International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:7, No:11, 2013

Vibration and Operation Technical Consideration before Field Balance of Gas Turbine Utilities (In Iran Power Plants SIEMENS V94.2 Gas Turbines)

Omid A. Zargar

Abstract**—**One of the most challenging times in operation of big industrial plant or utilities is the time that alert lamp of Bently Nevada connection in main board substation turn on and show the alert condition of machine. All of the maintenance groups usually make a lot of discussion with operation and together rather this alert signal is real or fake. This will be more challenging when condition monitoring vibrationdata shows 1X(X=current rotor frequency) in *fast Fourier transform*(FFT) and vibration phase trends show 90 degree shift between two non-contact probedirections with overall high radial amplitude amounts. In such situations, CM (condition monitoring) groups usually suspicious about unbalance in rotor. In this paper, four critical case histories related to SIEMENS V94.2 Gas Turbines in Iran power industry discussed in details. Furthermore, probe looseness and fake (unreal) trip in gas turbine power plants discussed. In addition, critical operation decision in alert condition in power plants discussed in details.

Keywords**—**Gas turbine, field balance, turbine compressors, balancing tools, balancing data collectors.

I. Introduction

THE condition monitoring system of most critical equipment is one of the major important topics in preventive maintenance. Bentley Nevada systems widely developed in recent years, and the option and capabilities of data collectors improve too much [1].

Higher speeds cause much greater centrifugal imbalance forces, and the current trend of rotating equipment toward higher power density clearly leads to higher operational speeds [2].

The small or medium size rotor usually sends to balance shop more easily because they carried easier to balance shop. Longer rotors just send to balance shop in more critical situations.That is why main plant usually has central balance shop inside factory. Wide types of balancing machines usually are available in the balance shop classified by hard bearing and soft bearings, flexible and rigid rotor. Most critical equipment is usually flexible and more challenging for balancing operation. The other consideration is length of the rotor. The main balance shops usually equipped with wide and sufficient types of balancing machines. In addition, there are different types of balancing machine related to overhang or middle hang rotors. The most critical equipment is usually middle hang [3].

Omid.A.Zargar is with the Department of Mechanical Engineering - Jawaharlal Nehru Technological University Hyderabad, Kukatpally, Hyderabad-500085,Andra Pradesh India .(phone +91-40-23158661 to 4, fax +91-40-23156184, e-mail: hyd2_jntuadm@sancharnet.in).

II. Material and Method

The rotor RPM is one of the most important consideration in balancing process. The RPM of balancing process is directly related to amount of rotor RPM and usually much fewer and introduced in machine technical document.

High speed rotor balance is usually one of the most challenging balancing processes in balance world and need special balancing machine [4].

High speed balance is one of the newer balance techniques that developed by modern balancing machines. There are numerous advantages to high-speed balancing of generator and turbine rotors [5]. These include, Smooth operation through the entire speed range up to over speed, The ability to access optimal weight planes, which are not normally accessible during operation in the machine, Verification of the mechanical integrity of the rotor and any assembled components—up to over speed and Electrical testing of generator rotors through their entire speed range can identify the existence of any speed related electrical faults [6].

The grade of a balance is a number that will calculate with the balancing engineer due to the technical documents of the rotor like material, geometry and weight. The grade of the balance will help the balance engineer to choose the most suitable balancing machine for the balance process [7]. There is a number of valuable information about other stage of balancing procedure in machine technical documents [8]. The field balance is applying for huge rotors that is hard to carrying to balance shop (like gas turbine power plants) by try and error process using trial weights and balancing data collectors. BM Console (VMI) and APT326 Balancing are two modern and user friendly balancing equipment recently introduced to thebalance shop and field balance activities respectively[9].

In this paper gas turbine online absolute and relative vibration monitoring system and Vibro 60 data collector used for gas turbine rotor balance. The main rotor of SIEMENS V94.2 Gas Turbines balancing process explained in details. The try and error methodology of balancing process explained in four case reports using some novel idea in balancing approaches. Besides, different aspects of gas turbine Siemens 162MW-V94.2 vibration analysis and its technical machinery specification discussed in reference 10 [10]. Furthermore, the basic principles of gas turbine power plants dynamic unbalance discussed in [11].

III. Results and Discussion

Case Report

In this part four case histories, all about SIEMENS V94.2 Gas Turbine related to Iran power industry explained.

Case History Number 1: Fars Power Plant-Unit 5
Duration:
From Tuesday, July 3, 2012 to Thursday, July 5, 2012

Due to the condition, monitoring reports archive the vibration was over 17 mm/s in the startup time in 980 RPM. This vibration is over dander limitation set up 14.7 mm/s therefore the gas turbine tripped and it cause that we could not measuring any vibration data in operation RPM or in base load condition[12].

