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CONTROL OF THE SOUND GENERATED BY A ROTARY COMPRESSORS

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ABSTRACT

The sound radiated by the compressor has been reduced by use of the special designed thrust bearing which has been pressfit within countersink recess in the rotor. This results in forming a single frictional pair, the lower surface of the thrust bearing against the upper end face of the compressor main bearing hub, thereby reducing the amount of sliding friction within the compressor. The polyamide material used to form the thrust bearing is characterized by a very low coefficient of static and kinetic friction. It helps to diminish the resultant friction thereby increasing the efficiency, reducing overall sound radiated by the compressor as during operation so at the start.

INTRODUCTION

There are a wide variety of compressors for use in air conditioners and refrigerators, including a reciprocating type, a screw type, a rotary type, and scroll type compressors. The rolling piston type rotary compressors are widely used because of their small size, lightweight, low cost and high performance. However vibration and noise characteristics of rotary compressors required further improvement to compete with scroll type compressors which has perfectly balanced motion, continuous suction and discharge flow with very low gas pulsations and absence of the dynamic valves.

A rotary rolling piston type compressor shown in Figure 1, has a cylinder mounted in the lower part of a hermetic shell and a rolling piston driven by a crankshaft carried by the rotor of an electric motor whose stator is fixed internally to the upper parts of the shell. The thrust bearing surface at the pump end of the crankshaft supports the pressfitted rotor weight and accepts dynamic and start loads associated with the shaft. An external part of the cylinder and the piston compression and a suction chamber. In operation the piston eccentrically rotates in the compression chamber and compresses the refrigerant gas which is discharged from the discharge port into the discharge muffler. The hot refrigerant gas (150° C; 3.5 MPa), flow through the motor stator-rotor gap to the cavity above the stator and farther - to the coils of the air conditioning unit. The lower part of the compressor shell contains oil necessary for the lubrication.

NOISE AND VIBRATION TEST RESULTS

Vibration and consequently sound radiation result from the collision and friction of the compressor components joined in kinematic pairs, suction and discharge gas pulsations, and electromagnetic forces of the motor.
Acoustic measurements have been performed in the anechoic room with rotary type vertical crankshaft compressors operating at conditions specified in ARI Standard 520 (USA) -7.2° C (45° F) evaporating temperature and 18.3° C (65° F) condensing temperature. The compressor has been operated at least 2h to reach thermal stabilization before the acoustical test. Figure 2 shows one-third octave band spectra of the rotary compressor running at 60 Hz power line frequency. The high level components of the sound spectra are located in the range 1600 Hz-6300 Hz with the maximum peak at 3150 Hz (2018 Hz - 3548 Hz of narrow band frequency limits).
The scheme of vibration and noise generation mechanism shown in Figure 3 has been developed on the base of the rotary compressor study in the Tecumseh Products Company Acoustic Laboratory. The rolling piston type compressor induces vibrations by the periodic change of the gas compression moment and fluctuation of the electric motor torque. The refrigerant gas pulsations taking place on both the low and high pressure sides, but the suction pressure pulsations are suppressed to some degree by the external accumulator.
The electric motor is held by shrink fit in the upper part of the housing and located on the high pressure side of the compressor so that stator winding and rotor are affected by a pulsatory magnetic field, as well as by pulsating discharge gas flowing through narrow stator-rotor gap. The discharge gas pulsation trigger resonance of the cavities located above and below the stator inside of the housing.

In the course of our experimental study we took the opportunity to map the surface of the housing for vibration magnitudes. The boundary points of the lines resulted from the intersection of the vertical and the horizontal planes have been chosen as measuring points (46 points total). The recorded (in dB) peak magnitudes of the acceleration at a single narrow band frequencies have been marked on the development of the compressor housing as shown in Figure 1. The McDonnell Douglas CAD/CAE system "Unigraphics" (Version 11) has been used to create, display, store, retrieve, and plot all graphical data. The contour plot for harmonics #3 (174 Hz) and #55 (3190 Hz) are shown in Figure 4. As the piston rotates in the compression chamber of a rotary compressor, the vane moves in and out of the chamber to maintain contact with the side of the piston and divide low and high pressure chambers. A varying pressure differential is applied to the exposed portion of the vane as the vane oscillate in the slot. The study performed by K. Sano and K. Mitsui [1] indicated that impulsive noise of the impact wave forms resulted from the slap of the vane against the walls of the slot have been contributed at frequencies above 2000 Hz. Another source of noise contributed at frequencies above 2000 Hz, is the collision of the discharge valve against the seat and retainer.

