

1986

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EXPERIMENTAL ANALYSIS OF SCREW COMPRESSOR  
NOISE AND VIBRATION

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ABSTRACT

Very few reports have been presented to date on the noise and vibrational characteristics of oil injected screw compressors.

This time, a series of extensive experimental analysis is performed. It includes the measurements and analysis of compressor casing vibrations, rotor shafts vibrations, torsional (rotational) vibrations, suction and discharge gas pulsations, pressure transient in one thread of rotors and compressor noise.

These experiments are done mainly with R-22 gas, and the operating speed of compressor continuously varied up to 4400 RPM in some test conditions. The tested compressors are mainly 160L(163mm ,long rotor).

Analyzed results of these experiments help to explain the major noise and vibrational characteristics of standard oil injected screw compressors.

Nomenclature

Fo = fundamental screw frequency, Hz  
Fm = male rotor operating frequency, Hz  
Ff = female rotor operating frequency, Hz  
k = specific heat ratio  
n = operating speed, RPM  
Pd = discharge pressure, Pa  
Ps = suction pressure, Pa  
Vi = built-in volume ratio  
Pi = built-in pressure ratio =  $V_i^k$  (adiabatic change)  
Zm = male tooth number  
Zf = female tooth number  
D = rotor diameter, mm  
L = rotor length, mm

## INTRODUCTION

Today, the oil injected screw compressors are known as high-performanced and highly durable compressor and their application range has become very wide. In spite of the amount of information concerning its applications and performances, technical papers dealing with the fundamental characteristics of noise and vibration of the compressors seem to be very few. In order to evaluate and reduce the noise and vibration of the compressors, their fundamental characteristics must be well understood. So, to aid the better understanding of these, we will present this summerized report of our series of experiments concerning noise and vibration of screw compressors.

### Compression Mechanism

The screw compressor is classified as a positive displacement rotary compressor. The noise and vibrations generated by a screw compressor have a distinct relationship to its gas compression mechanism. So it is important to understand this mechanism. A general arrangement of an oil injected screw compressor is shown in Fig. 1.

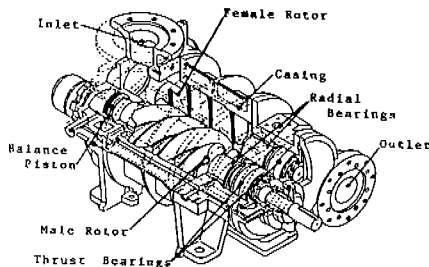


Fig. 1 Oil Injected  
Screw Compressor

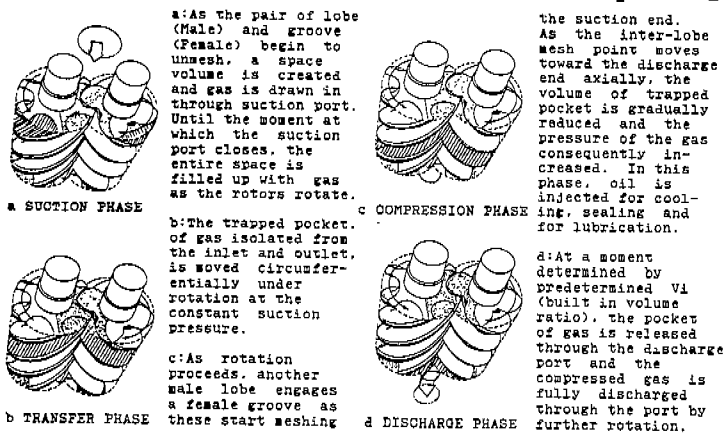


Fig. 2 Compression Mechanism (Ref. [1])

The compression of a gas is attained by the direct volume reduction of the inter-lobe space as the rotors rotate. This is illustrated in Fig. 2.

## EXPERIMENTAL TECHNIQUE

### Instruments (Ref. [2])

A number of analytical tools and procedures are available to analyze noise and vibration. The FFT (Fast Fourier Transform) analysis is considered as a versatile technique within these methods. For a practical point of view, stand-alone FFT analyzers are used as the main instrument to analyze signals from compressors. The followings are the signals studied in detailed experiments.

- pressure transient in one thread of female rotor
- discharge and suction gas pulsations
- rotor shaft radial vibrations (relative to casing)
- rotor shaft axial vibrations (relative to casing)
- torsional vibration
- relative rotational vibration (male and female)
- casing vibration acceleration (in three directions)
- compressor noise (normally at 1 meter)

Fig. 3 shows general set-up of instruments and transducers.

