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Vibration Analysis of Rotary Compressor Was Evaluated Experimental and Numerical Analysis

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ABSTRACT

This paper deals with By analyzing the motion of moveable machine elements in rolling piston rotary compressors, the rotatory behavior of the crankshaft and the piston was clarified and constraint forces and sliding speed at each pair were determined. Furthermore, unbalanced inertia forces and compressor vibrations were evaluated. It is concluded that theoretical results are in good agreement with experimental ones. Moreover, it was revealed that one of the major factors, which cause compressor vibrations, is speed variation of the crankshaft caused by a pressure fluctuation in the cylinder. By theoretically analyzing dynamic behavior of the crankshaft, the rolling piston and the blade in Rolling-Piston Rotary Compressors, constraint forces and sliding speed at each pair of movable machine elements are obtained, and unbalanced inertia forces and compressor vibrations are evaluated. Structural behavior (Static and Dynamic) for compressor housing with and without Base frame are estimated using experimental and Ansys is validated.

INTRODUCTION

Rolling-piston rotary compressors have the advantages of high volumetric efficiency and small mechanical loss and they are compact and light in weight, compared to corresponding reciprocating compressors.
In rotary compressors, moreover, vibrations are comparatively small in amplitude as they have few reciprocating elements, and hence have been considered suitable for lowering the noise in air-conditioning equipment. Due to these properties, most air-conditioning compressors presently used in Japan are of the rolling-piston rotary type.

It is likely that the popularity of rolling-piston compressors will continue to increase, and at the same time strong demands for reducing vibration and noise which arise from the compressors will also ~is. To cope with these demands, unbalanced inertia forces due to the motion of machine elements, and vibrations caused by those forces have to be evaluated before a design which reduces the revealed vibrations most effectively can be developed. In this paper, an analytical method to evaluate the vibrations is established, and the experimental confirmation is shown. Movable machine elements in a rolling-piston compressor the rotating crankshaft, the rolling piston and the reciprocating blade. Each machine element moves in connection with the others. Now, the blade motion is a function of the turning angle of the crankshaft, provided that the blade top moves in contact with the piston.

Rolling-Piston Rotary Compressor:

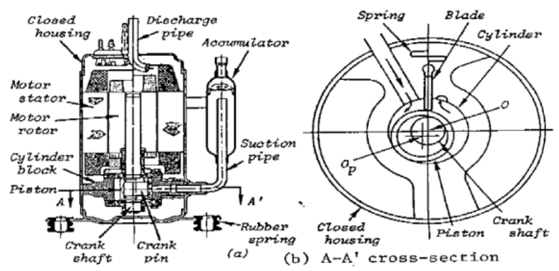

Fig.1 Construction of Rolling-Piston Compressor

Fig.l shows the construction of a rolling-piston rotary compressor which is used for air-conditioners of the refrigerating capacity 1755 kcal/h.

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The motor stator and the cylinder block are fixed inside the closed housing which is suspended with three rubber springs on a base. The refrigerant (R22) is sucked into the cylinder through the accumulator. The compressed refrigerant is discharged inside the closed housing and transferred to a condenser through the discharge pipe on the top of the closed housing. The dimensions of the closed housing are 110 mm diameter and 212 high, and the mass of the whole compressors 8.7 kg. The motor is a single-phase induction motor. The synchronous speed is 3600 rpm and the power is 0. 55kW. The machine part compressing the refrigerant is soaked in the lubricating oil and the gas leakage from the compression chamber is prevented by oil sealing. Fig.l(b) shows the A-A' crosssection of the machine part. The machine part consistsof the cylinder with bore 39 mm, the reciprocatingblade with thickness 3.2 =, the piston withoutside diameter 32.5 mm and the crankshaft systemwhich is composed of the crankshaft, the crankpin andthe motor rotor.

