1984

Improved Suspension System for Reciprocating Hermetic Compressors

V. Bacci

M. Zgliczynski

G. Gili

IMPROVED SUSPENSION SYSTEM FOR RECIPROCATING HERMETIC COMRESSORS

Dott.Vieri Bacci
Aspera Eng.Dept.

ing.Marek Zgliczynski
Aspera R&D

ing.Giorgio Gili
Aspera Labs

INTRODUCTION

Domestic room and window A/C conditioners market is one of more interested in compressors with low external vibrations and high suspension reliability. The capacity range involved is between 2000 and 4000 Kcal/h(R22).
The more and more inquiry by users of European, Middle-East and of tropical countries has obliged the compressor designers to consider the real operating conditions, which are more severe for high ambient temperature (high running load and start pressures) and for the line voltage wide variation range (-20 / +6% of nominal value).
Both mechanical and electrical components must be accurately studied to avoid costly solutions and sometimes useless oversized components.
Aspera decided to face the problems connected to this market(EER,pulsations,vibrations and reliability) by mean of a substantial redesign of some hermetic compressor in production in all the capacity ranges requested.
In this paper we will describe the work done to optimize the vibration level and internal suspension reliability of one cyllinder compressor (J serie).

STARTPOINT AND FIRST TEST

The most critical compressor in the context of the same serie is always the one with the largest capacity: this is true when we look specially at mechanical parts life .In our case such a compressor was about of 35 cc/rev. The equivalent compressor is in production with the performance showed in tab.1.
The method used by Aspera to evaluate external vibrations is to take one measure on the top and on eight lateral points, four on the upper and four on the lower part of the shell, along orthogonal axes.

Tab.1

VIBRATION SHELL mm/sec	lat. avg	4.75
	lat.min	3.50
	lat.max	9.00
	op	6.10
SUSPENSION LIFE (0.7-2.7 MPa) Failure at start-stop(cycles)		5000
FIRST DESIGN SPRINGS		

So the lateral average value can be considered an acceptable index for vibration level. The lateral minimum and maximum can be interpreted as directional degree. In our case the maximum was along the cylinder axis and the minimum along the orthogonal axis.
To improve the suspension life we first increased the diameter of spring wire of ten percent. In the next table are summarized the results:

Tab.2

VIBRATION SHELL mm/sec	lat. avg	8.00
	lat.min	4.20
	lat.max	13.00
	top	7.40
SUSPENSION LIFE (0.7-2.7MPa) Failure at start-stop(cycles)		16000
Oversized spring (+10% wire diameter)		

The number of cycles we can consider practically acceptable in such start stop tions is about 20000.This solution did not reach such limit and the increase of vibrations was relevant and unacceptable.

BALANCING TEST

We have hypothized to reduce the vibrations by mean of optimal balancing of the moving parts. The theory of balancing suggests to choice two counterweights placed between the bearings and symmetrical with reference to the cylinder axis:the centrifugal effect will be the same of the rotating masses plus about fifty percent of alternating masses supposed concentrated on the eccentric. We have wanted to verify this assertion experimentally.It has been used the method of dynamical balance which implies (see fig.1) the measure of vibrations on two different planes : the measures are taken three times,first for the compressor alone and after with known trial test masses in the correction planes.
With a fast calculation we have obtained the correction masses.The percentage of alternating masses resulted of seventy percent and with a small phase displacement

