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# A Proposed Method for Assessing the Susceptibility of Axial Compressors to Fouling

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## ABSTRACT

Although the overall effect of compressor fouling on engine performance has been recognized for many years, remarkably little has been published on the quantifiable effects. Mathematical modelling of compressors using stage stacking methods has recently been used for a systematic study of compressor fouling and earlier investigations led to an interest in the effects of engine size and compressor stage loading. This paper presents a proposed index showing the susceptibility of compressors to fouling, which could be useful in helping operators to determine clean-up intervals. Three engines of widely differing performance were used in developing this index and additional operator experience would be useful in confirming its validity.

## NOMENCLATURE

| | |
|---|---|
| $C_a$ . . . . | Stage inlet axial velocity |
| $C_p$ . . . . | Specific heat |
| $\dot{m}$ . . . . | Mass Flow Rate |
| $PR_s$ . . . . | Stage pressure ratio |
| $T_{os}$ . . . . | Stage inlet total air temperature |
| $\Delta T_{os}$ . . . . | Stage total temperature rise |
| $U$ . . . . | Tangential blade speed at mean radius |
| $\gamma$ . . . . | Ratio of the specific heats |

## INTRODUCTION

It has been well recognized for many years that the performance of industrial gas turbines can be significantly affected by compressor fouling (Upton, 1974). Fouling can result from operation in dirty industrial environments, which is to be expected, but can equally be found in remote rural locations due to problems such as insects, tree saps, pollen etc. Another source of fouling can be due to oil leakage from bearings. Effective inlet filtration can alleviate, but not eliminate, the problem and operators are still faced with the decision on what basis to schedule compressor cleaning. If cleaning is delayed too long full performance recovery cannot be obtained, but excessively frequent cleaning causes considerable disruption to operations.

Compressor fouling leads to reduction in both power output and overall efficiency of an engine. For example, a 1 percent reduction in axial compressor efficiency can account for a $1\frac{1}{2}$ percent increase in the heat rate for a given power output (Scott, 1977). Since neither the efficiency nor the power output of engines is usually measured directly, fouled engines can operate inefficiently for a long period of time. In the past, the operators did not consider the inefficient operation of an engine as a serious economic problem, mainly due to the availability of cheap fuels. However, rising fuel costs in the past two decades have caused the efficient operation of gas turbines to become a matter of prime importance. Presently, compressor fouling has been identified by both users and manufacturers of gas turbines as the most dominant factor contributing to performance deterioration in industrial applications. It is noted that the economic loss as a result of compressor fouling runs into millions of dollars annually (Reid, 1977).

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In the last few years discussions have been held with a number of operators and it became clear that a better physical understanding of the fouling process was needed. In one case an operator claimed that fouling was not a problem, but the manufacturer recognized that the engine was extremely badly fouled and the problem did not show up simply because of the continuous operation at a relatively low power level.

Operators require a simple and reliable method for detecting fouling at a level where performance can be restored by compressor cleaning. Recent studies have been aimed at modelling compressor behaviour so that a systematic study of fouling effects can be carried out to provide a knowledge base for further operations

In order to run a gas turbine at a particular power setting at its best efficiency, some users have adopted the policy of compressor cleaning, regardless of local factors such as environment, season and engine condition, on a fixed-interval basis such as approximately 250-hour interval for an on-stream washing and 1000-hour interval for a soak wash (Reid, 1977). An on-stream wash can return 75% of the lost efficiency. A soak wash can restore the engine virtually to "as new" condition. Dry cleaning using nut shells, carried out at full speed reduces the losses due to the fouling of the engine compressor by only approximately 20-30% and may clog up internal passages.

On-condition cleaning of the compressors is used by some users. This method requires a close monitoring of the engine parameters and development of a system that can predict performance degradation as a result of compressor fouling. The major obstacle in the way of successful on-condition cleaning is the development of a performance monitoring system which has to be tailored to the needs of a particular user for a particular application and environmental condition; this puts considerable economic constraints on the development of such systems which makes the adoption of the on-condition cleaning method practically impossible by some users. It is probable that if performance is allowed to deteriorate to the point where it can be detected, a considerable waste of fuel will be incurred. Despite such difficulties, on-condition cleaning is gaining popularity with users. The programs developed for this purpose vary in their complexities as well as in their scope; some are completely automatic and others are semi-automatic. Also, the number of parameters that are monitored vary from one system to another. Scott (1977) reports on a simple system of monitoring the intake depression as a means of predicting the compressor fouling of two Avon powered units with satisfactory results. There is not enough data available on the complex automated monitoring systems to demonstrate their capabilities of reaching the required levels of accuracy and reliability for detecting the small changes in performance which could indicate compressor fouling. MacIsaac (1981) presents a vivid and comprehensive review of some of the well known monitoring systems.

