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Experimental Investigation of Wash Fluid Preheating on the Effectiveness of Online Compressor Washing in a Suction Tunnel Compressor Cascade

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Abstract
Online compressor washing is a promising method of preventing/ recovering the effect of fouling on compressor blades. However, proper strategies need to be implemented for online compressor washing to be effective since it is conducted when the engine is in operation. This study presents an experimental investigation of wash fluid preheating on the effectiveness of online compressor washing in a suction wind tunnel compressor cascade. Crude oil was uniformly applied on the compressor cascade blade surfaces using a roller brush, and consequently carborundum particles were ingested into the tunnel to create accelerated fouled blades. Demineralized water was preheated to 50°C using a heat coil provided in the tank. Washing of the fouled blades were conducted using single flat fan nozzle, where the preheated and non-preheated demineralized water were used separately to wash the fouled blades. When fouled blades washed with preheated demineralized and the non-preheated were compared, it was observed that there was little or no difference in terms of total pressure loss coefficient and exit flow angle. However, when the fouled and washed cases were compared, there was a significant difference in total pressure loss coefficient and exit flow angle.

Keywords: *Compressor cascade; compressor washing; demineralized water; exit flow angle; fouled blades; total pressure loss coefficient*

1. Introduction

Gas turbines, being air breathing machines, ingest large volume of airflow which contains contaminants that foul the compressor blade surfaces, thereby degrading the overall performance of the gas turbine. Online compressor washing is a promising method of preventing/ recovering the effects of fouling on the compressor blades. However, proper strategies need to be implemented for online compressor washing to be effective since it is conducted when the engine is in operation. One of such strategies is the preheating of the fluid prior to washing.

Some studies have proclaimed that heating of the wash fluid is beneficial, especially in offline compressor washing where there is need for cooling down before washing to prevent thermal stresses [1]. Foss [2] added that solubility of deposits increases with water temperature. According to Stalder [3], wash fluid is preheated to enable earlier injection so as to reduce downtime associated with offline washing.

For heavy duty industrial gas turbines, it is a common practice to preheat the wash fluid to about 60° to 70°C to reduce waiting time for the engine to cool down, especially for offline compressor washing [4]. Fielder [1] stated that improved washing efficiency was achieved in marine application due to wash fluid preheating. However, Engdar et al. [5] reported in their numerical study that preheating the wash fluid plays insignificant role regarding the cleaning effectiveness. The authors attributed their claim to the fact that droplet temperatures adjust close to air flow temperature before it gets to the compressor.

From literature research, it is obvious that there is a misconception about the influence of wash fluid preheating on the washing effectiveness. This is because some researchers have proclaimed that it is beneficial to preheat the wash fluid prior to washing, while others are of the contrary view. This study presents an experimental investigation of wash fluid preheating on the effectiveness of online compressor in a suction wind tunnel compressor cascade.

2. Materials and Methods

2.1 Injector System Setup

An injector system employed in this study to conduct the washing investigation comprises a high pressure piston pump, tank and mechanical traverse unit where the nozzle is attached (see Figures 1). The tank is capable of containing 40litres of wash fluid and it has a heat coil which can be used for heating the wash fluid before being injected (see bottom right of Figure 1). A piston pump of 4.1kW as shown in bottom left of Figure 1 was used to inject the wash fluid from the tank through the nozzle tip at high pressures. A knob in the control panel is used to regulate the pressure at which the fluid is being injected. Top right of Figure 1 shows the control panel for the washing system. Also, a thermocouple shown in Figure 2, located externally from the tank, was used to measure the temperature of the wash fluid. Thermocouple readings are in degree Celsius. Figure 3 shows the schematic of the experimental setup.

Figure1: Washing System

Figure 2: Thermocouple

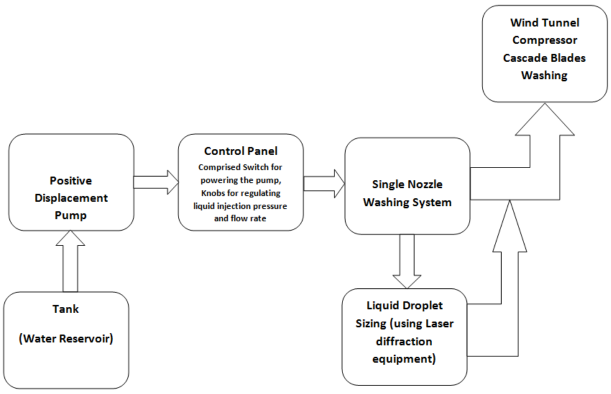

Figure 3: Schematic of Experimental Setup

2.2 Droplet Sizing

Prior to the investigation wash fluid preheating on the effectiveness of online compressor washing in the suction wind tunnel compressor cascade, the injection fluid droplets for the preheated and non-preheated cases were sized using a laser diffraction particle analyzer shown in Figure 4. The droplet size measurements were conducted to ascertain if there will be significant variation in droplet size between the preheated and non-preheated. This is because droplet size is one the parameters that influences the effectiveness of online compressor washing [6]. Hence, if there is a significant variation in droplet size between the preheated and non-preheated cases, this would provide a guide when analyzing the investigation on washing effectiveness of the preheated and non-preheated cases.