The eccentricity of the piston centreOp from the rotating crankshaft centre0 is 3.26mm and the cylinder depth is 28 mm. The center axisof the blade coincides with the cylinder center 0 andthe blade tip with radius 3. 2 mm is pushed on the pistonby the spring force and the gas force which areexerted on the back end. The minimum value of thepiston-cylinder wall clearance is about 10 vm. Thepiston-crankpinclearance is about Jill and thispair is lubricated by an oil pump attached to thelower end of the crankshaft. The arrows shown in thefigure express the direction of the refrigerant gasflow. The refrigerant is sucked in the suction chamberand discharged inside the closed housing aftercompressed in the compression chamber.

LITERATURE REVIEW

The performance test of a modified miniature rotary compressor in upright and inverted modes subjected to microgravity Rui Ma Yu-ting Wu, Chun-Xu Du, Xia Chen, De-lou Zhang, Chong-fang Ma Vapor compression heat pump is a new concept of thermal control system and refrigerator for future space use. Compressor is a key component in the vapor compression heat pump. Development of compressor capable of operating in both microgravity (10 E-6 g) and lunar (1/6 g) environments is urgently needed for space thermal control systems based on heat pump technique. In this paper, a miniature rotary compressor by ASPEN company was modified to realize acceptable compressor lubrication and oil circulation in microgravity environments. An experimental system was built up to check the performance of the modified compressor subjected to microgravity. A performances comparison of inverted compressor with upright one was made. The influences of operating parameters such as refrigerant charge, cooling water temperature as well as compressor speed on the performances of vapor compression heat pump were investigated. The results show that the modified miniature rotary compressor in inverted mode can operate stably in a long period, which indicates that the modified compressor can be employed in microgravity environments.

Compressor discharge temperature increased or decreased while COP changed more obviously with cooling water temperature and speed in microgravity. In most cases ,performance of the upright compressor is superior to that of the inverted one. But when the compressor speed is from 1500 rpm to 2500 rpm or the coolant temperature is between 20 and 25 degrees, the performance of inverted compressor is better. The highest discharge temperature of the inverted compressor can be as high as 1.7 times that of the upright one. The maximum of heating COP and cooling COP of upright compressor is respectively 1.21 and 1.16 times that of inverted compressor. The study of modified miniature rotary compressor on gravity-independence provides an experimental basis and laysfoundation for space applications.

Article · January 1970 author Noriaki Ishii By theoretically analyzing dynamic behavior of the crankshaft, the rolling piston and the blade in rolling-piston rotary compressors, constraint forces and sliding

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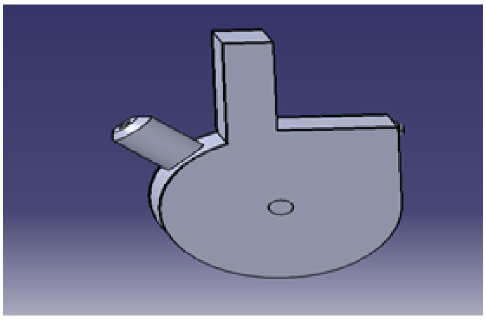
speed at each pair of movable machine elements were obtained, and unbalanced inertia forces and compressor vibrations were evaluated. It was concluded that theoretical results have a good agreement with experimental ones. Moreover, it was revealed that one of major factors which cause compressor vibrations is speed variation of the crankshaft and compressor vibrations are not affected by rolling behavior of the piston.

Experimental analysis:

Fig shows a schematic view of the experiment compressor. The rotational fluctuations of the rotating part are measured with an eddy current type displacement sensor fixed on the upper shell which detects the rotation of a gear mounted on the shaft. On the other hand, the tensional vibration of the stationary part about the shaft axis is measured with two eddy current type displacement sensors fixed on the base whichdetect displacements of the plates radically fixed on the shell. During the experiment, the compressor is supported by three rubber mounts on the base and linked with flexible pipes to a refrigeration cycle using R22(CHCIF2) as the working fluid at first vibration at a steady state operation are measured. Next the electric power to the compressor is powered on again and vibration during the stopping operation are measured. Operating pressure at steady state. The main dimensions of the compressor and the value imperial constants as shown in the table. The compressor is driven by a single phase two pole induction motor which power source of 100v 60hz. Its electrical characteristics are shown in fig



