THERMODYNAMIC MODELLING AND EFFICIENCY ANALYSIS OF A CLASS OF REAL INDIRECTLY FIRED GAS TURBINE CYCLES

by

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Indirectly or externally-fired gas-turbines (IFGT or EFGT) are novel technology under development for small and medium scale combined power and heat supplies in combination with micro gas turbine technologies mainly for the utilisation of the waste heat from the turbine in a recuperative process and the possibility of burning biomass or "dirty" fuel by employing a high temperature heat exchanger to avoid the combustion gases passing through the turbine. In this paper, by assuming that all fluid friction losses in the compressor and turbine are quantified by a corresponding isentropic efficiency and all global irreversibilities in the high temperature heat exchanger are taken into account by an effective efficiency, a one dimensional model including power output and cycle efficiency formulation is derived for a class of real IFGT cycles. To illustrate and analyze the effect of operational parameters on IFGT efficiency, detailed numerical analysis and figures are produced. The results summarized by figures show that IFGT cycles are most efficient under low compression ratio ranges (3.0-6.0) and fit for low power output circumstances integrating with micro gas turbine technology. The model derived can be used to analyze and forecast performance of real IFGT configurations.

Key words: *indirectly fired gas turbine, thermodynamic analysis, bioenergy, biomass utilization, micro gas turbine*

Introduction

In recent years, renewable energy utilization has grown rapidly. Electricity generated by utilizing renewable energy sources occupied 19% of world electricity gross used in 2005 [1]. The state-of-the-art renewable energy sources include large hydropower and other sources such as wind, biomass, solar, geothermal, and small hydros. However, related energy conversion technologies should combine high conversion-efficiencies with low emissions, especially CO_2 emission. Modern energy-conversion technologies like internal combustion engines and ordinary gas turbines demand clean fuels as the combustion gases are in direct contact with the moving parts of the machine. In order to use "dirty" fuels including biomass fuel and waste fuel, indirect combustion systems which separate combustion process and thermodynamic conversion cycles are a class of state-of-the-art and promising research objects. In addition to conventional coal-fired steam cycle plants and Stirling engines, indirectly or externally-fired gas-turbine (IFGT or EFGT) also forms a novel technology under development for small and medium scale combined power and heat (CHP) supplies in combination with micro gas turbine technologies [2]. The most attractive advantage of IFGT units is that they open a new option to utilize biomass fuel for CHP and contribute to reduce greenhouse-gas emissions. Up to now, studies of externally fired gas turbines have been performed by a number of different research organizations and power equipment manufacturers. Most available publications are devoted to various cycle configurations to attain high conversion efficiencies [3-7], to different types of furnace arrangements [8-11], and to the development of possible high-temperature heat exchangers [12-18]. In addition, several excellent review papers and reports on IFGT applications are available [19-21].

In this paper, by assuming that all fluid friction losses in the compressor and turbine are quantified by a corresponding isentropic efficiency and all global irreversibilities in the high temperature heat exchangers (HTHE) are taken into account by an effective efficiency, a one-dimensional model including power output and cycle efficiency formulation is derived for a class of real IFGT cycles based on thermodynamic analysis method. Detailed numerical analysis and figures are produced to illustrate and analyse the effect of operational parameters and components' parameters on IFGT efficiency.

Cycle model and analysis

The schematic and *T-s* diagrams of a class of IFGT cycles are shown in figs.1 and 2, respectively. The basic components of this class of cycles are the compressor, high temperature heat exchangers, combustion chamber, and gas turbine as shown in fig. 1. The working air enters the compressor at state 1 and is non-isentropically compressed to state 2. After state 2 (ideally to state 2s), the air leaving the compressor enters the HTHE and is heated to state 3 by the high temperature burnt biomass gas flow. Then the heated air enters the gas turbine and expands non-isentropically to state 4 (ideally to state 4s). After leaving the gas turbine, the still hot air enters the combustion chamber and takes part in the combustion process with biomass fuel to the highest temperature point of the cycle, *i. e.*, state 5. The hot gas exits the combustion chamber and enters the HTHE, where it adds heat to the air and cooled to state 6 at constant pressure. Finally the cycle is completed by cooling the gas to the initial state. In fig. 2, the process 1-2s is an isentropic compression and 1-2 takes into account the non-isentropic nature of a real compressor; the process 3-4s is an isentropic expansion and 3-4 a real non-isentropic expansion in a real turbine. We consider the real IFGT cycle 1-2-3-4-5-6-1.

