N86-30170

CO_2 COMPRESSOR VIBRATION AND CAUSE ANALYSIS

Ying Yin Lin
Liaohe Chemical Fertilizer Plant
Panjin, Liaoning Province
The People's Republic of China

This paper covers the operational experience of a large turbine-driven CO_2 compressor train in a urea plant with capacity of 1620 tons/day.

After the initial start-up in 1976, the vibration in the HP cylinder was comparatively serious. The radial vibration reached 4.2 to 4.5 mils and fluctuated around this value. It was attributed to the rotating stall based on our spectrum analysis. Additional return line from the 4th to 4th and higher temperature of the 4th inlet has cured the vibration.

This paper also describes problems encountered in the operation, their solutions, and/or improvements.

1. INTRODUCTION

The CO_2 compressor described here is a key installation of whole sets of the equipment in a 1620 ton/day urea plant. The steam turbine-driven compressor consists of two cylinders and four stages as shown in the flow diagram, Figure 1.

CO_2 gas from the CO_2-removing section of the ammonia plant is pressurized in this compressor to 144-150 kg/cm^2, sent to an HP condenser via a stripper and finally reacts with liquid ammonia forming urea. Main parameters and characteristics of the unit train are listed in Table 1.

2. CO_2 COMPRESSOR VIBRATION

From the commissioning in September, 1976, to the overhaul in 1980, vibration of the CO_2 compressor remained the bottleneck in urea production. During commissioning, radial vibration was found at comparatively high value, on the HP cylinder in particular.

The CO_2 compressor train has nine radial probes for vibration sensing on its turbine and gearbox and three axial displacement probes on the rotor's shaft of the LP cylinder, HP cylinder and turbine: twelve points in total.

1st	point	turbine	front	bearing	radial	probe
2nd	point	turbine	rear	bearing	radial	probe
3rd	point	LP cylinder	front	bearing	radial	probe
4th	point	LP cylinder	rear	bearing	radial	probe
5th	point	gearbox	front	bearing	radial	probe
6th	point	gearbox	rear	bearing	radial	probe
7th	point	HP cylinder	front	bearing	radial	probe

8th and 9th points: HP cylinder rear bearing radial probe (symmetrical about vertical axis and 90° included angle)

10th, 11th, 12th points: LP and HP cylinders, turbine rotor axial displacement probes.