The Spraytec particle analyzer employed to measure the droplet size of the nozzle under preheated and non-preheated conditions, uses laser diffraction method. The equipment utilizes angular intensity scattered light to measure the droplet when spray injected across the laser beam (see Figure 4). The appropriate optical model is then used to analyze the scattered light pattern recorded to yield a size distribution.

In this study, injection distance was also considered based on the need to account for distance from the cascade inlet plenum to the blades. Injection distance was varied from 50 to 200mm, in steps of 50mm to account for the effect of injection distance on droplet sizes.

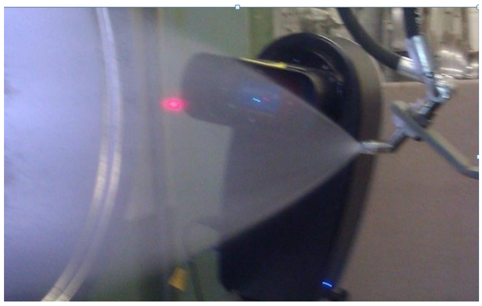

Figure 4: Spray Particle analyzer system

2.3 Compressor Cascade Description

To carry out the investigation of wash fluid preheating on the effectiveness of online compressor washing, a suction wind tunnel compressor cascade shown in Figure 5 was employed. The wind tunnel has nine untwisted NACA 65 series blades and when operated at full valve opening, it has a mass flow handling capacity of 5kg/s through an inlet area of 0.043m2. In addition, when the control valve fully opened, the cascade operates at a Mach number of 0.3 and Reynolds number of 3.8 x 105. Each of the 9 two dimensional blades in the cascade has a length, chord and a pitch-to-chord ratio of 180mm, 60mm and 0.8 respectively. In addition, to achieve high pressure rise, all the blades were positioned at zero incidence angle. A 45kW electric motor which runs at 2995rpm is used to drive the centrifugal fan which produces the suction effect of the tunnel.

Figure 5: Suction wind tunnel compressor Cascade

Design specifications of the cascade blades are presented in Table 1.

Table 1: Cascade blade design Specifications

Design Parameter	Value
Blade Inlet Angle (degrees)	51
Blade Outlet Angle(degrees)	34
Camber (degrees)	30
Stagger Angle (degrees)	36
De Haller Number	0.7
Profile Shape	NACA 65 series
S/c Ratio	0.8
Inlet Mach Number	0.3
Passage Width(mm)	48

The experimental study was carried out by taking measurement of the flow at the mid-span of the three middle blades to avoid interference of boundary layer on the measured results based on the suggestion of Dixon [7]. For inlet flow conditions, total and static pressures measurements were obtained using a pitot static tube at one chord upstream of the three middle blades. Measurements were taken at this point to ascertain the inlet flow conditions. While for the exit flow conditions, a three-hole probe was employed at one chord downstream of the three middle blades so as to take measurement of exit flow angle, velocity, total and static pressures. One chord downstream of the blades was chosen because at this point, information about exit flow conditions can be obtained. A reference point between blades 4 and 5 was chosen and the three-hole probe was nulled at this point. Measurements were taken at every one millimeter by traversing the three-hole probe between -40mm to 120mm. In addition, to ensure that relatively accurate results are obtained for the measurements, the readings were taken thrice at every measurement point. Consequently, averaged values of the reading were recorded. According Gostelow and Pollard [8] taking measurements at one chord downstream of the blade is reasonable in the sense at that point the flow is fully mixed.

2.4 Test Uncertainty Analysis

To ensure that relatively accurate results are obtained in the analysis of fouling and washing effect on the aerodynamic performance of the compressor blades in the wind tunnel. An attempt has been made herein to reduce measurement inaccuracies by conducting a measurement uncertainty on the three-hole probe. In this study, the methodology proposed by Abernethy et al.[9] and used by ASME standards for test uncertainty was used to ascertain the measurement uncertainty of exit flow angle, total and static pressures of the three-hole probe. The uncertainty analysis of the data obtained for clean condition was calculated using equations 1, 2 and 3.