Therefore balancing operation was more complicated and faced serious challenging. The vibration of the gas turbine in the shut down or trip moment was as following:

TABLEI
Relative Vibration Micron before Balancing

	Before Balancing				
Relative Vibration µm	980 RPM Start Up	CE	OCE	COMP	TUR
		17.4	16.5	67.8	77.9

TABLE II
Absolute Vibration MM/S before Balancing

	Before Balancing		
Absolute Vibration mm/s	980 RPM Start Up	COMP	TUR
		5.9	17.3

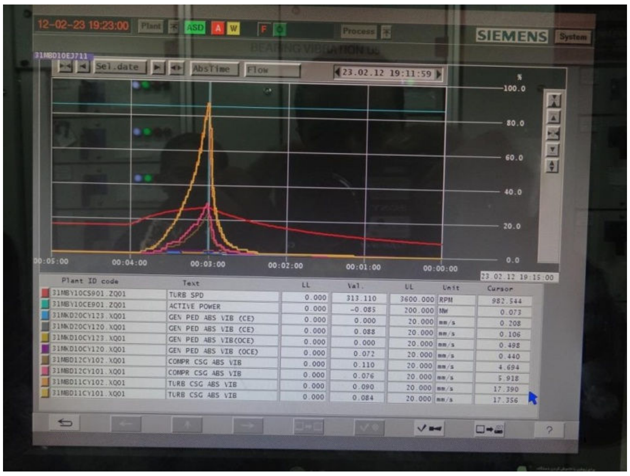

Fig. 1 Trip vibration condition

We analyzed the condition monitoring data in trip moment and realized that the main vibration peak was in 1X. Two direction amplitude overall both considerably high and the phase data shows 90 degree shift between two main direction in that moment. All those evidences leaded us to unbalance. We decided to operate balancing program in two steps because gas turbine was on trip condition and we could not operate the machine in process RPM.

In first step, we were planning to operate the balancing process in 3000 RPM between the period of Saturday, May 12, 2012 and Tuesday, May 15, 2012. In this stage, we installed 19 balancing weights in Compressor Disk Stage ten 165˚ and 27 balancing weight in turbine Stage four 165˚. However, unfortunately, process has some considerations and they could not run the gas turbine in the operation condition then we had to waiting for the suitable condition to operate the next stage of balancing program [13].

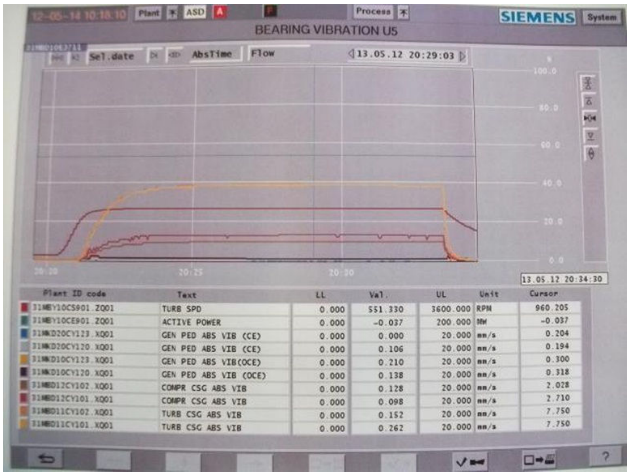

Fig. 2 Vibration condition in first balancing stage

In the next stage process operated the gas turbine in 3000 RPM and balancing operation continue between the period of Tuesday, July 3, 2012 and Friday, July 6, 2012,finally we decided to install 15 balancing weight in Compressor Disk Stage ten 150 ˚ and 23 balancing weight in turbine stage four 155˚.

Fig. 3 Vibration condition in 3000 RPM

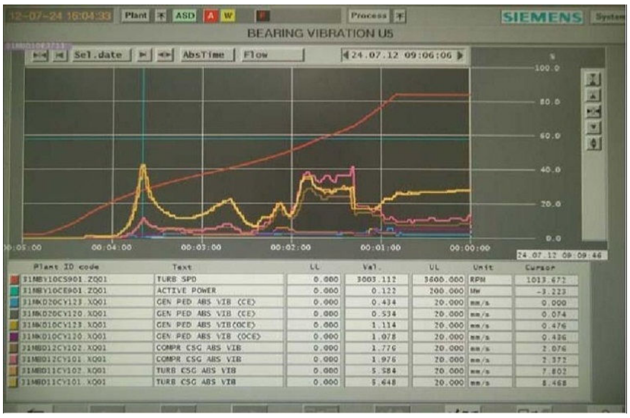

Fig. 4 Vibration condition in 3000 RPM after synchronize the unit by process

Absolute turbine and compressor vibration after balancing shown in following table:

TABLE III
ABSOLUTE TURBINE AND COMPRESSOR VIBRATION AFTER BALANCING

Fars utility (Unit 5)		COM.(abs)	TUR.(abs)
1000 RPM	Start up	2.3	8.4
	Shut down	2	8
1400 RPM	Start up	1.7	4
	Shut down	2	6.3
3000 RPM (116 MW)		2.5	6.4

In conclusion after synchronizing the unit by process in Monday, July 23, 2012 and analysis the vibration trends in the base load we could operate the gas turbine in process condition with close monitoring [14].

