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Evaluation of component characteristics of a reheat cycle gas turbine using measured performance data

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Abstract

In this work, component characteristics of a reheat cycle gas turbine in a commercial combined cycle power plant were evaluated. An inverse performance analysis, in which component characteristic parameters were estimated based on measured performance data, was carried out. The measured parameters were the power, the fuel flow rates of two combustors, and the temperatures and pressures at various locations such as the compressor discharge, exits of both the high- and low-pressure turbines. The estimated parameters from the analysis include the compressor and turbine efficiencies and the inlet air flow rate. The analysis was performed for a wide operation range in terms of the ambient temperature and load, providing a database for the variations of the characteristic parameters with changes in the operating condition. In addition, a sensitivity analysis was performed to examine the influence of the uncertainties of the measured parameters on the estimated parameters. The analysis program can be further developed into a performance diagnosis tool and the obtained component characteristic data can be used as reference database.

Keywords: Reheat cycle gas turbine; Characteristic parameters; Performance parameters; Measurement; Sensitivity

1. Introduction

During the past few decades, the demand for both high-efficiency and high-power-density of power generation gas turbines has resulted in diverse efforts by manufacturers. The recent performance enhancement of industrial gas turbines is due to increase in both the turbine inlet temperature and the pressure ratio. As a solution that increases the average heat addition temperature of a gas turbine, reheating in the middle of the turbine is an option. Fig. 1 shows the reheat cycle on a temperature-entropy diagram. The reheat clearly increases the specific power, but this requires a higher overall (compressor) pressure ratio in order to avoid the efficiency penalty. While the reheat upgrades the engine performance, it also causes several problems, especially those related to the operation of the gas turbine. Due to the existence of the reheat combustor, the reheat cycle gas turbine requires more operation parameters compared to a simple cycle gas turbine. The increased number of parameters makes engine control and performance diagnosis more complicated and challenging.

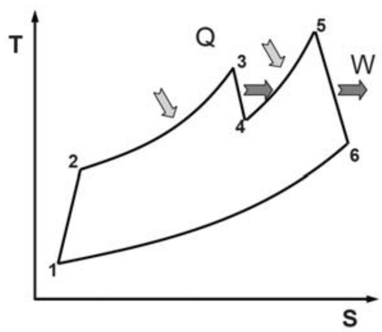

Fig. 1. Temperature-entropy diagram of the reheat gas turbine cycle.

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Accurate information regarding the performance of a gas turbine engine is very essential to end users. In particular, the characteristics of major components such as the flow rates and efficiencies should be estimated accurately. Their usefulness appears in the monitoring of engine conditions. Condition monitoring is enabled by performance diagnosis, which estimates component characteristic parameters such as the air flow rate, the compressor efficiency, and the turbine efficiency from measured performance parameters such as engine power, fuel flow, temperatures and pressures at locations within the engine. To set up a performance diagnostic system, two important factors must be considered. First, a good estimation tool is needed. There are various tools available including model based gas path analysis methods and artificial intelligence methods [1]. Second, reference data for the component characteristic parameters, usually those of healthy operation conditions without degradation, should be given prior to the diagnosis. In particular, the reference component parameters are extremely important because comparing them to actual running data provides information regarding the malfunction of each component.

The reference component characteristic parameters are rarely provided by gas turbine manufacturers. Although some parameters may be provided, they are of only nominal value and should be evaluated onsite for the specific engine. In general, the parameters can be obtained with model-based simulation. Furthermore, they should be evaluated for individual engines of a fleet, as performance differences exist among different engines for several reasons including engine-by-engine manufacturing tolerance and the different settings or tunings of each engine during operation (mainly after the engine overhaul). Therefore, accurate estimation of the component characteristic parameters based on specific operation data must be a primary step in a successful performance diagnosis. Once the parameters are evaluated for a wide operating range and saved into a database, they can be used further as reference values for engine performance diagnosis and condition monitoring.

An engine performance diagnosis predicts deviation of the estimated component parameters, from the reference (or healthy) data. One of the common schemes is the adaptation method, in which the characteristic parameters are modified (adapted) to match the measured performance parameters. In general, the adaptation is an inverse problem [2], in which the component characteristics (independent parameters in usual analyses) are obtained based on measured performance parameters (dependent parameters in usual analyses). A representative method can be found in the literature [3,4]. Other schemes, conceptually similar to those mentioned, can also be found in recent publications. Most of them utilize off-design simulation and attempt to adapt the reference parameters to match the measured engine performance optimally. Therefore, accurate information regarding the reference characteristic parameters prior to a diagnosis is very important in any diagnostic tool. Roth et al. [5] reviewed the currently available methods of matching engine simulation models to available test data and suggested a new method. In addition, Li et al. [6] suggested estimation of the component parameters at the design point based on an adaptation method, and Lee et al. [7] estimated component parameters of a micro gas turbine adopting an inverse calculation method based on measured performance data.

