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Technical Challenges for Compressors and Steam Turbines for Efficient and Sustainable Operation in Mega Ethylene Plants

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ABSTRACT

Changing markets, industry demands for increased efficiency and long term operation, and availability of new technology have all contributed in development of more efficient and reliable turbomachines. The authors present ethylene plants trends, demonstrate the challenges faced by the turbomachinery equipment manufacturers and highlight various advancements in the turbomachinery technology. History maps are introduced for design advancements, verification tests, and application results in terms of transient fluid dynamics, thermodynamics, rotor dynamics, and blade vibration strength evaluation. In addition, after recognizing the need for long term operation and related typical damage and deterioration modes, the authors explain various practical technologies (such as effective on-line washing, flow path surface treatment, combination of anti-corrosion and erosion prevention, stage performance enhancement by partial component replacement, NDE techniques for both compressors and steam turbines, and a unique casing replacement technique on the same footprint for increasing capacity) used to provide more efficient and reliable machines.

INTRODUCTION

During this decade, the capacity of ethylene plants has tended to increase continuously. Consequently, flow rates in the high- and low-pressure stages of the steam turbine driver have also increased. To meet this capacity increase, it has been necessary to apply larger sub-components including 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 capacities in excess of 1.5 MTPY have already been designed, manufactured, and are successfully operating. Therefore, in order to minimize power consumption and downtime losses, it is necessary to conduct machine monitoring for prevention and diagnosis of performance deterioration.

Turbomachinery equipment manufacturers have been moving forward to apply state of the art technologies for critical components in order to provide more efficient and reliable machines corresponding to the high power requirements for ethylene services. The authors introduce the technology development history map for typical designs, verification tests, and application results in terms of transient fluid dynamics, thermodynamics, rotor dynamics, and blade vibration strength evaluation. In addition, after commissioning turbomachinery performance and reliability tends to deteriorate during long-term operation. The typical damage and deterioration map for machinery is introduced, and the authors explain practical technologies that are applied, such as flow path surface treatment, effective on-line washing, combination of anti-corrosion, erosion prevention and stage performance enhancement by partial component replacement, NDE techniques for both compressors and steam turbines, and a unique casing replacement technique on the same footprint for increasing capacity.

Mega Ethylene Plant Trend

In the past 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 driver have also increased. To meet this capacity increase, it is necessary to apply larger sub-components including governing valves with larger flow capacities, longer blades, and longer bearing span rotors. For compression equipment, larger impellers with longer bearing spans are required. Compressor and turbine trains with capacities in excess of 1.5 MTPY have already been designed, manufactured, and are successfully operating.

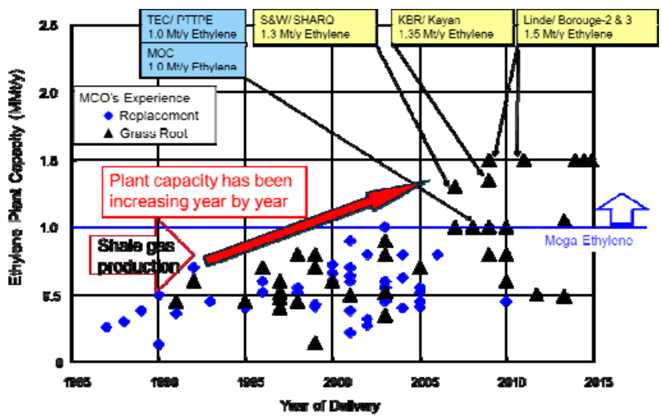

Figure.1 Ethylene Plant Capacity Increase Supported by Upgraded Equipment and Technology

Figure 2 shows the relationship between MCR (max. continuous rotating speed) and steam turbine maximum power. This relation is basically exponential. The required power for a Mega ethylene plant charge gas compressor (CGC) MP/LP/HP driver is 95MW, and can potentially be as high as 120MW. An optimized MCR for steam turbine and compressor performance and rotor dynamics is around 4,000 rpm.

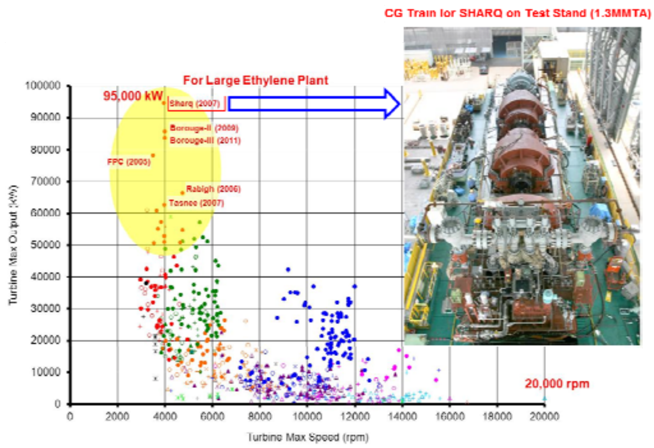

Figure.2 Recent Steam Turbine Range for Mega Crackers - Simple Single Train Solution

In Figure 3, below, the application range of each service and its corresponding Mitsubishi Advanced Compressor (MAC) frame size is shown on a compressor suction flow and discharge pressure map. Large frames (9H and larger) are applied for mega ethylene plants. These same sizes are used in other applications, such as LNG main refrigerant compressors.

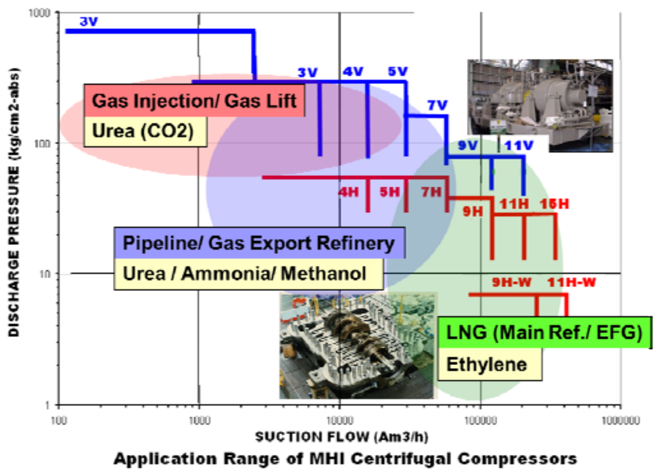

Figure.3 Application Range for Mitsubishi Advanced Compressor frame sizes

Figure.4 Recent Steam Turbine Range for Mega Crackers - Simple Single Train Solution

Every compressor's performance, including rotor dynamics, is measured and verified as a part of the manufacturing process. In addition, the steam turbine and compressors can be coupled and the complete train can be tested to confirm combined mechanical performance, such as vibration and bearing condition, on a large test stand prior to shipment. Figure 4 shows a 1.5MMTA class mega ethylene CGC train in the same cooperation condition as that at the site.

Challenging Technologies for Critical Components

As plant capacities become larger and process flows increase the corresponding compressor impeller and casing size must become larger. The required power also increases. Controlled manufacturing processes for 1 piece or 2 piece impellers are also important to meet the design and quality requirements for these large components. Figure.5 shows key technologies such as 3D impellers and FFD (Flexible Frame Design) which are utilized in compressor design. To increase the rotor dynamic stability a swirl canceling labyrinth seal and a special damper are applied depending on rotor rigidity.

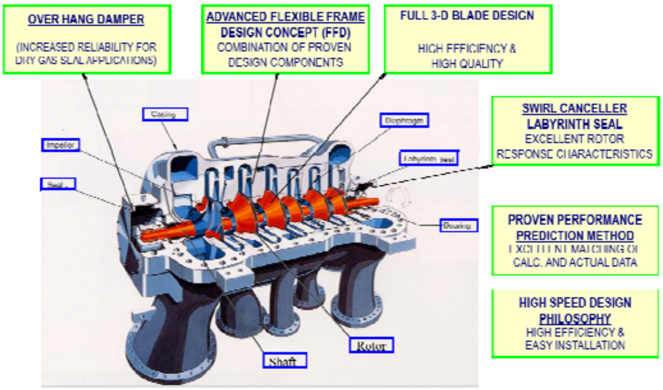

Figure.5 Compressor Key Technologies

To produce the increase power, the required inlet steam flow (HP section) and exhaust steam flow (LP section) become larger and a larger component size for the flow path is necessary. Figure.6 shows key technologies to meet the requirement for large plant and machines which are used in steam turbine design. A reaction optimized impulse blade is applied for the inlet and middle stage flow paths in order to increase efficiency and minimize the thrust load due to differential pressure across each blade inlet and exit.

The last 3 stages in the LP section utilize a z-lock integral shroud and high rigidity blade profiles are applied to reduce resonance vibration stresses in the blade and to increase stage efficiencies.

A critical design concept in modern rotor design is to increase rigidity as much as possible by optimizing blade stage number, bearing span, and HP/LP section efficiency at the required operating speed. A high rigidity rotor can reduce vibration response due to unbalance and increase vibration stability.