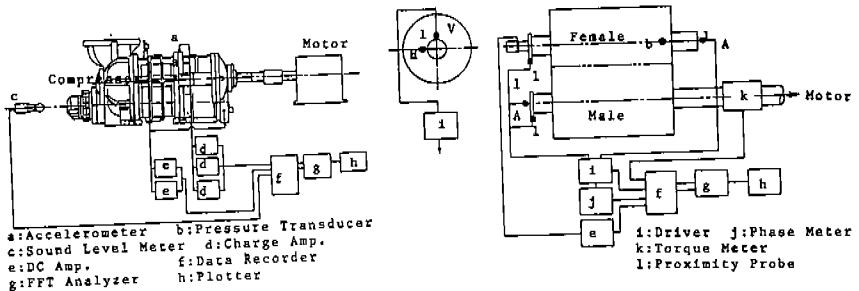


Fig. 3 Instrument General Set-Up

### Test Facilities

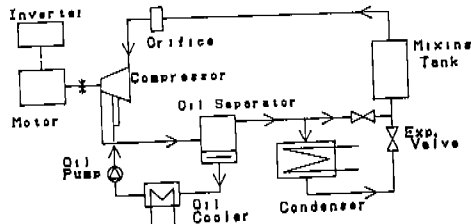


Fig. 4 Test Facility Arrangement

Detailed experiments including variable-speed operation are performed in laboratory test facilities mainly on 160L. Also some measurements are taken on operating compressor in actual plant. Shown in Fig. 4 is the general arrangement of the test facility.

## FUNDAMENTALS

This section will present materials found by experiments on the fundamental characteristics of screw compressor noise and vibration.

### Fundamental Screw Frequency

The fundamental frequency of noise and vibration can be determined by the following equations.

For male drive  $F_o = n/60 \times Z_m$

For female drive  $F_o = n/60 \times Z_f$

These equations indicate that fundamental frequency is simply determined by the drive rotor operating speed and its number of lobes. In this report we call this frequency  $[F_o]$  as "fundamental screw frequency".

Another fundamental frequencies are operating speed of male rotor and that of female rotor. They are calculated by next equations.

For male drive  $F_m = n/60$

$F_f = n/60 \times Z_m/Z_f = F_m \times Z_m/Z_f$

For female drive  $F_f = n/60$

$F_m = n/60 \times Z_f/Z_m = F_f \times Z_f/Z_m$

### Pressure Transient in Rotor Thread

Fig. 5 illustrates an example of in operational period relationship between male rotor turning angle and pressure transient in one rotor thread, pulsation of discharge and that of suction. All these are pro-

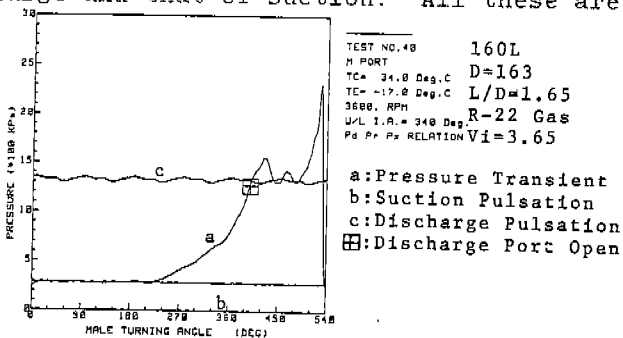


Fig. 5 Operation-Period of Pressure & Pulsation

bable causes of noise. The pressure transient dominates gas forces induced on rotors and thereby controls the fundamental noise and vibration characteristics.

### Pressure Pulsations

Pulsations are caused by intermittent gas discharge and suction according to the compression mechanism. The discharged gas pulsation is influenced by the pressure difference between operating discharge pressure and thread pressure at the moment of the beginning of the discharge phase. Example of operating pulsations from 160L is shown in Fig. 6. According to the Figure, their waveforms are similar to sawtooth / triangular wave and these periods indicate that their frequencies are at the fundamental screw frequency.

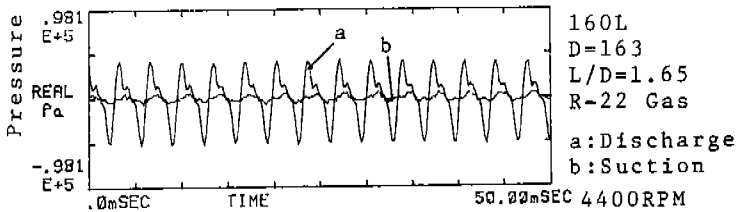


Fig. 6 Operating Pulsation Waveform

### Torsional Vibration (Torque Fluctuation)

Fig. 7 shows a typical example of operating torsional vibration signal from the output of the torque meter. The vibration is caused by dynamic gas torque and its frequency is also at the fundamental screw frequency. In general, the measured torsional amplitudes from torque meter would be well below 5% of the static torque when operated under normal conditions with R-22 gas.