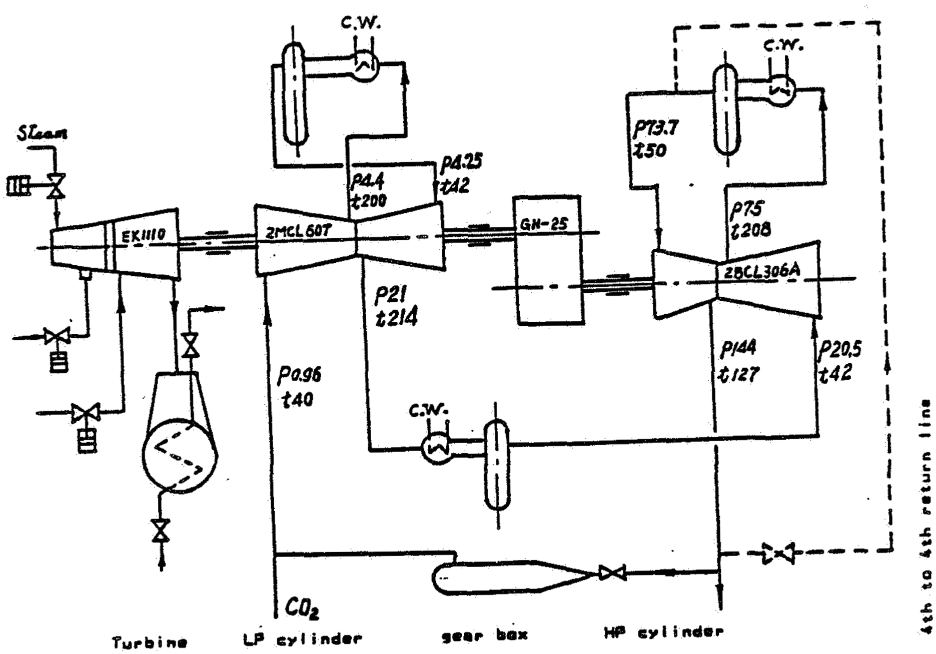

Figure 1. FLOW DIAGRAM OF CO_2 COMPRESSOR

description	LP cylinder		HP cylinder	
	1st stage	2nd stage	3rd stage	4th stage
inlet P kg/cm2	0.96-1.02	4-4.5	20-24	74-80
outlet P kg/cm2	4-4.5	20-24	74-80	144-150
inlet T °C	40	42	42	40-45(55-60)
outlet T °C	220	220	200	120-130
impeller No.	3	4	4	2
rated speed RPM	7200		13900	
max. continuous speed RPM	7560		14600	
1st order critical speed RPM	4100		8000	
2nd order critical speed RPM	9200		17500	
designed cylinder P kg/cm2	26		195	
designed cylinder T °C	250		250	
working medium	CO2			
rated power KW	7210			
molecular weight of CO2	43.17			
inlet flow NM3/h	27636			

Table 1. Main Parameters and Characteristics of the CO_2 Compressor

The unit train has proved satisfactory in producing 1620 tons of urea per day since commissioning; however, some problems were also found during these years. In September, 1976, the O-ring between the 3rd and 4th stages was blown off by the CO_2 stream; it thus caused increased axial displacement of the HP rotor and thrust bearing shoes burnt. During the overhaul in 1978, the brace ring of the last stage and the next to last were found broken. In April 1979, scalings on the LP cylinder impeller and subsequent ill balance of the rotor brought the radial vibration up to 5 mils, resulting in a forced shutdown and replacement of the rotor.

Drains and instrument tubes of the HP cylinder were broken by vibration several times. Vibration on the HP cylinder was one of the most difficult problems. Shortly after commissioning, vibration on the 7th point was as high as 2.7 mils which was already beyond the limit specified in API 617. However, the HP cylinder was still kept running at this high vibration value since the CO_2 stream of greater molecular weight had significant influence on rotor running, and the specified vibration range of API 617 is applicable during manufacturer's shop test rather than end user's production. The vibration value in the HP cylinder went higher in February 1979 and fluctuated sometimes. Table 2 gives the record made on 26 February.

Table 2. Vibration Record on February 26, 1979

<table>
<tr><td></td><td colspan="2">1st stage</td><td colspan="2">2nd stage</td><td colspan="2">3rd stage</td><td colspan="3">4th stage</td></tr>
<tr><td>P kg/cm2 inlet/outlet</td><td colspan="2">1.1/3.8</td><td colspan="2">3.8/24.5</td><td colspan="2">24.5/82</td><td colspan="3">82/140</td></tr>
<tr><td>T °C inlet/outlet</td><td colspan="2">32/210</td><td colspan="2">30/250</td><td colspan="2">29/160</td><td colspan="3">47/112</td></tr>
<tr><td>point/mils</td><td>1/1.2</td><td>2/0.4</td><td>3/0.8</td><td>4/0.6</td><td>5/0.7</td><td>6/0.7</td><td>7/2.5</td><td>8/2.9</td><td>9/4.5</td></tr>
</table>

Rotor vibration of the HP cylinder was indicated by the 7th point's high value, sometimes by the 8th and 9th point's values or high values of these three points. During the replacement of the LP cylinder rotor in April 1979, the bearing of the HP cylinder was readjusted. The bearings of 8th and 9th points were replaced. The clearances between shoes and shaft were smaller (=0.124), whereas the shoe clearance of 7th point became a little bigger (=0.175), as no replacement was made on it. The normal clearance would be 0.12-0.155. After start-up, the vibration value on the 7th point was 4.0-4.2 mils and 0.7-0.8 mils on the 8th and 9th points. But the vibration of these three points went towards the same level half a month later, i.e., 3.4-4.0 mils. Since then, each time the train was temporarily shut down and started up without replacement or adjustment, vibration values of the three points did have small changes but fluctuated synchronously ranging between 0.5-1.5 mils. Intervals between adjacent fluctuations were not equal; the shortest was several times per minute, the longest fifteen times per minute. The most severe case occurred after the overhaul in 1979. Drain tubes and process lines vibrated as the HP cylinder vibrated. One of the drain tubes in the HP cylinder was vibrated to breakage four times around the weld's heat-affected zone and resulted a forced shutdown.

