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# Hydraulic Unbalance in Oil Injected Twin Rotary Screw Compressor Vibration Analysis (A Case History Related to Iran Oil Industries)

Omid A. Zargar

***Abstract*—**Vibration analysis of screw compressors is one of the most challenging cases in preventive maintenance. This kind of equipment considered as vibration bad actor facilities in industrial plants. On line condition monitoring systems developed too much in recent years. The high frequency vibration of ball bearings, gears, male and female caused complex fast Fourier transform (FFT) and time wave form (TWF) in screw compressors. The male and female randomly are sent to balance shop for balancing operation. This kind of operation usually caused some bending in rotors during the process that could cause further machining in such equipment. This kind of machining operation increased the vibration analysis complexity beside some process characteristic abnormality like inlet and out let pressure and temperature. In this paper mechanical principal and different type of screw compressors explained. Besides, some new condition monitoring systems and techniques for screw compressors discussed. Finally, one of the common behavior of oil injected twin rotary screw compressors called hydraulic unbalance that usually occurred after machining operation of male or female and have some specific characteristics in FFT and TWF discussed in details through a case history related to Iran oil industries.

***Keywords*—**Vibration analysis, twin screw compressor, oil injected screw compressor, time wave form (TWF), fast Fourier transform (FFT), Hydraulic unbalance and rotor unbalance.

## I. 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 Fig. 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].

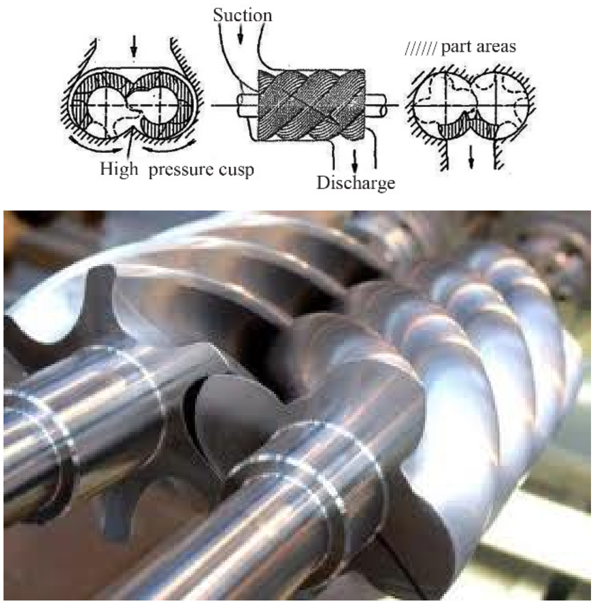


Fig. 1 Typical twin rotary screw compressor main rotor clearances down and basic operation principals up

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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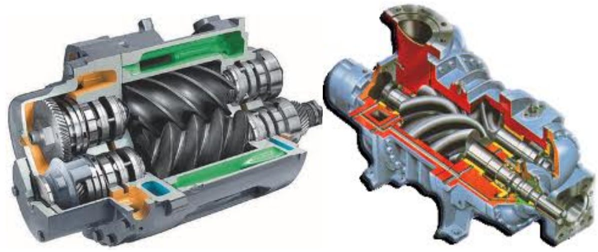

Fig. 2 Types of twin-screw compressors (left equipped with oil gear pump and right gearless)

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 meshing 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.

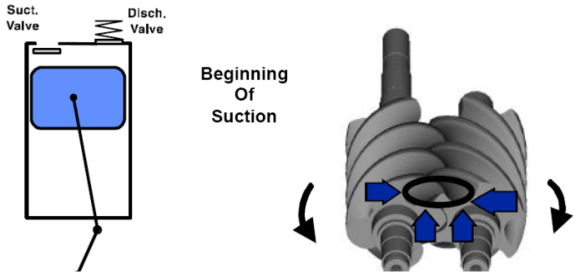


Fig. 3 How compressors are started suction of fluid in left reciprocating and right twin rotary screw compressor

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, (Vs). In the reciprocating analogy, the piston reaches the bottom of the stroke and the suction valve closes, trapping the suction volume, (Vs).