The present study aims to construct a cycle thermodynamic model of a reheat cycle gas turbine and estimate the component parameters, such as the air flow rate as well as the compressor and turbine efficiencies. The object gas turbine is one of several gas turbines (Alstom GT24) in a commercial combined cycle power plant [8]. As detailed component-related data are not provided by the manufacturer as usual and a preliminary review revealed that the operating performance is lower than that reported to the public, a detailed engine-by-engine analysis should be performed to understand the running condition precisely. The estimation is an inverse calculation to match the predicted performance parameters to the measured counterparts. As a preliminary study, the present authors have applied this method to the same gas turbine using commercial performance analysis software [9] and demonstrated the feasibility of the method to obtain the characteristic parameters. Confirmed by the proven feasibility of the method, an in-house program with similar analysis capability has been developed. The program can be extended to a wide range of studies such as engine condition monitoring more easily than the commercial software. This paper describes the modeling, analysis procedure and results performed by the program. Also described is the sensitivity of estimated component parameters to the measured parameters, which illustrates the influence of the measurement accuracy on the predicted component characteristics. The database obtained in this study can be used as reference data and the simulation method can be further developed into a diagnostic tool.

2. Gas turbine

A schematic layout of the reheat cycle gas turbine is shown in Fig. 2. The gas turbine adopts a 22-stage axial flow compressor. The hot gas from the primary combustor flows into a single-stage high-pressure turbine (HPT), after which the gas is further heated in the second combustor. The gas then expands at a four-stage low-pressure turbine (LPT). The compressor, the HPT and the LPT are mounted on a single shaft. Eight gas turbines are operating in the combined cycle power plant, and they are first versions of the engine model. A common summary of several accounts of nominal performance data of the early versions of the engine model [8, 10, 11] is that the power and thermal efficiency are 165 MW and 37.9% (at 15^{o}C and 1 atm), respectively, with a pressure ratio of 30 and an air flow of 378 kg/s. However, the observed power and efficiency onsite are lower than the nominal values. Thus, the engines must be running at conditions that deviate from the original design (or nominal) conditions. For this reason, investigations of the accurate operating conditions of the engines are very important. Once the operating characteristics of all components are derived from the analysis, they can be used as a reference database for further studies such as diagnostics, engine condition monitoring and performance upgrades. To cover a wide operation range, one of the eight gas turbines in the power plant, operating at the load following mode instead of the base load mode, was selected for the analysis.

Various performance data during the commercial operation (not during a well-organized test run) were used for the analysis. The data acquisition system records numerous operating parameters. From the logged database, parameters relevant to the performance analysis were selected. The data measured at the low-temperature sections include the inlet guide vane angle, the ambient pressure and temperature, and the compressor discharge pressure and temperature. Measured at the hot sides were the fuel flow rate at the first combustor, the exit pressure and temperature of the high-pressure turbine, the fuel flow rate of the second combustor, the exit pressure, and temperature of the low-pressure turbine. The onsite data processing system also provides the turbine inlet temperatures of both the high-pressure and low-pressure turbines. However, as usual, these values are not directly measured but are estimated by using a correlation embedded by the engine manufacturer as a function of the relevant measured parameters including the exit temperature and the pressure ratio of the turbine. Thus, they are pseudo-measured parameters and may not perfectly represent the actual thermodynamic mean temperature. Therefore, they were not used as given parameters that should be exactly matched by the analysis, as will be explained later.

As shown in Fig. 2, the coolant flow paths are very complicated. Accurate modeling of the hot-section cooling is an important task in a precise simulation of a modern high temperature gas turbine. Thus, all of the detailed flow paths were simulated precisely in this work, as depicted in Fig. 2. The cooling air from the last stage (the 22th stage) of the compressor is provided to both the nozzle and the rotor of the high pressure turbine. Before being injected, the cooling air is pre-cooled. The cooling air from the 16th stage exit is also pre-cooled and then supplied to the nozzle and the rotor of the first stage and the rotor of the second stage of the low-pressure turbine. The cooling air from the 11th stage is supplied to the nozzles of

