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Vibration Analysis of Gas Turbine SIEMENS 162MW - V94.2 Related to Iran Power Plant Industry in Fars Province

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***Abstract*—**Vibration analysis of most critical equipment is considered as one of the most challenging activities in preventive maintenance. Utilities are heart of the process in big industrial plants like petrochemical zones. Vibration analysis methods and condition monitoring systems of these kinds of equipments are developed too much in recent years. On the other hand, there are too much operation factors like inlet and outlet pressures and temperatures that should be monitored. In this paper, some of the most effective concepts and techniques related to gas turbine vibration analysis are discussed. In addition, a gas turbine SIEMENS 162MW - V94.2 vibration case history related to Iran power industry in Fars province is explained. Vibration monitoring system and machinery technical specification are introduced. Besides, absolute and relative vibration trends, turbine and compressor orbits, Fast Fourier transform (FFT) in absolute vibrations, vibration modal analysis, turbine and compressor start up and shut down conditions, bode diagrams for relative vibrations, Nyquist diagrams and waterfall or three-dimensional FFT diagrams in startup and trip conditions are discussed with relative graphs. Furthermore, Split Resonance in gas turbines is discussed in details. Moreover, some updated vibration monitoring system, blade manufacturing technique and modern damping mechanism are discussed in this paper.

***Keywords*—**Gas turbine, turbine compressor, vibration data collector, utility, condition monitoring, non-contact probe, Relative Vibration, Absolute Vibration, Split Resonance, Time Wave Form (TWF), Fast Fourier transform (FFT).

I. Introduction

THERE are two set of probes in gas turbines. The shaft relative vibrations (micrometer peak to peak) are measured by none contact probes. The condition monitoring systems of these kinds of probes are usually Bently Nevada. Besides, Absolute Vibration (mm/s RMS) is measured by contact probes. Both systems are equipped with alert and danger facilities in process main board sub stations.

The gas turbine is tripped in danger condition. The parallel condition monitoring system can reduce the risks like probe installation mistakes. System reliability is increased by this method.

The condition monitoring (CM) group data collectors like Easy viber and Vibro 60 are connected with these kinds of board facilities. Data collector software like spectra pro and XMS options are adjusted for gas turbine specification properly. Vibration data and trends like time wave form (TWF), fast Fourier transform (FFT) and phase values are measured. These kinds of data support the gas turbine vibration analysis. The regular and close monitoring helps the CM group to have better evaluations and make more efficient decisions. Relative vibration trends and values are measured by none contacted Eddy current probes. The basic principles of Eddy current probes are discussed in [1].

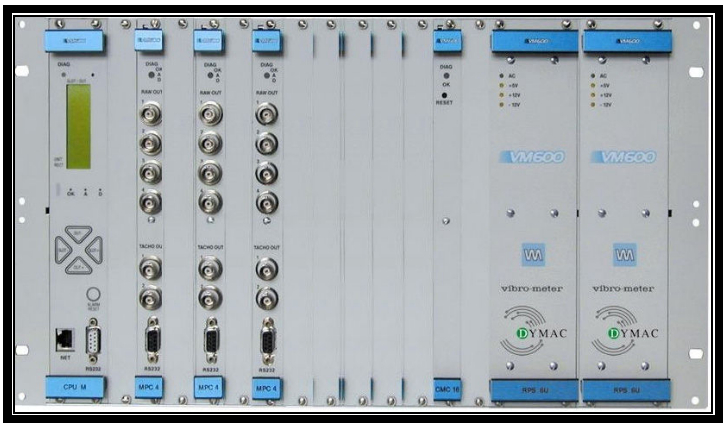

Fig. 1 None contact probe connection main board in substation

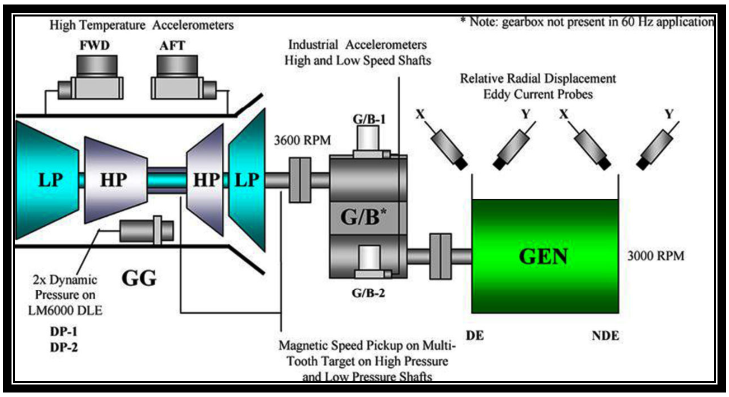

Fig. 2 Typical relative vibration monitoring system

Gas turbines vibration monitoring systems are usually equipped with some contact piezoelectric probes. Absolute vibrations in mm/s RMS are measured. The absolute and relative vibration data are compared. Fake trips are distinguished easily by this method. Probe installation mistakes are usually caused such fake trips. CA303 and CE136 Accelerometer are shown in Fig 4.

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Gas turbine systems are usually equipped with some vibration indicators. These indicators are usually installed in main board substations. The absolute and relative vibration data and trends are presented [2]. Besides, the basic principles of piezoelectric probes are discussed in [3].