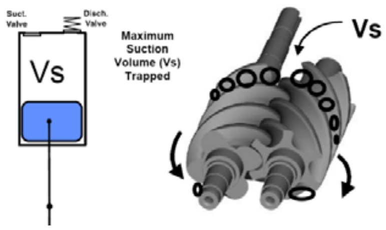


Fig. 4 When compressors have Maximum suction volume trapped of fluid in left reciprocating and right twin rotary screw compressor

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.

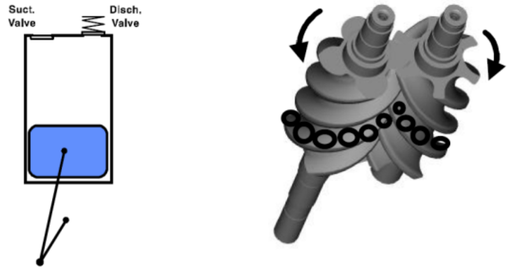


Fig. 5 How compressors are started Compression back the fluid in left reciprocating and right twin rotary screw compressor

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 reciprocating. That is starting up the cylinder and compressing the gas ahead of it.

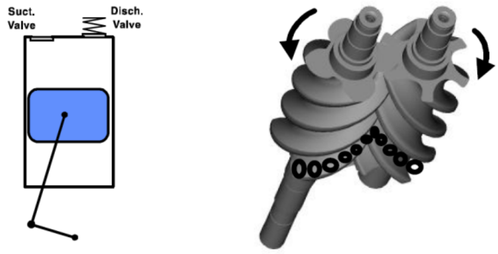


Fig. 6 How compressors are continued fluid Compression in left reciprocating and right twin rotary screw compressor

In the reciprocating 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, (Vd).

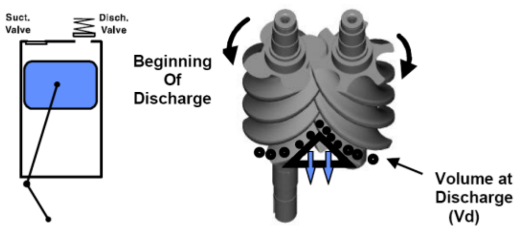


Fig. 7 How compressors are started fluid discharge in left reciprocating and right twin rotary screw compressor

A radial discharge port is used on the outlet end of the slide

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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].

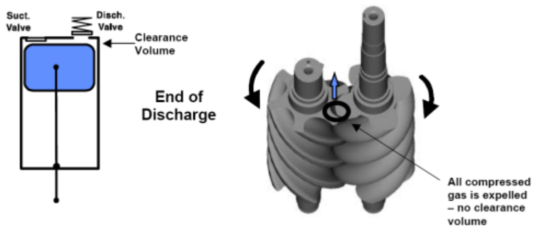


Fig. 8 How compressors are finished fluid discharge in left reciprocating and right twin rotary screw compressor

## II. Experimental Details

Screw compressors usually connected to an increasing RPM heavy gearbox with several main shafts and a high KW electromotor (usually more than 38 KW). The couplings are usually rubber type to damping the related vibrations. The foundation could be rigid or flexible in different industrial plants. The flexible foundation screw compressors vibration behavior is usually more complicated and that is because of the annoying noises generated. The monitoring system is connected to some on line vibration monitoring system in board substation.

Each part (motor-gearbox-compressor) has a separated vibration monitoring system.

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. Therefore, all vibration analysis techniques and methods like shaft centerline analysis [3] could be applied in critical conditions.

A new wireless condition monitoring system also developed in recent years. The main advantages of such system are in reducing the installation errors and increasing the monitoring accuracy and speed [4].

The gear boxes usually equipped with shock pulse measurement (SPM) on line monitoring [5]-[7]. The bearing condition unite (BCU) trends could be evaluated the condition of machine in different loads, speeds and situations [8]-[10].

In addition, traditional vibration measuring programs could be performed for all parts and components in screw compressors. It is worthy to know that these methods could be effective in gear box [11]-[13].

