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Train Performance - Evaluation and Monitoring by Torque Meter Applications and Process Gas Compressor / Steam Turbine Train Fouling and Washing Mitigation

by

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ABSTRACT

The systematic monitoring and evaluation of turbomachinery is an important diagnostic tool in the execution of a long-term maintenance plan as well as the prevention of unexpected outages due to machinery breakdown. This paper introduces the typical causes of performance deterioration in compressors and steam turbines and the phenomena related to their causes.

In addition, this paper also introduces how to evaluate performance, based on site monitoring techniques are explained by combining typical evaluation results with torque measurement and maximum power limit control. Evaluation of these results can determine if deterioration in performance, caused by a change in the internal flow conditions, has occurred.

Finally, typical results from on-line washing techniques for compressors and steam turbines will be introduced and the power recovery rate after on-line washing, measured by the torque meter, will be discussed.

INTRODUCTION

During this decade, the capacity of ethylene plants has tended to increase continuously as shown in Figure-1. Consequently, flow rates in the high and low-pressure stages of the steam turbine have also increased. To meet this capacity increase, it is necessary to apply larger sub-components including the large flow governing valves, longer blades and longer bearing span rotors. For compression equipment, larger impellers with longer bearing spans are required. Compressor and turbine trains, with capacity in excess of 1.5 MTPY have already been designed manufactured and are successfully operating as shown in the photograph of Figure-1. Therefore, in order to minimize power consumption and downtime losses, it is necessary to conduct machine monitoring for prevention and diagnosis of performance deterioration.

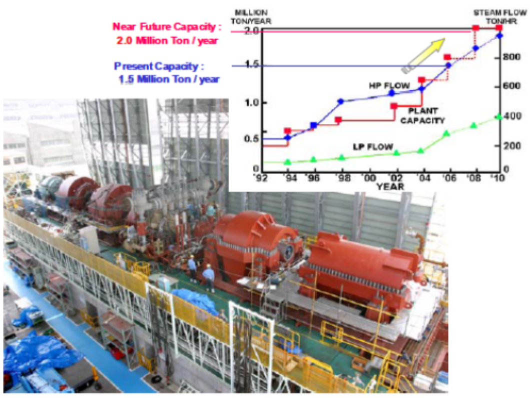


Figure-1 Mega Ethylene Plant Capacity Trend and Actual Mega Train

Performance deterioration for Compressor and Steam turbine

Figure-2 shows the Typical Fouling Condition for a Cracked gas compressor. The cracked gas compressor treats the hydrocarbon-containing gases such as mixtures containing acetylene, and they induce the adherence of fouling on the compressor flow path. Usually, the washing media is injected into the suction pipe line and return bend of each stage to wash the fouling. However, the fouling adhered to the compressor flow path even though washing media is injected.

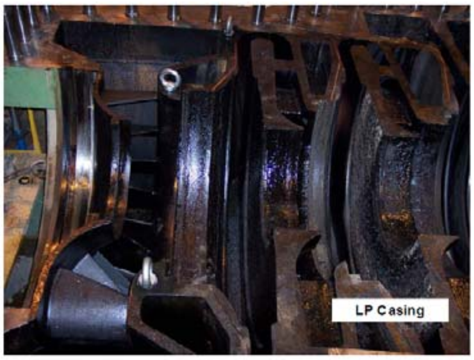


Figure-2 Typical Fouling Condition for Charge Gas Compressors

In Table-1 and Figure-3, damage to the steam turbine occurs during the course of normal operation which also leads to deterioration in performance. Failure modes of mechanical drive steam turbines are classified in terms of main flow path components, in either the rotating components such as rotor, groove, disk, and blade and/or in the stationary parts such as control valves, or bearings. The basic root causes for each damage mode are listed in the figure the physical damage conditions are shown in a steam turbine cross section. The 1st stage nozzle profiles at the trailing edge are eroded by hard solid particles. And, the diaphragm flow paths and blade profiles at the leading edge in the LP high moisture section are eroded by water droplets. In some cases, the welded zone of nozzles has a combination of erosion and corrosion. In the LP section blades, corrosion fatigue failure can occur at the wet and dry enrichment zone under corrosive conditions, if proper water treatment cannot be maintained during either steady or transient conditions, or if corrosive chemical leakage occurs at a heat exchanger.

