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Exergy Based Performance Analysis of a Gas Turbine at Part Load Conditions

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Abstract: A gas turbine can run at off-design due to change of ambient conditions or load demand. This needs to study how part load affect the gas turbine performance. The gas turbine considered for this study consists of variable stator vanes (VSVs). The VSVs are re-staggered for load greater than 50% so that it would increase exhaust gas temperature. For load less than 50% the VSVs are fully opened. The merit of the gas turbine system should be determined using exergy analyses because energy analyses tend to overestimate performance. Therefore, exergy model of each gas turbine component is formulated. A 4.2 MW gas turbine is analyzed for a wide range of part load condition. It is found that, at full load, the exergy destruction in the combustor is 48.9% and the loss with the exhaust gas is 35.8%. The reminder is being destroyed in the turbine and compressor is 10.8 and 4.3%, respectively. For comparison purpose both the first and second law efficiencies of each component are represented together. This analysis would help to identify the equipment where the potential for performance improvement is high, and trends, which may aid in the design of future plants.

Key words: Gas turbine, variable stator vanes, exergy destruction, part load performance

INTRODUCTION

A gas turbine is a complex assembly of various components and the design operation theories of these individual components are complex. The gas turbine often runs at off-design situations due to change of load or ambient condition. Therefore, when evaluating the overall performance of the gas turbine, it is important to account for all operating conditions that can be encountered. Furthermore, the merit of the gas turbine system should be determined using exergy analysis because energy analysis tends to overestimate performance.

Exergy, also known as availability, is a measure of the maximum useful work that can be obtained when a system is brought to a state of equilibrium with the environment in reversible processes. Therefore, a system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment (Cengel and Boles, 2006). The properties of a system at the dead state are denoted by subscript zero. Such information is useful when designing a thermal system or reducing sources of inefficiency in an existing system.

The specific exergy on a mass basis, ψ , expressed as the sum of thermomechanical and chemical contributions, is given as (Moran and Sciubba, 1994; Moran and Shapiro, 2006):

$$\psi = (h - h_O) - T_0(s - s_0) + \frac{V^2}{2} + gz_i + \psi^{CH} \quad (1)$$

where, h, s, and ψ^{CH} are the specific enthalpy, entropy, and chemical exergy, respectively. Furthermore, h_0 and s_0 denote the specific enthalpy and entropy, respectively, at the restricted dead state while V and z are the velocity and elevation of the bulk flows entering and exiting the control volume.

The maximum theoretical work obtainable as a system initially at T_0, p_0 achieves chemical equilibrium with the environment is called the chemical exergy, ψ^{CH} (Moran and Sciubba, 1994). Moreover, for hydrocarbons the fuel heating value roughly approximates the chemical exergy.

The thermomechanical exergy (ψ^{TM}) is the maximum theoretical work obtainable as the system passes from some given state to the restricted dead state. When evaluating the thermomechanical contribution, one can think of bringing the system without change in composition from the specified state to T_0, p_0 the condition where the system is in thermal and mechanical equilibrium with the environment. When a difference in exergy or flow exergy between states of the same composition is evaluated, the chemical contribution cancels, leaving just the difference in the thermomechanical contributions. For such a calculation, it is unnecessary to evaluate the chemical exergy explicitly.

The use of exergy analysis in power plants or generally in thermal design has been discussed and

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demonstrated by numerous researches (Cengel and Boles, 2006; Moran and Shapiro, 2006; Annamalai and Puri, 2002; Boehm, 1987; Boehm, 1997). An exergy analysis of a Braysson (consists of Brayton and Ericsson cycles) cycle for different cycle temperature and pressure ratios with ideal gas assumption was done by Zheng *et al.* (2001). Moreover, the results indicate the Braysson cycle specific work output and exergy efficiency were higher than that of Brayton cycle. Exergy based performance characteristics of heavy duty gas turbine in part load operating conditions was investigated by Song *et al.* (2002). The performance of a waste heat recovery based power generation system using the second law of thermodynamics for various operating conditions such as gas composition, specific heat, pinch point and gas inlet temperature was investigated by Butcher and Reddy (2007). Exegetic performance of a fixed geometry gas turbine cogeneration system with constant compressor and turbine isentropic efficiencies at full load for different compressor inlet temperatures was investigated by Oh *et al.* (1996).