Besides, all vibration data such as overall vibrations, TWF, FFT and phase values could be evaluated in vibration analysis [14]. Besides, these methods could be effective in rotor unbalance identification [15], [16].

These kinds of data are available in on line vibration monitoring systems in most critical screw compressors.

Furthermore, a condition based monitoring system using ultrasonic signal recently developed for gear boxes. High frequency ultrasonic sensor is used as a transducer to collect data base on run frequency and gear mesh frequencies. Finally signals could filter within an ultrasonic range [17].

The phase analysis could be evaluated the condition of coupling or misalignment, soft foot and high low flanges. In addition, vibration modal analysis could be identified the machine overall condition with assist of data collector VDAU-6000. On the other hand, noise-monitoring systems usually performed in screw compressors.

Moreover, oil analysis is one of the other critical checkpoints always recommended in screw compressors.

Nowadays thermography developed to evaluating the bearing temperature in screw compressors and gearbox.

The lubrication system of different parts of screw compressors considered as a main technical category in monitoring systems. 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, cooled recrystallization and homogenization. The lubrication performance tests result 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 [18].

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) [19] for strongly nonlinear problems is used to give explicit analytic solution for lubrications problems in slider bearings [20].

These kind of bearings usually equipped 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 could effects directly on vibration limits.

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 like vibration modal analysis that is based on the phase values [21].

Different maintenance strategies such as corrective, time based, preventive, condition-based and predictive 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. Maintenance strategies have been modeled with consideration of four fuzzy parameters in the 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. This kind of modeling systems could be selected the types of maintenance strategy [22].

Reliability Centered Maintenance (RCM) is a process to ensure that assets continue to do what their users require in their present operating context. It is generally used to achieve improvements in fields such as the establishment of safe minimum levels of maintenance, changes to operating procedures and strategies and the establishment of capital maintenance regimes and plans. Successful implementation of RCM will lead to increase in cost effectiveness, machine uptime, and a greater understanding of the level of risk that the organization is managing.

RCM considered as a new revolution in maintenance strategies. Ball bearings and roller bearings are used both in screw compressors and gearboxes. Gears, male and female are also produced high frequencies. That is why the screw compressors FFT are usually complicated 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 are most effective methods in roller bearings and ball bearings fault diagnosis. The small hill shape type frequencies are appeared around bearing high frequencies in first stages. After that by developing bad bearing condition the frequencies shifted themselves to ball or roller pass frequency and its small sidebands. In this stage bearing is completely damaged and noises are appeared during operation [23].

Besides, monitoring overall acceleration and bearing condition units (BCU) could be effective in fault diagnosis. The thermography and sound analysis could be helpful in roller bearings and ball bearings fault diagnosis. These methods are developed too much in recent years. The couplings in both sides of gear box are considered as most challenging parts of screw compressors. The couplings usually are rubber types and produced run frequencies.

The couplings are usually under high tension. High vibration rates may loosen the rubbers, both in compressor and electromotor sides. These phenomena could produce coupling unbalance that is hard to diagnose in complex screw compressors FFT and TWF. In addition, coupling abnormal noises could be disappeared in noise pollution of screw compressor.

It is strongly recommended to performed phase analysis between two sides of both couplings especially between gearbox and compressor. The traditional strobe light method could be helpful in some urgent conditions but the disadvantages of this method is the danger of working with naked critical coupling (without cover coupling) [24].

Shaft crack is also is 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 could be detected by amplitude, 1X, 2X phase and second harmonics of RPM monitoring. Shaft crack is hard to diagnose in complex FFT shape of screw compressors.

Some modeling techniques and strategies developed in recent years and could be detected the shaft cracks in different complex rotor shape or process condition base on mathematical methods with assist of technical software's [25].

Moreover, the axial clearances of male and female installation are considered as most challenging machinery concepts and maintenance issues in screw compressors.

Maladjustment of rotors will ruin tolerances and caused abnormal axial vibrations. Effect of an axial force and shaft characteristics on the lateral natural frequencies of a flexible rotating shaft with a cubic nonlinearity is also 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 discussed. In addition, the natural frequencies are plotted as damping coefficients functions. In addition, lateral natural frequencies increases by applying tension axial loading and decreases by applying compression axial loading at the ends of the rotating shaft [26].