Table-1 Failure Damage Mode and Root Cause

Components		Damage Mode	Root Causes
Flow Path Parts	Control Valve	Valve Stem Bending Fatigue Failure	High Steam Velocity Fluid Excitation
	Nozzles (HP)	Solid Particle Erosion Fouling	Hard Foreign Materials Inside of Pipes
	Blades (HP)	Solid Particle Erosion Fouling	Hard Foreign Materials Inside Steam Impurity
	Nozzles (LP)	Fouling	Steam Impurity
	Blades (LP)	Drain Attack Erosion Fouling	High Moisture & High Velocity Steam Impurity
Rotating Parts	Blades (HP)	High Cycle Fatigue Failure Centrifugal Force Failure	Excessive Excitation Force Excessive Over Speed
	Blades (LP)	High Cycle Corrosion Fatigue Centrifugal Force Failure	Improper Water Treatment Excessive Over Speed
	Rotor	Rubbing & Shaft Bow & High Vibration Disk SCC	Improper Start Up Drain Intake, Steam Impurity
	Thrust Collar Journal Shaft	Rubbing & Scratch/Wear	Excessive Thrust Force Improper Oil Supply (UPS)
	Bearings	Rubbing & Melting	Excessive Thrust Force Improper Oil Supply (UPS)
Stationary Parts	Casing	Creep Deformation/ Creep Rapture, Erosion/Corrosion	Operation Over Allowance Excessive Drain / Galvanic
	Diaphragm	Diaphragm Bending Deformation Erosion & Corrosion	Operation Over Allowance Excessive Drain / Galvanic
	Seals	Rubbing & Erosion	Improper Start Up Excessive Drain

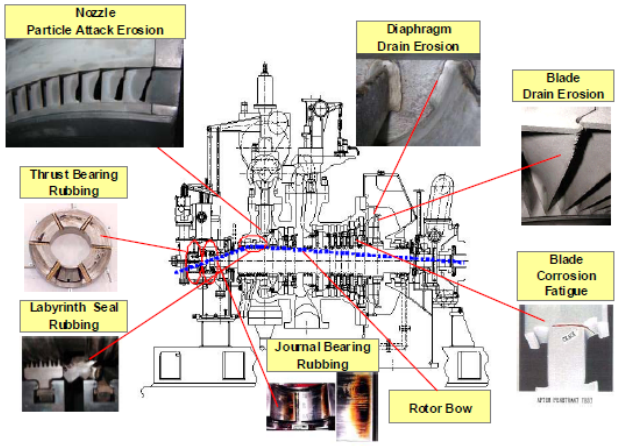


Figure-3 Failure Damage Map

In Figure-4, the main factor causing deterioration of the steam turbine performance is fouling. The steam flow paths tend to accumulate deposits on the profile of the nozzles and blades and cause a reduction in the area profile, increase surface roughness and change the velocity ratio related to the stage efficiency.

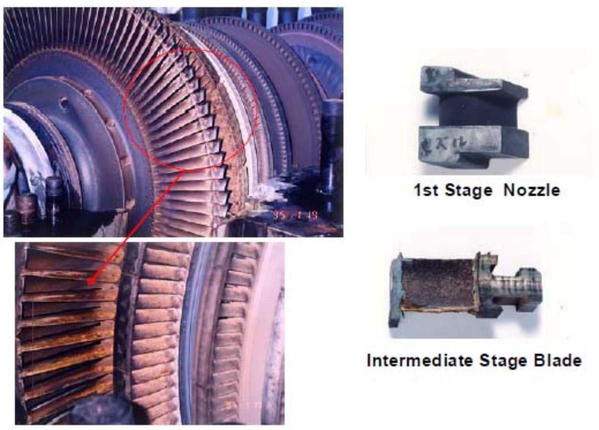


Figure-4 Fouling Condition After 7 year Continuous Operation

Monitoring and Evaluation Performance for Compressor and Steam turbine

In Figure-5 and Figure-6, the actual torque measurement between driver and compressors is very useful to evaluate the total performance of compressor and steam turbine train. In some case, torque meters are installed between each compressor individually. Steam turbine performance is calculated according to HP and LP section operation and steam conditions, and the steam consumption is evaluated by comparing measured torque and power. Performance deterioration due to changes in the internal flow conditions can also be evaluated and analyzed based on the comparison of monitoring data and design data. Operational performance of the compressor is calculated using actual process data such as molecular weight for each section in comparing each stage performance curve using basic equations. Comparison of the total calculated power for the steam turbine/compressor and the measured power obtained from torque monitoring should correlate within a small error tolerance. If the calculated versus actual power comparison falls outside of acceptable tolerances, further investigation is required.

