Vibration	Higher Harmonic Vibration of Reciprocating Compressor	Reciprocating Machinery
Resonance		

Object Machine

Reciprocating air compressor (about 100kW, 555rpm, belt-driven, refer to Fig.1), Unit A and B, two machines.

Observed Phenomena

Vibrations with a frequency five times the rotational speed (5N = 46.25Hz) occurred. Although Unit A and B are of the identical specification, the situation was different, that is, Unit A had large pulley vibrations, while Unit B had large cylinder vibrations. After measuring the natural frequencies, it was found that a vibration mode of the pulley in vertical direction (pulley mode) was Unit A: 46.75Hz, and Unit B: 41.25Hz, while a vibration mode of the cylinder in axial direction (cylinder mode) was Unit A: 42.5Hz, and Unit B: 47.75Hz, that is, they were inverted (Fig.2).

Cause Estimation

As these are displacement type compressors, the vibrations in question were assumed to be higher harmonic vibrations associated with pressure fluctuation, but the reason why their characteristics were different was not clear.

Analysis and Data Processing

From the compressor indicator diagram, a piston reaction force was obtained with crank angles on the horizontal axis, and was developed into a Fourier series expansion. As a result, it was found that the five times component of the rotational speed was larger than other higher harmonic components (Fig.3). These are close to the natural frequency of the pulley mode for Unit A and of the cylinder mode for Unit B, and were almost in the resonance conditions. Thus, in order to avoid resonance with 5N, a mass was added for detuning of the natural frequencies. However, individual addition of mass to the pulley or cylinder was not effective at all (Table 1), which is also similar in Unit B. To investigate the above reason, a two degree of freedom coupled model as shown in Fig.4 and equations of motion (1) were examined.

$$m_1\ddot{x}_1 + k_1(x_1 - x_0) = F_1(t), \quad m_2\ddot{x}_2 + k_2(x_2 - x_0) = F_2(t)$$
$$k_0x_0 + k_1(x_0 - x_1) + k_2(x_0 - x_2) = 0 \qquad (1)$$

Natural frequencies of a coupled system may be organized by the following four parameters: natural frequency of a pulley isolated system f_1, natural frequency of a cylinder isolated system f_2, pulley spring ratio $k_1/\Sigma k_i$, and cylinder spring ratio $k_2/\Sigma k_i$, (i = 0 ~ 2). On an assumption that each spring ratio of pulley & cylinder is 0.1 (that is, a base stiffness k_0 is relatively high, and 8 times of k_1 and k_2), the natural frequencies of a coupled system were organized with f_1 and f_2 as parameters (refer to Fig.5), with the following results identified.

(1) A slight reversal of f_1 and f_2 causes a mode changeover, which gives an explanation as to the difference between Unit A and B.

(2) By looking 3-dimensinal display of f_1 and f_2 of the natural frequencies of the coupled system, in case of a single weight added to reduce the natural frequency of a 2nd mode having a higher natural frequency, a significant effect cannot be expected if not simultaneously added on both sides.

Countermeasures and Results

For both Unit A and B, a mass 75kg was added to the pulley (changed to a thick type), and a mass 90kg to the cylinder as countermeasures (Table 1). Consequently, the vibration amplitude was greatly reduced, and the resonance problem was solved.

Lesson

As for resonance, many countermeasures involve additional masses or stiffening for detuning of natural frequencies. However, for a coupled system as in this case, only a single countermeasure may not be so effective in many instances.

References

Nothing in particular.

Keywords

Reciprocating compressor, two-degree of freedom coupled system, simulation, Fourier component, additional weight, higher harmonic resonance, detuning, multiple countermeasures

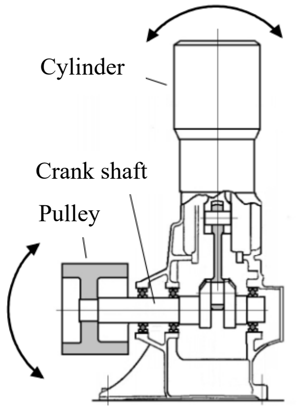


Fig.1 Schematic diagram of reciprocating air compressor and two vibration modes

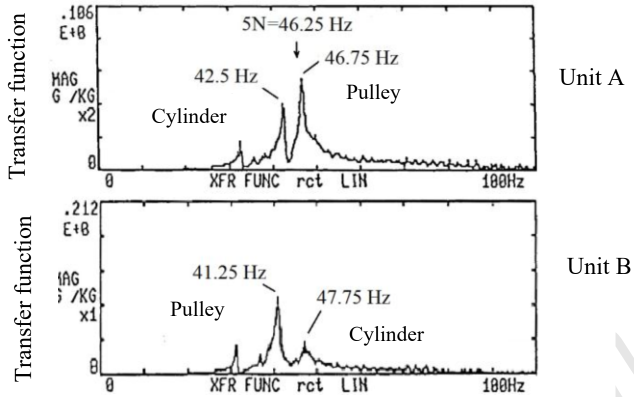


Fig.2 Transfer functions of two units (pulley response/pulley hitting)