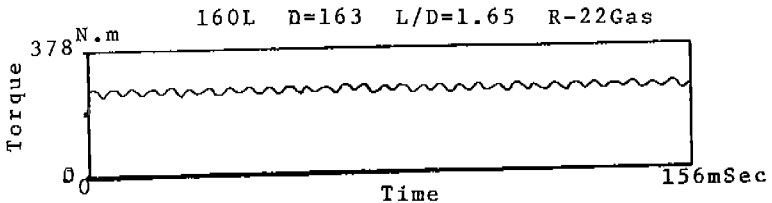


Fig. 7 Operating Torsional Vibration

### Rotor Shaft Vibrations

Resulting from the gas forces and rotor contact

forces, dynamic forces are induced on the rotors and thereby radial and axial shaft vibrations of the rotors occur. These vibrations are transmitted through the bearings to the casing of compressor. The fundamental frequency component of rotor shaft vibrations both in radial and axial directions are identical to the fundamental screw frequency. A typical example of operating axial shaft vibration is shown in Fig. 8.

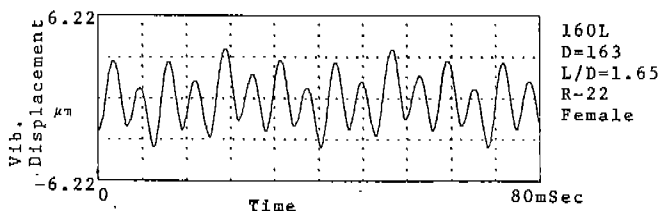


Fig. 8 Operating Axial Shaft Vibration

General Noise Characteristics

Fig. 9 shows the noise spectrum from 250L operating at 3000 RPM. Several discrete peaks, well above 1kHz, can be noted, and other peaks are broad and not well defined. The dotted line in the Fig. 9 indicates a broad-band random noise radiated by gas flow.

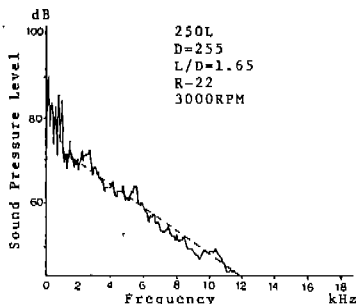


Fig. 9 Overall Operating Noise Spectrum

In general, the broad-band random noise of a frequency spectrum is heard as a rushing sound.

Fig. 10 shows the spectrum of another compressor, 200L, operating at 3000 RPM. This compressor again generates noise at discrete frequencies and are all harmonically related. They are higher harmonics of the fundamental screw frequency (200 Hz) and the broad-band random noise

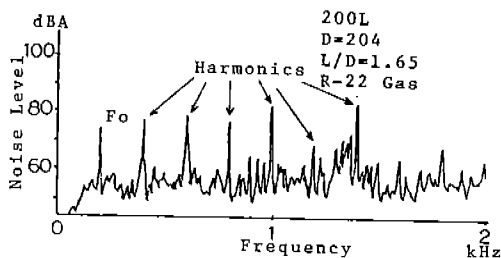


Fig.10 2 kHz Range Operating Noise Frequency Spectrum

indicates gas flow noise. It should be noted that the magnitudes of harmonics are approximately at a constant level up to several order harmonics.

General Vibration Characteristics

Fig.11 shows examples of vibration from 320L operating at 3600 RPM employing a multiplying gear box.

The displacement spectrum, Fig.11-a, contains 3 dominant discrete spectrum and the peak occurring at 240 Hz is the fundamental screw frequency. The spectrum at 50Hz shows operating speed of input shaft and its unbalance(including motor,coupling and drive gear ). Equally, the peak at 60 Hz indicates output shaft operating speed and its unbalance. And the waveform of displacement indicates the "beat" between 50Hz and 60 Hz operating frequency signals.

The velocity spectrum, shown in Fig.11-b, indicate the fundamental screw frequency and harmonics decreasing their amplitude with frequency. The waveform of velocity shows appearances of sawtooth wave hence its harmonics may decrease approximately -6dB/octave in amplitude.

In contrast with the velocity signal, the acceleration signal shown in Fig.11-c indicates more definite tendency of series of pulses. And according to its waveform, the acceleration spectrum contains more harmonics. The higher harmonics should come

