A computer-based model for gas-turbine power augmentation by inlet-air cooling and water/steam injection

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The paper presents a computer-based thermodynamic model that estimates the effects of power augmentation methods on the gas turbine power and heat rate. The model takes into consideration the effect of evaporative or refrigerative inlet-air cooling and allows for water or steam injection into the combustion chamber. The formulation of the model improves upon the conventional "standard-air" analysis by including the effects of water on the working-fluid properties, by accounting for the mass of fuel, and by adopting the exact variable specific-heat approach. The model also improves the accuracy and extends the scope of the analysis by solving an energy equation for the combustion process, which enables either the combustion temperature, or the required rate of fuel consumption, to be specified and the other to be calculated. The paper describes the thermodynamic model and the computer program based on it. Verification of the computer-based model against relevant published data is also presented and discussed.

Key words: power generation, gas turbine, inlet-air cooling, power aumentation, computer model

1. Introduction

Gas turbines have become the favoured power generation machines in many parts of the world. Standing alone in open-cycle, gas turbines are used to meet the peak-loads and combined with steam turbines in combined cycles they are used as efficient base-load machines. The easy-to-start capability, fuel-flexibility, low-fuel cost, and low environmental pollution offer the gas-turbine clear advantages over other power-generation machines. Unfortunately, the performance of these machines is greatly degraded by adverse ambient conditions due to the high air temperatures and dusty environment. Being a constant volume-flow machine,

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the power of the gas turbine is directly proportional to the mass flow rate of air through the turbine. A high ambient temperature (or humidity) reduces the air density and, accordingly, the mass-flow rate through the gas turbine. Therefore, the turbines loose significant portions of their design generation capacity at standard ISO condition (15 ℃) when operating in hot and humid climate. The gas turbinés output typically decreases by 10 % to 18 % for every 10 K of increase in inlet air temperature [1, 2]. A high ambient temperature also increases their heat rate and, according to McCracken [2], gas turbines produce 25–35 % less power in summer than in winter at 5–10 % higher heat rate, which means an average increase of 6 % in fuel consumption.

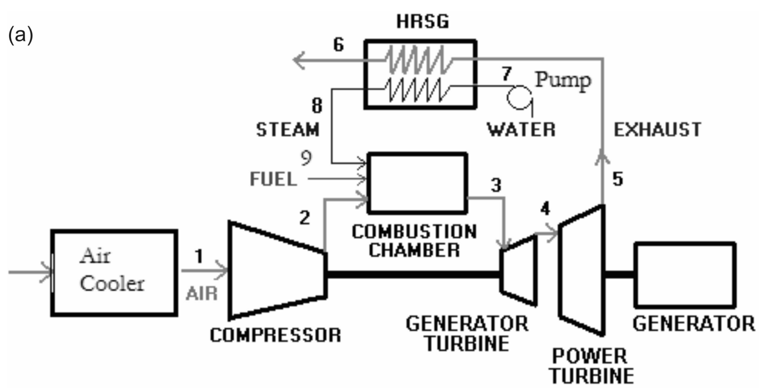
Power producers apply various power augmentation methods in order to compensate for the effect of high temperatures on the gas-turbine output [2–7]. These methods include refrigerative or evaporative inlet-air cooling. The refrigerative cooling can be of the mechanical or the absorption type and can be applied with or without thermal energy storage [3–5]. The evaporative cooling can be achieved by a conventional media-type method or by fogging [6, 7]. By applying inlet-air cooling, additional megawatts can be obtained from existing gas turbines at a fraction of the cost of installing new generation plants. According to Boyce [8], a new power plant costs 500 $ per kilowatt while inlet air-cooling costs 110–120 $ per kilowatt of augmented power. Besides inlet-air cooling, power-augmentation can also be achieved by injecting water or steam into the combustion chamber [9–11]. Unfortunately, water injection, which is usually used for NO_x control, increases the heat rate of the plant and, therefore, the generation cost [11]. The problem can be solved by injecting steam, rather than liquid water. Since the flue gases are usually exhausted at a temperature exceeding 500 ℃, the generation of steam in a heat-recovery steam generator enables some of this lost energy to be recovered and the plant efficiency to be improved.