3. VIBRATION CAUSE ANALYSIS AND SOLUTION

3.1 Imperfect Balance of Rotors

The changes of vibration during running were found to have a certain relationship to rotor speed and could be attributed to its imperfect balance. Since it was not possible to perform dynamic balancing at working speed and the urea product was in urgent demand then, the test on the HP cylinder was delayed until the overhaul in

1980. The vibration of the HP rotor decreased a little, but was far from elimination.

3.2 Misalignment of the Unit Train

Poor hot alignment of the unit train shafts connected could provide a cause for vibration. Therefore, realignment of the coupling between the HP cylinder and gearbox was performed during several overhauls but it made little improvement. During overhaul, inspection of the rotor before disassembly indicated that the points of high vibration lost their original alignment values, as did the other points. Based on our experience during these years, the alignment of the train gives little effect to the vibration with the self-centering gear coupling. So alignment between rotor connection does have some tolerance but shouldn't be neglected. Misalignment was not the main cause of HP cylinder vibration. Stresses of pipelines and large deviation when resting longitudinal slide key between casings were first suspected as the source of the unit's misalignment and the vibration of the HP cylinder. The stainless steel pipe at the CO_2 inlet of the 4th stage had to be replaced due to serious corrosion. During the replacement, fabrication errors resulted 10 mm offset between centerlines of the pipe flange and the equipment flange. In order to produce urea earlier than expected, the flanges were forced to alignment and then bolted. During the overhaul in 1979, the piece of pipe was removed, cut, and rewelded according to the requirement of initial routing until full alignment was reached without residual stress of piping system to the unit train. Besides this, regularity was found when realigning, i.e., HP cylinder changed its position toward gearbox. As the position of resting longitudinal slide key was doubtful, a pow jack was used to force alignment. Once the train was put into operation, the original position as restrained by the slide key was restored and hot alignment changed. During the overhaul in 1978, two longitudinal slide keys were chiselled off to free the unit from restriction and rewelded after alignment.

Two measures taken, as mentioned above, didn't eliminate the vibration of the HP cylinder.

3.3 Possibility of Oscillation of Oil Film

The radial shoes of the HP cylinder are of the tilt type and are capable of protecting the unit from vibration and oscillation of oil film. The oscillation couldn't be terminated unless the viscosity of the lub oil and the load on bearing shoes were changed. Although the temperature of the lub oil, i.e., its viscosity, was changed, the vibration value remained the same as before. In addition, 0.4-0.5 multiple frequency did not appear during frequency measurement. Therefore, the vibration of HP cylinder was not caused by oscillation of oil film.

3.4 Influence of Pulsating Gas Flow

In order to reduce CO_2 inlet temperature and raise gas transferring capability of the unit, a directly countflowing water cooler had been added on the inlet pipeline. Its side effect was introducing C_1 into the CO_2 system via cooling water, hence, the corrosion and leakage of stainless steel tubes in CO_2 intercoolers. Among them the 3rd cooler came first and received the worst damage. Did this leakage and pulsation of gas flow in CO_2 system cause the vibration? Furthermore, leakage of the relief valve on the 4th stage outlet was found several times. Sometimes the valve popped under its rated pressure. The pressure pulsation raised our doubt that subsequent popping of the outlet relief valve in the 4th stage might have resulted in the unit's vibration. That's why the relief valve was reset or blinded off for a period during overhauls. Unfortunately, replacement of the interstage cooler and reset of the relief valve or even blind-off didn't terminate or cure vibration of the HP cylinder.