Case History Number 2: Kerman Power Plant-Unit2 Duration: From Wednesday, October 24, 2012 to Thursday, October 25, 2012

The process of Kerman utility unit 2 reported that the relativevibration of OCE journal pass over 159 micrometer peak to peak after the turbine passing 1800 RPM (first critical speed) in shut down process. After analyzing the vibration data, we decided to operate the single panel balancing process on middle shaft at Wednesday, October 24, 2012.

After adding trial weights finally we achieve to 19 balancing weight 40 grams 155˚ therelativevibration of OCE journal reduce too much when turbine passing 1800 RPM [15] .

TABLE IV
RELATIVE VIBRATION MICRON BEFORE BALANCING

Before Balancing					
Relative Vibration µm	1800 RPM	CE	OCE	COMP	TUR
	Start up	64	152	71.4	35.3
	Shut down	51	159.3	72.3	51.4

TABLE V
ABSOLUTE VIBRATION MM/S BEFORE BALANCING

Before Balancing			
Absolute Vibration mm/s	1800 RPM	COMP	TUR
	Start up	6.1	6.2
	Shut down	6.1	5.8

TABLE VI
RELATIVE VIBRATION MICRONAFTER BALANCING

AfterBalancing					
Relative Vibration µm	1800 RPM	CE	OCE	COMP	TUR
	Start up	30.7	37.2	30.7	24.6
	Shut down	12	14	37.7	34.5

TABLE VII
ABSOLUTE VIBRATION MM/S AFTER BALANCING

AfterBalancing			
Absolute Vibration mm/s	1800 RPM	COMP	TUR
	Start up	2.9	3.5
	Shut down	2.7	3.1

The vibration trends before and after balancing operation as following:

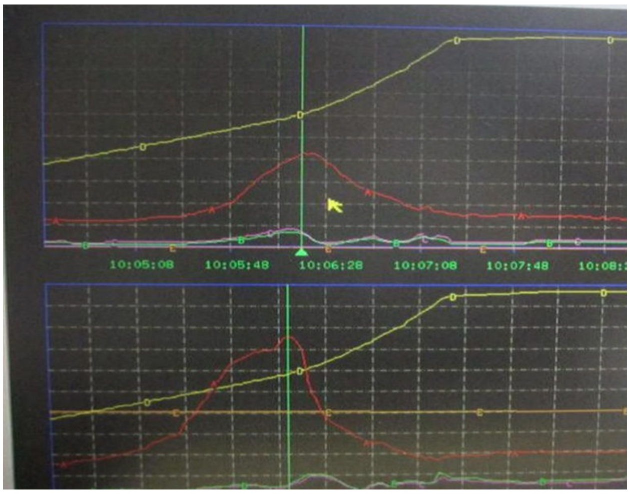

Fig. 5 Startup vibration of generator journal before balancing process

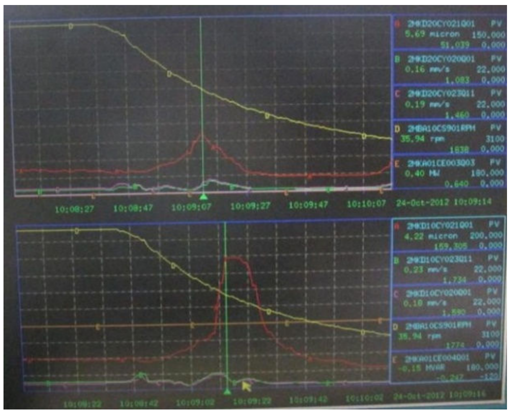

Fig. 6 Shutdown vibration of generator journal before balancing process

International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:7, No:11, 2013