Fig. 9 Typical on line monitoring systems for screw compressors equipped with contact piezoelectric sensors

## III. Result and Discussion

In this part a case history about oil injected twin rotary screw compressor explained. The ratio of male /female is 4/5.

This screw compressor provided air for the special electronic tools of main process board facilities and considered as most critical equipment. This compressor has not timing gear and oil provided by male and female rotation.

The couplings are rubber type. The electro motor is 38 KW and the gear box is 3000/4500 (increasing). The schematic diagram of screw compressors is shown in Fig. 10. In addition, vibration limits calculated based on standard ISO2372 (BS 4675, VDI 2056).

Values are shown in Table I. These kinds of standards usually work based on foundation type (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 vibrations were increased in all parts, locations and directions considerably compared to previous trending data. The vibration data are shown in Table II.

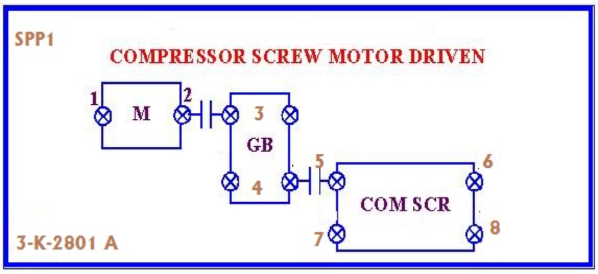


Fig. 10 Twin rotary screw compressor motor driven with increasing gear box vibration-measuring points

TABLE I
Screw Compressors Vibration Limits

| Location | Good condition | Alert condition (fair) | Danger condition (rough) |
|---|---|---|---|
| | Vibration limits | | |
| Driver | less than 2.8mm/s | between2.8-7.1mm/s | above7.1mm/s |
| Gear box | less than 4.5mm/s | between4.5-11mm/s | above11mm/s |
| driven | less than 4.5mm/s | between4.5-11mm/s | above11mm/s |

TABLE II
Highest Amplitudes Measured Air compressor / Pj-K-2801 C

| Position | Type | displacement in micrometer p-p | Velocity in mm/sec (r.m.s) | Acceleration (mm/s2) | Location | health condition |
|---|---|---|---|---|---|---|
| | | | Highest Amplitudes Measured | | | |
| Driver | motor | 19 | 2.3 | 5.1 | motor | Allowable |
| Gearbox | Gear box | 16 | 3 | 4.9 | Gearbox | Allowable |
| Driven | compressor | 19 | 2.8 | 4 | compressor | Tolerable |

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 and all special electronic tools paths should be checked accurately.

Furthermore, the run frequency was dominated based on all FFT, TWF, overall vibration and phase values. Besides, male rotor is fall from crane during balancing activates in balance shop according to the maintenance history. In addition, falling caused bent shaft in male. Therefore, the maintenance group sent male rotor for machining.

The machining operation caused hydrodynamic unbalance in screw compressor. It was strongly recommended to check compressor unloading valve, its seat, plug, cylinder, and their piping systems for possible leakage accurately. Fig. 11 shows FFT in highest amplitudes measured in compressor. Run frequency, male frequency and their harmonics are dominated. In addition, TWF shows high amount of impacts because of rotor hydrodynamic unbalance.

