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THERMODYNAMIC ANALYSIS OF SUPPLEMENTARY-FIRED GAS TURBINE CYCLES

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ABSTRACT

This paper presents an analysis of the possibilities for improving the efficiency of an indirectly biomass-fired gas turbine (IBFGT) by supplementary direct gas-firing. The supplementary firing may be based on natural gas, biogas, or pyrolysis gas. The interest in this cycle arise from a recent demonstration of a two-stage gasification process through construction of several plants. A preliminary analysis of the ideal recuperated Brayton cycle shows that for this cycle any supplementary firing will have a marginal efficiency of unity per extra unit of fuel. The same result is obtained for the indirectly fired gas turbine (IFGT) and for the supplementary-fired IFGT. Both results show that the combination of external firing and internal firing have the potential of reducing or solving some problems with the use of biomass both in the recuperated and the indirectly fired gas turbine: The former requires a clean, expensive fuel. The latter is limited in efficiency due to limitations in material temperature of the heat exchanger. Thus, in the case of an IBFGT, it would be very appropriate to use a cheap biomass or waste fuel for low temperature combustion and external firing and use natural gas at a high marginal efficiency for high

temperature heating. However, it is shown that this is not the case for a simple IBFGT supplementary fired with natural gas. Instead, other process changes may be considered in order to obtain a high marginal efficiency on natural gas. Two possibilities are analysed: Integration between an IFGT and pyrolysis of the biofuel which will result in a highly efficient utilization of the biomass, and integration between external biomass firing, internal biomass firing and internal natural gas firing. The marginal efficiency of the natural gas is in this case found to be independent of temperature ratio and lower than for the recuperated gas turbine.

NOMENCLATURE

c_p	Specific heat [kJ/kgK]
\dot{m}	Mass flow rate [kg/s]
PR	Pressure ratio [–]
\dot{Q}_i	Input heat flow rate [kJ/s]
\dot{Q}_o	Rejected heat flow rate [kJ/s]
T	Temperature [K]

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T_h	Maximum temperature for internal combustion [K]
T_l	Ambient temperature [K]
T_m	Maximum temperature for external combustion [K]
TR	Temperature ratio [–]
\dot{W}	Power [kW]
$\bar{\eta}$	Marginal efficiency [–]
η	Efficiency [–]
κ	Isoentropic exponent [–]

1...7 Indices defined in figures 2, 4 and 6

INTRODUCTION

The indirectly fired gas turbine (IFGT) has currently not reached a technological level making it commercially competitive. The main reason for this is that the cycle involves a heat exchanger transferring heat from the hot combustion products to the turbine inlet air. This requires the heat exchanger to operate at the highest temperature in the cycle. Material considerations and present designs limit the temperature of the heat exchanger to 700-800°C for metallic materials. The attainable efficiency will be limited by this maximum cycle temperature, and other ways to raise the process efficiency have to be explored. In this paper we propose supplementary direct firing as a way of raising the maximum cycle temperature, and thereby the efficiency of an IFGT, without exceeding the heat exchanger temperature limitation. This idea has been introduced in [1]. More recently, it has been studied by [2].

The paper addresses three different gas turbine cycles, i.e., a directly-fired, recuperated gas turbine; a simple cycle IFGT; and an IFGT with supplementary direct firing. For the latter we consider different fuels for supplementary firing in order to obtain either high efficiency on the biomass or high marginal efficiency on the more expensive fuel, the natural gas. The primary interest in this IFGT with

supplementary firing is prompted by the development and demonstration through construction of two-stage biomass conversion plants. By careful control of temperatures this concept has the potential of retaining the environmentally objectionable and corrosive chemicals in the ashes while producing gas with tar contents of an acceptable magnitude.

This paper addresses only the ideal cycles, with reversible turbo machinery and ideal heat transfer. Gases are assumed to be perfect and having constant specific heat. The various gas turbines are described by air standard cycles.

History

The Indirectly Fired Gas Turbine (IFGT) has been under consideration for a long time [1, 3]. The driving force has almost exclusively been the possibility for using it for coal [4, 5, 6, 7], but also wood has been considered. However, a successful design has not materialized. This is due to the limited durability of the heat exchanger material in the presence of corrosive gases, the problem of fouling, and also because of the very high efficiencies achieved by the most important competitor, the modern coal-fired steam power plant.