In order to investigate and analyze how IFGT cycle efficiency is affected by cycle operating parameters and components performance parameters, following assumptions are made

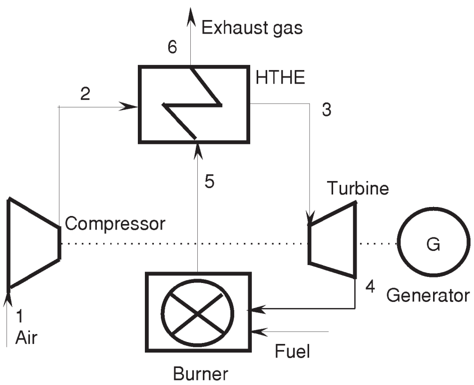

Figure 1. Schematic diagram of the IFGT cycle

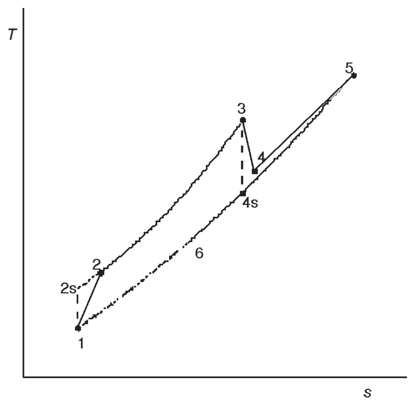

Figure 2. *T-s* diagram of the IFGT cycle

for the one-dimensional ideal gas steady flow and application of the principle of energy conservation to each component:

– the working fluids (air and combustion gases) are ideal gases and each possesses constant specific heat; the working fluids flow through the system in a one-dimensional quasistatic-state fashion,
– the compression process 1-2 and expansion process (4-5) are adiabatic and irreversible; the deviations from the isentropic processes are accounted for by the isentropic efficiencies; the gas turbine's isentropic efficiency, η_t, gives the ratio of the actual expansion process producing power to the isentropic one; the isentropic efficiency of the compressor, η_c, is defined as the ratio of the power requirement during the isentropic compression to that of the actual compression; η_t and η_c have the values between zero and unity:

$$\eta_t \quad \frac{T_3 \quad T_4}{T_3 \quad T_{4s}} \tag{1}$$

$$\eta_c \quad \frac{T_{2s} \quad T_1}{T_2 \quad T_1} \tag{2}$$

– the heat transfer in the real high temperature heat exchanger (HTHE) is described by using the HTHE efficiency η_{HE}; the η_{HE} is defined as:

$$\eta_{HE} \quad \frac{T_3 \quad T_2}{T_5 \quad T_6} \tag{3}$$

which is the ratio of the actual heat transfer process 2-3 to the maximum ideal amount of heat transfer (5-6); the value of η_{HE} varies from 0 to 1,

– the cycle has a pressure ratio, which is the ratio of the maximum pressure of the cycle obtained at the compressor exit (state 2) to the minimum pressure of the cycle at the turbine exit which corresponds to the compressor inlet (state 1), *i. e.*:

$$\pi_c \quad \frac{p_2}{p_1} \tag{4}$$

– the pressure drop in the flow piping is reflected by using a total pressure recovery coefficient, that is:

$$D \quad 1 \quad \frac{\Delta p}{p} \tag{5}$$

$$\frac{p_3}{p_4} \quad D\frac{p_2}{p_1} \tag{6}$$

The cycle power output P_{IFGT} and efficiency η_{IFGT} are defined as:

$$P_{IFGT} = P_t - P_c \tag{7}$$

$$\eta_{IFGT} \quad \frac{P_{IFGT}}{\dot{Q}_{in}} \tag{8}$$

where P_t, P_c, $\dot{Q}_{in}$ are power output by the gas turbine, compression power consumed by the compressor, and cycle heat absorption, respectively, and can be written as follows:

$$P_t \quad \dot{m}c_p(T_3 \quad T_{4s})\eta_t \tag{9}$$

$$P_{\mathrm{c}} \quad \frac{\dot{m}c_{\mathrm{p}}(T_{2\mathrm{c}} \quad T_1)}{\eta_{\mathrm{c}}} \tag{10}$$

$$\dot{Q}_{\mathrm{in}} \quad \dot{Q}_{45} \quad \dot{m}c_{\mathrm{p}}(T_5 \quad T_4) \tag{11}$$

Define x as the isentropic temperature ratio in the compressor. According to the knowledge of thermodynamics, one can obtain:

$$x \quad \frac{T_{2\mathrm{s}}}{T_1} \quad \frac{p_2}{p_1}^{\;m} \quad \pi_{\mathrm{c}}^{\mathrm{m}} \tag{12}$$

where $m = (k - 1)/k$ and k is adiabatic exponent (specific heat ratio).

According to the definition of total pressure recovery coefficient, one can obtain the isentropic temperature ratio in the gas turbine:

$$\frac{T_3}{T_{4\mathrm{s}}} \quad \frac{p_3}{p_4}^{\;m} \quad (\pi_{\mathrm{c}}D)^m \quad xD^m \tag{13}$$

The temperature differences, ΔT_{H} and ΔT_{L}, in HTHE are very important parameters in the design and performance optimization of the IFGT cycles [3, 20]. In general, a small temperature-difference improves the utilization of the heat and the efficiency, but increases the size of the heat exchanger, the costs and the pressure drop. As from fig. 1, ΔT_{H} and ΔT_{L} are written as:

$$\Delta T_{\mathrm{H}} = T_5 - T_3 \tag{14}$$

and

$$DT_{\mathrm{L}} = T_6 - T_2 \tag{15}$$

Defining y is equal to the ration of these two temperature differences:

$$y \quad \frac{\Delta T_{\mathrm{L}}}{\Delta T_{\mathrm{H}}} \quad \frac{T_6 \quad T_2}{T_5 \quad T_3} \tag{16}$$

Let

$$\alpha \quad \frac{T_1}{T_3} \tag{17}$$

From eqs. (17) and (13), one can get T_1 and T_4, respectively:

$$T_1 = \alpha T_3 \tag{18}$$

$$T_{4s} \quad \frac{T_3}{xD^m} \tag{19}$$

Substituting eqs. (18) and (19) into eq. (2) yields:

$$T_4 \quad T_3 \;\; 1 \quad \eta_t \quad \frac{\eta_t}{xD^m} \tag{20}$$

Combing eq. (12) and eq. (18), one can get:

$$T_{2\mathrm{s}} = x\alpha T_3 \tag{21}$$

Using eqs. (1), (18), and (21), T_2 is seen to be:

$$T_2 \quad T_3\alpha \;\; 1 \quad \frac{x}{\eta_c} \quad \frac{\eta}{c} \tag{22}$$

Solving eqs. (3) , (16), and (22), one can get:

$$T_5 \quad T_3 \; \frac{\eta_{\mathrm{HE}}^{1} \quad y}{1 \quad y} \quad \frac{\alpha(1 \quad \eta_{\mathrm{HE}}^{1})(1 \quad x\eta_{\mathrm{c}}^{1} \quad \eta_{\mathrm{c}}^{1})}{1 \quad y} \tag{23}$$

$$T_6 \quad \frac{T_3}{1 \quad y}[y(\eta_{\mathrm{HE}}^{1} \quad 1) \quad \alpha(1 \quad y\eta_{\mathrm{HE}}^{1})(1 \quad x\eta_{\mathrm{c}}^{1} \quad \eta_{\mathrm{c}}^{1}) \tag{24}$$