In Table 2, the typical evaluation sheet for process gas compressor of large ethylene plant is shown. The efficiency, head and power for each section are calculated and compared with the actual numbers based on the measured data. In this case, the close correlation for each performance factor is confirmed and the calculated power for steam turbine and compressors is approximately same as the measured power, within the instrument accuracy. Proper measurement of process data, such as molecular weight, is very important for performance evaluation. Piping design and location of instruments have to be checked and confirmed between End Users and OEM's.

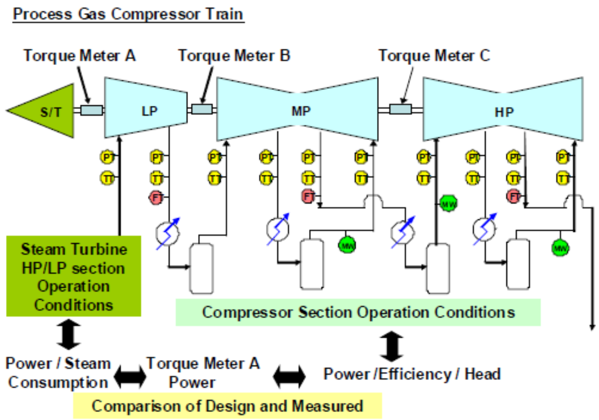


Figure-5 Toque Meter Arrangement and Evaluation Base Data

1) Polytropic head (Wp)

$$Wp = \left(\frac{n}{n-1}\right) f \cdot P_i \cdot V_i \left(\left(\frac{P_d}{P_i}\right)^{\frac{n-1}{n}} - 1\right) \times 10^4$$

$$n = \frac{\ln(P_d/P_i)}{\ln(V_i/V_d)}$$

$$f = \frac{h'_d - h_i}{\left(\frac{ns}{ns-1}\right)(P_d \cdot V'_d - P_i V_i)} \times J \times 10^{-4}$$

$$n_s = \frac{\ln(P_d/P_i)}{\ln(V_i/V'_d)}$$

2) Polytropic efficiency (η_p)

$$\eta_p = \frac{Wp}{h_d - h_i}$$

3) Brake horsepower (kW)

$$P_{sh} = \frac{W \cdot (h_d - h_i)}{860} + \text{Mech. losses}$$

W: Weight flow (kg/h)

Figure-6 Compressor Performance Evaluation Equations

Table-2 Typical Compressor Performance Evaluation Results

		Sect-1	Sect-2	Sect-3	Sect-4	Sect-5
Efficiency [%]	Measured at MCO Shop Test	η 1shp	η 2shp	η 3shp	η 4shp	η 5shp
	Measured at site on Oct, 2010	η 1site	η 2site	η 3site	η 4site	η 5site
	Difference by	-2.8	-2.6	-3.5	0.0	+0.9
Head [m]	Measured at MCO Shop Test	7600	7600	7600	6400	4600
	Measured at site on Oct, 2010	7570	7540	7570	6140	4770
	Difference	99.6%	99.6%	99.5%	96%	104%
Power [kW]		7300	7700	7700	5200	4000

Total: 32,000 kW

<Reference>
Steam Turbine power from the site data : 32,900 kW
Torque meter measurement power : 32,000 kW

As shown in Figure-7 and Figure-8, the steam turbine performance is systematically evaluated according to the steam condition of HP and LP section and operation conditions. The steam flow for each section can be estimated from valve lift and the exit pressure of 1st stage for each section. Accuracy of the measured steam flow should be checked in comparison to the expected flow, based on valve lift and the exit pressure of 1st stage. The steam turbine performance is compared using expected performance curves by applying the correction coefficients of steam temperature, pressure and speed in case of actual operating conditions.

