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PERFORMANCE ANALYSIS AND PREDICTION OF COMPRESSOR FOULING CONDITION FOR A TWIN-SHAFT ENGINE

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ABSTRACT

Performance of a twin-shaft Industrial Gas Turbine (IGT) at fouling condition is simulated via a gas turbine model based on fundamental thermodynamics. Measurements across the engine during compressor fouling conditions were considered to validate the outcomes. By implementing correlation coefficients in the compressor model, the performance of the IGT during compressor fouling conditions is predicted. The change in the compressor air flow and the compressor efficiency during fouling conditions is estimated. The results show that the reduction of air flow rate is the dominating parameter in loss of generated power under fouled conditions. The model can provide an insight into the effect of compressor fouling conditions on IGT performance.

INTRODUCTION

Application of Industrial Gas Turbines (IGTs) for power generation has surged in recent years. Modularity and extreme flexibility are among several benefits enabling IGTs to remain the logical choice for a majority of new power plants in the coming years. Competitiveness of the market, increasing fuel prices and economical challenges have motivated the business to put more emphasis on higher operating efficiencies and reduced emission levels to reduce the costs. Operations and Maintenance (O&M) costs constitute an important portion of the total lifetime costs of a typical gas turbine and therefore better maintenance strategies would help increasing the efficiency and thus reducing the costs.

Healthy operation of a gas turbine is the result of the fine-tuned function of many different components. Degradation on any of these components in isolation can result in the performance reduction of the entire gas turbine. Fouling is one of the most common reasons for engine performance deterioration and it affects all open-cycle gas turbines. The level of fouling depends on several factors but the main ones include the level of dirt and particles in the environment, quality of air filtration and, to some extent, the power output of the gas turbine. Deposited dirt and dust particles on the blades add to the surface roughness and affect the compressor performance by altering the shape of the airfoil. Modern turbine compressors are even more susceptible to fouling [1] due to the physical characteristics of the blades which are now comparatively thinner to achieve higher pressure ratios. Fouling can be controlled by appropriate air filtration system, and often reversed to some degree by detergent washing of components.

Due to its importance, fouling condition has been subject of several studies. Effects of compressor fouling to the performance of industrial gas turbine and their possible control strategies are investigated in [2]. It is concluded that fouling behaviour is influenced by inlet air filter selection and maintenance. A judicious combination of off-line and on-line cleaning usually provides the best results in helping operators fight this common and insidious operating problem. Close monitoring of compressor performance can help optimise compressor washing regimes and improve plant profitability. The influence of fouling on the performance characteristics of an axial compressor model is investigated in [3] by developing an analytical model. A linear fouling model was introduced in [4] that simulates the progressive build-up of contaminants in the compressor by modifying the appropriate stage flow and efficiency characteristics in a stepwise methodology. A preliminary study into how severely a given level of fouling will affect engines of different size then concluded that stage loading may be a more critical parameter.

This paper considers analysis of the compressor fouling condition at both high and low operating points (HOP and LOP respectively) in twin-shaft engine design where one shaft is used for the gas generator while the other is used for the free turbine that powers a driven load, and can operate at speeds independent of the gas generator. Based on the non-linear trends for the sensor measurements (pressure, temperature, rotational speed across the IGT), inputs are fed into a SIMULINK model to estimate the compressor efficiency and the discharged airflow in fouling conditions.

METHODOLOGY

Sensor measurements for a twin-shaft IGT indicating a performance reduction due to compressor fouling condition are considered. The measurements are obtained for three cycles of steady state operation at high and low operating points by various physical sensors across the engine (e.g., temperature, rotational speed, and pressure). The three cycles are chosen such that they indicate operation of the engine from clean to a highly fouled compressor condition. Measurements obtained from the engine exhibit high frequency variations and therefore a filtering stage is necessary before the data is further processed. Here, a low pass finite impulse response filter with a length of 200 minutes is applied to smoothen the measurements.

