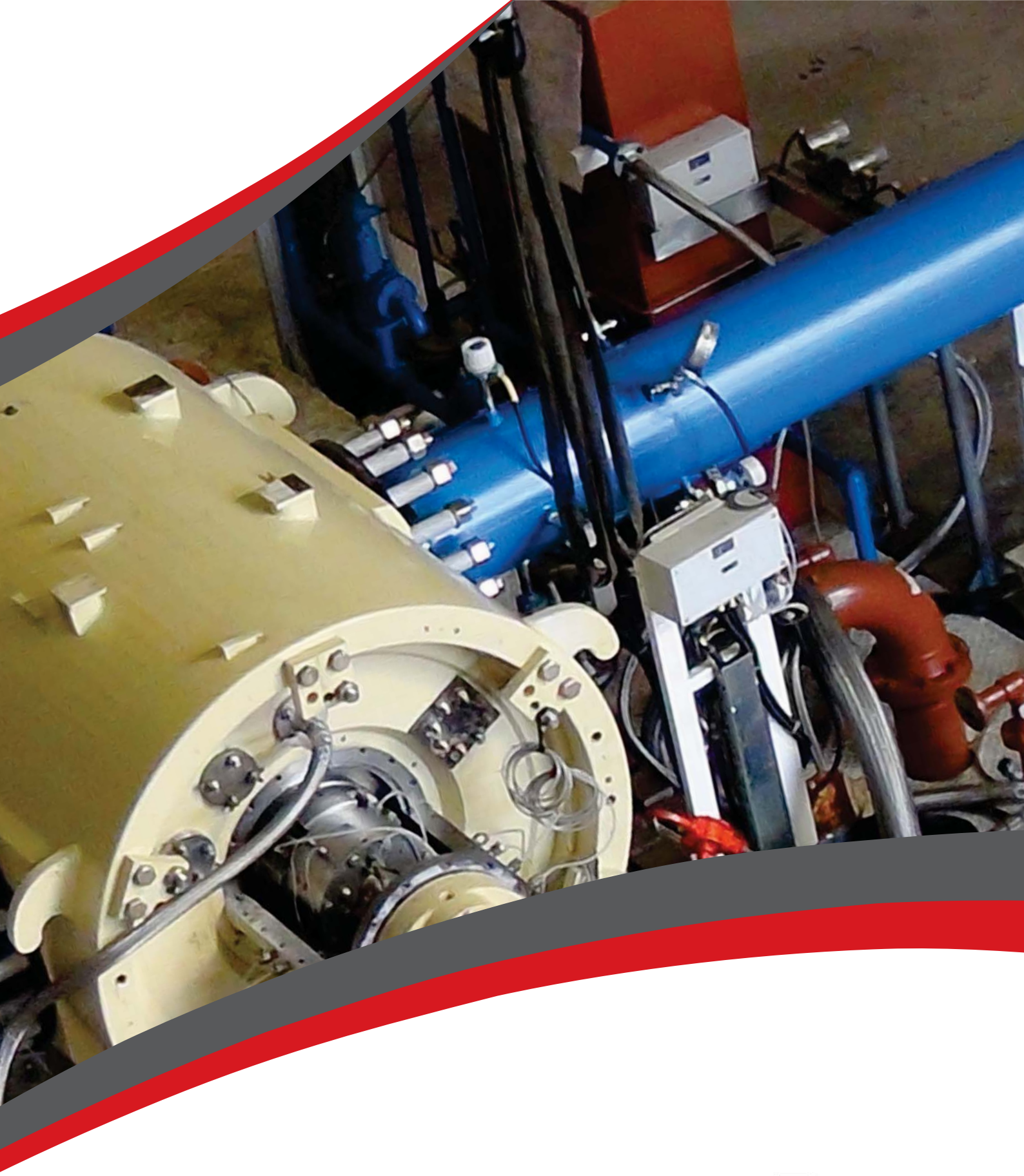


Technical Review

MAPNA TURBINE ENGINEERING & MANUFACTURING CO. (TUGA)

TUGA
MAPNA GROUP

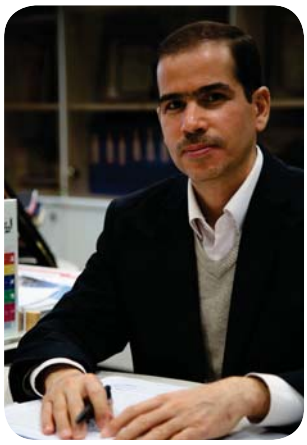
Willpower to Empower Generations

No.5 - March 2016

Cover Page

TUGA Centrifugal Compressor Test Bench

EDITORIAL



"If you always do what you've always done, you'll always get what you've always got"

Henry Ford

Dear Customers, Partners and Professionals,

At MAPNA Turbine people perpetually strive to practice Willpower to Empower Generations, true to our slogan. As a result, we are perfectly aware that an integral part of willpower is the courage to recognise changes, admit them and take the risk to modify our approaches accordingly. This 5th issue of MAPNA Turbine Technical Review presents a brief account of a few recent achievements in our everlasting journey towards excellence and finding new answers to deal with existing and upcoming challenges.

As a result of climate change and the subsequent water shortage in some areas, sustainable use of steam turbines with wet cooling systems is not practicable any more in the affected regions. Converting from wet cooling to dry cooling, on the other hand, requires a redesign of the core engine, steam turbine, and can be as demanding as replacing the whole rotor with a newly designed one. This redesign, primarily involves the last stage blade, but a verification of machine integrity regarding other design parameters is also necessary. This has been addressed for a specific machine in the first essay of this issue, to beautifully show that extreme redesign can be avoided through following an innovative approach looking at the bigger picture.

Given the OEM-class knowledge acquired at MAPNA Turbine, to run continual redesign and reanalysis of machines for improvement purposes, we need to develop new packages. MAXCD is a package primarily developed to verify intended design modifications to be introduced in axial compressors. The second item of this issue provides a rather detailed description of the functionalities.

CFD models to enable analyses of the flow in gas turbines need to be updated using the practically acquired results. Given that they are used in designing the heat recovery process in CHP and combined cycle applications, producing accurate results when analysing the exhaust gases is important in particular. Our test set-up to acquire the flow properties of the exhaust gas are discussed in the third essay.

Article four covers the hardware/software provisions and the knowledge acquired in-house and through technology transfer to get the customer requirements covered when designing and manufacturing centrifugal compressors.

Last but definitely not least, the fifth article outlines a simple approach to significantly increase the output power in the rather vast gas turbine fleet of MGT-70 with minimal effort. IGV+ is the title for the effective initiative that has been made available to the national grid, to economically boost production capacity in short time, with low cost and no intrusive action.

The material in Technical Review issues is based upon the information meticulously produced by our research engineers of different technical disciplines, whose invaluable sharing is hereby appreciated.

Respectfully,

Mohammad Owliya, PhD

Vice President for Engineering and R&D

MAPNA Turbine Company (TUGA) - March 2016

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Switching from Wet Cooling Systems to Dry Ones; A Breakthrough in Management of Change in Steam Turbine Design

Introduction

There are eight units of 320 MW Steam Turbines (similar to our MST-70) in Iran, currently in operation having HP-IP and LP sections in two separate cylinders. The HP section accommodates one Curtis stage together with 11 reaction stages. The IP section in turn incorporates 10 reaction stages and finally the LP cylinder includes two back-to-back LP sections with 6 reaction stages each.

These machines are originally designed for high vacuums with condenser pressures around 7 kPa for which 33.5in blades have been incorporated for the last rotary row of the low pressure sections. Although designed for such low back pressures, the blades are able to tolerate downstream pressures up to 28 kPa as the turbine trip value. However an alarm goes off at 19 kPa implying that the machine had better be operated with lower back pressures in order to protect the last stage blades (LSB) against damages due to fluttering. Being welded together in groups of eight with two lasing wires, the LSB are still vulnerable to high condenser pressures.

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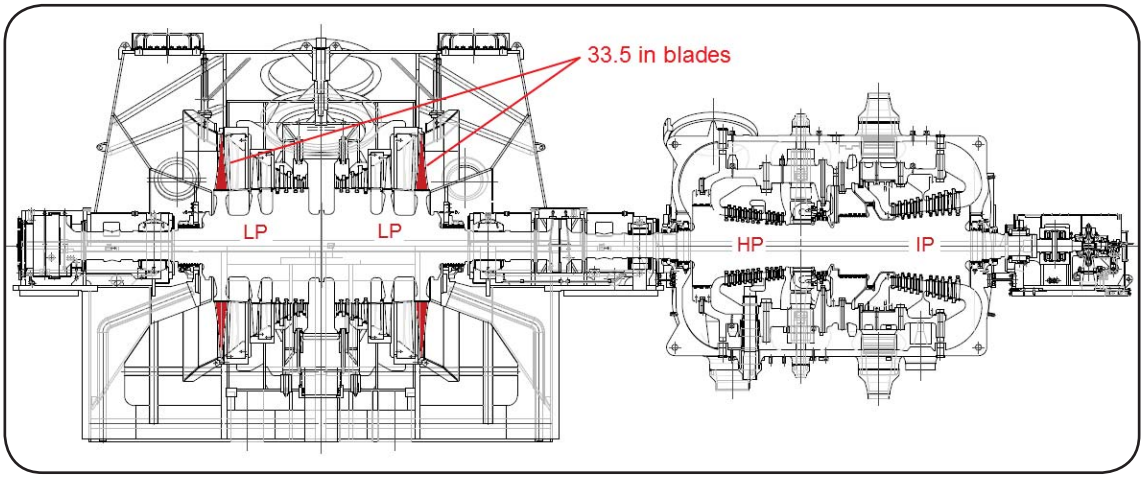

Figure 1: Cross sectional drawing of the machine

Water shortage problem is gradually deteriorating over time in the Middle East and some other regions in the world and this pushes the power plant owners to switch from wet cooling systems to dry ones that consume far less water. However, turbines need to catch up as well, with redesigned LSB withstanding higher condenser pressures.

Among the aforementioned eight units in Iran, the Islam-Abad power plant in Isfahan was selected to start with. The target is to replace the old wet cooling system with a dry one of Heller type. The design back pressure is assumed 15 kPa and the turbine trip value is desired to be higher than 35 kPa in order to have an economical design for the cooling system.

Challenges Ahead

Since the machine is in operation, there are some constraints imposed on this problem such as the following:

- Bearing span must be kept unchanged.
- LP outer casing must be kept unchanged as it is directly connected to the condenser and also to the HP-IP cylinder via two cross-over pipes.
- Weight of the LP rotor should not increase as it may have an impact on the bearings size.
- Three extractions in the LP section must still exist with the original pressures.

With the bullet points mentioned, the first solution would be to retrofit the LP sections with shorter standard LSB, with a smaller exhaust area so as to increase the exhaust axial velocity which rules out turn-up losses and unwanted fluid stream lines due to low volumetric flow. The turn-up losses lead to lower power output of the last stage, and in extreme cases, the output of LSB dies down to zero beyond which, it would work like a compressor stage that consumes power. This turning point is the threshold of fluttering problem that can cause vibrations and blade fatigue fracture.