Thus

$$\Delta T_{\mathrm{H}} \quad \frac{T_3(\eta_{\mathrm{HE}}^{1} \quad 1)[1 \quad \alpha(1 \quad x\eta_{\mathrm{c}}^{1} \quad \eta_{c}^{1})]}{1 \quad y} \tag{25}$$

$$\Delta T_{\mathrm{L}} \quad \frac{y(\eta_{\mathrm{HE}}^{1} \quad 1)[1 \quad \alpha(1 \quad x\eta_{\mathrm{c}}^{1} \quad \eta_{\mathrm{c}}^{1})]}{1 \quad y} T_3 \tag{26}$$

$$P_{\mathrm{c}} \quad \frac{\dot{m}c_{\mathrm{p}}(T_{2\mathrm{s}} \quad T_1)}{\eta_{\mathrm{c}}} \quad \frac{(x \quad 1)}{\eta_{\mathrm{c}}}\dot{m}c_{\mathrm{p}}T_3 \tag{27}$$

$$P_{\mathrm{t}} \quad \dot{m}c_{\mathrm{p}}(T_3 \quad T_{4\mathrm{s}})\eta_{\mathrm{t}} \quad \eta_{\mathrm{t}}(1 \quad x^{\;1}D^{\;\mathrm{m}})\dot{m}c_{\mathrm{p}}T_3 \tag{28}$$

$$\dot{Q}_{\mathrm{in}} \quad \dot{Q}_{45} \quad \dot{m}c_{\mathrm{p}}(T_5 \quad T_4) \quad \dot{m}c_{\mathrm{p}}T_3 \quad \frac{(\eta_{\mathrm{HE}}^{1} \quad 1)[1 \quad \alpha(1 \quad x\eta_{\mathrm{c}}^{1}\eta_{c}^{1})]}{1 \quad y} \quad \eta_{\mathrm{t}}(1 \quad x^{\;1}D^{\;m}) \tag{29}$$

$$\eta_{\mathrm{IFGT}} \quad \frac{P_{\mathrm{t}} \quad P_{\mathrm{c}}}{\dot{Q}_{\mathrm{in}}} \quad \frac{\eta_{\mathrm{t}}(1 \quad x^{\;1}D^{\;\mathrm{m}}) \quad \alpha\eta_{\mathrm{c}}^{1}(x \quad 1)}{\dfrac{(\eta_{\mathrm{HE}}^{1} \quad 1)[1 \quad \alpha(1 \quad x\eta_{c}^{1} - \eta_{c}^{1})]}{1 \quad y} \quad \eta_{\mathrm{t}}(1 \quad x^{\;1}D^{\;\mathrm{m}})} \tag{30}$$

Results and discussions

Equation (30) indicates that the efficiency of IFGT cycles η_{IFGT} studied is a function of the isentropic efficiency of the gas turbine η_{t}, the isentropic efficiency of the compressor η_{c}, the HTHE efficiency η_{HE}, the total pressure recovery coefficient D, the total pressure ratio π_{c} ($x = \pi_c^m$), the ration of low temperature side temperature difference (ΔT_{L}) to high temperature side temperature difference (ΔT_{H}) y, and the ratio of compressor inlet temperature to gas turbine inlet temperature α. As well, it is known that the key component in the IFGT configuration is the counter-flow high temperature heat exchanger (HTHE). Its regenerative efficiency and the temperature difference between the hot and cold gas are most important parameters affecting the total cycle performance. At the hot end of the HTHE, where the hot gas enters and the heated air exits, the hot-side temperature difference is defined by $\Delta T_{\mathrm{H}} = T_5 - T_3$ (fig. 2). For a realistic hot-side temperature difference $\Delta T_{\mathrm{H}} = 70$ K, in combination with design data for Turbec T100 (ABB/Volvo) micro gas-turbine whose gas temperature at turbine inlet, gas temperature after recuperator, and air temperature at compressor outlet are 1223, 523, and 487 K, respectively [3], on can compute the temperature difference ratio of the HTHE, *i. e.*, $y = 0.80$. To see how these different parameters affect IFGT cycle performance, detailed numerical solution of eq. (30) is processed and illustrated with a preset of y to be 0.80.

