

## Hydraulic Unbalance in Oil Injected Twin Rotary Screw Compressor Vibration Analysis

*Omid Ali Zargar*

Department of Mechanical Engineering, Jawaharlal Nehru Technological University  
Hyderabad Kukatpally, Hyderabad-500085 Andhra Pradesh, India

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**Abstract:** Vibration analysis of screw compressors is one of the most challenging cases in preventive maintenance and this kind of equipment consider as one of the vibration bad actor facilities in industrial plants. on line condition monitoring systems develop too much in recent years for such equipment. The high frequency vibration of ball bearing, gears, male and female cause complex FFT and TWF in screw compressors. The male and females usually and randomly sending to balance shop for balancing operation and it usually cause some bending in rotors during the balancing operation that cause further machining in such equipment. These kind of machining operation increasing the vibration analysis complexity of such equipment beside some process characteristic abnormality like inlet and out let pressure and temperature. In this paper first I briefly explain mechanical principal and different type of screw compressors beside this I tried to explain some new condition monitoring system and techniques of such equipment. Finally I discusses one of the common behavior of oil injected twin rotary screw compressor called hydraulic unbalance usually occur after machining operation of male or female and have some specific characteristics in FFT and TWF. Hope this paper will helpful for the readers to achieve better understanding of rotary screw compressor vibration analysis in real industrial process conditions.

**Key words:** Vibration analysis • Twin screw compressor • Oil injected screw compressor • Time wave form (TWF) • Fast Fourier transform (FFT) • Hydraulic unbalance and rotor unbalance

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### INTRODUCTION

Rotary screw compressors are widely used today in industrial refrigeration for compression of ammonia and other refrigerating gases. Simple in concept, the screw geometry is sufficiently difficult to visualize that many people using screws today have only a vague idea how they actually work. An understanding of the basics of their operation will help in applying them correctly, avoiding nuisance problems in operation and achieving the best overall system designs. A typical oil flooded twin screw compressor consists of male and female rotors mounted on bearings to fix their position in a rotor housing which holds the rotors in closely tolerance intersecting cylindrical bores shown in Figure 1. The rotors basic shape is a screw thread, with varying numbers of lobes on the male and female rotors. The driving device is generally connected to the male rotor with the male driving the female through an oil film. In refrigeration, four or five lobed male rotors generally

drive six or seven lobe female rotors to give a female rotor speed that is somewhat less than the male speed. Some designs connect the drive to the female rotor in order to produce higher rotor speeds thus increasing displacement. However, this increases loading on the rotors in the area of torque transfer and can reduce rotor life [1].

Compressors may be simply classified as dynamic compressor and displacement compressor; the displacement compressors confine successive volumes of gas within a closed space and increase the pressure by reducing the volume of the space. The displacement compressors are also classified as two types: rotary compressor and reciprocating compressor. As a major type of rotary and positive displacement compressor, the screw compressor has been playing more and more important role in the applications of compressors. Some of the twin screw compressor equipped with timing gear otherwise oil provide the only necessary rotating role.

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**Corresponding Author:** Omid Ali Zargar, Department of Mechanical Engineering - Jawaharlal Nehru Technological University Hyderabad Kukatpally, Hyderabad-500085 Andhra Pradesh, India.

