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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. Intuitively, supplementary firing is expected to result in a high marginal efficiency. The paper shows that depending on the application, this is not always the case.

The interest in this cycle arises from a recent demonstration of the feasibility of a two-stage gasification process through construction of several plants. The gas from this process could be divided into two streams, one for primary and one for supplementary firing. 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. These results show that the combination of external firing and internal firing have the potential of reducing or solving some problems associated 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 appear 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. 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. Instead, other process changes may be considered in order to obtain a high marginal efficiency on natural gas. Two possibilities are discussed: 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, and combined internal biomass and natural gas firing.

Key words: gas turbine cycles, external firing, supplementary firing, biogas firing

1. 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 in-let 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 Wilson (1993). More recently, it has been studied by Riccio et al. (2000).

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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 last one, 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 (Biomass Gasification Group, Technical University of Denmark). 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 have constant specific heat. The various gas turbines are described by air standard cycles.

1.1. History

The indirectly fired gas turbine (IFGT) has been under consideration for a long time (Wilson, 1993; Most and Hagen, 1977). The driving force has almost exclusively been the possibility for using it for coal (Jahnig, 1986; LaHaye and Bary, 1994; Solomon et al., 1996), 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.

IFGTs 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 (Jahnig, 1986; LaHaye and Bary, 1994; Solomon et al., 1996; Edelmann and Stuhlmüller, 1997; Leithner and Ehlers, 2000; Ehlers and Leithner, 2002). One interesting reference on coal-fired IFGTs is the Ackeret-Keller-based plants, of which the oldest is the closed-cycle 2 MW plant, in operation since 1956 operating at a maximum temperature of 700°C (Bammert et al., 1956; Bammert, 1975). This is a low temperature for gas turbines in general, also for operation on coal, but it has been shown that operating temperatures in this range may still be interesting for micro gas turbines fueled with wet biomass (Elmegaard and Qvale, 2002).

The efficiency of open simple cycle IFGTs 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 (Ruyck et al., 1994) and a number of novel concepts have been subject to theoretical and experimental studies (Eidensten et al., 1996; Evans and Zaradic, 1996; Stevanovic, 2001).

1.2. 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 (Jørgensen, 1997; Olsen and Foldager, 1995) 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 (Elmegaard and Qvale, 2002; Elmegaard et al., 2001). 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.

2. 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,
- 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:

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

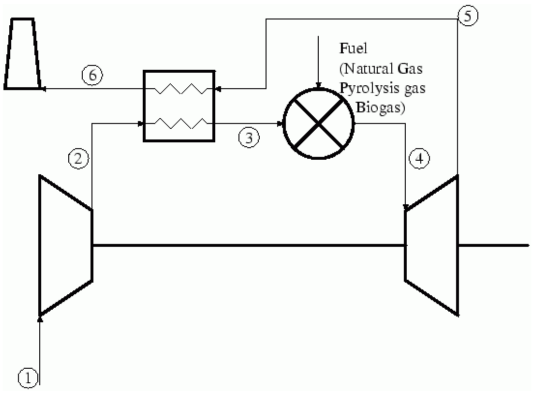

Figure 1. Flowsheet of a recuperated gas turbine

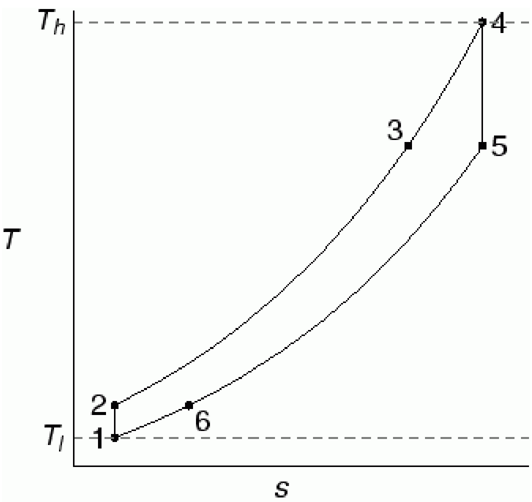

Figure 2. T-s-diagram for the recuperated gas turbine cycle

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

$$TR_{is} = PR^{\frac{\kappa-1}{\kappa}} \tag{3}$$

it is found that:

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \tag{4}$$

$$\frac{T_5}{T_4} = \frac{1}{PR^{\frac{\kappa-1}{\kappa}}} = PR^{\frac{1-\kappa}{\kappa}} \tag{5}$$

and thereby after setting $T_1 = T_l$ and $T_4 = T_h$, equations (1) and (2) can be written as:

