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Reduction of Compressor Vibrations by Optimizing the Locations of the Counterweight and the Internal Springs

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ABSTRACT

Until now the rotating and oscillating imbalance in the hermetic compressor has been balanced to and greater or lesser extent, but in addition internal springs were added where this was expedient from a design viewpoint. It is clear, however, that the position of the springs is an important factor in the transmitted vibrations. Generally, it proves in this paper that for a single-cylinder compressor balanced by a counterweight there is one and only one plane at right angles to the axis of rotation, where the vibrations in the plane are at a minimum. If the points of application are placed in the plane it is clear that the forces transmitted through the springs will be minimal.

INTRODUCTION

The noise level of compressor driven refrigerators and freezers is determined by several factors. The compressor emits

Noise
Pulsations
Vibrations

furthermore, the noise level is effected by flow noise in the refrigeration system itself and by installation conditions.

In order to obtain a minimum noise level it is necessary to give equal consideration to all factors. It serves no purpose to reduce, for example, system noise and compressor pulsations, if it is vibrations or compressor noise which is the dominating influence in the total result. General competition factors and the use of effective insulation materials means, that refrigerators and freezers are to day considerably lighter than before. These facts coupled with an increasing tendency to build appliances into kitchen modules makes it especially important to focuse attention to the vibration level of the compressor (1).

A thorough analysis of the vibrations from compressor/refrigerator/kitchen module would be very complicated.

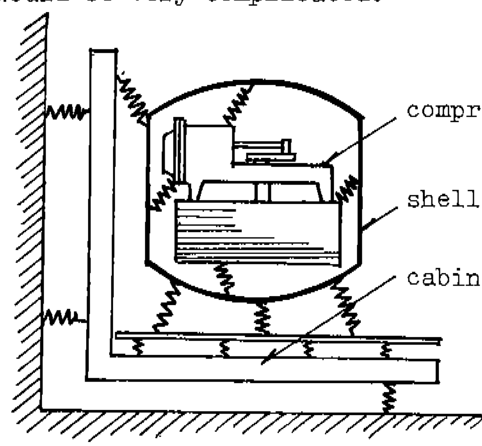

Fig. 1

As appears from fig. 1 the compressor and its shell alone constitute a system with 12 degrees of freedom.

To examine the possibilities for reducing compressor vibrations it is therefore wise to regard the problem in simplified form. With small approximations we find, that the forces and torques affecting the appliances are dependent on

The compressor
The tube connections
The compressor feets

Compressor vibrations are determined by

The mass inertia of the compressor shell
The forces acting on - - -

The forces with which the compressor unit acts on the shell are determined by

The stiffness of the internal springs
The translations and angular displacements of the springs.

If we look at the compressor as a whole the following is necessary, if we are to obtain minimum vibrations:

Large mass inertia of compressor shell.
Low stiffness of the internal springs.
Small translations and angular displacements of the internal springs.

A good result will require action on all three points. With regard to the two first points, the limitations are of a purely economic and practical nature. We shall not pursue these further, but instead elaborate on the last point.

THEORY

It is well known that a single-cylinder compressor cannot be balanced loo percent for both rotating and oscillating forces, without the use of expensive elements. However, it is possible, based on curtain mechanical constants of the compressor to calculate where the counterweights should be placed and where the springs should be fastened in order to transmit minimum first order horizontal forces through the springs.