Fouling conditions result in the reduction of air flow rate and the compressor efficiency which leads to higher compressor exit temperatures. Figures 1 and 2 show the normalised temperature measurements of the air discharged from the compressor and engine's fuel demand for the considered operational cycles considered at LOP and HOP respectively.

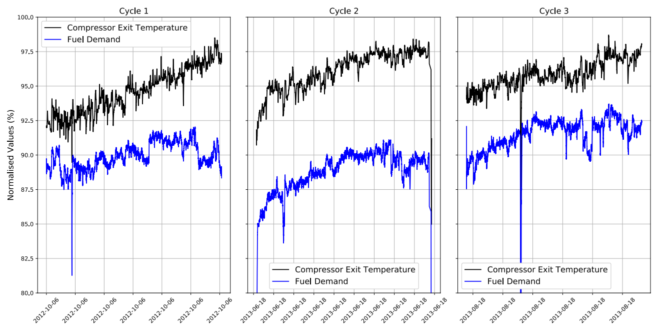


Figure 1 Normalised measurements of engine parameters at LOP for three cycles.

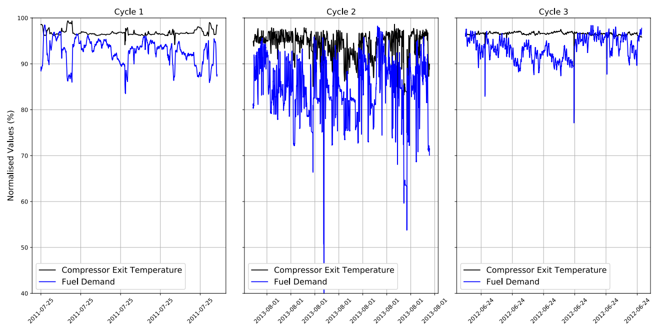


Figure 2 Normalised measurements of engine parameters at HOP for three cycles.

To estimate the reduction in airflow and efficiency, a SIMULINK model [5] is used that simulates the performance of a twin-shaft engine at optimal conditions (i.e., no degradation in any components including the compressor). The model is shown in Figure 3 and the component map for each module is also implemented in the modelling architecture.

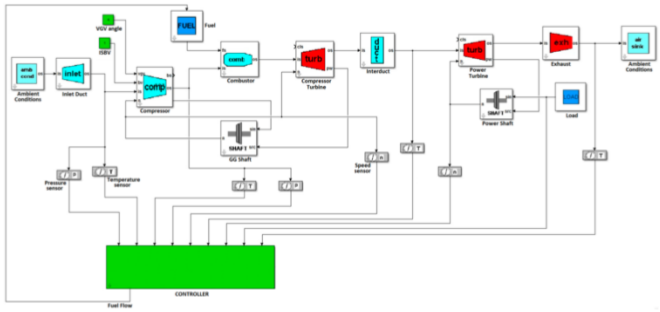
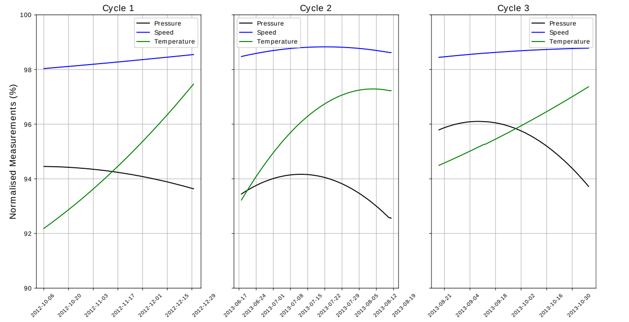


Figure 3 The SIMULINK model.

At the compressor, air enters the inlet at ambient conditions and is delivered to the combustor to be mixed with fuel for combustion which drives the compressor turbine through a bearing-shaft.

Gas path components that comprise the gas turbine model are divided into two groups: gas generator and power turbine shaft. Gas generator itself consists of the compressor, combustor, compressor turbine and the shaft connecting the compressor and the high pressure turbine. The generator is governed by a thermodynamic process from the air intake in the compressor to the hot gas entering the compressor turbine. On the other hand, power turbine comprises the power turbine, interduct, load, and the mechanical shaft connecting the power turbine and load. As inputs, the model requires, load and the ambient conditions (temperature and pressure) for the air entering the compressor.