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

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

By applying the First Law of Thermodynamics, this leads to a power output of:

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

The efficiency of the process is defined as the ratio between power output and heat input, and may be calculated from:

$$\eta \equiv \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_h} PR^{\frac{\kappa-1}{\kappa}} \tag{9}$$

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 the ratio between the partial derivatives of the power output, equation [8], and the heat input, equation [6], with respect to the maximum temperature, T_h:

$$\overline{\eta} = \frac{\dfrac{\partial \dot{W}}{\partial T_h}}{\dfrac{\partial \dot{Q}_i}{\partial T_h}} = \frac{\dot{m}\, c_p\, (1 - PR^{\frac{1-\kappa}{\kappa}})}{\dot{m}\, c_p\, (1 - PR^{\frac{1-\kappa}{\kappa}})} = 1 \tag{10}$$

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

For completeness of the description, it should be noted that recuperation is of interest only when there is a positive temperature difference between the turbine outlet and the compressor outlet. This restricts the pressure ratio to being below the value where these two temperatures, i.e., when T_2 and T_5, are equal. This sets an upper limit on the pressure ratio:

$$PR_{max} = \left(\sqrt{\frac{T_h}{T_l}} \right)^{\frac{\kappa}{\kappa-1}} \tag{11}$$

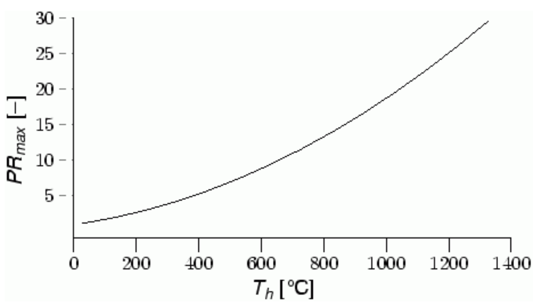

Figure 3. Upper limit of the pressure ratio for a gas turbine with recuperation (T_l = 298 K)

The relation between maximum cycle temperature and maximum pressure ratio is shown in *Figure 3*. The optimal pressure ratio will be significantly lower than this limit.

3. Indirectly Fired Gas Turbine

The flowsheet and the T-s-diagram of the indirectly fired gas turbine cycle are shown in *Figures 4* and *5*, 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.

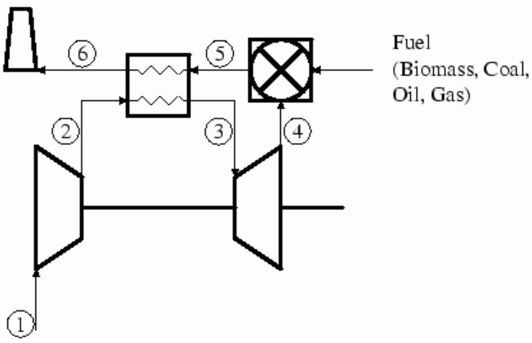

Figure 4. Flowsheet of an IFGT

It is seen that in the IFGT, the addition of heat to the working fluid by external combustion is occurring after the expansion, whereas in the recuperated cycle (*Figure 2*), the heat is added internally before expansion.

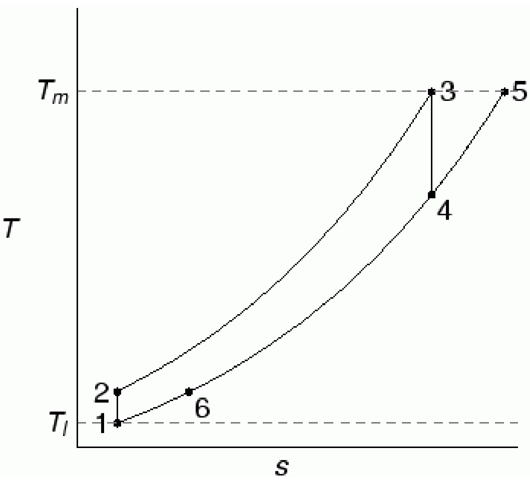

Figure 5. T-s-diagram for the indirectly fired gas turbine cycle

In this case, the net heat input to the cycle is:

$$\begin{aligned}\dot{Q}_i &= \dot{Q}_{45} = \dot{m}\,c_p(T_5 - T_4) \\ &= \dot{m}\,c_p\,(T_3 - T_4)\end{aligned} \tag{12}$$

and the net heat rejected is:

$$\begin{aligned}\dot{Q}_o &= \dot{Q}_{61} = \dot{m}\,c_p(T_1 - T_6) \\ &= \dot{m}\,c_p(T_1 - T_2)\end{aligned} \tag{13}$$