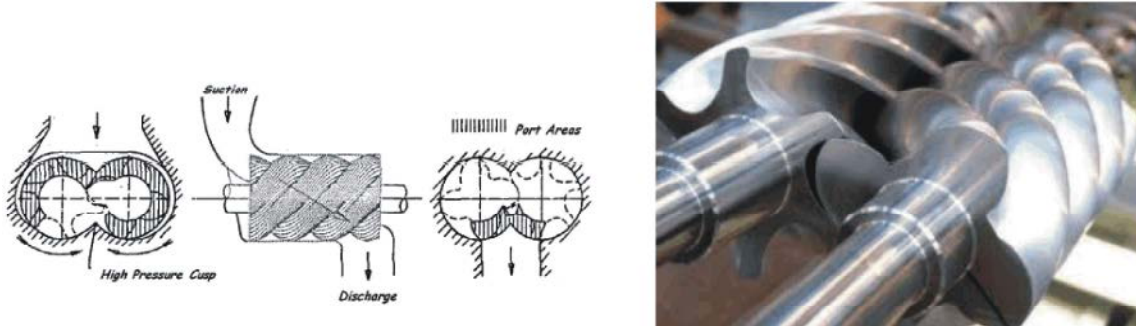


Fig. 1: Twin rotary screw compressor

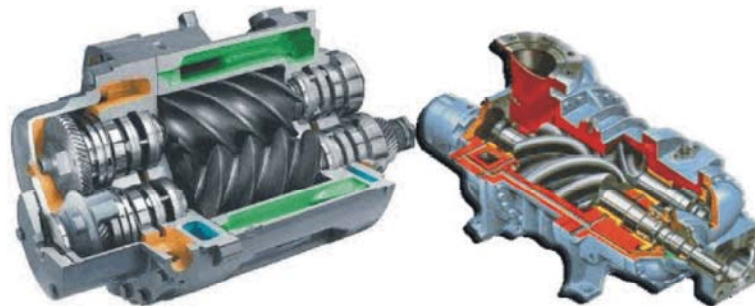


Fig. 2: Typical twin screw compressors

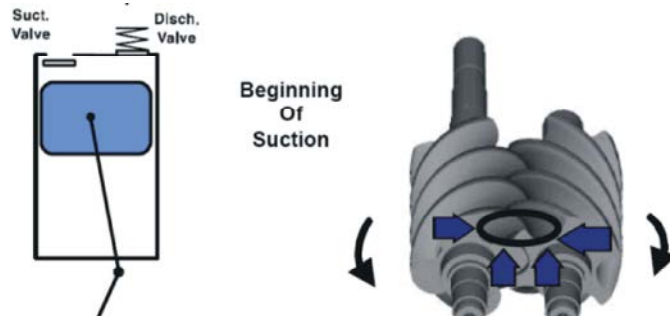


Fig. 3: Screw and reciprocating compressor at beginning of suction

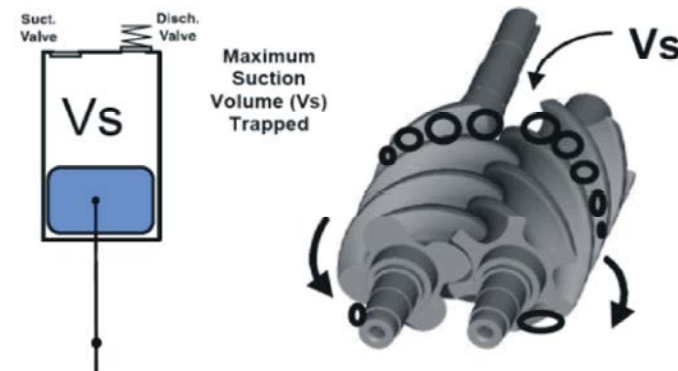


Fig. 4: Maximum suction volume trapped in reciprocating and screw compressor

Suction gas is drawn into the compressor to fill the void where the male rotor rotates out of the female flute on the suction end of the compressor. Suction charge fills the entire volume of each screw thread

as the unmeshing thread proceeds down the length of the rotor. This is analogous to the suction stroke in a reciprocating compressor as the piston is drawn down the cylinder.