Some of the monitoring systems are nothing but trend monitoring and analysis of a few engine performance parameters. Saravanamuttoo (1974, 1979) describes a very simple and effective system for on-site monitoring of the engines. These systems are based on the premise that the test data are reliable. In order to isolate faulty readings, Agrawal *et al* (1979) have published a method of identifying instrument faults. The method proposed by Williams (1981b) is similar to the gas path analysis of Saravanamuttoo (1979) except that he has assumed the compressor efficiency can be considered constant over a good part of the operating range. Several authors (Scott, 1979; Agrawal *et al*, 1979 and Saravanamuttoo and MacIsaac, 1983) attempted to derive correlations that link the engine problems to changes in the monitored engine parameters.

## SIMULATION OF COMPRESSOR PERFORMANCE

With the advent of faster and cheaper micro-computers the manual logging of engine data is gradually being replaced by remote data-acquisition systems. This has made it possible to extend the scope of research to predictive simulation techniques aimed at producing an accurate computer model of engines and their various components and processes. A systematic investigation of compressor fouling, requires that the effect of fouling of a single stage be modelled correctly and hence the overall effects of all stages of an engine compressor can be obtained from the effects of the fouling of the constituent stages.

The stage stacking technique, which was extensively studied by Stone (1958) and improved by Howell and Calvert(1978), can be used as a tool to produce the compressor map for an engine from the performance of its individual stages. A successful application of the stage stacking technique depends to a great extent on the knowledge of the stage characteristics for all stages of a compressor. The stage characteristics for the compressor of an engine are in turn obtained by the engine manufacturer during its development phase; such information, if not published, would remain proprietary material of the manufacturer. When stage characteristics from test data are not available, there are alternate methods that can estimate them fairly accurately from the geometry of a stage, blade rows, and the flow pattern. The performance of a single stage is normally presented in the form of non-dimensional stage characteristics as follows:

$$\phi = \frac{C_a}{U} \qquad \text{Flow coefficient;}$$

$$\psi = \frac{C_p T_{os}\left(PR_s^{\frac{\gamma-1}{\gamma}} - 1\right)}{U^2} \qquad \text{Pressure rise;}$$

$$\varsigma = \frac{C_p \Delta T_{os}}{U^2} \qquad \text{Temperature rise;}$$

and

$$\eta = \frac{T_{os}\left(PR_s^{\frac{\gamma-1}{\gamma}} - 1\right)}{\Delta T_{os}} = \frac{\psi}{\varsigma} \qquad \text{Efficiency}$$

A set of parameters called 'generalized characteristics' can be defined; these are the ratios of each characteristic parameter and the value of the same parameter at an arbitrary operating point, such as the point of maximum efficiency that may be designated by subscript ref:

$$\frac{\phi}{\phi_{ref}}, \quad \frac{\eta}{\eta_{ref}}, \quad \frac{\varsigma}{\varsigma_{ref}}, \quad \frac{\psi}{\psi_{ref}}, \quad \text{and} \quad \frac{\varsigma/\varsigma_{ref}}{\phi/\phi_{ref}}$$

Figure 1 shows a plot of $\psi/\psi_{ref}$ versus $\phi/\phi_{ref}$ assembled from a number of openly published sources using stage pressure rise data (Muir *et al*, 1988). A generalized efficiency relation developed by Howell (1950) as shown in Figure 2, which has also been used by Muir *et al* (1988), was found to be adequate for modelling the off-design performance of gas turbines due to compressor fouling.

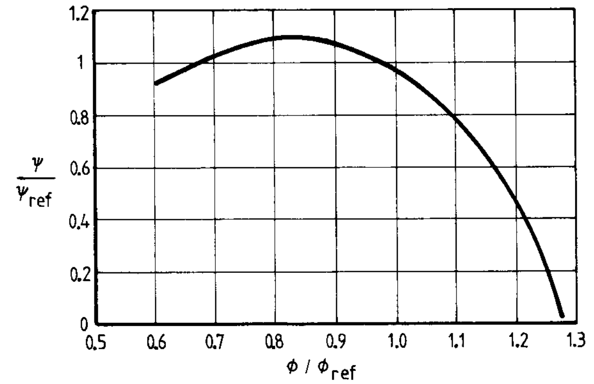


FIGURE 1: GENERALIZED STAGE PRESSURE COEFFICIENT CURVE (MUIR et al, 1988).