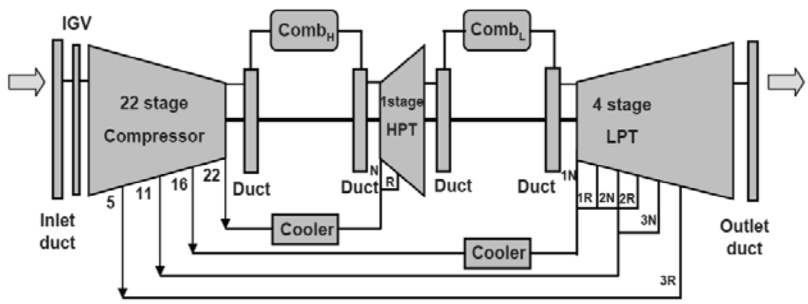

Fig. 2. Schematic layout of the reheat cycle gas turbine.

the second and the third stages of the low pressure turbine, and the cooling air from the fifth stage exit is provided to the third stage rotor. The fourth stage of the low-pressure turbine is not cooled. The coolant temperatures after the pre-coolers are also measured. Since the individual coolant flow rates are not measured, they must be estimated as a result of the analysis with the aid of proper assumptions as will be explained later.

Given these measured performance parameters, component characteristic parameters such as the compressor efficiency, efficiencies of both the HPT and the LPT, and the air and coolant flow rates should be obtained. In particular, the air flow rate is a very important parameter of the gas turbine, but it is not usually measured onsite. Measuring the air flow with sufficient accuracy in a large size gas turbine during commercial operation is nearly impossible; therefore, its prediction for every operating condition by analysis is very useful in many ways. First, combined with the pressure ratio, it can be used as a decision parameter for the compressor malfunction situations such as fouling. It can also be used for other purposes such as a thermal process analysis of the elements located downstream of the gas turbine. Hence, in this work, focus is given to an estimation of the air flow rate and other component efficiencies.

3. Analysis

3.1 Modeling

In usual simulation programs, the component characteristic parameters are not the dependent variables that are determined as a result of the simulation but instead are the independent parameters. Thus, the analysis adopted in this work is an inverse analysis because the component characteristic parameters should be found as a result of the analysis. A gas turbine analysis program [12], originally developed for the analysis of simple cycle gas turbines, was adopted as the framework of the program. Various modifications were made to revise the program into a reheat cycle version. Major modifications include the addition of a reheat combustor and the split of the turbine into two parts (HPT and LPT). As detailed modeling for basic elements and numerical schemes can be referred to the literature, they are not repeated here. Rather, the modeling of major components and the additional modeling required to simulate the current

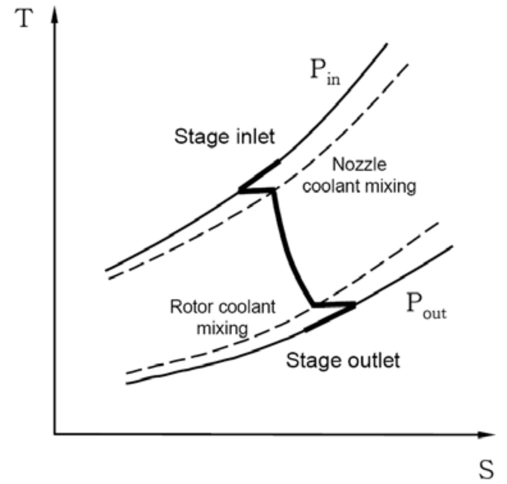

Fig. 3. Cooled turbine stage model.

specific engine is given briefly.

As no specific design guideline regarding the power split or pressure ratio split inside the compressor and the turbine is known, assumptions are required to determine the inter-stage parameters. In the compressor, the inter-stage temperature and pressure are required to simulate the properties of the extracted coolant. In the turbine, they enable a practical thermodynamic model of the mixing between the gas and the coolant. In the compressor, pressure ratios of the individual stages are assumed equivalent and a constant stage efficiency, defined as follows, is assumed for all stages.

$$\eta_{cs} = \frac{h'_{s,o} - h_{s,i}}{h_{s,o} - h_{s,i}} \tag{1}$$

As the stage pressure ratio is very small in a multistage axial compressor, the stage efficiency is nearly equivalent to the polytropic efficiency. All coolant bleed points are simulated as in the actual engine (see Fig. 2), and the coolant air temperature after the cooler is matched to the measured data.