Fig. 3 Typical Eddy Current Probe System (CMSS 68)

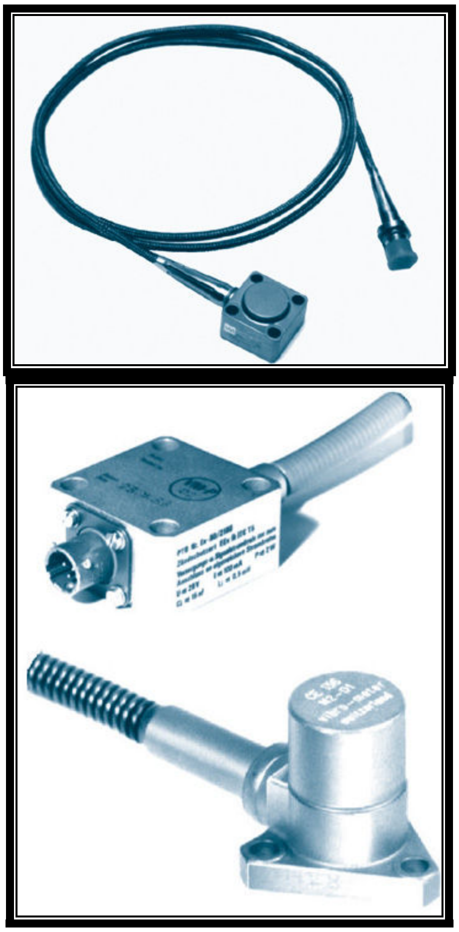

Fig. 4 CA303 and CE136 Accelerometer

Misalignment and unbalance is the most cause of machine vibration. An unbalanced rotor always cause more vibration and generates excessive force in the bearing area and reduces the life of the machine. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs and increasing machine uptime. Misalignment and unbalance have unique characteristics in FFT, TWF or phase behavior of machine. Vibration analysis could predict unexpected shut down in industrial plants [4].

Some successful vibration analysis case reports discussed in [5]-[8]. Besides, field balance of gas turbine utilities discussed in [9]. In addition, the shaft crack is one of the most challenging concepts in most critical equipment fault diagnosis. The shaft crack characteristics are unique. The guidelines are identified in different ways. This fault is distinguished with different techniques. Moreover, some parametric study has been conducted to discuss the effect of the crack location and material gradient on both the natural frequencies and the corresponding mode shapes [10].

The journal bearing wear is considered as an important factor in preventive maintenance. This factor is affected by lubrication. The oil sample locations are identified by CM engineers. Nowadays new lubricants are introduced to lubrication world. An environmental friendly palm-grease has already been formulated from modified RBDPO (Refined Bleach Deodorized Palm Oil) as base oil and lithium soap as thickener. Such palm-grease is dedicated for general application and equipment working in different industries (such as oil and petrochemical industries). This type of lubricants are improved the machine performance. Besides, these lubricants are usually used for high speed machines. Tribology performance, especially the anti-wear property of this lubricant are considerably improved [11].

The signal processing methods are developed too much in recent years. One of the alternative statistical analyses is known as I-kaz Multi Level method. This method was originally developed base on I-kazTM but with higher order of signal decomposition. The new I-kaz Multi Level method was proven very sensitive and detects very well in amplitude and frequency changes of a measured signal. Nowadays high speed machines TWF and FFT are achieved with more resolutions [12].

Robust real time surveillance and secure system, for critical oil pipeline infrastructures, with combination of conventional network and wireless sensor network along with microwave network shows significant improvement with eleven times more efficient to conventional systems by reducing leakage and loss reporting time to control center. This system will be more efficient to detect any threats in real time and can report to central control room without any further delay. These techniques are recently developed for gas turbine condition monitoring specially in remote areas [13].

Different maintenance strategies such as corrective, time based, preventive, condition-based and predictive maintenance are used for different equipment. New fuzzy multi criteria model is introduced and it is used for the optimization decision making of the complex systems. Maintenance strategies have been modeled with consideration of several fuzzy parameters. Different machine parts with respect to introduced criteria may need different maintenance policies. Suitable maintenance policies can be selected [14].

Several modern vibration monitoring systems are recently developed in USA for gas turbines. The vibration monitoring system acquires vibration data from an engine and processes the data with advanced algorithms to determine engine component health, both in a diagnostic and prognostic fashion. The method includes the steps of measuring an operating parameter and a corresponding set of vibration amplitudes for a plurality of rotating components during a period of operation and normalizing the set of measured vibration amplitudes based on established amplitude limits.

According to one embodiment, the method comprises, receiving engine data from the turbine engine while in service, where the engine data include vibration data measured by one or more sensors disposed on the turbine engine. The method further comprises receiving user input through a user interface, processing the vibration data in response to the user input and displaying the processed vibration data through the user interface.

The processed data being displayed is a function of a time parameter. According to another embodiment, the method comprises, receiving engine data from the plurality of turbine engines while in service, where the engine data include vibration data measured by a plurality of sensors disposed on the turbine engines. The method further includes receiving user input through a user interface, processing the vibration data in response to the user input and displaying the processed vibration data through the user interface. The processed data being displayed as a function of a time parameter associated with at least one of the turbine engines.

According to still another embodiment, the system comprises a general data module configured to receive periodic data from a controller of a turbine engine. The periodic data representing operational states of the turbine engine, a vibration data module configured to receive vibration data from a measurement module associated with the turbine engine and generate a functional relationship between the vibration data and a time parameter according to user input.

The vibration data including information about vibration of the turbine engine provided by a plurality of sensors associated with the turbine engine. The database configured to store periodic and vibration data. This method provides the historical data in response to the user input. Historical data module configured to retrieve the periodic and vibration data from the database. Some display device configured to display periodic, vibration, and historical data [15].

II. Experimental Details

The case history about vibration and operation behavior of gas turbine SIEMENS 162MW - V94.2 Related to Iran Power Plant Industry in Fars Province is explained in this part. Gas turbine technical specifications are briefly discussed. Gas Turbine Features are represented in Fig. 5.