MODEL: rotary compressor catia model

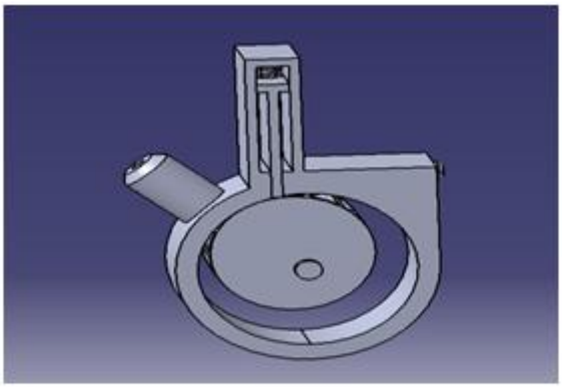


Fig: : inner process of compressor catia model

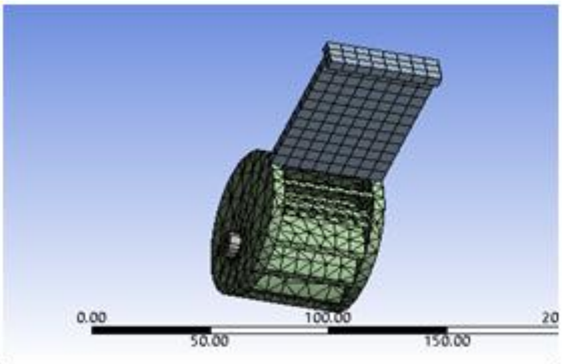


Fig: ansys model

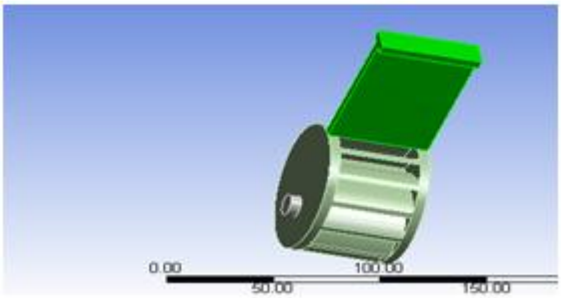


Fig: ansys model

SOLUTION AND RESULTS

The aim of modal analysis for rotor-bearing system of the rotor compressor by using the finite element software is to find out the natural frequency and its corresponding vibration mode shapes of the system. These parameters are mainly influenced by the structure types and constraint conditions without loads. In this paper the models were analyzed by using the block Lanczos method provided by ANSYS package to determine the eigen value of the Equation (1), and

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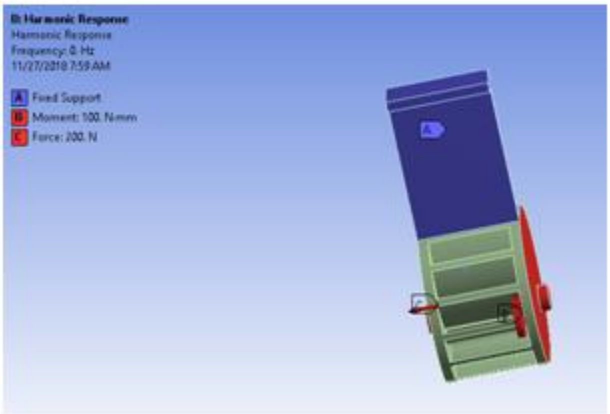
set the order number of modal extraction for solution. The bearing condition was terrible meanwhile all traditional signals like overall vibrations, BNC data, TWF, FFT, phase trends and gap voltages were shown nothing abnormal. These data were not shown any dramatic increase. Nevertheless, the shaft centerline monitoring method was provided enough evidences for suitable and on time recommendations. This method caused successful fault prediction.