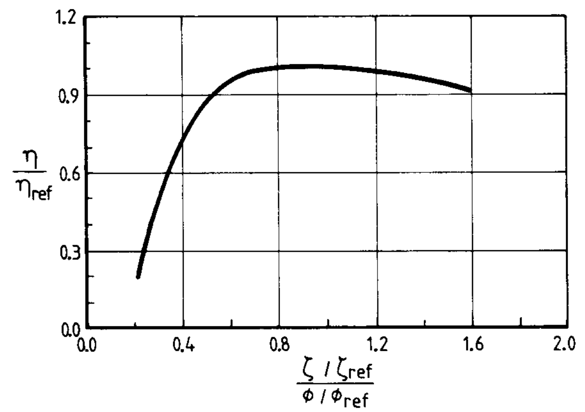


FIGURE 2: GENERALIZED STAGE EFFICIENCY CURVE (MUIR et al, 1988).

The stage characteristics for the compressor of an engine can be estimated by the use of a numerical search procedure; these characteristics with the aid of stage-stacking technique allows one to find the overall compressor map for the engine. At a given compressor speed and mass flow established by the operating line data points, the numerical search procedure begins by assuming specific values for the reference point of each stage. From the generalized characteristics curves (Figures 1 & 2) the performance map for each stage is obtained and the performance of the individual stages are then stacked to find the overall compressor pressure ratio, PR, and the temperature rise ratio, $\frac{\Delta T}{T}$. If these values do not agree with the operating line data, the procedure is repeated. Muir *et al* (1988) and Aker and Saravanamuttoo (1988) using the above procedure have produced the compressor performance maps for two industrial gas turbines that agree with the manufacturer's data. The present authors have also used the same procedure for producing the compressor performance map for another industrial engine whose specification data appear in Table 1; this is shown in Figure 3, which is also in conformity with the manufacturer's data.

**TABLE 1**

**Design Specification Data for the Three Engines**

| DESCRIPTION | | Engine # 1 | Engine # 2 | Engine # 3 |
|---|---|---|---|---|
| $\dot{m}$ | kg/s | 17.2 | 65.8 | 115.7 |
| PR | | 9:1 | 17.2:1 | 8.7:1 |
| No. of Stages | | 11 | 16 | 17 |
| Power Output | kW | 2,850 | 20,134 | 28,200 |
| $N_{gg}$ | rpm | 15,015 | 9,160 | 5,100 |
| $R_m$ | m | 0.176 | 0.279 | 0.556 |
| U | m/s | 277 | 268 | 297 |

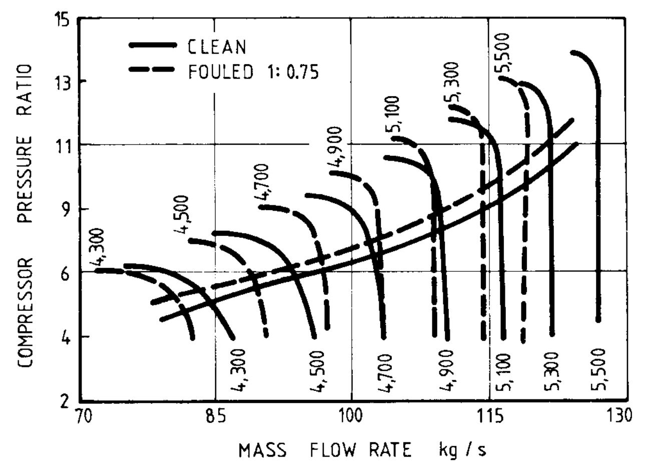


FIGURE 3: SIMULATED CLEAN AND FOULED COMPRESSOR MAPS FOR ENGINE NO. 3.