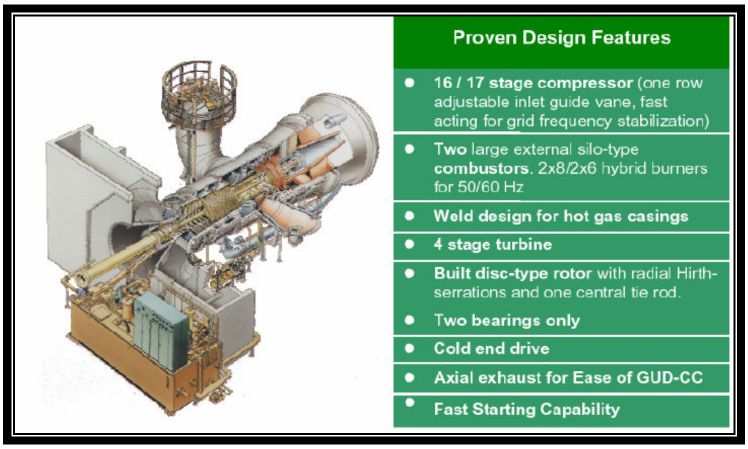

Fig. 5 SIEMENS V94.2 Gas Turbine Features

Gas Turbine Combustion chamber features are represented in Fig 6.

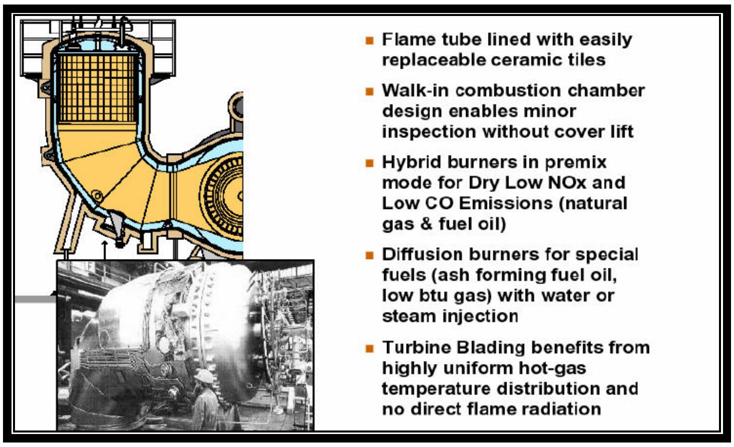

Fig. 6 V94.2 Gas Turbine Combustion chamber features

The blading system is SI3D. The blades are described as vane #1 and vane #2.

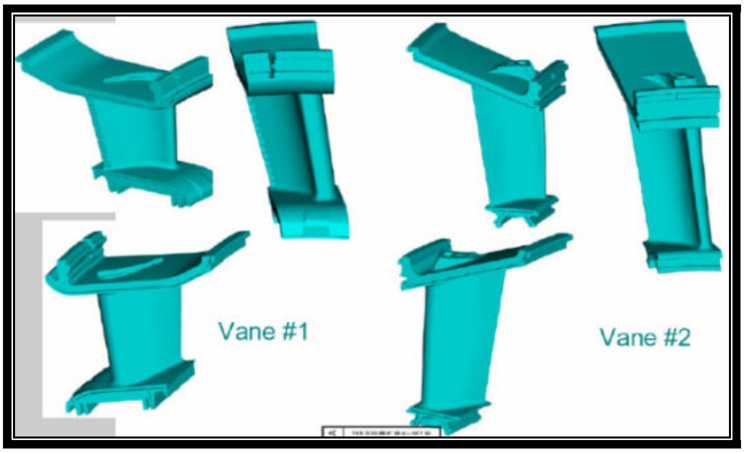

Fig. 7 Some SI3D Blading Samples

Main rotor and typical blades related to gas turbine SIEMENS162MW - V94.2 are shown in Fig 8. In addition, blades design features plot is shown in Fig 9.

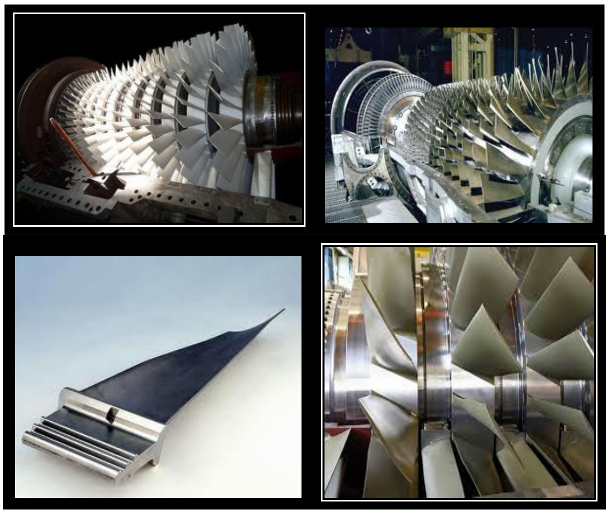

Fig. 8 Typical blades related to gas turbine SIEMENS162MW - V94.2

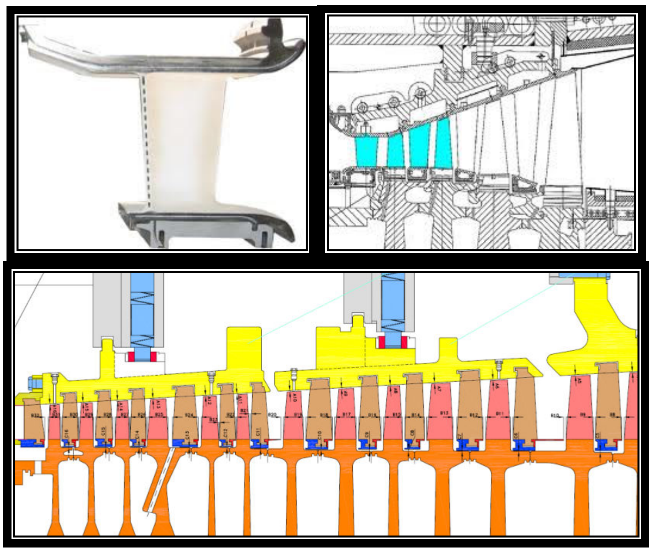

Fig. 9 Blades design features plot

Employing wet compression system is available in market since 2003.