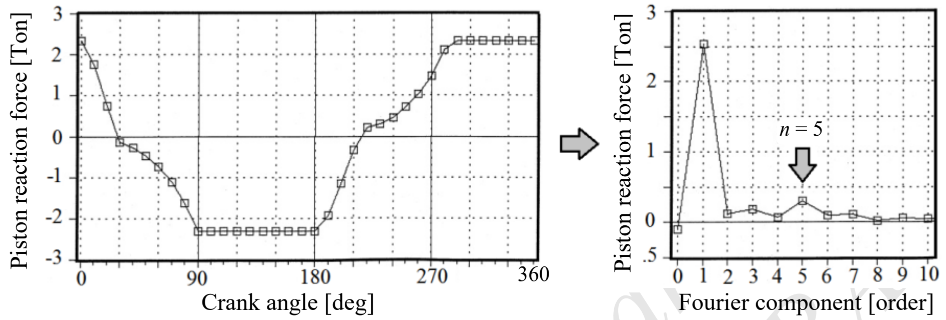
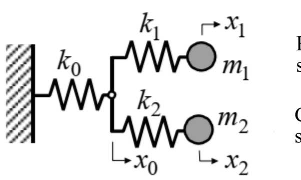


Fig.3 Piston reaction force and Fourier component obtained by indicator diagram (calculation)

Table 1 Mass (kg) added to pulley and cylinder, and changes in natural frequencies (Hz) of two modes (in case of Unit A)

Pulley \ Cylinder	0	40	90
0	46.75 (0) 42.5 (0)	46.5 (−0.25) 42.0 (−0.5)	46.5 (−0.25) 41.25 (−1.25)
21.5	–	–	45.0 (−1.75) 41.0 (−1.5)
44.5	46.25 (−0.5) 41.0 (−1.5)	–	44.25 (−2.5) 39.5 (−3.0)
75	–	–	44.0 (−2.75) 39.5 (−3.0)



$$f_1 = \frac{1}{2\pi}\sqrt{\frac{k_1}{m_1}}$$

$$f_2 = \frac{1}{2\pi}\sqrt{\frac{k_2}{m_2}}$$

Fig.4 Simulation model of pulley & cylinder two-degree of freedom system

a) In case natural frequency f_1 of single pulley system is fixed at 48Hz

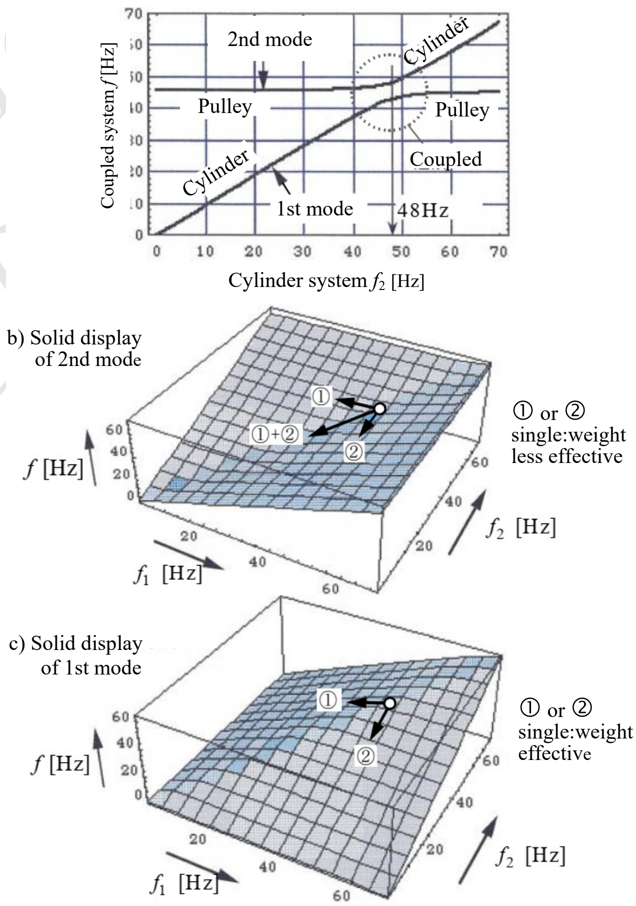


Fig.5 Description of mass addition effect for two-degree of freedom (pulley & cylinder) coupled system