# Investigating of Turbine Blades Geometrical Change Effects on Dynamic Performance of IGT-25 Gas Turbine

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## ABSTRACT

In the paper, the effect of gas generator turbine blades' geometrical change has been studied on the overall performance of a twin-shaft 25MW gas turbine with industrial application, under dynamic conditions. Geometrical changes include change of thickness and height of gas generator turbine blades which in turn would result in the change in mass flow rate of passing hot gas, as well as isentropic efficiency in each stage of the turbine. Gas turbine modeling in the paper is zero-dimensional and takes place with consideration of dynamic effects of volume on air compressor components, combustion chamber, gas generator turbine, power turbine, fuel system, as well as effects of heat transfer dynamics between blades, gas path, and effects of operators on inlet guide vanes, fuel valves, and air compressor discharge valve. In the mathematical model of each of the components, steady-state characteristics curves have been used, extracted from 3-Dimensional computational fluid dynamics (CFD). To do so, characteristic curves of the first and second stages of the four-stage turbine have been updated through 3-D fluid dynamic analysis so that the effect of geometrical changes in turbine blades would be applied. Results from effects of these changes on characteristics of transient gas flow including output power of gas generator turbine and power turbine, inlet and outlet temperatures of turbine stages, as well as air and fuel mass flow rates have been provided from the start-ups until reaching the nominal load would be achieved.

**Keywords:** *Gas Turbine; Geometrical Changes; Blade; Dynamic Conditions*

## Introduction

Dynamic behavior modeling of gas turbines is of special importance due to the complicated nature of the device and fast dynamics of its destructive phenomena. Under the dynamic and transient condition, some parameters such as rated speed, flow rate, and pressure are prone to faster changes, compared to inertia parameters such as temperature. Considering the above, the real control system has to provide a quick response as for assuring safe behavior of this set [1]. Work conditions of gas turbines are transient or outside design point, most of the times. Therefore, analyzing transient condition due to the function of gas turbines under various dynamic conditions in a normal work cycle including putting into operation, taking speed and off mode is of importance. Under such conditions, there is a high probability of existing stresses in different sections to cross authorized limit and/or become close to instable conditions such as Surge and Stall in the compressor.

Also, the lack of enough cooling flow rate of blades may result in an increase of thermal-pressure stresses in blades located in the hot section. From among other parameters, reference could be made to changes in fuel injection in the combustion chamber, followed by changes in flame temperature which has to be considered in analyzing transient behavior. Different factors can change the expected work condition of the gas turbine, the most important of which includes geometrical changes in different sections of gas path in the motor of gas turbine which would be mainly addressed in the paper.

Dynamic modeling of an approximately 25MW turbo generator is provided in the text. Turbo generator includes a 10-stage axial compressor, a two-stage gas generator turbine, and a two-stage power turbine and the set is used to rotate the generator. In a developed dynamic model, volume dynamics, heat transfer dynamics, shaft dynamics, operator dynamics, and cooling of blades have been taken into consideration. The full control system of the turbo generator also has been coupled with the dynamic model of motor [2].

## Modeling the Components

Turbo generator's components have been simulated in the 3D form with the help of computational fluid dynamics (CFD); and, simulation results include performance curves used in zero-dimensional cycle modeling. Any kind of change in work condition of each of the main parts in the gas turbine could be presented through modification of the aforementioned performance curves. In continuation, the modeling method of the main components of the gas turbine would be provided [3].

**Compressor:** Pumping capability in compressor and turbine is defined based on classic relation between pressure (expansion) ratio and modified gas flow along fixed speed lines. Also, turbine and compressor's efficiencies would be determined through the relationship between temperature and pressure ratios along fixed speed lines [4]. In axial compressor modeling, the same characteristic curves have been used. Characteristic curves are used as the ratio of dimensionless pressure $(\frac{PR}{PR_D})$ based on dimensionless mass flow $(\frac{\frac{\dot{m}\sqrt{T_0}}{P_0}}{(\frac{\dot{m}\sqrt{T_0}}{P_0})_D})$ and dimensionless efficiency $(\frac{\eta-\eta_D}{\eta_D})$ based on dimensionless pressure. Using the method is of some advantages, including a simple method of solving, and lack of need to geometrical details of the motor. The characteristic curve obtained for compressor would be created by axial compressor modeling through 3D streamline curvature method, capable of approximating compressor performance in low speeds up to 25% of designed rated speed[4]. Conditions at zero speed for turbine and compressor would be defined with zero mass flow and unit pressure ratio; and, fixed speed lines between zero speed and 25% of designed rated speed would be achieved from interpolating these two conditions [5, 6].