This should be noted that shorter standard LSB means shorter blades on a thinner rotor as the new blades have to be installed on a lower base diameter. This means that the old rotor will not be able to carry on its duty and should be replaced with a new one obviously with new blades for all the 6 stages. The downside is the enormous expense that the owner has to bear mainly for the new rotor. All the technical advantages might fade into insignificance compared with the last off-putting item.

To make the budgetary proposal more attractive to the customer, decision was made to find a solution that does not entail using a thinner rotor, and eventually the target was reached. The new LSB (Figure 2) are shorter with new profiles, though with the same base diameter as the original ones.

Figure 2: Original and modified LSB

Technical Description

As mentioned earlier, the profile and the stacking of the sections were modified. This modification focuses on the last stage (rotary and stationary) and the downstream diffuser. The rotor, all the upstream LP stages, and the LP outer and inner casings are kept unchanged. The stationary blades of stage 6 will be shorter, and so will be the modified blade holder depicted in Figure 3.

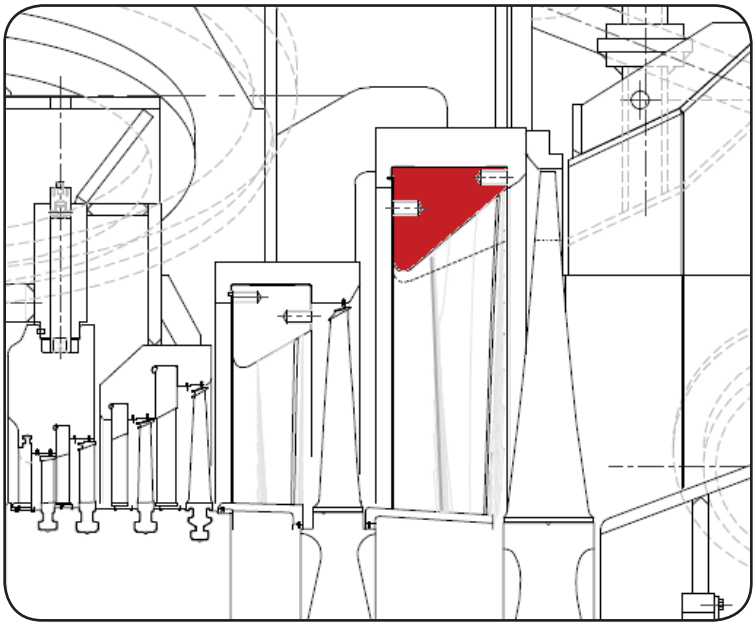

Figure 3: Stationary blade holder

The modified blade is 25 in long for which two-phase 3-dimensional CFD calculations have been performed. In Figure 4 the output power of the last stage is shown for the original and modified blades.

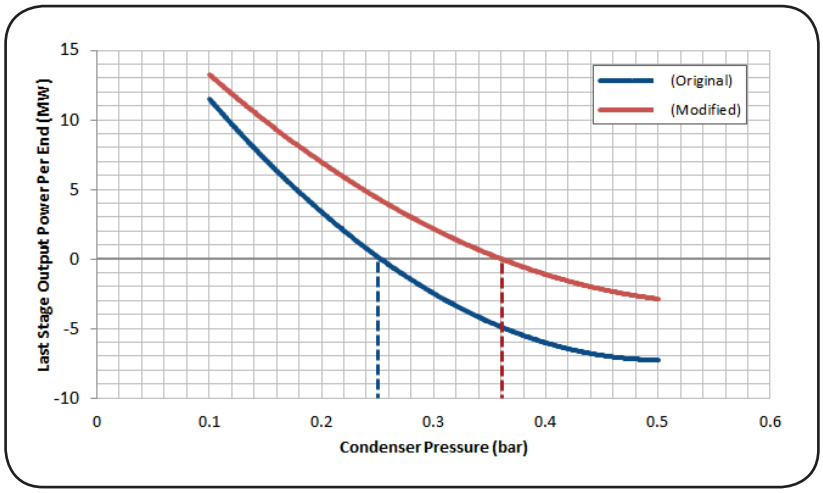

Figure 4: Output power of LSB versus condenser pressure

As can been seen, the output power of the modified blade becomes zero when the condenser pressure reaches 36 kPa which is more than what was expected at the beginning.

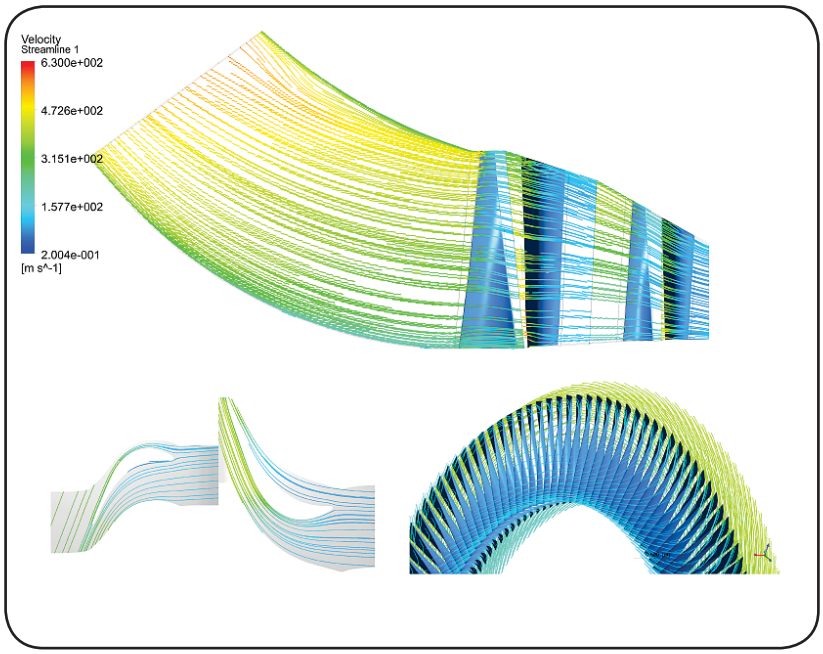

Figure 5: Regular stream lines with modified LSB

In addition to the CFD calculation which is a "NICE-TO-HAVE", there is also a "MUST-HAVE" which is the mechanical integrity, basically natural frequencies and strength withstanding operation conditions.

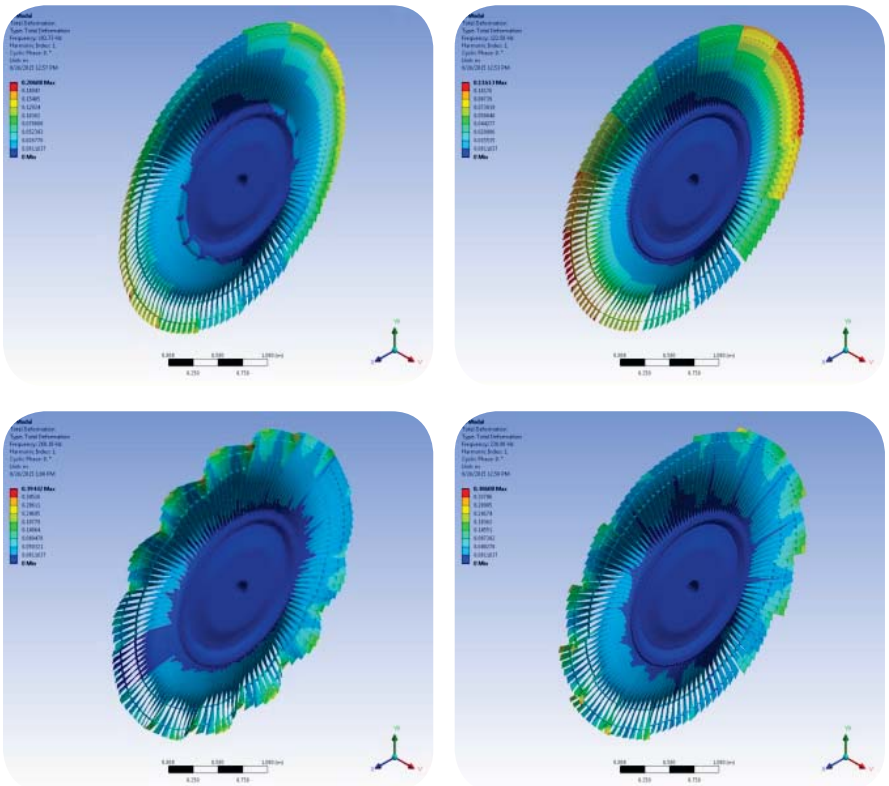

Figure 6: Natural frequencies of grouped blades

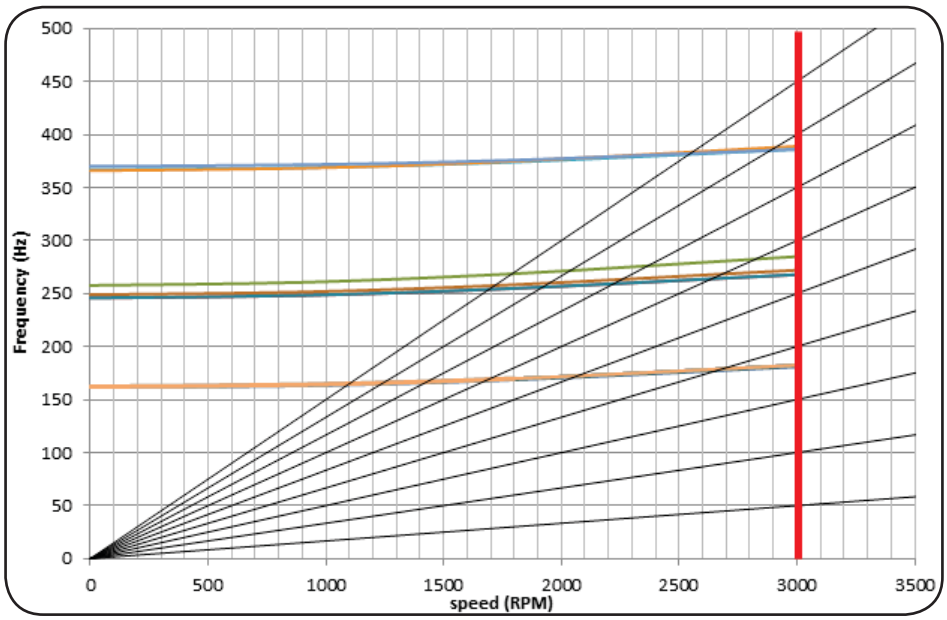

Figure 7: Campbell diagram of the modified LSB

According to the Campbell diagram, none of the natural frequencies of the modified blades coincide with the running speed harmonics which shows that they can operate safely at 3000 rpm.