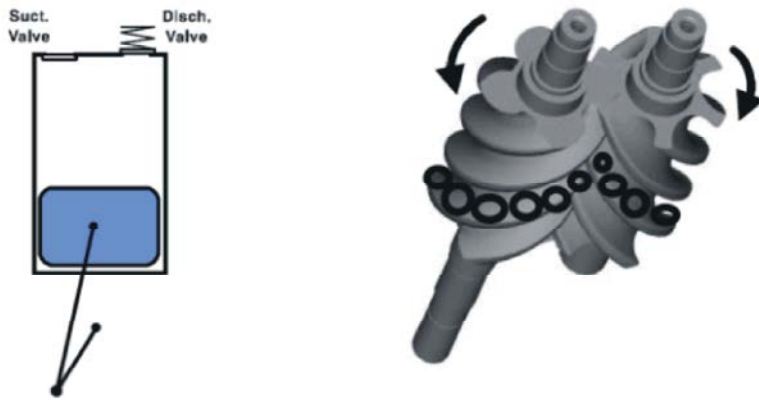


Fig. 5: Compression in reciprocating and screw compressor

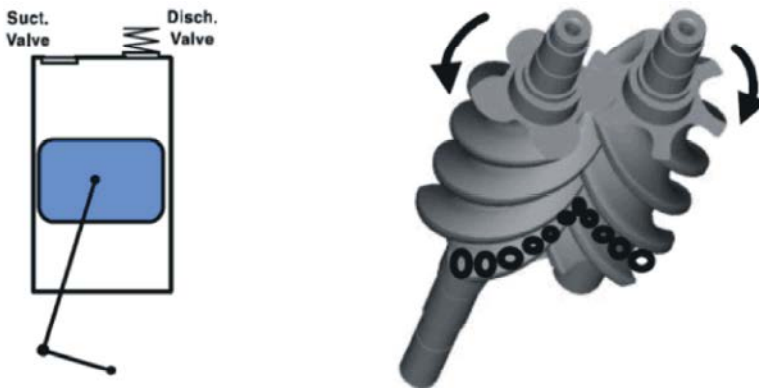


Fig. 6: Compression in reciprocating and screw compressor

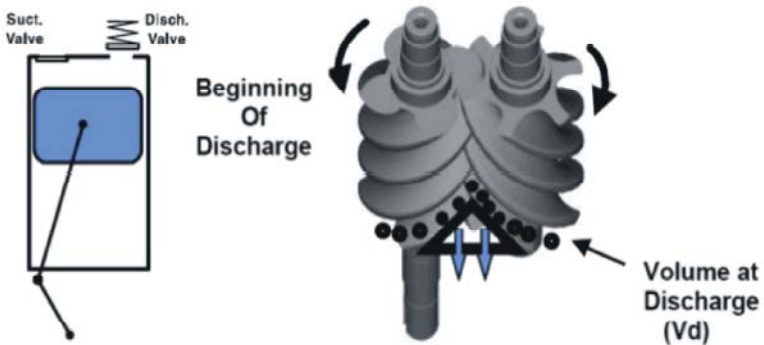


Fig. 7: Beginning of discharge in reciprocating and screw compressor

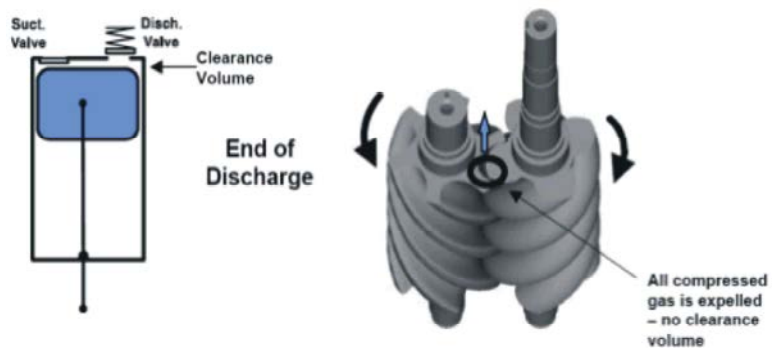


Fig. 8: End of discharge in reciprocating and screw compressor

The suction charge becomes trapped in two helically shaped cylinders formed by the screw threads and the housing as the threads rotate out of the open suction port. The volume trapped in both screw threads over their entire length is defined as the volume at suction, ( $V_s$ ). In the recip. Analogy the piston reaches the bottom of the stroke and the suction valve closes, trapping the suction volume, ( $V_s$ ).

The displacement per revolution of the reciprocating compressors. Is defined in terms of suction volume, by the bore times the stroke times the number of cylinders. The total displacement of the screw compressor is the volume at suction per thread times the number of lobes on the driving rotor. After that The male rotor lobe will begin to enter the trapped female flute on the bottom of the compressor at the suction end, forming the back edge of the trapped gas pocket. The two separate gas cylinders in each rotor are joined to form a "V" shaped wedge of gas with the point of the "V" at the intersection of the threads on the suction end.