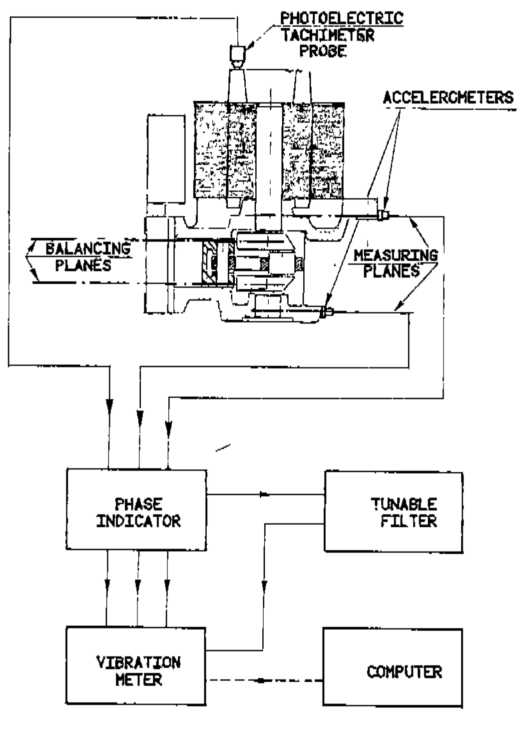

Fig.1 Balancing equipment

with reference to simmetry axis of crankshaft. The phase displacement resulted opposite inverting the rotating speed.
We cannot say to have sure arguments to explain these two phenomena; we can only consider that we are dealing with a physical system with many factors that can influence the measures and the results(friction, clearances,lubrication,suspensions, etc.).
However the results have been repeated in constant manner on all the compressor types we tested.
The vibration improvement measured on the compressor(see table 3) has been of about 25% with the original springs and with the spring wire increased.In the second case the improvement has not been enough to gain the results obtained with the original springs.

Tab.3

		Original	Oversized
		(mm/sec)	
VIBRATION	Lat. avg	3.55	5.70
	Lat. min	1.80	3.60
	Lat. max	6.00	8.90
	Top	4.50	7.00

Another way to obtain an improvement of vibration level was given from a possible lightening of alternating masses.Before to follow this way,which implied different production technologies and materials , we have investigated the physical conditions and the reasons of the spring failure in start-stop tests.

SUSPENSION ANALYSIS

Experimental system - description

The compressor has been mounted on a rigid base paying much attention that the suspension system was the same it was normally mounted on the shell.(see fig.2)
Three triaxial piezo transducers have been put between the base and the device on which were fixed the lower part of the springs. Three accelerometers have been fixed on the support with parallel axes to the force transducer.
A trigger device allowed the voltage cutoff and the data collection at the same crank angle to obtain the maximum of repeatibility. That was necessary because the signals from the force and acceleration transducers

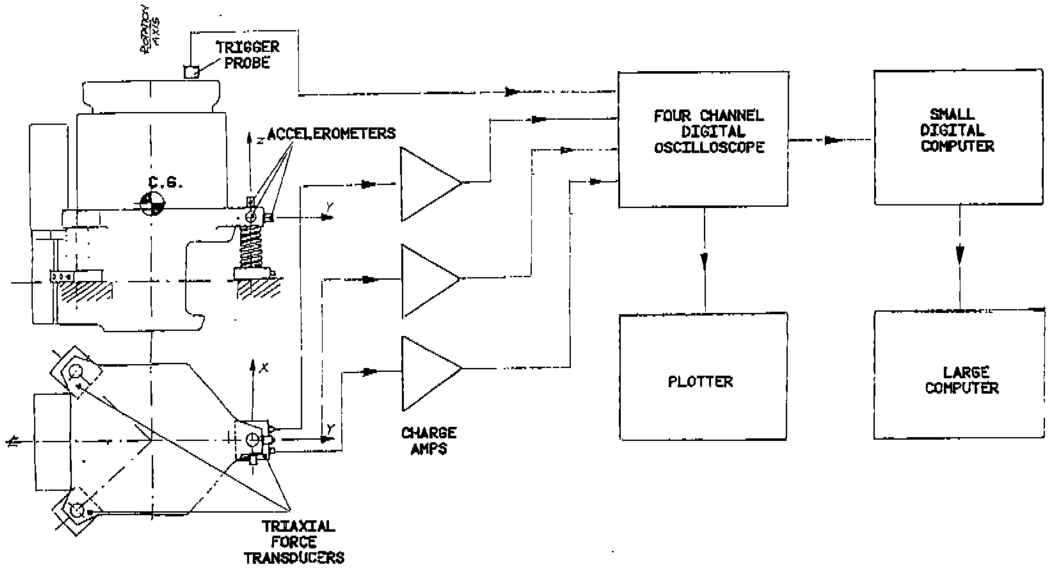

Fig.2 Schematic diagram of experimental arrangement

were sent to a four channel oscilloscope with digital storage : the fourth channel was used to take a RPM signal also to compare the data acquisitions . The equipment showed a very good repeatibility and a con-