The values of the uncertainty analysis obtained for exit flow angle, static and total pressure are +/-0.7°, +/-0.097, and +/-0.17Pa respectively. These values fall within acceptable range of measurement uncertainty found in Arguelles et al.[10].

Mean,
$$\bar{X} = \frac{\sum_{i=1}^{N} X_i}{N} \qquad (1)$$

Standard deviation,
$$S = \sqrt{\frac{\sum_{i=1}^{N}\left(X_i - \bar{X}\right)^2}{N-1}} \qquad (2)$$

Standard uncertainty,
$$\bar{S} = \frac{S}{\sqrt{N}} \qquad (3)$$

2.3 Description of Fouling Device

The fouling device shown in Figure 6 is a rectangular box with length, height and width of 430, 355 and 130mm respectively (external dimensions) and is designed to contain up to 7.5kg of particles. The device consists of two flat plates that are separated by a distance of 3mm. The first plate is fixed and it faces the inlet of the cascade. It consists of porous holes with diameter of 6mm and 8mm spacing between each circular hole, while the thickness of the plate is 3mm. The particle injection rate is determined by the velocity at which the tunnel is operated, governed by the fan. Hence, running the tunnel at a higher velocity will result in higher particle ingestion rate since the mode of the operation of the device depends on the suction pressure. The second plate is completely solid and used to control the flow of particles into the tunnel during operation. However in most cases the solid plate is moved up or is completely removed during operation.

Figure 6: Fouled Device

2.5 Compressor Cascade Fouling

This study employs the ingestion of particles into a wind tunnel compressor cascade to create an accelerated roughness (fouling) that degrades the blade profile. The fouling device described earlier was employed with the aim of having control of the fouling level, so as to be fairly repeatable in the accelerated fouling process [11].

In this study, crude oil was applied uniformly on both sides of the blades by using a roller brush, to ensure repeatability of the fouling process before ingesting about 1.5kg of carborundum (100microns) particles on the blades to create the accelerated fouled blades (see Figure 7). The accelerated fouled case achieved can be compared to a gas turbine having lube oil leakage operating in a desert environment. Figure 7 shows the three middle fouled blades, which can be related to a severely fouled industrial gas turbine compressor blades operated in a desert for over 8000hrs without any filtration system or maintenance activity. Meher-Homji and Bromley [12] stated that the deposition of particles in a gas turbine compressor is increased when oil vapour and oil leakages are present. A similar procedure was adopted herein to increase the particle deposition rate through the application of crude oil on the three middle blades.

Figure 7: Fouled and Washed Blades

2.6 Compressor Cascade Washing

Two washed cases, namely fouled blades washed with preheated demineralized water and the ones washed with non-preheated demineralized water were considered. In the first scenario (fouled blades washed with preheated demineralized water), about 40litres of demineralized water was preheated to 50ºC before it was injected to wash the fouled blades. Heating of the wash fluid was achieved by first pouring 40liters of demineralized water into tank, followed by switching on the heat coil in the tank as shown in bottom right of Figure 1. After allowing for some minutes, the demineralized water was stirred using rectangular shaped plastic, to achieve uniform temperature distribution of the wash fluid. Consequently, the wash fluid temperature was measured using a thermocouple. After ensuring that the temperature of the wash fluid was at 50ºC, the fouled compressor cascade blades were then washed by switching on the pump where the wash fluid was injected 90bar injection pressure through the single nozzle, positioned at mid-span of the tunnel intake (see top left of Figure 1). Similarly, for the non-preheated case, 40 litres of demineralized water was poured into the tank, followed by switching on the injector system in order to wash the fouled blades. It is worth mention that each was regime lasted for five minutes and the washing for the two different cases were carried out under the same operating conditions such as injection period, pressure, quantity of water etc.

2.7 Correlation of Cascade Data to a Theoretical Compressor Stage Performance

To obtain the mean theoretical mean stage performance of the compressor from the cascade experimental data, equations derived by Howell's [13], to account for losses were used in this study.