IFGT's have been the subject of many theoretical studies, but the published efficiencies have been quite low. The main parameter of optimization has been the highest temperature of the cycle which is that of the combustion products at the inlet of the heat exchanger. Moreover, many studies have focused on the IFGT for application to coal-generated power [4, 5, 7, 8, 9]. One interesting reference on coal-fired IFGT's is the Ackeret-Keller-based plants, of which the oldest is the closed-cycle 2 MW plant which has been in operation since the 1950's operating with a maximum temperature of 700°C [10]. This is a low temperature for gas turbines in general, also for operation on coal, but it will be shown below that operating temperatures in this range may still be interesting for micro gas turbines fueled with wet biomass.

The efficiency of Open-cycle simple IFGT's for biomass does not appear to be better than coal cycles. They are both limited by the same temperature. Also, in the smaller sizes, the

high specific cost of high-temperature heat exchangers rules out the use of these. It should be noted that one gas turbine has been modified for “humid air” (HAT) application [11] and a number of novel concepts are subject to theoretical and experimental studies [12, 13, 14].

Activities in the Field at the Technical University of Denmark

The reported study is one of a number of projects aimed at the utilization of biomass for generation of electric power. These include fundamental research, modeling and demonstration of gasification of wood and straw, cofiring of biomass with coal, biomass-fired Stirling engines, and biogasification of biomass. The activities in the area of the IFGT started in 1994 and have resulted in two Master Theses [15, 16] and a few informal reports, but a very modest rate of progress. However, quite recently, theoretical studies of a power plant design based on the combination of a biomass drying unit with the IFGT have shown results with considerable promise [17, 18]. Results to date point to conversion efficiencies, that are higher by a factor of 3 to 10, relative to the best current competing technology, namely biogasification.

RECUPERATED GAS TURBINE

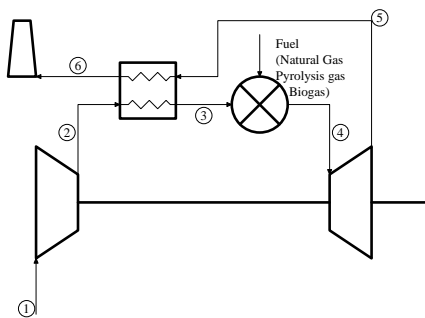


Figure 1: Flowsheet of a recuperated gas turbine

The flowsheet and the T - s -diagram of the recuperated gas turbine cycle are shown in figures 1 and 2, respectively. It consists of

- isentropic compression from 1 to 2
- isobaric heating in recuperator from 2 to 3

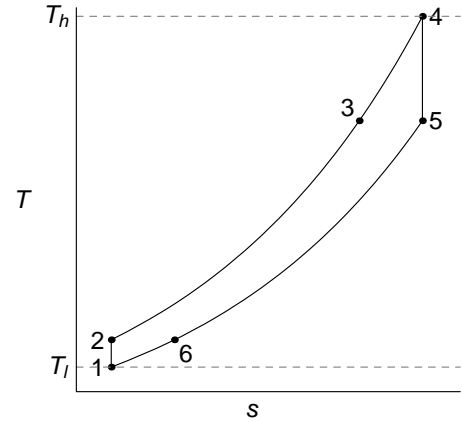


Figure 2: T - s -diagram for the Recuperated Gas Turbine Cycle

- isobaric temperature increase in combustor from 3 to 4
- isentropic expansion from 4 to 5
- isobaric cooling in recuperator from 5 to 6

The net heat input to the cycle is:

$$\dot{Q}_i = \dot{Q}_{34} = \dot{m}c_p(T_4 - T_3) = \dot{m}c_p(T_4 - T_5) \quad (1)$$

The net heat rejected is:

$$\dot{Q}_o = \dot{Q}_{61} = \dot{m}c_p(T_1 - T_6) = \dot{m}c_p(T_1 - T_2) \quad (2)$$

By application of the relation between pressures and temperatures for isentropic state changes

$$TR_{is} = PR^{\frac{\kappa-1}{\kappa}} \quad (3)$$

it is found that:

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \quad (4)$$

$$\frac{T_5}{T_4} = \frac{1}{PR^{\frac{1-\kappa}{\kappa}}} \quad (5)$$

and thereby after setting $T_1 = T_l$ and $T_4 = T_h$

$$\dot{Q}_i = \dot{m}c_p T_h \left(1 - \frac{1}{PR^{\frac{\kappa-1}{\kappa}}} \right) \quad (6)$$

$$\dot{Q}_o = -\dot{m}c_p T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \quad (7)$$

This leads to a power output of:

$$\begin{aligned} \dot{W} &= \dot{Q}_i + \dot{Q}_o \\ &= \dot{m}c_p \left(T_h \left(1 - PR^{\frac{1-\kappa}{\kappa}} \right) - T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \right) \end{aligned} \quad (8)$$

The efficiency of the process is defined as:

$$\eta \equiv \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_h} \left(PR^{\frac{\kappa-1}{\kappa}} \right) \quad (9)$$

which equals the Carnot efficiency in the limiting case with $PR = 1$.