## SIMULATION OF COMPRESSOR FOULING

Lakshminarasimha and Saravanamuttoo (1986) simulated the fouling of individual compressor stages in the NACA five-stage compressor described by Sandercock *et al* (1954). In order to reflect the loss in performance due to fouling, the inlet flow was reduced by an arbitrary factor of 3% and 8% for light and heavy fouling respectively, while the efficiency was lowered by 1%. Aker and Saravanamuttoo (1988) investigated the linear progression of fouling in compressor stages and how it could be best simulated. Basing their arguments on field data and on their discussions with the users of gas turbines, they noted that the impact of fouling on the front stages of a compressor is more than that on the rear stages. They concluded that it was more appropriate to adopt a model that could simulate linear progression of fouling into compressor stages. For this purpose, they reduced the flow coefficient, $\phi$, by a certain percentage say $k_1$ %, through a linear progression from $n \times k_1$ for the first stage to $k_1$ for the $n^{th}$ stage. In a similar manner, the stage efficiencies were reduced using another factor $k_2$. They have referred to this model of fouling by designation $(k_1 : k_2)$. After modifying the flow coefficients and efficiencies for the " clean stages", they applied stage stacking technique in order to produce a compressor map for a fouled compressor whose match point with the existing turbines produced the modified operating point for the engine. In this way they were able to simulate fouling for two industrial engines; the stage stacking procedure gave excellent correlation with field data for clean engines and manufacturer's data, but no field results were available for fouled compressors. Similarly, the present authors simulated the fouling of an industrial gas turbine in the power range of 30 MW. The simulated compressor maps for the "clean and fouled" compressors for this engine, using the fouling model (1: 0.75), are shown in Figure 3.

## FOULING SEVERITY

Quantifying the effects of compressor fouling has a number of obvious applications and benefits to the users for diagnostics, engine performance monitoring, maintenance and operations. Such applications can be further extended by investigation of the sensitivity of engine performance to compressor fouling; this requires the identification of the significant engine parameters which can indicate the detrimental effects of compressor fouling. There have been references in the literature to stage loading as being one of the significant parameters in the impact of fouling on engine performance. Power output has also been mentioned. Saravanamuttoo and Lakshminarasimha (1985) were of the opinion that the performance of a small engine of less than 1 MW is expected to deteriorate more as a result of fouling than does a larger machine. Blauser (1984) states that the performance loss due to fouling of the gas turbine compressor, ranges from 3 to 10 percent of the output horsepower for axial design and substantially less for centrifugal designs. The amount of loss depends on the control parameters employed. If the gas generator's primary control parameter is speed, a lower degradation, in terms of power, will be observed than that which occurs if exhaust temperature is the primary control.

The existing knowledge concerning an engine's sensitivity to the fouling of its compressor has remained, to a large extent, speculative and suffering from contradictions; these conclusions were generally drawn from observations of engine operation without examining the repeatability of such observations. Therefore this study is intended to fill the vacuum through the development of a systematic method for the examination of the likely parameters which could influence the severity of performance degradation due to fouling. For this purpose three industrial gas turbines, ranging in power output from 3MW to 30 MW, were selected for fouling simulation. The significant parameters for these engines, which have been designated by numbers 1 through 3, are tabulated in Table 1.

Furthermore, the fouling model (1: 0.75) for the simulation of fouling of these engines was adopted. The fouling simulation results for engines number 1 and 2 have already been reported by Aker and Saravanamuttoo (1988). Similar results for engine number 3 have been produced, using the same fouling model. A critical examination of these results forms the basis for the investigation of engine sensitivity to fouling.

## RESULTS AND DISCUSSION

The computer model for the "clean compressor" of engine number 3 was produced by stage stacking. The specifications for the compressor and the engine were obtained from its manufacturer and from published literature such as McQuiggan (1975). Figure 3 shows the simulation of the compressor map and the predicted operating line for this engine. A linear progressive fouling model of (1: 0.75) through the $8^{th}$ stage of the compressor was applied and a compressor map for the fouled compressor was obtained and subsequently an operating line for the fouled engine was predicted. These results are also displayed in Figure 3. Although a slight shift in the surge line is observed, the shift of the operating line is more significant and it is clear that the fouled engine operates at a lower surge margin. This engine actually makes use of variable power turbine nozzles, so the CDP can be maintained at a fixed value making fouling hard to predict.

In Figures 4 through 7, the simulation of the impact of the fouling model (1: 0.75) on compressor delivery pressure, power output, mass flow rate, and thermal efficiency for the three engines are shown. The general pattern exhibited in these graphs for various parameters is consistent, regardless of the performance parameter selected. The pattern indicates that the three engines can be listed on the basis of the severity of the performance degradation as a result of the compressor fouling in the ascending order of engine number 1, 2, and 3. For further analysis Figure 4, which shows the pattern in the compressor delivery pressure drop as a result of compressor fouling, is considered. Decreases in CDP for engines 2 and 3 are closer to each other compared to those for engines 1 and 2. The decrease in the CDP as a function of % of number of stages fouled for the three engines is almost linear up to 25% progress into the compressor stages. For other performance parameters shown in Figures 5, 6, and 7, more divergence from linearity at higher levels of fouling is observed, particularly for engine number 3 .