However, in the published literatures, the effect of part load engine operation on the performance of gas turbine with variable geometry compressor has not been studied widely. Further analysis is needed. Therefore, the objective of the present work was to examine the performance of a gas turbine using exergy concept for a wide range of gas turbine part load operation.

COMPONENT EXERGY MODEL AND ANALYSIS

Figure 1 shows the basic components of the gas turbine that is considered for this study. Assuming one-dimensional flow at locations where mass enters and exits with single inlet and outlet flow the steady exergy rate balance for a system interacting with the surrounding temperature T° is given by (Moran and Shapiro, 2006):

$$0=\sum_{j}\left(1-\frac{T_o}{T_j}\right)\dot{Q}_j-\dot{W}_{cv}+\dot{m}(\psi_i-\psi_e)-\dot{I} \tag{2}$$

where: e and i are subscripts referring an exit and inlet of a component, respectively; Q_j is the heat transfer rate at the location on the boundary where the instantaneous temperature is Tj and W_{cv} is control volume work.

The exergy destroyed in the rate form is proportional to the rate of entropy generated, and can be expressed as:

$$I=T_0\dot{S}_{gen} \tag{3}$$

For a general steady state single stream flow process the rate of entropy generated is:

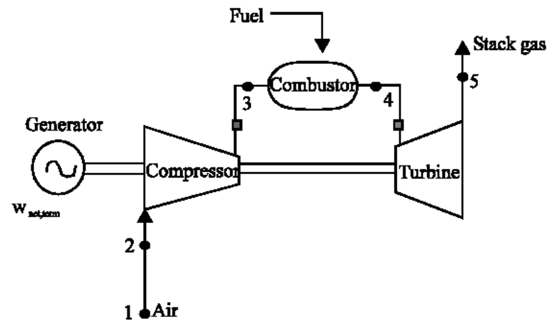

Fig. 1: Schematic representation of a single shaft gas turbine

$$\dot{S}_{gen}=\dot{m}(s_e-s_i)-\frac{Q_j}{T_j} \tag{4}$$

The corresponding change in the flow exergy based on a unit mass is given by:

$$\psi_e-\psi_i=(h_e-h_i)-T_0(s_e-s_i)+\frac{V_e^2-V_i^2}{2}+g(z_e-z_i) \tag{5}$$

Once the exergy change formulated, the exergy destruction or exergy loss within a particular component can be determined by applying the exergy rate balance (3) on that component. All real processes are irreversible due to effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion and friction (Merle and Somerton, 2006). The subscript numbers in the components exergy equation formulation and analysis refer inlet and exit of a component according to the designation given in Fig. 1.

Compressor exergy destruction: Applying exergy destruction rate (3) to the compressor and air with perfect gas behavior, the exergy destruction rate is given as:

$$\dot{I}_c=T_o\dot{m}_a(s_3-s_2)=T_o\dot{m}_a\left(c_{p2-3}\ln\frac{T_3}{T_2}-R_a\ln\frac{p_3}{p_2}\right) \tag{6}$$

The compressor second law efficiency is given as:

$$\eta_{II,c}=\frac{w_{rev,in}}{w_{ac,in}}=\frac{\psi_3-\psi_2}{h_3-h_2}=1-\frac{i_c}{h_3-h_2} \tag{7}$$

The isentropic or adiabatic efficiency, which is a measure of the deviation of actual processes from the corresponding idealized ones, is given by:

$$\eta_{i,c}=\frac{w_{is,in}}{w_{ac,in}}=\frac{h_{3,is}-h_2}{h_3-h_2} \tag{8}$$

where $w_{rev,in}$ $w_{ac,in}$ and $w_{is,in}$ are the reversible, actual and isentropic work input to the compressor, respectively.