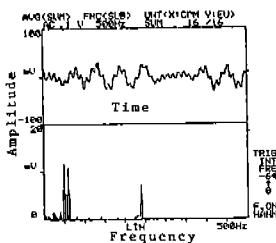


Fig.11-a Displacement

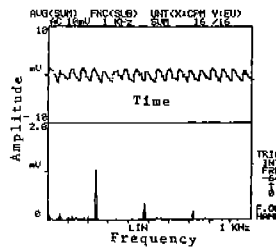


Fig.11-b Velocity

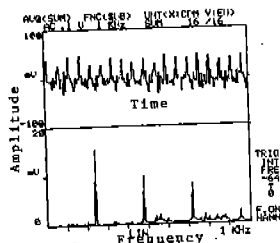


Fig.11-c Acceleration

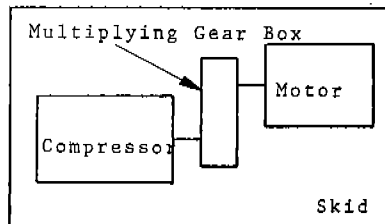


Fig.11-d Arrangement

Fig. 11 Operating Vibration Waveform and Spectrum



mainly from rotor meshing. Generally, measured vibration acceleration values may not exceed  $9.8 \text{ m/sec}^2$  ( $= 1 \text{ G}$ ). The "Axial" direction shows the maximum vibration acceleration level from normally operating compressors. It should be noted that, in general, larger vibration level result in higher sound level.

### Vibration Parameter Selection

According to extensive vibration measurements, data indicate that the vibration of screw compressor has several harmonics of the fundamental screw frequency at a constant level in acceleration in certain range of frequencies. An example is shown in Fig.12.

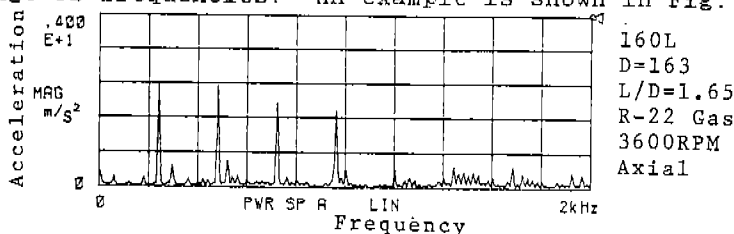


Fig. 12 Operating Acceleration Spectrum

So, regarding the selection of vibration measurement parameter, displacement is not the preferred parameter except in case of unbalance problem because of its low sensitivity at high frequency, as shown in Fig.13.

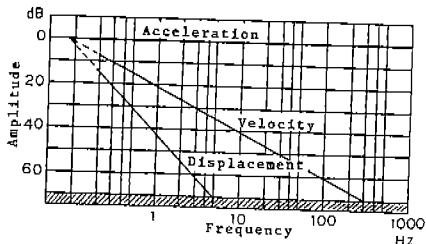


Fig. 13 Displ., Vel. & Acc. Relation

It is recommended that one should choose acceleration as a parameter when measuring screw compressor vibration in detail. Care must be taken to avoid mounted resonance of accelerometers within the frequency range of interest.

As a separate consideration, the displacement probe (proximity probe) is the only transducer when measuring shaft vibration relative to the casing.

## EFFECT OF OPERATING CONDITIONS

### Operating Pressure Condition

The operating pressure condition has a distinct

influence upon compressor noise, casing vibrations and performance. Operating pressures also affect torsional vibration, shaft vibration and pressure pulsations. The use of economizer also has some effect on above mentioned vibrations.

As the pressure in a rotor thread, just prior to the start of discharge phase, is only determined theoretically by the multiplication between suction pressure and built-in pressure ratio ( i.e.  $P_s \times P_i$  ) without economizer, so there exists only one actual operating discharge pressure which will exactly agree with the theoretically determined discharge pressure. Regardless of the magnitude, there exists a certain pressure difference between these two discharge pressures. In general, the operating compressor noise and casing vibration level will increase with the increase in the pressure difference.

A measured example of compressor noise and casing vibration shown in Table 1 indicates above mentioned tendency under an operating discharge pressure condition. Another example under a constant operating suction pressure condition is shown in Table 2.

Table 1 (dB)

Noise & Vibration at Varying  $P_s$

No.	$P_s$ MPa	V.Acc.	H.Acc.	A.Acc.	SPL
1	0.50	+4.1	+2.2	+2.9	+3.1
2	0.30	0	0	0	0
3	0.16	+0.1	+1.1	+3.1	+3.4

\* $P_d=1.37\text{MPa}$ , No.2 as a reference value

Table 2 (dB)

Noise at Varying  $P_d$

No.	$P_d$ MPa	SPL
1	0.91	+0.2
2	1.07	0
3	1.37	+2.4

\* $P_s=0.24\text{MPa}$ , No.2 as a ref. value

In addition, the discharge port size (H, M and L), H-port generally shows the best effect on measured noise and vibration from refrigeration compressors because of its least gas mass flow rate within three ports under normal operating condition.

### Operating Speed

Operating data is used to determine not only the level of noise and vibration but also, more importantly, the frequency components of these signals. By varying the operating speed of the compressor, the effect on frequency spectrum can be seen and the various resonances, inherent in the compressor system, can be determined.