C_{Dp}, C_{Da}, C_{Ds} represent Profile, Annulus and Secondary drag coefficients in equations 4, 5 and 6 respectively.

$$C_{DP} = \frac{s}{c}\left(\frac{\Delta P}{1/2\rho V_1^2}\right)\frac{COS^3\alpha_m}{COS^2\alpha_1} \tag{4}$$

$$C_{Da} = 0.02\frac{s}{h} \tag{5}$$

$$C_{Ds} = 0.018C_L^2 \tag{6}$$

Lift coefficient (CL) can be obtained using equations 7 and 8

$$C_L = 2\frac{s}{c}\cos\alpha_m(\tan\alpha_1 - \tan\alpha_2) - C_D\tan\alpha_m \tag{7}$$

$$\tan\alpha_m = 0.5(\tan\alpha_1 + \tan\alpha_2) \tag{8}$$

Equation (9) below is the summation of the equations (4), (5) and (6) which gives the stage overall drag coefficient.

$$C_D = C_{Dp} + C_{Da} + C_{Ds} \tag{9}$$

Also, an assumption of 50% stage reaction was made so as to minimize adverse pressure rise from either the rotor or stator blade surfaces [14]. With this assumption, it implies that the pressure rise is equally distributed between the stator and rotor where α1=α3 and α0=α2

The temperature rise coefficient is given by Equation 10.

$$\frac{C_p\Delta T_s}{0.5U^2} = 2\lambda\left(\frac{V_a}{U}\right)(\tan\alpha_1 - \tan\alpha_2) \tag{10}$$

Polytropic or stage efficiency can be calculated using Equation 11.

$$\eta_p = 1 - \left(\frac{2}{\sin\alpha_m} * \frac{C_D}{C_L}\right) \tag{11}$$

While Pressure rise coefficient is obtained using Equation 12.

$$\frac{\Delta P_s}{0.5\rho U^2} = \eta_P \left(\frac{c_p \Delta T_s}{0.5U^2} \right) \quad \textbf{(12)}$$

It is worth mentioning that all the blades in the cascade were assumed to have same aerodynamics in order to calculate the isentropic efficiency for the different cases. Hence, design pressure ratio of the adopted engine was used to calculate the isentropic efficiency for the different cases investigated. The isentropic efficiencies and flow coefficients were calculated using Equations 13 and 14 respectively.

$$\eta_c = \frac{PR^{\left(\frac{\gamma-1}{\gamma}\right)} - 1}{PR^{\left(\frac{\gamma-1}{\gamma\eta_p}\right)} - 1} \quad \textbf{(13)}$$

$$\phi = Va/U = 1/(\tan\alpha 1 + \tan\alpha 2) \quad \textbf{(14)}$$

2.8 Engine Performance Simulations

The output parameters of flow capacity and calculated isentropic efficiency obtained from the correlations of the cascade data were implanted into gas turbine performance simulation software, to simulate the overall performance of the engine for the different conditions investigated. To simulate the different conditions, a twin shaft engine specification data obtained from open domain (Courtesy General Electric), were used to model the engine configuration in the GASTURB simulation software.

Figure 8 shows the industrial gas turbine engine configuration model adopted for the investigations, while Table 2 presents the design point performance specifications. Screen shot of modeled design point simulation interface is presented in Figure 9.

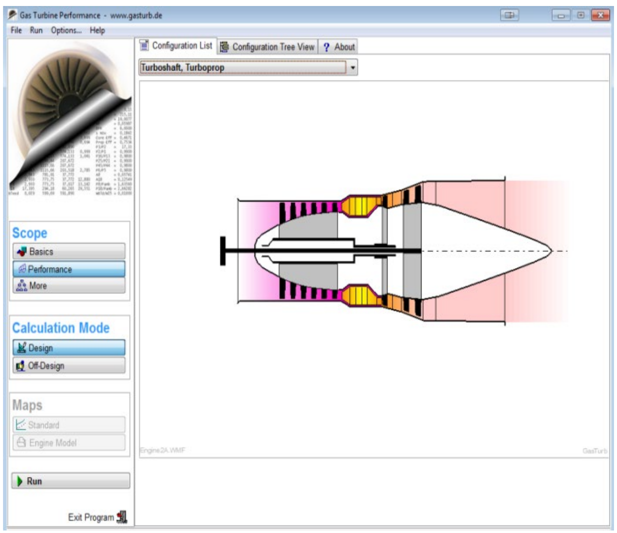

Figure 8: Industrial gas turbine engine configuration in GASTURB simulation software interface [15]

Table 2: Engine design specifications (Courtesy of General Electric)

Design parameters	Units
Power output	25MW
Thermal efficiency	36
PR	18
Exhaust temperature	839K
Exhaust flow	70.5kg/s
Heat rate	9705kJ/kWh

