The Effect of Surface Roughness on Thermodynamic Performance Parameter of Axial Flow Compressor

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Abstract: In axial flow compressor, blade surface roughness is affected by many failure modes such as fouling, erosion, corrosion and foreign object damage. But with development of filter performance, particles with diameter larger than 2 μm cannot enter into compressor, so fouling is the most important influence factor which results in the variation of surface roughness. This study respectively discusses the effect of surface roughness on performance parameter when surface roughness is constant and linearly distributed. Finally, based on experiment result, reverse design method is applied to reconstruct the fouled compressor by combining laser triangulation sensor with compressor fouling test rig and then reconstructed solid model is imported into ANSYS CFX to simulate flow field. Result shows that the increase of surface roughness results in the decrease of pressure ratio, mass flow and efficiency.

Keywords: Axial flow compressor, fouling, reverse design, surface roughness, thermodynamic performance parameter

INTRODUCTION

Axial flow compressor is an important device in gas turbine plant because it consumes a large portion of the total turbine work. Many researches found that approximately 50 to 60% of the total work produced in the turbine is consumed by its axial flow compressor. So maintaining high compressor efficiency is important for the plant.

In many failure modes of axial flow compressor, blade surface roughness is affected by fouling, erosion, corrosion or FOD (Foreign Object Damage). At present, because of high performance filters mounted in inlet, particles with diameter greater than 2 μm are prevented from entering into compressor. In general, particles with diameter smaller than 10 μm may cause fouling, but not erosion. Smaller particles (including dust, unburned hydrocarbons and insects) can be deposit on blade surface to form fouling when lubricating oil and water is existed simultaneously. Fouling changed compressor blade geometry and increased surface roughness so that influenced its aerodynamic performance. So, fouling is most important influence factor which results in the variation of surface roughness.

Experience has shown that axial compressors will foul in most operating environments. There are many industrial pollutants and all kinds of environmental conditions that play an important role in the fouling process. Compressor fouling is typically caused by (Table 1):

- External contaminants
- Internal contaminants
  - Internal gas turbine oil leakage from the front bearing of the axial compressor is a common cause. Oil leaks combined with dirt ingestion causes heavy fouling problem
  - Impure water from evaporative coolers
  - Vapor plumes from adjacent cooling towers
  - Corrosion and erosion of filter panel

Because gas turbine ingest extremely large quantities of air, study found that at a 10 ppm foul ant

| Table 1: Contaminants source | Type of particle | Size (μm) |
|------------------------------|-----------------|-----------|
| F1                           | Ground-dust     | 1–300     |
| F2                           | Oil smokes      | 0.02–1    |
| F3                           | Fly ash         | 1–200     |
| F4                           | Salt particles in mist | Less than 10 |
| F5                           | Smog            | Less than 2 |
| F6                           | Fume            | Less than 1 |
| F7                           | Clay            | Less than 2 |
| F8                           | Rosin smoke     | 0.01–1    |
| F9                           | Coal dust       | 1–100     |
| F10                          | Metallurgical dusts and fumes | 0.001–100 |
| F11                          | Ammonium        | 0.1–3     |
| F12                          | Carbon black    | 0.01–0.3  |
| F13                          | Contact sulphuric mist | 0.3–3    |
| F14                          | Paint pigments  | 0.1–5     |
loading rate, 204 tones of foul ant would be ingested which reduced performance parameter such as mass flow, efficiency.

Previous research (Lakshminarasimha et al., 1994) found that if roughness increased from 55 to 120 μm, fuel consumption rate increased 0.13%, mass flow decreased 5% and efficiency decreased 2.5%. Another study (Kurz and Brun, 2001) found that rotor blade roughness increased from 4 to 8 μm and stator blade roughness increased to 8 μm after long time operation. Researcher (Zwebek, 2002) demonstrated that mass flow decreased 5% and fuel consumption rate increased 2.5% because of fouling.

Some researchers simulated the effect of surface roughness on performance by adding constant surface roughness. But experiments showed that fouling level is not uniform along chord length and blade span, so above methods can’t reflect the real status of fouled compressor. In this study, reverse design method is applied to reconstruct solid model of fouled compressor and then flow simulation is implemented based on reconstructed solid model in order to reveal the real effect of surface roughness on performance parameter.