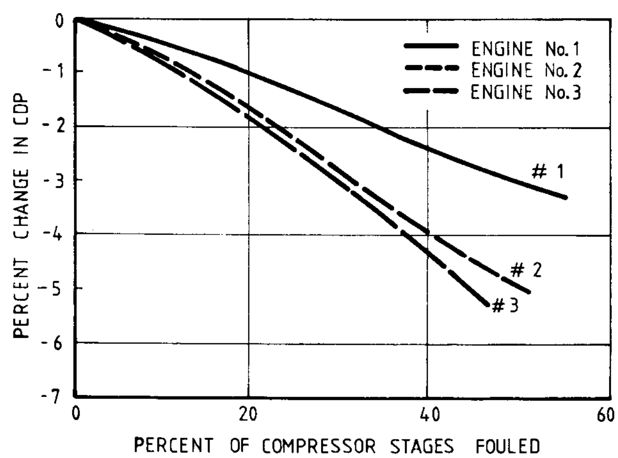


FIGURE 4: VARIATIONS OF THE PERCENTAGE CDP DROP WITH THE PERCENTAGE OF STAGES FOULED FOR THE THREE ENGINES.

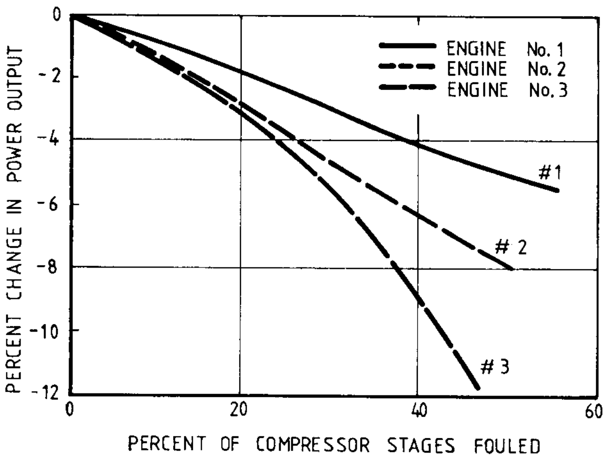


FIGURE 5: VARIATIONS OF THE PERCENTAGE POWER OUTPUT DROP WITH THE PERCENTAGE OF STAGES FOULED FOR THE THREE ENGINES.

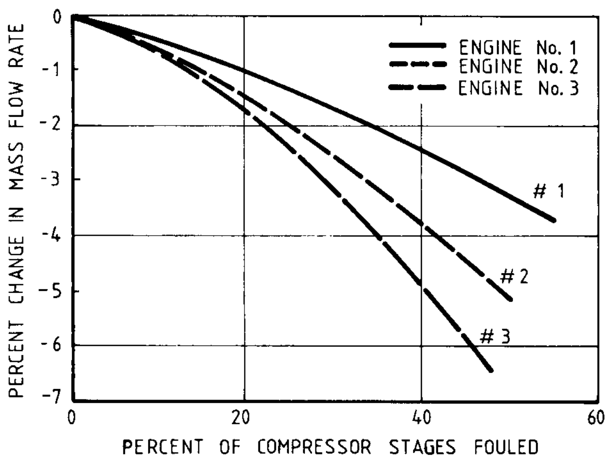


FIGURE 6: VARIATIONS OF THE PERCENTAGE MASS FLOW RATE DECREASE WITH THE PERCENTAGE OF STAGES FOULED FOR THE THREE ENGINES.

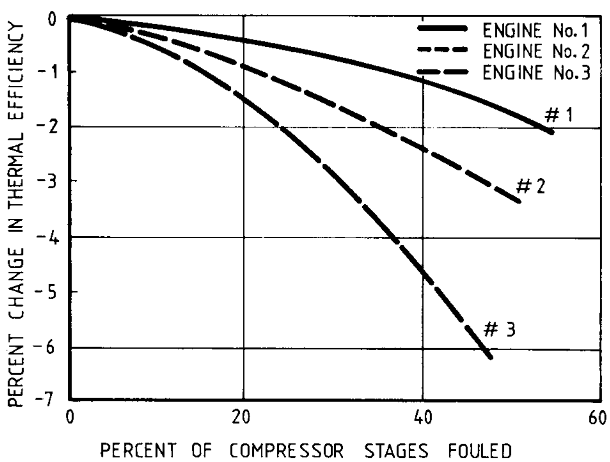


FIGURE 7: VARIATIONS OF THE THERMAL EFFICIENCY DROP WITH THE PERCENTAGE OF STAGES FOULED FOR THE THREE ENGINES.

