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AXIAL-THRUST RESPONSES DUE TO A GAS TURBINE'S ROTOR BLADE DISTORTIONS

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The axial thrust imposed on the shaft of a gas turbine depends upon its rotor blade inlet inclination to the turbine's axial direction: this inclination can change due to the distortions resulting from fouling, aging, tip rubbing, erosion, thermal-fatigue cracks, and corrosion. Relevant influential parameters for an operational gas turbine were measured. Theoretical predictions for the behavior of the same turbine were obtained from computer simulations. The results of both measurements and theoretical predictions were compared and showed qualitative correspondence. The rotor blade profile distortions result in significant increases in the axial thrust on the compressor, which adversely affects the gas turbine's thermodynamic performance, reliability, and operational life.

Keywords: *gas turbine, compressor, rotor blade, axial thrust, vibration.*

Introduction. The release of a mechanical energy in combustion of a fuel is generally accomplished by a thermodynamic cycle involving the processes of induction, compression, heat addition, expansion, and heat rejection/exhaust. In a gas turbine, atmospheric air is drawn in through an intake duct into the compressor and delivered at a higher pressure to the combustor [1]. This is accomplished by the gas turbine compressor consisting of a cascade of several stages of blades located radially on a single axle. Maintenance of each blade's shape design is fundamental to sustaining high performance of the compressor: distortions of each blade profile lead to changes in the air flow incidence angle on the blade and the thinning of its trailing edges. The latter is detrimental to the fatigue strength of the blade. Fouling, aging, tip rubbing, erosion of the compressor blades, thermal-fatigue cracks, and corrosion have been found to be responsible for compressor blade distortions [2]. Flow distortions can result in the vibration of the rotor blades. Of particular concern is the manner in which these distortions propagate through each blade row. The distorted flow emerging from one stage serves as the driving force for the subsequent blade row [3]. The cumulative effect of these distortions is a progressive increase in the axial thrust on the compressor, which can result in serious vibrations occurring in the rotating blade rows when they move through a nonuniform flow field and are excited by alternating bending forces. Such nonuniformities can be generated by flow distortions and trailing edge vortices from the struts in the compressor [4]. The most intensive vibration response occurs if the flow distortions are equally spaced over the rotating blades. This forms harmonic multiples of the rotational speed, especially if these harmonics coincide with the natural frequencies of the rotating blades. Blade-passing frequencies, combined with the rotation of blades, can lead to a condition known as high-cycle fatigue, which is a primary mechanism resulting in blade failure because of vibrations at levels exceeding the material endurance limit of the blades. The foregoing makes the problem of compressor blade profile distortions a major source of concern, as it adversely affects the thermodynamic performance, reliability, and life duration of a gas turbine.

Lee and Kim [5] presented a numerical optimization technique, combined with a three-dimensional, thin-layer, Navier–Stokes analysis, to study the effects of profile shape in an axial compressor on the basis of calculations for a single-stage flow between the rotor and stator. Ghaly and Mengistu [6] found that the aerodynamic performance of each blade depends on (i) the blade angle of incidence relative to the axial direction, (ii) the leading- and trailing-edge radii, (iii) the wedge angles, and (iv) the blade stagger angle and (v) spacing. Xu and Chen [7] studied the effects of aerodynamic sweep on the compressor performance and the turbine stability (aerodynamic sweep is the redistribution of flow using either backward- or forward-swept leading-edge compressor blades).

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TABLE 1. Predicted Values of the Turbine Parameters

β_1, deg	β_2, deg	C_{a1}, m/s	C_{a2}, m/s	$\dot{m}$, kg/s	R_a, kN
45	42	74.3	70.7	200.009	720
48	39	77.7	66.9	200.000	2160
52	36	80.9	61.5	199.999	3880
55	33	83.3	57.3	199.998	5300
60	28	88.2	50	199.999	7640
63	25	90.6	45.3	199.998	9080
65	23	92.0	42.3	200.001	9960
68	20	93.9	37.4	199.999	11300

TABLE 2. Measured Values of the Turbine Parameters

T_{01}, K	P_{01}, N/m^2	β_1, deg	β_2, deg	C_{a1}, m/s	C_{a2}, m/s	$\dot{m}$, kg/s	R_a , kN
306	107313	48	21.6	92.9	66.9	122.1	3174
307	107293	48.5	20.1	93.9	66.2	121.8	3373
308	107272	49	18.6	94.7	65.6	121.3	3529
309	107252	49.5	17.1	95.5	64.9	120.9	3699
310	107232	50	15.6	96.3	64.2	120.5	3868
311	107212	51.4	13	97.4	62.3	120.1	4215

The present investigation was carried out using an operational gas turbine at Ughelli, southern Nigeria. The plant operates on a Brayton cycle. The compressor and turbine components have 17 and 4 stages, respectively.

Method of Investigation. The values of the parameters relevant to this analysis were obtained from observation of the meters in the test room. These direct measurements were carried out at various points, such as the inlet and outlet of the compressor, as well as the inlet and outlet of the turbine. Large amounts of performance data were obtained from the daily log sheets for the period of study. The design parameters for the turbine were taken from the manufacturer. Then calculations were made using existing and derived formulas to obtain the values of the parameters that could not be measured directly or derived from design manuals or log sheets. Predictions from computer simulations of the turbine behavior were obtained (see Table 1), and comparison was made with the corresponding results of measurements (see Table 2).