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

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \tag{14}$$

$$\frac{T_4}{T_3} = PR^{\frac{1-\kappa}{\kappa}} \tag{15}$$

and thereby after setting $T_1 = T_l$ and $T_3 = T_5 = T_m$, equations (12) and (13) yield:

$$\dot{Q}_i = \dot{m}\,c_p\,T_m\,(1 - PR^{\frac{1-\kappa}{\kappa}}) \tag{16}$$

$$\dot{Q}_o = \dot{m}\,c_p\,T_l\,(1 - PR^{\frac{\kappa-1}{\kappa}}) \tag{17}$$

The First Law of Thermodynamics leads to a power output of:

$$\begin{aligned}\dot{W} &= \dot{Q}_i + \dot{Q}_o \\ &= \dot{m}\,c_p\left(T_m\,(1 - PR^{\frac{1-\kappa}{\kappa}}) + T_l(1 - PR^{\frac{\kappa-1}{\kappa}})\right)\end{aligned} \tag{18}$$

Thus, the efficiency of the process is:

$$\eta = \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_m} PR^{\frac{\kappa-1}{\kappa}} \tag{19}$$

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

$$\overline{\eta} = \frac{\frac{\partial \dot{W}}{\partial T_m}}{\frac{\partial \dot{Q}_i}{\partial T_m}} = \frac{\dot{m}\,c_p\,(1 - PR^{\frac{1-\kappa}{\kappa}})}{\dot{m}\,c_p\,(1 - PR^{\frac{1-\kappa}{\kappa}})} = 1 \tag{20}$$

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

4. 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 may 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, and usually this is 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 favorable.

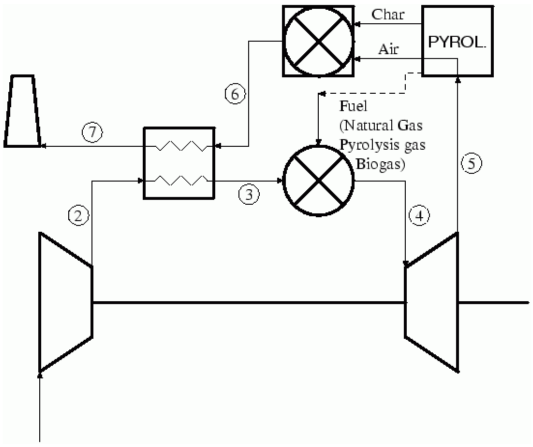

Figure 6. Flowsheet of an IFGT with supplementary firing

Secondly, the introduction of heat into the IFGT is achieved in a high temperature heat exchanger. Several studies (Kumada, 1999; Luzzatto et al., 1997; Meunier, 1991; Orozco, 1993) 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 in *Figures 6* and *7*, 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.

From the T-s-diagram it is seen that in this case addition of heat is carried out both before and after the expansion, allowing the turbine inlet temperature to be raised above the limit prescribed by the heat exchanger material.

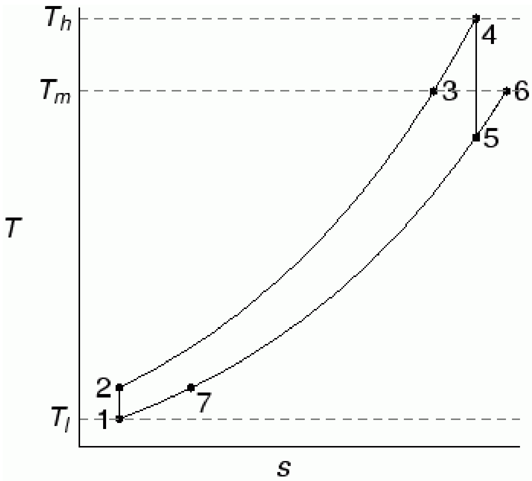

Figure 7. T-s-diagram for the indirectly fired gas turbine cycle with supplementary firing

The net heat input to the cycle is in this case:

$$\dot{Q}_i = \dot{Q}_{34} + \dot{Q}_{56} = \dot{m}\,c_p\,(T_4 - T_3) + (\dot{m}\,c_p\,(T_6 - T_5) \quad (21)$$

The net heat rejected is:

$$\dot{Q}_o = \dot{Q}_{71} = \dot{m}\,c_p\,(T_1 - T_7) = \dot{m}\,c_p\,(T_1 - T_2) \quad (22)$$