For the mechanical analysis, it can be concluded in a short look that since the weight of the blades has decreased, the centrifugal stresses are lower. In a similar way, for the dynamic stresses which are affected by mass flow and enthalpy, the same holds true; as those parameters have not been intensified. Therefore the new LSB can cope with High Cycle Fatigue (HCF) loadings. Nevertheless, to make sure we are on the safe side, the calculations were performed and the results were compared with the ones for the original blades as illustrated in Figure 8.

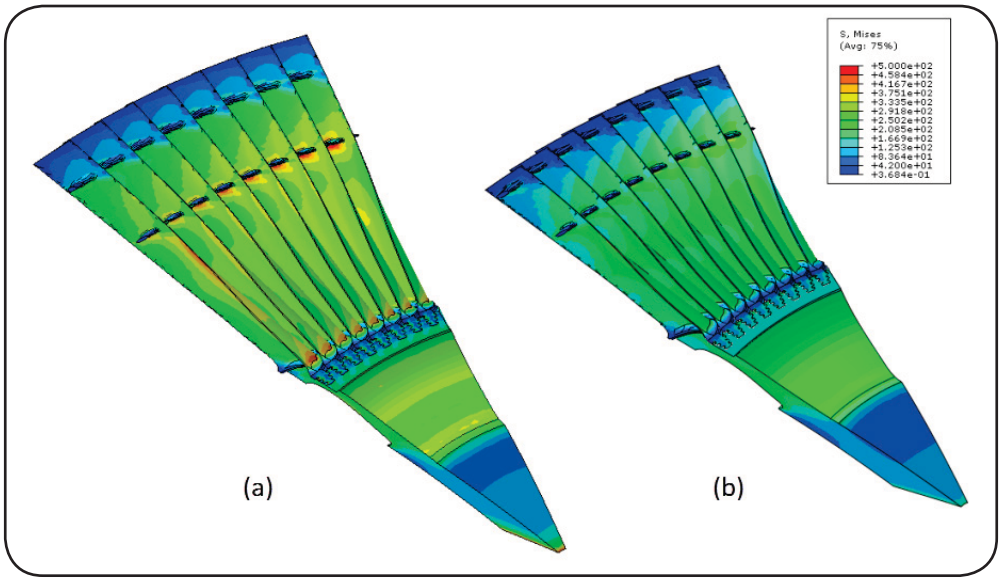

Figure 8: Mechanical analysis; (a): original blade; (b): modified blade

Conclusion

The devised solution is quite feasible and far less expensive than the solution in which the rotor has to be replaced. Moreover, this method can be applied to other power plants which use wet cooling system. This move is inevitable to respond to environmental concerns as our social responsibility.

MAXCD: MAXimum Effort to Pinch the Best Possible Service out of Axial Compressors

Introduction

In recent decades, a multitude of studies and research activities have been conducted to improve knowledge in design and optimization of axial compressors with various industrial applications. TUGA, as a major manufacturer of turbo-machines in Iran and in the region, has also set off for developing the knowledge and design tools in this area.

There are commercial packages specifically developed by various manufacturers and designers. Taking into account its own requirements and market demand, TUGA has also developed a local package customized to design and conduct the preliminary optimization of axial compressors: MAXCD, MAPNA Axial Compressor Design.

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There are particular functionalities making MAXCD package stand out among its counterparts. Although some of these functionalities are offered by similar packages, but integrating them all into MAXCD has provided it with quite a few outstanding features including but not limited to the following:

- Analysis of the existing compressors using the geometrical information as the input
- Compressor design using the initial inputs to produce the 3D geometry
- Optimizing the design through statistical algorithms
- Partial optimization to be used in upgrading projects in the existing products
- Producing multi-variable objective functions based upon design requirements
- High precision thanks to employing various built in empirical correlation curves
- Minimal elapsed time to produce the results, thanks to employing optimized algorithms
- 3D geometry generation and producing files that are identifiable by well-known analysis packages, ANSYS and NEMECA

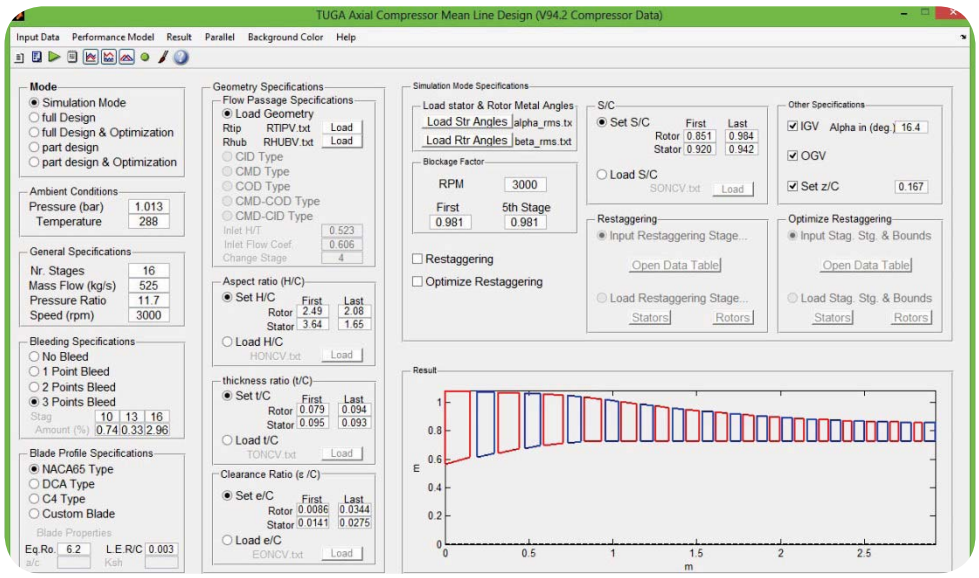

Figure 1: View of the Main Dialogue Box of MAXCD

These are the general functionalities of the package:

- Analyzing all axial compressors in their mean radii to extract the performance parameters
- Producing performance curves for axial compressors
- Design and optimization of axial compressors in their mean radii
- Developing the design to include other radii and optimizing the distribution of various parameters both in transition from various stages and along blade spans
- Making use of a user-friendly graphic interface
- Producing blade/vane geometries based upon conventional airfoil families such as NACA65, DCA, etc.

This software package was developed based on Mean Line method and so are the relevant design and analysis. Design and optimization modules were further developed making use of Through Flow Calculation technique to enable the user to generate 3D geometry and later on export it into commercial analysis packages. The package can also be used to conduct preliminary and basic design of axial compressor. When in the analysis mode, geometric inputs only are provided for the software and their associated performance parameters are produced as the outputs. The outputs encompass all design parameters of an axial compressor including Flow Coefficient and Load Coefficient along with its performance curve. Another fundamental feature of the package is generating performance curve of axial compressors in different operational speeds.

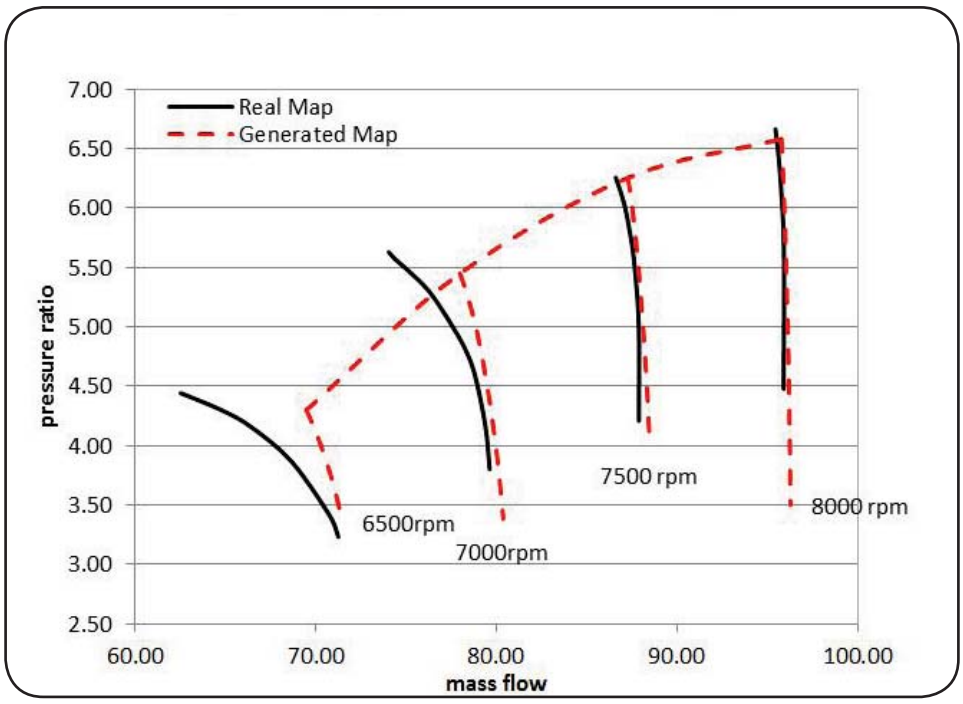

Figure 2: A sample of the produced performance curves in the analysis mode of the package and how they compare with experimentally produced curves

Using thermodynamic relations and the gas dynamics governing the flow regime inside the machine, and a variety of empirical correlations like Koch-Smith and Wright-Miller, to estimate: flow loss, optimum incidence angle and flow deviation angle both in design and off-design conditions, effort has been made to come up with an appropriate model to serve three purposes:

- Approximation of the performance for existing machines
- Designing new machines, and
- Redesigning parts of the existing machine to upgrade their performance

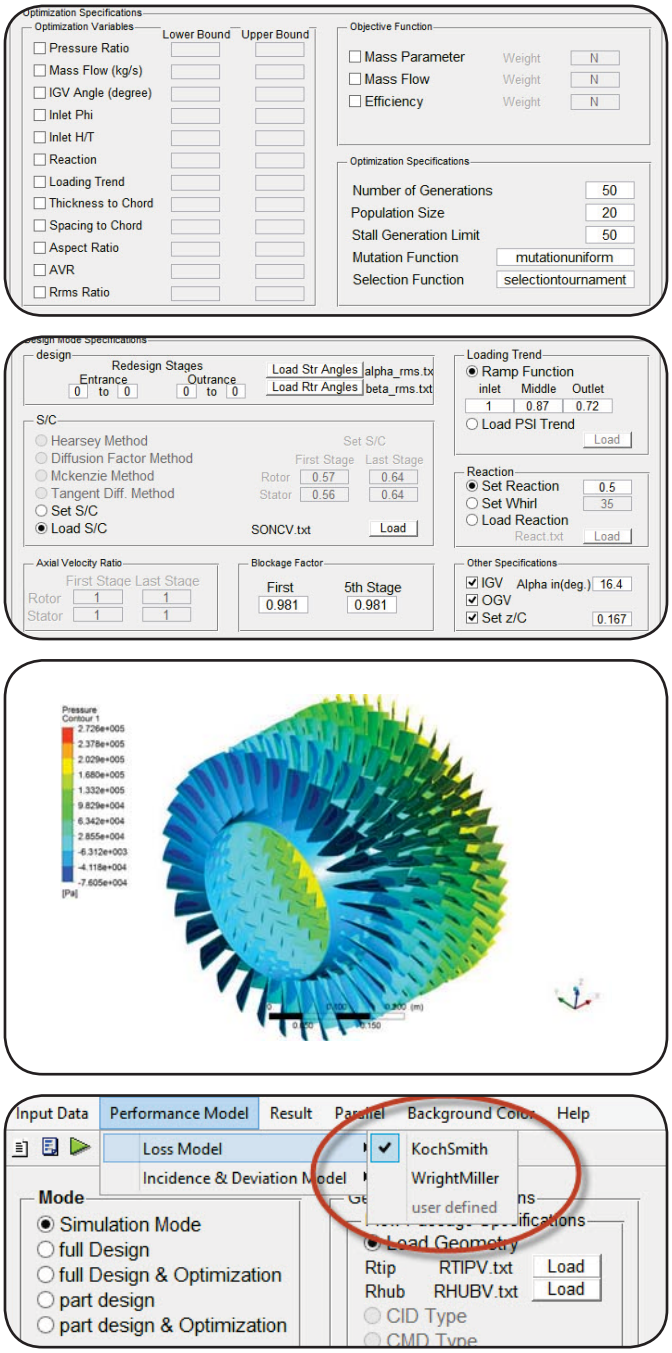

Figure 3: The package features in using different loss estimation models and design and optimization tools

For optimization purposes, the developed package was coupled with Genetic Algorithm to enable optimization of parameters like mass flow rate, pressure ratio and efficiency, through modified blade loading, share of reaction in compression in different stages and a few other design variables. The objective function can be defined as a single-variable or a multivariable function depending on the required application, and each single influencing parameter can be optimized through allocating a different weight coefficient from those allocated to the others.