A row-by-row calculation considering the blade cooling, as incorporated in the existing program [13], was applied to every stage of the turbine. Fig. 3 shows the adopted stage model of a cooled turbine. The mixing of coolant with the main gas stream at both the nozzle and the rotor, which leads to a drop in the temperature and pressure, is modeled. The isentropic stage efficiency is applied to the expansion process, which is assumed to be free of coolant mixing.

$$\eta_{ts} = \frac{h_{e,i} - h_{e,o}}{h_{e,i} - h'_{e,o}} \quad (2)$$

The high-pressure turbine consists of a single stage, while the low-pressure turbine consists of four stages. As in the compressor, design guidelines for the inter-stage parameters for the low-pressure turbine do not exist. If all of the stage pressure ratios and efficiencies of the low-pressure turbine (eight parameters in total) were to be estimated independently, the number of searched characteristic parameters would be too great (the number of parameters to be solved would be larger than the number of given parameters), resulting in multiple solutions. Therefore, proper assumptions are required to obtain a feasible unique solution. In this work, the pressure ratios and the efficiencies of the four stages were assumed constant. As the individual flow rates at the coolant injection ports were not measured, they should be obtained as a result of the analysis, that is, they are also characteristic parameters that must be obtained. However, the use of all of the eight coolant flows as independent parameters would provide too many degrees of freedom in the simulation. Therefore, proper assumptions are needed for the total amount of coolant and its distribution among various lines. The distribution of the coolant between the nozzle and the rotor in a stage is assumed as 1:1. Considering that more coolant is generally used in the hotter stages, a decreasing distribution function from the first stage to the third stage is adopted in the low-pressure turbine.

The flow rates of the four coolant source lines from the compressor do not exist in the logged database. However, information for reasonable assumption could be deduced from several samples of off-line data, where the ratio between the coolant for the HPT (from the compressor exit) to the coolant for the LPT (the sum of the coolant flows from 15th, 11th and 5th stage of the compressor) could be evaluated. For a very wide operation range (full to 50% power), the following ratio was found to be nearly constant [9].

$$\alpha = \frac{\dot{m}_{cl,LPT}}{\dot{m}_{cl,HPT}} \quad (3)$$

The average value is 2.45 with relatively small deviations. Therefore, this value is used to divide the total coolant into two parts in the simulation. At this point, only the total coolant amount is left undetermined. Again, from the sample off-line data, the fraction of the total coolant flow rate relative to the compressor inlet air flow rate can be evaluated. The fraction is defined as follows:

$$\beta = \frac{\dot{m}_{cl,total}}{\dot{m}_{c,in}}, \quad (4)$$

where $\dot{m}_{cl,total} = \dot{m}_{cl,HPT} + \dot{m}_{cl,LPT}$

This fraction was found to merge into a narrow band from 0.33 to 0.35. Accordingly, a constant value of 0.34 is assumed for the ratio. Finally, all of the coolant flow rates are now linked to the inlet air flow rate. Thus, the total number of searched parameters is greatly reduced. Once the air flow rate is assumed, the total coolant flow is calculated and then the HPT and LPT coolant values are determined by using Eqs. (2) and (3). Subsequently, the flow rates of individual coolant injections at each turbine section are determined by the local distribution function.

For both the first and second combustors, the exact mass and energy balances considering reasonable combustion efficiencies are applied. The right composition for the natural gas fuel, provided by the supplier, is used.

The power output of the gas turbine is calculated as follows:

$$\dot{W} = \left(\dot{W}_{HPT} + \dot{W}_{LPT} - \dot{W}_c / \eta_m\right) \cdot \eta_{gen} \quad (5)$$

Since no detailed information about the mechanical and generator efficiencies was available, they were assumed to be 0.99 as a representative value in large gas turbines. The thermal efficiency of the gas turbine is calculated as follows:

$$\eta_{th} = \frac{\dot{W}}{(\dot{m}_{f1} + \dot{m}_{f2}) \cdot LHV_f} \quad (6)$$

Here, the subscripts 1 and 2 denote the first and the second combustor, respectively.

3.2 Analysis procedure

The analysis is an inverse calculation to estimate the component characteristic parameters (the air flow, the coolant flows, and the compressor and turbine efficiencies) that match the predicted performance parameters to the measured ones (the gas path temperature and pressures, the power, and the fuel flows). Table 1 lists all of the parameters. In the measured parameters, the compressor inlet temperature and all pressures such as the compressor inlet temperature and pressure, the compressor discharge pressure, and the LPT exit pressure, and the HPT exit pressure are classified as given parameters. This implies that they were directly given as inputs in the simulation. As a result, the number of remaining performance parameters is six.

Since all stage efficiencies of the LPT are assumed equal, only one variable exists for the LPT efficiency. Similarly, as coolant flow rates are linked to the inlet air flow as explained in 3.1, only one degree of freedom, represented by the inlet air flow, exists for the flow rates. Consequently, we have six characteristic parameters to obtain. As there are exactly matched numbers (six) between the to-be-matched and to-be-estimated parameters, a single solution set for every operation condition is possible mathematically.