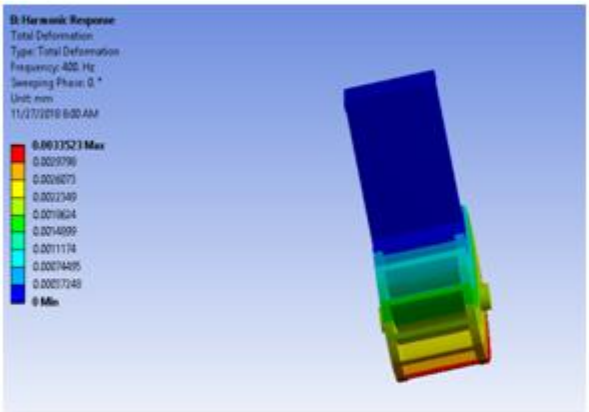
If CM group did not perform shaft centerline monitoring, the alignment will faced serious problems in next step. The valuable turbine blades might have some touches or damages. Furthermore; some damages might occurred in multi stage compressor parts. This would cause high economical maintenance and production costs for the factory (petrochemical zone).

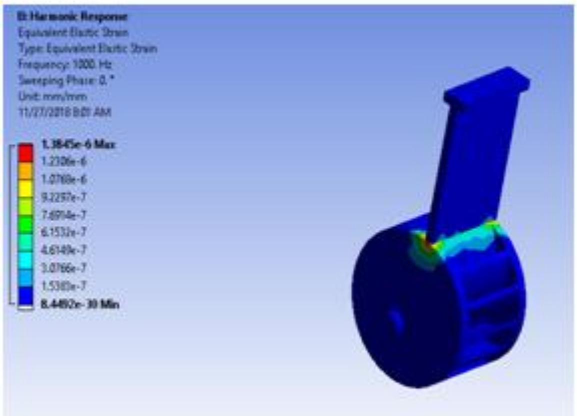
In addition, the plant might face an unexpected shut down for several hours. The process may have 4 to 5 hours high-pressure work to start up the spare compressor. Moreover, thousands of dollars wasted in these kinds of shutdowns (because of quit of the Olefin production). In addition, Olefin is the base material for tens of other petrochemical companies in the petrochemical zone. These plants faced serious problems in production.

Besides, petrochemical companies should pay some delay fines to the export ships that coming from far countries every hour. All in all this 4 to 5 hours may costs thousands of dollars wasted in production. Beside this, mechanical part damages like turbine parts could waste large amount of time and money. All these evidences have shown the effectiveness of shaft centerline method for future applications.

Case 1: Rotor Rotation angle (degrees) at 0deg







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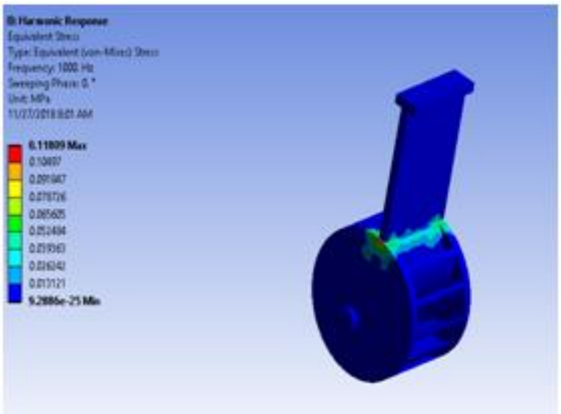
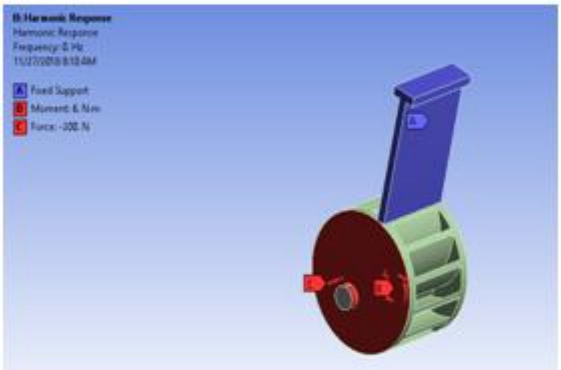
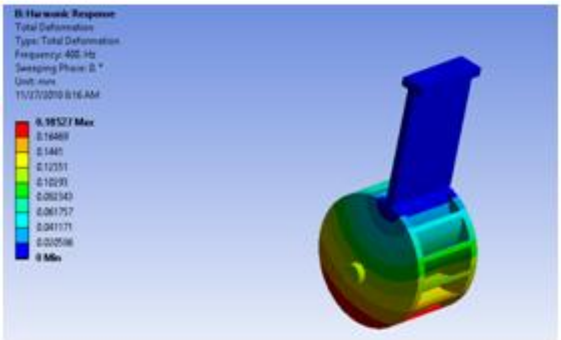
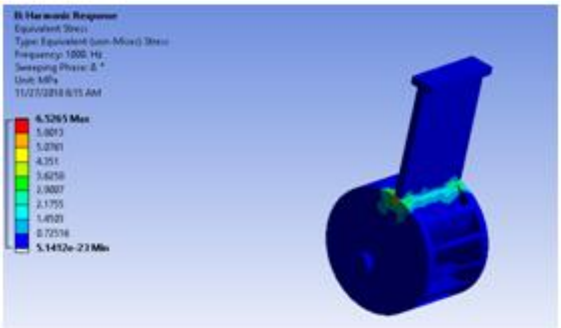
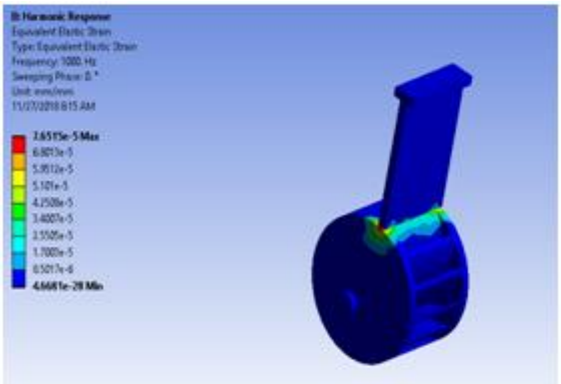