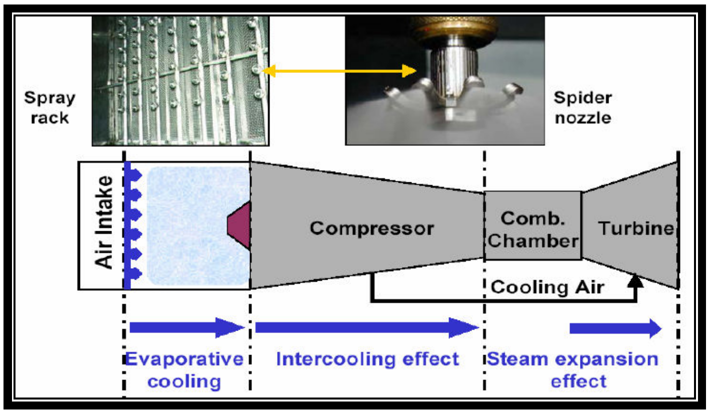

Fig. 10 Wet compression process

Machinery features and basic principles related to gas turbine SIEMENS162MW - V94.2 are determined. Vibration analysis will be discussed in details in the next stage. Before, it is worthy to discuss some new and interesting patent and innovation in gas turbine world [16].

Turbine blade for industrial gas turbine is used which includes a blade substrate formed of a single-crystal heat-resistant alloy containing C: 0.06 to 0.08%, B: 0.016 to 0.035%, Hf: 0.2 to 0.3%, Cr: 6.9 to 7.3%, Mo: 0.7 to 1.0%, W: 7.0 to 9.0%, Re: 1.2 to 1.6%, Ta: 8.5 to 9.5%, Nb: 0.6 to 1.0%, Al: 4.9 to 5.2%, Co: 0.8 to 1.2%, and the remainder substantially consisting of Ni with reference to mass, and includes a diffusion barrier layer, a metal layer, a bond coat, and a top coat, these layers and coats being stacked in this order on a surface of the blade substrate, the metal layer having a thickness of 5 to 30 micrometer.

Therefore, the turbine blade can be provided which has a thermal barrier coating formed without loss of a function of the diffusion barrier layer [15].

Damper system recently developed in gas turbine world. The damper includes a width dimension, a height dimension, and a length dimension, and a forward plate and an automatic fine tuning (AFT) plate. The AFT plate is larger than the forward plate along the width and height dimension and includes an upper portion extending in the height dimension, the upper portion having a non-symmetric configuration. The damper further includes a longitudinal structure extending in the length dimension and connecting the forward plate and the AFT plate.

The damper includes a width dimension, a height dimension, and a length dimension, and a forward plate. The damper further includes an AFT plate including a larger area than the forward plate along the width and height dimension, an upper portion having an upper point that is offset with respect to a central axis of the AFT plate extending in the height dimension, and a rectangular-shaped discourager extending AFT in the length dimension from the AFT plate. The damper also includes a longitudinal structure extending in the length dimension and connecting the forward plate and the AFT plate.

The turbine rotor assembly includes a turbine rotor having a plurality of turbine blade slots, and a plurality of turbine blades having an airfoil, a platform, and a root structure, the root structure of each turbine blade shaped to be received in a corresponding turbine blade slot of the turbine rotor.

The turbine rotor assembly also includes an under-platform gap formed adjacent and below the platforms of adjacent turbine blades, and an under-platform cavity formed between an outer radial surface of the rotor and adjacent turbine blade root structures, and below adjacent turbine blade platforms.

The turbine rotor assembly further includes a turbine damper located within at least one of the under-platform cavities, the turbine damper including a width dimension, a height dimension, and a length dimension. The damper further includes a forward plate sized to provide a forward flow gap into the under platform cavity and the under-platform gap, and an AFT plate sized to cover a portion of the under platform cavity and a portion of the under-platform gap [15].

Gas-turbine engines used in transportation, energy, and defense sectors rely on high-temperature thermal-barrier coatings (TBCs) for improved efficiencies and power. The promise of still higher efficiencies and other benefits is driving TBCs research and development worldwide. An introduction to TBCs-complex, multi-layer evolving systems are presented, where these fascinating systems touch on several known phenomena in materials science and engineering [17].

Facilitates removal of the patent pending lateral generator from between the turbines for maintenance or repair is already investigated by several gas turbine manufacturers [18].

Energy efficiency is one of the main objectives for the development of new power plant technology in order to reduce fuel consumption and emissions. In response to the increasing, worldwide need for reliable, low cost and environmentally compatible generation. New Siemens SGT5-8000H Gas Turbine is recently introduced by Siemens Company.

The SGT5-8000H will have a net power output of at least 340 MW and will be optimized for the combined cycle process with a net power output of more than 530 MW. A major benefit for the customer is the high efficiency of 60%. Efficiency not only plays an important role with respect to environmental aspects, but also for the profitability of the power plant.