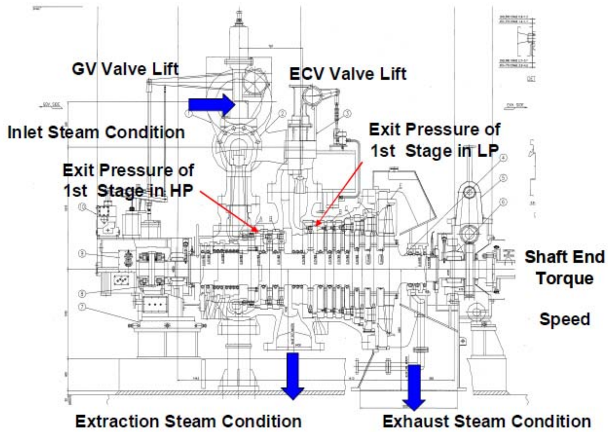


Figure-7 Steam Turbine Evaluation Basic Data

a) G_{AH} shall be the measured steam flow

b) G_{OH} is defined as the steam flow which is expected to be measured if the steam operates under original running conditions (i.e. original speed, original design inlet pressure, etc.)

c) Φ_{nH}, Φ_{PO}, Φ_{TO}, Φ_{P2}, Φ_{VG} are correction factors according to MHI-diagrams and must be determined

d) Calculate $G_{OH'} = G_{AH} \times (\Phi_{nH} \times \Phi_{PO} \times \Phi_{TO} \times \Phi_{P2} \times \Phi_{VG})$ (6)

e) The value $G_{OH'}$ is the expression for the expected steam flow figure and is determined out of the vendor's expected steam turbine performance curve (issued for original operating conditions).

f) Ultimately the ratio $Y = (G_{OH} / G_{OH'})$ (7)

is calculated, which is the ratio between actual steam flow and expected steam flow for specified steam conditions.

Figure-8 Steam Turbine Evaluation Procedure

A typical evaluation sheet and performance curve is shown in Figure-9.

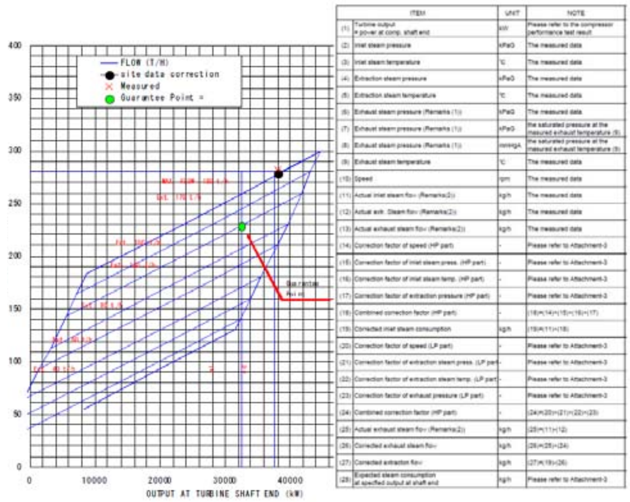


Figure-9 Typical Steam Turbine Performance Evaluation Result

Fouling Phenomenon for Compressor and Steam turbine

In Figure-10, the steam flow is regulated by the governing valve to control speed, power and extraction pressure depending on the type of turbine. The steam flow is known by valve lift and linkage system including control signal to the E/H actuator. In addition, when the actual measurement of the valve lift and steam flow is compared to the design curve for valve lift and steam flow, it is possible to detect fouling conditions or mechanical problems with the valve linkage system. This typical relationship of valve lift and actual measured steam flow is also a possible indicator that the 3rd valve could have a flow area reduction due to deposits on the valve seat.

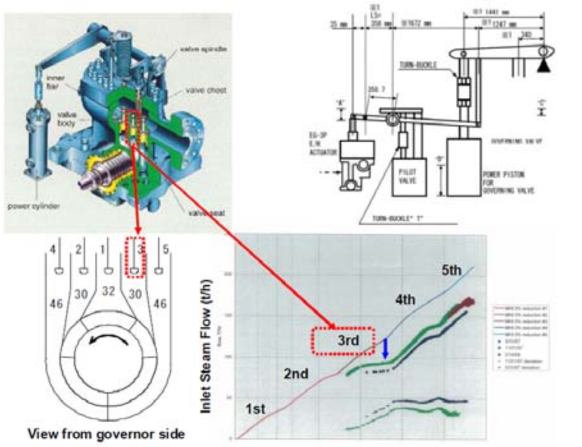


Figure-10 Governing Valve Linkage System and Relationship of Valve Lift and Comparison of Steam Flow

In Figure-11, another fouling indicator can be found in the comparison of the measured flow in the HP section and lift valve, the exit pressure of 1st stage and expected efficiency based on the correlation of inlet and exit steam condition changes over several months of operation. The exit pressure of 1st stage for each section gradually increases due to nozzle area reduction caused by fouling and efficiency decreases corresponding to the change in the internal flow condition.