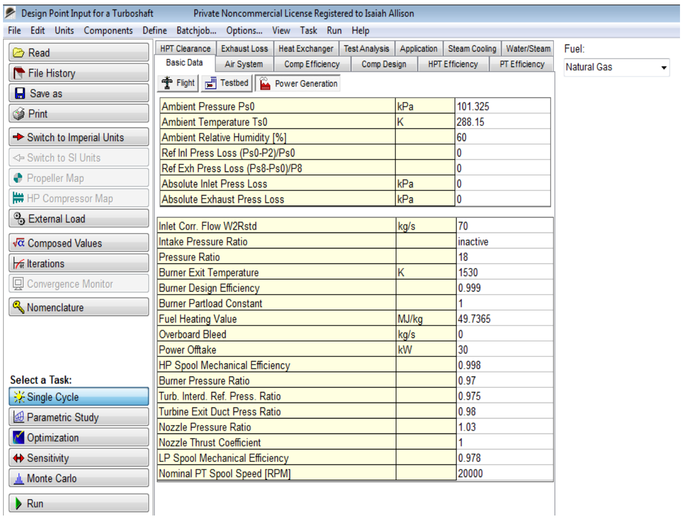

Figure 9: Modeled design point Screen shot

3. Results and Discussion

3.1 Droplet Sizing Analysis

Figure 10 shows the droplet size cumulative distribution curves for the preheated wash fluid and non-preheated cases. As can be seen from the figure, the curves are similar. However, it was observed that there was interlapping of the curves between the preheated and non-preheated plots. This interlapping effect can be attributed to the fluid temperature being warmer than the surrounding air, thereby resulting in transfer of heat between the wash fluid and ambient air. At 90bar injection pressure, when the wash fluid was heated from 15 to 50°C, the mean droplet size reduced from 81 to 78µm. This reduction in droplet size can be attributed to the reduced viscosity of the wash fluid due to heating, thereby resulting in finer droplet sizes. Though, the reduction in droplet size with heating of wash fluid is relatively small because of the low viscosity of water; for high viscosity fluid, heating can result in significant reduction in droplet sizes.

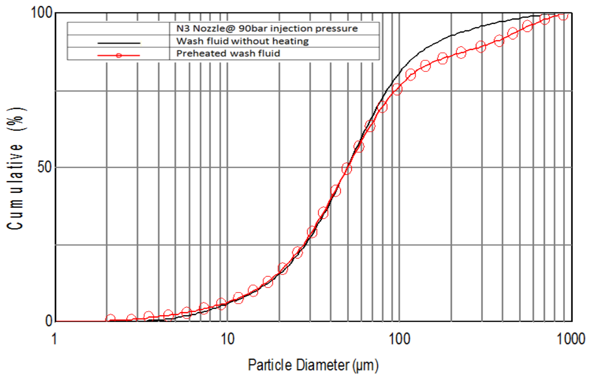

Figure 10: Cumulative distribution curves preheating the wash fluid and that without preheating at a given injection pressure

Figure 11 shows the cumulative distribution curves for varying injection distances. As can be seen, droplet size distributions increased with injection distance. Considering particle diameter of 100µm from the figure, it is obvious that the cumulative percentage of droplet size at an injection distance of 200mm is larger than that of 50mm. The increase in droplet size with injection distance can be attributed to a secondary process of droplets collision and coalescence. These findings agree with the study Jasuja [16].

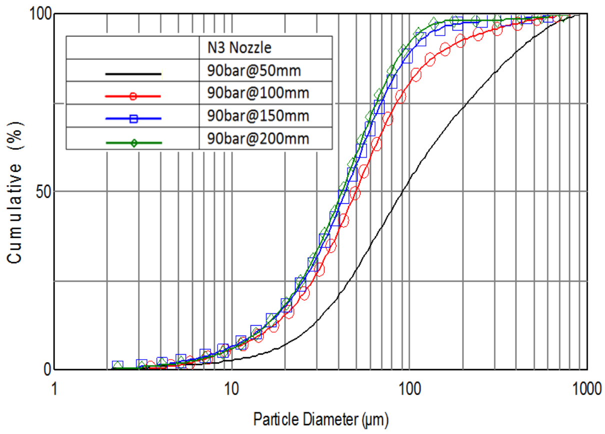

Figure 11: Cumulative distribution curves for varying injection distance

3.2 Compressor Cascade Result Analysis

Figure 12 shows the blade aerodynamic performance plot of total pressure loss coefficient for different conditions. When fouled blades were washed separately with preheated and non-preheated demineralized water, the plots show a decrease in total pressure loss coefficient for both washed conditions from the fouled case. The mean total pressure loss coefficient decreased from a fouled case of 0.109 to 0.079 and 0.082 for preheated wash fluid and non-preheated respectively. However, when the preheated and non-preheated cases were compared, there was slight difference in total pressure loss coefficient for different cases. The mean total pressure loss coefficient for blades washed preheated demineralized water is 0.079 as against non-preheated case of 0.82.