Figure 4 Normalised measurements used as input for the SIMULINK model at LOP.



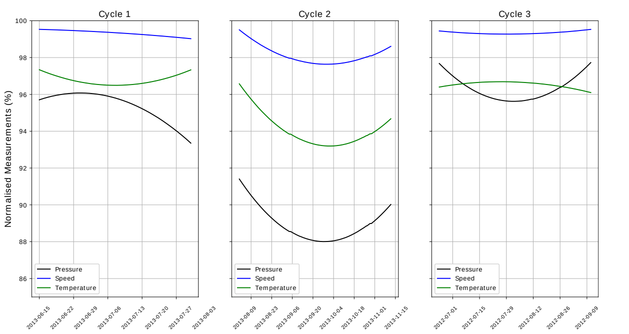


Figure 5 Normalised measurements used as input for the SIMULINK model at HOP.

Fuel demand is worked out via a controller unit that considers as inputs the shaft speeds and measurements at several stations across the engine.

A simplified representation of fuel flow control logic has been considered. The following control limiting loops have been implemented in simulation model:

- Power Turbine speed controller
- Turbine Temperature limit controller
- Power output limit

The outputs of control loops are the input in a minimum value selector, so that the loop which takes control, is the one whose output is the lowest. Power turbine speed control is the main control loop during normal operating conditions, and input to this control logic is the PT speed deviation. Temperature limit controls the fuel flow to limit the turbine operating temperature, and usually is active when engine is running at high operating point. Power limit provides a maximum continuous fuel flow limit for the engine for the present ambient temperature. This limit may however be exceeded for a short duration. (See [5] for a more detailed description of the controller).

Analysis of the fouling condition requires performance simulation of the IGT at fouling conditions over certain periods. To reduce computational complexity during the simulation, the model is converted to a stand-alone executable which enables simulation of 83 days operation in only 65 hours.

Parameter estimation for fouling analysis

The compressor module of the model is governed by a performance map that relates the rotational speed, pressure ratio, mass flow rate, and efficiency at clean conditions. Rotational speed and pressure ratio are injected as inputs to the compressor map to calculate airflow and the discharge temperature from the pre-defined efficiency. Since the map only considers clean condition operation, an empirical analysis has been carried out to introduce a pair of coefficients for the compressor module to enable simulation of the fouling conditions by reducing the air flow and compressor efficiency (Figure 7).

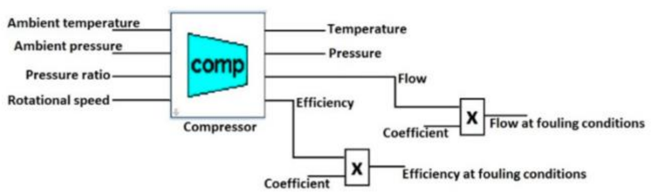


Figure 6 Coefficients to be tuned for simulation of fouling conditions.

The initial values for the coefficients were estimated by analysis of several measurement points along the full operation range. Then an optimisation technique (available in SIMULINK Design Optimisation Toolbox) is used to fine-tune the initial estimations considering several parameters across the engine.

Moreover, to enable the simulation, a decreased pressure ratio is considered in the input which is calculated from the pressure drop across the system and the pressure at the ambient conditions.

Model validation

Field data from twin shaft IGTs at fouling conditions were considered to validate the simulation after the coefficients were introduced to adjust the compressor's performance. Engines were operated at steady-state conditions under partial and full load. The field data as well as the simulated data were normalised with respect to parameters at clean conditions (day 0). Figures 8 through 10 show the comparison between normalised measured and simulated compressor exit pressure, interduct pressure and exhaust temperature at low operating point respectively whereas figures 11 through 13 show the same parameters at high operating point. It is easy to verify that the model can predict effects of fouling in the chosen engines. At both operating points, interduct and compressor exit pressure drop as the fouling increases while the exhaust temperature increases due to the drop in the compressor efficiency and airflow.