The required initial investments and running costs of the different power augmentation systems vary considerably [1, 8]. The suitability and economical feasibility of power augmentation methods depend strongly on the existing power--plant equipment and layout, the ambient atmospheric conditions, the effect of the method on the turbine power and heat rate, and the electricity-consumption pattern. Therefore, a given power augmentation method may have certain advantages or disadvantages, compared to other methods, when a particular gas-turbine power plant is considered. The present work is part of an on-going effort to develop a computer-based model that estimates the effect of the different power augmentation methods on the turbine output. The model, which takes into consideration the effect of inlet air cooling on the turbine power and heat rate and allows for water or steam injection into the combustion chamber, can assist in the selection of the suitable power augmentation method. Sections 2 and 3 of this paper describe the analytical thermodynamic model and the computer program based on it while Section 4 deals with the model verification against published data.

2. The thermodynamic model

Figure 1a,b shows the schematic diagram of a two-shaft gas-turbine plant together with the cycle T-s diagram. The system consists of an axial-flow air compressor followed by the combustion chamber and the two-shaft gas turbine. In the present model, it is assumed that the gas exhausted by the gas-generator (i.e. the high-pressure) turbine is passed directly to the power (i.e. the low-pressure) turbine without reheating. An air-cooler cools the air inlet to the compressor and, therefore, increases its density and mass flow rate through the plant. The air--cooler can be of the evaporative or refrigerative type. Before discharged to the atmosphere, the flue gases pass through a heat-recovery steam generator (HRSG), which uses the heat of the flue gases leaving the turbine to produce steam. Some of this steam can be injected into the combustion chamber for power augmentation and NO_x reduction. A pump is needed to raise the water pressure before the HRSG to a suitable value.

Standard textbooks of thermodynamics usually analyse the gas-turbine cycle by adopting the "standard-air" method, according to which the working fluid is



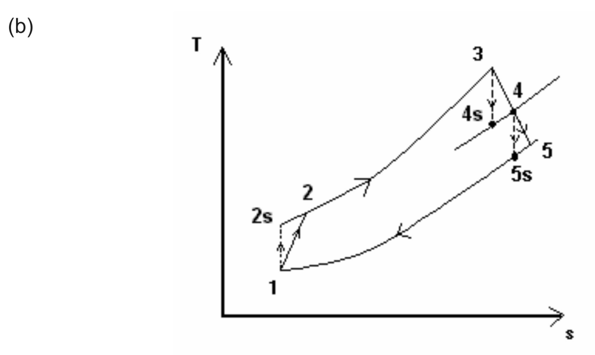


Fig. 1. The steam-injected gas-turbine plant (a) and the cycle (b).

treated as pure air whose specific heats are evaluated at some average temperature or even kept constant at their room-temperature values [12, 13]. Moreover, the effect of the fuel mass on the turbine work is completely ignored. These assumptions allow an "approximate method" of analysis to be applied, which greatly simplifies the analysis. However, the results of the approximate method are known to deviate from the exact values by significant margins [13]. The present thermodynamic model includes the effects of water on the working-fluid properties, adopts the exact variable specific-heat approach [13] and accounts for the fuel mass. By solving an energy equation for the combustion process, the model extends the analysis scope since this enables either the combustion temperature or the required rate of fuel consumption to be specified and the other calculated. In order to simplify the model, pressure and heat losses are ignored anywhere in the cycle.

2.1 The air compressor

Given the inlet-air temperature (T_1) and humidity (ω_1), the enthalpy of humid air (per kg of dry air) at the compressor inlet (h_1) is determined from:

$$h_1 = h_{1\mathrm{da}} + \omega_1 h_{1\mathrm{w}}, \tag{1a}$$

where $h_{1\mathrm{da}}$ and $h_{1\mathrm{w}}$ are the specific enthalpies of dry air and water-vapour, respectively. The enthalpy of humid air after an ideal isentropic compression ($h_{2\mathrm{s}}$) is determined using the compressor pressure ratio (P_{rc}) and the relative pressure (P_{r}) as follows [12]:

$$T_1 \to P_{\mathrm{r}1}; \qquad P_{\mathrm{r}2} = P_{\mathrm{r}1} \times P_{\mathrm{rc}}; \qquad P_{\mathrm{r}2} \to T_{2\mathrm{s}} \to h_{2\mathrm{s}}. \tag{1b}$$

Note that an iterative method is needed to find $T_{2\mathrm{s}}$ from $P_{\mathrm{r}2}$ of the humid air. The actual enthalpy after compression (h_2) is calculated from the isentropic head as follows:

$$h_2 = h_1 + \frac{h_{2\mathrm{s}} - h_1}{\eta_{\mathrm{c}}}, \tag{1c}$$

where η_{c} is the compressor isentropic efficiency.