**TABLE 2**

**Dimensionless Groups and Simulated Results for the Three Engines**

| DESCRIPTION | Engine # 1 | Engine # 2 | Engine # 3 |
|---|---|---|---|
| $\Delta T$ per Stage °C | 28.2 | 25.9 | 17.9 |
| $\frac{C_p \Delta T_{stg}}{0.5U^2}$ | 0.737 | 0.724 | 0.407 |
| $\frac{kW}{\dot{m}U^2}$ | 2.16 | 4.26 | 2.76 |
| $\frac{kW}{\dot{m}C_p\Delta T_{stg}}$ | 5.88 | 11.77 | 13.56 |
| CDP Drop at 40% fouling | 2.40 | 3.95 | 4.45 |

On the other hand some parameters calculated from the specification data for the three engines, as tabulated in Table 2, follow a pattern consistent with the above-mentioned observations. This indicates that a likely relationship might exist between the corresponding parameters and the fouling sensitivity of the engines. A few dimensionless groups can be formed from the engine parameters, which in addition to the stage loading, $\Delta T_{stg}$, are discussed below; the tangential velocity of the first row of blades at the meanline, the mass flow, and the power output are denoted by U, $\dot{m}$ and kW respectively.

- $\Delta T_{stg}$ This parameter is a measure of stage loading. For ease of reference and analysis, the percentage drop in CDP as a result of fouling 40% of the stages for all three engines have been recorded along with the other data in Table 2. The tabulated data indicate that on the basis of higher stage temperature rise, the engines follow the order of: engine no. 1, engine no. 2, and engine no. 3; while on the basis of sensitivity to fouling as apparent from the % CDP drop, the engines follow a reverse order. The relationship between these data become more clear from a glance at Figure 8a. This figure shows that a clear-cut correlation between the

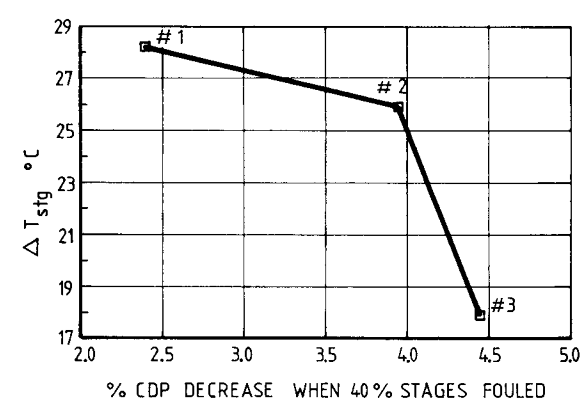


FIGURE 8a: VARIATIONS OF STAGE TEMPERAURE RISE WITH PERCENTAGE CDP DECREASE FOR THE THREE ENGINES WHEN 40% OF STAGES FOULED.

stage loading of an axial compressor and its sensitivity to fouling does not exist, though it may be stated that the performance degradation of gas turbines with smaller stage loadings is more severe as compared to those with larger stage loadings.

- $\frac{C_p \Delta T_{stg}}{0.5U^2}$ This is the stage-temperature-rise coefficient based on the tangential velocity of rotor blades at their mean radius for the first stage, which is a non-dimensional value for the stage loading. A glance at Table 2 and Figure 8b reveals that practically what was said for stage temperature rise holds also true for this coefficient; all three engines had very similar values of U.

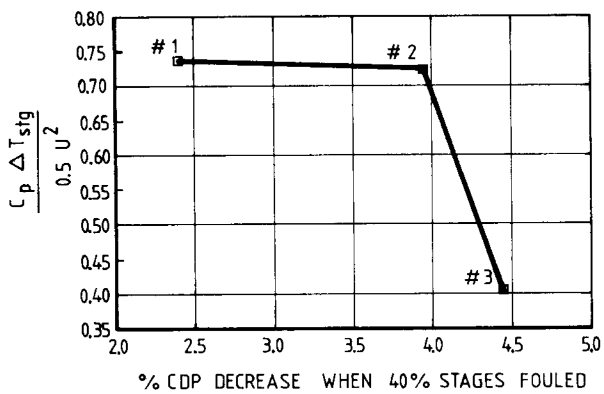


FIGURE 8b: VARIATIONS OF STAGE LOADING COEFFICIENT WITH PERCENTAGE CDP DECREASE FOR THE THREE ENGINES WHEN 40% OF STAGES FOULED.