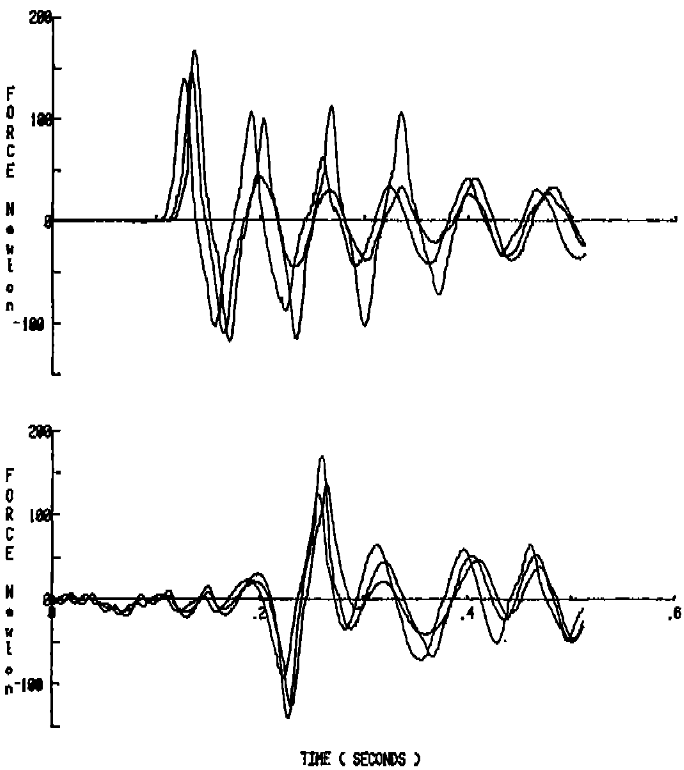

Fig.3 Forces transmitted during start (above) and stop (below) operations

sistent reliability of the measures.
Digital data were sent to a large computer for comparison and further processing.
First experimental data showed that forces detected from the three piezo-transducers at start and stop conditions (see fig.3) were nearly the same : this has been interpreted as a good choice of the location points for the springs with reference to the center-gravity of the body and to rotation axis ,almost coinciding.

First experimental results

During the starting operation we've noticed the remarkable effect of electric motor start and breakdown torque .
Influence of this two parameters is shown in fig.4a, where we've repeated the same start procedure with three different start capacitors (20, 50, 70 μF) and then with a high start torque three fase motor.
Having the same inertia moment of body and and no load pressure conditions , we can compare the forces trasmitted along the X axis.The intensity of force peak depends in first approximation on the maximum torque and the time to get the rated rpm depends on the start torque. Fig.4b shows how the instant torque changes during the first rounds.
In the shut down conditions, when the motor torque is setting to zero,the shaft rotation continues until the inertia can win the pressure forces. This problem has been successfully studied by other authors.

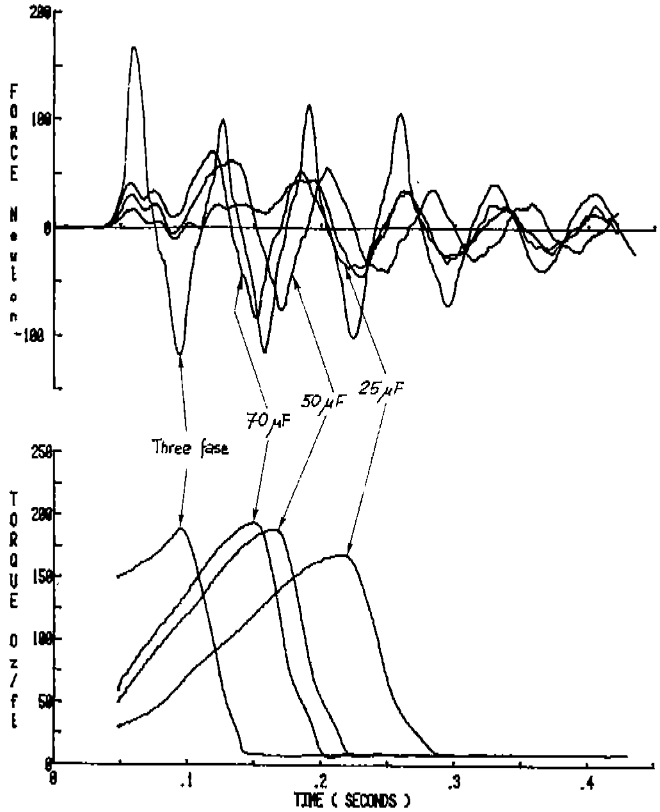

Fig. 4 Force and torque during start operations with different motors

Anyway, we measured these forces, and we've found that their intensity are very high and depends only from the load and not from motor performance.