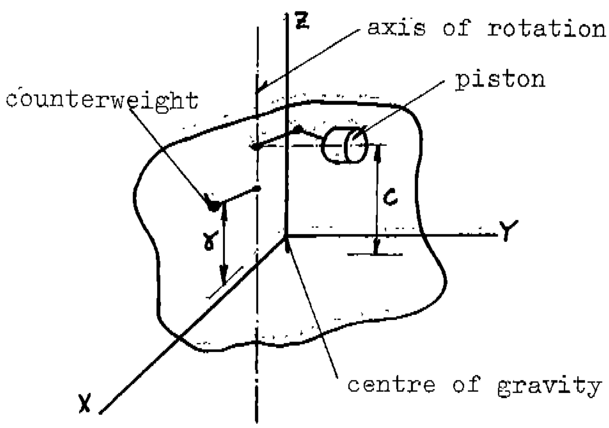

Fig. 2

Let us regard a stiff body equipped with a cylinder, a piston mechanism and a counterweight. With the piston in arbitrary position, a cartesian coordinate system is plotted, fixed in relation to the surroundings. The zero point of the system lies in the gravity point of the body, the Y-axis is parallel to the cylinder axis, and the Z-axis is parallel to the rotation axis of the piston mechanism. The cylinder is at a distance c from the X-Y plane and the corresponding distance for the rotation plane of the counterweight is γ. Finally, let us assume that the mass of the body is M and the moments of mass inertia corresponding to the X and Y axes are I_x and I_y.

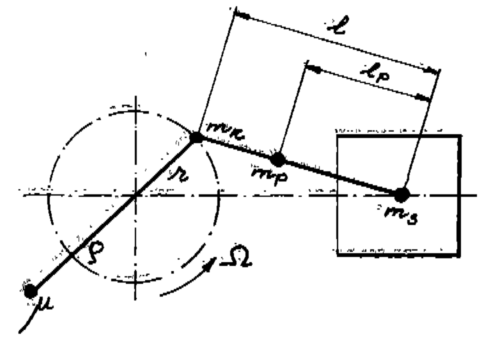

Fig. 3

In the above drawing of the piston mechanism the half stroke is r. The piston and the crank pin have the masses m_s and m_k. Furthermore, the piston rod has the mass m_p, and the point of gravity at distance l_p from the gudgeon of the piston. The total length of the piston rod is l. Finally, the counterweights has the mass μ, and ρ is the radius of the circle, which the point of gravity of the counter weight describes.

The following abbreviations are used

$$M_1 = \frac{l_p}{l} m_p + m_k$$

$$M_2 = m_s + m_p + m_k$$

We shall assume provisionally that the body is floating freely in a weightless room. We shall regard points lying in a plane at an arbitrary distance z from the X-Y plane. If the angular velocity of the crank is Ω, it can be demonstrated that the first order translations in the X and Y directions of the points in the above mentioned plane can be written

$$(1)\quad d_x = \left\{ -\frac{M_1 \cdot r - \mu \cdot \rho}{M} - z \frac{M_1 \cdot c \cdot r - \mu \cdot \gamma \cdot \rho}{I_y} \right\} \sin(\Omega t)$$

$$(2)\quad d_y = \left\{ \frac{M_2 \cdot r - \mu \cdot \rho}{M} + z \frac{M_2 \cdot c \cdot r - \mu \cdot \gamma \cdot \rho}{I_x} \right\} \cos(\Omega t)$$

Let us assume, that springs are placed in between the surroundings and the points regarded. Equations (1) and (2) are made under the conditions, that the body is not affected by forces of the springs. If this condition is to be retained we must have

$$(3)\quad d_x = d_y = 0$$

If these two conditions can be fulfilled the springs are stationary in the two directions, and as a consequence <u>the first order forces are not transmitted to the surroundings in these directions</u>.

We have the following equations at our disposal

$$(4)\qquad -\frac{M_1\cdot r-\mu\varrho}{M}-z\,\frac{M_1\cdot c\cdot r-\mu\cdot\gamma\cdot\varrho}{I_y}=0$$

$$(5)\qquad \frac{M_2\cdot r-\mu\cdot\varrho}{M}+z\,\frac{M_2\cdot c\cdot r-\mu\cdot\gamma\cdot\varrho}{I_x}=0$$

Following quantities are regarded as constants

The mass, M
The moments of mass inertia, I_x and I_y
The masses, M_1 and M_2
The half stroke, r