Experimental Measurement of Exhaust Gas Parameters for MGT-30 Gas Turbine, A Way Paver to Add more Accuracy to Analytical Models

Introduction

Efficient approaches to generate electric power and recover thermal energy in power plants are important, and have led to further development of systems like combined heat and power technologies (CHP) and combined cycle power plants. Having on hand a precise model of gas turbine exhaust gas stream as a heat source for the downstream steam cycle is a prerequisite when designing an optimized system. Regarding the intrinsic restrictions of computational techniques and numerical models and the inaccuracy involved, using experimental methods and real measurements is considered a useful supplement in development process of those models.

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MAPNA Turbine Engineering & Manufacturing Co. (TUGA) produces the industrial gas turbine MGT-30, with two applications: mechanical derive of centrifugal compressors and generator drive in turbo-generator sets. For turbo-generator applications of this gas turbine, the study of properties of the exhaust gas stream is an integral part of the optimal design of boiler, in combined cycles and desalination plants. Analysis of exhaust gas of 25 MW MGT-30 machine was carried out on the exhaust gas stack which is installed in the gas turbine factory test bench, in order to fine_tune developed numerical models of the gas stream for this machine.

The parameters to be measured in this experiment are total temperature, total and static pressure and velocity vector of exhaust gas stream. The general characteristics of the stream are given in table 1.

Table 1: Characteristics of MGT-30 Gas Turbine Exhaust Gas Stream

Flow content	Gas turbine exhaust gas
Flow temperature	≈ 500 °C
Flow total pressure (at the lowest measurement section of stack)	≈ local atmosphere pressure20+ mbar
Flow velocity	Max: 120 m/s (Numerical Predicted)
Predicted flow pattern	Turbulent with overall swirling (specially, after the gas volute of stack)
Steady approach of flow	Steady
Pulsation condition of flow	≈0.01 Hz (sampling frequency)

Three measurement sections of the exhaust stack are chosen for this study as shown in figure 1. At the first step, flow measurement will be performed at the second considered section (the Spacer section) involving a turbulent stream. In the second phase, the measurement setup will be developed for the stream with a more complex pattern at section 1 and the more homogenous one at section 3.

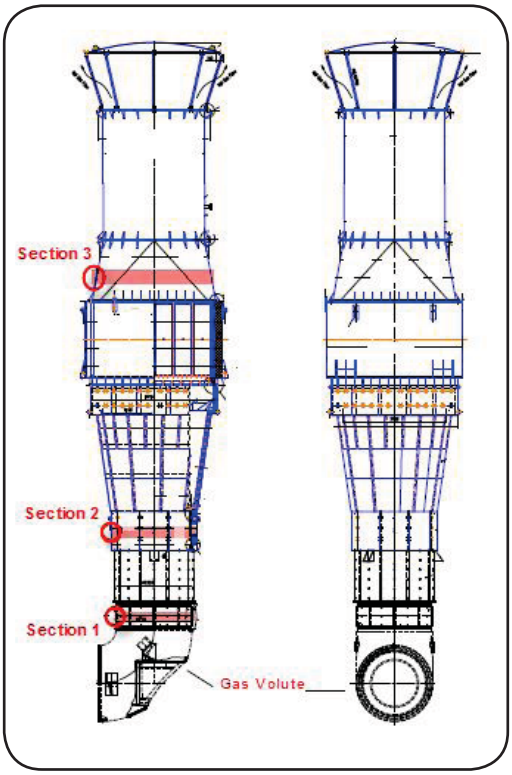

Figure 1: Measurement Sections of Exhaust Stack

Design of Sampling Points

The first step for design of test procedure is definition of number and arrangement of required sampling points. In this section, Sampling Points Arrangement has been designed according to EPA standard. This standard has been developed by the Environmental Protection Agency of America. The EPA-2F document proposes the methods for 3D measurement of gas flow in ducts, stacks and pipes (internal flows). According to this standard, the number of sampling points in a flow could be defined based on the flow angle by two different methods: Simple Method and Alternative Method.

The simple method can be used when the overall stream lines are parallel to the main flow axis and do not form a large angle and when the flow is not swirling. If some components such as bends, expansion or contraction joints, etc. cause disturbance in flow, it will be necessary for measurements (especially for velocity) to be carried out at least at the 8D[1] downstream and 2D upstream of such components in order to get apart enough from the disturbance affected area. In such cases the minimum number of sampling points could be defined by figure 2 according to EPA standard.

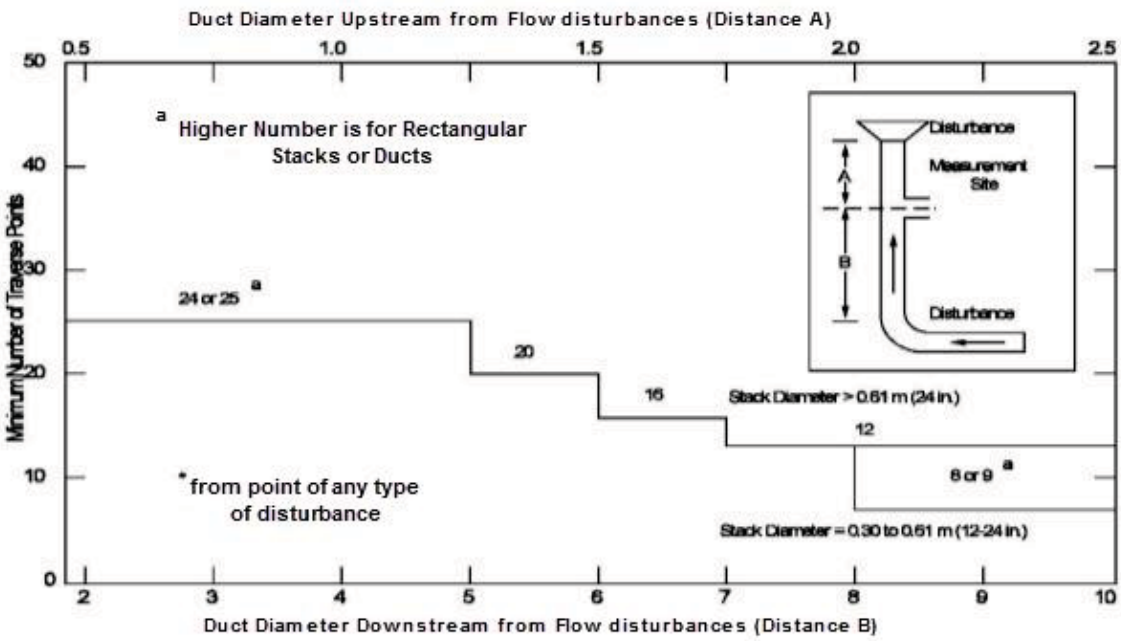

Figure 2: Measurement Sections of Exhaust Stack

In some measurement sections, if the inclination angle of stream, is more than 20°, the Simple Method is not applicable any more, especially to measure the velocity. In such cases the Alternative Method can be used where directional probes are made use of at least at 40 points and the Pitch and Yaw angles of stream lines are defined by these probes. At the present measurement the Alternative Method has been used to identify an arrangement pattern for the sampling points. Table 2 shows the geometric parameters for measurement sections of the exhaust stack. In this table the distance between section and downstream/upstream disturbances are shown according to EPA standard definition (Figure 2).

[1] Section diameter

Table 2: Geometric Parameters for Measurement Section of Exhaust Stack

	Diameter (mm)	A	B
Section 1	2010	2466	1927
Section 2	2170	408	2796
Section 3	3115	805	805

A: distance between section and downstream disturbance
B: distance between section and upstream disturbance

It is necessary for the stack flow angles to be defined in order to get exact definition of number and arrangement of sampling points. The approximate flow angles can be derived from numerical models. Referring to Figure 3, the return flows can be seen at sections 1 and 2 so the velocity direction will be negative at some points. But section 3 placed after the silencer, returned a more homogeneous pattern. Figure 4 shows the presence of swirling flow in addition to return flow at sections 1 and 2.