When exit airflow angle of the two cases were compared as shown in Figure 13, the plots show that blades washed with non-preheated demineralized water produced a lower mean exit flow angle of 34.15degrees as against 34.45degrees for preheated case (see Table 3). The mean values of aerodynamic parameters obtained in this study are similar to the findings of Fouflias et al.[17] and Igie et al.[18]. Although, slight differences were observed; it could be attributed to the level degradation applied in the different studies or discrepancies arising from the measuring instrument. For instance, Fouflias [17] applied different levels of roughness, ranging from clean condition (0µm) to particle sizes of 354µm for roughened blades. For clean condition of 0µm and blades roughness of 354µm, the author reported total pressure loss coefficient for both cases as 0.16 and 0.35 respectively. While the exit flow angle for the two conditions were 34 and 39.5 degrees respectively. Similarly, Igie et al.[18] reported total pressure loss coefficient of 0.056, 0.136 and 0.097 for clean, fouled and washed cases respectively.

Table 3: Mean Values of blade aerodynamic parameters

Conditions	α total (deg)	ω
Clean	33.35	0.050
Fouled	35.44	0.109
Washed with Preheated demineralised Water	34.15	0.082
Washed with non- preheated demineralised water	34.45	0.079

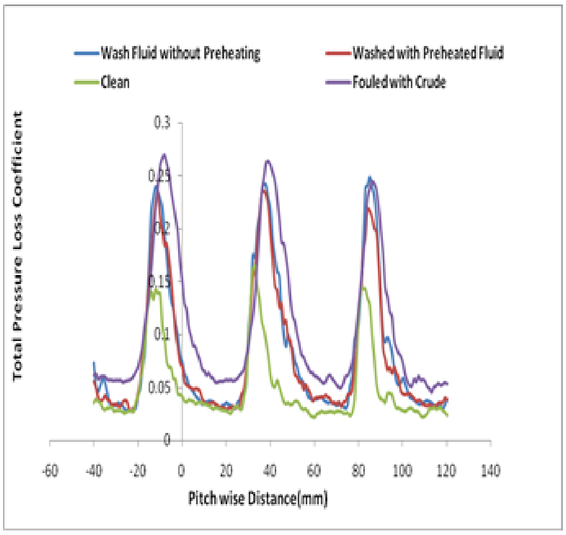

Figure 12: Total pressure loss coefficient against Pitch distance

From the aerodynamic performance plots analyzed, it is obvious that the preheated washed case produced a slightly better recovery in total loss coefficient than the non-preheated washed case. However, for the plots exit flow angle shown in Figure 13, the fouled blades washed with non-preheated wash fluid produced a lower exit flow than the case washed with preheated washed fluid. The difference seems so insignificant for aerodynamic performance parameters for both cases. Therefore, a valid conclusion cannot be drawn in relation to which case produced a lower/ higher total pressure loss coefficient and/or exit flow angle. Also, these findings validate the measurement of droplet size earlier conducted for the preheated wash fluid and non-preheated condition, where the difference in measured droplet sizes for the two cases were insignificant. Since there were no significant differences in measured wash fluid droplets for two cases, one can agree with outcome of the aerodynamic performance results obtained. In addition, when the preheated and non-preheated conditions were compared visually, using the blade aerodynamic performance, there was little or no difference. Despite the fact that slight discrepancies were observed, regarding the trend of total pressure loss coefficient and exit flow for the two cases, the

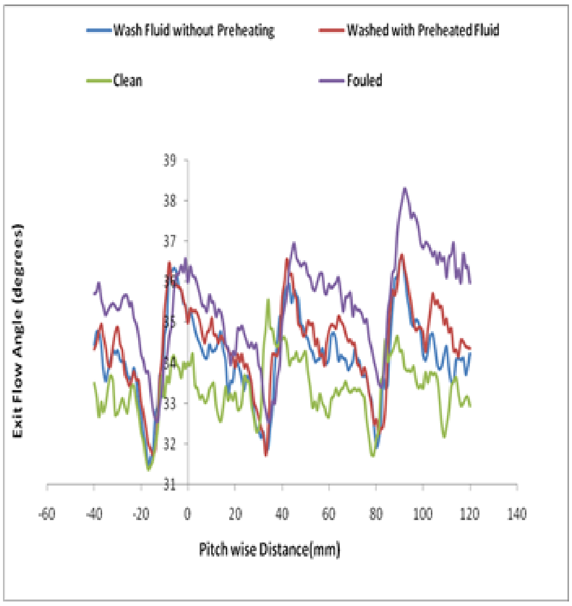

Figure 13: Exit flow angle against Pitch distance

3.3 Engine Performance Analysis

Table 4 presents the values of flow coefficient, polytropic and isentropic efficiencies obtained, using the Howell's method to correlate the cascade readings to an actual stage performance data. While Table 5 shows the variation/reduction in isentropic efficiency and non-dimensional flow values implanted into the software, to simulate the performance of the different cases.