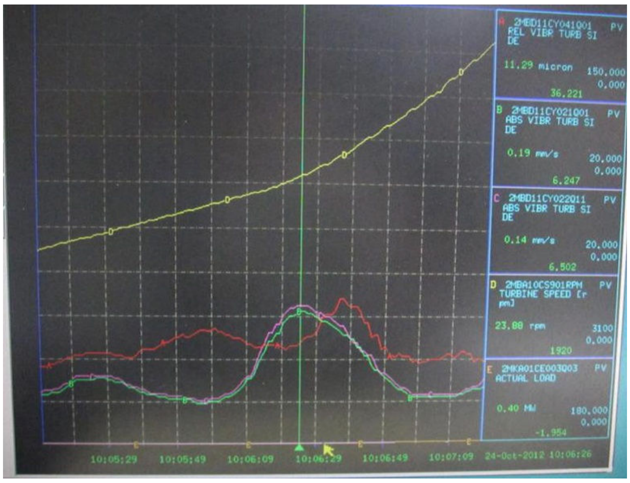

Fig. 7 Startup vibration of turbine journal before balancing process

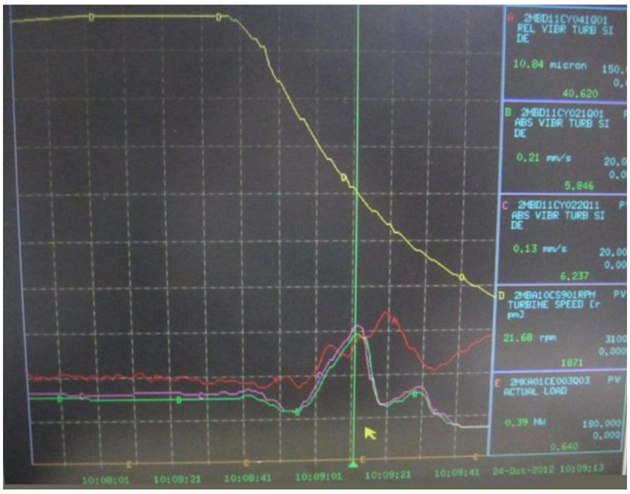

Fig. 8 Shut down vibration of turbine journal before balancing process

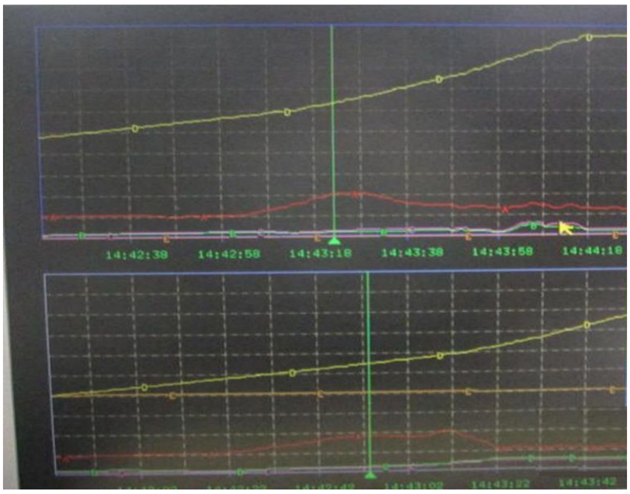

Fig. 9 Startup vibration of generator journal after balancing process

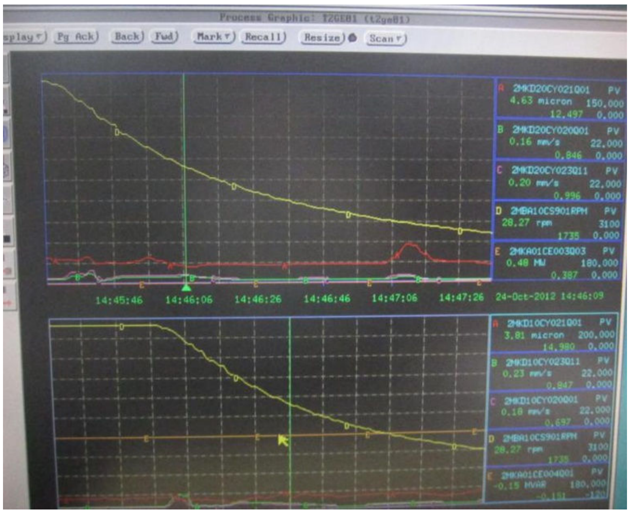

Fig. 10 Shutdown vibration of generator journal after balancing process

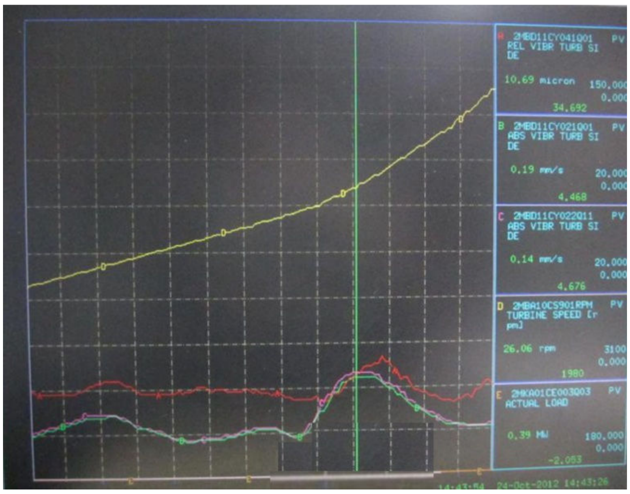

Fig. 11 Startup vibration of turbine journal after balancing process

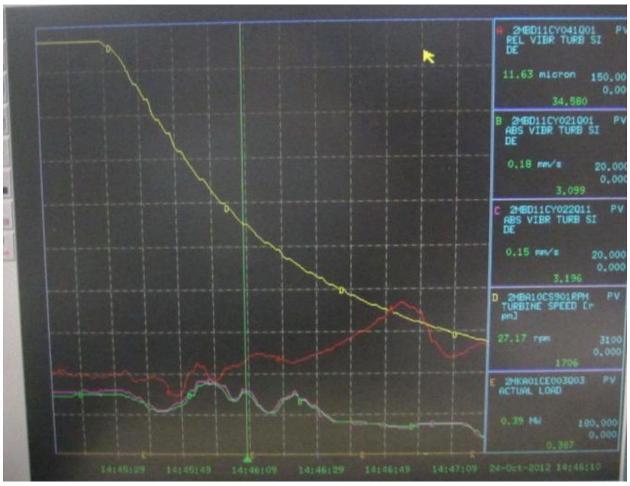

Fig. 12 Shut down vibration of turbine journal after balancing process

Due to the vibration, overall amount after balancing process and vibration limits in technical document of this gas turbine the vibration condition of this machine is in good condition therefore this unit can continue operation without any problem [16].