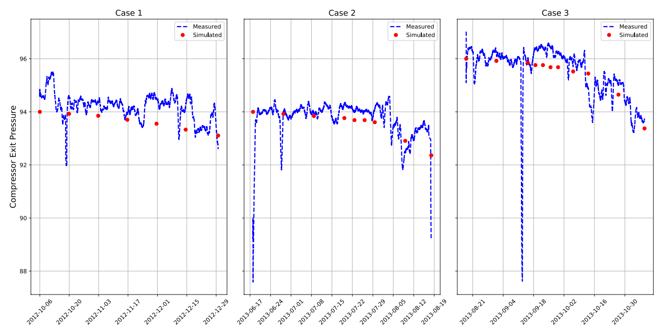


Figure 7 Comparison between normalised simulated and measured compressor exit pressure at low operating point.

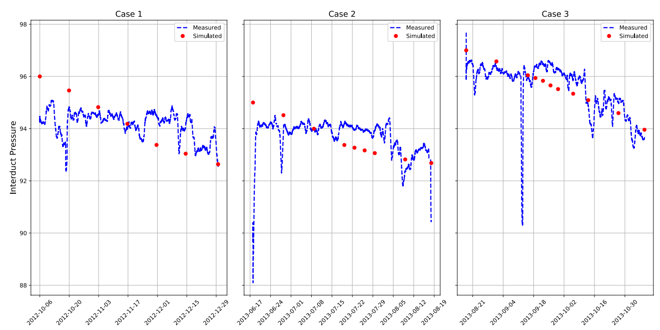


Figure 8 Comparison between normalised simulated and measured interduct pressure at low operating point.

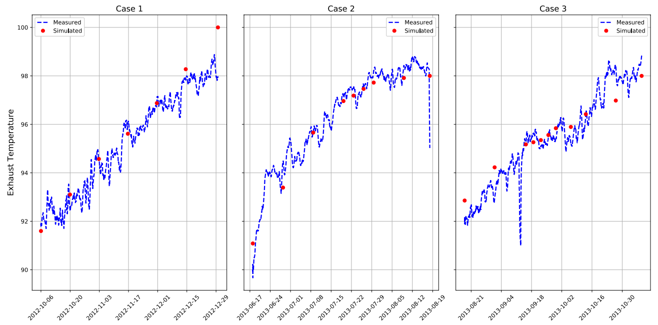


Figure 9 Comparison between normalised simulated and measured exhaust temperature at low operating point.

The comparison between measured and simulated data is carried out based on the nonlinear trends extracted from the field data. Tables 1 to 3 show this comparison for the three cases examined at low operating point and tables 4 to 6 illustrate the error for high operating point simulation. At low operating point and highly fouled conditions the maximum simulation error for interduct pressure is 2 %, 0.2%., and 0.06% for cases 1, 2, and 3 respectively. While, at high operating point and highly fouled conditions the maximum simulation error for interduct pressure is 0.1%, 3%., and 0.87% for cases 1, 2, and 3 respectively.

Table 1 Simulation error for case 1 at low operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
2012/10/06 00:00:00	0.057970803	0.005998376	0.092930433
2012/10/19 22:00:00	0.011410842	0.438562809	0.290608966
2012/11/02 19:20:00	0.007505527	1.146571641	0.523207786
2012/11/16 16:40:00	0.033025101	1.861831327	0.382042206
2012/11/30 14:00:00	0.030833721	2.612760604	0.547912577
2012/12/14 11:20:00	0.065402444	2.587402916	0.921874003
2012/12/29 21:59:00	0.026663149	2.068718632	1.576636967