Table 1. List of parameters.

Measured performance parameters	Given (fixed) parameters	Compressor inlet pressure Compressor inlet temperature Compressor discharge pressure HPT exit pressure LPT exit pressure
	To-be-matched parameters	Compressor discharge temperature Fuel flow at the first combustor HPT exit temperature Fuel flow at the reheat combustor LPT exit temperature Power
To-be-estimated characteristic parameters	Inlet air flow Compressor efficiency HPT efficiency LPT efficiency HPT inlet temperature LPT inlet temperature	

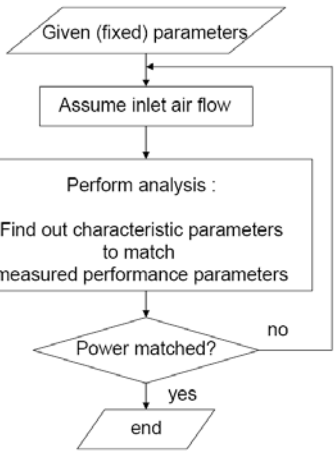

Fig. 4. Schematic flow chart for the simulation.

The simulation process starts with an assumed inlet air flow rate. The remaining five characteristic parameters can then be estimated by perfectly matching the five measured performance parameters excluding the power. The iteration proceeds until the calculated power is matched perfectly to the measured power. Fig. 4 shows a brief analysis procedure.

4. Results and discussion

4.1 Characteristic parameters

The performance analysis was applied to various operating conditions of the gas turbine and the characteristic parameters were estimated. The analysis was shown to produce stable solutions for all operating conditions that were been arbitrarily selected. In particular, the predicted air flow rate is very reasonable in terms of both the absolute value and its variation with operating conditions. Fig. 5 illustrates variations of the estimated air flow rate at the full-load condition with ambient temperature. Also shown is the variation of the gas turbine power. The gas turbine is equipped with a variable inlet guide vane (IGV); the angles of the first two stator vanes are also variable. At a full-load condition, the IGV angle is +2 degrees. During the load reduction procedure, the vanes are closed to reduce the air flow rate. The data of Fig. 5 are those with the IGV fully open. At operation conditions close to ISO design condition (15^{o}C), the estimated air flow rate is 367 kg/s. This can be evaluated as sufficiently close to the actual flow rate considering published reference data [8,10,11]. The variation with the air flow with ambient temperature also appears feasible. This validates the feasibility of the current analysis scheme.

From now on, variations of estimated parameters during load change will be discussed. For illustration purpose, data for three different ambient temperatures were selected:

- Case 1: 24.3^{o}C
- Case 2: 15^{o}C
- Case 3: 3.4^{o}C

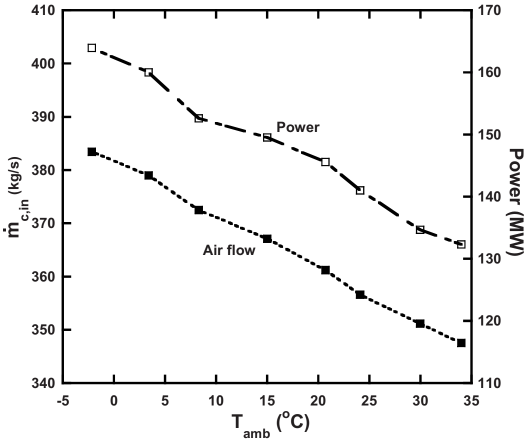

Fig. 5. Variations of measured power and estimated air flow rate at the full load condition with ambient temperature.

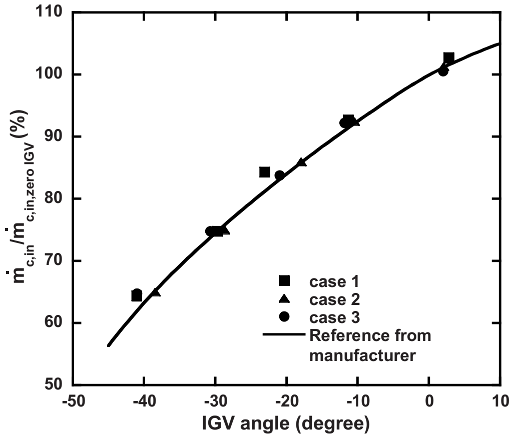

Fig. 7. Variation of the compressor air flow with the IGV angle.