**Combustion chamber:** Pressure drop inside chamber and diffuser within the full operational bounds of the turbine would be estimated from pressure drop curve based on loading. These curves have been created based on the 3D dissolution of flow inside the chamber through software packages provided for solving computational fluid dynamics. Stagnation pressure drop inside the chamber also has been calculated with CFD simulation and approximated through a relationship [7]. Moreover, thermal energy resulted from combustion can be obtained as heat transfer into the combustion chamber, and energy balance relationship as follows [8, 9]:

$$Q = \eta_{comb}\dot{m}_f(LHV + \Delta H_{T_0 - T_f}) \tag{1}$$

Where, 0 is reference point showing low heat value (LHV), $T_f$ is flame temperature, $\dot{m}_f$ is mass flow rate of fuel, and $\eta_{comb}$ is combustion efficiency obtained through 3D analysis, entering in zero-dimensional models as combustion efficiency curves, based on loading parameter [10]. Loading parameter is a relationship based on temperature, pressure, and mass flow rate of inlet air into the combustion chamber, calculated through CFD simulation [11].

**Turbine:** It is modeled like a compressor; however, the difference is characteristic curves of turbine almost being fit, under different rated speeds. However, high cooling technology in today's axial turbines turns to model to a complicated procedure. Air undertow from the compressor will enter rows in the stage so that turbine blades would be cooled; and, it affects specifications of the stage. So, a row by row cooling prediction method has to be considered for the turbine so that its performance specifications would be obtained. In the cooling method of the turbine, integrated modeling of the rotor would be made and outlet flow after the rotor would be mixed with cooling flow. To calculate the cooling flow rate of each row, its type of technology including a layer or inner cooling has to be specified. Then, the maximum gas temperature would be calculated, considering the pattern factor in the chamber. If temperature crosses authorized temperature of the blade surface, cooling flow for that row would be taken into consideration [12]. Under the outside design and dynamic conditions, expansion pressure ratio would be obtained through the following relationship:

$$\frac{\dot{m}_{in}\sqrt{T_{T,in}}}{P_{in}} = K\sqrt{1-(\frac{P_{out}}{P_{in}})^2} \tag{2}$$

Where, *in* and *out* are indicative of inlet and outlet conditions and K is resulted from a design point, and will remain fixed outside this point [4]. Cooling flow rate entering rows of the blade in turbine under outside design and dynamic conditions would be calculated through the following formula:

$$\dot{m}_c = K_c P_c \sqrt{\frac{2(1-\frac{P_T}{P_c})}{RT_c}} \tag{3}$$

Where, (c) is related to cooling fluid, (T) is related to gas flow inside the turbine, and R is gas global constant [4].

### Gas Dynamics

Gas dynamics is one of the important phenomena in the transient condition of the turbine which considers mass, energy, and the momentum stored in the chamber of main components. In fact, the problem is equal to the inertia of motion inside a pipe. Inertias resulted from gas dynamics in the compressor, combustion chamber, and turbine would result in the behavioral difference between outside design point and dynamic conditions of performance. Considering the phenomenon in designing control system is important for such instabilities as Surge and Humming to be taken into consideration [8].

Gas dynamics would be modeled through one-dimensional equations of mass, energy, and conservation of momentum for every certain component from among main components of motor that are placed within control volumes (plenum).

Equations related to mass and energy conservation for control volume are written as follows:

Continuity
$$\dot{m}_{in} - \dot{m}_{out} = \frac{d}{dt}(m_{vol}) = \frac{V}{\sqrt{RT}}\frac{dP}{dt} \tag{4}$$

Momentum
$$H_{in} - H_{out} + \dot{q} = \frac{d}{dt}(m_{vol} \cdot C_v \cdot T) = C_v(\frac{Tdm_{vol}}{dt} + m_{vol}\frac{dT}{dt}) \tag{5}$$

Where, *in* and *out* respectively are inlet and outlet conditions into and from control volume, $m_{vol}$ is remained mass in control volume, and $C_v$ is gas thermal capacity in a fixed volume. Solving differential equations, equations, and outputs related to these two equations i.e. temperature and pressure would be calculated. Program's input would be the output from the chamber.