Table 2 Simulation error for case 2 at low operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
2013/06/18 00:00:00	0.802236266	1.395538237	0.789237996
2013/06/28 09:58:00	0.03152212	0.345115886	0.588807274
2013/07/08 19:58:00	0.065199441	0.091566392	0.040773496
2013/07/19 05:58:00	0.036437271	0.722185738	0.120003646
2013/07/22 17:18:00	0.151999003	0.769377089	0.036468634
2013/07/26 04:38:00	0.040995427	0.756813797	0.058635829
2013/07/29 15:58:00	0.0185275	0.715648288	0.114767813
2013/08/09 01:58:00	0.113897916	0.344323511	0.008435448
2013/08/18 00:00:00	8.54729E-05	0.201591191	0.016835963

Table 3 Simulation error for case 3 at low operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
2013/08/18 00:00:00	0.04	0.27	0.019372768
2013/08/31 21:18:00	0.043	0.041	0.1978
2013/09/15 08:44:00	0.019	0.515	0.4233
2013/09/18 20:04:00	0.087	0.596	0.5910
2013/09/22 07:24:00	0.062	0.668	0.7465
2013/09/25 18:44:00	0.108	0.802	0.7655
2013/09/29 06:04:00	0.057	0.885	0.7063
2013/10/06 04:44:00	0.065	0.889	1.110209932
2013/10/13 03:24:00	0.102	0.880	1.038804437
2013/10/27 00:44:00	0.132	0.515	1.535377787
2013/11/08 00:00:00	0.035	0.063	1.679827771

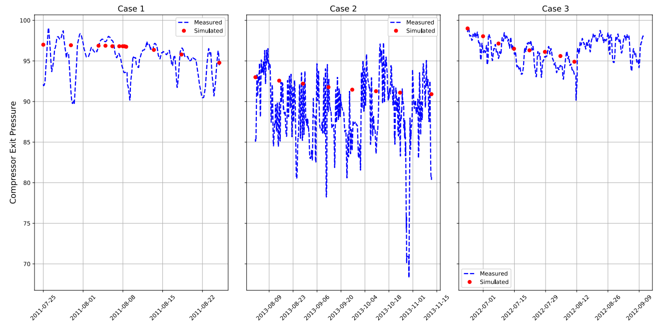


Figure 10 Comparison between normalised simulated and measured compressor exit pressure at high operating point.

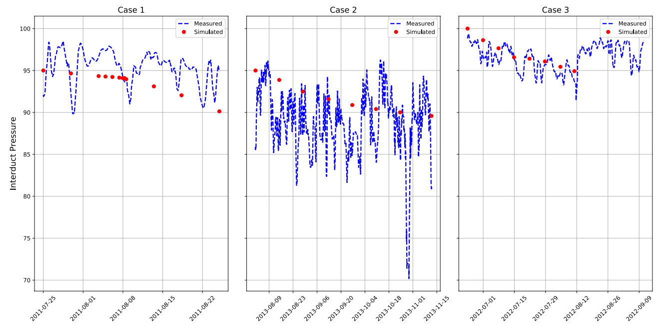


Figure 11 Comparison between normalised simulated and measured interduct pressure at high operating point.

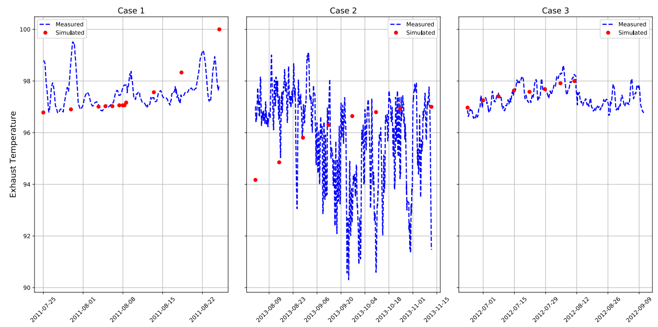


Figure 12 Comparison between normalised simulated and measured exhaust temperature at high operating point.