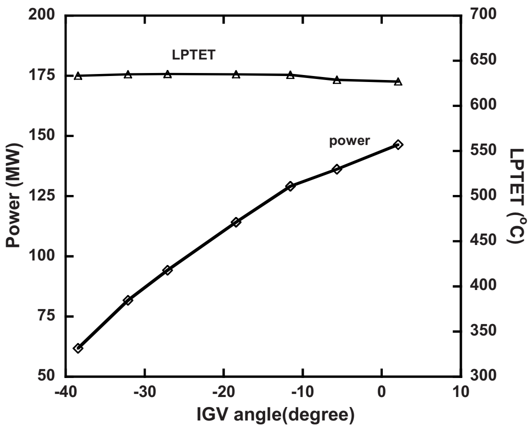

Fig. 6. Variations of the exit temperatures of the LPT with the IGV angle (case 2).

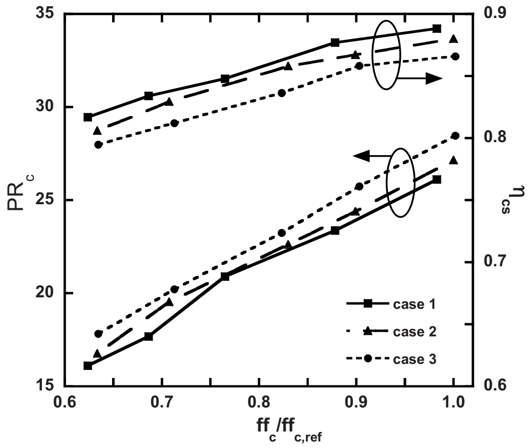

Fig. 8. Operation characteristics of the compressor.

As explained above, during a load reduction process, the vanes are closed to reduce the air flow rate. The angle is +2 and -40 degrees for the fully open and closed conditions, respectively. For each ambient temperature case, five operating points covering the full IGV angle range were analyzed. The purpose of reducing the IGV angle is to reduce the air flow rate flowing into the engine, aiming to maintain a high turbine exit temperatures. Fig. 6 illustrates the exit temperatures of the LPT and power for case 2. For all of the other ambient conditions, the LPT exit temperature pattern is very similar. Thus, the low-pressure turbine exit temperature (the system exit temperature) seems to be controlled as constant during IGV variations for all power and ambient temperature conditions.

Fig. 7 shows the result of the estimated inlet air flow rate. For each case, the air flow rates of all points are normalized by that of the zero IGV angle. The relative variation of the air flow with the IGV angle tested by the engine manufacturer [14] is also shown along with the estimated air flow rate. The relative variations of the air flow rate with the IGV angle for different cases are very similar. Moreover, they coincide very well with the manufacturer's data. Not only this qualitative trend but also the absolute value of the estimated air flow rate is sufficiently accurate. For example, at the full-load condition (full IGV) shown in case 2, which is the ISO ambient design condition, the estimated the air flow rate is sufficiently close to the published reference data, as explained previously. The results shown in Fig. 5 and 7 confirm that the

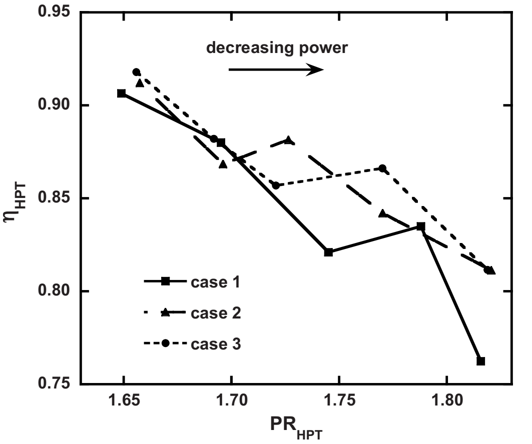

Fig. 9. Variation of the HPT efficiency with the pressure ratio.

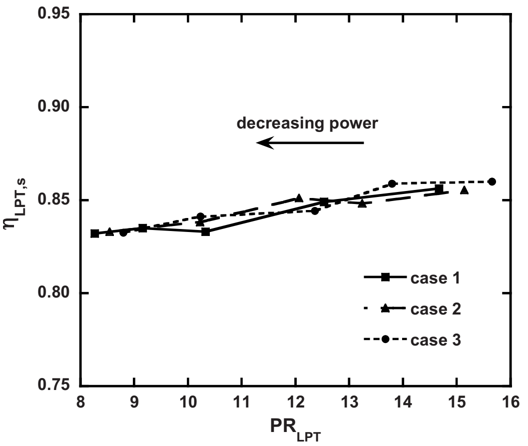

Fig. 10. Variation of the LPT stage efficiency with the pressure ratio.

present analysis estimates very accurate air flow rates in terms of both the absolute values and the variations with changes in the operation conditions.