- $\frac{kW}{\dot{m}U^2}$ This non-dimensional coefficient is the ratio of the specific power output and the square of the tangential speed of the frontal rotor blades at the mean radius. From the comparison of the tabulated results, as shown in Figure 8c, no consistent correlation between this parameter and the fouling sensitivity of the engines emerges.

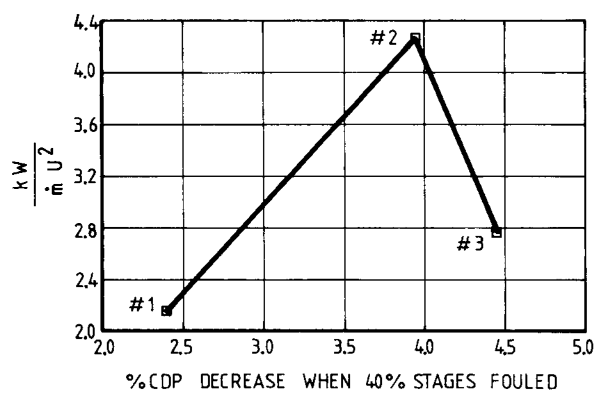


FIGURE 8c: VARIATIONS OF THE POWER COEFFICIENT, $kW/(\dot{m}U^2)$, WITH PERCENTAGE CDP DECREASE FOR THE THREE ENGINES WHEN 40% OF STAGES FOULED.

- $\frac{kW}{\dot{m}C_p\Delta T_{stg}}$ This non-dimensional coefficient is the ratio of the specific power output of an engine and the enthalpy rise for a stage. Figure 8d shows the variation of this parameter versus the % CDP decrease due to fouling 40% of the stages. It appears that this parameter presents the best consistency among the dimensionless coefficients listed for these engines and the corresponding fouling sensitivity. The variation of this parameter with the sensitivity of an engine to fouling is linear and consistent. This parameter incorporates important specification data for an engine, i.e. the engine size manifested by its design power output, its design mass flow rate and its stage loading.

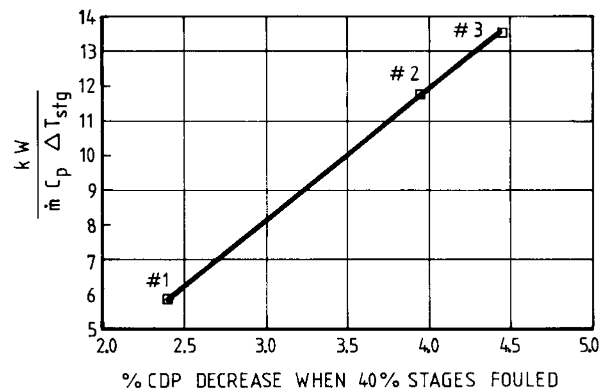


FIGURE 8d: VARIATIONS OF $kW/(\dot{m}C_p\Delta T_{stg})$, WITH PERCENTAGE CDP DECREASE FOR THE THREE ENGINES WHEN 40% OF STAGES FOULED.

As is evident from figures 4 through 7, the relationship between the progress of fouling in compressor stages and performance degradation is non-linear for all of the engine parameters except for CDP. The nearly-linear relationship exhibited in Figure 4 and the ease of CDP measurement are the two factors for selecting CDP as the most suitable parameter in fouling studies, whose extent of degradation would determine the level of fouling progress in an engine. Further examination of the presented results indicates that the dimensionless quantity $kW/(\dot{m}C_p\Delta T_{stg})$ stands out as the most promising parameter for the comparison of different engines on the basis of their sensitivity to performance degradation due to compressor fouling.

It appears that susceptibility to compressor fouling is related to the parameter $kW/(\dot{m}C_p\Delta T_{stg})$, which could be referred to as a *fouling index*. The parameters are readily identifiable for any engine and the values for a wide range of commonly used gas turbines are shown in Table 3. The data in Table 3 were obtained from the 1989 Edition of "Gas Turbine World Performance Specifications," assuming a polytropic efficiency of 88% in all cases to estimate the overall compressor temperature rise; engines with outputs from about 1000-150,000 kW are included, including both older designs and some of the latest designs. It is interesting to note that the larger units tend to have higher values of the fouling index than the smaller units; this may well be a controversial result, but has been arrived at from a systematic investigation without reliance on pre-conceived ideas.