Fig. 14 shows an example of a three-dimensional RPM spectrum map of generated sound pressure levels from 160L. The peaks ( at 100 Hz, 220 Hz, 275 Hz and 545 Hz ) are indications of various resonances.

In this case, the maximum amplitude peak at 545 Hz comes from a relative rotational resonance between

the male and female rotors, and can be determined by the method of detecting the phase difference between two sinusoidal tooth passing signals from involute gears attached to each rotors.

The peak at 100Hz in Fig.14, finally determined at 87.5 Hz in higher frequency resolution analysis with more fine pitch change in the operating speed, is coming from a torsional resonance in the system.

And the peaks at 220 Hz and 275 Hz are confirmed to be resonances of skid members by hammering method.

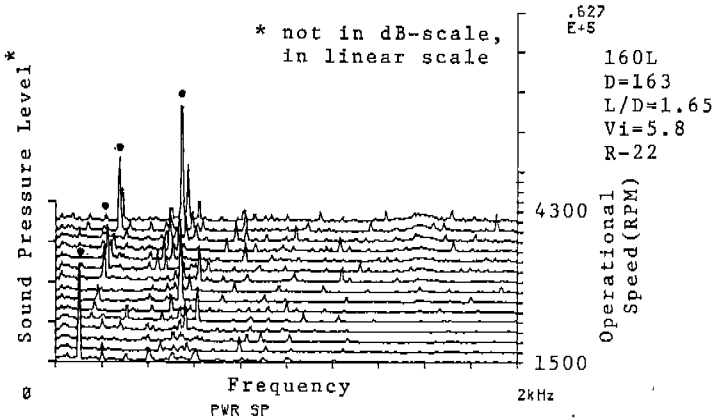


Fig. 14 3-D RPM Spectrum Map of Generated Noise

For the discharge gas pressure pulsation, a typical example is shown in Fig. 15. The magnitude increases with operating speed, but on the basis of our data, it cannot be accounted for its effect on compressor noise.

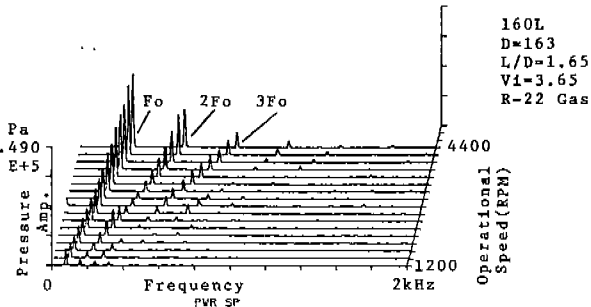


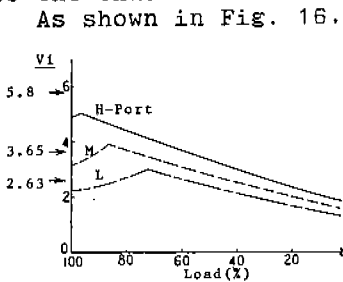
Fig. 15 3-D RPM Pulsation Spectrum Map

Relative to the casing vibration acceleration, as a result of our experiment, it can be assured that the acceleration level will increase proportionally with the increase in operating speed when no resonance

has occurred.

Part Load Condition

An outstanding feature of the standard screw compressor is the ability of stepless capacity control with the slide valve system. The change in  $V_i$  resulting from the axial movement of the slide valve affects not only the performance of compressor but also the characteristics of noise and vibration.



As shown in Fig. 16, under part load operations the change in  $V_i$  ranges from 3.0 to 1.25 even in the L-port, so the difference between theoretically determined discharge pressure ( $P_s \times P_i$ ) and the actual operating discharge pressure also would be varied with unloading.

Fig.16 Vi Change in Unloading

As the result of above, it appears that a complex change occurs in the characteristics of noise and vibration under part load operation.

Fig. 17 shows a 3-D vibration spectrum plot from 160L operating at 3600RPM under part load.

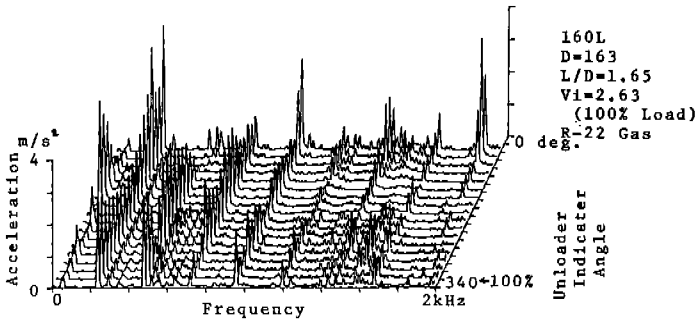


Fig. 17 3-D Axial Vibration Spectrum Map Under Part Load Operation

From full unloaded operation data, it can be assumed that the returning oil from the bearings and the mechanical seal will sometimes obstruct a stable operation of compressor under reduced gas flow. Moreover the slide valve sometimes vibrates at the fundamental screw frequency and, by impacting on the casing, it would cause an excessive noise and / or vibration problem in extreme case.