Compressor characteristics are estimated as in Fig. 8. The pressure ratio and compressor efficiency are plotted as a function of the compressor flow function defined as follows:

$$ff_C = \frac{\dot{m}_{C,i}\sqrt{T_{C,i}}}{P_{C,i}} \tag{7}$$

The abscissa is the flow function normalized by a reference value, which is the flow function at the full IGV condition of case 2. The relationship between the pressure ratio reduction and the flow reduction during the IGV closing process is nearly linear. The compressor stage efficiency decreases as the IGV is closed, as expected. Furthermore, its variation with the air flow (and with IGV angle) is nearly linear. It is also notable that its variation is very similar in all ambient temperature conditions. The absolute value of the efficiency is higher at a higher ambient temperature. The maximum efficiency reduction from the fully opened to the fully closed IGV is about 7 percent point.

Figs. 9 and 10 show the variations of the estimated efficiencies of HPT and LPT with the turbine pressure ratios. The LPT efficiency denotes the stage efficiency that is assumed constant for all stages, as explained in the component modeling. Before examining the efficiency, it should be noted that the HPT pressure ratio increases as the engine power reduces, whereas the LPT pressure ratio decreases. In all cases, the efficiencies generally decrease as the operation condition deviates from the design condition (to the right for HPT and to left for LPT), which must be a quite reasonable trend. The maximum efficiencies are slightly higher than 90% for HTP and about 86 % for LPT, respectively. The relative reduction of the LPT efficiency is less than 5 %. However, the HPT efficiency varies by much more. The relatively large and irregular variations of the HTP efficiency may be partly related to a high dependence of its prediction on the accuracy of the measured parameters, which will be demonstrated in the following section.

4.2 Sensitivity analysis

The previous section exemplified the usefulness of the present analysis scheme to predict the characteristic parameters of arbitrary operation conditions. The key factor dominating the accuracy of the predicted characteristic parameters is the correctness of the measured performance data. Using the data measured during commercial operations instead of during well-organized performance tests may have some shortcomings, particularly regarding the uncertainties of the acquired data. First, although steady state data were sought, some transient effects may exist. Second, issues can arise if some parameters, provided after data reduction such as averaging, exactly represent the actual thermodynamic mean properties that are required for a precise performance analysis. In general, several gas path properties are mean values of outputs from a number of sensors. For example, the turbine exit temperatures of the current gas turbine

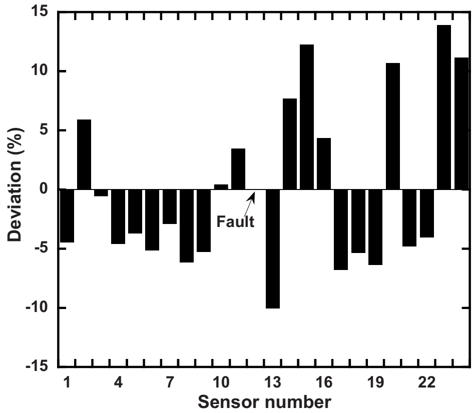

Fig. 11. Example of the distribution of the measured HPTET.

were measured at more than 20 points, with sensible distributions among individual values. Fig. 11 illustrates an example of the distribution of the measured temperature values taken from sensors located at the high-pressure turbine exit. The figure represents the deviation of each sensor value from the average value used as the HPTET in the analysis in the previous section. The maximum difference between the highest and the lowest values of the HPTET is more than 20 percent of the average value. In addition, a few sensor faults exist. Although smaller than those of the HPTET, the deviations of the local temperatures at the low-pressure turbine exit are also considerable. Therefore, it is quite possible that the average values may not precisely represent the thermodynamic mean temperatures that satisfy the performance analysis.

Once there is an error in estimating the true thermodynamic mean value of any measured performance parameter, the solution (characteristic parameters) may be altered. Therefore, it is worth examining how sensitively each estimated characteristic parameter reacts to variations in the measured performance parameters. The characteristic parameters of the full-load condition of case 2 were re-calculated by artificially altering each of the performance parameters. The effects of single performance parameters on the estimated characteristic parameters were analyzed and the following index was calculated.

$$\text{Sensitivity}=\frac{\Delta Y_j / Y_j}{\Delta X_j / X_j} \tag{8}$$

This index represents the relative variation of each estimated characteristic parameter (Y_j) with the relative variation (uncertainty or error) of each measured performance parameter (X_j). As the variations in the characteristic parameters are nearly linear, the value calculated by the above equation is sufficient as a sensitivity index.