**TABLE 3**

**Design Specifications and Estimated Data**

**for Some Industrial Gas Turbines**

| ENGINE | No. of Stages | Power kW | PR | Mass Flow kg/s | kW/kg/s | $\Delta T_{compressor}$ °C | $\Delta T_{stage}$ °C | $\frac{kW}{\dot{m}C_p\Delta T_{stg}}$ |
|---|---|---|---|---|---|---|---|---|
| Mars | 15 | 8,840 | 16.0 | 37.2 | 237.6 | 421 | 28.1 | 8.43 |
| GT8 | 12 | 47,100 | 16.3 | 177.0 | 266.1 | 425 | 35.4 | 7.52 |
| GT9 | 15 | 34,400 | 9.0 | 161.1 | 213.6 | 300 | 20.0 | 10.68 |
| GT13E | 17 | 147,900 | 13.9 | 491.4 | 301.0 | 389 | 22.9 | 13.14 |
| 501-KB5 | 13 | 3,924 | 15.1 | 15.7 | 249.9 | 408 | 31.4 | 7.96 |
| 570-KC | 13 | 4,731 | 12.1 | 18.6 | 254.3 | 360 | 27.7 | 9.18 |
| 570-KA | 13 | 5,910 | 12.7 | 19.6 | 300.8 | 370 | 28.4 | 10.59 |
| PG 5371 | 17 | 26,300 | 10.2 | 122.5 | 214.7 | 325 | 19.1 | 11.24 |
| PG 6541 | 17 | 38,340 | 11.8 | 136.6 | 280.7 | 354 | 20.8 | 13.50 |
| PG 7191 | 17 | 150,000 | 13.5 | 416.5 | 360.1 | 383 | 22.5 | 16.00 |
| PG 9161(E) | 17 | 116,900 | 12.1 | 403.3 | 289.8 | 360 | 21.2 | 13.67 |
| LM 2500 | 16 | 21,230 | 18.7 | 66.2 | 320.5 | 458 | 28.6 | 11.21 |
| AVON | 17 | 14,240 | 9.2 | 77.6 | 183.5 | 304 | 17.9 | 10.25 |
| TORNADO | 12 | 6,185 | 12.1 | 27.7 | 223.5 | 360 | 30.0 | 7.45 |
| SATURN | 8 | 1,080 | 6.7 | 6.4 | 168.8 | 246 | 30.8 | 5.48 |
| CENTAUR | 11 | 3,130 | 9.4 | 17.6 | 178.3 | 309 | 28.1 | 6.35 |
| CENTAUR H | 11 | 3,880 | 9.3 | 17.5 | 222.1 | 307 | 27.9 | 7.96 |
| V-64 | 16 | 53,300 | 15.8 | 170.1 | 313.3 | 418 | 26.1 | 11.94 |
| V-84 | 17 | 103,200 | 10.7 | 349.4 | 295.4 | 334 | 19.7 | 14.96 |
| CW 251 | 19 | 42,500 | 14.7 | 159.7 | 266.0 | 402 | 21.2 | 12.55 |
| W 501 D5 | 17 | 104,400 | 14.2 | 358.9 | 290.9 | 394 | 23.2 | 12.54 |
| W 501 F | 17 | 145,000 | 14.2 | 413.8 | 350.4 | 394 | 23.2 | 15.10 |
| MW 701 | 17 | 130,550 | 14.0 | 444.6 | 293.6 | 391 | 23.0 | 12.77 |
| CW 352 MAA | 15 | 22,463 | 7.3 | 99.4 | 226.1 | 262 | 17.4 | 12.99 |
| CW 352 MA | 16 | 26,119 | 8.2 | 111.2 | 235.0 | 283 | 17.7 | 13.28 |
| CW 352 MB | 17 | 29,701 | 9.3 | 126.6 | 234.6 | 307 | 18.0 | 13.00 |

## CONCLUSIONS

From this investigation it appears that both the specific output of an engine and its compressor stage loading are important factors in determining its response to fouling effects. This study has been an attempt in the direction of quantifying the severity of performance degradation of an engine due to its compressor fouling; very little information is available on this topic. In the light of what has been presented and expressed, it warrants examining the simulated and field data for a few more engines similar to those in this study. Certainly, the validity of these results can be strengthened and the status of the presented hypothesis will be enhanced if more engines of diverse size are studied in future, using the same methodology for their fouling simulation.

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