As fuel is the largest single cost item for running a power plant, an increase of two percentage points can save the operator millions of Euros over the entire life cycle of a combined cycle power plant with a capacity of 530 MW [19].

III. Results and Discussion

In this part a case history related to Gas Turbine SIEMENS V94.2 is explained. This gas turbine is related to Iran Power Plant Industry in Fars Province and is in main operation board high vibration.

A. Case History

Duration: Tuesday, April 24, 2012 to Friday, May 4, 2012.

After facing some problems in gas turbine Fars utility startup period, the gas turbine was tripped in 1000 RPM. The trip was because of high amounts of vibrations. The vibration groups were take part a session for analyzing the balancing condition of rotor and gas turbine foundation.

Both the balancing and foundation maintenance data was in good conditions then vibration groups decided to close monitoring the gas turbine. Further investigations in maintenance history showed that gas turbine middle shaft was falling during machinery maintenance actions and installed after repair in machinery workshop. There was a strong hypothesis that this may the main root of all vibration problems in this machine. The vibration analysis of hypothesis is discussed.

Besides, other possibilities are mentioned. First, the electronic group checked the absolute and relative conditioners to adjust the related options with technical documents. Then the analyzer group readjusts all the adaptation numbers in turning gear condition with technical document. After that, all the relative vibrations in turbine, compressor and generator were monitored in 600 RPM in the next stage.

The operation condition were not agree to continue increasing RPM then in the next day the gas turbine first start up to 600 RPM then process were decided to reduce the RPM up to 450 and after one hour they were increased the RPM little by little up to 990 RPM gradually. In this stage Gas turbine trip occurred because of high amount of absolute vibrations. As we checked all the trend of vibration monitoring data it was clear that the gas turbine did not have any problem up to 750 RPM.

In addition, there was an acceptable adaptation between all monitoring systems in process board and CM vibration trends in both absolute and relative monitoring systems. Unfortunately, after 750 RPM the related vibrations were increasing gradually. In addition, the absolute vibrations were increased dramatically after 940 RPM but relative vibrations were decreased slightly.

Absolute vibrations were increased and relative vibrations were decreased simultaneously. This increasing in absolute vibration in turbine side was 17.5mm/s RMS that caused vibration trip in 990 RPM.

The absolute and relative vibrations were decreased slightly in the same manner in trip period. Furthermore, All journals temperature monitoring trends were normal and in range during this period. The decreasing relative vibration with increasing in absolute vibration was never seen in other gas turbines by the vibration groups and it seems to be a new vibration behavior!

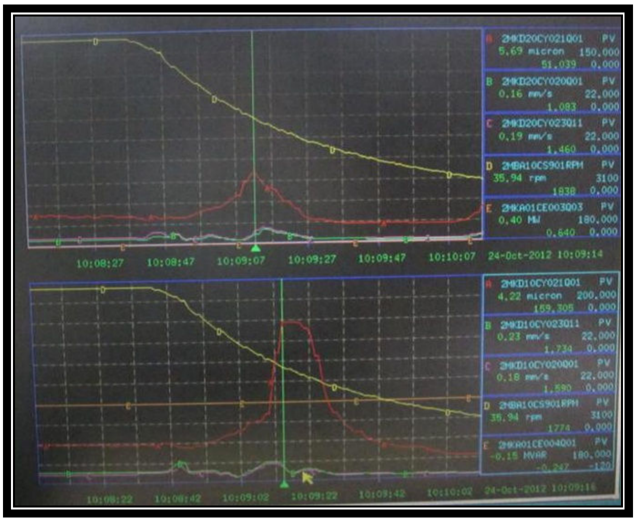

Fig. 11 Absolute and relative vibration trends from control monitoring system process main board

Because of low frequency condition, the absolute sensors cannot help in vibration analysis but there is some valuable information in relative sensors data that called slow roll in CM texts and will help us in shaft centerline analysis.

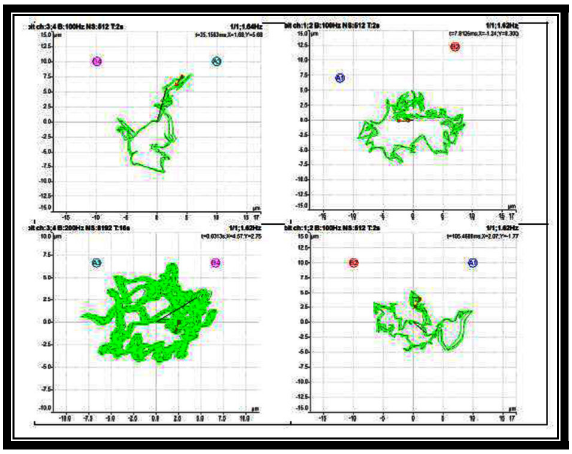

Fig. 12 Turbine and compressor orbit OCE and CE in turning gear condition

TABLE I
SHAFT POSITION BY 9-VOLT INITIAL INSTALLATION

Y	X
-40	-75
-45	115

Vibrations in 750 RPM should be focused because of the higher amplitudes. Besides, FFT Absolute vibrations turbine side in 600 RPM (horizontal, vertical and axial) is shown in Fig 13.