**MATHEMATICAL MODEL**

This study chooses NASA rotor37 as study object, its detailed design parameters are as following:

- **Type:** Transonic axial compressor
- **Blades:** 36 blades with tip clearance
- **Fluid:** Air
- **Working point:** Rotation speed 17188.7 rpm, Mass flow 20.19 kg/s, Pressure ratio 2.106, Adiabatic efficiency 87.7%

The geometry of rotor37 is shown in Fig. 1. This study simulates the flow field of axial flow compressor by using ANSYS CFX. In this simulation, J-grid is chosen and grid node number is 25 million. And then SST model is chosen as turbulent model. The grid node is refined in stream wise location near to wall and the computational model of clean rotor37 is shown in Fig. 2. Finally, compressor inlet boundary is given by mass flow rate (20.19 kg/s) and outlet boundary is given by static pressure (Zhou and Wang, 2007; Wang et al., 2009; Li et al., 2007).

**SIMULATION ANALYSIS OF DIFFERENT BLADE SURFACE ROUGHNESS DISTRIBUTION**

**Constant blade surface roughness:** Assuming blade surface roughness is uniform in whole blade; flow simulation is implemented by ANSYS CFX based on three different surface roughnesses such as 50, 100 and 150 μm, respectively. Simulation result is shown in Table 2.

Simulation result shows that pressure ratio decreases 5.37%, temperature ratio decreases 0.49%, isentropic efficiency decreases 4.11% and output power decreases 3.69% when surface roughness increases from 0 to 50 μm. Pressure ratio decreases 5.9%, temperature ratio decreases 0.59%, isentropic efficiency decreases 4.53% and output power decreases 4.03% when surface roughness increases from 0 to 100 μm. Pressure ratio decreases 6.2%, temperature ratio decreases 0.61%, isentropic efficiency decreases 4.8% and output power decreases 4.17% when surface roughness increases from 0 to 150 μm. Simulation result simultaneously shows that thermodynamic performance parameter is dramatically reduced when blade is rough.

**Table 2: Simulation results of fouled compressor when surface roughness is constant**

| Thermodynamic parameter          | Blade surface roughness (micron) |
|----------------------------------|----------------------------------|
|                                  | 0      | 50     | 100    | 150    |
| Pressure ratio                   | 2.0167 | 1.9085 | 1.8977 | 1.8917 |
| Temperature ratio                | 1.2686 | 1.2626 | 1.2611 | 1.2608 |
| Isentropic efficiency (%)        | 84.6359| 81.1543| 80.8045| 80.5763|
| Polytrophic efficiency (%)       | 86.0435| 82.7451| 82.4111| 82.1938|
| Output power (KW)                | 1627.0900| 1567.1200| 1561.4700| 1559.2200|
Linear distribution of surface roughness: Figure 3 shows the fouling on the IGV surface from the gas turbine. Fouling is not uniform in whole blade, so blade surface roughness is not constant and above result doesn't accurately reflect the real status of fouled compressor. Experiment found that contaminant particles are easily deposited on the blade root and fouling in suction surface is more severe than in pressure surface. At the same time, particles are more easily deposited in leading edge. Based on experiment result, assuming blade surface roughness in section at specified span is linear distributed from leading edge to trailing edge.

Surface roughness, often shortened to roughness, is a measure of the texture of a surface. It is quantified by the vertical deviations of a real surface from its ideal form. If these deviations are large, the surface is rough; if they are small the surface is smooth. There are many different roughness parameters such as Ra, Rz, Rq, but profile arithmetic average is by far the most common result, assuming blade surface roughness in section at specified span is linear distributed from leading edge to trailing edge.

Surface roughness, often shortened to roughness, is a measure of the texture of a surface. It is quantified by the vertical deviations of a real surface from its ideal form. If these deviations are large, the surface is rough; if they are small the surface is smooth. There are many different roughness parameters such as Ra, Rz, Rq, but profile arithmetic average is by far the most common which is shown in Fig. 4. The formula is shown in Eq. (1):

$$Ra = \frac{1}{L} \int_{0}^{L} y(x) \, dx$$
$$Ra = \frac{1}{n} \sum_{i=1}^{n} |y_i|$$

Table 3: Simulation results of fouled compressor when surface roughness is linear distributed

| Thermodynamic parameter                  | Surface roughness in leading edge (micron) |
|-----------------------------------------|------------------------------------------|
| Pressure ratio                          | 2.0167                                   |
| Temperature ratio                       | 1.2686                                   |
| Isentropic efficiency (%)               | 84.6359                                  |
| Polytropic efficiency (%)               | 86.0435                                  |
| Output power (KW)                       | 1627.0900                                |