Recently, operating intervals of over 8 to 10 years have been required to meet customer demand. For this requirement, several new technologies have been developed and applied to existing machines.

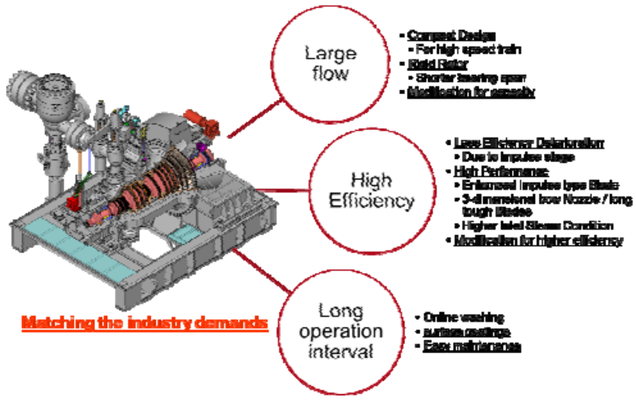

Figure.6 Steam Turbine Key Technologies

Figure 7 shows the technical matrix for present and future steam turbine designs. For long term operation and increased reliability, several surface treatments are applied to protect the flow path against environmental damage from solid particles or wetness in the LP stages.

During operation, the flow path tends to develop deposits caused by degraded steam quality and insufficient control. These deposits cause performance deterioration and potential reliability issues. On-line wash technology can clean and wash the flow path in full power operation in less than one hour.

To simplify the governing valve control system, the oil free governing valve was developed.

Application of Key Technologies (Today) for steam turbines results in higher performance, strength, and reliability as shown below. MCO applies new technologies for the continuous improvement of their designs.

	Performance	Strength	Reliability
Turbine	High speed application	-	On-line washing
Casing	Application for super high press / temp steam		No leakage
TTV	Quick shut down		No stick
Rotor	Higher rigidity	Solid disk	No crack
Blades	Hybrid improved impulse blading	ISB	Ceramic coating PTA welding Hybrid coating of fluorine resin & ceramics
Diaphragms / Nozzles		No bending	Boronize coating Hybrid coating of fluorine resin & ceramics
Governing system	Higher controllability	-	Oil Free governing valve control system

Figure.7 Technical Matrix for Present and Future Steam Turbine Designs

Technological Development History Map

Figure 8 shows the latest technologies for steam turbine development history from 1980 to the present. Flow path and profile design concepts and procedures have been developing from 2D to 3D by improvement of computational technologies and computer calculation speed/data capacity. Due to larger machine requirements, a longer blade is necessary for the last stage. Integrally shrouded blades (ISB) for the high pressure section were developed. This ISB grouping concept for impulse type blading is different from conventional twisted designs and creates one robust blade grouping. TTV and governing valves were developed for larger sizes and to meet higher inlet steam temperatures. Direct lubricant tilting pads for journal and thrust bearings instead of flood types reduce mechanical losses and reduce pad metal temperatures in high velocity situations. Rotor repair technologies have been developed for planned and emergency outages to minimize the risk of no spare rotor. Several types of seals have been developed to reduce leakage and increase efficiency. For long term operation, surface treatment technologies are applied. NDE techniques were also developed to inspect internal damage on rotor disk grooves. One of the latest technologies is PAUT, phased array ultra-sonic testing. Based on measured crack length, location, and orientation the allowable residual operational life can be estimated for turnaround and asset planning.

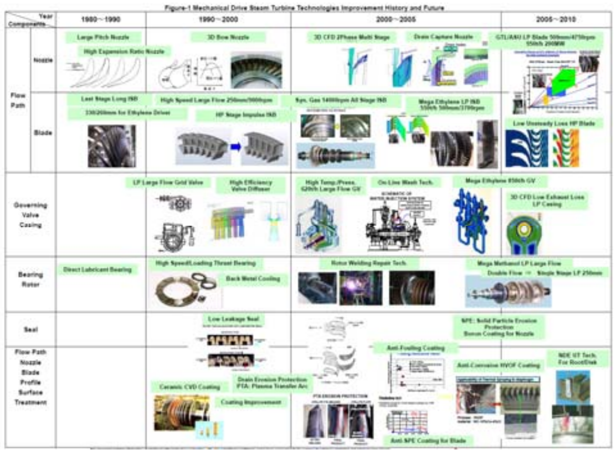

Figure.8 Latest Technologies for Steam Turbine and History

Figure 9 shows a centrifugal compressor research and development map. Similar to the steam turbine, flow path and profile design concepts and procedures have been developing from 2D to 3D as per the improvement of computational technology. Compressor rotor stability is critical in high pressure applications in large bearing span machines. To increase rotor rigidity and maintain efficiency, the high boss ratio impeller was developed. To increase the damping of the rotor/bearing/seal vibration system and rotor stability, a mechanism for gas excitation force reduction and a high damping seal were developed. In terms of manufacturing processes, new methods were developed such as 1 piece impellers by 5-axis machining for large high flow coefficient impellers and EDM/ECM for small throat, low flow coefficient designs.

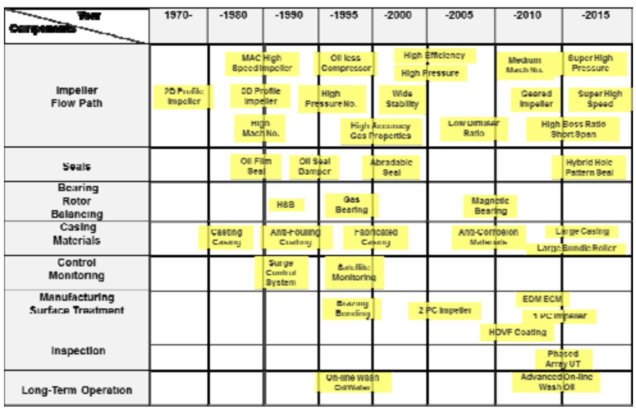

Figure.9 Centrifugal Compressor Research and Development Map

Design, Verification Tests, and Application in terms of Transient Fluid Dynamics, Thermodynamics, Rotor Dynamics and Blade Vibration Strength Evaluation

Machine components are designed to meet the required specification which is decided by process data. In some cases, the compressor impellers and steam turbine blades need to be newly developed. Here, several designs and performance verification case studies are explained to show the typical development process.

Compressor Development
Figure 10 shows the Compressor Flexible Frame Design (FFD) Concept. The fixed frame consists of flow path suction parts, discharge parts and bearings. These are high reliability and well-proven components. Flow path aerodynamics design parts are flexible to achieve optimum performance with high efficiency.

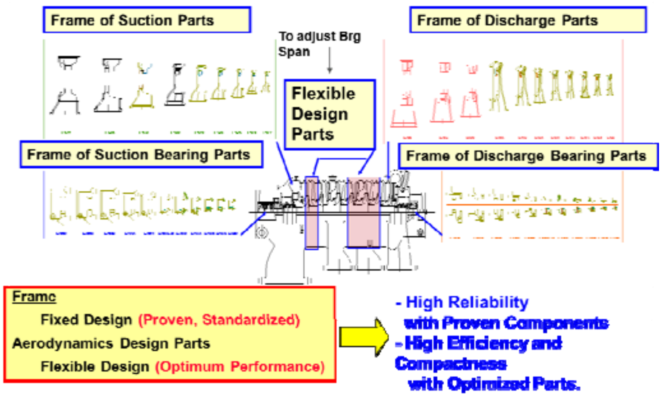

Figure.10 Compressor Flexible Frame Design (FFD) Concept

Figure 11 shows the verification process for a new impeller for a compressor. As the first step, the impeller profile, disk and cover structure dimensions are decided according to the results of performance predictions by CFD and static/dynamic strength evaluation by FEA. The actual impeller size is scaled to the required diameter using the similarity rule of turbomachinery flow paths (velocity triangles) and strength (vibration, stress, power). Next, a standard size test impeller is manufactured according to this similarity rule. In the third step, a single stage impeller performance test is conducted to characterize the performance and compare it to calculated results as shown in Figure 12. This data is added to the impeller performance database and is used for actual application design in FFD process.