## Type of Gas

Change in type of gas to be compressed sometimes indicates distinct effect on the compressor noise and vibration characteristics. Here, as a typical example, Table 3 is shown as reference. Other than this Table, data indicates an existence of large difference in axial shaft vibrations and the amplitude under operation in some test conditions during which the amplitude with helium gas is four times greater than the amplitude with air.

The accurate cause of the difference in noise and vibration cannot be confirmed now, but it can be assumed that it is resulting from a change in the molecular weight of the gas to be compressed which may affect the leakage inside the compressors as well as the effect of oil.

Table 3 Noise and Vibration Difference

Gas	Vert. Acc.	Hori. Acc.	Axial Acc.	SPL
Air	1.46m/s <sup>2</sup>	1.53m/s <sup>2</sup>	1.86m/s <sup>2</sup>	85.5dB
Helium	4.80	3.50	7.34	93.5

Operating Condition:  $P_s=0.049\text{MPa}$ ,  $(P_d/P_s)=P_i$ , 3000RPM  
320L,  $D=321$ ,  $L/D=1.65$ ,  $V_i=5.8$

## Injection Oil Quantity

Injection oil flow rate sometimes affects the noise level experienced in large refrigeration compressors. It is said that under a reduced injection operation overall noise level sometimes can be reduced by several dBs. Other than R-22, and some light molecular weight gas, this phenomena is clearly confirmed by some experiments.

An example shown in Fig.18 is an operating noise from 320L operating at 3000RPM with helium gas. Although in this spectrum plot, reduced injection oil operation is quieter by 4 dB, in some tests, the attained improvement in sound pressure level is well above 8 dB.

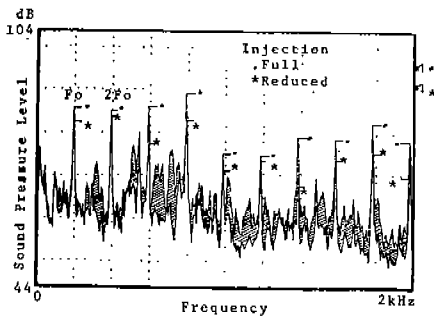


Fig. 18 Injection Oil Effect

## AFFECT OF SURROUNDING EQUIPMENTS

In most cases, the measured operating noise from

compressors contains many components from various individual sources. The motor, gear box, coupling, piping, oil separator, skid and so on. They all may combine to yield a complex frequency spectra plot.

Here, some simplified examples are discussed.

### Oil Separator

As already shown in Fig. 9, the accompanying broad-band random noise component decreases with frequency in frequency spectrum plot. In contrast with the Figure, in some cases as the typical example shown in Fig. 19, a broad peak appears in operating noise spectrum.

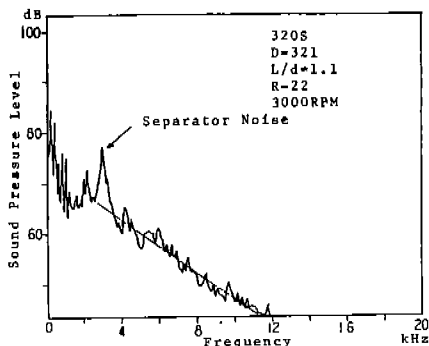


Fig. 19 Separator Noise

The broad peak present is seen to emerge from general background. This may be due to resonance noise radiated from a source other than the compressor. In this operating noise spectrum, that peak has come from separator outlet resonance.

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### Piping

Fig. 20 shows an example of operating noise from 320L test stand. The contained broad-band noise spectrum resulting from gas flow noise radiated from suction piping, can be reduced by changing arrangement of the piping.

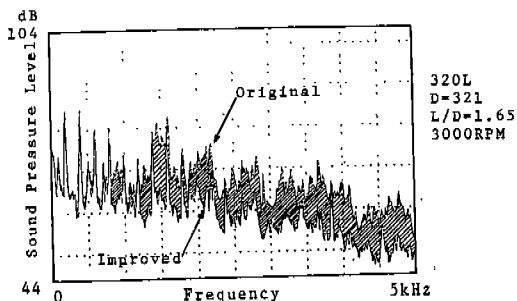


Fig. 20 Suction Piping Gas Flow Noise

In some applications, a multiplying gear box is employed to attain larger capacity than that of direct drive operation. Gear set within gear box sometimes produces noise problem because of its improper mesh, as shown in Fig. 21. In the Figure, the peak at about 3300 Hz is corresponding to the mesh frequency of this gear set. And around the primary mesh frequency, there is a series of equally spaced components of sidebands. These sidebands indicate existing gear pitch error.