The product of the following two quantities is chosen as a variable

The mass of the counterweight, μ
The rotation radius of the counterweight, ϱ

and as dependent variables we chose

The position of the counterweight γ
The position of the springs z

If we solve equation (4) and (5) with regard to γ and z , we get

$$(6)\quad \gamma=\frac{c\cdot r(M_1 r-\mu\varrho)(M_2-M_1)I_y}{\mu\cdot\varrho\{(M_1 r-\mu\varrho)I_y-(M_2 r-\mu\varrho)I_x\}}+\frac{M_1\cdot c\cdot r}{\mu\varrho}$$

$$(7)\qquad z=\frac{(M_1 r-\mu\varrho)I_y-(M_2 r-\mu\varrho)I_x}{M(M_2-M_1)\,c\cdot r}$$

and these two equations are examined more closely in the following.

We shall examine counterweights in the interval

$$0\le\mu\cdot\varrho\le m\cdot M_2\cdot r$$

in other words from imbalance to n times full balance of oscillating and rotating parts.

<u>$\mu\cdot\varrho=0$</u>

From (6) and (7) we get

$$(8)\qquad \lim_{\mu\cdot\varrho\to 0}\gamma\to\infty$$

$$(9)\qquad z=\frac{M_1 I_y-M_2 I_x}{M(M_2-M_1)\cdot c}$$

There is no contradiction in (8) since there is no counterweight that cannot be placed anywhere !

<u>$\mu\cdot\varrho=M_1\cdot r$</u>

From the equations (6) and (7) we obtain

$$(10)\qquad \gamma=\frac{M_1\cdot c\cdot r}{M_1\cdot r}=c$$

$$(11)\qquad z=\frac{-(M_2\cdot r-M_1\cdot r)\cdot I_x}{M(M_2-M_1)\cdot c\cdot r}=\frac{-I_x}{M\cdot c}$$

Thus the counterweight must be placed adjacent to the cylinder.

<u>$\mu\cdot\varrho=M_2\cdot r$</u>

We find

$$(12)\quad \gamma=\frac{c\cdot r(M_1\cdot r-M_2\cdot r)(M_2-M_1)I_y}{M_2\cdot r(M_1\cdot r-M_2\cdot r)I_y}+\frac{M_1\cdot c\cdot r}{M_2\cdot r}=c$$

$$(13)\qquad z=\frac{(M_1\cdot r-M_2\cdot r)I_y}{M(M_2-M_1)c\cdot r}=\frac{-I_y}{M\cdot c}$$

The same remarks as above.

<u>$\mu\cdot\varrho=m\cdot M_2\cdot r$</u>

We obtain

$$(14)\quad \gamma=\frac{c}{m}\left\{\frac{\left(\frac{M_1}{M_2}-m\right)\left(1-\frac{M_1}{M_2}\right)I_y}{\left(\frac{M_1}{M_2}-m\right)I_y-(1-m)I_x}+\frac{M_1}{M_2}\right\}$$

$$(15)\qquad z=\frac{(M_1-m M_2)I_y-(1-m)M_2 I_x}{M(M_2-M_1)\cdot c}$$

Finally, by differentiation of (7) with respect to $\mu\varrho$, we obtain

$$(16)\quad \frac{\partial z}{\partial(\mu\varrho)}=\frac{-I_y+I_x}{M(M_2-M_1)\cdot c\cdot r}=\text{CONSTANT}$$

The function z = z ($\mu\rho$) represent then a straight line, where the sign of inclination is dependent on the size of moments of mass inertia.