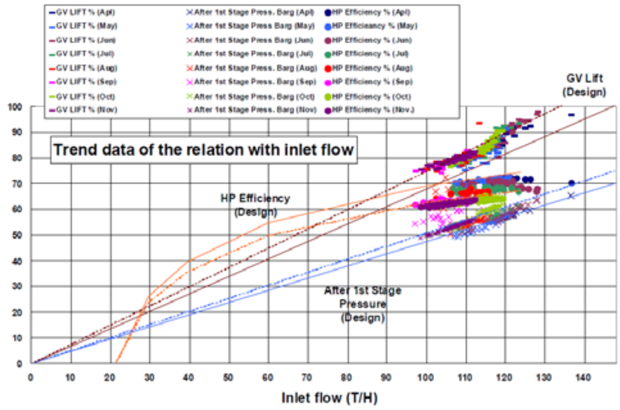


Figure-11 Relationship of Steam Flow and Governing Valve Lift, Exit Pressure of HP section, HP Efficiency in Comparison of Design and Measured

In Future-12, the nozzle area reduction down stream causes an increase in the exit pressure of 1st stage for HP and LP section and this pressure rise must not increase over allowable limit of the pressure vessel design. In some case, the inlet steam flow has to be decreased in order to decrease the exit pressure of 1st stage below that limit. This causes power reduction and production limitations in the plant operation.

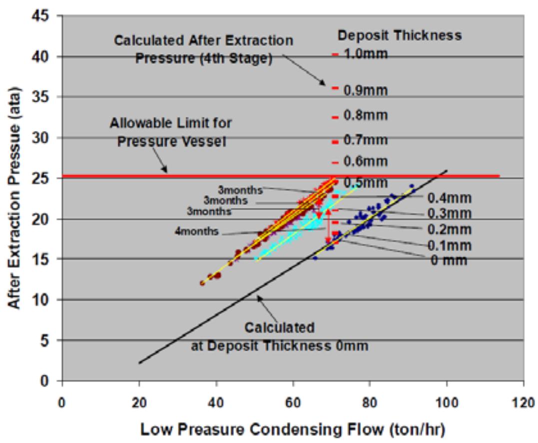


Figure-12 Operation Condition and Deposits Profile Calculation Results

In Figure-13, a more effective wash procedure is necessary in considering internal flow during wash oil injection. The compressor efficiency is decreased remarkably if oil wash is improperly applied in a very short duration.

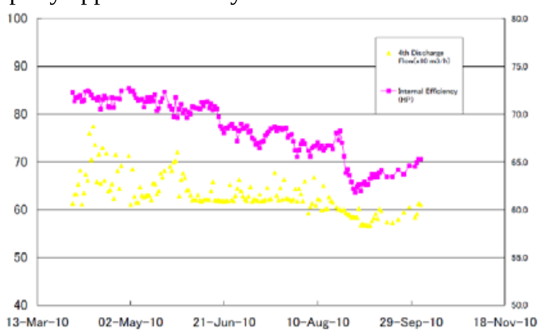


Figure-13 Typical Compressor Efficiency Deterioration for Short Term Operation

Steam turbine Online washing

In Figure-14 to Figure-16A, the steam turbine online wash procedure had already been established and applied in more than 30 actual turbine applications. The water is directly injected inside of the turbine valve chest and typically it is not necessary to reduce the power and speed. During online washing, the LP section condition becomes wet and washable chemicals are washed out as indicated by a high spike of conductivity. After several hours of washing, the extraction pressure decreases to a level much lower than the allowable pressure limit and the turbine performance increases. Before and after washing, the operating conditions, including power based on measured torque are compared and typical results are shown. A large increase in the power recovery ratio is confirmed.