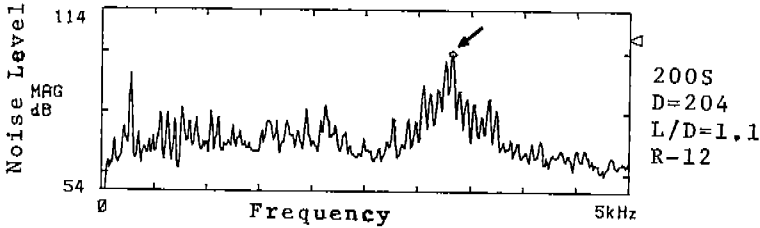


Fig. 21 Gear Noise Spectrum

DETECTION OF FAILURE

Thrust Bearing Flaking

In screw compressors, rolling-element bearings are usually employed as rotor thrust bearings to ensure precise axial positioning. These bearings have finite fatigue life and sometimes fail unexpectedly by flaking of raceway as a result of an abnormality.

The example shown in Fig. 22 is a typical operating vibration from 160L at 3600 RPM. In the

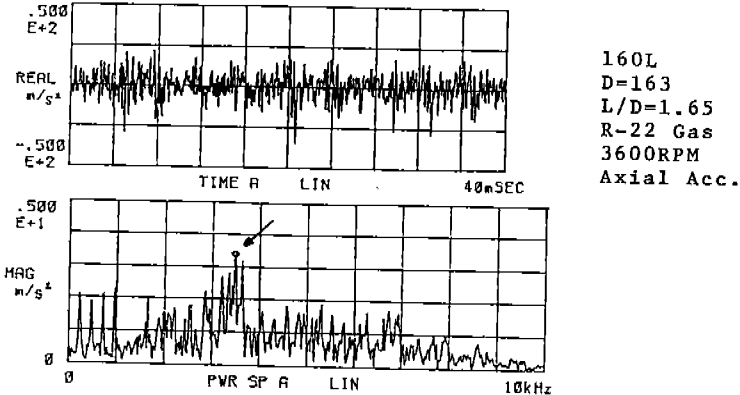


Fig. 22 Operating Abnormal Vibration

vibration acceleration spectrum, a cluster of peaks at approximately 3.5 kHz indicates an abnormality in this compressor. And the waveform indicates that the unusual signal is generated in a series of impacts repeating at relatively low frequency.

In this case, the technique of the absolute averaging in the time domain signals is employed, and by its spectrum, shown in Fig. 23, the repeating frequency is confirmed at 322.5 Hz. According to this frequency, the damage in a bearing inner raceway, shown in Fig. 24, can be detected. Concerning the frequencies derived by damage of rolling-element bearing, refer to the Table 4.

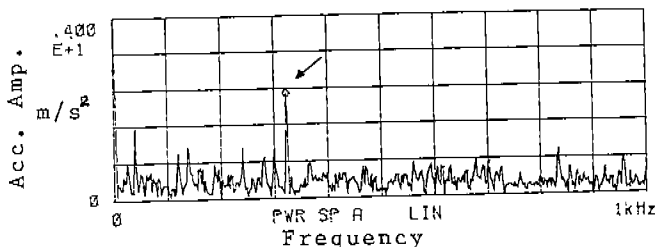


Fig. 23 Absolute Averaged Spectrum

Table 4 Bearing Frequencies

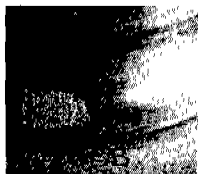


Fig. 24 Inner Raceway Flaking

Types of Damage	Damaged Part	Frequency
Eccentricity*1	Inner Raceway	$n \cdot Fr$
Rough Spot	Inner Raceway	$n \cdot Z \cdot F_1$ **
	Outer Raceway	$n \cdot Z \cdot F_5$
	Rolling Element	$2 \cdot n \cdot F_b$ **

\*1:and / or Wear,

\*\* :Amplitude Modulation may occur,

Where  $F_1 = 1/2 \cdot Fr \cdot (1 + d/D \cdot \cos \beta)$

$F_5 = 1/2 \cdot Fr \cdot (1 - d/D \cdot \cos \beta)$

$F_b = 1/2 \cdot Fr \cdot D/d \cdot [1 - (d/D)^2 \cdot \cos^2 \beta]$

$Fr = F_m$  or  $F_f$

$d$  = Diameter of Rolling Element

$D$  = Pitch Diameter

$\beta$  = Contact Angle

$Z$  = Number of Rolling Elements

$n = 1, 2, 3, \dots$

[Inner Raceway Rotates and Outer is Stationary]

### Scuffed Rotor Tooth

In a certain operating conditions, scuffing of rotors sometimes take place.