Further rotation begins to reduce the trapped volume in the "V" and compress the trapped gas. The intersection point of the male lobe in the female flute is like the piston in the recip. That is starting up the cylinder and compressing the gas ahead of it.

In the recip. Compressor, the discharge process starts when the discharge valve first opens. As the pressure in the cylinder exceeds the pressure above the valve, the valve lifts, allowing the compressed gas to be pushed into the discharge manifold. The screw compressor has no valves to determine when compression is over. The location of the discharge ports determine when compression is over. The volume of gas remaining in the "V" shaped trapped pocket at discharge port opening is defined as the volume at discharge, ( $V_d$ ).

A radial discharge port is used on the outlet end of the slide valve and an axial port is used on the discharge end wall. These two ports provide relief of the internal compressed gas and allow it to be pushed into the discharge housing. Positioning of the discharge ports is very important as this controls the amount of internal compression. In the reciprocating compressors the discharge process is complete when the piston reaches the top of the compression stroke and the discharge valve closes. The end of the discharge process in the screw occurs as the trapped pocket is filled by the male lobe at the outlet end wall of the compressor. The reciprocating compressors always has a small amount of gas, (clearance volume), that is left at the top of the stroke to expand on the next suction stroke, taking up space that could have

been used to draw in more suction charge. At the end of the discharge process in the screw, no clearance volume remains. All compressed gas is pushed out the discharge ports. This is a significant factor that helps the screw compressor to be able to run at much higher compression ratios than a reciprocating compressor [2].

**Experimental Details:** Screw compressors usually connected to an increasing RPM heavy gear box with several main shafts and stages and also a high KW electromotor usually more than 38 KW will rotate the gear box. The coupling usually robber type to damping the related vibration as well as possible. The foundation may be rigid or flexible in different industrial plants. The flexible foundation screw compressors vibration behavior usually more challenging. Because of the annoying noises usually generated with these king of equipment. The monitoring system is connected to some on line vibration monitoring usually in some part of process board each part have some separate vibration monitoring systems the electro motor usually equipped with journal bearings. These kind of bearings also equipped with some none contact probes and bently Nevada vibration monitoring systems. Then all vibration analysis techniques like shaft center line analysis in this field will help us in critical vibration conditions a new wireless condition monitoring system also developed in recent years the main advantages of such system is reducing the installation errors and increasing the monitoring accuracy and speed [3]. The gear boxes usually equipped with shock pulse measurement (SPM) system equipped with on line monitoring system. The bearing condition unite (BCU) trends will help us to evaluate the condition of machine in different load, speed and situation also the traditional vibration measuring program perform to all parts of screw compressors and all vibration data such as overall vibrations, TWF, FFT and phase values will help us in vibration analysis of these kind of machines more than other monitoring systems. Sometimes these kind of data are available in on line vibration monitoring system in some most critical screw compressors. beside this a condition based monitoring system using ultrasonic signal recently developed for gear boxes a high frequency ultrasonic sensor is used as a transducer to collect data base on run frequency and gear mesh frequencies finlly signal will filtered within an ultrasonic range [4]. The phase analysis will help us to evaluate the condition of coupling or misalignment, soft foot and high low flanges. also vibration modal analysis will help us in any foundation vibration analysis if related

data collectors like VDAU-6000 is available. On the other hand noise monitoring systems usually perform in such equipment. Beside this oil analysis is one of the other critical check point always recommended for such equipment. Nowadays thermography developed to evaluate the bearing temperature of this kind of equipment in minimum time for evaluating the bearing condition of screw compressors in gear box and compressor bearings and these kinds of techniques found so effective in these matters. The lubrication system of different parts of screw compressors consider as a main technical category in monitoring systems of such equipment 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 or equipment working in different industries. The grease was manufactured via 4 steps of processes: saponification in pressurized reactor, soap dilution by heating, re-crystallization by cooling and homogenization. The result of lubrication performance tests using 4-ball wear-test showed that the amount of wear on ball specimen was smaller in test with the palm- grease than the test with mineral (HVI 160S) grease. This ability of the palm-grease to provide better surface protection or anti wear property will be helpful in different type of machine lubrication like screw compressors [5]. The electro motors of screw compressors usually equipped with slider bearings several modeling techniques developed in recent years for simulate such bearing conditions. The homotopy analysis method (HAM) for strongly nonlinear problems is used to give explicit analytic solution for lubrications problems in slider bearings [6]. Also these kind of bearing usually equipped minimum with a vertical direction none contact probe or two classical none contact probe for most critical equipment. The foundation of screw compressor may be flexible or rigid types of foundation have direct effects on vibration limits in condition monitoring trends. Damping is a complex phenomenon which acts in the form of absorption and dissipation of the energy in the vibrational systems. Different factors effect on the damping such as type of joints in the connections. There are several methods for foundation modeling and design one of the most effective method is vibration modal analysis based on the phase analysis [7]. Different maintenance strategies such as corrective maintenance, time based maintenance, preventive maintenance, condition-based maintenance and predictive maintenance exist for different equipment like screw compressors. A new fuzzy multi criteria model