The marginal efficiency, i.e., the efficiency obtained by adding a small amount of fuel to reach a combustion temperature of $T_h + \Delta T_h$ is:

$$\bar{\eta} = \frac{\frac{\partial \dot{W}}{\partial T_h}}{\frac{\partial \dot{Q}_i}{\partial T_h}} \approx \frac{\Delta \dot{W}}{\Delta \dot{Q}_i} = \frac{\dot{m} c_p \Delta T_h (1 - PR^{\frac{1-\kappa}{\kappa}})}{\dot{m} c_p \Delta T_h (1 - PR^{\frac{1-\kappa}{\kappa}})} = 1 \quad (10)$$

This shows, that in the ideal case any supplementary firing will be thermodynamically favourable for the recuperated gas turbine.

INDIRECTLY FIRED GAS TURBINE

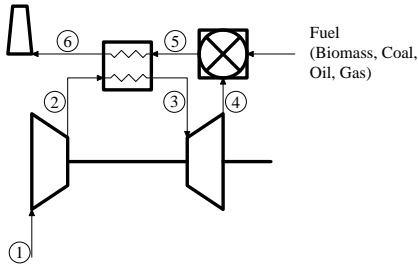


Figure 3: Flowsheet of an IFGT

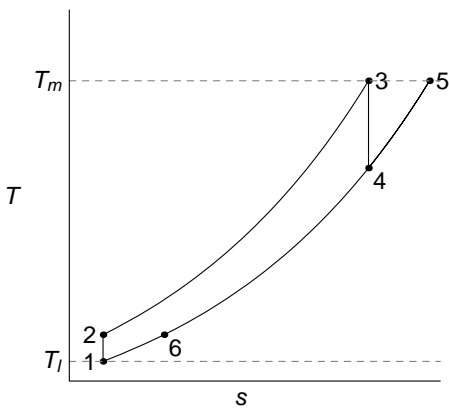


Figure 4: T - s -diagram for the Indirectly Fired Gas Turbine Cycle

The flowsheet and the T - s -diagram of the indirectly fired gas turbine cycle are shown in figures 3 and 4, respectively. It consists of:

- isentropic compression from 1 to 2
- isobaric heating in high-temperature heat exchanger from 2 to 3
- isentropic expansion from 3 to 4
- isobaric temperature increase in combustor from 4 to 5
- isobaric cooling in high-temperature heat exchanger from 5 to 6

The net heat input to the cycle is:

$$\dot{Q}_i = \dot{Q}_{45} = \dot{m} c_p (T_5 - T_4) = \dot{m} c_p (T_3 - T_4) \quad (11)$$

The net heat rejected is:

$$\dot{Q}_o = \dot{Q}_{61} = \dot{m} c_p (T_1 - T_6) = \dot{m} c_p (T_1 - T_2) \quad (12)$$

By application of equation (3) it is found that:

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \quad (13)$$

$$\frac{T_4}{T_3} = \frac{1}{PR^{\frac{1-\kappa}{\kappa}}} \quad (14)$$

and thereby after setting $T_1 = T_l$ and $T_3 = T_5 = T_m$

$$\dot{Q}_i = \dot{m} c_p T_m \left(1 - \frac{1}{PR^{\frac{\kappa-1}{\kappa}}} \right) \quad (15)$$

$$\dot{Q}_o = -\dot{m} c_p T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \quad (16)$$

This leads to a power output of:

$$\begin{aligned} \dot{W} &= \dot{Q}_i + \dot{Q}_o \\ &= \dot{m} c_p \left(T_m \left(1 - PR^{\frac{1-\kappa}{\kappa}} \right) - T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \right) \end{aligned} \quad (17)$$

The efficiency of the process is:

$$\eta = \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_m} \left(PR^{\frac{\kappa-1}{\kappa}} \right) \quad (18)$$

which is the same as for the recuperated cycle except for difference in maximum cycle temperature. Thus, the marginal efficiency found for raising the temperature to $T_m + \Delta T_m$ is also:

$$\bar{\eta} = \frac{\frac{\partial \dot{W}}{\partial T_m}}{\frac{\partial \dot{Q}_i}{\partial T_m}} \approx \frac{\Delta \dot{W}}{\Delta \dot{Q}_i} = \frac{\dot{m} c_p \Delta T_m (1 - PR^{\frac{1-\kappa}{\kappa}})}{\dot{m} c_p \Delta T_m (1 - PR^{\frac{1-\kappa}{\kappa}})} = 1 \quad (19)$$

Similar to the recuperated cycle, this indicates, that in the ideal case any supplementary firing would be thermodynamically favourable for the IBFGT.