The input of the compressor ($\dot{W}_{\mathrm{c}}$) is calculated from the mass flow rate and air enthalpy increase within the compressor as follows:

$$\dot{m}_{\mathrm{ha}} = (1+\omega)\dot{m}_{\mathrm{da}}; \tag{1d}$$

$$\dot{W}_{\mathrm{c}} = \dot{m}_{\mathrm{ha}}(h_2 - h_1), \tag{1e}$$

where $(\dot{m}_{\mathrm{ha}})$ and $(\dot{m}_{\mathrm{da}})$ are the mass flow rates of humid air and dry air, respectively. The compressor discharge temperature (T_2) is found from the discharge enthalpy of humid air (h_2) by iteration.

2.2 The combustion process

The presence of excess air into the combustion chamber, which is usually the case with gas turbines, assures complete combustion of the hydrocarbon fuel. Neglecting kinetic energy and potential energy changes between inlet and exit of the combustion chamber, the energy balance over an adiabatic chamber is given by:

$$\begin{aligned} &\left(1 \times H_{\mathrm{f}}^{0}\right) + \left(n\widetilde{h}_2\right)_{\mathrm{da}} + \left(n\widetilde{h}_2\right)_{\mathrm{w}} + \left(n\widetilde{h}_8\right)_{\mathrm{wi}} \\ &\quad = \left(n\widetilde{h}_3\right)_{\mathrm{w}} + \left(n\widetilde{h}_3\right)_{\mathrm{wi}} + \left(n\widetilde{h}_3\right)_{\mathrm{H_2O}} + \left(n\widetilde{h}_3\right)_{\mathrm{CO_2}} \\ &\quad + \left(n\widetilde{h}_3\right)_{\mathrm{excess\ da}} - \left(n\widetilde{h}_3\right)_{\mathrm{O_2}}, \end{aligned} \tag{2a}$$

where H_{f}^{0} is the molar heat (enthalpy) of formation of the hydrocarbon gas-fuel, n is the number of moles of the reactant or product of combustion per mole of the fuel, and $\widetilde{h}$ is the molar enthalpy. The suffices H_2O and CO_2 refer to the two products of combustion (water and carbon dioxide) and O_2 refers to oxygen. The suffices da, w and wi refer to the dry air, air-borne moisture, and injected water/steam, respectively. Suffices 2 and 3 refer to the conditions (i.e. temperatures) before and after the combustion chamber. The number of oxygen-moles appearing in the product side is obtained from:

$$(n)_{\mathrm{O_2}} = (n)_{\mathrm{CO_2}} + \frac{1}{2}(n)_{\mathrm{H_2O}}. \tag{2b}$$

Equation (2) is used to find T_3 when the fuel-air ratio is specified, or to find the fuel-air ratio (mass of fuel per unit mass of dry air) when the combustion temperature (T_3) is specified.

2.3 The gas-generator turbine

The enthalpy after the gas-generator turbine (h_4) is obtained by equating the gas-generator turbine work with the compressor work. The discharge temperature (T_4) is then obtained from h_4 by iteration:

$$h_4 = h_3 - \frac{\dot{W}_c}{\dot{m}_{ha}}; \qquad h_4 \rightarrow T_4. \tag{3a}$$

The enthalpy (h_{4s}) and temperature (T_{4s}) after an ideal, isentropic expansion are then obtained using the generator turbine efficiency (η_{gt}):

$$h_{4s} = h_3 - \frac{h_3 - h_4}{\eta_{gt}}; \qquad h_{4s} \rightarrow T_{4s}. \tag{3b}$$

The pressure at the outlet of the gas-generator turbine is determined from the known temperatures T_3 and T_{4s} using the isentropic expansion relationship:

$$T_3 \rightarrow P_{r3}; \qquad T_{4s} \rightarrow P_{r4}; \qquad P_4 = P_3 \times \frac{P_{r4}}{P_{r3}}. \tag{3c}$$