CASE 2:Rotor Rotation angle (degrees) at 180 deg



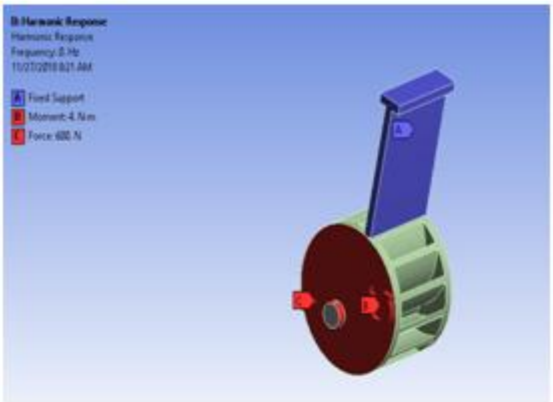


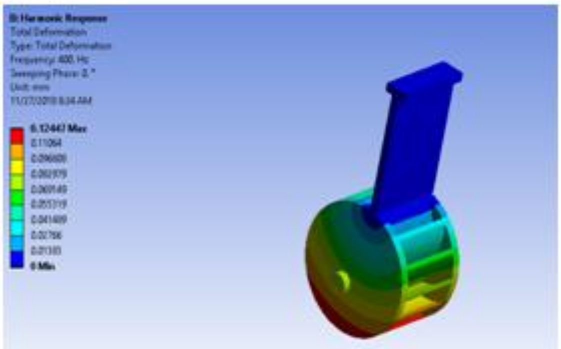




CASE 3:Rotor Rotation angle (degrees) at 270deg

	CASE	DEFORMATION (MM)	STRESS (MPA)	STRAIN (MM/MM)
1	C1	0.003	0.12	1.38E-06
2	C2	0.185	6.52	7.65E-05
3	C3	0.12	4.38	5.14E-05





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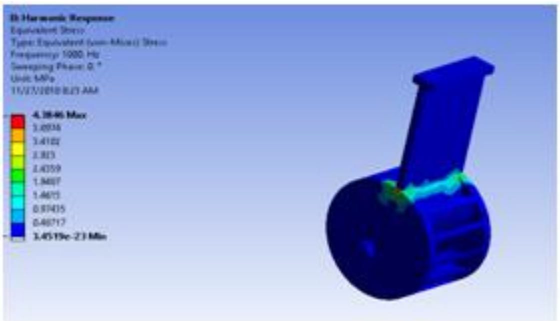