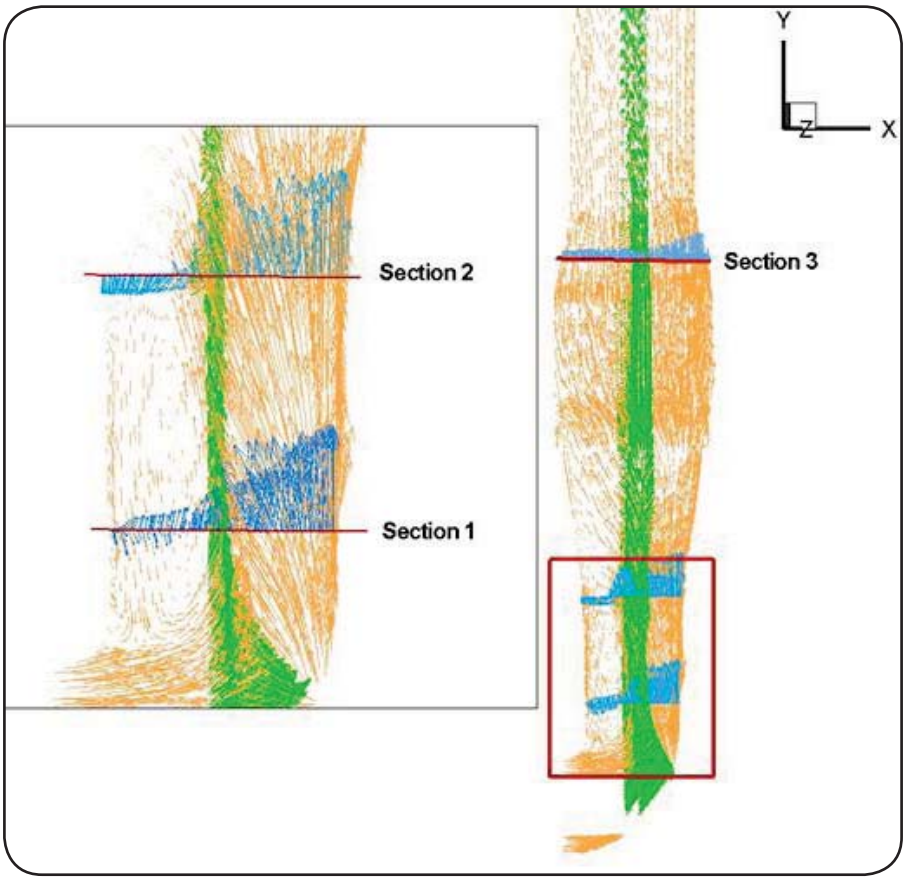

Figure 3: Exhaust Gas Stream Line Vector in Stack (X-Y Plane)

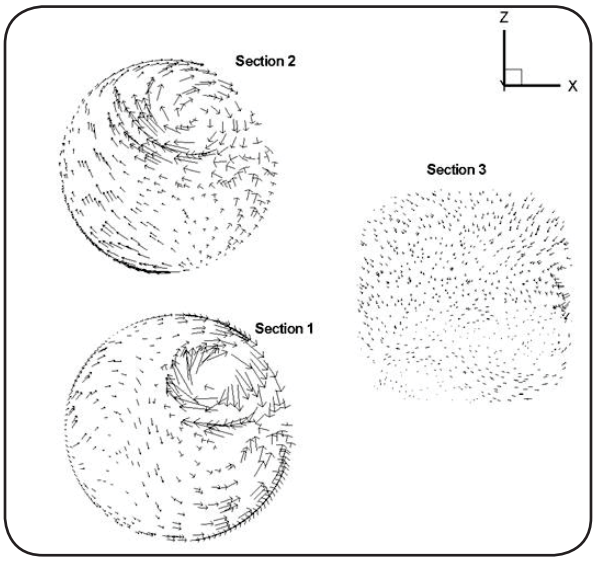

Figure 4: Exhaust Gas Stream Line Vector in Stack (X-Z Plane)

In order to define flow angles from the numerical models, the velocity component must be used. The Pitch angle is the main criterion of flow complexity. The pitch angle and other angles are shown in Figure 5.

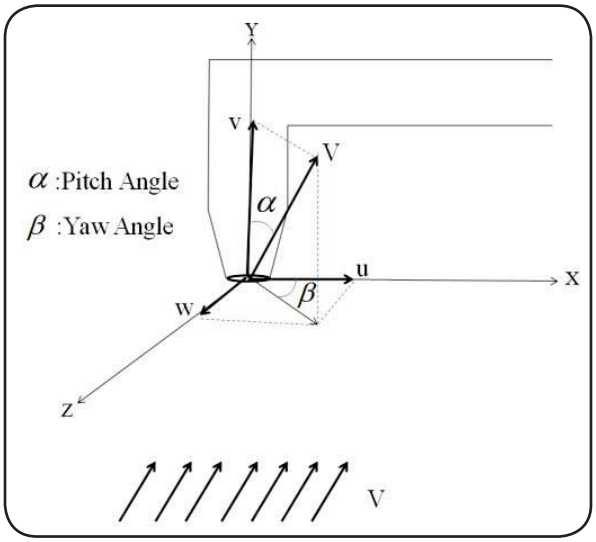

Figure 5: Pitch and Yaw Angles Definition of Exhaust Gas Flow

The calculated Pitch angles of exhaust flow are represented in Table 3 for all three measurement sections of the exhaust stack.

Table 3: Calculated Pitch Angles of Flow in Exhaust Stack

	Flow angles more than 20 ° (%)	**Flow angles more than 50 ° (%)**
Section 1	55	10
Section 2	40	8
Section 3	15	5

According to these flow angles and the minimum number of sampling points proposed in Alternative Method (40 points) and in order to get more accurate results, in the present work, 48 points are used for sampling at each section 2 and 3 (totally for two diameters at each section).

Positioning of sampling points on a circular section can be defined as a percentage of the distance between the point in question and the stack inner wall. It is necessary for stack wall effects to be considered in order to define the exact arrangement of sampling points. In the present work we calculated the wall correction factor through velocity measurements at least at three points on the boundary layer. Table 4 shows final number of sampling points for the three sections of the present measurement.

Table 4: Final number of sampling points for Measurement Sections of the Exhaust Stack

	Section 1	**Section 2**	**Section 3**
Number of Sampling Point for Each Diameter	29	29	32

Measurement Instrument

Complex pattern and high temperature of turbine exhaust gas flow cause a serious limitation in selection of measurement instrument. In this project the following instrument (Table 5) is to be used.

Table 5: Instrument for Turbine Exhaust Gas Measurement

Parameter to be Measured	**Measurement Instrument**
Gas Temperature	Thermocouple Type K with Sheath
Gas Total Pressure	5 Hole probe + Pressure Transmitter
Static Pressure	Wall Taps + Pressure Transmitter
Gas Velocity Vector	5 Hole probe + Pressure Transmitter
Gas Velocity Vector	High Temperature Hot wire

Data Sampling Method and System

According to exhaust stack flow condition and importance of provided data, a traverse system was used for sampling at sections 1 and 2 and a fixed grid of instruments at section 3.

The traverse system includes two main parts. The first is traverse arm carrying the measurement instruments and data transfer equipment and the second is drive mechanism including mechanical and electrical pieces of equipment.

The traverse arm needs to be moved into the hot gas and along the sampling points' path. As a main design challenge, the blockage effects of the arm on the exhaust gas stream should be considered in traverse arm design. The other challenge is the vibration problem of traverse arm which needs the frequency of Von-Karman vortex shedding formation and Strouhal Number to be determined. Some other analyses such as static deflection, stress analysis, thermal expansion and heat transfer of traverse system must be taken into account in the mechanical design of the traverse.

Traversing arm is made of two concentric tubes and the space between two pipes will be filled with insulation. The following figures represent the traverse system and its components.

In addition to the parameters extracted by the traverse system, the static pressures need to be covered. For measuring the static pressures some taps need to be mounted on the stack wall close to traverse system as shown in Figure 8.

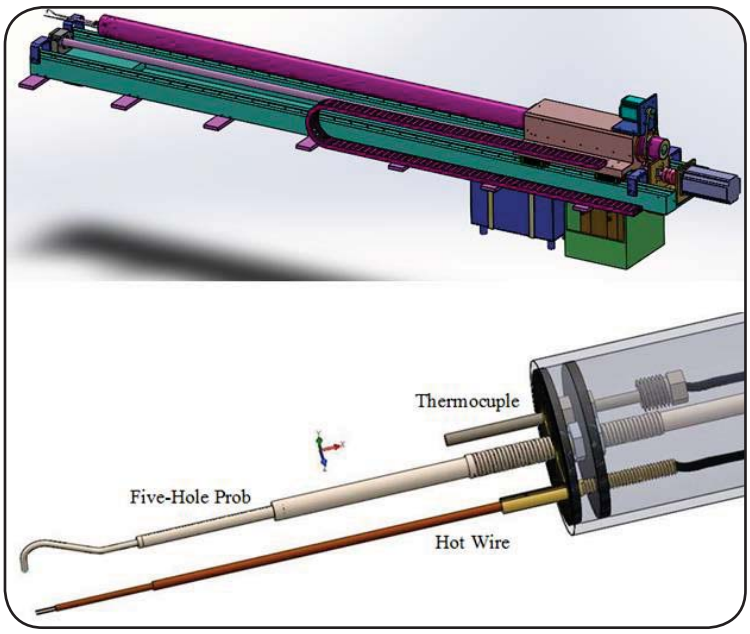

Figure 6: Traverse System for the Exhaust Gas Measurement

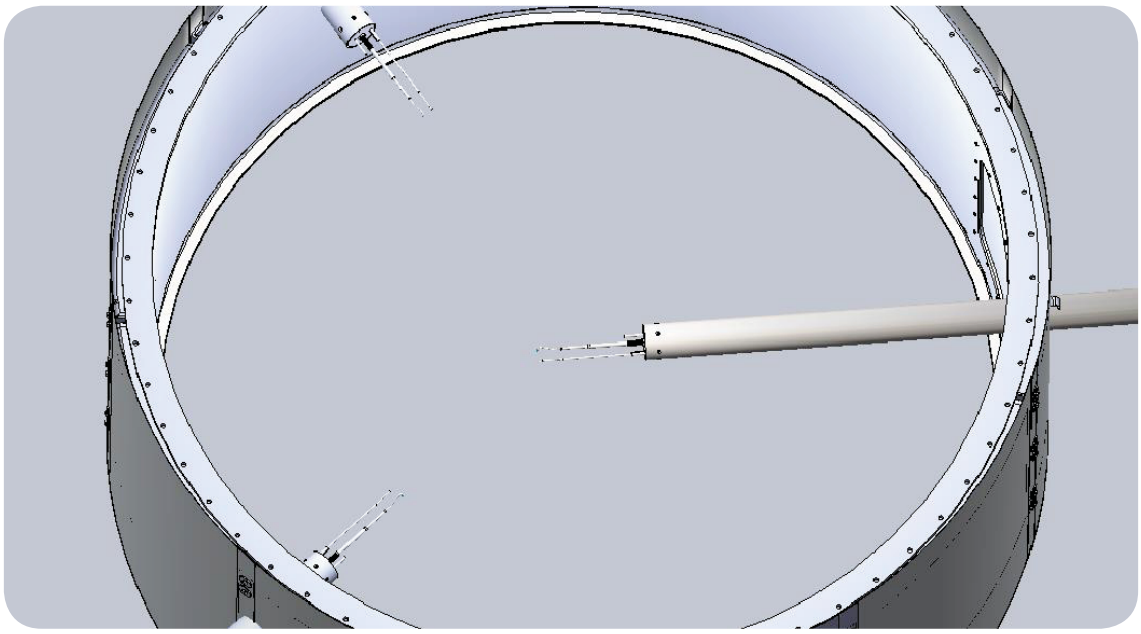

Figure 7: Schematic Illustration of Traverse System installation on Stack Wall

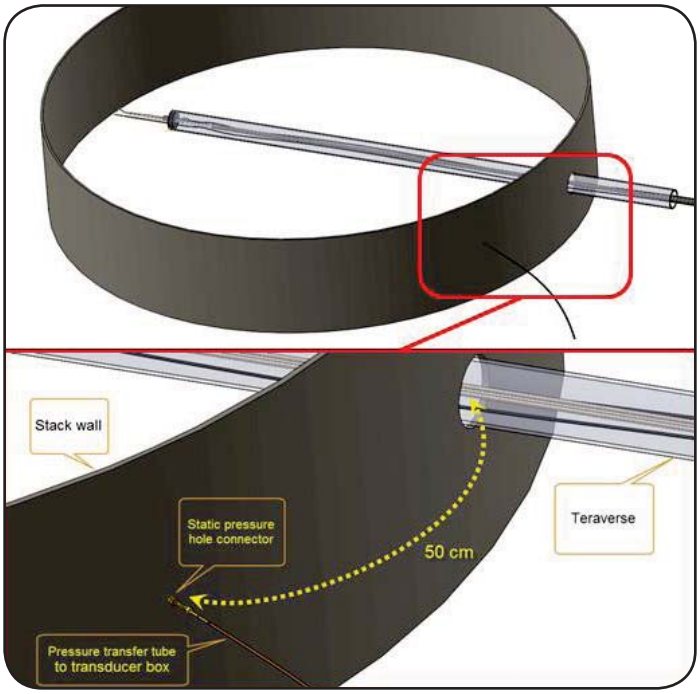

Figure 8: Illustration of Static Pressure Measurement Taps on Stack Wall

Conclusion

The present work outlines significant measurements on the gas stream of MGT-30 gas turbine in the exhaust stack. Some different issues were addressed:

- Selection of instruments and sensors for harsh measurement environment
- Using guidelines of standards related to stream measurements in gas stacks
- Exhaust stream properties by numerical analyses
- Designing a flexible data sampling system according to restrictions in the test stand

The experimental data will be later on used for verification and tuning of numerical models of exhaust gas stream and could also serve design optimization purposes in HRSG equipment.