3.5 Influence of Rotating Stall and Solution

We have explored some possible sources causing serious vibration through the practice of several years. Among them, rotating stall has been the doubtable factor since fluctuation and high vibration value with 3.8-5.6 mils changes existed. Flow pulsation was revealed by local indication of outlet flow. The pulsation was about 3000 NM_3/H and was accompanied by low and deep humming from the compressor like surging but not so strong. Indication of temperature and pressure changed little at that time. When fluctuation went out, vibration became stable; the strange sound disappeared, and the indication of the 4th outlet flow came to rest.

The sound we heard usually came from the unit or pipe vibration rather than from turbulent steam inside the unit or pipes. In the morning of September 19, 1980, gas inside rushed out from a suddenly-cracked thermowell on the 4th stage outlet and a strange sound was generated which was never heard before. Vibration values fluctuated during the sound; gas gushed from the cracked area in intermittent puffs strong or weak. When the vibration became stable without change, escaping gas snorted continuously. The sound of the gas stream strongly suggested rotating stall.

In order to find out the real source of vibration on the HP cylinder, we invited some research institutes and colleges, specialists in joint analysis. Table 3 is the record measured with frequency detector.

Rotative speed of the turbine and LP cylinder was 7200 rpm, whereas that of the HP cylinder was 13,900 rpm. During selection, 240 H_2 was taken as the same frequency of HP cylinder, so 120 Hz was the half frequency, 60 Hz the quarter, and 480 Hz the double.

It is evident from Table 3 that base frequency was always present, which suggested the HP cylinder rotor was probably not in balance. Since the rotor was still running then, it was difficult to trace the source causing unbalance. Fitting parts coming off, impurities blocked in the channel of impellers, scaling on rotors or unbalance as manufactured were possibly the causes.

Table 3. Records of Frequency Spectrum

No.	location	pass frequency	frequency selection component				
		mils	60Hz	120Hz	240Hz	360Hz	480Hz
1	turbine inlet	1.23	0.1	1.15	0.275	0.085	0.10
2	turbine outlet						
3	LP cylinder in	0.8	0.336	0.142	0.48	0.48	0.06
4	LP cylinder out	0.8	0.38	0.48	0.2	0.225	
5	gear box in	1.0	0.145	0.83	0.145	0.064	
6	gear box out	1.0	0.22	0.10	0.708		0.310
7	HP 4th stage in	3.4	2.0	1.1	2.8		0.11
8	HP 3rd in(south)	3.6	1.15	0.9	3.2		0.4
9	HP 3rd in(north)	3.4	0.6	0.8	3.0		0.4
10	LP displacement	16	3.2	10	2.6	3.7	
11	HP displacement	12	3.7		9		3.3

Besides the base frequency, 120 H_2 and 60 H_2 come next and the third, the double, and other multiple frequencies also stood in a certain proportion. The lower frequencies of 120 H_2 and 60 H_2 were considered caused by rotating stall since a strange sound, changed flow and pressure were observed as vibration fluctuated. Big clearances between shoes and shaft or outer ring would result in half-frequency vibration; however, not all vibration values changed after replacement of radial shoes. Eddy turbulence of bearings at half speed might bring lower frequencies, but after raising oil temperature, vibration did not change. Eddy turbulence was also discounted.

Double and multiple frequencies, in minor proportion as they were, shouldn't be neglected. Misalignment of rotors being the source of double frequency was not the main cause of the vibration of the HP cylinder. The decisive factor was the base frequency resulting in rotor's unbalance and half and quarter frequencies resulting in rotating stall. Of course, this should be proved through practice including dynamic balancing of the rotor and increased inlet flow.

As mentioned above, dynamic balance at high speed done during the overhaul in 1980 brought down vibration a little. In order to solve the problem completely, two measures were taken as follows.

A. Erection of a return pipe from and to the 4th stage. A length of pipe being 32.5*3.25 size, 1Cr18Ni9Ti material was routed from the outlet and to the inlet of the 4th stage.

B. Raising the inlet temperature of the 4th stage. It was increased from 55°C to 65°C to get more volumetric flow at the 4th stage inlet and thus cleared up stalling.