Table 4: Summary performance data of clean compressor

| Quantity   | Inlet | LE cut | TE cut | Outlet | TE/LE | TE-LE |
|------------|-------|--------|--------|--------|-------|-------|
| Density    | 1.1542| 0.9993 | 1.5376 | 1.5422 | 1.5387| 0.5383 |
| P static   | 94536.900 | 81829.2000 | 151403.0000 | 151351.0000 | 1.8502 | 69573.9000 |
| P total    | 112291.0000 | 108559.0000 | 229552.0000 | 226462.0000 | 2.1145 | 120993.0000 |
| P total (rot) | 112756.0000 | 105305.0000 | 201453.0000 | 201822.0000 | 0.9634 | -3852.2700 |
| T static   | 284.6910 | 272.5300 | 338.5050 | 338.8530 | 1.2421 | 65.9742 |
| T total    | 299.6910 | 301.6220 | 380.5200 | 380.1980 | 1.2616 | 78.8977 |
| T total (rot) | 300.0760 | 300.3940 | 300.4100 | 300.3600 | 1.0001 | 0.0162 |
| H static   | -13517.9000 | -25732.3000 | 40532.2000 | 40882.2000 | -1.5752 | 66264.5000 |
| H total    | 1547.2800 | 3487.7000 | 82732.5000 | 82409.3000 | 23.7212 | 79244.8000 |
| Rothalpy   | 1933.9800 | 2269.8600 | 105305.0000 | 105305.0000 | 1.0072 | 16.2798 |
| Entropy    | -22.1182 | -1.4787 | 9.7369 | 12.9574 | 11.2156 | 1.5387 |
| Mach (abs) | 0.5080 | 0.7300 | 0.7844 | 0.7799 | 0.7228 | 0.5255 |
| Mach (rel) | 1.2705 | 1.3791 | 0.7248 | 0.7228 | 0.5255 | 0.6543 |
| U          | 391.9840 | 391.3060 | 390.1770 | 0.9996 | -0.1707 | 0.6543 |
| Cm         | 171.4960 | 225.4720 | 175.9940 | 181.9840 | 0.7806 | -49.4783 |
| Cu         | 0.9917 | -11.8294 | -230.2880 | -214.4570 | 19.4674 | -218.4580 |
| C          | 171.6840 | 240.0940 | 298.5870 | 2253.5800 | 1.0745 | 58.4924 |
| Distortion parameter                    | 1.0663 | 1.0466 | 1.0613 | 1.0125 | 0.0140 | 0.0147 |
| Flow angle: alpha                       | -0.2125 | 2.8925 | 38.4182 | 50.0662 | 1.3282 | 35.5258 |
| Wu        | 392.9750 | 379.4770 | 160.8470 | 176.2600 | 0.4239 | -218.6290 |
| W         | 429.6930 | 442.5470 | 243.3650 | 256.5760 | 0.5499 | -199.1820 |
| Flow angle: beta                        | -66.4963 | -60.1496 | -46.7381 | -44.5495 | 0.7770 | 13.4115 |

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Table 3: Summary performance data of fouled compressor