2.4 Power turbine and thermal efficiency

Assuming the discharge pressure from the power turbine (P_5) to be the same as the inlet pressure P_1, the enthalpy after an ideal, isentropic expansion (h_{5s}) is determined from the known temperature (T_4) and pressure ratio (P_5/P_4) as follows:

$$T_4 \rightarrow P_{r4}; \qquad P_{r5} = P_{r4} \times \frac{P_5}{P_4}; \qquad P_{r5} \rightarrow T_{5s} \rightarrow h_{5s}. \tag{4a}$$

The actual enthalpy at the outlet of the power turbine (h_5) is then calculated from the isentropic efficiency of the turbine (η_{pt}) and enthalpy difference ($h_4 - h_{5s}$):

$$h_5 = h_4 - (h_4 - h_{5s}) \times \eta_{pt}. \tag{4b}$$

The net work of the gas turbine per unit mass flow rate of dry air is equal to that of the power turbine ($\dot{W}_{pt}$), which is calculated as follows:

$$\dot{W}_{pt} = (\dot{m}_{ha} + \dot{m}_{fuel} + \dot{m}_{wi})(h_4 - h_5), \tag{4c}$$

where $\dot{m}_{fuel}$ is the fuel consumption and $\dot{m}_{wi}$ is the rate of water/steam injection per unit mass of dry air. The heat energy of the fuel ($\dot{Q}_{in}$) and thermal efficiency (η_{th}) are then obtained from:

$$\dot{Q}_{in} = \dot{m}_{fuel} \times LHV; \tag{4d}$$

$$\eta = \frac{\dot{W}_{\mathrm{pt}}}{\dot{Q}_{\mathrm{in}}}, \tag{4e}$$

where LHV is the fuel lower heating value. Finally, the exhaust-gas temperature (T_5) is obtained from the calculated enthalpy (h_5) of the gas mixture by iteration and the energy balance over the HRSG is given by:

$$(\dot{m}_{\mathrm{ha}} + \dot{m}_{\mathrm{fuel}} + \dot{m}_{\mathrm{wi}})(h_5 - h_6) = \dot{m}_{\mathrm{wi}}(h_8 - h_7). \tag{5}$$

3. The computer program

A computer program has been developed based on the above thermodynamic model to analyse the gas-turbine cycle with power augmentation by inlet-air cooling and/or water/steam injection. Developed in FORTRAN, the program initially reads the basic data of the gas turbine (total pressure ratio, compressor and turbine efficiencies, combustion temperature, and rate of fuel injection) and requires the user to specify a reference atmospheric condition as described by the inlet values of the pressure, temperature and humidity. It then asks the user about the actual inlet-air condition and the method of power augmentation to be considered. If inlet-air cooling is to be used, the program determines the dew-point and wet-pulp temperatures and informs the user with their values since these two temperatures determine the limiting values for refrigerative and evaporative cooling, respectively. If water/steam is to be injected, the program asks about the temperature of water/steam to be injected in order to calculate the maximum rate of injection that saturates the air at the compressor discharge. It then asks the user to give the required rate of water/steam injection, which should be less than the maximum value, before proceeding to calculate, from the relationships described in the preceding section, the compressor work, combustion temperature (or fuel rate), gas-generator turbine work and exit pressure, power turbine work and exit temperature, and thermal efficiency of the plant. The important details of the computer program are given below.

3.1 The reference atmospheric condition and reference volume flow rate

The mass flow rate through the gas turbine at a particular site depends on the air density, which depends on the atmospheric pressure, ambient temperature, and humidity. To be able to compare the gas-turbine performance at different atmospheric or inlet-air conditions, the present computer model analyses the cycle on the basis of volume flow rate rather than mass flow rate. A reference volume flow rate ($\dot{V}_{\mathrm{R}}$) is defined as the volume that contains a unit mass flow rate ($1\ \mathrm{kg \cdot s^{-1}}$) of dry air at a chosen "reference" inlet-air condition. The reference inlet-air condition can be the design condition or the ISO standard condition of 101.325 kPa, 15 ℃, and 60 % humidity. But it can also be a certain atmospheric conditions at the gas-turbine site, say in a winter day. At atmospheric or inlet-air conditions which are different from the reference condition, the reference volume will contain different mass flow rates of dry air and moisture. Applying thermodynamic relations for a mixture of two ideal gases (dry air and water-vapour), the reference volume flow rate is given by [12]:

$$\dot{V}_{\mathrm{R}} = \frac{R_{\mathrm{a}} T_{\mathrm{R}}}{P_{\mathrm{R,da}}} = \frac{R_{\mathrm{a}} T_{\mathrm{R}}}{-\dfrac{\phi_{\mathrm{R}} P_{\mathrm{sat}}}{100}}, \tag{6}$$

where R_{a} is the gas constant for dry air. T_{R}, P_{R}, ϕ_{R}, and $P_{\mathrm{R,da}}$ are, respectively, values of the pressure, temperature, relative humidity (%), and partial pressure of dry air at the reference atmospheric condition. Psat is the saturation pressure for water at the reference temperature. The total mass flow rate of humid air (dry air + moisture) entering the compressor through the reference volume at the reference condition is equal to $(1 + \omega_{\mathrm{R}})$ kg · s^{-1}, where ω_{R} is the absolute humidity (or humidity ratio) of air at the reference condition.

3.2 Actual mass flow rate of air through the compressor

At atmospheric condition other than the reference condition, the total mass flow rate of dry air through the reference volume is calculated from the given values of atmospheric pressure (P_{atm}) and dry-bulb temperature (T_{atm}) as follows:

$$\dot{m}_{\mathrm{da}} = \frac{P_{\mathrm{da}}}{R_{\mathrm{a}} T_{\mathrm{atm}}} \dot{V}_{\mathrm{R}}. \tag{7a}$$

Using Eq. (6), this becomes:

$$\dot{m}_{\mathrm{da}} = \frac{P_{\mathrm{da}} T_{\mathrm{R}}}{P_{\mathrm{R,da}} T_{\mathrm{atm}}}. \tag{7b}$$

The total mass flow rate of humid air ($\dot{m}_{\mathrm{ha}}$) is then given by:

$$\dot{m}_{\mathrm{ha}} = (1 + \omega_{\mathrm{atm}}) \dot{m}_{\mathrm{da}}. \tag{7c}$$

The specific humidity at actual atmospheric condition (ω_{atm}) is either given or determined from the relative humidity (ϕ_{atm}) according to the known thermodynamic relations for a mixture of air-water vapour mixture [12].

3.3 Effect of water/steam injection on the combustion temperature

The injection of liquid water or steam into the combustion chamber causes the combustion temperature to drop unless more fuel is added. Two options have been considered and included in the program:

1. to maintain the combustion temperature at its value prior to water/steam injection,

2. to keep the fuel consumption constant and allow the combustion temperature to drop.

The amount of additional fuel required to maintain the combustion temperature equal to that prior to water/steam injection depends on the temperature and injection rate of water/steam. More fuel is consumed when the injection rate of water/steam increases and/or when its temperature decreases. The required fuel flow rate is obtained from the energy equation (Eq. 2). If no fuel is supplied in addition to that prior to water/steam injection and the combustion temperature is allowed to drop, Eq. (2) is solved to give the new combustion temperature. In this case, the work from air and the combustion products will be reduced as a result of the reduced temperature and enthalpy at the turbine inlet. However, additional work is provided by the injected water/steam, which increases the mass flow rate in the turbine. Therefore, both the turbine power output and thermal efficiency depend on the additional work of the injected steam or water.

3.4 Modelling the working fluid

The thermodynamic model described above requires values of the enthalpy (h) and relative pressure (P_{r}) for the working fluid, which consists of two gases or more. The computer program evaluates the required thermodynamic properties by treating the pre-combustion working fluid as a mixture of two ideal gases, viz. dry air and water vapour. After combustion the working fluid consists of these two components plus the products of combustion (CO_2, N_2 and H_2O). If the temperature is known, the thermodynamic properties of the pro-combustion working fluid can also be found easily by treating it as a mixture of ideal gases. However, if an inverse solution is required (e.g. finding T from a known value of h) it will be difficult to perform an iterative solution for more than two components (see section 3.6 below). Therefore, the pro-combustion working fluid is also treated as a mixture of dry air and water vapour, but the program offers the user two options for modelling the water vapour part:

1. Model 1: all products of combustion – including the combustion water – are treated as dry air. Thus, the water vapour part includes the original air-borne moisture plus the water vapour that results from the injected water/steam.

2. Model 2: the combustion water is also included into the water vapour component. Thus, the dry air component includes only CO_2 and N_2.

For the purpose of code verification and comparison with available data, the program offers a third option that treats all the working fluid as dry air. This option, called Model 0, is similar to the air-equivalent method of Bathie [13], who also used variable specific heats but did not take into consideration the effect of inlet-air humidity.