### Integration of turbo-compressor dynamic models in the form of a single model

In figure (1), the integration method of turbo generator's various dynamics is shown. Gas dynamic phenomenon in control volumes would be taken into consideration. As far as the compressor is divided into four groups, four control volumes are considered for it. Combustion chamber has been divided into three groups; and, the control volume is shown in figure (1). In fact, three control volumes have been taken into consideration. The turbine containing two high and low pressure sections is a four-stage turbine as a whole, with four control volumes considered for it [11].

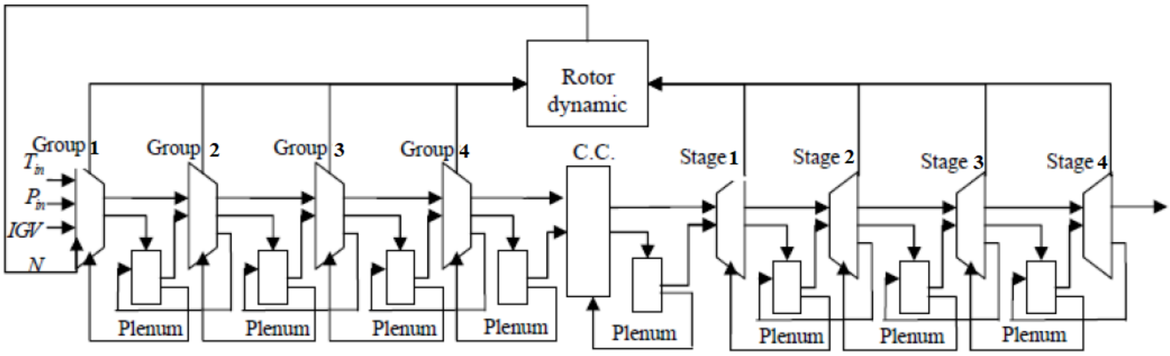


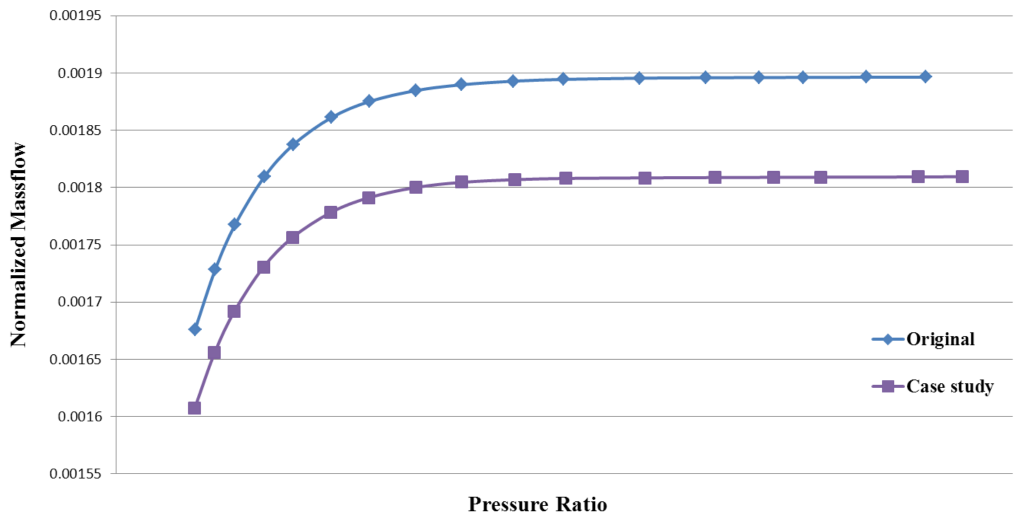

**Figure 1-** Integration method of gas turbine different dynamics

Heat transfer and stress dynamics in the compressor, combustion chamber, and turbine have been separately considered in the developed model. To study the effect of changes in each part of the model on the model as a whole, each of control volumes would be linked to other volumes, through usage made of compatibility equations. At the end of each time step, gas turbine and passing gas flow characteristics would be converged so that all compatibility equations would be satisfied.

**Geometrical Changes in Turbine**

In the study, geometrical changes in relation to blades in the first and second stages of the motor's turbine have been studied. Geometrical changes can include changes in thickness and height of blades. Changes in blade thickness can be due to technological manufacturing restrictions, and/or sediments formed by air pollution on blades. In general, effective mechanism of these changes includes cross-section change in the channel through which hot gas flows pass through blades, as well as the roughness effect of the layer located over gas flow. In the present study, an approximate increase of 0.5mm in the thickness of sediments on turbine blades would be resulted in 9.15% reduction in channel area where hot gases are passing through.