On September 19, 1980, two experiments were made as follows: Opened the block valve on the 4th to 4th return line and the vibration of the 7th point decreased gradually. It was performed in 3 steps. Ten minutes after two turns opening of the valve, the vibration of the 7th point went down from 2.5-2.6 mils to 2.2-2.3 mils but came back just two seconds later. Two more turns made the vibration down again 2.2-2.3 mils with obvious increased indication of flow percent on the control panel. At the same time the outlet pressure from the 3rd stage increased from 82 kg/cm^2 to 87 kg/cm^2. Full opening of the valve on the 4th to 4th return line brought vibration of the 7th point down from 2.2-2.3 mils to 1.7 mils; fluctuation and the strange noise disappeared; vibration of the 4th stage outlet pipe decreased. The 8th and 9th points had corresponding changes to the 7th point.

Raising the inlet temperature of the 4th stage had the purpose of increasing volumetric flow at the 4th stage inlet. Temperature increase from 50-55°C to 65°C brought the vibration of the HP cylinder down and produced stable running of the train. On the contrary, decreasing the 4th stage inlet temperature from 65° to 55°C made the vibration of the 7th point increase from 1.7 mils to about 2.5 mils; a strange noise appeared and vibration of pipes increased.

Two measures were taken for increasing volumetric flow at the 4th stage and curing the rotating stall of the stage. Our practice proved them effective but adverse to energy saving.

On April 14, 1982, for the purpose of lowering steam consumption, the valve on the 4th to 4th return line was closed gradually; vibration of the HP cylinder went up slowly. The vibration of the 9th point was up to 4.5 mils when the valve was fully closed and the strange noise was heard again. Finally, the valve on the 4th to 4th return line had to be opened to ease the train.

4. IMPROVEMENTS ON THE CO_2 COMPRESSOR TRAIN

4.1 Retrofit on Power Supply System

After put into operation, the motor-driven oil pump tripped several times due to instant fluctuation in the power supply system. If the spare oil pump couldn't be started promptly, the oil pressure became too low, the interlocking system would work immediately and finally the CO_2 compressor train got tripped. Therefore, the power supply of the interlocking system was rearranged from the ordinary civil power source to the uninterrupted power system from the ammonia plant. This cured the problem.

The interlocking solenoid valve worked normally energized. Fluctuated voltage would make the solenoid valve release its action due to lower tension and trip the train because of oil system cutoff. The interlocking solenoid valve after improvement worked normally; deenergized and got energized when abnormal. This cured trips from false action of solenoid valve due to fluctuation in the power system and eliminated shutdowns when quick on-off in power source occurred possibly in thunder and storm on a summer's day. Two years' practice had proved the measures effective.

4.2 Retrofit on Oil Supply System

There is a cubic "U-tube" oil heater in the lub oil sump to heat cool oil with LP steam after protracted shutdown of the train. Every year sludge layers about 50 mm thick accumulated on the inside surface of the oil sump and should be cleaned up during overhaul because of corrosion and scaling resulting from comparatively high temperature and small amount CO_2 and water in the oil. The cleaning was difficult and the oil might be damaged. A stainless steel partition plate was added in the lower portion of the oil sump. The oil heater was changed to a coil-type heater and put under the partition plate like a flat drawer which could be pulled out or pushed in as needed. Two-millimeter stainless steel lining was provided on the surface of the sump. No dirt or scalings from the heater and the inside surface have been found in the oil sump since then.

4.3 Backing Ring to the Rubber Seal in the HP Cylinder

The 3rd and 4th stage are in the HP cylinder which is of a barrel type. The outer casing and end covers are forgings, whereas the inner casing is a cast piece. The inner casing is half split horizontally. The two halves are bolted together and put into the outer casing. Rubber O-rings were provided to prevent leakage through and between inner and outer casings. During the trial running in 1976, the "O" rubber ring was blown off and displacement of the rotor in the HP cylinder occurred due to overload of axial force. This resulted in a serious incident in which thrust bearings, rotor, diaphragms and labyrinth at different stages were damaged. Another ring of PTFE was added to back up the rubber one which was already blown off. The backing ring was added as illustrated in Figure 2.