3.5 Thermodynamic properties of the working fluid

The computer program obtains the thermodynamic properties of the working-fluid components using various relations. Most of these relations were adopted from the published literature [12, 13], but some relationships have been derived directly from thermodynamic property tables. Properties of the dry air, water vapour, products of combustion, and liquid water are determined from the following relations:

Air:

Treated as an ideal gas, the enthalpy of air depends on the temperature alone and is obtained from the following relationship [12]:

$$h = R_{\mathrm{a}} \times \left\{ a \times T + \frac{b}{2} \times T^2 + \frac{c}{3} \times T^3 + \frac{d}{4} \times T^4 + \frac{e}{5} \times T^5 \right\} - 302.67, \qquad [\mathrm{kJ \cdot kg^{-1}}], \tag{8a}$$

where a, b, c, d, and e are constants and T is the absolute temperature in Kelvin. The constant -302.67 is added to the original equation given by Moran and Shapiro [12] in order to unify their reference temperature (originally taken as 0.0 K) with that of Bathie [13], taken as 298.15 K or 25 ℃.

The relative pressure is also obtained from a polynomial in temperature. For the temperature range 500–1000 K, this is given by the following 4$^{\text{th}}$ order polynomial:

$$P_{\mathrm{r}} = a + b \times T + c \times T^2 + d \times T^3 + e \times T^4, \qquad [\mathrm{kJ \cdot kg^{-1}}], \tag{8b}$$

where the constants a, b, c, d, and e have the values of 8.2488968, -0.060512094, 0.00016927346, $-1.8744272 \times 10^{-7}$ and $1.8445728 \times 10^{-10}$, respectively. For temperatures higher than 1000 K, the 4$^{\text{th}}$ order polynomial requires a different set of coefficients. For temperatures lower than 500 K, a 10$^{\text{th}}$ order polynomial is used due to the rapid change with temperature.

Water vapour and products of combustion:

The enthalpies of the water vapour and products of combustion (CO_2 and O_2), which also depend on the temperature alone, are obtained from the following relation [12]:

$$h = R_{\mathrm{u}} \times T \times \left\{ a + \frac{b}{2} \times T + \frac{c}{3} \times T^2 + \frac{d}{4} \times T^3 + \frac{e}{5} \times T^4 + \frac{f}{T} \right\}, \qquad [\mathrm{kJ \cdot kmol^{-1}}], \tag{9}$$

where R_u is the universal ideal gas constant and T is the absolute temperature in Kelvin. The values of the constants a, b, c, d, e, and f are obtained by fitting polynomials to the tabulated values of h for each gas/vapour. For a better accuracy, two sets of these constants are used depending on the temperature level, the levels being 300–1000 K and 1000–5000 K. The model also requires the relative pressure for the air-water vapour mixture. This is obtained by fitting 4th order polynomials to the required data of the air and water vapour with different coefficients for three temperature levels: less than 500 K, 500–1000 K and 1000–1500 K.

Liquid water:

Two thermodynamic properties of liquid water are required in the model, which are the enthalpy (h) and saturation pressure at a given temperature (P_{sat}). The enthalpy of water at the injection temperature is approximated by that of saturated liquid at the given temperature (h_f), which is obtained from the following relationship:

$$h \approx h_f = h_{steam} - h_{fg}, \qquad [\mathrm{kJ \cdot kg^{-1}}], \tag{10a}$$

where h_{steam} is the value of enthalpy for water vapour as obtained from Eq. (9) and h_{fg} is the latent heat of vaporisation $[\mathrm{kJ \cdot kg^{-1}}]$ at the given temperature obtained from the following third-order polynomial:

$$h_{fg} = a + b \times T + c \times T^2 + d \times T^3, \qquad [\mathrm{kJ \cdot kg^{-1}}], \tag{10b}$$

in which T is in ℃ and values of the coefficients a, b, c, and d are obtained by fitting a polynomial to the tabulated values of h_{fg} for water [12]. The respective values of these constants are 2501.3687, -2.3680605, 0.00056653491 and $-1.3212214 \times 10^{-5}$. The polynomial is applicable in the range of temperatures 5–100 ℃.