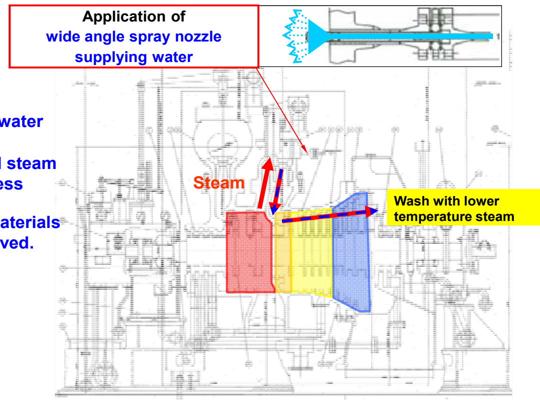


Figure-14 Typical On-Line Washing Procedure

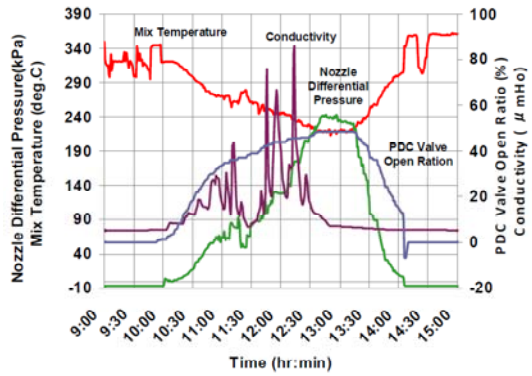


Figure-15 Typical Operation Data during On-Line Washing

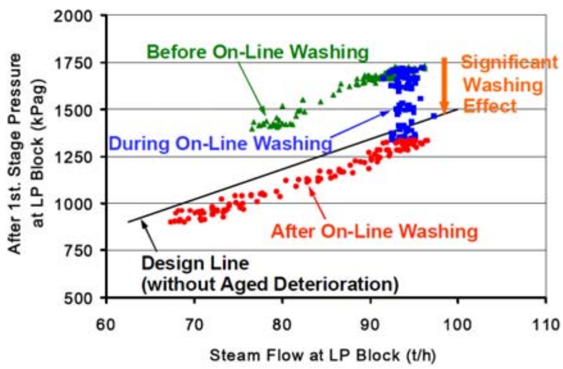


Figure-16 Performance Recovery and Operation Window Improvement after Online Wash

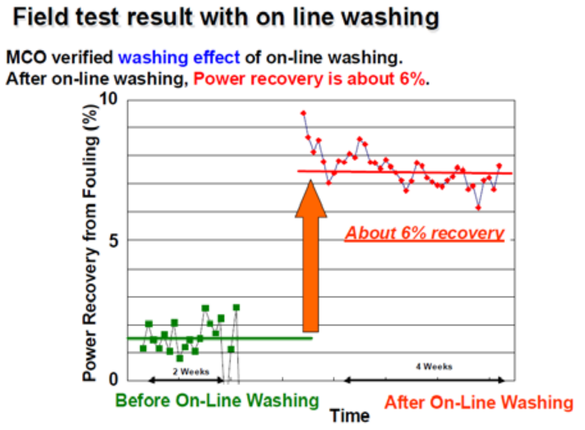
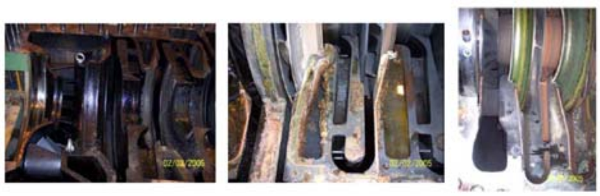


Figure-16A Power Recovery after Online Wash

Optimization of Oil Injection Configuration for Compressor

Figure 17 shows the Typical Fouling Condition for Cracked gas compressor. The cracked gas compressor treats the hydrocarbon-containing gases such as mixtures containing acetylene, and they induce the adherence of fouling on the compressor flow pass. Usually, the washing media is injected into the suction pipe line and return bend of each stage to wash the fouling. However, as shown in Figure.1, the fouling adhered on the compressor flow pass even if washing media is injected.

Since, in order to improve the wash efficiency, the authors carried out the simulation of oil injection by application of a unique CFD modeling and analysis technique and testing of the injection flow pattern on a rotating impeller.



a) Low Pressure Casing b) Middle Pressure Casing c) High Pressure Casing

Figure 17: Typical Fouling Condition for Crack Gas Compressor

Figure.18 shows the CFD analytical model in case washing media is injected from suction piping. The compressor internal from suction piping to the inlet of 1st impeller was modeled and simulated by particle / gas continuous two phase flow model analysis to understand the distribution of washing media around the compressor suction line. The various injection condition such a location of injection point and particle size of washing media were simulated. The result of CFD simulation under typical injection condition is shown in Figure.19. Only small amount of droplet is entered into the 1st impeller, and almost all of oil seems to be drained from suction piping. Figure.20 shows a part of CFD simulation result which varied the injection location, direction and particle size. It is hardly affected by initial particle size, injection location and direction. This is because the droplet may be shrunk by inertia of main gas flow, and wash media does not expand remarkably.