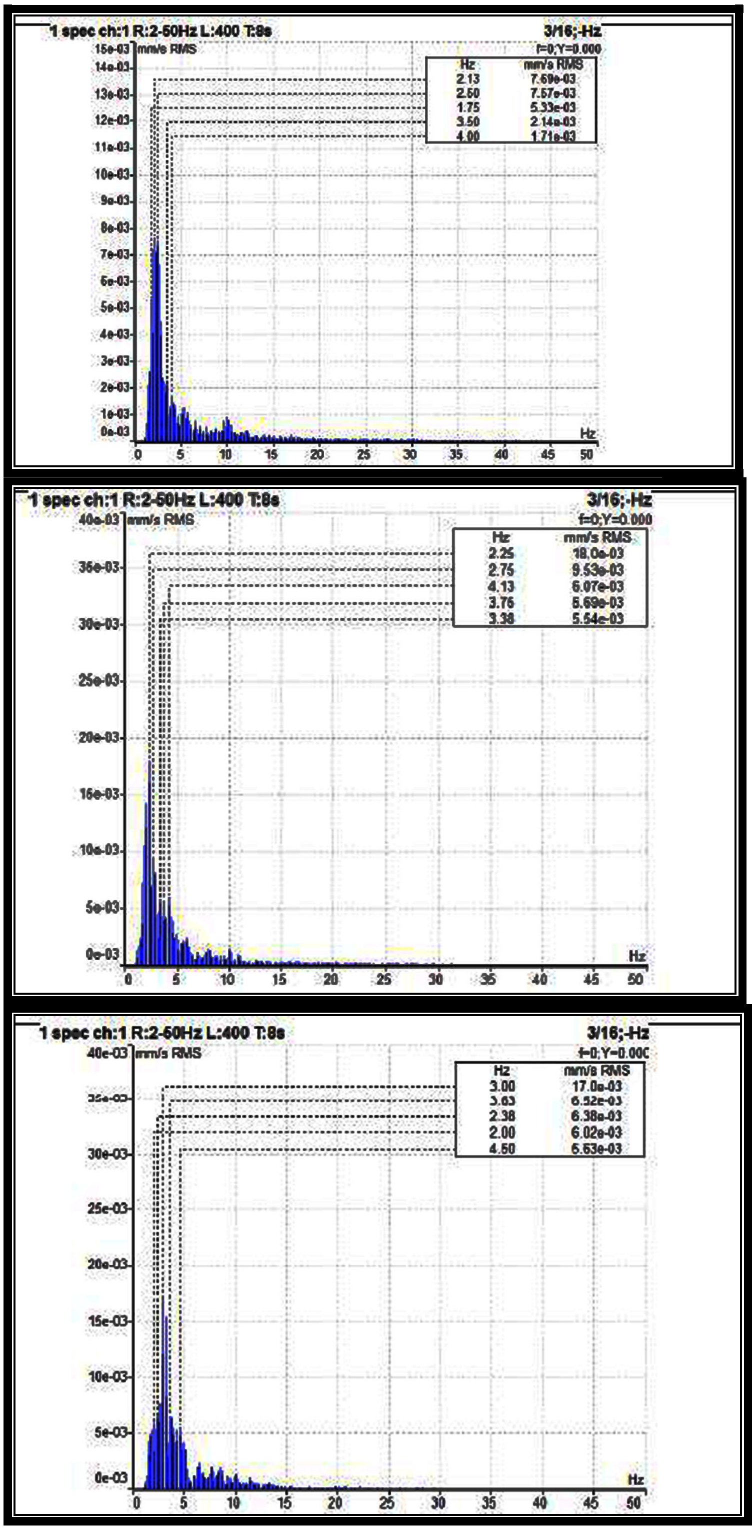

Fig. 13 FFT Absolute vibrations turbine side in 600 RPM (horizontal, vertical and axial)

750 RPM approximately is beginning of the first gas turbine SIEMENS162MW - V94.2 critical speed and the vibration data in this condition can help too much for the gas turbine vibration analysis.

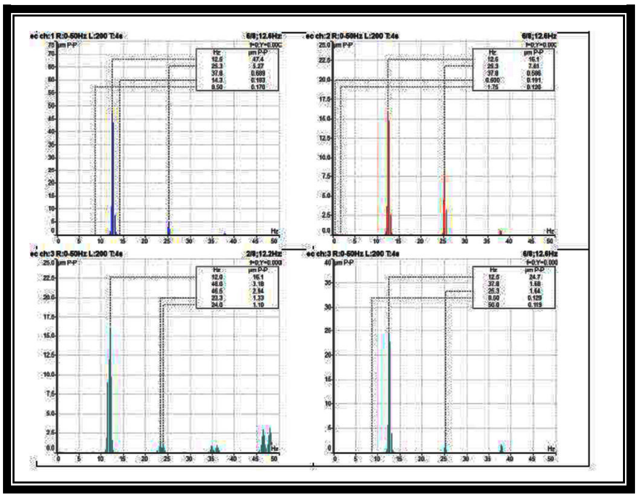

Fig. 14 Relative vibrations FFT turbine 1, turbine 2, compressor and OCE in 750 RPM

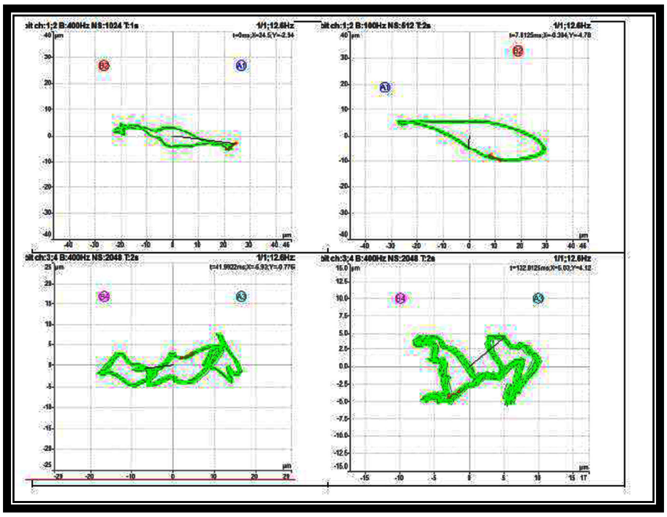

Fig. 15 Orbits turbine, compressor, OCE and CE in 750 RPM

Vibration modal analysis 1X (left is related to turbine side) is shown in Fig 16. Vibration modal analysis is obtained by using the phase values and their trends in different locations (points).