The saturation pressure of water (P_{sat}) at a given temperature is obtained from the following relation [12]:

$$\ln(P_{sat}) = 70.4346943 - \frac{7362.6981}{T} + 0.006952085T - 9.0\ln(T), \tag{10c}$$

where T is in Kelvin and P_{sat} is in atmospheres. The relationship is applicable for temperatures in the range 0–200 ℃.

3.6 The iterative solution procedure

In all the thermodynamic property relations given above, the properties are functions of the temperature. When the value of a given property (say h) is known and the corresponding temperature is to be determined, an inverse solution of these relations is required which the present computer program obtains by an iterative procedure. This iterative solution is particularly needed since the working fluid in the present model is a mixture of gases and not simply dry air. First, a value of the temperature (T_{guess}) is assumed, which must be lower than the expected temperature (if a better guess cannot be made, this initial guess is made equal to 273 K, or 0 ℃). The calculated value of the required property at the guessed temperature (say h_{guess}) is then compared with the known value of the property (h_{known}).

Since the thermodynamic properties increase with temperature, the calculated property value will be lower than the required value. The temperature is then increased by a reasonably large increment ΔT (typically 10 K) and h_{guess} is recalculated and compared with h_{known}. If h_{guess} is still less than h_{known}, the temperature is increased again by ΔT. This is repeated until h_{guess} exceeds h_{known}. Once this happens, the increment ΔT is reduced (typically by a factor of 10) and the last step of the iterative solution is repeated using the smaller ΔT. The increment ΔT is reduced until a prescribed criterion $\varepsilon = (h_{\text{guess}} - h_{\text{known}})/h_{\text{known}}$ is met at which the value of T_{guess} is taken as the required solution for T. A similar iterative solution is applied when the rate of fuel combustion is fixed and the combustion temperature is to be determined from the energy balance equation (Eq. 2).

4. Validation of the model

In order to verify the computer program with its property relationships, it was initially tested against the solutions given by Bathie [13] for the following two-shaft gas turbine:

Inlet air pressure = 101.35 kPa
Inlet air temperature = 15 ℃
Inlet air humidity = 0.0 % (dry air)
Compressor pressure ratio = 12 : 1
Fuel = n-octane (C_8H_{18})
Fuel mass flow rate at rated load = 0.0214773 $\text{kg} \cdot \text{kg}^{-1}$ dry air
Flue gas temperature at generator-turbine inlet (at rated load) = 1400 ℃
Flue gas temperature at power-turbine exit (at rated load) = 497.63 ℃
Compressor efficiency = 87 %
Gas-generator and power turbines efficiency = 89 %

Bathie [13] analysed the cycle of the above gas turbine with and without steam injection. Table 1 shows seven primary quantities in the cycle without steam injection as computed by the present computer model compared to their respective values obtained from Bathie [13]. The table shows three solutions for the present model (M0, M1 and M2), which represent the different options given by the computer program for modelling the working fluid as discussed in section 3.4. Bathie

Table 1. Comparison of the main cycle parameters as computed by the present computer program with those given by Bathie [13]

Key Parameters	Bathie [13]			Present model		
	SA	AE	AF	M0	M1	M2
Compressor work [$kJ \cdot kg^{-1}$]	343.0	343.0	343.0	344.78	344.79	344.79
T_2 [K]	623.0	623.0	623.0	622.92	622.90	622.90
T_4 [K]	1109.0	1115.8	1122.2	1114.50	1114.50	1123.70
P_4 [kPa]	408.2	418.5	422.8	419.50	419.49	432.85
T_5 [K]	815.0	815.0	831.1	814.31	814.30	818.67
Power turbine work [$kJ \cdot kg^{-1}$]	333.0	347.5	355.6	346.14	346.15	362.69
Thermal efficiency [%]	37.7	36.4	37.2	36.30	36.30	38.03

[13] also gave three solutions using standard-air (SA), air-equivalent (AE) and actual fluid (AF) assumptions. As the table shows, the values of the two parameters prior to combustion (the compressor work and T_2) as obtained by the present model are close to one another and agree well with the three solutions given by Bathie [13] who also used variable specific heats but did not take into consideration the effect of air humidity. Therefore, the present agreement between all six solutions for the first two parameters is anticipated since the inlet air humidity is zero and the inlet air is assumed to be dry.