Case History Number 3: Shahrod Utility-Unit 2
Duration:
From Tuesday, May 1, 2012 to Friday, May 4, 2012

Due to the process report about high relative vibration in different point of gas turbine specially in OCE journal, approximately 134 micrometer peak to peak after overhaul the session between different maintenance group in Tuesday, May 1, 2012 take placed. These vibrations caused the gas turbine shut downed. The trends of process parameter was in the range of technical documents of both turbine and compressor also the machinery bearing clearances was in the range of the document;therefore, the session result was make a vibration committee by both condition monitoring (CM), electrical special toolsand balancing group together to make decision about the signals accuracy[17].

First of all CM records showed that the frequency shift randomly but the overall amplitude is relatively in same situation all over the startup and shut down process. This kind of behavior reject the hypothesis of probe looseness and also the 1X vibration was not always dominated in different RPM. Besides, the phase have not 90 degree shift between two main direction therefore the unbalance hypothesis also rejected therefore they may have some problem in adjusting probe installation or any related electrical or electronic considerations[18].

The electrical special tools group then planning theinstallation check in Wednesday, May 2, 2012 they first checked the distance between probesand the shaftby digital volt meter all the records was correct by 0.01 volt accuracy then they check the angle that was45˚ by 0.1 accuracy all was acceptable and in the range of technical document of gas turbine but there was a ridiculous mistake that the transfer cable must be 1 meter but the installation guy install 1.5 meter cable that was the recommended cable of gas turbine unit 6 by mistake and the electrical resistance of these length of cable caused unreal signal and unreal signal cause unreal shut down or trip[19].

The electrical special tools reinstalled the probes with new cables and the gas turbine restarted for operation. The vibration trends in startup and shut down before and after change the cables was as following.

