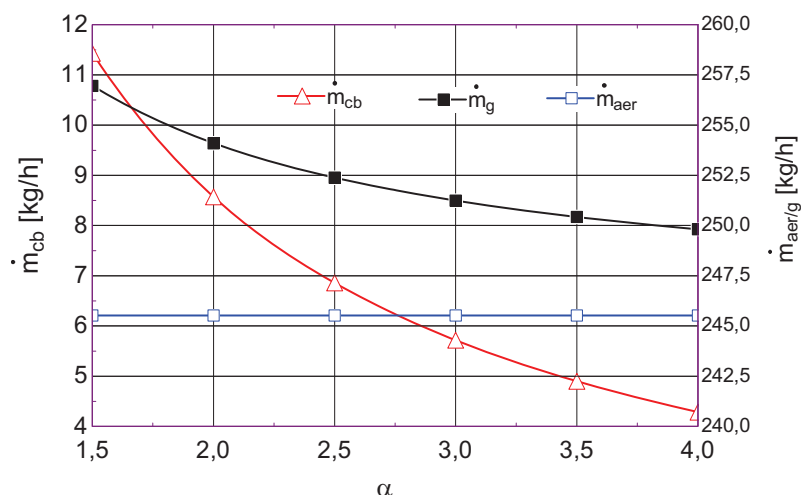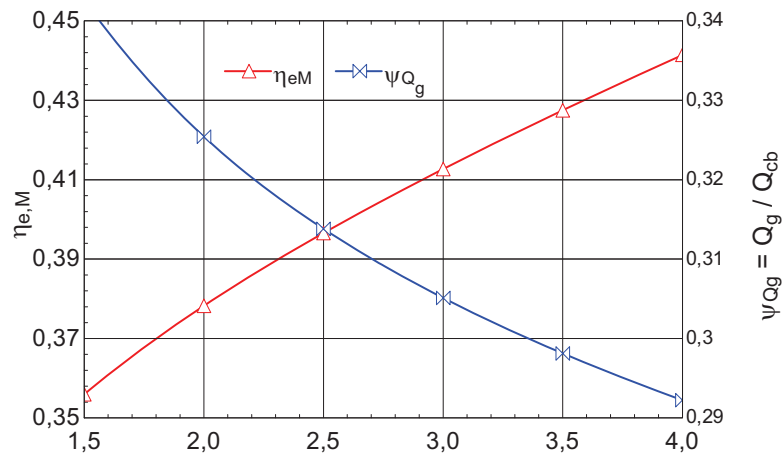

Fig.3 - The influence of the air excess coefficient α on hourly fuel, air and exhaust gas flows
(p_s=1,5 bar, t_0=25°C)

As it was to be expected, increasing the air excess coefficient leads to a rapid decrease in fuel consumption, with a corresponding reduction in engine load. In the illustration in figure 3, the engine intake air flow is not affected by the change in the air excess coefficient. This is probably due to the fact that the mathematic model, on which the engine running is simulated, holds that the other determining functional engine parameters are unchanged. The experimental study will provide precious information on how the engine behaves when the decision parameters vary, which will serve in correlating and improving the mathematical model.
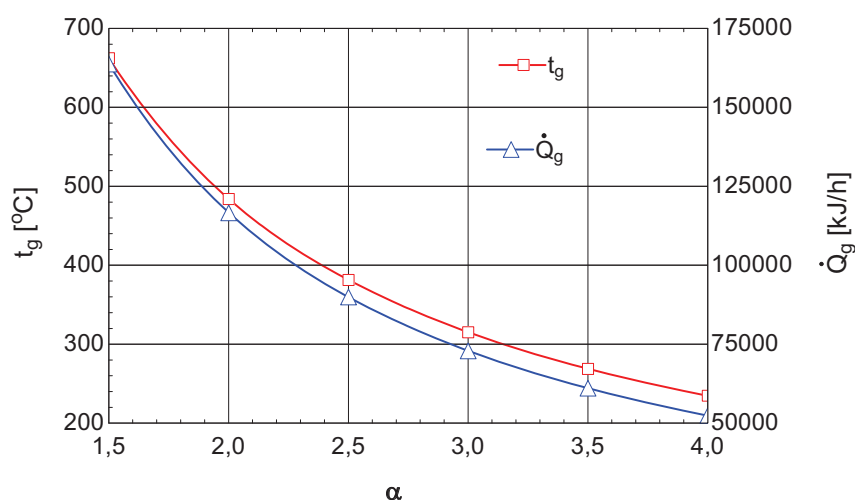
Fig.4 - Influence of the air excess coefficient α on the exhaust gas temperature and
the heat available in the exhaust gas (p_s=1,5 bar, t_0=25°C)

Fig.5 - Influence of the air excess coefficient α on the η_{eM} engine efficiency and on the ψ_{Qg} percentage of available heat in the exhaust gas from the heat generated by burning the fuel

Increasing the air excess coefficient α leads to a rapid decrease of the exhaust gas temperature, a decrease which reduces, at the same pace, the amount of recoverable heat from the exhaust gas. Figure 5 presents the influence of the excess air coefficient α on the η_M engine efficiency and on the ψ_{Qg} percentage of available heat in the exhaust gas from the heat generated by fuel burning.

Decreasing the exhaust gas temperature and as a consequence the potential of recoverable heat from the exhaust gas leads to a decrease in the fraction of available heat in the exhaust gas compared to the heat introduced by fuel burning. A comparison of the results obtained by simulating the engine operation and those presented by the manufacturer are listed in Tables 2.

Table 2

Comparison between simulation results and manufacturer's data

α		$\dot{m}_{cb}$ [kg/h]		$\dot{m}_g$ [kg/h]		t_g [°C]	
Model	Design	Model	Design	Model	Design	Model	Design
2	2.06	8.573	8.5075	254.1	259.55	483.9	480
$\dot{Q}_g$ [kJ/h]		η_M %		ψ_g %		t_s [°C]	
Model	Design	Model	Design	Model	Design	Model	Design
116610	102369	37.83	37.98	32.54	28.64	369	366.29

The comparative analysis shows a very high correlation between the experimental data and the simulation results for an engine running at 100% load. Instead of the energy potential for recovering heat from exhaust gas, which in no way reflects the ability to generate electricity using the ORC cycle, an exergetic analysis is recommended [7,8]

4. Conclusions

The paper presents a part of the results obtained in a first stage of a Research Grant called "Hybrid micro-cogeneration group of high efficiency equipped with an electronically assisted ORC" acronym GRUCOHYB. The micro-cogeneration group is equipped with a 40 kW overcharged Diesel engine. The aim of the research is to recover heat from the exhaust and cooling systems of the Diesel engine and convert it into electricity using an ORC based system. A mathematical model of the internal combustion engine is presented. The mathematical model is developed based on the Laws of Thermodynamics and takes into account the real, irreversible processes of scavenging during the intake, free evacuation of exhaust gasses, pressure losses

during the engine cylinders intake and exhaust stages and pressure losses at the inlet and outlet of the turbocharger and compressor. Also the combustion process and the change in composition of the working agent as it traverses the engine work cycle were taken into account. Based on the mathematical model a computation program was developed and solved using the EES software, simulating the engine behavior based on variations of the operating parameters. The results obtained were compared with those provided by the Diesel engine manufacturer. Results show a very high correlation between the manufactures data and the simulation results for an engine running at 100% load. Future developments could involve using an exergetic analysis to show the ability of the ORC to generate electricity from the recovered heat.

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