SIMULATION OF GAS TURBINES OPERATING IN OFF-DESIGN CONDITION

Arnaldo Walter: awalter@fem.unicamp.br
University of Campinas – Dept. of Energy
DE/FEM/Unicamp – P.O. Box 6122 - ZIP code 13083-970 - Brazil

Abstract. *In many countries thermal power plants based on gas turbines have been the main option for new investment into the electric system due to their relatively high efficiency and low capital cost. Cogeneration systems based on gas turbines have also been an important option for the electric industry. Feasibility studies of power plants based on gas turbine should consider the effect of atmospheric conditions and part-load operation on the machine performance. Doing this, an off-design procedure is required. A GT off-design simulation procedure is described in this paper. Ruston RM was used to validate the simulation procedure that, general sense, presents deviations lower than 2.5% in comparison to manufacturer's data.*

Key words. *Gas turbines; simulation.*

1. INTRODUCTION

Gas turbines market has risen continuously during last two decades. In many countries combined cycles are the main option for centralized power sector expansion. On the other hand, distributed power generation has played a very important role as well and, in this category, cogeneration plants based on gas turbines have been a choice of preference.

Gas turbine performance at ISO basis (burning a reference fuel, as natural gas, at 15ºC, 1 atm, and 60% relative humidity), is an information always provided by machine manufacturers. Nevertheless, a GT frequently operates under an off-design condition as, for instance, at part load or under different atmospheric conditions, and in these conditions changes of performance can be dramatic. A good prediction of GT off-design performance is essential for accurate technical and economic assessments.

Off-design simulation of gas turbines requires a special procedure. This paper presents the basics of GT off-design simulation, highlighting the impact over the results of some parameters evaluation, such as (i) new compressor pressure ratio, (ii) corrected compressor air mass flow and (iii) correction of compressor efficiency. Two different off-design situations are analyzed: (i) GT operation under variable ambient temperature, and (ii) part load GT operation at ISO basis.

2. GAS TURBINE OFF-DESIGN SIMULATION PROCEDURE

A gas turbine off-design simulation procedure evolves two steps. The first one is a tuning phase, while the second is the simulation itself under an off-design condition. The tuning step corresponds to the evaluation of unknown GT parameters, usually not provided by manufacturers, suchlike compressor and turbine efficiencies, GT firing (maximum) temperature, pressure and thermal losses and cooling air fraction. Tuning procedure requires both an appropriate set of equations modeling energy conversion inside the machine and the knowledge of some GT operational and performance parameters (turbine inlet temperature, thermal efficiency, gas exhaust temperature, etc.) to check tuning adequacy. This shall be done for a well-known operation condition, as GT full load operation at ISO basis, for which basic data are presented at the open literature.

In predicting machine off-design performance a special procedure that uses the ISO known solution (modeling and setup parameters) as the start point is applied. Vis-à-vis the basic modeling, additional functions are required to predicted revising of some GT parameters. Responding to ambient temperature variation or GT de-rating (reduction of maximum gas temperature) a GT can present substantial changes in compressor pressure ratio, compressor air mass flow and operational efficiencies of both compressor and expander.

The machine considered in this paper is the first version of former Ruston RM model, an industrial machine, single-shaft, precursory of current ABB Alstom Typhoon, for which a detailed performance map was available (Gas Turbine World, 1987). Gas turbine manufacturers generally present off-design data on their products in terms of performance parameters at different ambient temperatures (Desideri, 1994), but detailed information on part load are not usual. The performance map was determined for the following hypothesis: intake and exhaust losses equal to 10 and 20 mbar, respectively, alternator efficiency 97% and gear box efficiency 98.2%. This map defines a reference of comparison for off-design cases that correspond to varying ambient temperature and part load operation at ISO basis. Basic performance data of Ruston RM at ISO basis are presented in Table 1 while off-design performance curves are presented in Figures 1 and 2. Table 1 also presents estimated parameters during tuning phase that vanish errors vis-à-vis declared ISO rated performance.