For a varying total compression ratio of the compressor (π_c), IFGT cycle efficiency η_{IFGT} with $\eta_c = 0.8$, $\eta_t = 0.85$, $\eta_{HE} = 0.90$, $y = 0.80$, and $D = 0.85$ under several specific α values are plotted in fig. 3. For ordinary IFGT configurations, compressor inlet temperature T_1 is usually equal to environmental temperature. Thus a smaller α value means a higher gas turbine inlet temperature T_3. Assuming $T_1 = 298$ K, α of 0.15, 0.20, 0.25, and 0.30 correspond to T_3 of 1987, 1490, 1192, and 993 K, respectively. As one can see, for a given compression ratio, IFGT cycle efficiency η_{IFGT} increases with the gas turbine inlet temperature. For an engineering-realizable state-of-the-art IFGT, α varies between 0.20 and 0.30. Thereafter it can be seen from fig. 3 that efficiency attains its maximum value for a compression ratio of 3.0-6.0, being the higher values referred to configurations with the higher gas turbine inlet temperature T_3.

The influence of the isentropic efficiency of the gas turbine and of the compressor ($\eta_t = = \eta_c$) on the IFGT cycle efficiency η_{IFGT} with respect to the total pressure ratio π, under the conditions of $\eta_{HE} = 0.90$, $D = 0.85$, $y = 0.80$, and $\alpha = 0.20$ is shown in fig. 4. It illustrates that the cycle efficiency η_{IFGT} increases with the isentropic efficiency of the gas turbine and of the compressor.

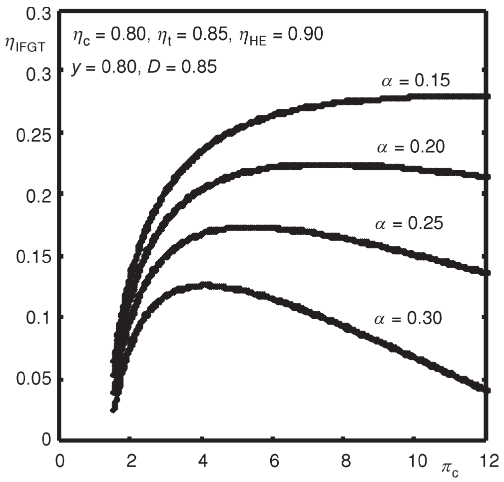

Figure 3. η_{IFGT} ***vs.*** π_c

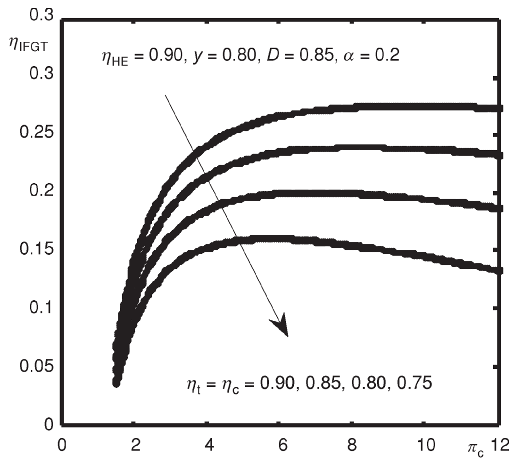

Figure 4. η_{IFGT} ***vs.*** η_t **and** η_c

Variations of IFGT cycle efficiency η_{IFGT} with the total pressure recovery coefficient D are illustrated in fig. 5, with, $\eta_c = 0.80$, $\eta_t = 0.85$, $\eta_{HE} = 0.90$, $y = 0.80$, and $\pi_c = 5.0$ under different gas turbine inlet temperature T_3. For a given total pressure recovery coefficient D, IFGT cycle efficiency η_{IFGT} increases with the gas turbine inlet temperature T_3. The total pressure recovery coefficient D affects η_{IFGT} linearly since the variable just describes piping flow losses.