is introduced and it is used for the optimization decision making of the complex system maintenance strategy with five Criteria. A maintenance strategy has been modeled with consideration of four fuzzy parameters in the figure of Multi Criteria Decision Making. One of the Criteria elaborates minimization of total Completion time. The second Criteria has been considered in this model due to describe minimize cost. The other Criteria are in regards to minimization of risk and working man and maximize retrieval parts. These kinds of modern modeling systems will help us a lot in choosing our maintenance strategy [8]. Also reliability centered maintenance (RCM) developed in recent years base on stochastically maintenance management and condition monitoring systems of all maintenance and engineering groups in industrial zone and considering as a new revolution in maintenance strategies ball bearings and roller bearings using both in screw compressor and gear box gears, male and female is also produce high frequencies.that is why the FFT of such equipment is usually complex in high frequencies. Roller bearing failure is a major factor in failure of rotating machinery. As a fatal defect is detected, it is common to shut down the machinery as soon as possible to avoid catastrophic damages. Performing such an action, which usually occurs at inconvenient times, typically results in substantial time and economical losses. It is therefore important to monitor the condition of roller bearings and to know the details of severity of defects before they cause serious catastrophic consequences. Traditional FFT and TWF is one of the most effective methods in this case. the small hill shape type frequencies will appear around bearing high frequencies in first stages. after that by developing bad bearing condition the frequencies shift themselves to ball or roller pass frequency and its small sidebands. In this stage bearing is completely damage and noise appears during operation [9] beside this monitoring overall acceleration and also BCU could help us in this matter. The thermography and sound analysis is also could help us a lot in roller bearing fault diagnosis and developed too much in recent years. the coupling in both side of gear box is consider as a most challenging parts of screw compressors. the couplings usually is rubber type and produce run frequencies. The coupling usually is under high tension and may lose some rubber parts. These phenomena produce coupling unbalance that is hard to diagnosis in complex FFT and TWF related to screw compressors also noise pollution of screw compressor will disappear coupling abnormal noise. It is strongly recommended to perform phase analysis between two



Fig. 9: Typical on line monitoring system of screw compressors [13]

side of both couplings specially between gearbox and compressor the traditional strobe light method could help us in some urgent conditions but the disadvantages of this method is the danger of working with naked critical coupling (without cover coupling) [10]. Shaft crack is also one of the usual faults in both male and female or in main rotor of single screw compressors. Small size crack usually produce because of bad operation condition. The cracks may be longitudinal or radial shaft crack can be detect by monitoring of amplitude and phase of 1X and 2X and second harmonics of RPM in theory but it is hard to diagnose in complex FFT shape of screw compressors. Some modeling techniques and strategies developed in recent years to detect the shaft cracks in different complex rotor shape or process condition base on mathematical methods and related soft ware's [11]. The axial clearances of male and female installation also is one of the most challenging machinery concept in screw compressor maladjustment of rotors will ruin tolerances and it will cause abnormal axial vibration in machine effect of an axial force and shaft characteristics on the lateral natural frequencies of a flexible rotating shaft with a cubic nonlinearity is recently investigated. The shaft is assumed to be uniform and the Euler-Bernoulli theory is used to model the rotating shaft. Method of multiple scales is used to solve the dimensionless partial differential equation of the motion. Linear and nonlinear lateral natural frequencies are plotted for various shaft parameters and effects of these parameters and cubic nonlinearity is also discussed. In addition the natural frequencies are plotted as functions of damping coefficients, shaft characteristics, axial force and amplitude. Also, lateral natural frequencies increases by applying tension axial loading and decreases by applying compression axial loading at the ends of the rotating shaft [12].