Table 1. Ruston RM performance parameters at ISO basis and estimated parameters at tuning

Manufacturer performance parameters			
Gas turbine power	3789 MW	GT thermal efficiency	29.84%
Turbine inlet temp. – TIT	1054.5ºC	Turbine exhaust temp. – TET	500ºC
Compressor pressure ratio	12.8	Exhaust gas mass flow	16.87 kg/s
Estimated and imputed parameters			
Compressor isentropic effic.	83.0%	Expander isentropic efficiency	87.69%
Cooling air fraction	18.06%	GT maximum temperature	1106.1ºC
Compressor inlet flow	16.6 kg/s	Fuel flow (LHV 47.45 MJ/kg)	0.2676 kg/s
Combustor pressure drop	5.5 dp/p	Combustor efficiency	99%

3. GT Modeling and Off-Design Equations

The set of equations modeling energy conversion within the main components of a gas turbine (compressor, combustion chamber and expander) are not presented here. Basically, compressor and expander equations are defined for polytropic processes of known efficiencies allowing the evaluation of gas outlet temperature and the power evolved with the compression and the expansion. Compressor equations are functions of the inlet parameters (air temperature, pressure and mass flow), the compressor pressure ratio and the isentropic efficiency. Conversely, expander equations are function of turbine inlet parameters (temperature – TIT and pressure – TIP), expander pressure ratio (different from compressor pressure ratio due to pressure losses at the combustion chamber) and the isentropic efficiency.

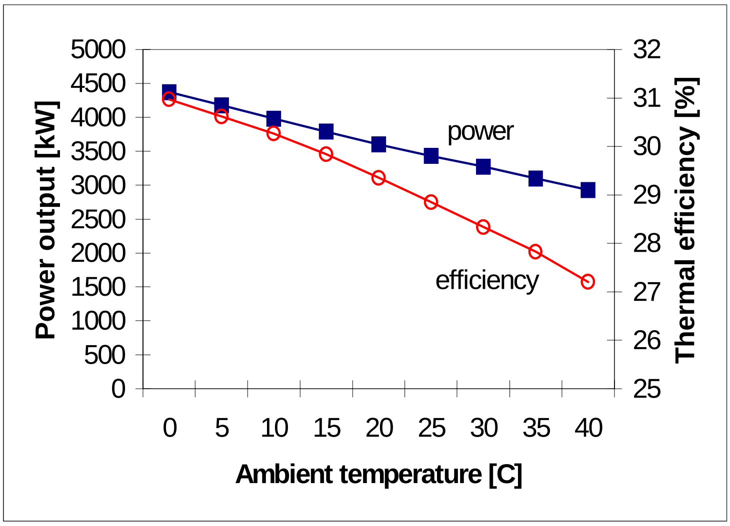

Figure 1. Ruston RM – power output and thermal efficiency as function of ambient temperature

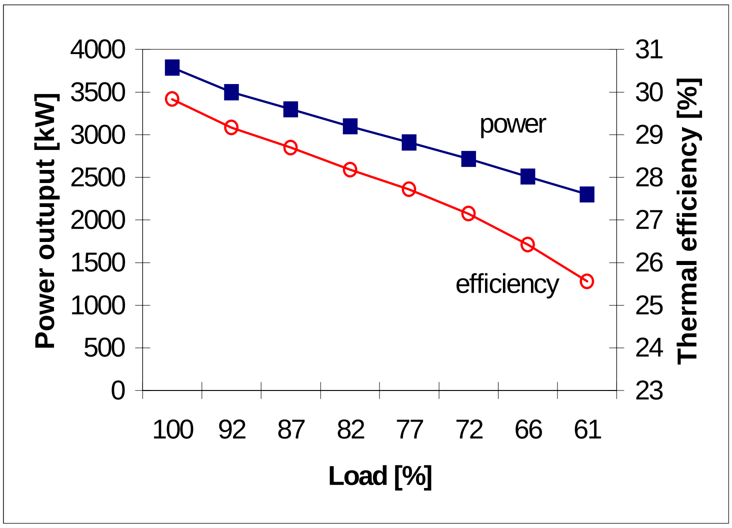

Figure 2. Ruston RM – part load power output and thermal efficiency – 15ºC

The fuel input is calculated through an energy balance at the combustion chamber, imposing that a specified maximum gas temperature should be verified. The maximum gas temperature is even evaluated as function of the known TIT or just specified to control GT power output.