Analysis of the contour map shows the following:

1. High levels of low frequency vibrations were recorded on the housing part adjacent, above and below the motor stator and on the accumulator strap.
2. High levels of vibration in the frequency range 2000-4000 Hz have been recorded on the surface of the housing below the motor stator particularly for harmonics #52 (3016 Hz), #55 (3190 Hz) and #59 (3422Hz), at the points located near the wire welds and suction line .

A compressor cycling noise (beats) are defined periodic increase and decrease of amplitude (beat frequency) that results from the superposition of two simple harmonics of different (but close) frequencies $\acute{\omega}_1$ and $\acute{\omega}_2$. The period of beats

$$T_B = 2\pi / | \omega_2 - \omega_1 | = = 2\pi / \Delta\omega \qquad (1)$$

in sec and the beat frequency, $f_B = S / T_B = \Delta\omega S / 2\pi$ in Hz, where S is slip frequency. We can observe sequence of major and minor maxima if n oscillations of the same amplitude with frequencies deviated successively by $\Delta\omega$ have been added.

The period of beats is independent of the number of oscillations that are added, but the number defines the principal maximum. The amplitude of the beat can be computed from the equation below.

$$A(t) = A_0 \sin(\omega t + \Psi) \qquad (2)$$

$$A_0 = [A_1 + A_2 + 2A_1A_2\cos(\omega_2 - \omega_1)t]^{1/2} \qquad (3)$$

where A_1 , A_2 correspondingly are amplitudes of first and second components, and Ψ is a phase angle.

The electromagnetic noise of the rotary compressor motor combine running frequency components, corrected by slip frequency S, and power line frequency components. As shown in the work of T. Uetsuji at all [2] and T. Mochizuki at all [3] the eccentricity and inclination of the motor rotor to stator is an important factor governing the generation of motor electromagnetic noise.

It is useful to mention that the compressors electric motor located on the high side and hot refrigerant flowing through the motor stator-rotor gap may trigger aerodynamic unbalance, in addition to the dynamic unbalance forces acting on the crankshaft with rotor on its end and thermal deformation forces.

THE THRUST BEARING AS A SOURCE OF NOISE

One of the sound sources within the compressor is the mechanical friction between the crankshaft thrust surface and facing surface of the outboard bearing. Noise produced by such

a hydrodynamic bearing become significant when a full oil film is not generated or when the bearing operating conditions are such that the self-generated instability known as oil whirl occurs. The rotary compressors crankshaft thrust surface has a half-moon shape and located on one side of the eccentric. Due to the limited space the thrust area is relatively small. It creates conditions for partial or total overloading of the bearing. The total axial force applied to the thrust surface $F = F_M + F_R + F_C$, where F_M is the motor axial (solenoid) force, F_R and F_C correspondingly is gravity force of the rotor and crankshaft.
The motor axial (solenoid) force can be computed from the equation below:

$$F_M = 0.0117P \ (60/f) \ (I_{M0} E_0 / L_0) \ (L_0/L)^2 [1 - 2\pi^{-1} \operatorname{ctn}^{-1} (h/g)] \qquad (3)$$

Where **P** - phase number (for single phase =2), **f** - line frequency, I_{M0} - magnetizing current in amperes, E_0-line voltage, L_0- stator core stock height, **L** - effective core height, **h** - misalignment, and **g** - rotor-stator air gap.
Another factor which significantly increases the vane is stick-slip motion of the mating surfaces, metal to metal contact due to poor oil film generation saturation of the refrigerant in the oil (holes in the oil film), and interrupted path in the oil film generation due to assimetry of the thrust surface . The dynamic of the thrust bearing during start and operation of the compressor is governed by the torques exerted on it.

When the lower end of the crankshaft is in contact with the facing part or the outboard bearing, then metallic friction can be assumed Bondary friction is considered for this thrust bearing which support the gravity of the rotor and shaft. Since the configuration of this thrust bearing is a parallel face, a geometric converging wedge for fluid friction is not shaped. The boundary friction loss F_L is

$$F_L = 2\mu \, W_S \, (R_{S2}^3 - R_{S1}^3) / 3 \ (R_{S2}^2 - R_{S1}^2) \qquad (4)$$

where μ - coefficient of friction R_{S1} and R_{S2} - inside and outside radius of the thrust surface W_S - weight of rotor and shaft. With the addition of the axial solenoid downward force the loss factor will be significantly higher.