INDIRECTLY FIRED GAS TURBINE WITH SUPPLEMENTARY FIRING

An indirectly fired gas turbine with supplementary direct gas firing is an interesting combination of the simple IFGT and the recuperated gas turbine. It has several technical advantages, because it will overcome the main problems with both of the two separate cycles.

Firstly, the recuperated gas turbine is directly fired and thus requires the fuel to be clean; usually natural gas in power applications. Natural gas is an expensive fuel, so an alternative of using a cheap fuel for the low-temperature part of the cycle, may be economically favourable.

Secondly, the introduction of heat into the IFGT is achieved in a high-temperature heat exchanger. Several studies [19, 20, 21, 22] have shown that the development of this component for very high temperatures for coal applications is very difficult. For biomass which may be more corrosive than coal, the maximum allowable temperature of the heat exchanger will be further constrained. With current technology this temperature should probably not exceed 700°C. This leads to a suggestion of a process with indirect biomass-firing and supplementary direct natural gas firing. A flowsheet and T - s -diagram of the cycle is shown figures 5 and 6, respectively. The process consists of:

- isentropic compression from 1 to 2
- isobaric heating in high-temperature heat exchanger from 2 to 3
- isobaric temperature increase in natural gas combustor from 3 to 4
- isentropic expansion from 4 to 5
- isobaric temperature increase in biomass combustor from 5 to 6
- isobaric cooling in high-temperature heat exchanger from 6 to 7

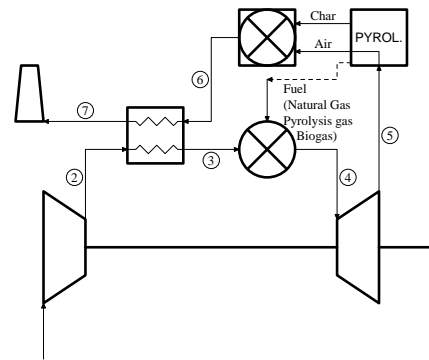


Figure 5: Flowsheet of an IFGT with supplementary firing

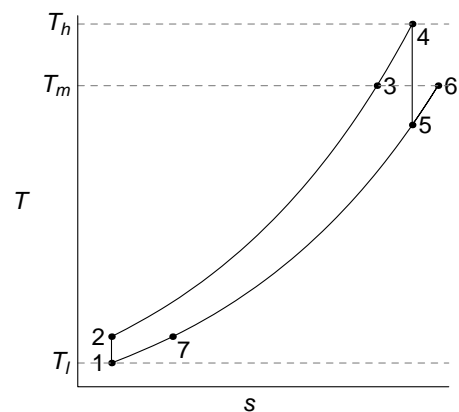


Figure 6: T - s -diagram for the Indirectly Fired Gas Turbine Cycle with supplementary firing

The net heat input to the cycle is:

$$\dot{Q}_i = \dot{Q}_{34} + \dot{Q}_{56} = \dot{m}c_p(T_4 - T_3) + \dot{m}c_p(T_6 - T_5) \quad (20)$$

The net heat rejected is:

$$\dot{Q}_o = \dot{Q}_{71} = \dot{m}c_p(T_1 - T_7) = \dot{m}c_p(T_1 - T_2) \quad (21)$$

By application of equation (3) it is found that:

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \quad (22)$$

$$\frac{T_5}{T_4} = \frac{1}{PR^{\frac{1-\kappa}{\kappa}}} \quad (23)$$

and thereby after setting $T_1 = T_l$, $T_3 = T_6 = T_m$ and $T_4 = T_h$

$$\begin{aligned} \dot{Q}_i &= \dot{m}c_p \left((T_h - T_m) + \left(T_m - T_h \frac{1}{PR^{\frac{\kappa-1}{\kappa}}} \right) \right) \\ &= \dot{m}c_p \left(T_h \left(1 - \frac{1}{PR^{\frac{\kappa-1}{\kappa}}} \right) \right) \end{aligned} \quad (24)$$

$$\dot{Q}_o = -\dot{m}c_p T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \quad (25)$$

This leads to a power output of:

$$\begin{aligned} \dot{W} &= \dot{Q}_i + \dot{Q}_o \\ &= \dot{m}c_p \left(T_h \left(1 - PR^{\frac{1-\kappa}{\kappa}} \right) - T_l \left(1 - PR^{\frac{\kappa-1}{\kappa}} \right) \right) \end{aligned} \quad (26)$$

The efficiency of the process is:

$$\eta = \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_h} \left(PR^{\frac{\kappa-1}{\kappa}} \right) \quad (27)$$

which is the same as for the recuperated cycle and the simple IFGT.