Blow-off of the rubber ring hans't happened again since the addition of PTFE backing ring, thus it gave trouble-free and safe production.

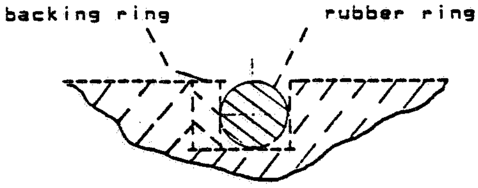

Figure 2

4.4 Rearrangement of Vent Pipes

Besides the vent pipe upstream from the stripper, at the side wall of compression building, there were other vent pipes with horseshoe-like nozzles toward manways between the process steel structure and control room and compression building. During starting, the HP cylinder was operated at low pressure and flow which was not in balance with the inlet flow of the LP cylinder. The vent pipe of the second stage in particular had to be opened in time to avoid over-pressure in the stage. Therefore, a large amount of dense CO_2 gas vented moved down to the ground and diffused around. At this point, operators in the urea plant were busy up and down the steel structure. Suffocating made operators uncomfortable. During the overhaul in 1980, the original vent pipes were extended by 1.5 m which was long enough to make them higher than the building. The horseshoe-like nozzles were turned to the opposite direction. This eliminated pollution and harm to operators when starting.

4.5 Shortening the Drains of HP Cylinder

The CO_2 compressor rested on the second floor with drain tubing from the compressor's bottom to the first ground floor, which was designed rationally and serviced. LOng drains and instrument tubing always vibrated when the HP cylinder vibrated seriously. The over-stress on the drains and instrument copper tubing made them break several times, hence forced shutdowns, e.g., in 1979. Changing the instrument tube material to plastics, shortening the drain tube from 6 m to 0.5 m, and moving the drain valve from downstairs to upstairs cured the trouble because the shortened drain tube would vibrate together with the HP cylinder; no shutdown and overstress occurred after that.

4.6 Improvements of the 3rd-Stage Cooler

The temperature of the CO_2 incoming stream from the ammonia plant increased after each compression stage; therefore, interstage coolers were provided. After several years' usage, the 3rd-stage cooler was found badly corroded due to two reasons. A direct cooler for incoming CO_2 was designed and put into operation; therefore, the radical C_1- in circulating water was introduced into the CO_2 system causing stress corrosion in the "U-type" cooling tubes. The inlet temperature of the 4th stage had been increased in order to cure the rotating stall. Under the same cooling area, reducing the quantity of cooling water would increase its outlet temperature. The outlet temperature of cooling water changed from 42-47°C to 70-80°C, which was beyond the limit 45°C specified to prevent stainless steel of TP304 and TP316 from stress corrosion. The 3rd-stage cooler was improved in two ways. The average length of welded "U" tubes was reduced from 13.3 m to 7.55 m and heat transfer area from about 100 M2 to 51.6 M2, whereas the quantity of cooling water increased from 30 T/H to 150 T/H. After that, the outlet temperature was down to about 45°C. Besides this, a bypass of Dg200 was provided on the inlet pipe of cooling water for the purpose of outlet temperature control. The 3rd-stage cooler was found to have almost no damage or trouble after the improvements in 1980.

REFERENCES

1. Vibration Analysis on the CO_2 Compressor Train in Liaohe Chemical Fertilizer Plant. Vibration Research Group, Mechanical Dept., Zheng Chou Engineering College, China, October 1980.

2. Vibration Problems of Rotors in Turbine Machines. Turbine & Compressor Teaching & Research Section, Sian Communication University, China, March 1980.

3. Tang Guo Fu: Analysis and Solution for Big Fluctuation of Vibrations on CO_2 Compressor HP Cylinder. Liaohe Chemical Fertilizer Plant, Liaoning Province, China, July 1980.

4. Procedure for Maintenance and Overhaul of CO_2 Compressor Train. Chemical Fertilizer Dept., The Ministry of Chemical Industry, China, 1983.