As previously stated, the problem solving strategy in zero-dimensional models is using performance curves. Accordingly, applying new geometry of blades in the first and second stages of the turbine would be updated through 3D CFD simulation of performance lines. Simulation results are presented in figures (2) and (3), respectively for a dimensionless mass flow rate of gas passing through gas generator turbine and polytropic efficiency turned dimensionless, based on design conditions.




**Figures 2** – The line specifying the change of dimensionless mass flow rate through pressure ratio in gas generator turbine (90% nominal speed)

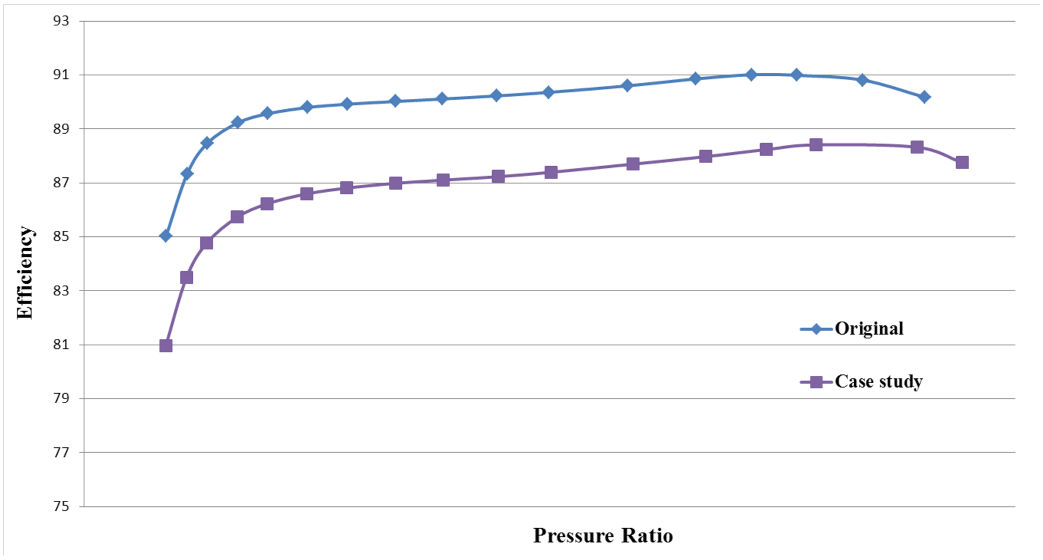



**Figure 3-** The line specifying the changes of polytropic efficiency through pressure ration in the gas generator turbine.

Considering the above figures, the differential value of passing mass flow rate and efficiency of turbine stages would be specified in terms of initial and transformed blades. Average difference percentage in different pressure ratios shows 4.09% and 3.26% decrease, respectively in mass flow rate and polytropic efficiency, compared to the initial state of blades. Overall performance results of turbo generator will be studied in the next section, applying new curves.

## Results

Figure (4) shows performance lines of compressor along with speed lines from the moment of speeding up, for initial blades and those transformed. As it is observed, under a fixed pressure ratio where gas cross-section through turbine blades is reduced; the value of flow rate passing through compressor also would be reduced. This reduction is more significant in higher pressure ratios (higher rated speeds). Gas flow pressure reduction rate when passing through transformed blades would be reduced. Effect of the relative increase in downstream pressure of turbine blades compared to initial blades on an air compressor is a performance closer to Surge line which would be resulted in an increase of instabilities.