Combustion chamber exergy destruction: Applying the exergy destruction rate (3) to the combustor gives:

$$\dot{I}_{cc} = T_o \dot{m}_g (s_4 - s_3) = T_o \dot{m}_g \left(c_{p3-4} \ln\frac{T_4}{T_3} - R_g \ln\frac{p_4}{p_3} \right) \qquad (9)$$

The second law efficiency of the combustion chamber is the ratio of exergy gain to the fuel chemical exergy value (the exergy input) and approximately the same as its lower heating value:

$$\eta_{II,cc} = \frac{\dot{m}_g (\psi_4 - \psi_3)}{\dot{m}_f LHV} \qquad (10)$$

Turbine exergy destruction: Applying (3) to the turbine and assuming perfect gas behavior the exergy destruction rate in the turbine is:

$$\dot{I}_t = T_o \dot{m}_g (s_5 - s_4) = T_o \dot{m}_a \left(c_{p5-4} \ln\frac{T_5}{T_4} - R_g \ln\frac{p_5}{p_4} \right) \qquad (11)$$

Its exergetic efficiency is given as the ratio of actual useful work output to the reversible work output:

$$\eta_{II,t} = \frac{w_{act}}{w_{rev,out}} = \frac{h_4 - h_5}{\psi_4 - \psi_5} = 1 - \frac{i_t}{(h_4 - h_5) - T_0 (s_4 - s_5)} \qquad (12)$$

The isentropic or adiabatic efficiency of the turbine, which is a measure of the deviation of actual processes from the corresponding idealized ones, is given as the ratio of actual useful work output to the isentropic work output:

$$\eta_{is,t} = \frac{w_{ac,out}}{w_{is,out}} = \frac{h_4 - h_5}{h_4 - h_{5,is}} \qquad (13)$$

Stack gas exergy loss: The rate of exergy loss with the stack gas to the surroundings is given by:

$$Stack_{exr,loss} = \dot{m}_g T_o \left((h_5 - h_o) - T_o (s_5 - s_o) \right) \qquad (14)$$

Applying the perfect gas behavior for the change in enthalpy and specific entropy from the exhaust state to the surroundings the rate of exergy loss with the exhaust gas is written as:

$$Stack_{exr,loss} = m_g T_0 [C_{p,exh} (T_5 - T_0) - (cp_{exh} \ln\frac{T_5}{T_0} - R_{exh} \ln\frac{p_5}{p_0})] \qquad (15)$$

SOLUTION METHODOLOGY

Prior to the exergy analysis of the gas turbine its components inlet and exit temperatures and pressures should be known. To predict these properties at a given ambient conditions energy analysis is required. Moreover, the approach followed to develop the energy analysis was modeling the various gas turbine components using conservation of mass and energy, component performance maps, and empirically developed relationships. Then, the modified component matching method was used to interlink them. A simulation model was developed in Matlab environment and used to solve the equations that represent each component process. Simulation was carried out for a set of input data and the inlet and exit properties of each component as well as the whole plant performance were predicted and compared with actual plant data and found valid (Baheta and Gilani, 2010). Furthermore, two modes of gas turbine operation are identified with the first mode being for part load less than 50% running to meet the part load demand. This is achieved by controlling the fuel flow and air bleeding downstream of the compressor to avoid surge formation. The second mode of operation is for part load greater than 50% and running to meet both the part load demand and the exhaust gas temperature set value by simultaneously regulating the fuel flow and the variable vanes opening. Consequently, the thermodynamic properties drastically change at 50% load that is manifested in the results. The inlet and exit properties (temperatures and pressures) of each component are obtained from the energy analysis and used to do the exergy analysis.

RESULTS AND DISCUSSION

In this analysis standard reference ambient pressure and temperature are assumed 101.3 kPa and 298.15 K, respectively. The results of the exergy analysis are discussed as follows for a wide range of load variations. The variation of exergy destruction or the loss work rate in the compressor with respect to load is shown in Fig. 2. The exergy destruction goes on decreasing with load and then suddenly increases at around 50% load and restarts to decrease. The sudden increase is the consequence of the VSVs repositioning to limit the airflow. Although repositioning of the VSVs give high exhaust gas temperature, which is useful for heat recovery, it increases the rate of exergy destruction.