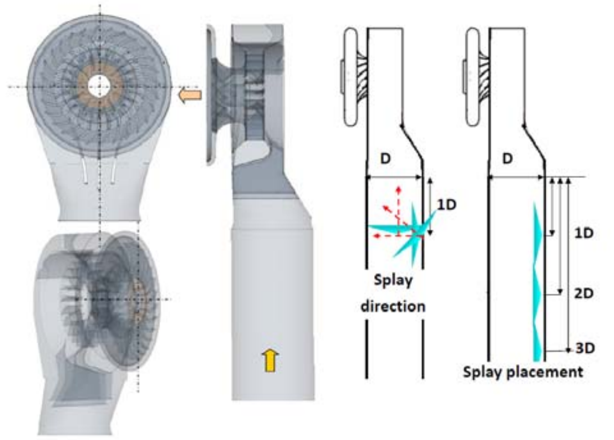


Figure 18: Injection from Suction Piping CFD model

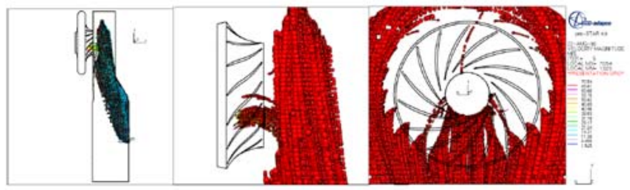


Figure 19: Result of CFD Simulation (Typical condition)

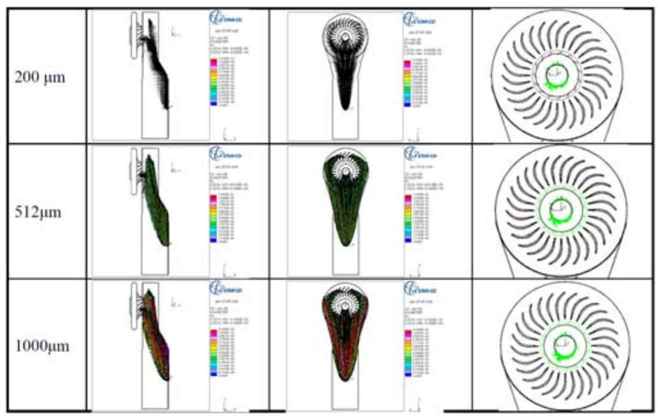


Figure 20: Result of CFD Simulation (different initial particle size case)

In addition to particle / gas two phase flow model analysis of suction line, water / gas continuous two phase flow model analysis is being done to understand the total flow path of stator and impeller. The injection nozzle was located in the return bend as typical condition, and wash spread area Index (WSA) was confirmed as the new concept of quantitative evaluation. Figure.21 shows the result of CFD simulation under typical injection condition. The colored area shows the Volume of Fraction of Water (VOF) in gas. As a result of CFD analysis, the water still does not expand to all surfaces of flow pass same as suction piping.

From the result of previous section, it was identified that the typical injection condition is not the optimum configuration, and it should be resolved practically. Hence, as a next step, in order to improve the wash efficiency, various injection conditions such as location, velocity and direction were simulated and studied based on particle / gas two phase flow model analysis as shown in Figure.22 which is a part of the investigation result of nozzle arrangement by CFD analysis.

From these simulations, the optimum oil injection condition was identified and confirmed. As a next step, to confirm the properness of CFD simulation, the test set up was designed to evaluate mix flow, water / oil distribution and performance change and detail test procedure is considered to simulate the actual flow condition as shown in Figure.23.

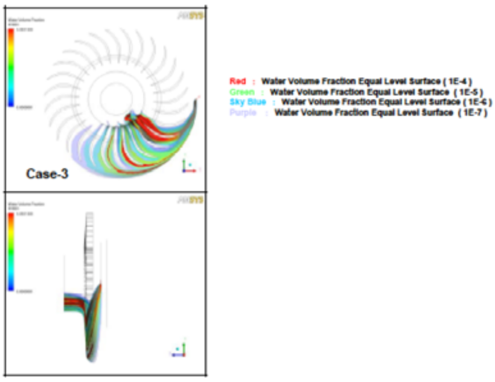


Figure.21: Result of CFD simulation (Typical condition)