Customized Design of Centrifugal Compressors and Launching Compressor Test Stand

Introduction

Centrifugal compressors have applications in various industries including but not limited to crude oil extraction, natural gas transmission/sweetening/refining and deriving hydrocarbon gases, as well as steel industries. There are a variety of sizes, forms and performance requirements for the compressors based upon the targeted application. Kicking off by a huge investment, MAPNA Turbine commenced playing its role in this sector since five years ago. It has ever since continually developed the knowledge and capability to go beyond manufacturing and set foot in customized design of centrifugal compressors taking into account specific customer requirements.

Based upon the in-demand models, aerodynamic analysis of compressors including Zero-D, 1D and 2D simulations were accomplished. The required packages to design mechanical elements of these machines such as casings, impellers, shafts, end covers and others were provided and a few prototypes covering specific customer demands were designed.

04

Given the current infrastructure and the so far developed engineering knowledge, the design limits for the service range of centrifugal compressors have been specified in table 1.

Table 1: Compressor service ranges

Quantity	Lower Limit	Upper Limit
Process Specifications		
Standard Discharge Flow, Sm3/min	60	80000
Discharge Pressure, bar absolute	N.A.	200
Pressure Ratio	>1	6
Machine Specifications		
Number of Stages	1	10
Speed, RPM	-	18000
Nominal Power Consumption, MW	0.2	30
Impeller/Diffuser Blade Geometry	2D 3-D	
End Seals	Dry Gas Seal / Mechanical Seal	
Bearing Type	Tilting Pad/Lobe Journal Bearings	
Process Gas Specifications		
Molecular Weight	16	39

To acquire the capability to manufacture these machines, MAPNA Turbine installed world-class general and specific-purpose hardware and launched local test bench for performance and mechanical running test of the manufactured/overhauled centrifugal compressors. The test bench is able to test centrifugal compressors following the requirements of PTC10 standard using Air as the fluid in a simulated power range of 100kW to 1MW which gives it the ability to cover the compressors used in natural gas transmission pipelines with a consumed power of up to 30MW.

This work provides an account of the developed packages to enable design and manufacturing of centrifugal compressors as integral pieces of equipment to oil and gas industry.

Zero-dimensional Approach

Once customer requirements are established and the input is provided, the first station in compressor design is conducting 0D design. The prerequisites for 0D design are the parameters identified when tender specifications are worked out. Given the customer's input, the very initial parameters like inlet and outlet gas properties (compressibility, enthalpy, entropy, heat capacity…) are calculated using gas state equations. Efficiency, impeller diameter, number of compressor stages, outlet temperature and rotational speed are also obtained out of trial and error iterations and eventually, the power consumption of the machine is calculated.

The performance curves can be later on generated based upon dimensionless impeller curves and performance calculations. These curves show a multitude of parameters like pressure ratio, outlet temperature, efficiency, polytropic head and power, for any given inlet or outlet gas pressure and mass/volume flow rate A sample is shown in figure 1.

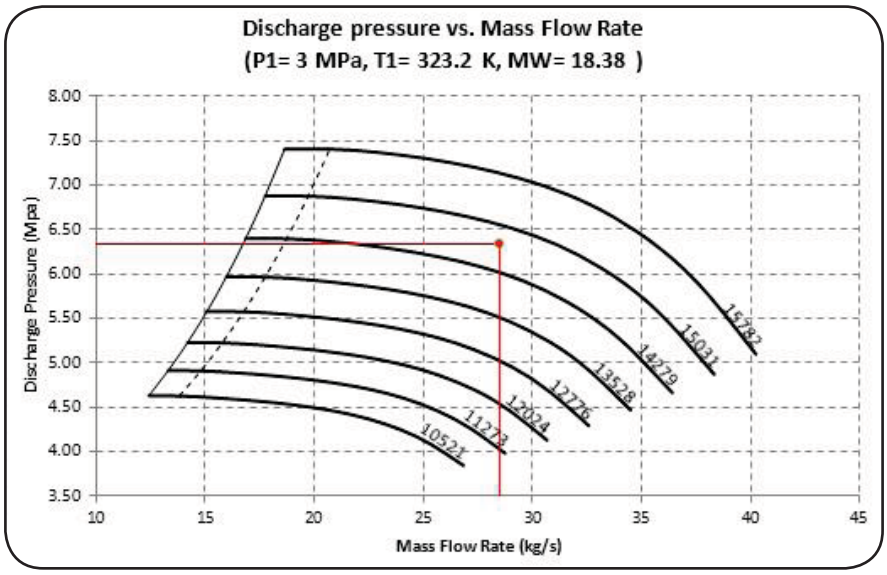

Figure 1: Compressor discharge pressure diagram

1D Approach

To predict the internal flow in compressors, MAPNA Turbine has adopted 1D flow analysis approach. In this method, using conservation equations, empirical coefficients, computational fluid dynamics coefficients and experimental loss coefficients taking boundary layer effects into account, the 1D flow analysis is accomplished along the flow mean plane.

For the bladed components like IGV (Inlet Guide Vane), impellers and bladed diffusers, CFD parameters are obtained at three flow path sections: inlet, outlet and middle. Boundary conditions at the inlet are thermodynamic properties and angular momentum and at the outlet they comprise outlet flow angle and total pressure loss of the flow. The analysis is based upon solving the system of continuity, momentum and energy equations. As some loss coefficients in the empirical models are functions of outlet conditions, the analysis has to occur following an iterative trial and error process. Binding to a given set of outlet boundary conditions, the step-by-step analysis is carried out until convergence is achieved between flow and outlet set of conditions. For non-bladed components, the equations are solved solely taking into account friction forces and other empirical coefficients. In other words, the system of equations is solved using a step-by-step formulated analysis.

1D analysis calculates the internal losses and accordingly estimates real compressor pressure ratio and efficiency making use of three groups of data: geometric data, working fluid data and velocity diagram data. In 1D analysis, the compressor performance is identified by two performance parameters like mass flow rate and impeller speed. Given the velocity triangle and the density at the inlet, conservation and momentum equations are solved using an iterative approach. As for the impeller duct, the velocity triangle is obtained using the formerly calculated parameters and initial conditions (e.g. inlet conditions). And eventually, using the calculated parameters at the inlet, velocity triangle and density at the outlet is obtained. The remaining performance parameters are all calculated using the so far obtained data.

Making use of the components of the velocity triangle, fluid properties at the outlet, required and wasted work and all internal loss factors for air compression are calculated. Other parameters and specific loss factors to the design point are iteratively derived. In figures 2 and 3, a comparison has been made between experimentally produced pressure ratios and efficiencies and those calculated using 1D approach.

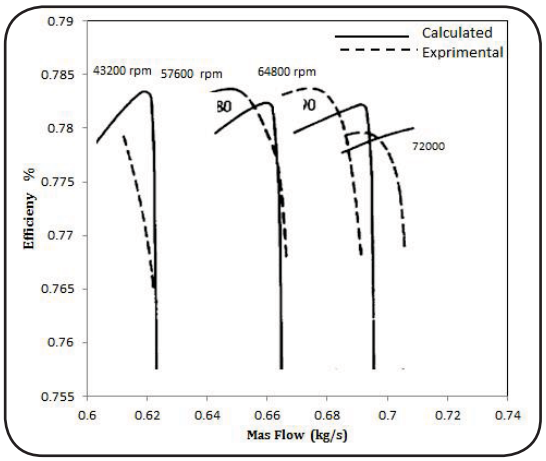

Figure 2: Efficiency curves

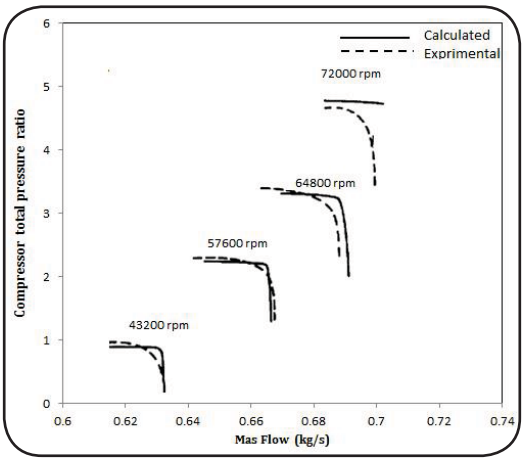

Figure 3: Pressure ratio curves

2D Approach

2D flow analysis is carried out using streamline curvature method. Fluid flow equations are solved in the area between two blades in the meridional plane. The objective is predicting the internal flow pattern and optimizing the compressor geometry. Solving the flow equations in these areas enables calculating velocity distribution along the blade and outlet angle of the flow. The lost work and impact waves' specifications are also obtained out of the 2D analysis. Boundary layer calculations can be used to estimate the losses and foresee the probable flow separations.

To separate Euler equations, CFD techniques are made use of. The governing differential equations in streamline curvature method are of elliptical type, which are difficult to solve; so, they are solved after mapping (m,θ) on to (x,y) coordinates system. In this method, the differential equations are solved through CFD techniques by generating the computational grid and separating the equations following hybrid methodology and other differential equation discretizing techniques, after converting the coordinates system.

Steady and non-viscid flow is the governing assumption. Maximum number of streamlines is assumed to be 9. One can increase this number, but 7 streamlines are enough to analyze the flow. This number has to be odd.