Figure 3 shows that both the isentropic and the second law efficiencies of the compressor follow the same profile with respect to load. In the first mode of operation

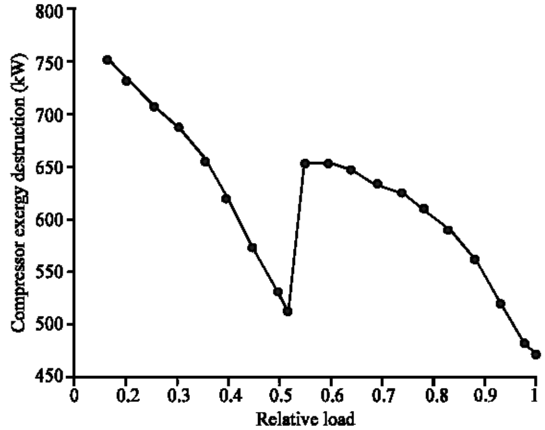

Fig. 2: The variation of exergy destruction rate in the compressor with respect to load

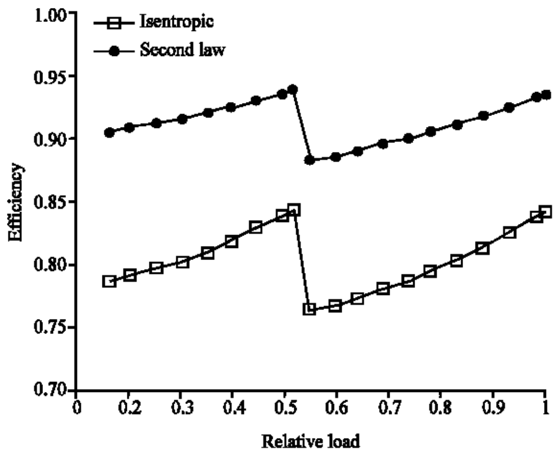

Fig. 3: Variation of compressor isentropic and second law efficiencies with respect to load

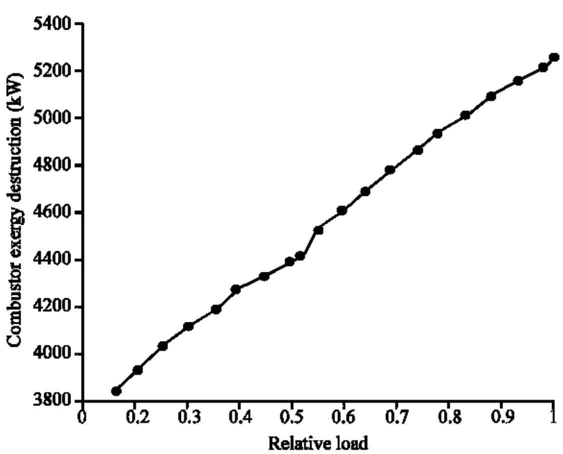

Fig. 4: Combustion chamber exergy destruction variation with respect to load

the efficiencies increase with load and around 50% load they drop suddenly. The reason is associated with the exergy destruction when VSVs is modulated to maintain the exhaust gas temperature set value. After 50% load the efficiencies keep on increasing. In general, the VSVs are useful to modulate the flow and maintain the exhaust gas set value temperature for heat recovery. However, they have also negative effect on the compressor efficiency, which would also affect the subsequent components efficiencies.

The same figure shows that the second law efficiency is greater than the isentropic efficiency. This is because in the isentropic efficiency the useful minimum work input is calculated based on reversible and adiabatic compression that leads to another final state condition; whereas the second law efficiency calculation considers the actual initial and final states and assume reversible compression. That is, the useful minimum compression work input of the second law analysis is higher than the corresponding isentropic work input hence the second law efficiency is higher than its corresponding isentropic efficiency.