Table 4 Simulation error for case 1 at high operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
2011/07/25 00:00	0.06557377	1.765695067	0.197380226
2011/07/29 20:38	0.054224228	1.595028833	0.164969151
2011/08/03 17:18	0.017723719	1.272629833	0.150555285
2011/08/04 22:28	0.023004008	1.215578226	0.155909037
2011/08/06 03:38	0.032245812	1.163226166	0.179162167
2011/08/07 08:48	0.013600188	1.09080461	0.166609061
2011/08/07 23:23	0.000269953	1.044530702	0.17823032
2011/08/08 06:41	0.007570677	1.05055897	0.184046571
2011/08/08 13:58	0.0495227	0.917233947	0.08248765
2011/08/13 10:49	0.153412633	0.279030455	0.235175441
2011/08/18 07:29	0.035406064	0.159724455	0.919881207
2011/08/24 23:14	0.932579852	0.167907405	2.450965686

Table 5 Simulation error for case 2 at high operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
01/08/2013 00:00	0	0.53428317	0.504231947
14/08/2013 21:18	0.250675561	0.353668716	0.584860413
28/08/2013 18:38	0.435895375	1.520127476	2.308206173
12/09/2013 16:44	0.688562985	2.181762867	3.642200513
26/09/2013 14:04	0.808727618	2.636006096	4.65365358
10/10/2013 11:24	0.79553161	2.85063447	5.056739761
24/10/2013 12:50	0.782278536	2.975403305	4.763655794
11/11/2013 21:06	0.720035676	3.062963332	2.998099192

Table 6 Simulation error for case 3 at high operating point.

Time	Compressor Exit Pressure (%)	Interduct Pressure (%)	Exhaust Temperature (%)
2012/06/24 00:00	0	0	0.07208506
2012/06/30 22:38	0.049637948	0.345709935	0.109509086
2012/07/07 21:18	0.179891605	0.619885505	0.042994231
2012/07/14 19:58	0.368365651	1.233392926	0.069981317
2012/07/21 18:38	0.188487295	1.018055751	0.167443833
2012/07/28 17:18	0.024505587	0.951330033	0.2218735
2012/08/04 15:58	0.064336425	1.055261935	0.129829981
2012/08/10 22:16	0.090436213	0.873498379	0.148275605

RESULTS AND DISCUSSION

Air flow rate, pressure ratio, and compressor efficiency decrease as a result of fouling [6]. The reduction percentage for air flow and compressor efficiency are calculated by selecting discrete points across the full range of measured data. Figures 14 to 16 show the reduction in compressor discharge air flow and compressor efficiency from day 0 to a day before the engines were washed for 3 cases at low operating point. It is immediately apparent that the reduction in air flow is more prevalent than the compressor efficiency in all cases. This has also been reported in [6]. The ratio between delta flow and delta efficiency, also decreases which is inversely proportional to the error between engine and simulated data for compressor delivery temperature. During fouling conditions, an increase in fuel is required to compensate the loss of generated power during engine operation at low operating point with an attendant reduction in overall cycle thermal efficiency. However, during high power engine operation, fuel demand is limited by the maximum turbine inlet temperature. A lower air flow rate during high engine operation and fouling conditions is expected, as shown in Figs. 17, 18 and 19.

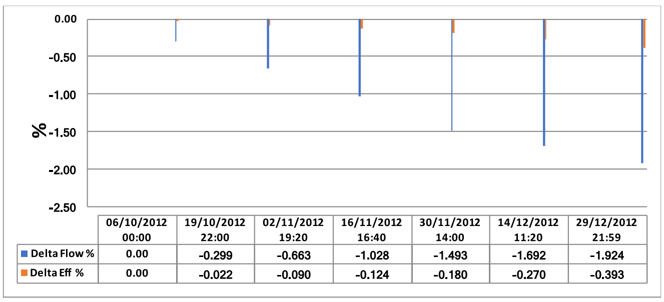


Figure 13 Reduction in air flow and compressor efficiency during fouling conditions and low operating point for case 1.

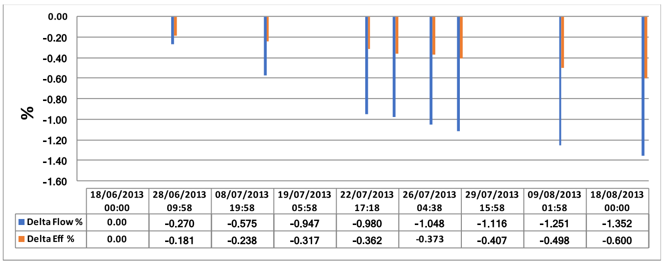


Figure 14 Reduction in air flow and compressor efficiency during fouling conditions and low operating point for case 2.