Table 2. Example of sensitivities defined by Eq. (8) for the full load condition of case 2.

Y \ X	HPTET	LPTET	$\dot{m}f1$	$\dot{m}f2$	CDT	HPTEP
$\dot{m}_{a,in}$	+0.07	-1.47	+0.64	+0.93	-0.33	+0.05
η_{HPT}	-8.96	+4.57	+1.02	-2.91	+3.48	+1.74
η_{LPTs}	+0.61	-0.77	-0.34	+0.04	+0.03	-0.52
η_{cs}	0.0	0.0	0.0	0.0	-0.78	0.0

Table 2 summarizes the results. For example, one percent uncertainty in the measured first combustor fuel flow rate results in a 0.64 percent change of the estimated air flow rate. The positive sign indicates that an increase in the measured value results in an increase in the estimated value, and vice-versa. For every measured parameter, the HPT efficiency is the most highly influenced estimated parameter. In other words, the sensitivity values of HP efficiency are far greater than those of other parameters. In particular, uncertainty of the HPTET greatly influences the HPT efficiency. Since the measured HPTET has considerable local variations, as shown in Fig. 11, one percent uncertainty in the averaged value is quite likely. According to Table 2, even this relatively small error significantly distorts the predicted HPT efficiency. Thus, the present measurement system has the possibility of a high level of uncertainty in predicting the HPT efficiency, and improving the measurement accuracy is a key factor for accurate turbine characteristic parameters. The sensitivities of the low-pressure turbine efficiency are relatively small. The estimated air flow rate is affected most highly by the LPT exit temperature. Thus, the measurement accuracy of the LPT exit temperature is very important in predicting the air flow rate, too. The air flow rate is also moderately affected by the fuel flow rates. Since the compressor efficiency can be calculated by the measurements around the compressor, uncertainties of other measured parameters do not affect its prediction.

4. Conclusion

An analysis program was set up to estimate the component characteristic parameters of a reheat gas turbine by using measured performance data. The analysis was applied to a wide operation range and the results can be summarized as follows.

It was found that the inverse calculation guaranteed a stable solution set for every operation condition. In particular, the predicted air flow rate was very reliable in terms of both its absolute value and its variation with respect to changes in the IGV angle. The accurate and consistent estimation of the air flow rate is a useful outcome, as it can be used for diagnostic purposes. The estimated compressor efficiency values were also very realistic. Moreover, the variation of the efficiency as the IGV changes was consistent for all ambient conditions. The variations of the estimated turbine efficiencies with the power were also feasible.

The sensitivities of the estimated characteristic parameters on the measurement uncertainties were also discussed. Indices showing how sensitively each estimated characteristic parameter reacts to variation (error) in the measured parameters were illustrated. It was found that the HPT efficiency is the most highly influenced parameter for all measured parameters. The estimated air flow rate has a relatively high correlation with the exit temperature of the low-pressure turbine. Consequently, in order to fully utilize the present methodology of estimating operating condition of the gas turbine, more reliable measurements, especially those of the turbine exit temperatures, are required.

Since the usefulness of the present simulation program has been demonstrated and a database for the characteristic parameters has been set up for a wide operation range, both the program and database can be further utilized as fundamental tools for the performance diagnostics. Also, since the simulation model used in this work is quite general, it can be applied to predicting component characteristics of any gas turbines based on measured performance data after a few additional modifications required to accommodate specific features of each engine.

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Nomenclature

CDT : Compressor discharge temperature [K or ^{o}C]
ff_c : Flow function of the compressor
HPT : High-pressure turbine
h : Specific enthalpy [kJ/kg]
IGV : Inlet guide vane
LHV : Lower heating value [kJ/kg]
LPT : Low-pressure turbine
$\dot{m}$: Mass flow rate [kg/s]
N : Nozzle
P : Pressure [kPa]
PR : Pressure ratio
R : Rotor
T : Temperature [K or ^{o}C]
TEP : Turbine exit pressure [kPa]
TET : Turbine exit temperature [K or ^{o}C]
$\dot{W}$: Power [kW]
X : Performance parameter
Y : Characteristic parameter
α : Coolant distribution ratio
β : Coolant fraction
Δ : Deviation
η : Efficiency

Superscripts

' : Isentropic

Subscripts

1 : First combustor
2 : Second combustor
amb : Ambient condition
c : Compressor
cl : Coolant
e : Expansion
f : Fuel
gen : Generator
j : Each parameter
i : Inlet
m : Mechanical
ref : Reference
s : Stage
t : Turbine
th : Thermal

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