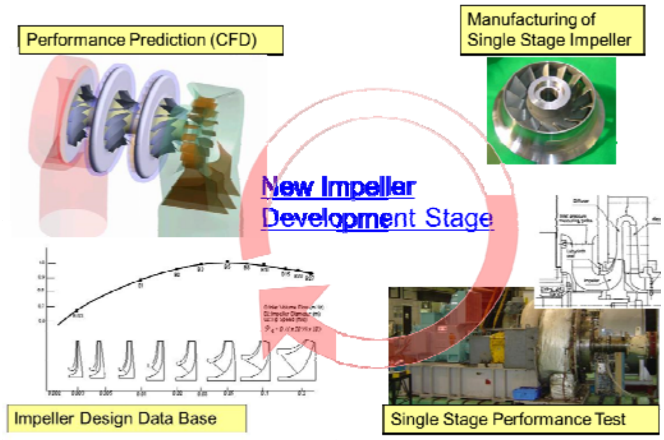

Figure.11 Verification Process for New Impeller

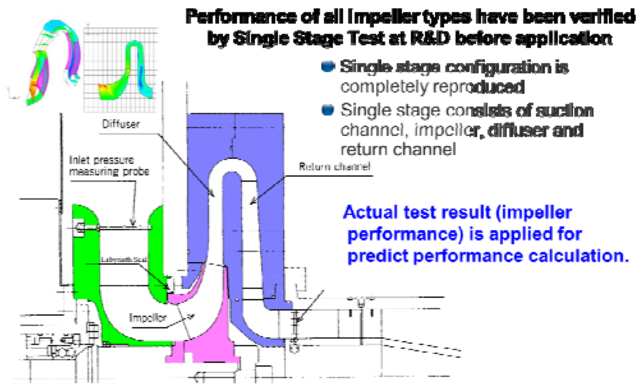

Figure.12 Development / Improvement of Impeller

Case Study: The High Boss Ratio Impeller
The latest impeller design concept combines a high boss ratio and short span as shown in Figure 13. In order to increase the rotating speed for compact designs the shaft rigidity needs to be larger for low vibration response and high stability. In addition, the bearing span needs to be shorter while the number of stages remains the same. In order to realize both, a new impeller standard series was developed and is in service in operational machines.

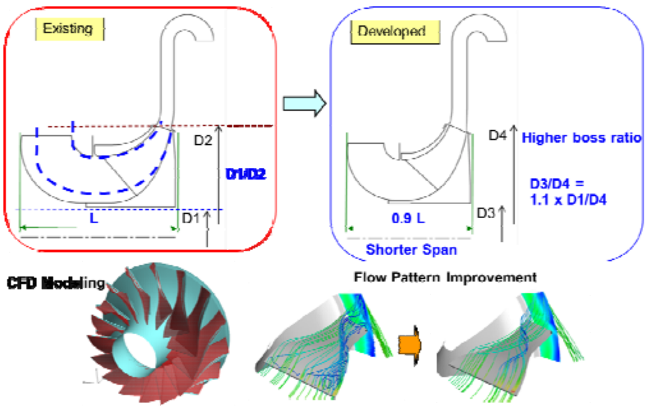

Figure.13 High Boss Ratio/Short Span Impeller Design

Figure 14 shows the profile modification for high boss ratio and short span impeller design and corresponding performance test results. Two concepts are applied: a reduced blade count for reduction of friction losses and a modified blade shape for optimum flow line. This design has a confirmed polytropic efficiency gain over a conventional impeller.

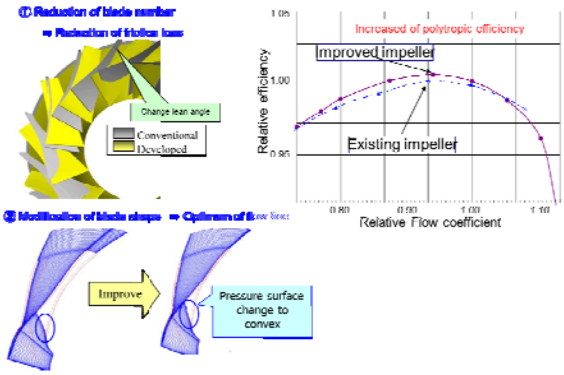

Figure.14 Profile Modification for High Boss Ratio and Short Span Impeller Design and performance test results

The advantage of the high boss ratio/short span impeller for an actual machine application is shown in Figure 15. A Mega ethylene plant PRC train is selected as a typical case. The number of compressor casings can be reduced from two to one; in addition, the weight and length can be reduced by roughly 20% and 50% respectively. This represents significant cost savings through reduced raw material requirements.

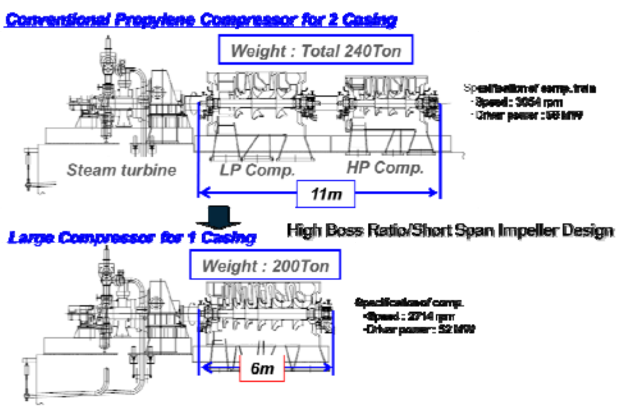

Figure.15 Advantage of High Boss Ratio Impeller

Steam Turbine Development

For steam turbines, Figure16 shows the high performance of the hybrid impulse blade design used to improve the efficiency of the HP section of the turbine. It is possible to minimize the friction losses of the steam path by conducting blade and nozzle CFD analyses and the actual performance is verified by a test machine and actual machine at the shop.

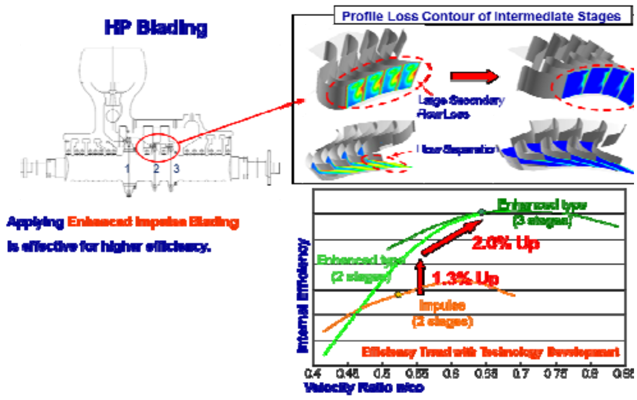

Figure.16 High Performance of Hybrid Impulse Blade Design

Figure 17 shows the high performance of the LP section long blades. This example details improvements to efficiency by optimizing the steam flow using CFD to prevent secondary flows between blades and the high Mach flow area. It was possible to improve the LP section efficiency by 8% in this case study.

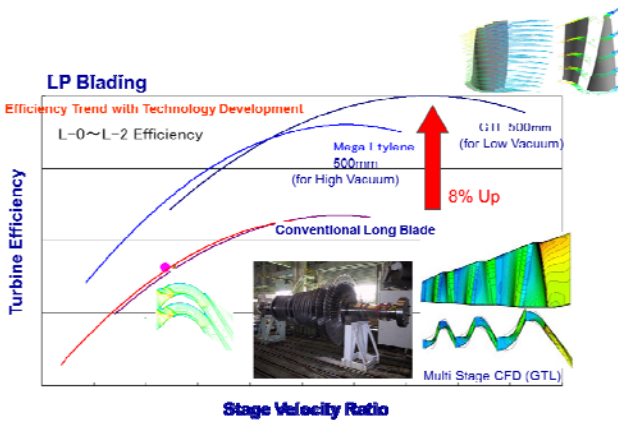

Figure.17 High Performance of LP section Long Blade

Figure18 shows the application for higher inlet steam conditions to achieve higher steam flow utilization.

Increasing steam and energy costs are resulting in the demand for improved steam consumption. Higher pressure and temperature steam reduces the steam consumption of the turbine. High pressure/temperature steam generally has higher enthalpy and a higher theoretical maximum efficiency. Figure 18 shows a relevant case study from a recent project. One challenge is to apply advanced materials for the high pressure/temperature steam and turbine design structure while maintaining the same dimensions as the original. The stress analysis is done to evaluate thermal stress under steady and transient conditions comparing design criteria including creep strength.

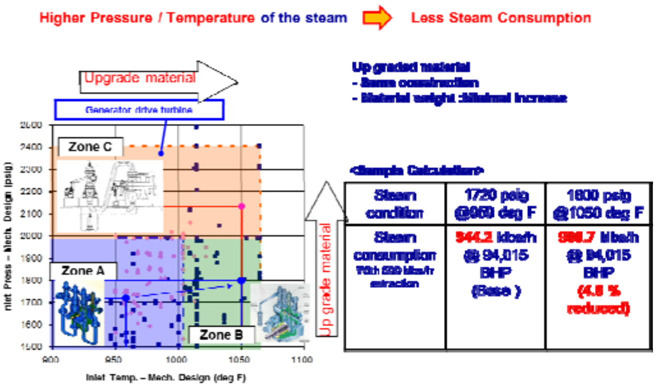

Figure.18 Application for Higher Inlet Steam Condition

Finally, Figure 19 shows the reliable operation for rotor dynamics. API standards are widely used in the petrochemical industry. Typically steam turbine designs must comply with this API standard, which recommends a maximum shaft vibration of 25.4 micron peak-to-peak during shop mechanical running tests. Figure 19 shows historical vibration records during mechanical running tests in the shop demonstrating the API criterion.