Table 4: Polytrophic and isentropic efficiencies and non-dimensional mass flow of clean, fouled and washed cases

Conditions	Polytropic Efficiency	% Variation Polytropic Efficiency	Isentropic Efficiency	% Variation Isentropic Efficiency	% variation Non-dimensional Mass flow Rate
Clean	91.9	0.0	88.1	0.0	0.0
Fouled	86.7	5.6	80.6	8.5	2.7
Washed with Preheated Demin. water	89.7	2.4	84.9	3.6	1.3
Washed with Demin water without Preheating	89.6	2.5	84.8	3.7	1.3

Table 5: Variation/reduction in Isentropic Efficiency and Non-dimensional mass flow rate

results seem valid. This is because significant differences in total pressure loss coefficient and exit flow angle were observed between the fouled and washed cases. Also, the slight discrepancies observed when the blades washed with preheated and the non-preheated case can be attributed to the resolution of the instrument used in obtaining the compressor cascade performance data in this study.

Conditions	% Variation Isentropic Efficiency	% variation Non-dimensional Mass flow Rate
Clean	0.0	0.0
Fouled	8.5	2.7
Washed with Preheated Demin. water	3.6	1.3
Washed with Demin water without Preheating	3.7	1.3

Figure 14 shows the engine performance of plot of thermal against the different conditions. As can been seen from the figure, when fouled blades washed with preheated and non-preheated demineralized water cases were compared, there was little or no difference in thermal efficiency. It was only a 0.1 percentage change that was recorded between the two cases. However, when the fouled and blades washed with preheated demineralized water were compared, the results show an improvement in thermal by 5.37 percent. The plot of fuel flow in Figure 15 follows a similar pattern to thermal efficiency when fouled blades washed with the preheated and non-preheated demineralized water cases were compared. Also, when the fouled and washed with preheated demineralized water were compared for the fuel flow plots, a percentage change of 5.38 was recorded. These results agree with the findings in Engdar et al.[5],where the authors stated that preheating the wash fluid has no significant effect on the cleaning effectiveness. Also, these results validate the blade aerodynamic performance results presented earlier.

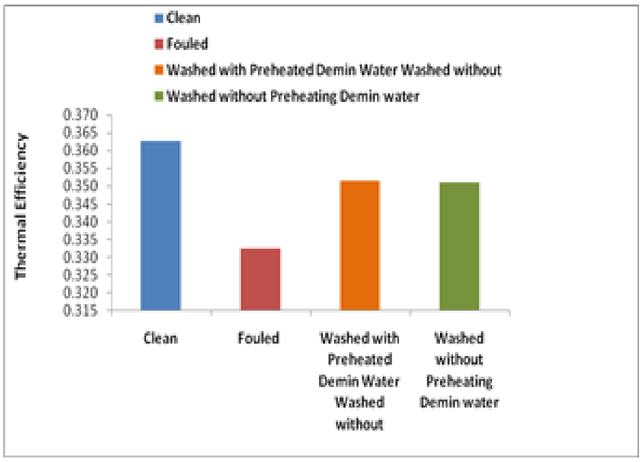

Figure 14: Plot of Thermal Efficient at Different Conditions

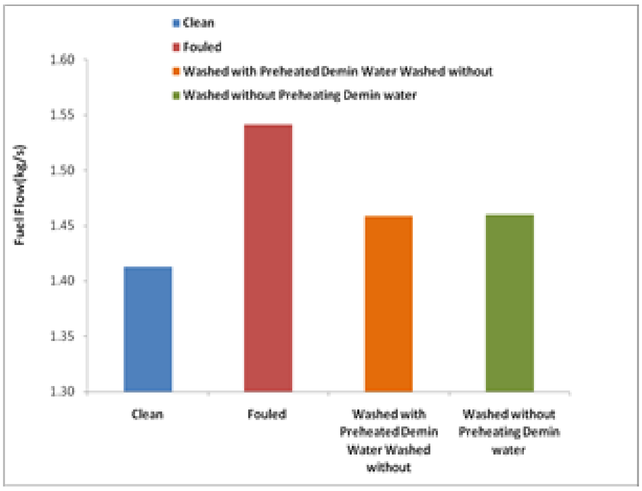

Figure 15: Plot of Fuel Flow at Different Conditions

4.0 Conclusion

An experimental investigation of the influence of increased wash fluid temperature on online compressor cleaning effectiveness in a suction wind tunnel compressor cascade is presented in this study. Compressor cascade blades of a suction wind tunnel were fouled by applying crude oil uniformly on the blade surfaces, and subsequently ingesting carborundun particles into the tunnel, to create an accelerated fouling. Washing of the fouled blades were conducted using single flat fan nozzle, where preheated and non-preheated demineralized water were used separately to wash the fouled blades. The outcome of the findings from the study is presented below:

i. Fouled blades washed separately with either preheated or non-preheated demineralized water produced a better blade aerodynamic performance than the fouled condition

ii. Little or no difference was observed visually, when fouled blades washed with preheated demineralized water and non-preheated cases were compared in terms total pressure loss coefficient and exit airflow angle.