MODIFICATIONS AND RESULTS

The sound and vibration absorbing damper have been developed to reduce sound radiation of the compressor. The area of the housing with the highest level of vibration identified in the initial acceleration survey of the structure (Figure 4) have been chosen for modification. The sound and vibration absorbing damper contained the metal wire loops wound close around the housing of the compressor so that conjugate loops and surface of the housing have had interface contacts [4, 5]. The noise reduction and damping of vibrations is due to slipping of wires - housing surface interfaces and pumping of air (gas) caused by transverse relative motion of the wire loops and housing [6]. Results of the experimental study shown in Figure 5 indicate up to 3 dBa reduction of overall sound. By changing location, quantity, gage, profile, or material of the wire we can achieve the necessary degree of vibration and noise reduction. The vibration and sound absorbing damper can be effectively used in aggressive medium for a wide range of temperatures and does not prevent heat exchange of the compressor (high side housing) with surrounding medium.

The mechanical friction associated with a vertical rotor and crankshaft combination as it rests upon and rotates about the frame bearing hub is reduced both at start up and during the compressor operation by utilizing a thrust bearing formed of a polyamide material [7]. By press fitting the thrust bearing within the counterbore formed in the rotor, rotation of the thrust bearing relative to the rotor is prevented. This results in rotational contact between a single frictional pair, the lower surface of the thrust bearing against the upper end face of the bearing hub, thereby reducing the amount of mechanical friction within the compressor. In the preferred embodiment (shown in Figure 1), the polyamide thrust bearing is formed of torlon as produced by Amoco or Vespel as produced by DuPont. By reducing the friction caused by the radial reaction of the crankshaft at compressor start up and during operation, the present modification increases overall compressor efficiency and reduces radiated sound.

The polyamide material used to form the thrust bearing of the present invention is characterized by a very low coefficient of static and kinetic friction. This results in reduced mechanical friction and reduced power consumption associated with starting and operation of the compressor. Another beneficial characteristic associated with polyamide is it's broad temperature range thermal stability. Even unlubricated polyamide thrust bearings are capable of withstanding approximately 300,000 lb. ft/in. minimum with a maximum contact temperature of 740° F. Lubrication oil is delivered by the crankshaft to the thrust bearing surface, thereby further reducing the coefficient of friction during compressor operation. Circular shape of new thrust bearing helps to form cercumferential periodic pattern of the oil film. The consequence of the flow pattern in the bearing is extremely important to the rotor stability. When an oil flow has a circumferential pattern it generates a dynamic effect which creates rotating forces that, in feedback, act on the shaft and cause lateral precession motion. New thrust bearing helps to eliminate occurance of self-exited vibrations associated with such phenomena as oil whirl which triggered by fluid dynamic forces generated in the bearing. Yet additional advantages of the bearing relocation and modification are: vibration dampening, lack of corrosion, broad temperature range thermal stability, and superior chemical and abrasion resistance. Results of the experimental study shown in Figure 6 indicate up to 3 dBA reduction of overall sound.

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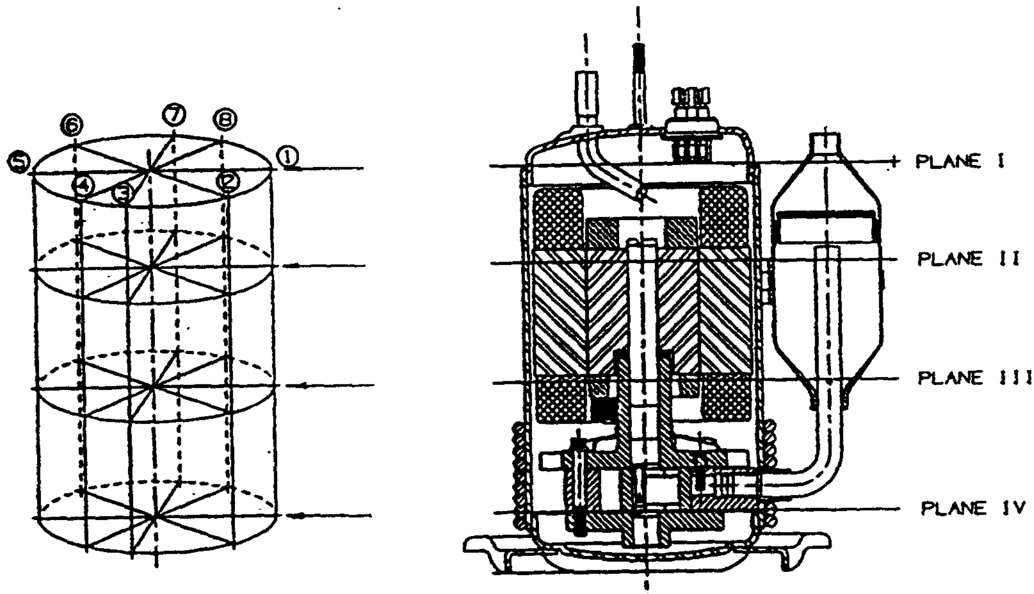