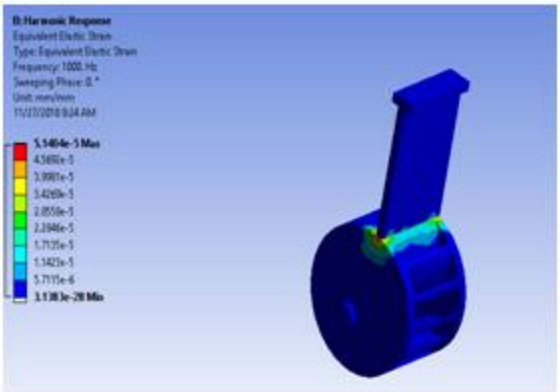

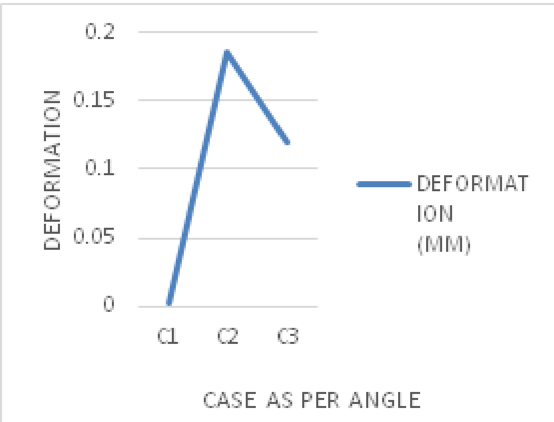

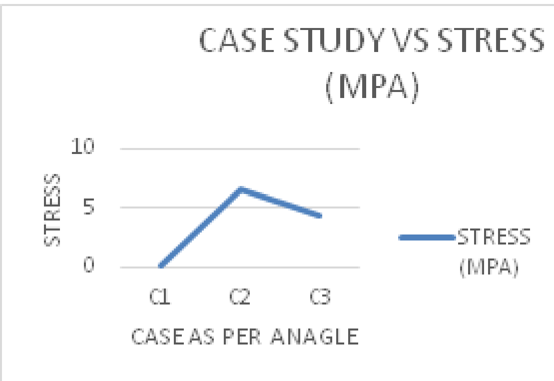

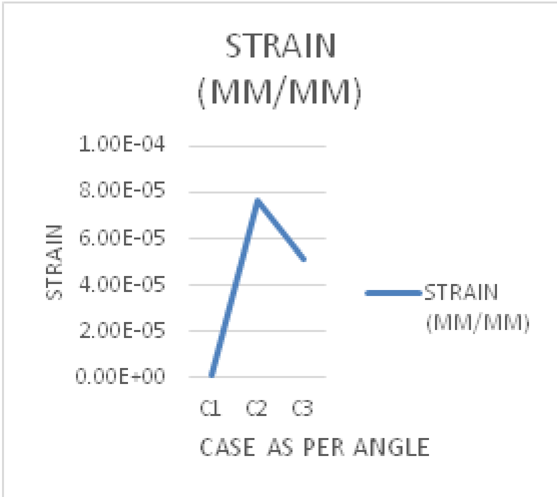

COMPARISION BETWEEN THEARITICAL AND EXPERIMENTAL ANALYSIS:

To examine the calculated results, the compressor vibration on the cylindrical r-loosed housing is measured and it is compared with the computer simulated results. The measured point is on the horizontal plane which passes the cylinder center 0, and it has the coordinate (5.04, -2.10, -6.65 em). The measured directions are the tangential and the normal (called 'radial') to the cylindrical shell, on the above horizontal plane. Fig.l7 shows the experimental results, in which (a) shows the radial vibratory acceleration and (b) the tangential. The calculated results corresponding to the above experimental results are presented in Fig.l8, in which the solid line shows the tangential acceleration Xst and the dotted line the radial Xsr· The calculated results cannot simulated the vibratory components of higher frequency, and it is seen that the measured vibration forms of the lower frequency are closely simulated by the computer calculation. Fig.l9 shows a comparison of the vibration power spectrum. The abscissa is the frequency order and the fundamental frequency is 57 .1 Hz. The ordinate is the vibration level expressed by decibel. 0 dB shows 1. 0 m /s 2 • The solid lnes show the calculated results, and 0 sings show the power spectrum of the measured tangential acceleration which was analyzed by the first fourier translator (Nicolle660).