iii. Engine Performance parameter plots of thermal efficiency and fuel flow show that the difference in performance between the preheated and non-preheated was so insignificant.

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NOMENCLATURE

C_D	drag coefficient
C_{Da}	annulus drag coefficient
C_{Dp}	profile drag coefficient
C_{Ds}	drag coefficient for secondary losses
c_p	specific heat at constant pressure
C_L	lift coefficient
P	total pressure
P_s	static pressure
PR	pressure ratio
T	total temperature
T_s	static temperature
TET	turbine entry temperature
TP	trailing passage or pitch
μ m	micrometer
W	non-dimensional air mass flow

References:

[1] J. Fielder “Evaluation of zero compressor wash routine in RN service,” *ASME Turbo Expo*, 16-19 June 2003, Atlanta, GA, pp. 543, 2003.

[2] G. Foss “On-line water Wash Test,” *Kvaerner Energy a.s., Test Cell*, Report No.690280, 1999.

[3] J. Stalder “Gas turbine compressor washing state of the art: Field experiences,” *Journal of Engineering for Gas Turbines and Power* 123: 363-370, 2001.

[4] F. C. Mund and P. Pilidis “Gas turbine compressor washing: Historical developments, trends and main design parameters for online systems,” *Journal of Engineering for Gas Turbines and Power* 128:344-353, 2006.

[5] U. Engdar, R. Orbay, M. Genrup and J. Klingmann “Investigation of the two-phase flow field of the GTX100 compressor inlet during off-line washing,” *ASME GT* 2004-53141, 2004.

[6] E. Syverud “Axial compressor performance deterioration and recovery through online washing,” *(PhD thesis), Norwegian University of Science and Technology*, 2007.

[7] S. L. Dixon “Fluid mechanics and thermodynamics of Turbomachinery,” 5th ed, Butterworth-Heinemann, U.S.A, 1998.

[8] J. Gostelow and D. Pollard “Some experiments at low speed on compressor cascades(Low speed cascade tunnel experiments for improvement of airflow and testing techniques, noting porous sidewall suction effect on axial velocity changes),” *ASME, Transactions, Series A-Journal of Engineering for Power* 89:427-436, 1967.

[9] R. B. Abernethy, R. P. Benedict and R. B. Dowdell "ASME measurement uncertainty," *Journal of Fluids Engineering,* vol. 107, no. 2, pp. 161-163. 1985.

[10] https://digibuo.uniovi.es/dspace/bitstream/handle/10651/6953/Cylindrical%20three-hole.pdf;jsessionid=EC9D149C281BDA9F349AA86AF2DE53F8?sequence=1; Accessed 24 November, 2020.

[11] R. Agbadede, P. Pilidis, U. L. Igie and I. Allison “Experimental and theoretical investigation of liquid injection droplet size influence on online compressor cleaning effectiveness for industrial gas turbines,” *Elsevier- The Journal of Energy Institute* 88:414-424, 2014.

[12] C. B. Meher-Homji and A. Bromley “Gas turbine axial compressor fouling and washing," 33rd *Turbomachinery Symposium,* Houston, TX, pp. 163, 2004.

[13] A. Howell “Fluid dynamics of axial compressors,” *Proceedings of the Institution of Mechanical Engineers* 153:441-452, 1945.

[14] A. Howell "Design of axial compressors," *Proceedings of the Institution of Mechanical Engineers* 153: 452-462, 1945

[15] J. Kurzke "GasTurb 11-Design and off-design performance of gas turbines," 2007,

[16] A. K. Jasuja "Structure of high throughput, dense pressure atomized elliptical sprays under high ambient air pressure conditions," *ILASS-Europe* 8-10 September, 2008.

[17] D. Fouflias "An experimental and computational analysis of compressor cascades with varying surface roughness," *Disertation, Cranfield university, UK*, 2009.

[18] U. L. Igie, P. Pilidis, D. Fouflias and K. Ramsden "On-line compressor cascade washing for gas turbine performance investigation," *Proceeding of ASME Turbo Expo, 6-10 June 2011.*