One typical example of abnormal noise from 200L operating at 3600 RPM is shown in Fig. 25. For the noise level of this compressor, 85 dBA was recorded at the beginnings of the plant operation and it has increased well above 100 dBA at the time of the analysis. The overall noise level indicates only how serious the problem is. To detect the origin of noise a frequency analysis is necessary. The analyzed noise spectrum contains many sub-harmonics of fundamental screw frequency, indicating tooth separation and



collision at the point of rotor mesh contact. The waveform also shows one collision takes place per every two mesh. After inspection, the cause of this abnormality is confirmed resulting from scuffed and worn rotor tooth flanks. (The scuffed pattern on the rotors can be reproduced by computer simulation, an example is shown in Fig. 26.)

In this case, scuffing of rotors is caused by oil compression at frequent start-ups. Operating sequence is revised to avoid unnecessarily long oil pump operation before compressor starts, and the problem was resolved.

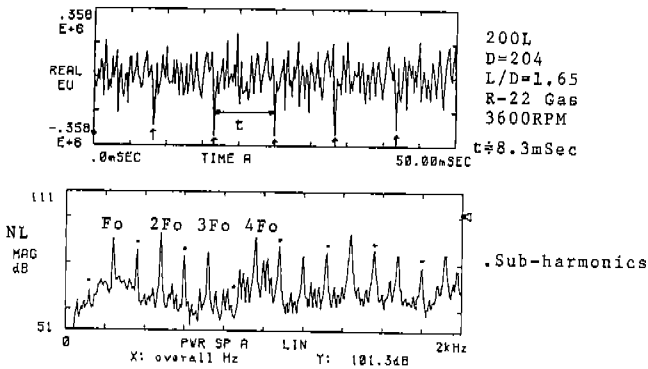


Fig. 25 Abnormal Noise Signal

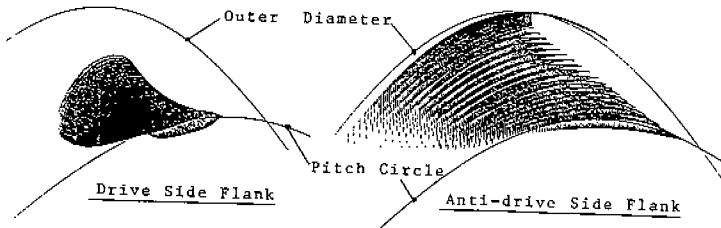


Fig. 26 Simulated Scuffed Pattern (Male Rotor)

### Unprecise Rotor

The screw rotors are so strictly controlled and inspected in manufacturing that the accuracy has been maintained over a certain level.

As a seldom example, if ever occur, Fig.27 shows an operating noise on 250S at 3000 RPM with unprecise rotor set. The sound is heard as a "rattling" noise.

The waveform indicates that three meshing pulses occur repeatedly in abnormal magnitude at every revolution of the female rotor. And the shown spectrum contains a series of discrete frequency components

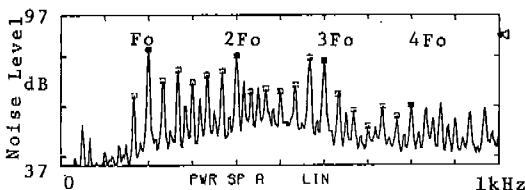
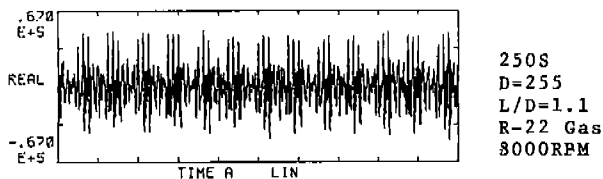


Fig. 27 Operating Noise with Unprecise Rotor

spacing approximately 33 Hz (=  $F_f$ ) between harmonics of the fundamental screw frequency. After inspecting the rotor set, it is confirmed that abnormal noise is caused by uneven tooth thickness of three of the six female rotor.

#### SUMMARY

The material in this report is directed toward providing a basic understanding of the fundamental characteristics of noise and vibration in the oil injected screw compressor.

With respect to frequency, it can be assumed that the fundamental screw frequency ( $F_o = F_m \times Z_m$ ; for male drive) dominates all aspects in compressor noise and vibration. From the results of these experiments it is believed that compression mechanism, design and manufacturing parameters, and operating conditions all influence on pressure pulsations of discharge and suction gas. More importantly, they influence on dynamic gas forces induced on rotors and the mechanical contact forces between rotors, thereby control all the characteristics of noise and vibration.

In our next report, we will present a theoretical analysis of screw compressor vibration.

#### REFERENCES

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