To analyze the flow in meridional plane the curvatures of hub and shroud have been taken into account. Like what was done in the blade to blade region, the flow is solved in this region through separation of equations and generation of the computational grid. Figure 4 shows a sample of such a solution in the meridional plane. Diffuser and return channel have been considered to come up with this flow solution. Figure 5 shows the pressure distribution on both suction and pressure sides of the blade.

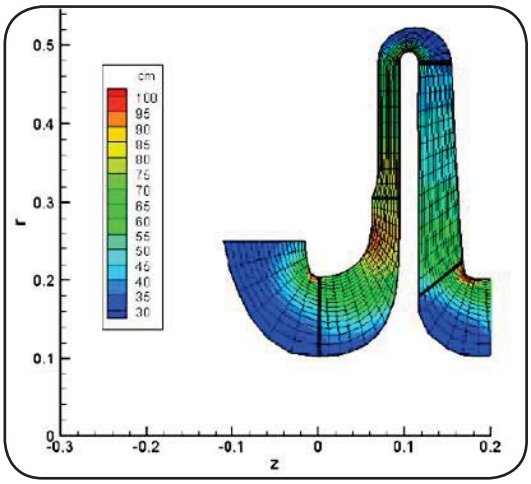

Figure 4: Flow solution in meridional plane

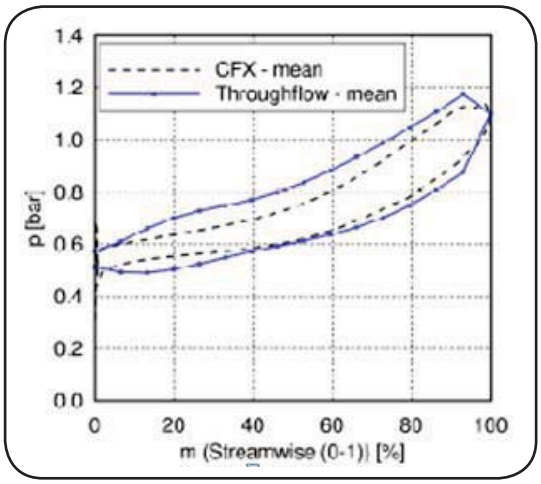

Figure 5: Pressure distribution on suction and pressure side of the blade

3D Approach

3D analysis is the last step when aerodynamically designing a compressor. At this stage, the results produced out of 1D and 2D analyses undergo a review and verification. Among the advantages the outstanding ones are its accuracy and its being economical to implement, compared to experimental methods.

A sample of a 3D analysis done on one stage of compressor will follow. This includes an impeller and the following volute in a centrifugal compressor. The general geometry is given in table 2.

Table 2: Impeller geometry

Number of Blades (mm)	Outlet diameter of the Volute (mm)	Outlet diameter of the Impeller (mm)	Inlet diameter of the Impeller (mm)
13	1000	600	240

In the present work, the boundary conditions include: inlet flow rate of 80 kg/s and outlet pressure of 72 Bar. The angular speed of the machine is assumed to be 900 Rad/s. Based on the already undertaken investigations, to acquire independency from the mesh density in the model, 2340000 mesh elements have been used to provide the numerical solution.

Figure 6 shows the streamlines and velocity/pressure contours within the investigated area. As expected, the fluid pressure has taken on its maximum at the outer diameter of the volute. In the same way, the maximum fluid velocity is observed at the outer edge of the impeller blades.

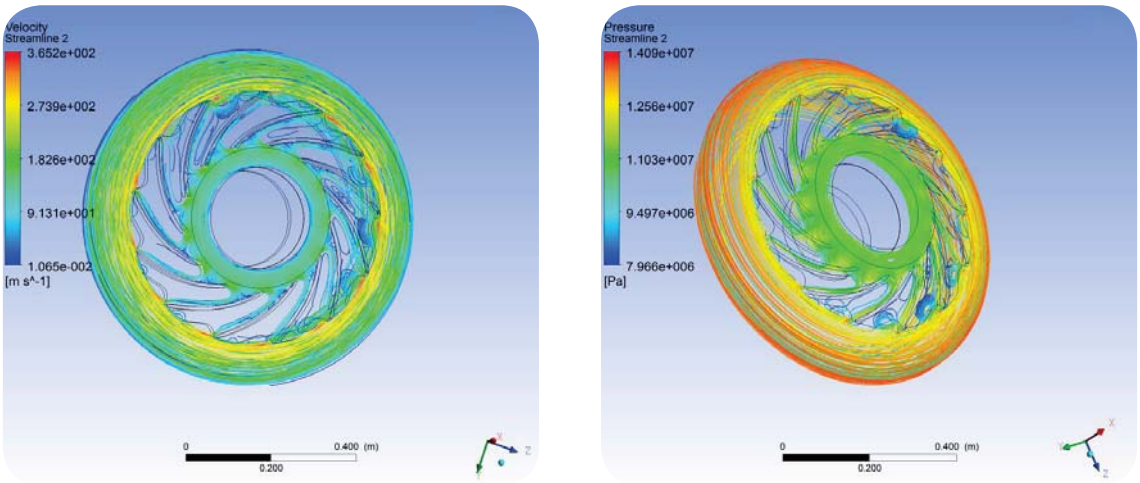

Figure 6: Streamlines and velocity/pressure contours

Mechanical Deign

Mechanical design of compressor elements entails analyzing all components taking into account static and dynamic loading, variation in dimensions due to temperature change, thermal stress, joint and assembly design, vibrations, etc.

When designing the internal parts of a barrel-type compressor, outer casing design is of utmost importance. And so it is when sour gas is the working fluid and corrosion crack resistance is considered.

At MAPNA Turbine, finite element simulation packages are used to analyze component stresses. A sample of the results of such analysis is shown in Figure 7.

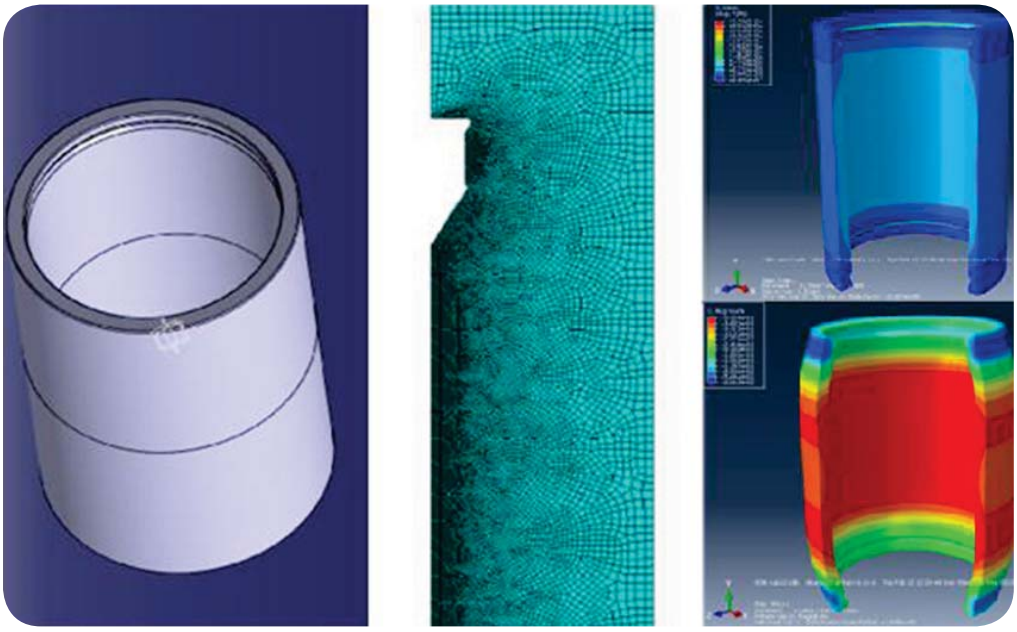

Figure 7: Casing analysis

In this analysis, the stress distribution on the casing under static forces has been calculated and the displacement thereof has been shown. Other components like end covers are analyzed in a similar way. Impellers, as the most important and the most sensitive of all internal components are dynamically analyzed.

An integral part of mechanical design is rotor-dynamics analysis. Unlike the design for other component classes, rotor-dynamics can be embarked on only after the design is accomplished for all rotating parts of the compressor as well as the shaft bearings and furthermore, coupling specifications and the driving machine characteristics are identified. So, one of the complications at this stage is inputting rather reliable information to conduct the vibrational analysis with a good degree of accuracy right at the early stages of design.

Manufacturers normally have a reliable forecast of the vibrational behavior of the system based upon their previous experience to help them design a stable system as far as practicable, but the fact is that without a complete shaft train analysis, an accurate investigation and a subsequent fine tuning is out of the question.

At MAPNA Turbine, local codes have been developed for torsional and lateral vibration analysis and a few commercial packages are also used to serve this purpose. A few codes have been also developed to simulate dynamic characteristics of various bearing systems. By using the tools, an estimation of the vibrational behavior of the system is provided and the required initial confidence in the design adequacy is obtained. After extraction of all required design parameters, the complete shaft train analysis is conducted based upon the details given in API617 and the results are accordingly presented to the customer.

Equipment

All required hardware to manufacture and test the compressors have been provided. As well, required laboratories for tests and verifications have been equipped, among which the following can be mentioned:

1) High speed milling machines
2) External/Internal grinding machine
3) EDM machine
4) Radial drilling machine
5) X-ray testing equipment
6) High speed balancing and over-speed facilities
7) Welding positioner and rotator
8) Welding machines
9) Pressure and flow test bench
10) Industrial compressor test bench
11) Calibration system
12) Coordinate measuring machine (CMM)
13) Profile projector
14) Multifunction data acquisition system
15) Spectrometer machine
16) Magnetic testing machine
17) Roughness measuring machine
18) Laser measuring machine

Centrifugal Compressor Test Bench

Performance and mechanical running test bench for centrifugal compressors was launched in 2014 at MAPNA Turbine. Numerous machines of various capacities have been successfully tested at the test bench. The test facility is equipped with a 2.5MW driving electromotor and a variable speed gearbox generating an output speed varying from 0 to 5000 rpm. Compressors of higher velocity are tested using an accessory step-up gearbox coupled to the output shaft of this main gearbox. The facility is also equipped with air intake and exhaust systems with varied dimensions to accommodate compressor inlet/outlet branch pipes of various diameters, silencers, flow rate control system, pressure/temperature/flow measuring instrumentation, integral lube oil system, remote control system, compressed air supplying system to feed the DGS system, and base plates to accommodate machines of different sizes.

All equipment pieces and inlet/outlet accessories as well as test instructions follow the requirements of PTC10. The test bench is also capable of generating compressor performance curves and identifying compressor Surge Point for tested machines.

The vibration measuring system is capable of measuring amplitude and phase of the detected vibrations with a time step of 0.5ms which enables it to be suitably used for compressors with speeds up to 20000 rpm. Figure 8 shows a view of the test bench.