| Quantity          | Inlet Interpolated | TE cut Interpolated | Outlet | TE/LE | TE/LE | Units       |
|-------------------|--------------------|---------------------|--------|-------|-------|-------------|
| Density           | 1.2774             | 1.1092              | 1.5443 | 1.5497| 1.3922| 0.4351 [kg m^-3] |
| P static          | 105191.0000        | 92375.7000          | 151374.0000 | 151332.0000 | 1.6387 | 58998.1000 [Pa] |
| P total           | 121757.0000        | 116749.0000         | 236703.0000 | 232320.0000 | 2.0274 | 119954.0000 [Pa] |
| P total (rot)     | 122732.0000        | 113004.0000         | 108415.0000 | 106568.0000 | 0.9594 | -4589.3100 [Pa] |
| T static          | 286.2090           | 275.4150            | 333.7480 | 334.1610 | 1.2118 | 58.3332 [K] |
| T total           | 290.2710           | 302.1120            | 378.0850 | 377.6230 | 1.2515 | 75.9737 [K] |
| T total (rot)     | 300.0030           | 300.7390            | 300.7480 | 300.7480 | 1.0002 | 0.0547 [K] |
| H static          | -11993.9000        | -22835.0000         | 35754.9000 | 36169.6000 | -1.5658 | 58589.9000 [J kg^-1] |
| H total           | 1125.6200          | 3979.1000           | 80287.1000 | 79822.9000 | 20.1772 | 76308.0000 [J kg^-1] |
| Rothalpy          | 1860.8300          | 2600.6800           | 2655.5600 | 2609.5700 | 1.0211 | 54.8799 [J kg^-1] |
| Entropy           | -44.8137           | -19.5862            | -6.0457 | -1.5581 | 0.3087 | 13.5405 [J kg^-1 K^-1] |
| Mach (abs)        | 0.4708             | 0.6909              | 0.8118 | 0.8058 | 1.1750 | 0.1209 |
| Mach (rel)        | 1.2557             | 1.3554              | 0.7682 | 0.7644 | 0.5667 | -0.5873 |
| U                 | 391.9840           | 391.3060            | 391.1350 | 390.7170 | 0.9996 | -0.1707 [m s^-1] |
| Cm                | 159.0300           | 211.4220            | 178.3260 | 182.8250 | 0.8435 | -33.0965 [m s^-1] |
| Cu                | 1.8612             | -13.3386            | -238.7790 | -216.5460 | 17.9014 | -225.4400 [J/kg] |
| C                 | 159.5450           | 227.8330            | 312.2570 | 293.6230 | 1.3705 | 84.4243 [m s^-1] |
| Distortion parameter | 1.0903              | 1.1400              | 1.0602 | 1.0974 | 0.9567 | -0.0498 |
| Flow angle alpha  | -0.3865            | 3.2204              | 26.8658 | 48.9872 | 8.3424 | 23.6454 [degree] |
| Wu                | 393.8450           | 377.9680            | 152.3560 | 174.1710 | 0.4031 | -225.6110 [m s^-1] |
| W                 | 425.8230           | 435.3020            | 239.6200 | 256.5120 | 0.5505 | -195.6820 [m s^-1] |
| Flow angle beta   | -68.0904           | -62.0062            | -48.8216 | -46.0173 | 0.7874 | 13.1846 [degree] |

According to Eq. (1), assuming surface roughness of leading edge is, respectively 50, 100, 150 μm; point coordinate of blade section is computed. And then, solid model is constructed. The simulation result is shown in Table 3.

Simulation result shows that the decrease of thermodynamic performance parameter when surface roughness is linear distribution is less than when surface roughness is constant. It can more accurately reflect actual situation of fouled compressor.

The application of reverse design method: Reverse design method is a modern design method to create a 3D virtual model of existing physical part or mechanical system. The reverse design process involves measuring an object using 3D scanning technologies such as coordinate measuring machine, laser scanners and then reconstructing it as a 3D solid model based on measured data. Because the geometry compressor blade is irregular, traditional contact measure method is difficult to measure it. At the same time, blade surface can be scratched by contact measure method (Chen et al., 2003, 2004, 2005).

Laser triangulation is a non-contact active vision measurement method which has many advantages such as no influence on object surface, high precision, simple structure and strong anti-interference ability. Laser triangulation sensor is mounted before the compressor inlet of compressor fouling test rig to measure rotor blade profile parameter online. This study analyzes the change of thermodynamic performance parameter when compressor is operated for 20 h. Point clouds measured from sensor are reconstructed into solid model and then imported into ANSYS CFX to implement flow simulation. Simulation result shows that that pressure ratio decreases 0.97%, temperature ratio decreases 0.05%, isentropic efficiency decreases 0.77% and output power decreases 0.52% when test rig is operated for 20 h. Table 4 shows the detailed performance parameter of clean compressor and Table 5 shows the detailed performance parameter of fouled compressor. From this table can be seen that pressure and temperature at leading edge and trailing edge of fouled compressor are
The increase of surface roughness results in the increase of pressure, but the mass flow, efficiency and pressure ratio are reduced. The variation of parameter such as pressure, temperature and mach number between clean compressor and fouled compressor is shown in Fig. 5 to 10:

- **Pressure**: Stream wise plots of Pt and P show that the increase of surface roughness resulted in the increase of Pt and P in whole stream wise location
- **Temperature**: Figure 7 and 8
- **Mach number**: Figure 9 and 10

**CONCLUSION**

Simulation analysis found that the effect of surface roughness on performance parameter is as following:

- Different distribution of surface roughness can result in different variation of performance parameters.
- Then increase of surface roughness may cause the decrease of pressure ratio, mass flow, efficiency.
- Reverse design method can be applied to reconstruct fouled compressor and simulate flow field. And this method can accurately reflect the true status of fouled compressor.

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