From this comparison, it is seen that the calculated results simulate precisely the measured vibration components which frequency order is lower than eight. The shaft centerline analysis performed successfully for main steam turbine multistage compressor MPC-C-8001 related to an Olefin plant in Iran oil industry (Maron petrochemical company) for point 3-compressor drive end bearing. The traditional methods like overall trends of vibration displacement micrometer peak to peak (P-P) by Bently Nevada board in substation, phase trends, TWF and FFT monitoring did not indicated any dramatic increase and enough evidence to any machinery maintenance recommendation (for example checking the bearings). The shaft center line analysis predicted the strong touch in down half of the bearing that was completely true.

Besides the shaft center line method could predict the exact location of touches successfully. After maintenance, the bearing changed, reinstalled and replaced. The main shaft was sent to metal spray work shop. This successful analysis caused protecting potential unexpected shutdowns that might pose huge mechanical and production costs on the factory. Therefore shaft centerline analysis could be effective if performed accurately and correctly on most critical equipment like main turbine compressors. In conclusion, rotor movement evaluation by shaft centerline method considered as one of the most effective tools in predictive maintenance and condition monitoring systems. Besides, low vibration does not always indicate a healthy machine.

The process condition of both turbine and compressor such as suction and discharge pressure and inlet and outlet temperature were remained constant. These values were remained in the range of turbine and compressor technical documents (according to the data trends in main board of olefin plant). In addition, the quality of steam was acceptable and remained in the range of turbine documentation.

Therefore, the problem of abnormal noise in point 3 should be mechanical rather than something related to the process. Main turbine compressor MPC-C-8001 working in 11000 RPM. This machine equipped with Bently Nevada 3500 series monitoring system. MPC-C-8001 is the heart of an olefin plant in Iran oil industry (Maron petrochemical company). The overall vibrations are measured in micrometer peak-topeak (P-P). The alert limit is 20 micron and the danger is 40. Then, less than 20 is good and between 20 and 40 is fair. Alarm lamp is appeared in BNC board in substation in fair conditions (alert). More than 40 is rough condition and compressor is automatically tripped to protect the rotary and stationary part of turbo compressor.

CONCLUSSION

By exact analysis of the dynamic behavior of the movable machine elements in rolling-piston rotary compressors ,a method of vibration analysis of the rotary compressors was presented, and it was applied to a small rolling-piston rotary compressor with a motor power of 550 W which is widely used for air-conditioners with the refrigerating capacity of 1755kcal/h. The conclusions obtained in these study areas follows:

(1) The speed variation of the crankshaft was about7.8% and the fluctuating peak to peak value of therotatory acceleration was 13350 rad/s2 • The fluctuatingwave from of the rotatory acceleration was closely related to that of the gas pressure in the282-compression chamber. The rotatory acceleration of the piston rapidly changed at the time t = 4.4 msand 12.1 ms when the direction of the sliding speed at the piston-blade pair changed, and the fluctuating value reached about 13540 rad/s2.

(2) The characteristics of the fluctuating constraintforces were revealed, and hence fundamental design criteria for manufacturing compressors which are more compact and lighter in weight were obtained.

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(3) The calculated results of the compressor vibrations were able to precisely simulate the measured vibration components which frequency order is lower than 8th. One major factor inducing compressor vibrationsis an unbalanced inertia force based on the fairly large speed variation of the crankshaft andhence the vibration component about the crankshaft center is fairly large in amplitude comparedwith the other vibration components.

(4) When only the compressor vibrations are discussed, the analysis of the piston rotator motionis negligible, since the inertia moment of the pistonis fairly small compared with of the rotatingcrankshaft system in general. Hence the method for vibration analysis can be fairly simplified.

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