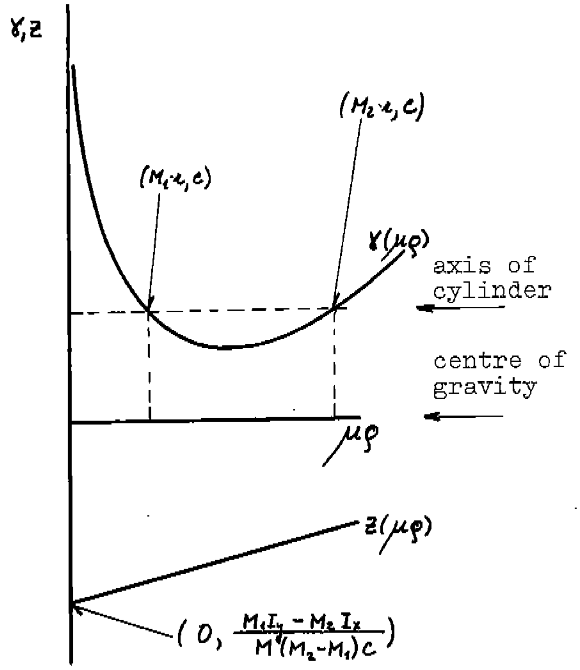

Fig. 4

Now we know the principal shape of the two functions. The shape of $\gamma(\mu\rho)$ appears from (8), (1o), and (12) as suggested in fig. 4. It appears from (16) that z ($\mu\rho$) is a straight line and in the figure we have chosen $I_x > I_y$ which corresponds to a positive inclination of the line.

It is clear that the figure offers several possibilities dependent on, if it is the size of the counterweight, its position or the fixing points of the springs, which is given.

Finally, it must be emphasized that if

$$c > 0$$

then

$$z < 0$$

and vice-versa. If the counterweight is on "the one side" of the centre of gravity, then the fixing points of the springs should be "on the other side" of the centre.

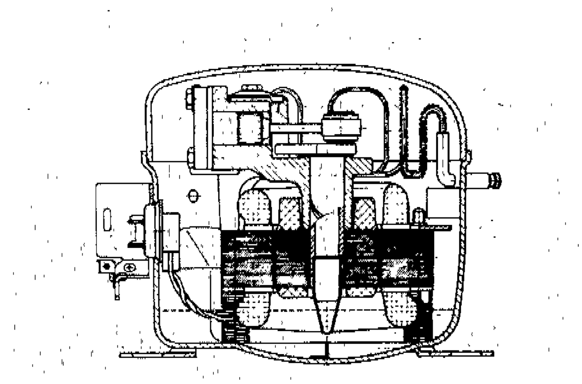

Fig. 5

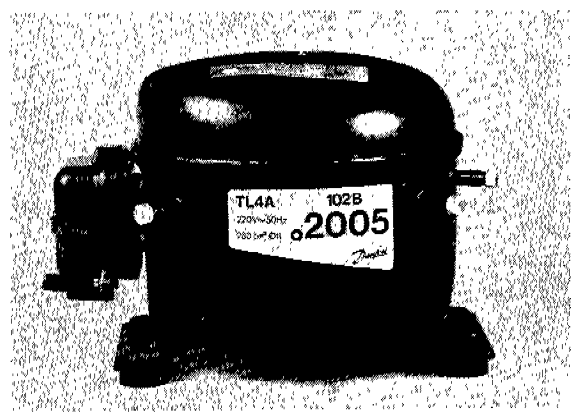

Fig. 6

CONCLUSION

It has been proved that with a compressor having a rotating and oscillating imbalance at a distance from the centre of gravity, it is possible to utilize the angular displacements of the compressor to counteract its horizontal translations.

For each individual degree of imbalance it is possible to calculate where the counterweight is to be placed and where the springs are to be fastened in order to achieve minimum horizontal displacements of the springs.

The Z-coordinate of the fastening points of the springs can be limited to an interval whose size is proportional to the difference between the moments of mass inertia with respect to the X and the Y axis. If the difference is zero, there is only one optimum plane for the spring fastening points regardless of the size of the counterweight.

Vertical movements of the fixing points cannot be counteracted, but in practice they are unaffected by the horizontal plane they are placed in.

Their size is dependent on the angular displacements of the compressor which means, in effect, dependent on the degree of imbalance and the horizontal distance from the centre of gravity.

The principle dealt with has been applied with great success by DANFOSS in a new generation of hermetic compressors, see fig. (5) and (6). The result is a marked reduction of the vibration level in relation to other known designs.

REFERENCES

(1) KJELDSEN and NISSEN: "Minimizing noise from compressor-driven refrigerators and freezers".
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