## RESULTS AND DISCUSSION

In this part I want to explain a case history about oil injected twin rotary screw compressor that the ratio of male /female is 4/5. This screw compressor provide air for the special electronic tools of main process board facilities thus considering as most critical equipment. This equipment does not have timing gear and oil provide the only necessary rotating role. The coupling is rubber type, the electro motor is 38 KW and the gear box ratio is 3000/4500 increasing type. The schematic diagram of screw compressors shown in Figure 10. Also The Vibration limits base on vibration standard ISO2372 (BS 4675, VDI 2056) tables calculated and used in vibration monitoring trends shown in Table 1. These kind of standards usually work base on types of foundation (rigid or flexible ) and the power of driver in KW(size of equipment).

**Case history Air compressor Pj-K-2801 C Tuesday, March 19, 2013:** The overall vibration increasing in all parts, locations and directions considerably compare to previous trending data.the vibration data was as following table.

Due to high amounts of run frequency in compressors area the process problem is possible in this case therefore all air and oil controlling paths like unloading valves also all special electronic tools paths should be check accurately also base to all FFT, TWF, overall vibration and phase values trend evaluations the run frequency was dominate. Beside this due to the maintenance history male rotor fall from crane during balancing activates in balance shop. And it cause bending in male then the maintenance group sent male rotor for machining. Therefore the machining operation may cause hydrodynamic unbalance in screw compressor [14, 15, 16]. It was strongly recommended to check compressor unloading valve, its seat and plug and its cylinder and their piping systems for possible leakage conditions accurately. Figure 11 shown FFT in highest amplitudes measured in compressor. Run frequency also male frequency and its harmonics are dominating as you see in this figure also TWF of this point show high amount of impacts due to rotor hydrodynamic unbalance.

After appropriate maintenance checks and activities the air compressor/ Pj-K-2801 C start up in full load process condition at Thursday, April 4, 2013 and the result of the vibration checks was as following.

With taking a glance on last two tables and compare the data, all vibrations velocity, displacement and acceleration reduce considerably after recommended