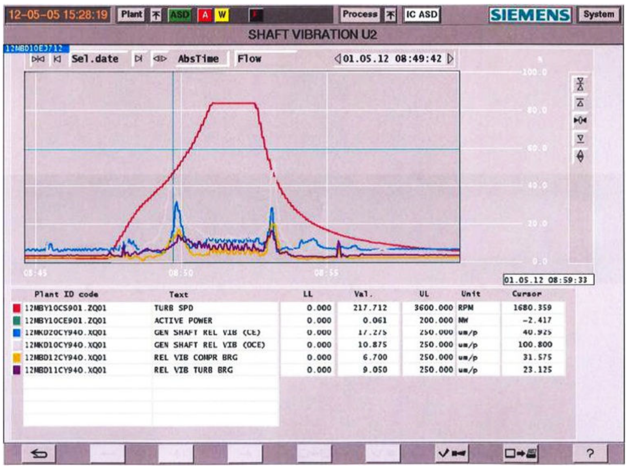

Fig. 13 Relative vibration trend startup turbine side before change the cables

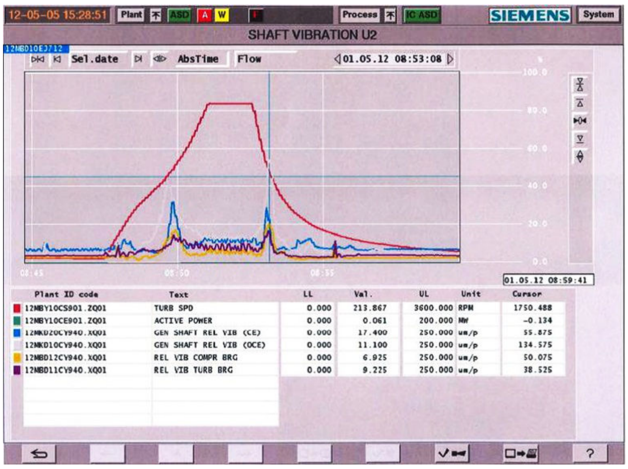

Fig. 14 Relative vibration trend shut down turbine side before change the cables

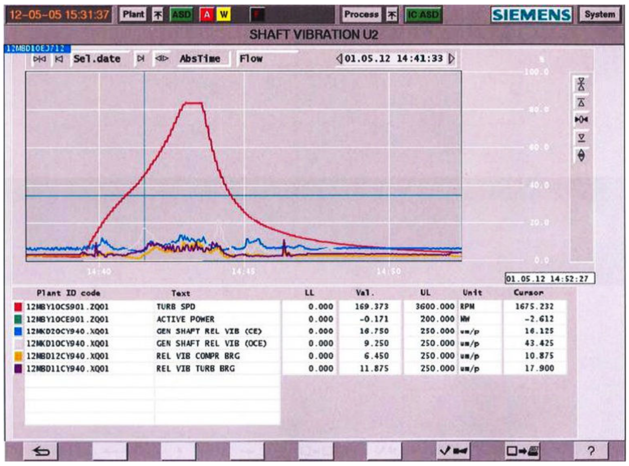

Fig. 15 Relative vibration trend startup turbine side after change the cables

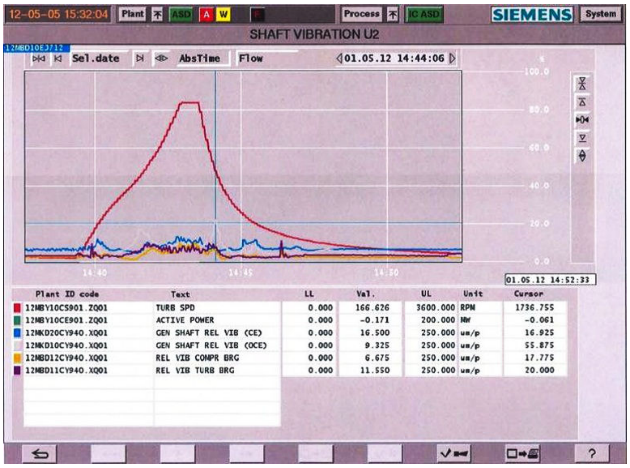

Fig. 16 Relative vibration trend shut down turbine side after change the cables

Dou to the vibration overall amount after changing the cables the vibration of this gas turbine was in good condition after overhaul and this gas turbine could continue operation and the trip vibrations was unreal or fake[20].

Case History Number 4: Kerman Utility-Unit 4 Durations:

First Period: From Wednesday, February 27, 2013 to Wednesday, March 6, 2013.

Second Period: From Monday, March 11, 2013 to Sunday, March 17, 2013.

Third Period: From Thursday, April 18, 2013 to Thursday, April 25, 2013.

Due to vibration trip unit, four of Kerman utility on Monday, February 25, 2013 after overhaul in 940 RPM the field balancing team went from Tehran to Kerman. The following operations occurred on unit four in first period (8 days).Vibration analysis from this gas turbine represented unbalance. It means vibration FFT was mainly in 1X, two main direction vibrations were both high and had 90 degree shift in phase trends. These evidences represented high amount of unbalance. The balancing operation was not successful and recommended high amount of balancing weights that was not possible according to the gas turbine technical document [21].

We decided to check the middle shaft run out in several points that everything was on machinery document ranges. The operation asked to field balance the rotor (Issued related work order) [22]. Therefore, the balancing operation continued. In previous RPM the vibration table was like below:

TABLE VIII
TURBINE AND COMPRESSOR VIBRATIONS

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
3	66	15	158

This time by adding, the balancing weights the gas turbine continues operation up to 1867 RPM and then trip [23]. Five balancing weight added in 133˚ in Front Hollow Shaft and 14 balancing weight added in 125 ˚ in Compressor Disk Stage ten finally 9 balancing weight added in 120˚ in Rear Hollow Shaft. The new vibration amounts were as following:

TABLE IX
TURBINE AND COMPRESSOR VIBRATIONS AFTER FIRST STAGE BALANCING PROCESS

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
2.7	32	15	162

In the next stage after adding, the balancing weights we could continue up to 3000 RPM. 9 balancing weight added in 35˚ in Front Hollow Shaft,8 balancing weight added in 88˚ in Front Hollow Shaft,21 balancing weight in 45˚ in Compressor Disk Stage ten and 10 balancing weight added in 205˚ in Rear Hollow Shaft the amount of new vibration recorded as following:

TABLE X
TURBINE AND COMPRESSOR VIBRATIONS AFTER SECOND STAGE BALANCING PROCESS

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
7.54	130	10.35	103

Finally by adding three weighting balance in Rear Hollow Shaft 177˚ we achieved following vibration behavior in no load condition.

TABLE XI
TURBINE AND COMPRESSOR VIBRATIONS AFTER THIRD STAGE BALANCING PROCESS

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
5.4	97	8.12	80

In these conditions, operation increased the load up to 40 MW and the overall vibration increased up to 1 mm/s.

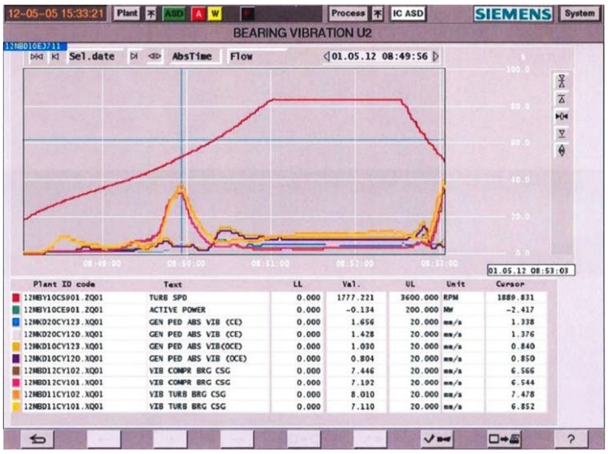

Fig. 17 Vibration condition SIEMENS V94.2 Gas Turbine utility 3000 RPM without any load