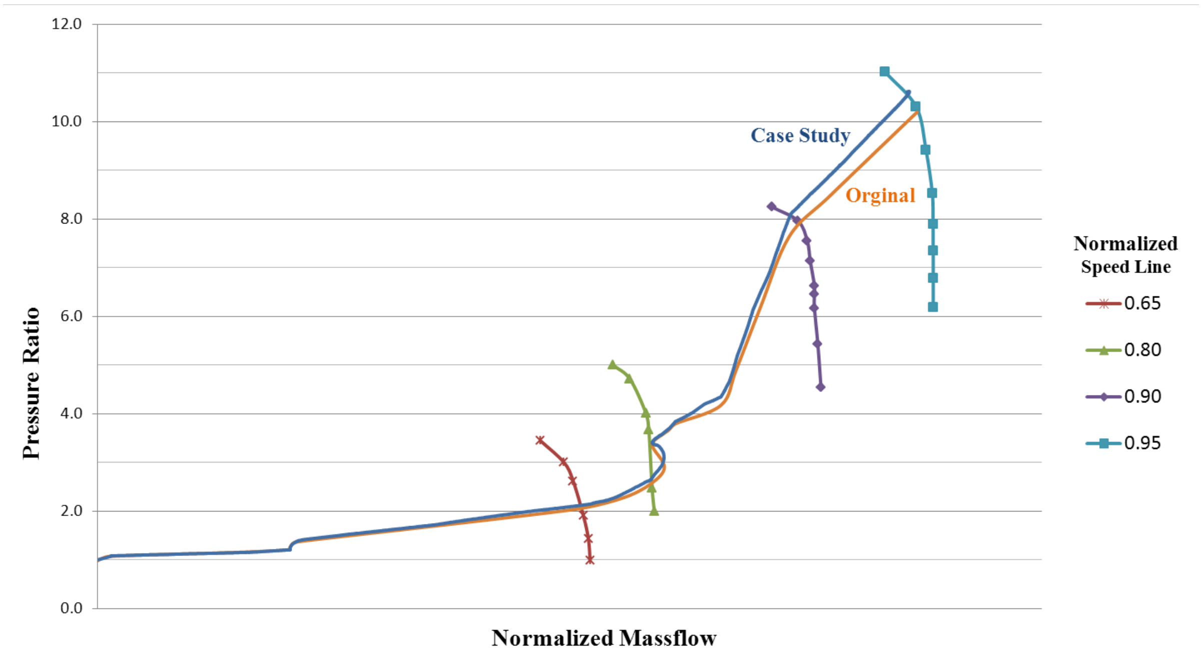



**Figure 4-** Comparing performance lines of initial and transformed blades.

Figures 5 and 6 show changes of air and fuel mass flow rates respectively, from the beginning of the loading process up to 95% of the nominal speed. Because of the reduction of the flow cross-section passing through turbine blades, reduction of air mass flow in terms of transformed blades seen in figure (5) would be expected. Required exit power from turbo generator is

13MW. Following reduction made in air mass flow rate, fuel mass flow rate would be increased by turbo generator control system so that required power would be supplied. As it is also observed in the figures, a 0.89% reduction in air mass flow rate would result in a 1.6% increase in fuel mass flow rate. Simultaneous reduction and increase respectively made in air and fuel mass flow rates would be resulted in an increase of fuel-air ratio (FAR), turbine input temperature (TIT), and turbine exit temperature (TET). In figure (7), curve related to changes in turbine inlet temperature is shown. Finally, the gas turbine control system supplies output power required through changes it makes in work and performance conditions. Comparing power change of power turbine through time is shown in figure (8).