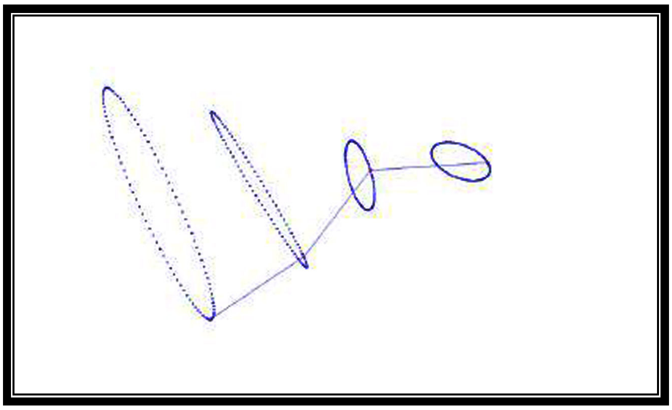

Fig. 16 Vibration modal analysis 1X (left is related to turbine side)

Vibration monitoring system in operation main board showed that the relative vibration decreased suddenly when absolute vibration caused gas turbine trip. In addition, CM vibration graphs indicated the same information.

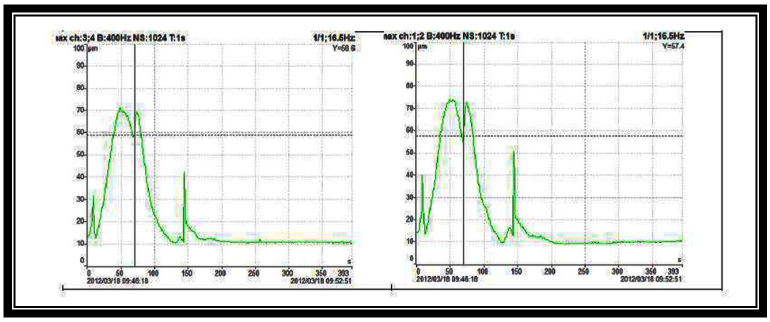

Fig. 17 Turbine and compressor start up and shut down condition (trip position are indicated by cursor)

The reason of this phenomena could be discovered by drawing bode diagrams for relative vibration sensors in both turbine and compressor.

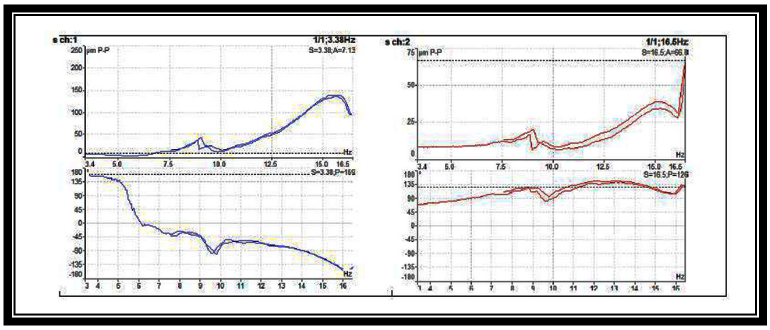

Fig. 18 Turbine relative vibrations Bode diagrams

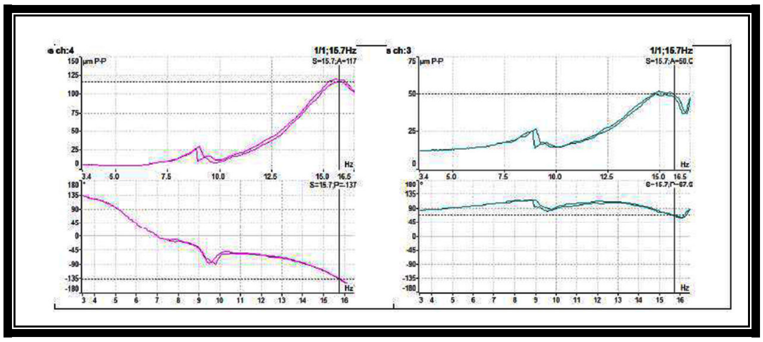

Fig. 19 Compressor relative vibrations Bode diagrams

As shown in above bode diagrams, overall amplitude vibration was decreased in one sensor and was increased in other sensor simultaneously. This increasing behavior is related to a turbine side mode of vibration for 940 RPM in 30˚ sensor but this mode of vibration was disappeared and was replaced by another mode of vibration in 90˚ sensor.

This phenomenon caused high amount of vibrations in turbine side and occurred after increasing RPM. Besides, these phenomena indicated in gas turbine Nyquist diagrams and are called Split Resonance. Split Resonance may cause Backward whirl between 940 to 980 RPM in both turbine and compressor journals [20].