The exhaust liner was engage and the machinery maintenance actions take place in the next stage. The alignment check operated by machinery group between compressor and generator due to the condition monitoring vibration analysis. In addition, we decided to continue the balancing operation to reduce the amount of vibration on Thursday, April 18, 2013. The initial vibration amounts were as following:

TABLE XII
INITIAL TURBINE AND COMPRESSOR VIBRATIONS

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
5	100	8	85

The balancing process continued by adding the weights ,six weighting balance 240 gram added in intermediate Shaft 70˚,one weighting balance 180 gram in Front Hollow Shaft 5˚,one weighting balance 115 gram in Front Hollow Shaft 60˚, one weighing balance 180 gram in Compressor Disk Stage ten 75˚,three weighting balance 540 gram in Compressor Disk Stage ten 154˚, three weighting balance 475 gram in Compressor Disk Stage ten 0˚, four balancing weight 720 gram in Rear Hollow Shaft 240˚ and two weighting balance 295 gram in Rear Hollow Shaft 180˚. The relative and absolute vibration change by loading the process condition was as following:

TABLE XIII
TURBINE AND COMPRESSOR VIBRATIONS FULL SPEED NO LOAD

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
4	73	6.2	76

TABLE XIV
TURBINE AND COMPRESSOR VIBRATIONS LOAD=40 MW

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
4.58	105	6.28	85

TABLE XV
TURBINE AND COMPRESSOR VIBRATIONS BASE LOAD

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
4.5	121	7.12	92

TABLE XVI
TURBINE AND COMPRESSOR VIBRATIONS BASE LOAD AFTER ONE HOUR

Compressor		Turbine	
Abs (mm/s)	Rel(μm)	Abs (mm/s)	Rel(μm)
4.7	138	9.25	106

After two hours the vibration over all turbine side passed over 11.5mm/s and gas turbine tripped. This caused turbine shut down. The next stage in weight balancing process caused to reduce both journal bearings of turbine and compressor relative vibration but unfortunately, the absolute vibration of turbine side increased too much [24].

Finally we had to remove all the weights(25 weighting balance 1000 gram in intermediate Shaft 77˚,three weighting balance 540 gram in Front Hollow Shaft 355˚,two weighting balance 360 gram Front Hollow Shaft 60˚ and seven weighting balance 1260 gram in Rear Hollow Shaft 274˚). In the next stage by removing, all above balancing weight and restart the turbine the vibration behavior of gas turbine isfull speed-no load 2.2 mm/s-1.84 mm/s, base load 4.42 mm/s-3.71 mm/s, after 45 minute at Base load 4.91 mm/s-4.04 mm/s and after 49 minute at Base load 5.11 mm/s-4.17mm/s in two main directions respectively.

As it is clear from the above vibration data, vibration in 3000 rpm and base load was increasing gradually related to no load condition. In addition, increasing was in both absolute and relative vibration. Furthermore, increasing occurred in both compressor and turbine sides [25].

Absolute vibration was decreasing suddenly in turbine side after 40 minute that turbine operate at base loadto 3 mm/s butabout 4 minute later it had a sharp raise up to 5 mm/s. Therefore, it was increasing gradually and finally caused turbine trip [26]. Due to the high initial overall vibration of the gas turbine, high amount of weights requested in balancing process and gas turbine document recommendations [15].

This was the optimal condition of the balancing process and the next step will requested out of rang weights. Therefore, it was not possible to reduce high overall vibrations by adding balancing weights or field balance and rotor should be replaced [27].

IV. CURRENT AND FUTURE DEVELOPMENT

The vibrations that can represent unbalance in most critical equipment like gas turbine utilities or steam turbine multistage compressors should have three main characteristic. Firstly, both two main direction overall vibrations should have high amplitude amounts. Secondly, the phase trends in recent days should have 90-degree shift between two main directions. Finally, the FFT should be completely or more than 90 percent in 1X. If any of these three conditions was not satisfied, you should not suspicious about unbalance. Rotor field balance sometimes faced some challenges.

The knowledge of the balance engineers about technical machinery and technical document of machine could help them to reduce the balancing process time by reducing the try and error stages.

Furthermore, the fluctuation of overall vibration may represent probe looseness or any process problem in gas turbine. In addition, The CM engineer should have a good knowledge of process condition of machine specially inlet and out let pressure and temperature trends of both turbine and compressor.

Besides, The random fluctuation in frequencies of FFT may represented any probe maladjustment, inaccurate probe installation or any possible mistake in related electrical or electronic areas causedunreal alert signal or unreal trip. Furthermore, in gas turbines the absolute vibration will help us to distinguish these kinds of unreal signals more easily by comparing absolute and relative vibration trends with each other.