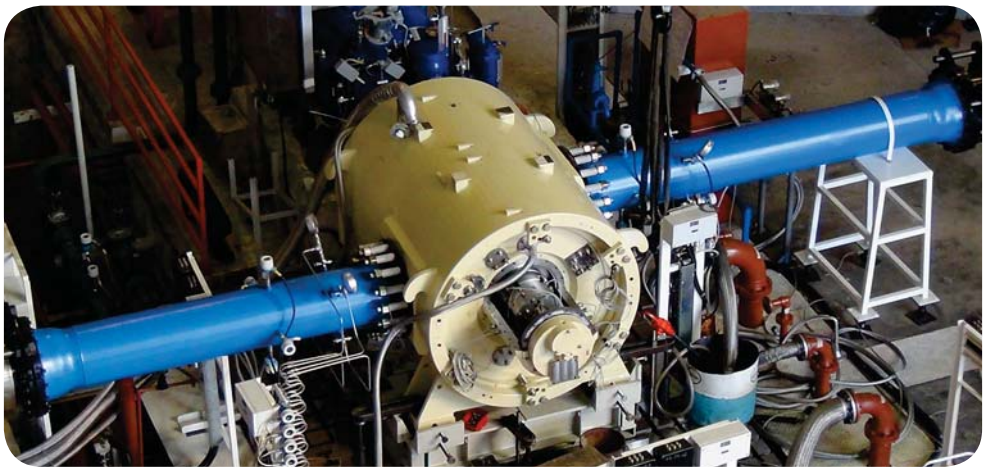

Figure 8: Compressor test bench

Rotor Balancing and Impeller Over-speed Test Equipment

MAPNA Turbine is also equipped with a high-speed equipment to balance rotors of centrifugal compressors as well as other turbo-machinery. Rotors with weights as high as 125 tones and service speeds as high as 10000 rpm, can be tested in vacuum. Figure 9 gives a view of the balancing facility. An over-speed test stand for compressor impellers is also installed to verify the mechanical strength of the impellers when rotated at 15% higher than their maximum intended service speed. A view of this test stand is shown in Figure 10.

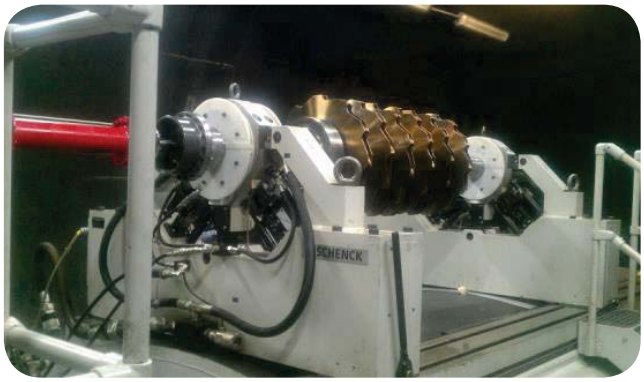

Figure 9: High/low speed balancing test

Figure 10: Impeller Over speed Test Stand

Conclusion

The knowledge required to design and manufacture centrifugal compressors has been a cquired and developed completely through huge investment on infra structures and engineering in the last 5 years. The result is reflected in the engineering tools, special hardware and design documentations practically available to conduct designs based on customer requirements.

As a consequence, MAPNA Turbine is now capable of manufacturing and testing a wide range of centrifugal compressors driven by electro-motors or gas turbines.The boom in Iranian oil and gas industry gives MAPNA Turbine a promising view in the coming years.

Boosting the Output Power through IGV+ Package

Introduction

MGT-70 machines make up a large portion of the gas turbine fleet being operated in Iran and all over the world. They are so attractive machines to invest in, when it comes to upgrading and maximizing power output and efficiency. It is thus considered as one of the major research and development endeavors at MAPNA Turbine, to put forth various upgrade and improvement packages for MGT-70 users.

Out of them, IGV+ power boosting package is one that takes the least possible time to implement and entails the fewest changes to cover to improve power output for these machines.

Approach

IGV+ package is based upon optimized adjustment of inlet guide vane angles in various versions of the machine. The immediate outcome of this upgrade is an increase in mass flow rate and subsequently a proportional increase in the power output. Increasing the air mass flow helps the operator enjoy two merits based upon their requirement and those of the grid:

1. Increasing the output power while maintaining the turbine inlet temperature
2. Longer life for hot section components subject to peak loads, in peak season through maintaining the turbine inlet temperature while still increasing the output by a higher mass flow rate

The governing logic on this upgrade package can be altered taking into account what is required by the customer to enjoy either of the merits outlined above. Opening rate for IGVs is controlled in a way that relative increase in mass flow and the subsequent power output is materialized all over the performance interval. It enables one to decrease the turbine inlet temperature at peak loads and thereby considerably increase the life expectancy for hot section components.

05

An integral part of the successful implementation this package is introducing modifications in the control system following the new positions for IGVs to improve the sought after flexibility against various environmentally inflicted conditions such as increases in ambient temperature. To achieve this, compressor performance curves in different versions of the machine were scrutinized and the required changes in the control system and the logic were introduced.

Given the design characteristics of the compressor, there are various aspects to cover when implementing this improvement. Determining the optimal opening angle of the IGVs taking into account compressor and turbine efficiency and the increased increment of mass flow rate, is one of the main challenges of the implementation.

Figure 1 schematically shows the changes in opening angle of the IGVs, the subsequent relative increase in mass flows and the impact on compressor efficiency.

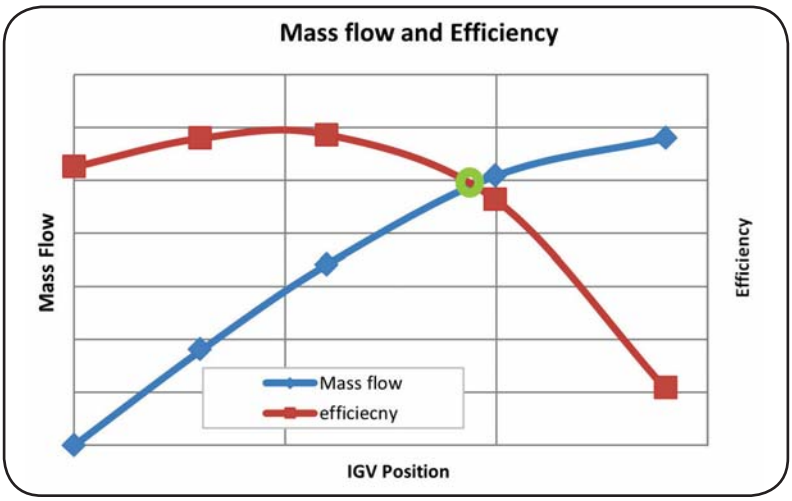

Figure 1: Scheme of compressor mass flow and efficiency versus changes in IGV position

Given the variation in compressor design in different versions of MGT-70, machine and its predecessors implementing this package produces different outcomes in terms of power output and efficiency increase for different versions of the machine. Gets a higher increase in power output compared with version 5. In table 1 a comparison is provided as for the power output increase in different versions of the machine.

Another important outcome of this package is increasing the overall output of combined cycle power stations due to the increase in turbine outlet mass flow rate and its subsequent effect in increasing the boiler capacity to generate hot steam for steam turbines. The quantities indicating the effects of this upgrade on increasing combined cycle power output are shown in table 1.

Table 1: Power output increase in different versions of MGT-70 family as a result of implementing IGV+

	V94.2 (V3)	V94.2 (V5)	MGT-70(1)	MGT-70(2)
Simple Cycle (MW)	6	3.9	3.9	3.9
Combined Cycle (MW)	9	5.85	5.85	5.85

IGV+ is an upgrade package with extremely high potentials to improve power output in the national grid of Iran. A power output increase as high as 1000MW can occur as a result of the implementation. Table 2 provides an account of the expected power output increase for different versions of this machine in the country's fleet.

Table 2: Power output increase (MW) in the grid as a result of implementing IGV+

	V94.2 (V3)	V94.2 (V5)	MGT-70 (1)	MGT-70 (2)	Total (MW)
Working in Simple Cycle	40	65	2	2	
Working in Combined Cycle	46	18	4	0	
Total Machines	86	83	6	2	
Total MW Increase After IGV+	**654**	**358.8**	**31.2**	**7.8**	**1051.8**

Verification

The package was temporarily implemented on two units in two power plants: ParehSar Power Plant (Unit 3, version 5 V94.2 machine) and Yazd Power Plant (unit 1, version 3 V94.2 machine). The practically produced results successfully verified the expected outcome of the package. The results show output increases of about 4MW and 6MW for version 5 and version 3 machines respectively (Figure 3 & 2).

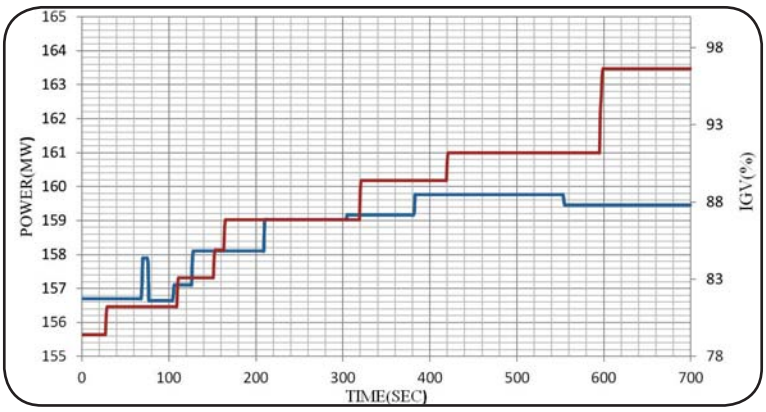

Figure 2: Results of the implementation on Unit 2 of ParehSar (V94.2 version 5 machine)

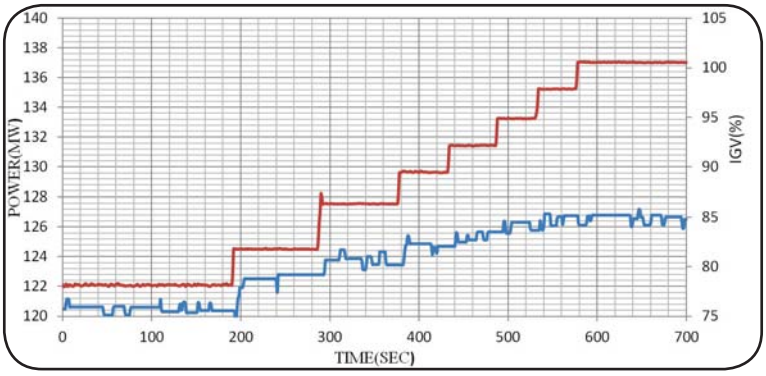

Figure 3: Results of the implementation on Unit 3 of Yazd (V94.2 version 3 machine)

Conclusion

To sum up, the achievement out of implementing IGV+ on MGT-70 machines is as follows:

- Approximately 6MW power output increase at base load for V94.2 version 3
- Approximately 4MW power output increase at base load for V94.2 version 5
- The ability to both increase output in summer peak and decrease turbine inlet temperature for life expectancy improvements out of the peak season
- An slight increase in machine efficiency as a bonus

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