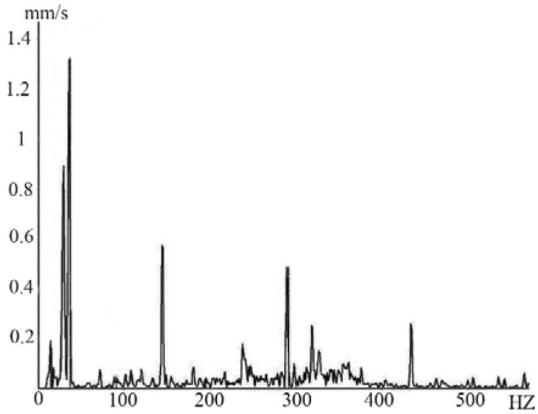


Fig. 11 FFT in highest amplitudes measured in air compressor/ Pj-K-2801 C

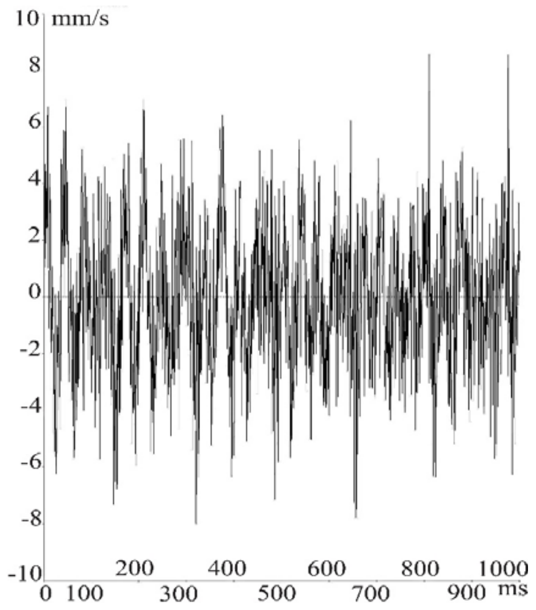


Fig. 12 Impacted TWF in highest amplitudes measured in air compressor/ Pj-K-2801 C (influenced by rotor hydrodynamic unbalance)

Recommended maintenance checks and activities applied. The air compressor/ Pj-K-2801 C started up in full load process condition at Thursday, April 4, 2013. The startup vibration values are shown in Table III.

TABLE III
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 (mm/s2) | Location | health condition |
| Driver | motor | 8.8 | 2 | 0.6 | motor | good |
| Gearbox | Gear box | 9 | 1.6 | 3.7 | Gearbox | good |
| Driven | compressor | 9 | 1.6 | 1.9 | compressor | good |

By taking a glance on last two tables and compared the data, all vibrations velocity, displacement and acceleration reduced considerably after recommended maintenance checks and activities.

In addition, the noise of screw compressor reduced too much and come back to its normal initial condition. Moreover, there are no impacts in TWF of different compressor points any more. Run frequency, male frequency and its harmonics are dominated in highest amplitudes measured FFT in screw compressor yet but the peak amplitudes and their sidebands reduced too much.

Furthermore, it is strongly recommended that this screw compressor considered under close monitoring during necessary operations because of rotor hydraulic unbalance history.

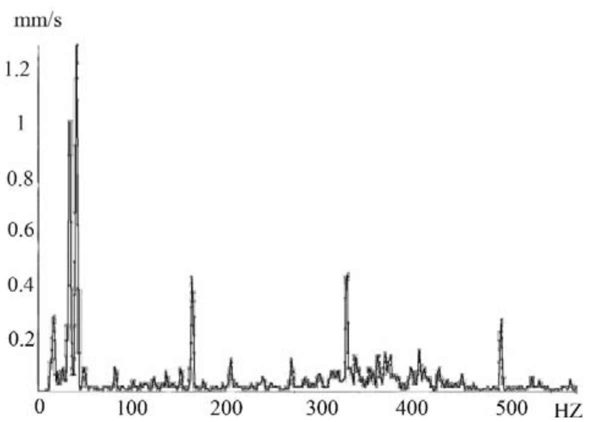


Fig. 13 FFT in highest amplitudes measured in air compressor/ Pj-K-2801 C after repair

## IV. CURRENT AND FUTURE DEVELOPMENT

All types of screw compressors vibration behavior considered as most challenging topics in preventive maintenance. That is because of gear, bearing, male and female high frequencies mixed together in FFT or TWF.

In addition, the noise pollution of screw compressors caused monitoring problems. The male and female rotor usually sent to balance shop for balancing operation. This usually caused bent shaft because of male and female sensitive geometry.

The further machining operation for improving tolerances caused hydraulic unbalance in screw compressor. The hydraulic unbalance shows itself in run frequency, its harmonics and sidebands. In such cases, it is strongly recommended to use the spare compressor if possible.

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

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