Mechanical drive steam turbines are designed based on impulse type blades. Typically an impulse turbine has a shorter bearing span than a reaction type blade design of similar power. Shorter bearing spans can increase the rotor rigidity and this higher rigidity contributes to low vibration levels during stable operation.

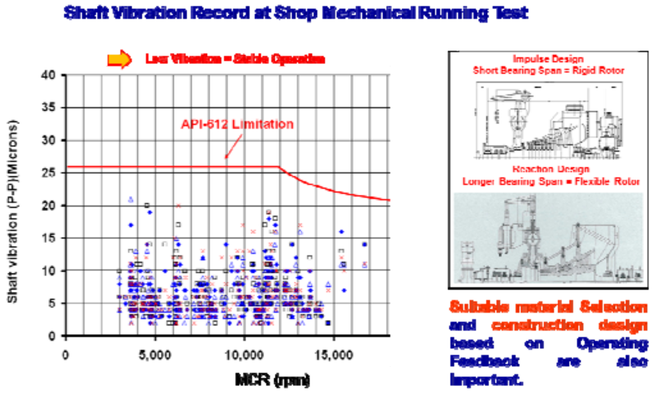

Figure.19 Reliable Operation for Rotor Dynamics

Case Study: LP Blading

In the following case study, the authors conducted an excitation test on two LP blade rows, using an actual scale model, and an actual load test with a scale model to verify the LP blades' reliability. With these tests, the authors could complete the verification program for the large steam turbine. Figure 20 shows the verification test for integrity of actual scale blades and confirmation of the principle of similarity. The rotating blades of steam turbines are essential elements in the conversion of fluid force to mechanical force, and high reliability in operation is a critical requirement. During the design/development stage, vibratory characteristics and strength of the actual blades were checked and verified with a rotating excitation laboratory test. However, an actual load test in a test turbine is the best method of verifying blade reliability. Due to the limited quantity of steam flow in the test facility, the actual loading test is usually carried out using a scale model turbine instead of a full scale model. Furthermore, mechanical and aerodynamic laws of similarity can be applied to the scale model turbine. This figure shows a summary of the principles of similarity.

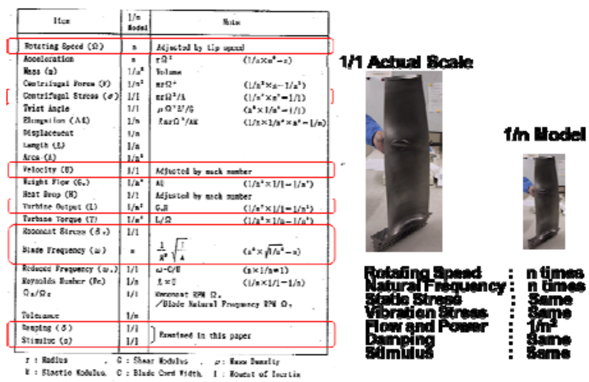

Figure.20 Verification Test for integrity of actual scale blades Confirmation for Principle of Similarity

Figure 21 shows the verification test for integrity of actual scale blades. Before a mechanical rotating test in a vacuum chamber typically used for high speed balance and the actual load test, tapping test of the LP blade rows in the free-free condition was carried out. The authors conducted a 3-D model study of the LP stage blades (actual/ scaled model) to verify the predicted natural frequency mode shape of the scaled low pressure stage blades. As a result, measured values for each mode showed a ±3% variation, compared with the calculated value.

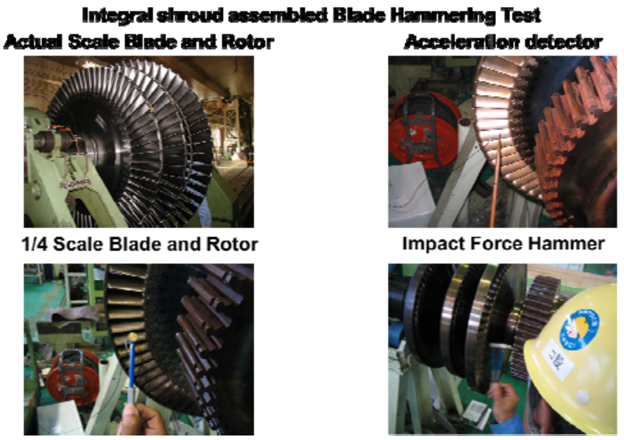

Figure.21 Verification Test for integrity of actual scale blades

Test Turbine

Figure 22 shows the test facility set-up schematic used for the scaled LP blade row verification. The major specifications of the test turbine and details of the manufactured test turbine and compressor are shown in this figure.

The test turbine was planned as a 5-stage straight condensing type, with 1 control stage and 4 LP stages. Its size was selected considering a steam flow for blade loading at the most severe case. For the inlet steam condition adjustment, a pressure control valve and temperature decrease devices were installed on the inlet steam line. Furthermore, the exhaust pressure was adjusted, and a vent valve was installed on the condenser. A 2-stage centrifugal type impeller was selected for the test compressor and air was used as the operating fluid in a closed loop.

Special Measurements

During the test, the authors took some special measurements for verification of the LP blade rows, including excited blades frequencies, vibration stress and flow path performance, in addition to normal measurements (speed, steam flow, pressure, temperature on piping, shaft vibration, etc.). In particular, to verify the LP blade rows vibration response, the authors computed blade vibration stress using strain gauges and thermocouples installed on the blades with a telemetry system. Figure 8 shows the special wiring for the verification test. Strain gauges were installed on the blade profile, tip or mean edge. Six strain gauges were installed on each stage (L-0 to L-3 stage). The authors adopted the latest available technology, an induced electromotive type telemetry system (no battery/self-generation type). Figure 23 shows the test rotor with the installed telemetry system. For reference, the cross section of the telemetry system is shown. The vibration response of the blades was available during testing through this system. Figure 24 shows a photo taken during measurement of the vibration response.

For verification of LP blade performance, the authors measured the Pressure and Temperature of each stage in the turbine casing and the turbine power through a coupling torque meter.

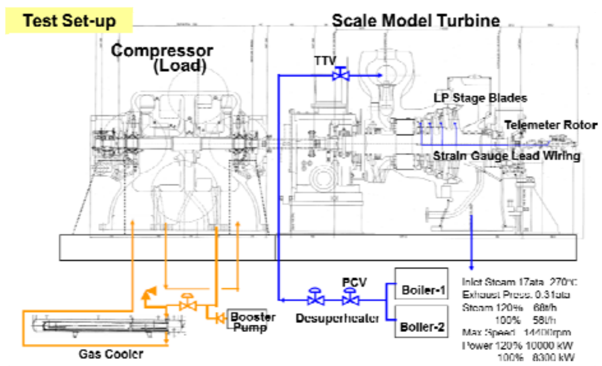

Figure.22 Verification Test for integrity of actual scale blades Full Load Blade Excitation Test

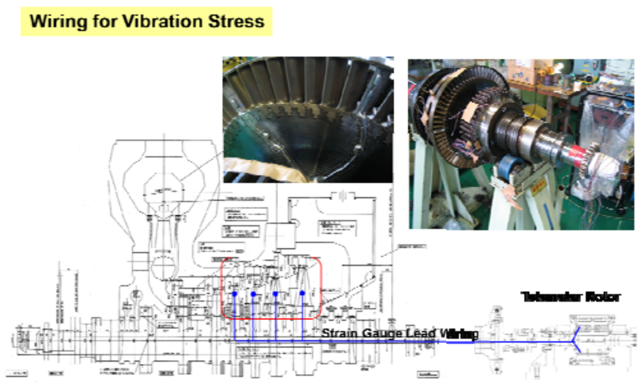

Figure.23 Verification Test for integrity of actual scale blades Full Load Blade Excitation Test

Test Conditions

The applicable operation range of the type 14 LP blade rows was evaluated for the case of a 1/4 scale test. Measurements were done at almost the whole applicable operating range. Turbine power and exhaust pressure at MCR (14,400 rpm) were adjusted and a stable operating condition was confirmed. The speed was then decreased to Minimum Governor Speed (MGS, 11,520 rpm), and then increased to MCR again and stable operating conditions were confirmed. During the stable condition at MCR, performance measurement was conducted, and as the speed changed, blade vibration response was measured (twice for each measurement)

These measurements were performed for a total of 50 hours of operation over seven days. Additionally, 1.5 hours of 110% load test at harmonic resonant speed was conducted to verify blade stress cycles in excess of 1×10^7.