The exergy destruction rate in the combustion chamber increases with load as indicated in Fig. 4. As the load increases the exergy destruction rate in the combustion chamber increases significantly. The destruction rate varies in the range of 51.4 to 63.8% of the overall system destruction rate corresponding to the lowest load and full load, respectively. This is because to meet the increase of part load the combustor should produce high temperature gas product associated with that the chemical reaction exergy destruction rate increases.

The combustion chamber exergetic efficiency is increasing as the load increases except the small drop at around 50% load as shown in Fig. 5. This drop is the consequence of the compressor VSVs reposition to modulate the air flow.

Figure 6 shows that the turbine exergy destruction increases throughout the load except the drastic decreases at around 50% load. In the turbine, unlike the compressor the VSVs air flow modulation has positive contribution to the turbine operation as the exergy destruction rate drastically decreases at this load. The reason is the exergy destruction rate is proportional to the flow rate that is reduced by closing the VSVs in the compressor. Consequently, the exergy destruction rate is drastically reduced in the turbine at around 50%.

The variation of the second law and first law efficiencies of the turbine is indicated in Fig. 7. Almost both efficiencies have similar profile with respect to load. However, the second law is higher than the first law efficiency. The reason is in the isentropic expansion process the maximum useful work output is calculated assuming the process is reversible adiabatic that leads to

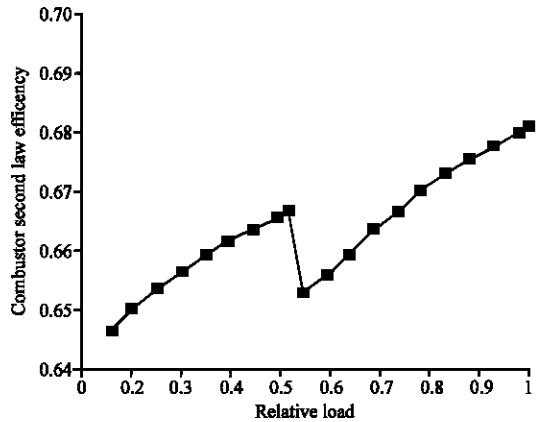

Fig. 5: Variation of combustion exergetic efficiency with respect to load

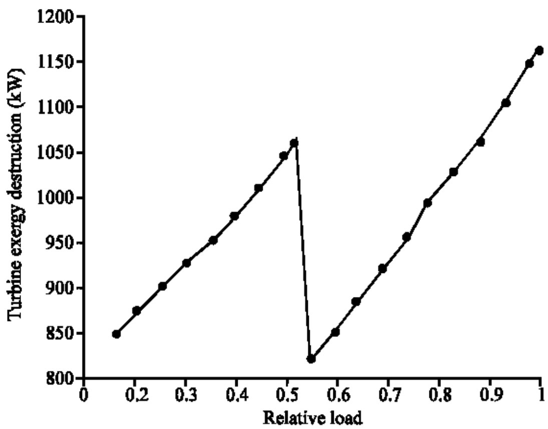

Fig. 6: Turbine exergy destruction with respect to load

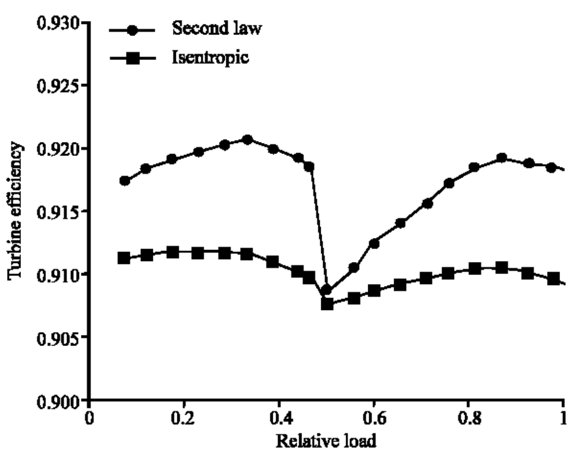

Fig. 7: The variation of turbine efficiencies with respect to load

another final state point that very much deviates from the real process. On the other hand, the second law which