The efficiency of the IFBGT cycle, i.e., the biomass part of the supplementary fired cycle, with a combustion temperature of T_m is as in (18)

$$\eta = \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_m} PR^{\frac{\kappa-1}{\kappa}} \quad (28)$$

which means that the value of the marginal efficiency for this cycle also is unity which also means that the marginal efficiency obtained by supplementary firing and thereby raising the maximum temperature to T_h equals 1. The fuel added to get from T_m to T_h is given by:

$$\Delta \dot{Q}_i = \Delta \dot{W} = \dot{m}c_p(T_h - T_m) \left(1 - PR^{\frac{1-\kappa}{\kappa}} \right) \quad (29)$$

but, the amount of natural gas added is larger than the change in fuel consumption. It is:

$$\dot{Q}_{ng} = \dot{m}c_p(T_h - T_m) \quad (30)$$

This gives a marginal efficiency for the natural gas of:

$$\bar{\eta} = \frac{\Delta \dot{W}}{\dot{Q}_{ng}} = 1 - PR^{\frac{1-\kappa}{\kappa}} \quad (31)$$

It is observed that this is only dependent on the pressure, and thus independent of the temperatures in the cycle. Furthermore, the value is lower than the efficiency achieved by a simple recuperated gas turbine cycle working between T_l and T_h (see equation 9). The simple supplementary firing scheme therefore is not an advantage when trying to increase the marginal efficiency of natural gas. However, it does raise the efficiency of an IFBGT and may be acceptable for this reason.

It should be noted that if the biofuel is costless, e.g., a waste stream from an industrial plant, the total power production may be considered as an output from the natural gas consumption making the marginal efficiency on natural gas exceed unity. This is a further complication of matters, however, and is not discussed further.

Two alternatives present themselves:

- If the biofuel can be divided into two streams, one for indirect and one for direct firing, a high marginal efficiency on supplementary internal firing is reached. The separation of the fuel in two parts, a "clean" gaseous fuel for internal firing and a "dirty" residue for external firing may be accomplished by pyrolysis, or by thermal or biological gasification.
- Furthermore, if an amount of "clean" gaseous biofuel, equal to or greater than:

$$\Delta \dot{Q}_b = \dot{m}c_p(T_h - T_m) PR^{\frac{1-\kappa}{\kappa}} \quad (32)$$

is available without cost (see equations 28 and 29), it may be fired internally concurrently with the natural gas. Under the assumption of no cost of the biomass, the marginal efficiency of electric power produced by the natural gas may even exceed

unity. In order to realize such a cycle, a number of constraints on the temperatures in the process stages will have to be introduced. This option is however, highly dependent definitions of efficiency and assignment of cost to the different fuels.

DISCUSSION

In this paper we have considered the overall thermal efficiency (total fuel input to total power output) and the marginal efficiency (added fuel input to increased power output) only. However, in a cycle with more than one fuel input several alternative measures of efficiency may be applied, depending on which fuel is considered to be the basic input and how much of the produced power that is considered to be produced by each fuel. Thus, depending on cost of the different fuels and power alternative measures of quality may be preferred. In any case, the most important factor for the evaluation of an IFGT with or without supplementary firing will be an assessment of the overall economics of the installation. In future studies we will incorporate both economic aspects and component data for real gas turbines.

CONCLUSION

We have shown that, in the ideal case, a recuperated, an indirectly fired and an indirectly/supplementary-fired gas turbine will have the same efficiency. This is naturally not the case for a real application, but the analysis has provided a deeper insight into the paths to follow in order to find IFGT cycles with sufficiently high efficiency compared to alternative options for biomass applications.

The IFGT with supplementary firing may both be applied for achieving a higher total efficiency than possible with external firing only, and for achieving a high marginal efficiency with an expensive fuel. In both cases there are restrictions however. In the former case the supplementary firing and the basic firing have to be provided by the same biomass requiring pretreatment of the fuel for instance by pyrolysis, thermal gasification, or biogasification. In the latter case, part of the cheaper fuel has to be

burned internally, concurrently with the expensive fuel. Thus, a pretreatment of the cheap fuel is also demanded in this case.

The conclusion must, however, be that in view of the recent demonstration of two-stage gasification, the IFGT should be given more attention in the future research on biomass applications. The present conclusive observation is that the main hindrance for its commercialization, the need to develop high-temperature heat exchangers, may to some extent be compensated for by process modifications.

ACKNOWLEDGMENTS

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