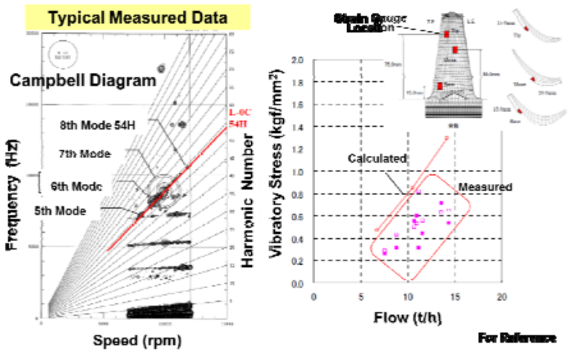

Figure.24 Verification Test for integrity of actual scale blades Full Load Blade Excitation Test

Test Results

Figure 24 shows the measured Campbell Diagrams of the L-0 last stage blades. In these diagrams, the size of the circle represents the vibration stress levels. By changing conditions during the verification test the authors evaluated a variety of vibration response stress levels including harmonic resonance, low vibration modes (below 20 harmonic), nozzle wake resonance, high vibration modes, and random vibration levels from a low exhaust vacuum condition (L-0 stage).

The authors then checked and obtained vibration response from harmonic resonance to low vibration mode (below 20 harmonic) test results. The diagram shows a comparison between calculated and measured results. This confirmed the natural frequency of the low vibration mode of the scaled blades, which agreed with the study results. (Other LP stage results also agreed.)

The largest resonant stress measured during the verification test for each vibration mode was tabled, and a safety margin of L-0 stage for each vibration mode was calculated based on an allowable stress and weakest point stress. This was considered as an actual stress measured point (tip or mean edge) in each vibration mode. This enables the evaluation of the safety margin for all blades. In the case of the L-0 stage, the safety margin of the 1st mode was the lowest, but met design requirements.

Typical Damage and Deterioration Maps

Figure 25 shows the typical damage modes of mechanical drive steam turbines. Steam flow path parts such as nozzles, diaphragms and blades are eroded by solid particles in the HP section and water droplets in the LP section. Corrosion fatigue failure can occur in the blade of the LP section at the transition zone of wet and dry conditions due to a corrosive environment. Excessive load due to drain carry over and poor quality of lubricant oil can induce melting damage to bearing pads.

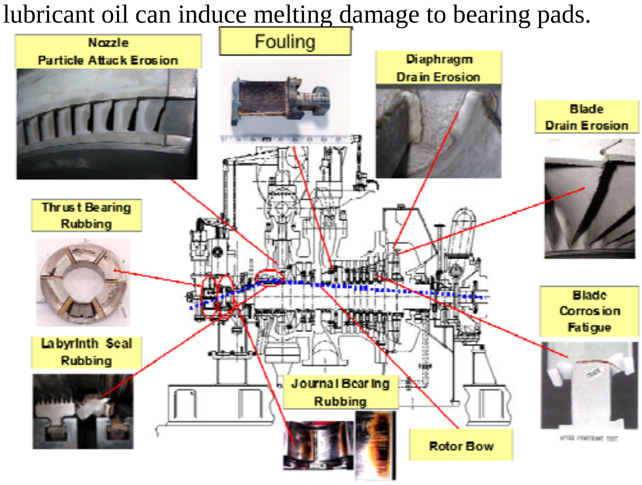

Figure.25 Steam Turbine Deterioration Map

Figure 26 shows the compressor deterioration map. Depending on the actual operating conditions, process gases can become wet and corrosive. As a result, the impeller tends to have flow path erosion, cracking at stress concentration critical areas and heavy chemical fouling which causes performance deterioration. At the balancing line, the differential pressure changes from design condition, excessive thrust force acts on the thrust pads and thrust bearing pads will see high metal temperatures. In worst case scenario pads can be damaged and compressor train may need to be tripped. Dry gas seals can have damage due to oil mist invasion or lower than dew point temperature of supply.

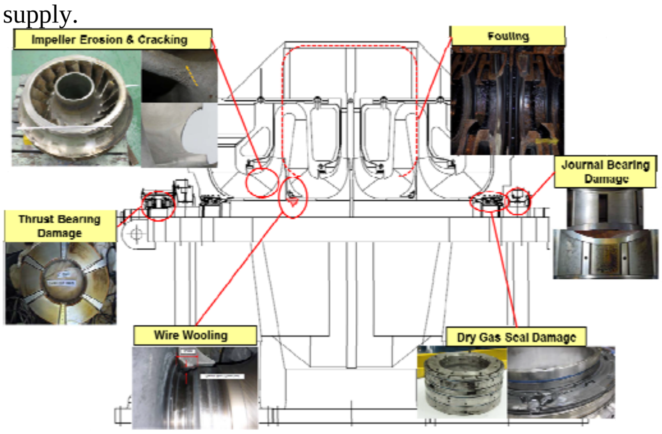

Figure.26 Compressor Deterioration Map

Practical Countermeasure Technologies Applied

Flow Path Surface Treatment

Figure 27 shows the applicable technologies for long term operation. Key technologies for steam turbine reliability improvement such as special blade coatings have been developed. In addition, applications of stainless steel materials to protect against drain erosion on the diaphragms, and the development of hybrid coatings of fluorine resin and ceramic have been utilized. One of the essential factors for long-term operation is coating technology. Regarding the deterioration phenomenon of internal parts, the two major causes are solid particle erosion and drain erosion. In the case of solid particle erosion, boronizing heat treatments are applied for high pressure stage nozzles, particularly the 1st stage nozzle. On the other hand, in case of drain erosion, Cr-TiN coatings and PTA welding are applied.

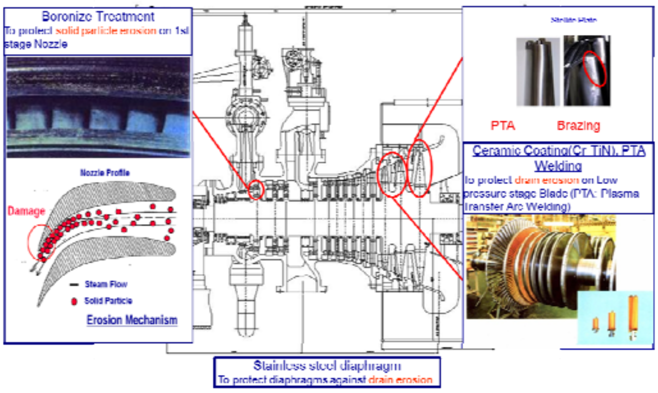

Figure.27 Applicable Technologies for Long Term Operation

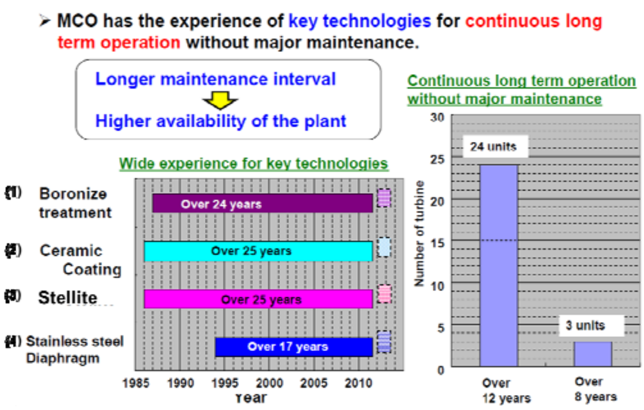

Figure.28 Experience for Long Term Operation

Figure 28 shows experience using applied technologies for long term operation. Key technologies, such as boronizing heat treatment, ion plating, Stellite brazing and stainless steel diaphragms, have decades of operational experience. Machines utilizing these countermeasures have achieved long continuous operation. Without major maintenance, 24 units, including drivers used in LNG plants, are running for continuous periods in excess of 12 years. Therefore, the turbines can be operated with longer maintenance intervals which will lead to achievement of higher availability of the plant.

Effective On-Line Washing

To compete in today's economic climate, petrochemical plants are strategizing on continuous long-term operation to reduce maintenance costs and increase productivity. This strategy has led some plants to go from eight years between turnarounds to 10 years. For rotating machinery such as mechanical drive steam turbines, one factor that affects this strategy is heavy deposition on steam turbine internals, caused by impurities in the steam. These impurities result in fouling on the blade and nozzle path surfaces due to contaminated materials such as silica and sodium as shown in Figure 29. As a result, turbine performance tends to deteriorate gradually.

As a countermeasure to performance degradation an innovative on-line washing technique can be utilized to minimize the impact caused by fouling of the steam path for large multi-stage condensing steam turbines. This technique, although applied here to extracting-condensing turbines, is also applicable to large condensing turbines. The new technology has water injection nozzles located at the steam chest of the extraction control valve rack. The injection nozzles are plumbed to a water supply source, which controls a set point temperature by controlling the water injection rate. The objective is to directly wash off deposits adhering to the blades and nozzles on the low-pressure side with minimal power turndown, and without impacting the turbine's long term performance. Erosion damage and thermal stress of internal parts such as chest valves and blades due to the injected water had to be taken into consideration. To properly achieve this objective, and considering the potential for damage during the online wash, a new extraction valve box had to be designed.