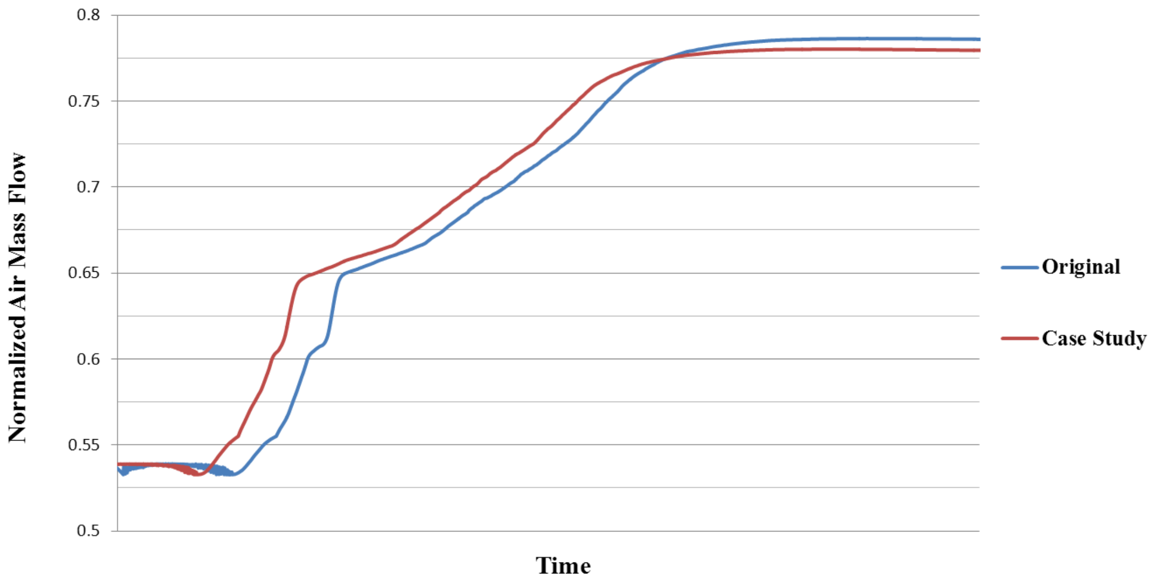



**Figure 5-** Changes in air mass flow rate through time

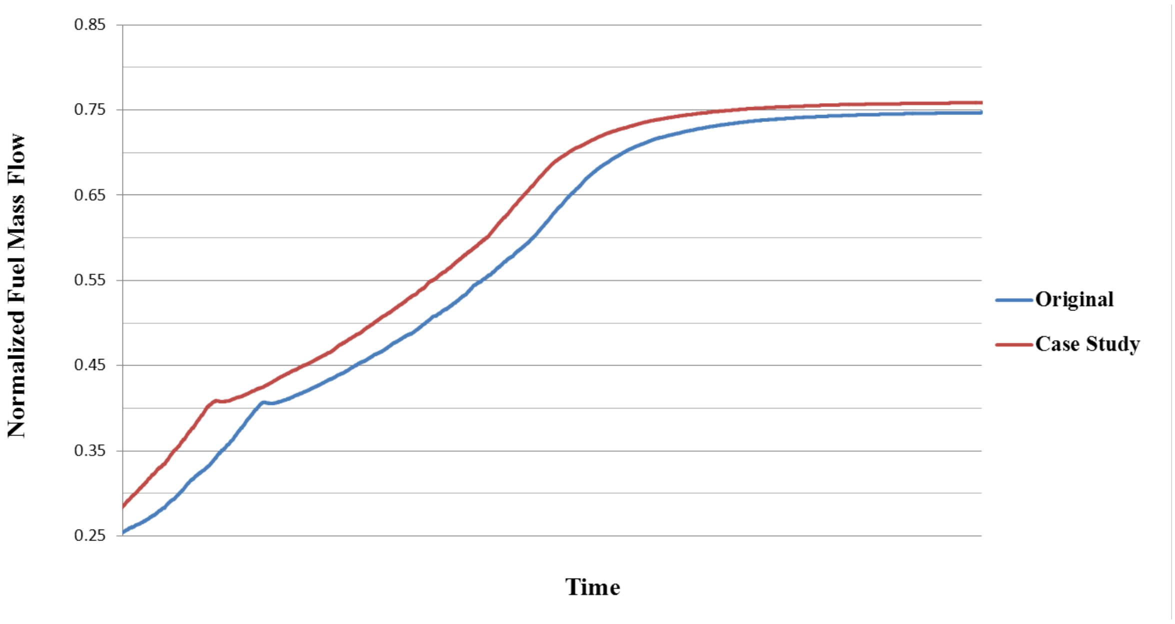



**Figure 6-** Change of fuel mass flow rate through time

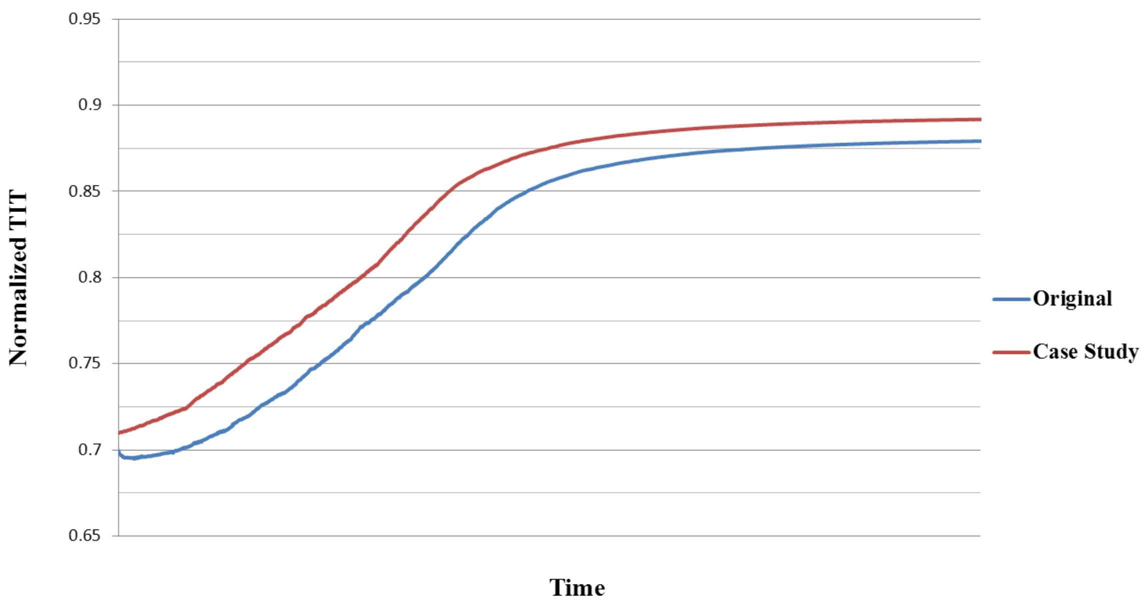



**Figure 7-** Temperature change of inlet gas flow to gas generator turbine through time

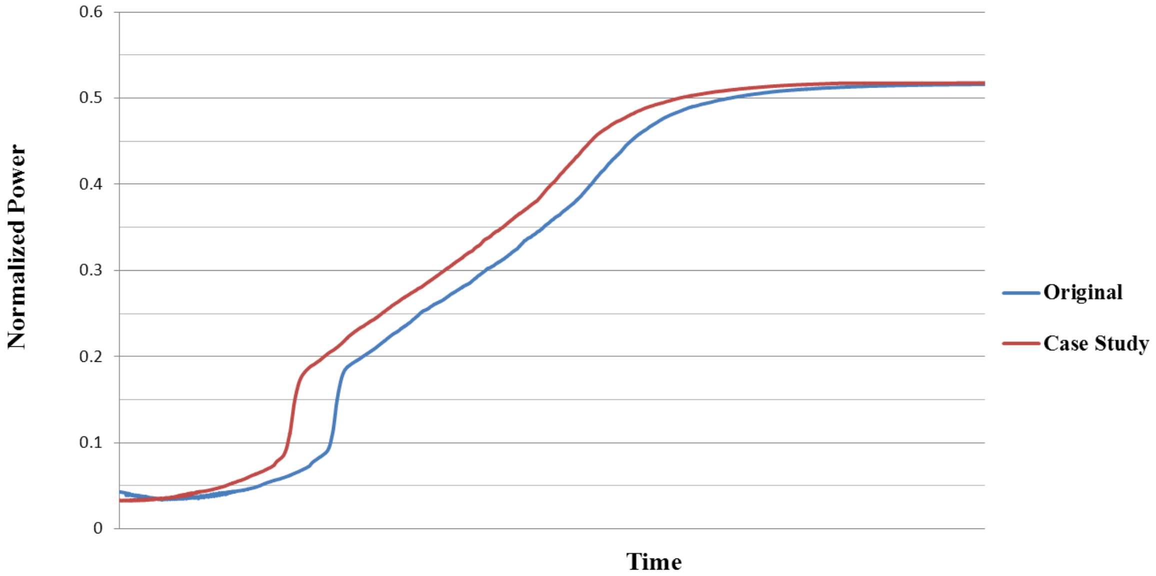



**Figure 8-** Change of generated power of power generator through time

## Conclusion

Effect of geometrical changes in gas generator turbine blades has been studied on the overall dynamic behavior of a 25-MW industrial gas turbine. The geometrical changes studied is the reduction of hot gas flow cross-section resulted from unwanted factors such as restrictions related to manufacturing technology and/or fouling phenomenon. According to the results, the reduction of cross-section would result in the reduction of mass flow rate passing through blades. Reduction of passing mass flow rate would result in compressor performance in areas closer to Surge line which in turn would be followed with the increase of instability. Motor control system in gas turbine makes the increase in fuel flow rate so that nominal expected output power would be achieved; and, the decrease in power resulted from the reduction of air mass flow rate would be compensated. Following these changes, the inlet gas temperature to gas generator turbine and outlet gas temperature from power turbine would be increased, resulting in optimum performance of gas turbine to be affected.

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