The different solution methods gave different solutions for the five parameters after combustion. The figures on the table show that both Model 0 (air-equivalent option) and Model 1 of the present model yield values for the five parameters, which are close to the air-equivalent (AE) solution given by Bathie [13]. The agreement between Model 0 and the AE solution of Bathie [13] is anticipated but the agreement with Model 1 is again due to the assumption used in the above case with zero inlet air humidity. It should be expected that, for the general case of humid inlet air, the results obtained by Model 0 and Model 1 will be different. The values obtained by Model 2 are significantly different from those of Models 0 and 1, giving higher values for T_4, P_4, T_5, the power turbine work and the thermal efficiency. This option is closest to the actual fuel (AF) solution of Bathie [13]. Also, note the difference between the standard air analysis (SA) of Bathie [13] and all other solutions.

Bathie [13] analysed the gas turbine cycle with steam injection at a temperature of 380 ℃ and a rate of 2.5 % to that of the dry air flow. For a dry air mass-flow rate of 1 $kg \cdot s^{-1}$, his values for the power and thermal efficiency were 375.4 kW and 37.5 %, respectively. The corresponding values obtained by Model 1 of the present model, which is similar to his AE method, are 377.2 kW and 37.7 %. The values obtained by the present Model 2 for the power and thermal efficiency are 394.6 kW and 39.4 %, respectively.

Figure 2a,b compares the model estimation for the power and heat rate of the gas turbine at different inlet-air temperatures with other estimates obtained from

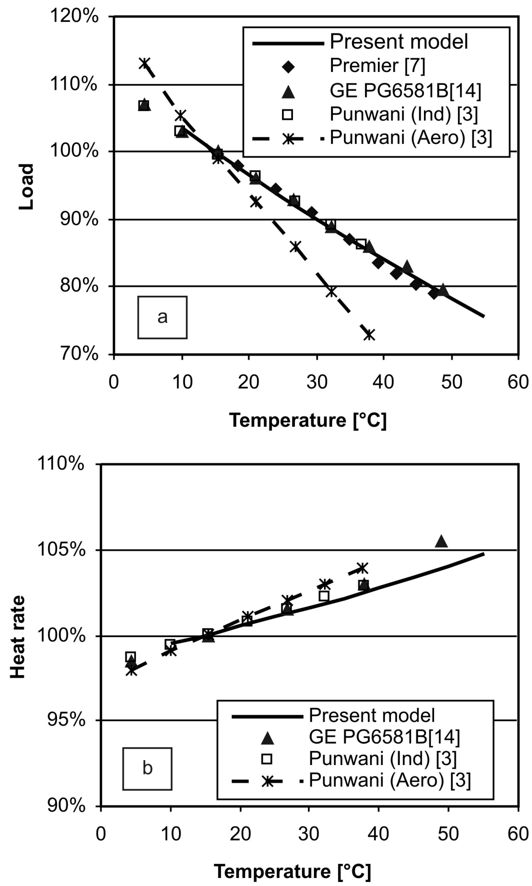


Fig. 2. Effect of ambient temperature on the performance of gas turbines: (a) power, (b) heat rate.

the literature [3, 7, 14]. Punwani [3] provided estimates for both industrial and aeroderivative gas turbines. As can be seen from the figure, the model estimate of the turbine power agrees well with those given by all three sources for industrial gas turbines. Although the model estimate for the heat rate is also comparable to the data of Punwani [3], it slightly underestimates that of the General Electric PG6581B model [14] at high inlet air temperatures. Figure 2 clearly shows that the model is suitable for industrial gas turbines rather than aeroderivative ones.

Since the effect of inlet-air humidity on the turbine power and heat rate is less significant than that of air temperature, the "dry-air" solutions shown on Fig. 2 are practically useful for humid air as well as dry air conditions. Also, since the effect of inlet-air temperature on the turbine power is more significant than that on the heat rate, an inlet-air cooling system will pay for itself by the increased megawatts more than by the reduced fuel consumption of the gas turbine.

5. Conclusions

Verification of the present computer-based thermodynamic model against relevant published data shows that the model estimates for the selected key cycle parameters compare well with these data. The effect of inlet-air cooling and steam injection on the power and heat rate of the gas turbine is also predicted accurately by the model. Therefore, the present model can be used to study the effect of evaporative or refrigerative inlet air cooling and water/steam injection on the gas turbine cycle and to analyse the effect of different combinations of these power augmentation methods on the gas turbine performance. The model can also be extended to calculate the expected annual revenues that result from the increased megawatts and reduced fuel consumption from inlet air cooling and/or water/steam injection.

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