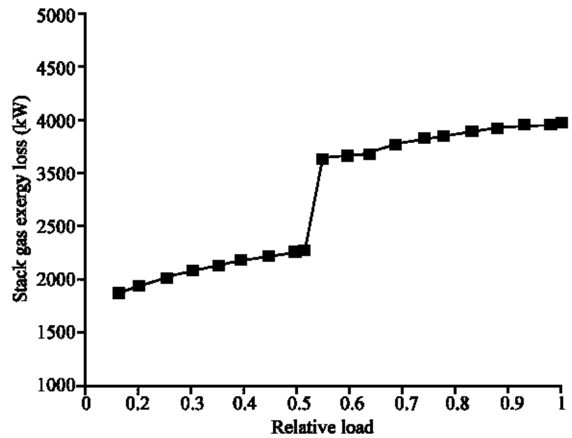

Fig. 8: Variation of the stack gas exergy loss with respect to load

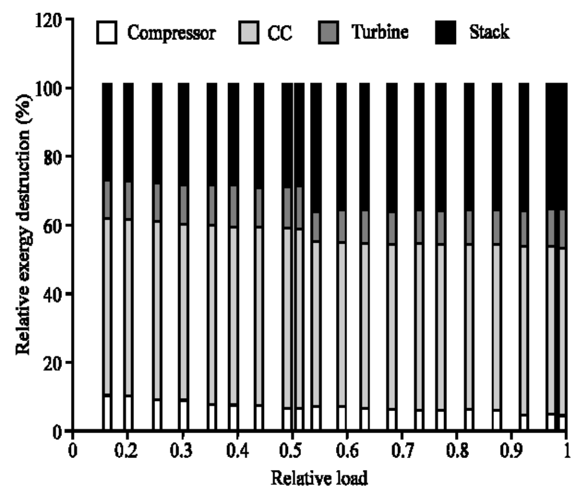

Fig. 9: Variation percent of percentage exergy destruction of the gas turbine components with respect to load

assumes only reversible process, but no actual process reversible, refers the same actual initial and final state points. In other words, the optimum expansion work obtained using the second law analysis is less than the isentropic work output hence the turbine second law efficiency is greater than its corresponding isentropic efficiency. Furthermore, the efficiencies are increasing in the first mode of operation but when the load is around 50% the efficiencies decreases and then again begin to increase. The efficiencies drop is the consequence of reduced mass flow rate in the compressor where the output power is directly proportional to the flow rate.

Figure 8 shows the exergy loss rate with the stack gas. In general the exergy loss with the stack gas is increasing with load. The sudden increase at 50%

happened VSVs modulation to increase the gas turbine exhaust gas temperature for heat recovery.

The gas turbine relative exergy analysis is shown in Fig. 9. In both mode of operations exergy destruction rate in the combustion chamber and the exhaust gas exergy loss rate are responsible for the major exergy losses. At full load the exergy destruction in the combustion chamber is 48.9% and the loss with the exhaust gas is 35.8% the reminder being destroyed in the turbine and compressor are 10.8 and 4.3%, respectively.

CONCLUSIONS

Exergy model of each gas turbine components, i.e., compressor, turbine, combustor and stack gas loss are formulated. The formulated equations are used to evaluate the components exergy destruction rates for a wide range of part load operation. It is found that the exergy destruction rate is increasing with load in all the components except in the compressor. Furthermore, there is exergy destruction rate and efficiency change in each component at 50% load.

This is because the VSVs are modulated to decrease the air flow rate entering into the compressor so that the exhaust gas temperature will increase that could be used for heat recovery. The VSVs modulation contributes to the increase of exergy in all the components except in the turbine. The exergy destruction decrease in the turbine is the result of the reduction of mass flow rate. At full load the percentage exergy destruction rates in the compressor, combustor, turbine and the exergy loss with the stack gas are 10.8, 4.3, 48.9, and 35.8%, respectively. Since devices with the largest exergy losses have the largest margins for efficiency improvement efforts to increase the efficiency of the combustion chamber and reduce the stack gas loss should be made.

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