Rogers and Mayhew [8] stated that the axial thrust R_a and the mass flow rate $\dot{m}$ can be obtained from the following equations:

$$R_a = \dot{m}\,(C_{a1} - C_{a2}) + (P_1A_1 - P_2A_2)\,, \tag{1}$$

$$\dot{m} = \rho_1 A_1 C_1\,, \tag{2}$$

where

$$\rho_1 = P_{01}/RT_{01}\,, \tag{3}$$

$$C_{a1} = C_1 \cos\alpha_1\,, \tag{4}$$

$$C_{a2} = C_2 \cos\alpha_2\,. \tag{5}$$

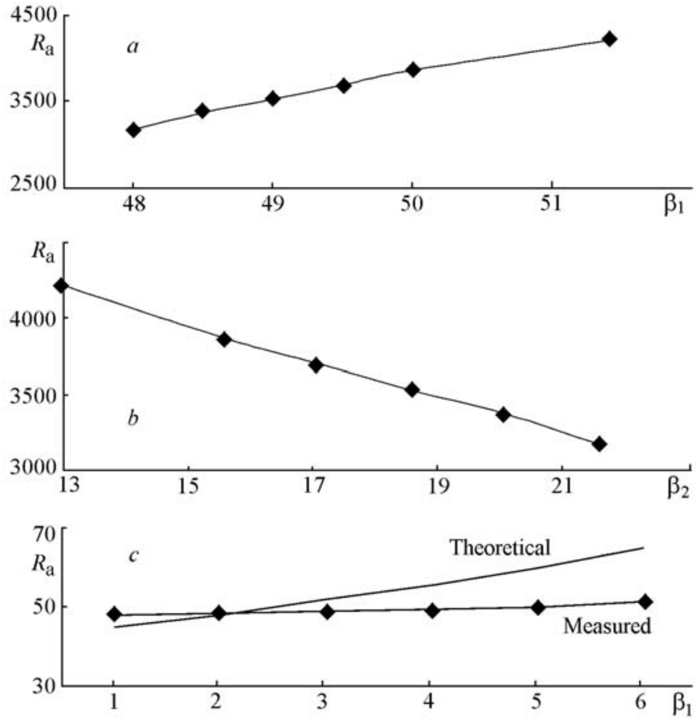

Fig. 1. Measured values of the axial thrust vs. the inlet (a and c) and outlet (b) angles. R_a, kN; β_1, β_2, deg.

For each stage, $C_1 = C_2 = 100$ m/s. Owing to the symmetry of the angles, $\alpha_1 = \beta_2$, $\alpha_2 = \beta_1$. Thus, Eqs. (4) and (5) can be rewritten as

$$C_{a1} = C_1 \cos \beta_2 ,$$

$$C_{a2} = C_2 \cos \beta_1 .$$

Next it follows from [1] that

$$T_{01} = T_1 + \frac{C^2}{2C_p} ,$$

$$P_{01} = P_1 \left(\frac{T_{01}}{T_1} \right)^{\frac{\gamma}{\gamma - 1}} .$$

The mass flow rate was computed numerically for substitution into Eq. (1) in order to obtain the values of the axial thrust. The variations of the axial thrust with the rotor blade inlet and outlet angles based on plant measurements are given in Fig. 1a and b, respectively. Figure 1a shows that, as the rotor blade inlet angle increases due to distortions, the axial thrust on the compressor also increases. Figure 1c gives the comparison of the measured and theoretical predictions of the variation of the axial thrust on the compressor.

Both trends in the measured and theoretical predictions are in qualitative agreement. Increases in the rotor blade inlet angle due to blade profile distortions result in increases of the axial thrust on the compressor and can lead to vibration problems. There is a direct correlation between blade profile distortions and vibratory stresses resulting from the axial thrusts. This corroborates the findings of Luedke [3] and Vahdati et al. [4].

Conclusions. Compressor blade profile distortions can result in an increase in the rotor blade inlet angle and, subsequently, in a significant increase in the axial thrust on the compressor. The increase in the rotor blade inlet angle leads to a reduction in the pressure ratios and air-mass flows across each stage. The cumulative effect is a drop in the thermodynamic performance, reliability, and operational life.

NOTATION

A_1 and A_2, flow area at inlet to the rotor and outlet from it, respectively, m^2; C, air velocity, m/s; C_1 and C_2, inlet and outlet air velocities, m/s; C_{a1} and C_{a2}, axial inlet and outlet components of velocity, m/sec; C_p, specific heat of air at constant pressure, kJ/(kg·K); $\dot{m}$, mass flow rate, kg/s; P_1 and P_2, inlet and outlet pressures, bar; P_{01}, inlet stagnation pressure, bar; R, gas constant, kJ/(kg·K); R_a, axial thrust, kN; T_{01}, inlet stagnation temperature, K; T_1, inlet temperature, K; α_1 and α_2, inlet and outlet angles of incidence of air flow on rotor, deg; β_1 and β_2, inlet and outlet angles of blade inclination with respect to axial direction of the turbine, deg; γ, specific heat ratio of air; ρ_1, inlet air density, kg/m^3. Subscripts: a, axial; 1, inlet; 2, outlet.

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