The new design had to consider the effects of optimizing the mixing zone of the steam and water injection to generate a specific particle size, moisture propagation through the condensing section, mechanical deflection of stationary components, and the overall thermodynamic analysis of each stage during the on-line wash. A prototype model was built and several experiments were carried out based on the practical operating conditions of actual steam turbines. Analysis (FEA) was used to evaluate the strength of the internal parts during actual online washing. The final design was a compact extraction box that could replace existing models without any machining of the casing.

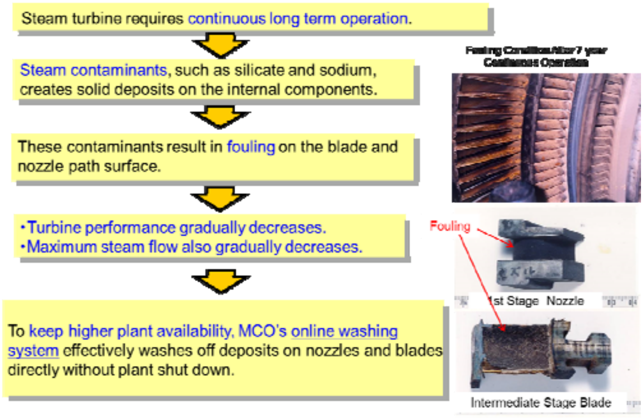

Figure.29 Background for development of On Line Washing

To keep plant availability, the online washing system is effective to wash off the deposits without a shut down. The washing system can be furnished on steam turbine itself, not on the inlet steam piping; it does not damage any other function, but improves the expected steam turbine performance. Figure 30 shows a drawing a map of typical online washing procedure. The steam turbine online washing system has already been used and applied over 30 existing turbine applications. First, hot steam comes from the high pressure section and the steam is mixed with injected water through spray nozzles in the turbine valve chest. The mixed steam is controlled in order to avoid the saturated condition, and fouling material on the nozzle/blade rows is washed by the steam in the wet condition.

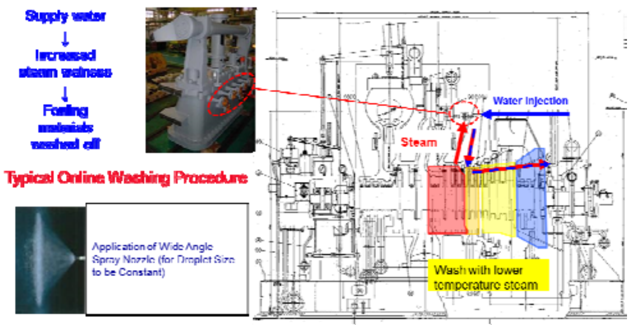

Figure.30 Online Washing for Steam Turbines

Figure 31shows typical operation data during online washing. The lower axis is steam flow, and the left axis is after 1st stage pressure for LP section. Before online washing, after 1st stage pressure is over the designed line, due to fouling condition. Though pressure gradually decreases during online washing, the pressure becomes stable in below the designed line after on-line washing. It means that performance was improved by on-line washing.

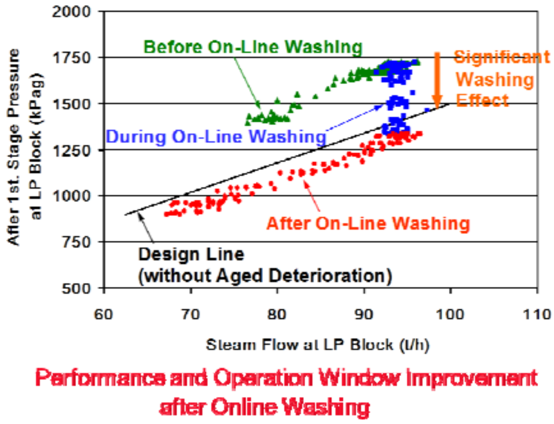

Figure.31 Performance Recovery by Online washing

Figure 32 shows the typical effect of steam turbine online washing at field execution. The result shows high power improvement after online washing, roughly a 6% recovery when compared to before washing.

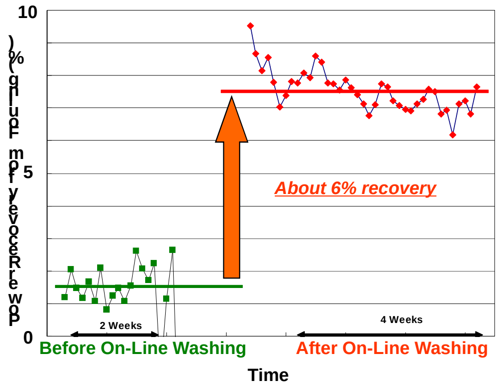

Figure.32 Typical Effect of Steam Turbine On-line

For the compressor, Figure.33 shows a typical fouling condition for a charge gas compressor in an ethylene application. Heavy hydrocarbon deposits will accumulate at the flow passage if the gas temperature rises above 90 °C. This fouling causes large flow path losses and a reduction in efficiency, and requires more power from steam turbine driver to maintain output.

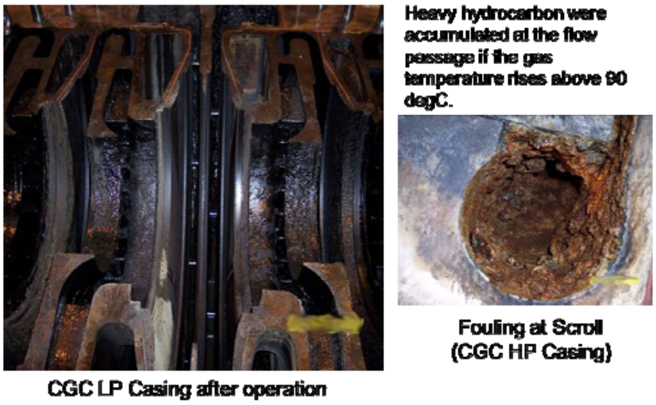

Figure.33 Typical Fouling in Charge Gas Compressor

CFD analysis was conducted in order to characterize the effectiveness of wash oil injection from the return bend, shown in Figure 34. As an initial result, the injected oil does not distribute remarkably. The effective oil wash area is only 3.6% of the return channel even if the quantity is increased. Another more effective injection method is necessary.

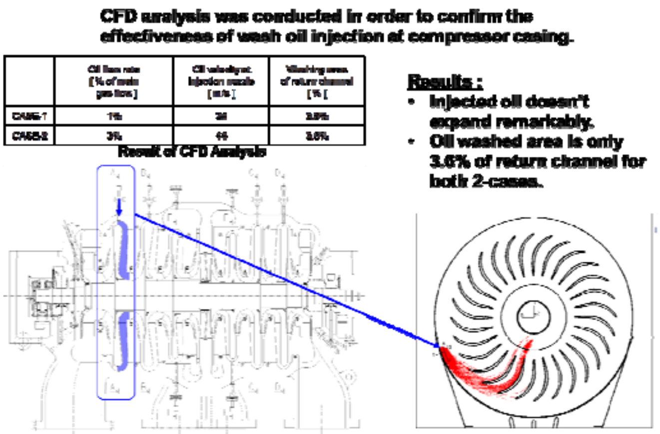

Figure.34 Verification of Oil injection at Return Bend

Figure 35 is the expected scenario of oil injection. First, oil is injected from return bend, but oil coverage is not expanded remarkably. However, when the oil reaches the impeller it washes the fouling deposit on the rotating impeller. After that the oil particles are scattered by the impeller. Therefore, the oil coverage is expanded widely to the next stage.

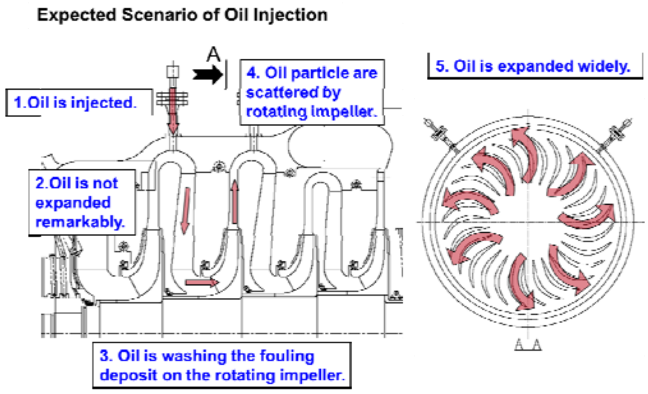

Figure.35 Expected Scenario of Oil injection at Return Bend

Figure 36 shows results of fouling survey at site successful oil/water injection. This is the result of a fouling survey at site. Due to effective oil injection, fouling of internal passages was minor. Internal hardware was very clean as shown on these pictures.