For a given gas turbine, the compressor pressure ratio is only known for the reference ISO case. As long as atmospheric properties change or de-rating is imposed, compressor pressure ratio varies. An essential feature of a GT off-design procedure is its ability in evaluating new compressor pressure ratio. Doing this it is assumed that the flow at the inlet to the gas turbine expander is always choked. This is a very reasonable assumption for a wide range of GT operation conditions (Cohen et al, 1996). The mass flow equation for choked flow of an ideal gas – Eq. (1) - is used to calculate the new pressure ratio.

$$m = pA^* \sqrt{\frac{Mol}{T}} \sqrt{\frac{\gamma}{R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{1}$$

where m is the gas mass flow, p is the total pressure, A* is the critical area where the flow is choked, Mol is the gas molecular weight, T is the gas temperature (absolute), R is the universal gas constant and γ is the ratio of specific heats of the gas. All properties are evaluated at the gas turbine expander inlet section. Equation (1) should be first applied to the reference case, allowing the evaluation of A* that is set constant hereafter.

A second important feature of an off-design procedure is the correct evaluation of compressor air mass flow. In the simulation model here presented, two different situations should be identified: (i) when the compressor operates with air inlet properties equal to the ISO case (15ºC, 1 atm), but with different pressure ratio due to GT de-rating (or lower LHV fuel), and (ii) when just air inlet properties change.

An industrial single-shaft gas turbine, as the one considered here, always operates over the same compressor running-line if it is connected to electric grid (changes in GT rotation are not possible) and, in addition, if atmospheric properties are kept constant. Running-line (RL) is defined by the compressor speed N and the air temperature at the compressor entrance; a running-line 100, or 100%, means that the compressor operates at its nominal speed being the inlet air at the compressor design temperature (in general, 15ºC). Changing inlet air temperature, but keeping the compressor velocity, the new running line can be evaluated by Eq. (2).

$$RL = 100\sqrt{\frac{T_r}{T}} \tag{2}$$

A compressor running-line can define how inlet air mass flow varies with pressure ratio, but compressor maps – and, consequently, their running-lines - are generally not known for a given gas turbine model. The procedure here proposed is just consider a generic running-line; the effect upon the simulation results of three different running lines, taken from generic compressor maps, are then analyzed. The running-lines considered are presented in Figure 3, relating compressor pressure ratio to its reduced air mass flow - RMF. For all three curves, RMF = 2.781 corresponds to compressor operation at 15ºC, 1 atm, when inlet air flow is 16.6 kg/s and compressing pressure ratio is 12.8. RMF is defined according to Eq. (3):

$$RMF = \frac{\dot{m}\sqrt{T}}{p} \tag{3}$$

For a defined compressor running-line, once the new compressor pressure ratio is known (from Eq. 1) it is then possible to evaluate the corrected air flow. The code used do this through a simple function relating the compressor reduced mass flow (RMF) to its pressure ratio. It should be noticed that air flow correction and the evaluation of new compressor pressure ratio is always an interactive procedure.

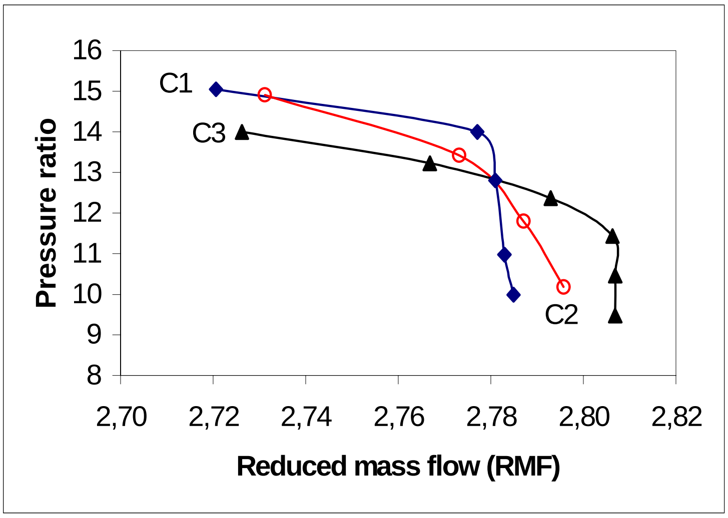

Figure 3. Considered running-lines for three different compressor maps

Changes in ambient temperature correspond to the second off-design case. As seen before, in this situation the compressor no longer operates over the same known running-line, and a new procedure is required to adjust air mass flow. For a constant-speed compressor, constant volumetric air flow equation could be used once its vanes are completely opened. To better fit to the air flow changes, the proposed equation (Eq. 4) incorporates a flow correction proportional to the deviation of inlet air temperature. Rigorously, correction factor CF must be evaluated for each gas turbine. The subscript *r* refers to a reference condition, e.g., the air mass flow at ISO basis.