Some theoretical considerations

Experimental test gave us all the data (force and deflection) that were necessary to evaluate by analytic way the stress on the springs.

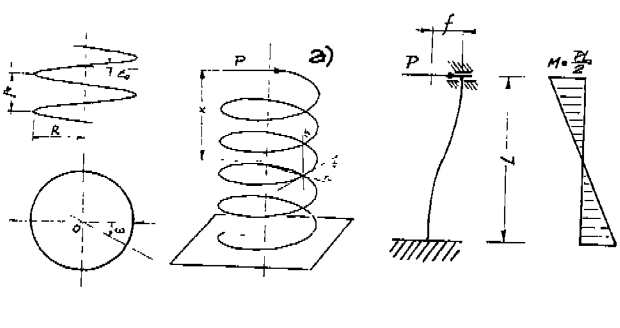

Fig.5 Spring scheme

As first approximation we have supposed the spring clamped at one edge and free at the other side following the scheme shown in fig.5a.
The basic equations of the moments acting on each section of the spring are given from :

$$M_t = -PR\sin\varepsilon_0 \sin\omega + Px\cos\varepsilon_0 \cos\omega$$
$$M_n = Px\sin\omega \qquad \text{(I)}$$
$$M_b = -PR\cos\varepsilon_0 \sin\omega - Px\sin\varepsilon_0 \cos\omega$$

and the lateral displacement for a spring with free lenght L

$$f_0 = \frac{2PL^3}{3\pi E r^4} \, \frac{2+\nu\cos^2\varepsilon_0}{\sin\varepsilon_0} \qquad \text{(II)}$$

Substituting the real values for P into (II) together with the contruction data, the results for the lateral displacement were wrong of about five times.
After careful assurance that the experimental data were correct we looked for a better modeling of boundary conditions.
So we have supposed that the sides of the spring move relatively in a prismatic coupling. In this case the bending moment in the fixed edge become a half and the deflection one to four respect the spring clamped at one side only:

$$M = \frac{M_0}{2} \;, \quad f = \frac{f_0}{4} \qquad \text{(III, IV)}$$

Experimental data confirmed this hipothesis In fact the amplitude of deflection along the z-axis is negligible against the x-axis displacement.
A further refinement has been brought to the (I) for the stress due to static vertical force Q (the following terms must be added):

$$M_t = -QR\cos\varepsilon_0$$
$$M_n = 0 \qquad \text{(V)}$$
$$M_b = QR\sin\varepsilon_0$$

To compare the global stress of the spring to the fatigue limits for the specific material it has been calculated the ideal stress by mean of Mohr fracture hypothesis (for circular section , r=radius)

$$\sigma_{id} = \frac{4}{\pi r^3} \cdot \left(\frac{m-1}{2m} M + \frac{m+1}{2m} \sqrt{M^2 + M_t^2} \right) \qquad \text{(VI)}$$