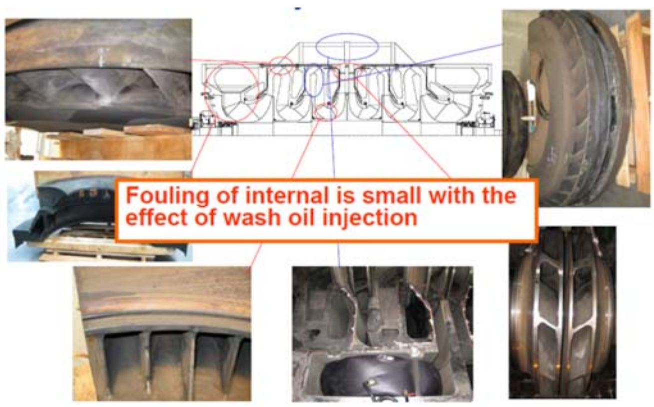

Figure.36 Results of Fouling Survey at Site Successful Oil Wash Injection

Combination of Anti-Corrosion and Erosion Prevention

A typical example of applications for robust technologies is shown in Figure 37. Application specific coatings are used to provide additional improvement against steam turbine efficiency deterioration. ISB designs were applied for all stages (impulse and reaction types) to increase overall efficiency and prevention of corrosion fatigue in the dry/wet transient zone. For the inlet 1st stage nozzles a boronizing heat treatment was applied to prevent solid particle erosion and L-1 and last stage blade surfaces were treated with TiN ion plating and PTA (plasma transfer arc welding) respectively to prevent drain erosion. In addition, wide pitch nozzles were applied for all stages to increase stage efficiency and to minimize after stage pressure due to fouling. In Figure 40, the difference in after stage pressure increase due to deposits on the nozzle profile is explained by comparing a fine pitch nozzle and wide pitch nozzle. If deposit thickness increase rate (mm/year) is the same, the nozzle area reduction rate of wide pitch nozzles can be decreased by half because of the smaller number of nozzles or smaller profile surface area than the fine pitch nozzle. The after-stage pressure increase is proportional to steam flow and the turbine can have a wider margin for operating time before reaching the pressure limitation under fouling conditions, while maintaining the required power.

Furthermore, if online washing is executed periodically, the operating time can be further extended by performance recovery. In this case study, the power recovery ratio after online washing is assumed to be 80%. The power decrease rate due to after-stage pressure increase caused by fouling is half of that of a fine pitch nozzle when using a wide pitch nozzle. If newly developed coating and other improvements, including online washing and wide pitch nozzles as explained above, are applied systematically the turbine integrity and performance can be maintained as same as the original even after 8 years of operation.

Figure.37 Surface Treatment Application for Steam Turbines

Figure38 is the typical fouling condition for charge gas compressors in ethylene service. To avoid fouling, water injection is done to keep the gas temperature low by water vaporization. A hydrophobic, low surface friction layer is applied as an anti-fouling coating for compressor flow paths to avoid the accumulation of polymer on internal surfaces for minimum efficiency losses. Oil (wash media) injection is done to clean the compressor internals and retain high performance without overhauls.

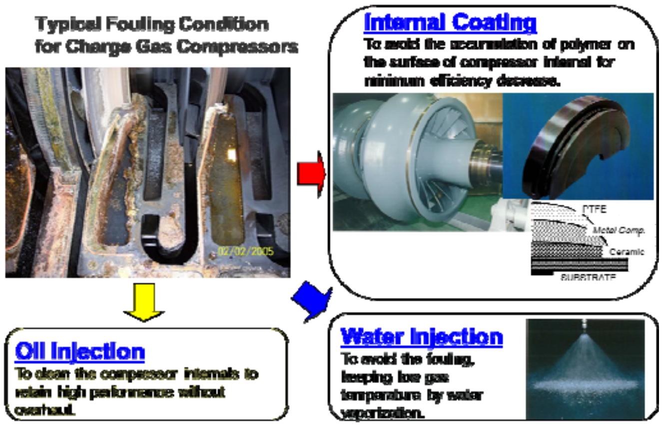

Figure.38 Anti-Fouling Coating for Compressor

Figure 39 is a typical example of an anti-erosion coating applied by high velocity oxygen fuel (HVOF) spraying. After four years of operation the 4th and 6th stage impellers had corrosion damage. Before changing the materials to anti-corrosion materials, for the spare rotor impellers of these stages an anti-corrosion coating by HOVF was applied.

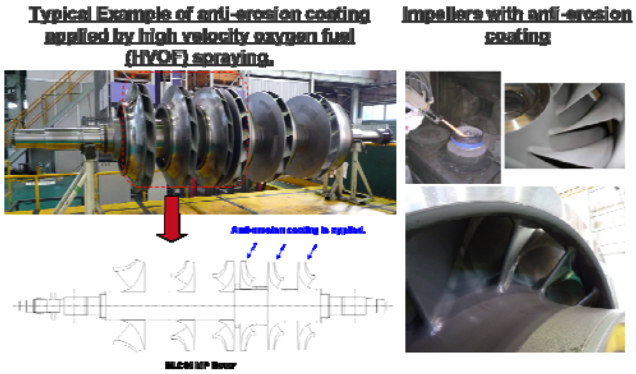

Figure.39 Anti-Erosion Coating HVOF Running Rotor

Figure.40 shows the evaluation results of anti-erosion factor for applied HVOF coating performance. The best composition of coating was decided from the point of view of anti-corrosion and erosion.

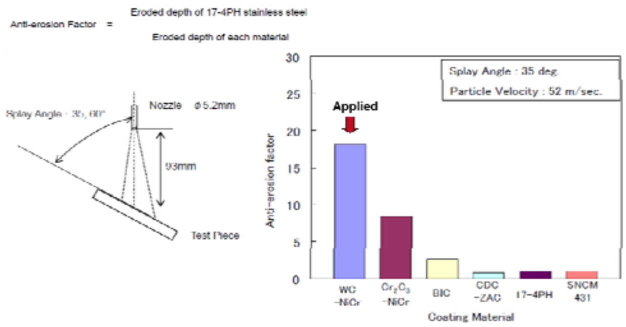

Figure.40 Evaluation of Anti-Erosion Factor for Applied Coating HVOF Performance

Stage Performance Enhancement by Partial Component Replacement

In case that existing machines need to be modified according to a desired process change or plant capacity increase, two options are proposed: one is a footprint replacement, casing with internals replacement; the other is the internal replacement of the rotor with impellers and stationary parts such as diaphragms. Figure 41 demonstrates the case of retrofit design of a flow field for a steam turbine LP section. Only the nozzle (stator) profiles of the diaphragms of the last 3 stages are modified to increase LP section efficiency. In this case study, total efficiency could be improved by 1.5%.

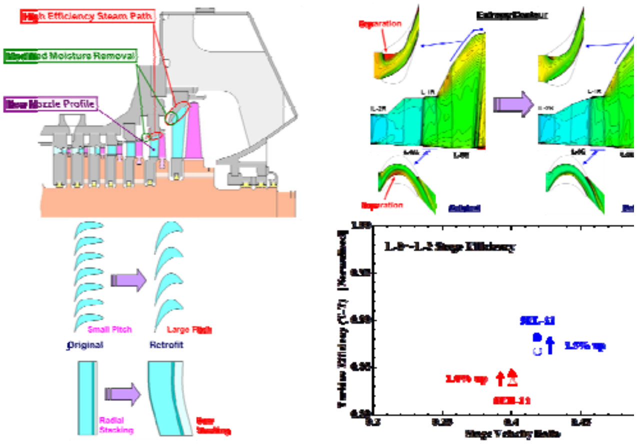

Figure.41 Retrofit Design of Flow Field for Steam Turbine LP Section

Figure 42 shows drain erosion prevention for a steam turbine LP section. The last stage moisture is about 10% to 14% and 20% to 30% of this moisture becomes water. Small water droplets of 50 to 300 micrometer diameter scatter to the blade with a high velocity and attack the blade profile surface. The number of such droplets has an adverse effect on the turbine last stage efficiency. If the number of such droplets is reduced, the drain damage is attenuated and efficiency can be increased. This nozzle profile has a slit to gather the condensation on the nozzle surface and an intake hole is provided in order to extract the drain outside of the flow path.

The geometry of the structure such as slit and hole locations is decided by analysis of the results of a 2-phase CFD (steam and water droplets).

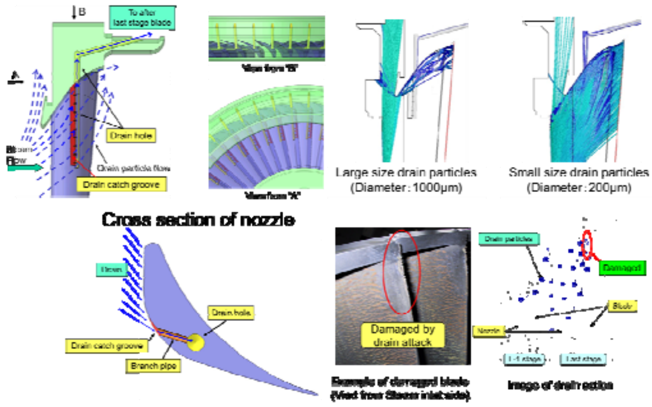

Figure.42 Drain Erosion Prevention for Steam Turbine LP Section

NDE Techniques

For long term operation, surface treatment technologies are applied. NDE techniques are also developed to inspect the internal damage of the rotor disk groove. One of the latest technologies is PAUT, phased array ultra-sonic testing. In contrast to traditional straight-beam UT, phased array sends and receives multiple UT signals simultaneously, allowing for greater detail and resolution. Based on measured crack length, location, and orientation the allowable residual operational life can be estimated for turnaround planning. Figure.43 shows the results of a PAUT inspection for steam turbine stress corrosion cracking (SCC) at the inside of the groove under the blade assembly.