$$\dot{m} = \dot{m}_r \left(\frac{p}{p_r} \frac{T_r}{T} \right) \left[1 + CF \left(\frac{T - T_r}{T_r} \right) \right] \tag{4}$$

The third correction that should be done to the basic set of equations corresponds to the compressor efficiency adjustment as far as pressure ratio changes. The first option here presented is based on the assumption that compressor isentropic efficiency varies with pressure ratio while polytropic efficiency is essentially kept constant (Cohen et al, 1996). Equation (5) shows the relation between isentropic (γ_i) and polytropic (β) efficiency as function of compressor pressure ratio (PR):

$$\eta_i = \left[\left(PR^{\frac{\gamma-1}{\gamma}} - 1 \right) \Big/ \left(PR^{\frac{\gamma-1}{\gamma\beta}} - 1 \right) \right] \tag{5}$$

In applying Eq. (5) it is first necessary to evaluate polytropic efficiency for the reference solution (ISO), considering a given tuning value of isentropic efficiency (see Table 1). Based on this equation the isentropic efficiency increases as far as pressure ratio drops; this tendency is just coherent for not high flowed GT compressor, particularly when reductions on compressor pressure ratio are caused by reductions on inlet air flow (Ragland, 1998) (e.g., when ambient air temperature rises).

Alternatively, exclusively for off-design cases that correspond to changes of ambient temperature Eq. (6) could be used to adjust compressor isentropic efficiency. This equation is based on GateCycle software (GateCycle, 1998) and imposes a correction as function of running-line changes. Evaluating new RL value, Eq. (2) can be used. As in Eq. (4), the subscript *r* refers to a reference condition, e.g., the compressor efficiency at RL_r (ISO basis). Again, if possible, correction factor ECF should be evaluated for each gas turbine modeling.

$$\eta_i = \eta_{ir}\sqrt{1 - \mathrm{ECF}\left(\frac{\mathrm{RL} - \mathrm{RL}_r}{\mathrm{RL}_r}\right)^2} \qquad (6)$$

Finally, as an alternative to Eq. (5) an option was defined to correct compressor isentropic efficiency for cases in which the running-line is constant but the pressure ratio changes (off-design with GT de-rating, for instance). In this case, a generic polynomial function on PR was defined based on the shape of generic efficiency curves available at the literature (Cohen et al, 1996). For the running-line that corresponds to ISO basis operation, its maximum value is 0.83 (see Table 1) at PR = 12.8.

4. OFF-DESIGN SIMULATION RESULTS AND DISCUSSION

To validate the proposed code for off-design situations Ruston RM was simulated considering variations of ambient temperature and GT part load operation at ISO basis. In the first case, compressor pressure ratio and compressor efficiency change just in response to the alteration of compressor air flow. In the 0-40ºC range it is supposed that no GT control operation, such as de-rating or closing inlet guide vanes, is applied. The difference between simulated cases 1 and 2, as summarized in Table 2, is the function used to correct compressor pressure ratio. Correction factor CF used in Eq. (4) was evaluated as the average value between those that allow error minimization in each simulation case vis-à-vis manufacturer declared power output in the whole range of temperatures.

Table 2. Summary of simulated off-design cases

Off-design case	Case	Equations used		
		Pressure ratio	Air flow	Compressor effic.
Ambient temperature	1	Equation 1	Equation 4 (CF = -0.647)	Equation 5
Ambient temperature	2	Equation 1	Equation 4 (CF = -0.647)	Eq. 6 (ECF=0.20)
Part load	3	Equation 1	Running-line C3	Polynomial
Part load	4	Equation 1	Running-line C2	Equation 5
Part load	5	Equation 1	Running-line C3	Equation 5

Predicted manufacturer results of power output and thermal efficiency presented in Figure 1 were used as basis of comparison. Simulation errors vis-à-vis these predicted results are shown in Figures 4 and 5. General sense, power output and thermal efficiency results in all range of temperatures fit well with predicted values as the maximum error is never larger than 2.5%; simulation errors bellow 3% are always considered acceptable. Case 1, in which Eq. (5) is used to correct compressor efficiency, overestimates power output and efficiency, while case 2 underestimates both parameters beyond 15ºC. As the difference between the two cases is just how compressor efficiency is corrected, it should be concluded that actual compressor efficiency is, approximately, an intermediate value between these two evaluations.