REFERENCES

[1] Vibration school website: http://www.vibrationschool.com, 2013.
[2] Kalmegh, A., and S.Bhaskar. Dynamic Balancing of Centrifugal Pump Impeller International Journal of Emerging Technology and Advanced Engineering 6(3): 2012; 221-223.
[3] Xie, D. rigid rotor balancing. wuhan university publication, 3(6): 2012; 331-337.
[4] High speed rotor balance facilities, Elliott High Speed Rotor Balance website: http://www.elliottturbo.com/High_Speed_Balance.asp,2012.
[5] Toshiba , Milwaukee Service Center , 6623 W Washington Street ,West Allis, WI 53214 , (414) 475-2800 , 2013.
[6] Intelligent Balancing Solutions, hofmann product website, www.hofmann-global.com, 2013.
[7] World's Leading Supplier of Soft-Bearing Balancing Machines & Instruments, IRD Balancing website, www.irdbalancing.com,2013.
[8] Becker,E.Mobile field balancing reduces vibrations in energy and power plants.PRÜFTECHNIK Condition Monitoring Power Tech 13(6):2012; 1-7.
[9] VMI website:http://www.vmiab.com, 2013.
[10] Ali Zargar O.2013. Different Aspects of Gas Turbine Siemens 162MW-V94.2 Vibration Analysis. Middle-East Journal of Scientific Research. 18: 546-556.
[11] Ali Zargar O.2013. Vibration Analysis of an Alstom Typhoon GasTurbine Power Plant Related to Iran Oil Industry. World Academy of Science, Engineering and Technology.International Journal of Mechanical, Industrial Science and Engineering Vol: 7 No: 11.
[12] Kumar, S., R. Sehgal, R. Kumar, and S. Bhandari. Vibrations Analysis of 4 Jaw Flexible Coupling Considering Unbalancing in Two Planes.International Journal of Science and Technology 11(4): 2012; 324-326.
[13] Donth, D., A. Grunwald, C. Heilmann, and M. Melsheimer. Multi plane on site balancing of wind turbine drive trains, BerlinWind journal 13(3): 2012; 4-6.
[14] Brüel&Kjærwebsite: http://www.bksv.com,2013.
[15] Siemens website: Siemens Gas Turbines, Technical Papershttp:/www.energy.siemens.com, 2012.
[16] Hariharan, V. and P.S.S. Srinivasan. Vibrational Analysis of Flexible Coupling by Considering Unbalance. World Applied Sciences Journal 8(3): 2010; 1024-1030.
[17] EftekhariM.,Javadi M. and Farsani R.E. 2011. Free vibration analysis of cracked functionally graded material beam. World Applied Sciences Journal, 12:1216-1222.
[18] Sukirno,A.R.Ajar, S. Bismo, and M. Nasikin. Bio grease Based on Palm Oil and Lithium Soap Thickener: Evaluation of Anti wear Property. World Applied Sciences Journal 6(4):2009; 402-406.
[19] Karim, Z., M.Z. Nuawi, J.A. Ghani , S.Abdullah , and M.J. GhazaliOptimization of Integrated Kurtosis-Based Algorithm for Z-Filter (I-KazTM) Coefficient Using Multi Level Signal Decomposition Technique. World Applied Sciences Journal 14(6): 2011; 1542-1546.
[20] Alsaade, F., N. Zaman, A. Abdullah, and M. ZafarDawood. Enhancing Surveillance and Security of Oil Pipelines Transportation Using Wireless Sensor Network. Middle-East Journal of Scientific Research 11(5): 2012; 1030-1034.
[21] Akhshabi, M. A New Fuzzy Multi Criteria Model for Maintenance Policy. Middle-East Journal of Scientific Research 10(3):2011; 34-37.
[22] Azadi, M., E. Azadi, and M. Roopaei. Adaptive Inverse Dynamics Control for Vibration Suppression of Flexible Panel with Piezoelectric Layers. World Applied Sciences Journal 12(4):2011; 2343-2347.
[23] Abu-El-Yazied, T.G., S.Y.Abu-El-Haggag, and F.S. Al-Fares. Identification of Vibrating Structures with Application to a Steel Tower. World Applied Sciences Journal 6(3):2009; 1674-1679.
[24] Hosseini, H., D.D. Ganji,M.Abaspour, and H.D. Kaliji. Effect of Axial Force on Natural Frequency of Lateral Vibration of Flexible Rotating Shafts. World Applied Sciences Journal 15(4):2011; 854-858.
[25] Afkhami, H., A. Argha, M.Ropaee, and M.A. Noori. Surveying the control loops of the governor of the V94.2 gas turbine. World Applied Sciences Journal 15(4): 2011; 1436-1437.
[26] Goodarziyan, H. and O.M. shobi. Effect of design parameters on the exergetic operation of gas turbine power plant. World Applied Sciences Journal 8(3):2010; 590-594.
[27] Liang, Z.Q. W.H. Qing, and Y.J. Fei. A Novel Method of Dynamic Balance Weighting for Single-Disk Rotor System. Diagnosis and Self-Recovery Engineering Research 11(4): 2012; 1-8.