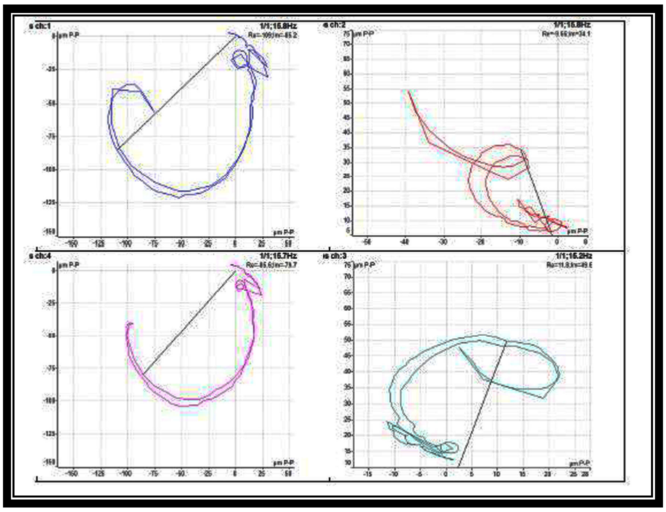

Fig. 20 Turbine and compressor relative vibrations Nyquist diagrams

Finally, turbine and compressor relative vibrations waterfall or three-dimensional FFT diagrams were obtained (for startup and trip). These diagrams were developed some hypothesis in gas turbine fault diagnosis.

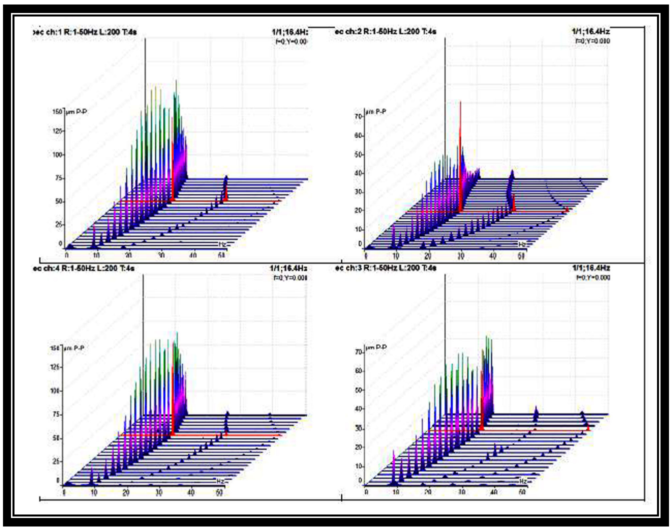

Fig. 21 Turbine and compressor three-dimensional FFT diagrams for startup and trip (waterfall graphs)

In this part, most critical equipment rotor faults are compared with the vibration evidences in mentioned gas turbine. Small amount of these kinds of faults usually are existed in normal behavior of industrial rotors but high amounts of these faults may cause trip in all somewhat most critical equipment.

The piezoelectric layers are used for sensors and actuators. Micro vibrations, generally defined as low amplitude vibrations at frequencies up to 1 kHz. An adaptive inverse dynamics control was used to suppress the vibration of a simply supported panel. These kinds of techniques can effectively utilize in most critical equipment condition monitoring systems [21].

In addition, stochastic identification technique is proposed to estimate both the parameters and order of multi-input and multi-output vibrating structural systems. In complex machinery systems like gas turbine hundreds of parts may originate hundreds of frequency vibration is such a complex condition time modeling systems like vibration modal analysis. These techniques are utilized in fault diagnosis and sometime cause some partial modification to improve the machine performance and machine vibration behavior [22].

Sub harmonic vibrations existed specially in higher RPM of gas turbine but trip was in low RPM and wear could not be a main problem. Angular misalignment is based on axial vibrations and there were no high axial vibrations in this machine. In addition, Phase analysis information provided no evidence of any misalignment. There was no 0.48 X (X=current rotor RPM) in this gas turbine FFT then oil whip/whirl could not be the problem.

Recent investigations showed that lateral natural frequencies are increased by applying tension axial loading and decreased by applying compression axial loading at the ends of the rotating shaft. Gas turbine vibration behavior was provided no evidences of natural frequencies. Besides, the axial vibrations were not too high [23].

Moreover, there were no shaft crack evidences like 2X in half critical speed (Due to three dimensional or waterfall FFT). Besides, there were no rotary looseness evidences in gas turbine TWF. In addition, foundation phase analysis showed that there is no looseness in the gas turbine.

Turbine limit load control is one of the most important parts in any gas turbine system that has direct effects in vibration behavior of machine. There are a number of methods recently are developed for these kinds of behaviors and characteristics in gas turbines [24].

Three conditions should exist simultaneously to represent unbalance [20]. Fist, all relative vibrations in different directions should be considerably high. After that, none contact main vibrations should have 90˚shift phase. Finally, the main frequency should be 1X in FFT.

All these evidences existed in this gas turbine. These evidences showed that the vibration analysis group should predicted rotor unbalance. Therefore, vibration analysis team recommended gas turbine field balance.

After field balance of the rotor, turbine started up (Some interesting case reports related to similar gas turbines field balances were discussed in [9]).

Both absolute and relative vibrations were reduced too much. All vibrations were in range of technical documents. There was no trip in Bently Nevada panels and there was no abnormal noise in gas turbine any more.

IV. Current and Future Development

The process parameters like inlet and outlet pressure and temperature trends should be monitored and make sure that all parameters are in the gas turbine technical document ranges. In addition, changing in load and RPM should be monitored accurately. These kinds of process abnormalities sometimes cause serious mechanical problems.

Besides, the condition monitoring groups should provide good trends of different vibration data and graphs like absolute and relative over all vibrations, FFT, TWF and phase characteristics in different points and directions.

Furthermore, vibration analysts should have good understanding, knowledge, experience, and background about machinery characteristics of gas turbine.

After comparing the vibration trends and data with machinery and process evidences for different main machine faults like wear, misalignment, oil whip/whirl, shaft crack, looseness and unbalance, the vibration analysts could recommend optimal maintenance action on gas turbine or any other most critical equipment.

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