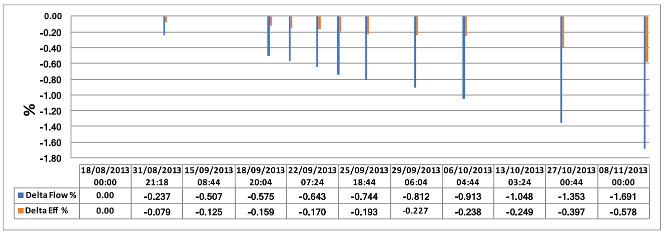


Figure 15 Reduction in air flow and compressor efficiency during fouling conditions and low operating point for case 3.

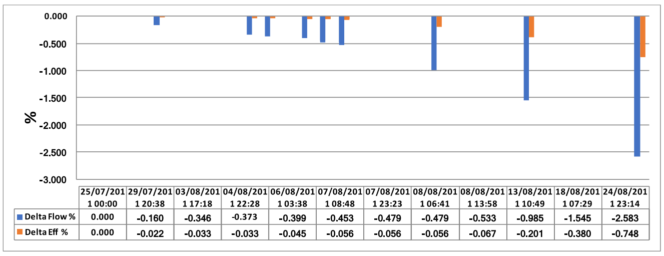


Figure 16 Reduction in air flow and compressor efficiency during fouling conditions and high operating point for case 1.

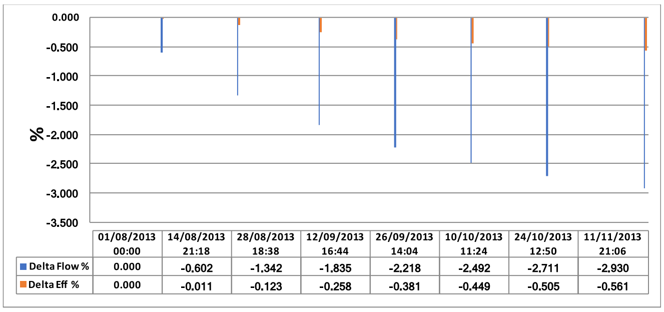


Figure 17 Reduction in air flow and compressor efficiency during fouling conditions and high operating point for case 2.

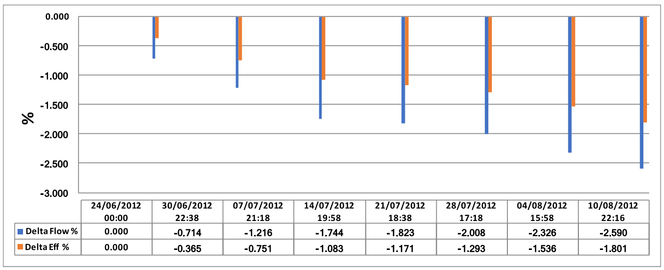


Figure 18 Reduction in air flow and compressor efficiency during fouling conditions and high operating point for case 3.

CONCLUSIONS

In the presented paper, the performance of twin-shaft IGTs at fouling conditions is predicted via a Simulink Model at low and high operating points. Measurements such as temperature, pressure, and rotational speed across two gas turbines were considered to validate the capability of the model for estimation of compressor efficiency and airflow rate. The IGTs were continuously running whether as a mechanical drive or a power generator unit at steady state conditions over certain periods of time from a compressor clean condition to a highly-fouled condition. A pair of coefficients were introduced to simulate effect of fouling. An empirical analysis based on the proposed coefficients from the measurements enabled the prediction of the changes in physical parameters of the IGT system at fouling conditions. The results demonstrate that the decrease of power generation and performance in an IGT at highly fouled conditions is attributed mainly to a reduced air flow rate.

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