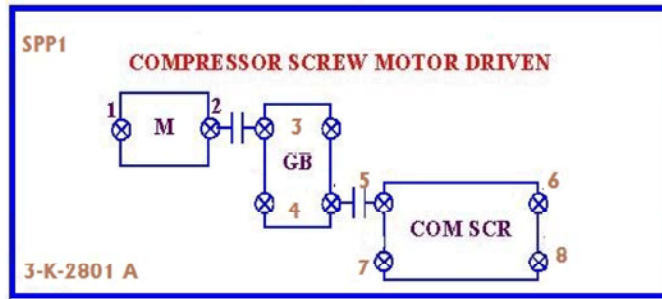


Fig. 10: Compressor screw motor driven vibration measuring points

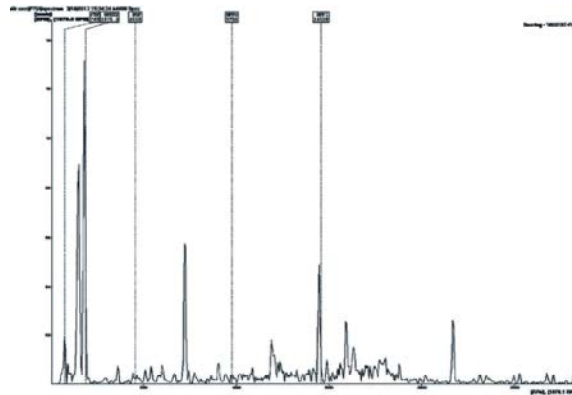


Fig. 11: FFT in highest amplitudes measured in air compressor/ Pj-K-2801 C

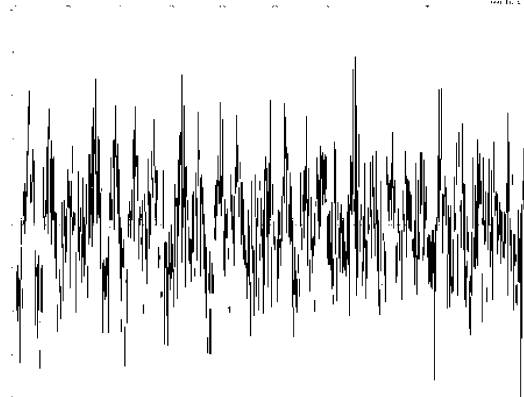


Fig. 12: Impact TWF in highest amplitudes measured in air compressor/ Pj-K-2801 C due to rotor hydrodynamic unbalance

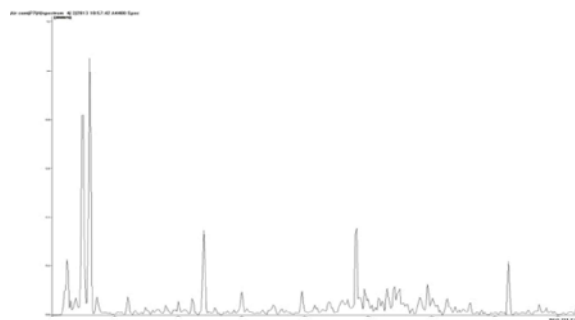


Fig. 13: FFT in highest amplitudes measured in air compressor/ Pj-K-2801 C after repair

Table 1: Screw compressors vibration limits.

Location	Vibration limits		
	Good condition	Alert condition (fair)	Danger condition (rough)
Driver	less than 2.8mm/s	between 2.8-7.1mm/s	above 7.1mm/s
Gear box	less than 4.5mm/s	between 4.5-11mm/s	above 11mm/s
driven	less than 4.5mm/s	between 4.5-11mm/s	above 11mm/s

Table 2: Highest Amplitudes Measured Air compressor / Pj-K-2801C

Highest Amplitudes Measured						
Position	Type	Displacement in micrometer p-p	Velocity in mm/sec (r.m.s)	Acceleration (m/s <sup>2</sup> )	Location	Health condition
Driver	Motor	19	2.3	5.1	Motor	Allowable
Gear box	Gear box	16	3	4.9	Gear box	Allowable
Driven	Compressor	19	2.8	4	Compressor	Tolerable

Table 3: Highest Amplitudes Measured Air compressor / Pj-K-2801 C after repair

Highest Amplitudes Measured						
Position	Type	Displacement in micrometer p-p	Velocity in mm/sec (r.m.s)	Acceleration (m/s <sup>2</sup> )	Location	Health condition
Driver	Motor	8.8	2	0.6	Motor	Good
Gear box	Gear box	9	1.6	3.7	Gear box	Good
Driven	Compressor	9	1.6	1.9	Compressor	Good

maintenance check and activities. Also the noise of screw compressor reduces too much and come back to its normal initial condition. Also there are no impacts in TWF of different compressor points any more. Run frequency, male frequency and its harmonics are dominate in highest amplitudes measured FFT in screw compressor yet but the peak amplitude reduce too much there fore because of the hydraulic unbalance history of this equipment rotor it is strongly recommended that this screw compressor will under close monitoring during necessary operations.

### CONCLUSION

All types of screw compressor vibration behavior are one of the most challenging topics in preventive maintenance. It is because of gear, bearing, male and female high frequencies mixed together in FFT or TWF of such machine also the noise pollution of such equipment cause these kinds of compressors as most bad actor machine in these types of equipment. The male and female rotor usually send to balance shop for balancing operation randomly. Because of this fact and the sensitive geometry of such male and female rotors. It causes some bending in some rotor areas. The further machining operation to improve tolerances cause possible hydraulic unbalance in screw compressor. The hydraulic unbalance show itself in run frequency and its harmonics. In such cases it is strongly recommend using the spare compressor as possible. But in the urgent maintenance

time it is recommended to check all air and oil controlling paths like unloading valves, its seat and plug and its cylinder and their piping systems for possible leakage conditions accurately.

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