where

$$M^2 = M_n^2 + M_b^2 \qquad \text{(VII)}$$

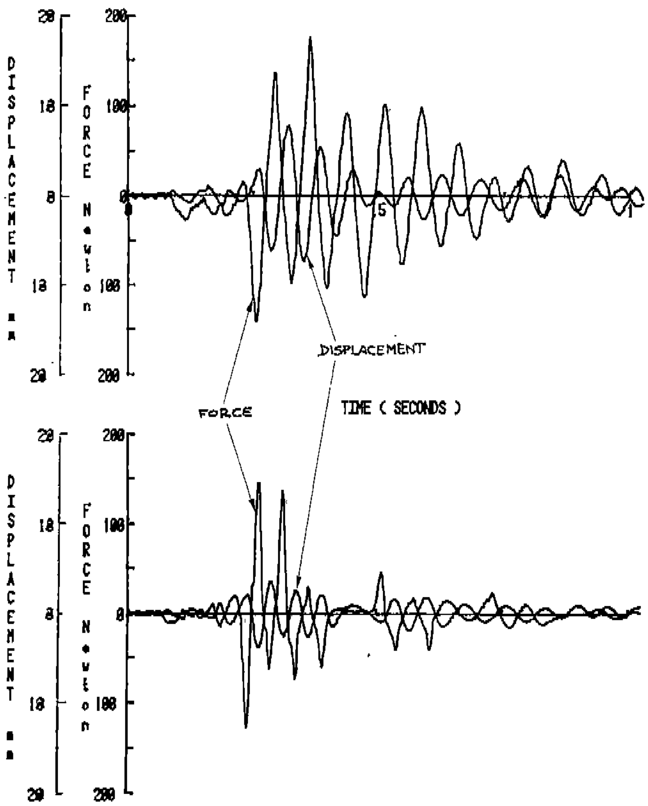

Fig.6 Force and deflection in old (above) and in improved configuration (below)

The new model showed itself very good to explain the experimental quantities.Moreover the stresses were too high if compared to the fatigue limits of material. The refinement of the model has showed us

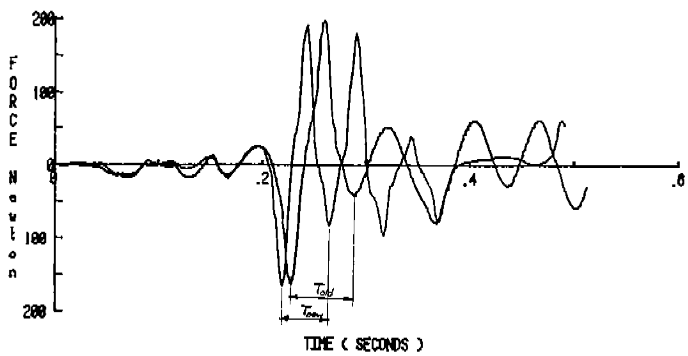

Fig.7 X-axis force comparison between old and new solution

the influence of boundary conditions.So we have verified via calcula many boundary possibilities until we have found a method to limitate the spring motion and guarantee a reduced stress to 1/2 under maximum load and that would be uninfluent in running conditions.Figure 6 shows the differences in force and displacement between the original design and the improved suspensions at stop load.

Fig.7 shows the difference between the lateral stifness on the same spring with different boundary conditions and the changed the free oscillating frequency.

On table 4 are summarized the results of new suspension on the compressor with optimal balance:

Tab.4

SHELL VIBRATION mm/sec	lat. avg	3.55
	lat.min	1.80
	lat.max	6.00
	top	4.50
SUSPENSION LIFE (0.7-2.7 MPa) Failure at start-stop(cycles)		40000

CONCLUSIONS

The example we presented points up the importance of spring sizing and spring limiting systems under heavy loads conditions. The experimental test facility instrumentation and data processing allow the study of many compressor types with different number and cylinder arrangement.

In any case the target is the use of softer suspensions but with increased spring life also in the most severe load conditions.

Nomenclature

r - spring wire radius
R - mean radius of spring coil
P - lateral load
Q - vertical load
L - free lenght of spring
ε_o - coil angle
ω - progressive angle
E - Young's modulus
ν - Poisson's ratio
σ_{id} - ideal uniaxial stress
M_n,M_b - flexural moments
M_t - torque moment
f - spring end lateral deflection

References

1) J.F.Hamilton - Modeling and Simulation of compressor suspension system vibrations, Short Course Purdue 82,
2) A.Futakawa, N.Muramatsu ,K. Tsuchiya Transient stress produced in internal springs of hermetic refrigeration compressor during start and stop operations, Purdue 80,
3) O.Belluzzi - Scienza delle costruzioni Vol.2 Zanichelli ,Bologna 1977,
4) R.Giovannozzi - Costruzione di macchine Vol.1 and 2. Patron, Bologna 1965,
5) Fundaments of balancing machines- Technical Review - Bruel&Kjaer 1981,
6) L.Buzzi - Equilibratura - CEMB 1971,

ACKNOWLEDGEMENT

The authors are grateful to Dott. G.Lampugnani for his contribution in setting up the lab facilities and result discussion.