FIG. 1. THE VIBRATION MEASUREMENT POINTS

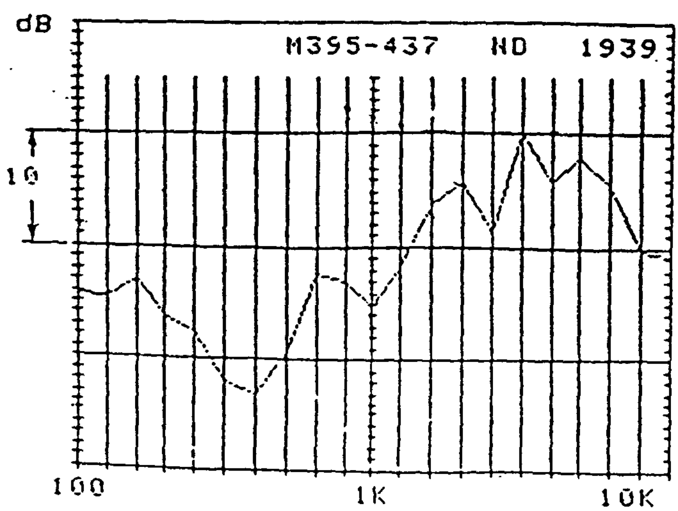

FIG. 2.1/3- OCTAVE BAND SOUND RADIATION SPECTRUM