Part load operation of Ruston RM at ISO basis was simulated considering continuos de-rating. Combination of three running-line curves and two different procedures to evaluate compressor efficiency gives the number of cases simulated. Nevertheless, considering the accuracy of simulation results on thermal efficiency and on TET as well, just the three best cases are here presented. These cases are defined according to the information shown in Table 2.

Predicted manufacturer results of thermal efficiency presented in Figure 2 were used as basis of comparison. Simulation errors vis-à-vis these predicted results are shown in Figures 6, while errors on TET are presented in Figure 7. For all non reported cases errors are slightly larger than those that correspond to case 3. From the point of view of efficiency, simulation results can be considered good, but from the point of view of TET, and especially for lower loads, evaluation is just acceptable. Simulations 4 and 5 give better results, but the use of Eq. (5) to adjust compressor efficiency is questionable in this case as isentropic efficiency continuously increase with reduction of pressure ratio. The use of a polynomial function to adjust compressor efficiency is more coherent, but this function should be defined with more precision. On the other hand, the use of running-lines seems to be reasonable, at least to give the right tendency to air flow correction. However, it is not possible to previously know which running-line will fit better in each case.

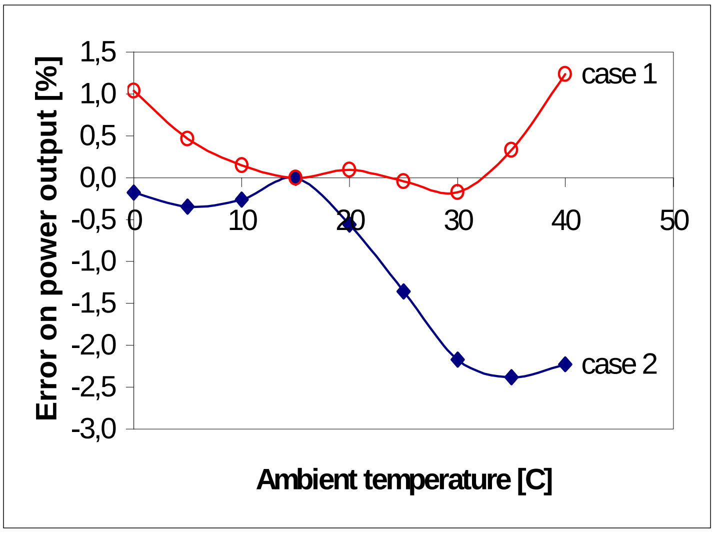

Figure 4. Simulation errors on power output

Part load off-design simulation deviates more from tuning solution than the case that corresponds to changes in ambient temperature. In this sense, the model could be further improved in some aspects, especially on the estimation of cooling air correction, correction of expander isentropic efficiency and evaluation of TIT in response to GT de-rating.