Figure 44 shows a section of part of steam turbine disk and an actual stress corrosion crack fracture analysis result. According to these results, the PAUT measurement accuracy of crack length and location is verified.

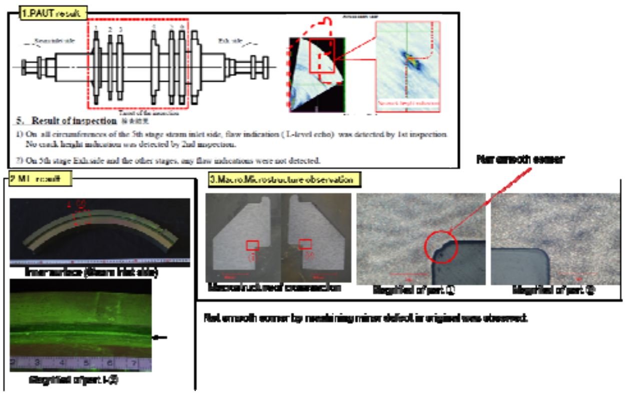

Figure.43 PAUT Phased Array Ultra Sonic Test for Steam Turbine SCC

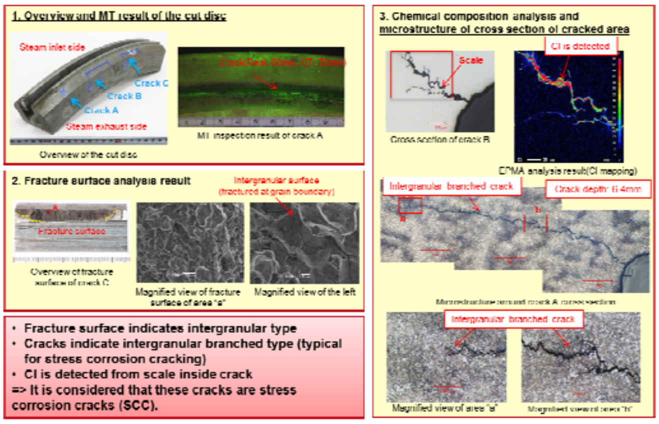

Figure.44 Typical SCC Crack for Steam Turbine Rotor Disk

Unique Casing Replacement Technique on Same Footprint

Figure 45 shows the basic steps for plant revamping considerations. As a first step, the end user should consider whether or not process improvements alone can satisfy their rerate targets. If so, this can be accomplished without equipment modifications. If process adjustments alone do not satisfy the requirements, the second step in this process is to consider re-rotoring the existing equipment. However, when mechanical revamping of the existing equipment will not achieve the plant rerate target, the end user will most likely need to consider equipment replacement and/or parallel train applications as shown in the slide.

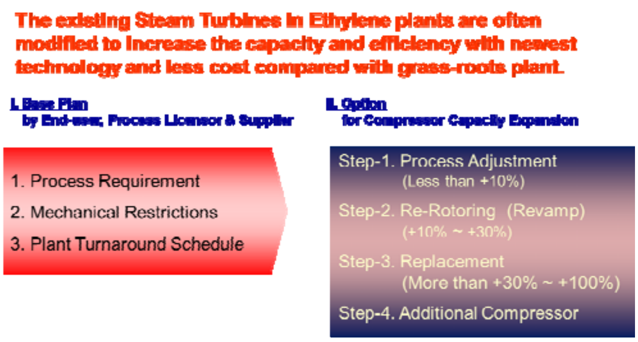

Figure.45 Basic Steps for Plant Revamping

In the replacement approach, an existing compressor is replaced with a new high efficiency design. In this case shown in Figure 46, it is important that the new large compressor is designed to allow the casing to be put on the existing foundation and the nozzles to set in the opening of the existing foundation. When a standard design compressor is applied to a replacement project, often nozzle positions will not fit in the existing main gas pipe positions and/or do not set in the narrow openings of the existing foundation. Modification of foundations significantly increases the difficulty, time, and cost of the installation.

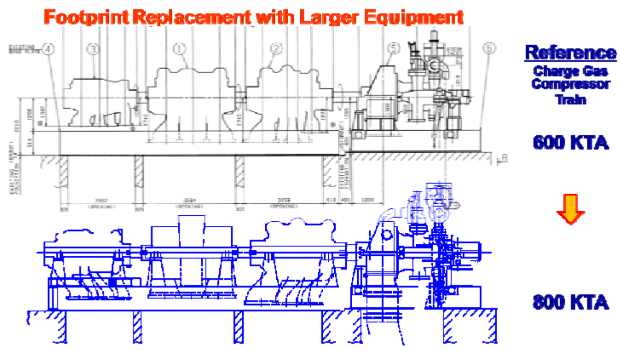

Figure.46 Replacement for Plant Capacity Increase

In terms of the cost side of revamping versus replacement, Figure 47 shows a comparison of initial investment cost based on 300 kiloton per annum ethylene plant. It is easily imaginable that the equipment cost of replacement is higher than that of revamping.

However, it is less well known that the total cost difference is not so large - only 18%. In the meantime, a 40% capacity increase could be achieved by replacement, while only a 20% capacity increase is achieved by revamping. So the initial investment cost difference may be able to absorbed by the profit difference from the production increase between the two cases. For the case at hand (300 KTA plant), production increase by replacement is 120 kiloton per annum.

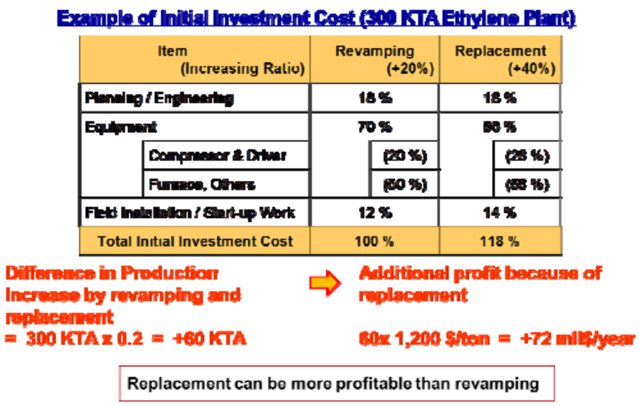

Figure.47 Comparison of Initial Investment Cost

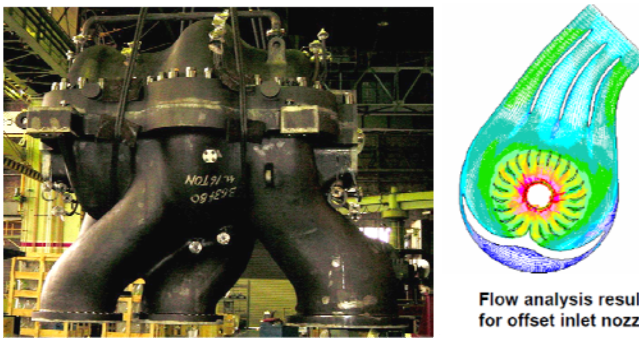

Figure.48 Flexible Casing Nozzle Design for Compressor

Figure 48 shows a flexible casing nozzle design for a replacement compressor. The nozzle design is very important for compressors. Compressor cases are typically formed by casting. Therefore, the configuration of the nozzle is very flexible if the location requirements are considered early in the design process. This is especially useful for a footprint replacement project. If the replacement is properly done, the base plate can be reused and the process piping positions remain fixed. Therefore, the compressor nozzle has to be flexible in order to replace the existing compressor to larger one without significant and costly modifications. The flow of the nozzle should be analyzed by CFD analysis in order to ensure the design is successful.

CONCLUSIONS

Turbomachinery manufacturers have been going forward to apply challenging technologies for critical components in order to provide highly efficient and more reliable machines corresponding to the requirement of higher power for larger ethylene services. The authors introduce the technology development history map in past and future for typical design, verification test and application results in terms of transient fluid dynamics, thermodynamics, rotor dynamics, and blade vibration strength evaluation. In addition, after commissioning, turbomachinery tends to have deterioration of performance and reliability for long-term operation. The typical damage and deterioration map for machine are introduced and the authors explain the practical technologies actually applied, such as flow path surface treatment, effective on line washing, combinations of anticorrosion, erosion prevention and stage performance enhancement by partial component replacement, NDE techniques for both compressors and steam turbines, and unique casing replacement techniques on an existing footprint for revamps to increase capacity.

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