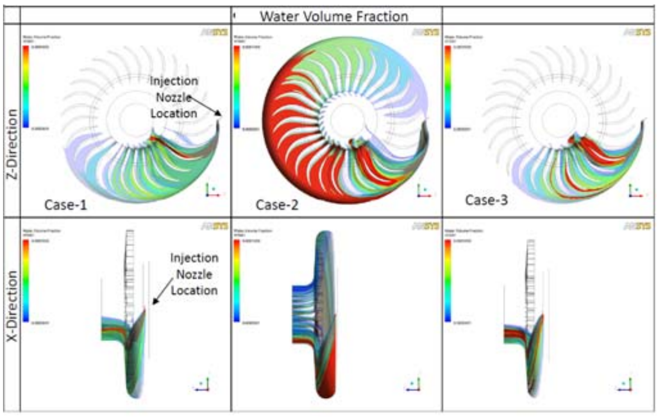


Figure 22: Investigation result of nozzle arrangement

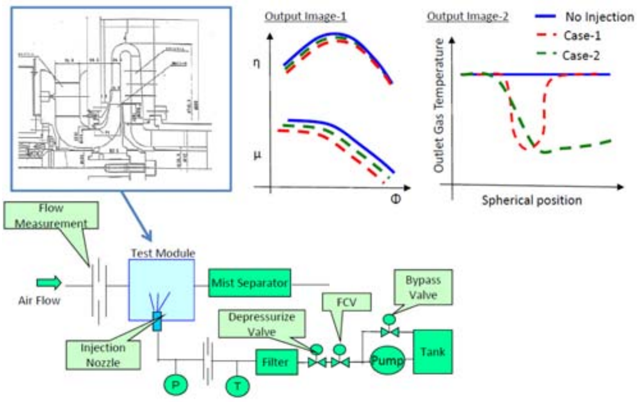


Figure 23: Test set up for water injection simulation experiment

In this test setup, in order to confirm the dispersion area of water droplet to the flow path, the temperature sensors were installed on the return channels of the diaphragm as shown on Fig.24. These sensors detect the temperature decrement when the water droplets reach to the sensor position.

Fig.25 shows the test result. The vertical axis shows the temperature decrement of each sensors and horizontal axis shows the circumferential direction. From this result, it was confirmed that the result of CFD analysis well agreed with this test result and the water droplet dispersion area depends on the injection condition.

This means that the optimum injection condition of the water/oil can be designed based on the CFD analysis results.

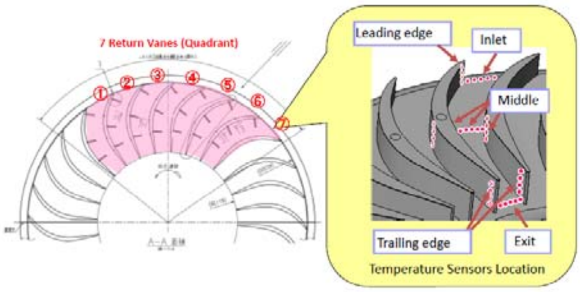


Fig.24 Temperature Sensors location at Return Vanes

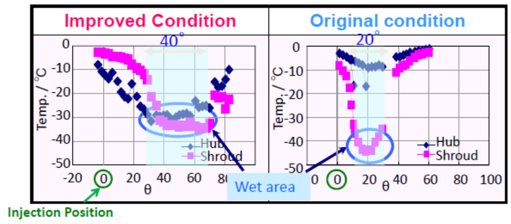


Fig.25 Test Results (Temperature decrement at each sensors)

From the result of all study, the optimum oil injection procedure is decided, and it will apply on the actual machine to evaluate the oil wash effectiveness. The results of this study will show that introducing the wash oil injection flow directly into the compressor casing is much more effective than injection into the suction pipe and the ratio of flow into the various injection points is critical to successfully mitigation.

CONCLUSIONS

The systematic monitoring and evaluation of steam turbines and compressors has been introduced to explain the importance of using the monitoring data for machine diagnosis and to develop a long-term maintenance plan for the prevention of unexpected outages by showing how to calculate the performance according to the measured data by using torque meter.

Typical performance evaluation results, actual torque measurement between driver and compressors, the examples of diagnosis for internal flow condition and performance change are explained in addition to online wash techniques in order to prevent performance deterioration.

From the result of all study, the optimum oil injection procedure is decided, and it will apply on the actual machine to evaluate the oil wash effectiveness. The results of this study will show that introducing the wash oil injection flow directly into the compressor casing is much more effective than injection into the suction pipe and the ratio of flow into the various injection points is critical to successfully mitigation.

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