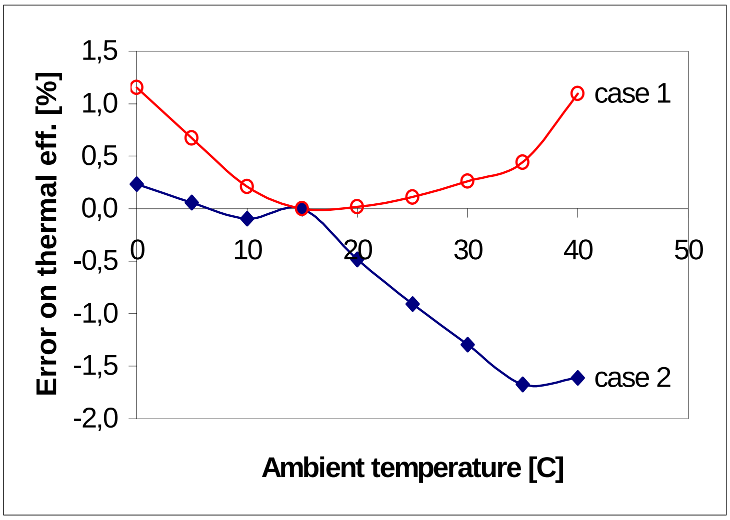

Figure 5. Simulation errors on thermal efficiency

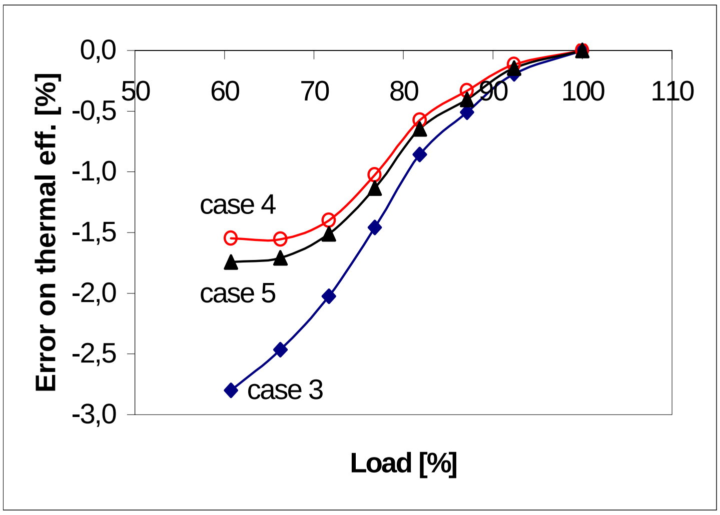

Figure 6. Part load operation – simulation errors on thermal efficiency

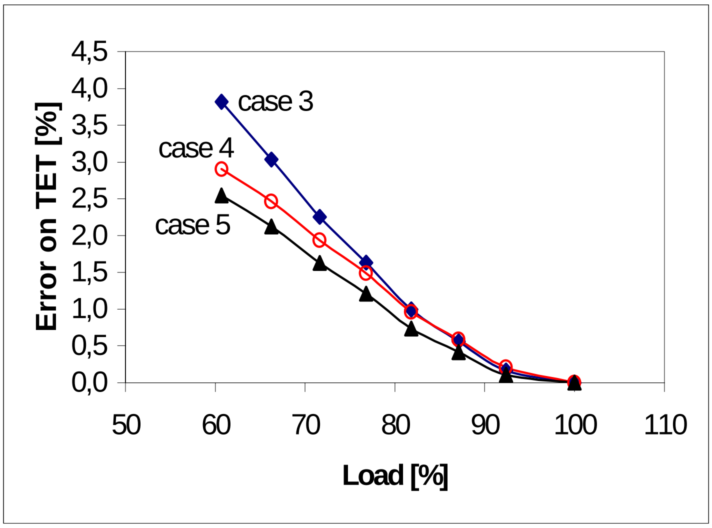

Figure 7. Part load operation – simulation error on TET

5. CONCLUSIONS

The code presented in this paper leads to quite reasonable results on off-design GT simulation. Deviations regard manufacturer performance data are larger as far is off-design condition from ISO point, i.e. higher ambient temperature and lower GT load. Nevertheless, for the majority of conditions simulated, errors are lower than 3%, that is the acceptable error in a commercial software.

Acknowledgments

The code presented in this paper was partially developed during a post-doctoral season at the National Renewable Energy Laboratory - NREL, Golden, USA, from 1997-98. The author is grateful to Fundação de Amparo à Pesquisa do Estado de São Paulo - FAPESP - for the financial support received at that time.

REFERENCES

Cohen H, Rogers G and Saravanamuttoo H., 1996, Gas Turbine Theory, Essex, Longman Group Limited.

Desideri U., 1994, Performance analysis of gas turbines operating at different atmospheric conditions, Proceedings of ASME Cogen Turbo Power '94, October, Portland, pp. 485-492.

GateCycle, 1998, Users Manual of GateCycle for Windows, Enter Software, Palo Alto, USA.

Gas Turbine World, 1987, Ruston working on a pair of small high efficiency engines, Gas Turbine World, Vol. 17, n. 3, pp. 48-49.

Ragland, T.L., 1998, Industrial gas turbine performance uprates: tips, tricks, and traps, Journal of Engineering for Gas Turbine and Power, Vol. 120, October, pp. 727-734.