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

$$\frac{T_2}{T_1} = PR^{\frac{\kappa-1}{\kappa}} \quad (23)$$

$$\frac{T_5}{T_4} = PR^{\frac{1-\kappa}{\kappa}} \quad (24)$$

and after introducing the conditions $T_1 = T_l$, $T_3 = T_6 = T_m$ and $T_4 = T_h$ into equations (21) and (22), the expression for heat input and heat rejection are calculated:

$$\begin{aligned}\dot{Q}_i &= \dot{m}\,c_p\left((T_h - T_m) + (T_m - T_h\,PR^{\frac{1-\kappa}{\kappa}})\right) \\ &= \dot{m}\,c_p\left(T_h\,(1 - PR^{\frac{1-\kappa}{\kappa}})\right)\end{aligned} \quad (25)$$

$$\dot{Q}_o = \dot{m}\,c_p\,T_l\,(1 - PR^{\frac{\kappa-1}{\kappa}}) \quad (26)$$

These are equal to the expressions that were found for the recuperated cycle.

This leads to a power output of:

$$\dot{W} = \dot{Q}_i + \dot{Q}_o$$

$$= \dot{m}\, c_p \left(T_h \,(1 - PR^{\frac{1-\kappa}{\kappa}}) + T_l\,(1 - PR^{\frac{\kappa-1}{\kappa}}) \right) \quad (27)$$

and thus the efficiency of the process is:

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

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 equation (19):

$$\eta = \frac{\dot{W}}{\dot{Q}_i} = 1 - \frac{T_l}{T_m} (PR^{\frac{\kappa-1}{\kappa}}) \quad (29)$$

which means that the value of the marginal efficiency for this cycle also is unity which furthermore 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 , ΔQ_i can be calculated by subtracting equation (16) from equation (25), and the increase in power output, ΔW, can be found by subtracting equation (18) from equation (27). It is found that both have the same values expressed by:

$$\Delta\dot{Q}_i = \Delta\dot{W} = \dot{m}\, c_p\, (T_h - T_m)(1 - PR^{\frac{1-\kappa}{\kappa}}) \quad (30)$$

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

$$\dot{Q}_{ng} = \dot{m}\, c_p\, (T_h - T_m) \quad (31)$$

This gives a marginal efficiency for natural gas of:

$$\overline{\eta} = \frac{\Delta\dot{W}}{\dot{Q}_{ng}} = 1 - PR^{\frac{1-\kappa}{\kappa}} \quad (32)$$

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 IBFGT and may be acceptable for this reason.

5. Discussion

5.1. Economic Considerations

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 of 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. However, this will require that the conversion to gas to take place under high pressure, or that the gas at high temperature be compressed after the conversion. Furthermore, cleaning of the gas probably will be required. All these issues will undoubtedly penalize the efficiency of the process.
- Furthermore, if an amount of “clean” gaseous biofuel, equal to or greater than:

  $$\Delta\dot{Q}_b = \dot{m}\, c_p\, (T_h - T_m)\, PR^{\frac{1-\kappa}{\kappa}} \quad (33)$$

  (found by subtracting Eq. (31) from Eq. (30)) is available without cost, 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 on definitions of efficiency and assignment of cost to the different fuels.

5.2. Definition of efficiency

In this paper we have considered the overall thermal efficiency (total work output to total heat input) and the marginal efficiency (increased work output to added heat input) 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 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.

6. 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. Moreover, the marginal efficiency of additional firing will be unity. However, if the aim is to utilize two fuels and reach a high marginal efficiency on the more expensive one, the results of the paper show that for the ideal IBFGT cycle, this is not beneficial. Supplementary natural gas firing has a marginal efficiency below the efficiency of a simple recuperated cycle. This may seem counterintuitive, but such a process will require more natural gas to be used than the obtained additional power output.

Further details will be obtained in future studies of non-ideal cycles. Probably the results will show that efficiencies of the different cycles will be different, but the present analysis has provided an insight into the paths to follow in order to identify IFGT cycles with the potential to compete with the presently available candidates for biomass applications.

Supplementary firing of the IFGT may be applied both 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 last 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 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.

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]
TR	Temperature ratio [-]
T	Temperature [K]
T_h	Maximum temperature for internal combustion [K]
T_l	Ambient temperature [K]
T_m	Maximum temperature for external combustion [K]
$\dot{W}$	Power [kW]
$\overline{\eta}$	Marginal efficiency [-]
η	Efficiency [-]

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