Since the counter-flow high temperature heat exchanger (HTHE) is the most important component in the IFGT configuration, to illustrate and analyze the effect of its performance parameters on IFGT total performance is always considered in the design and operation stages. Figure 6 shows the influence of the HTHE efficiency η_{HE} on the IFGT cycle efficiency η_{IFGT} with respect to the total pressure ratio π_c under the conditions of $\eta_c = 0.80$, $\eta_t = 0.85$, $D = 0.85$, $y = 0.80$, and $\alpha = 0.20$. One can see the HTHE efficiency η_{HE} remarkably affects the IFGT cycle efficiency η_{IFGT}. In the ideal condition with $\eta_{HE} = 1.0$, the IFGT cycle efficiency η_{IFGT} reaches its maximum about 50% under total compression ratio $\pi_c = 2.5$. In real circumstances considering losses, a small value of decrease of η_{HE} from 1.0 to 0.95 causes the maximum IFGT cycle efficiency η_{IFGT} a sharp decrease from about 50% to nearly 30%.

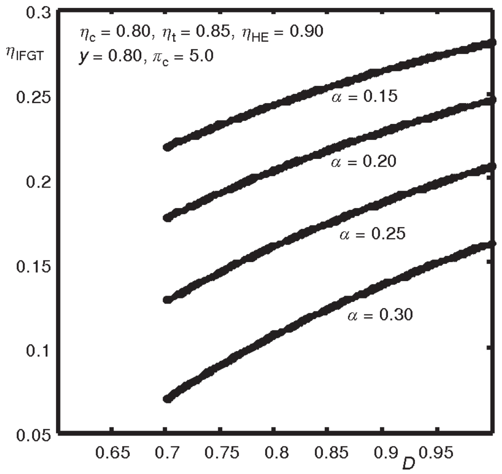

Figure 5. η_{IFGT} *vs.* D

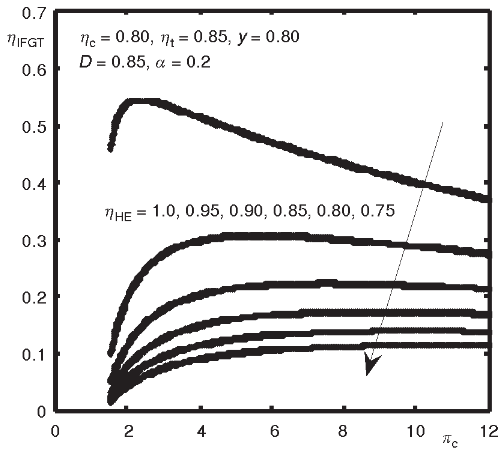

Figure 6. η_{IFGT} *vs.* η_{HE}

Conclusions

By taking several assumptions, a one-dimensional model including power output and cycle efficiency formulation is derived for performance analysis of a class of real IFGT cycles. The assumptions include that all fluid friction losses in the compressor and turbine are quantified by a corresponding isentropic efficiency and that all global irreversibilities in the HTHE are taken into account by an effective efficiency. Based on detailed numerical analysis on a set of preset parameters affecting IFGT performance and corresponding illustrations, it can be concluded that IFGT cycles are most efficient under low compression ratio ranges (3.0-6.0) and fit for low power output circumstances integrating with micro gas turbine technology. The IFGT cycle efficiency is always below 30%. The HTHE efficiency and temperature differences at both hot-side and cold-side affect the IFGT cycle efficiency η_{IFGT} sharply. The formulae derived can be used to analyze and forecast performance of real IFGT configurations.

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