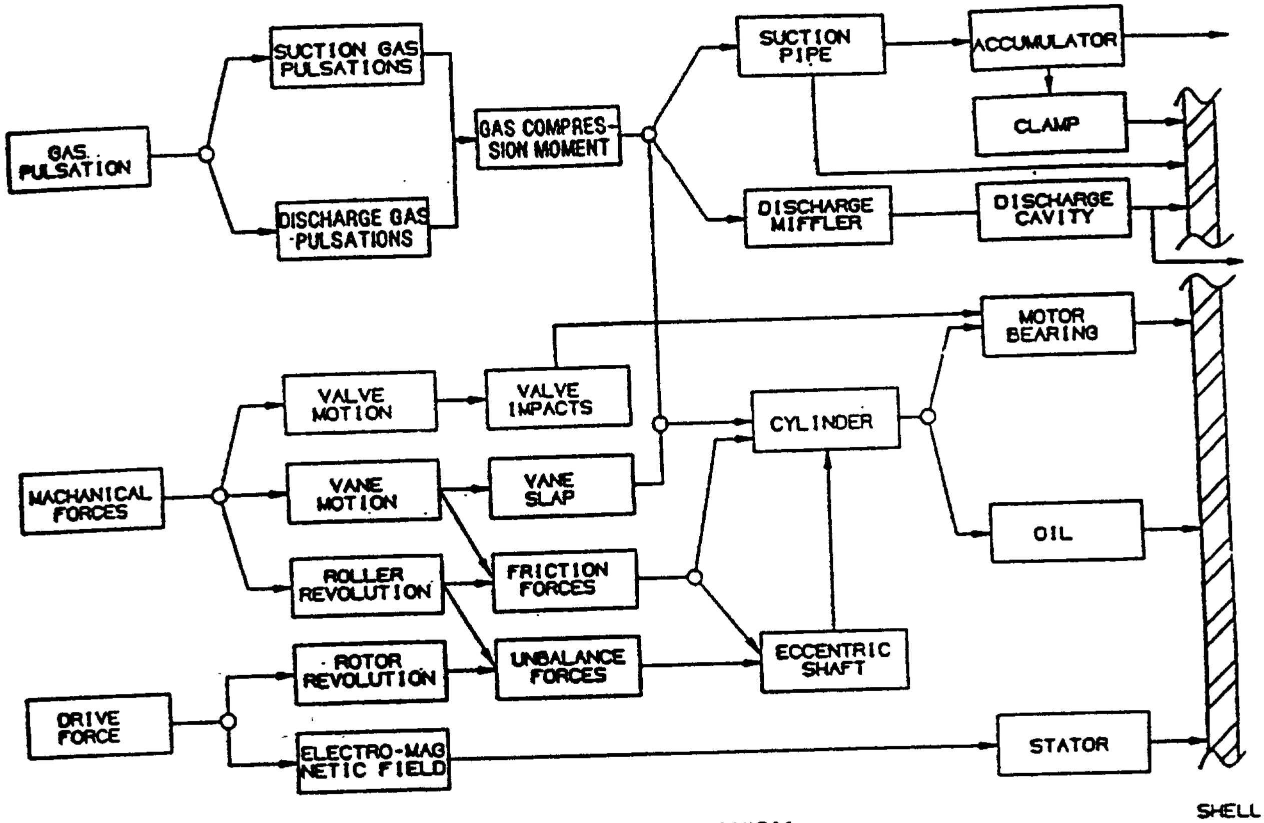

FIG. 3.VIBRATION AND NOISE GENERATION MECHANISM

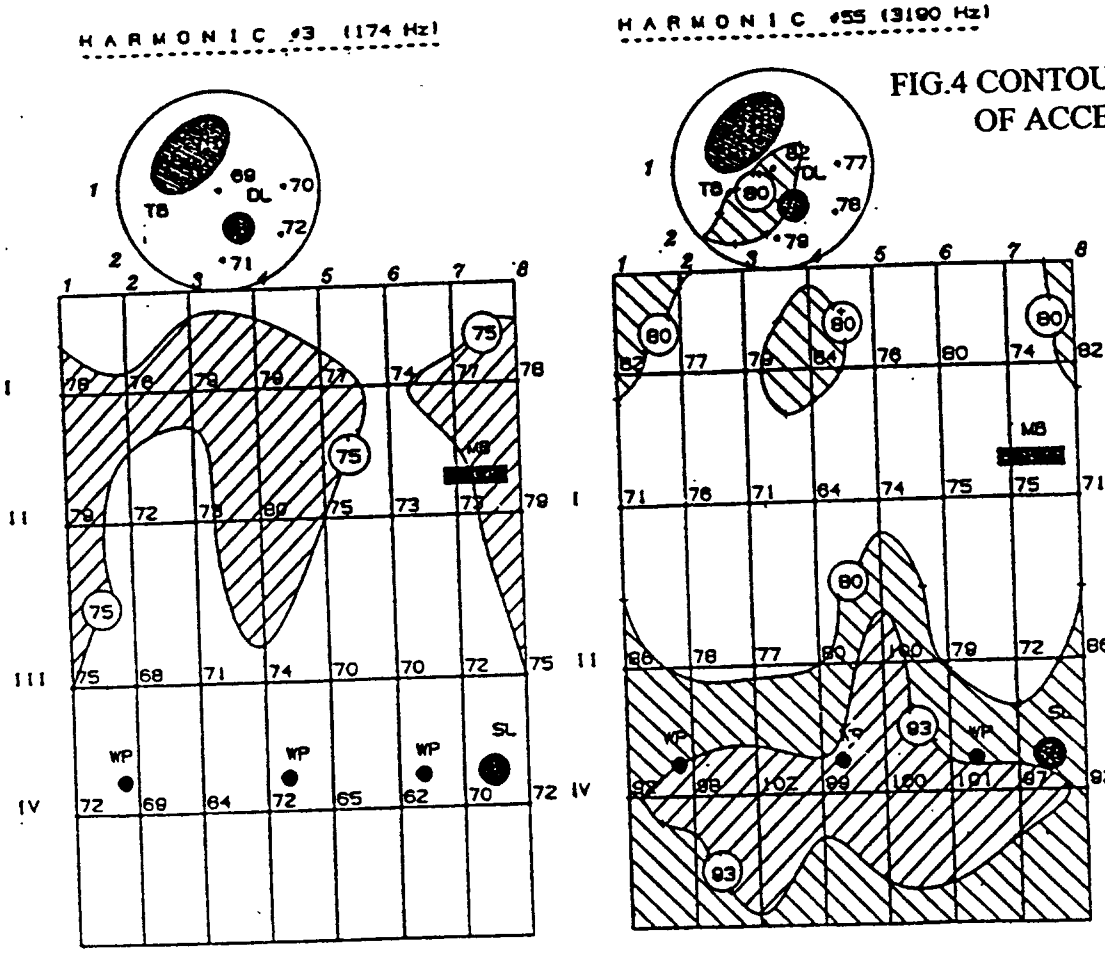

FIG.4 CONTOUR PLOT OF ACCELERATION

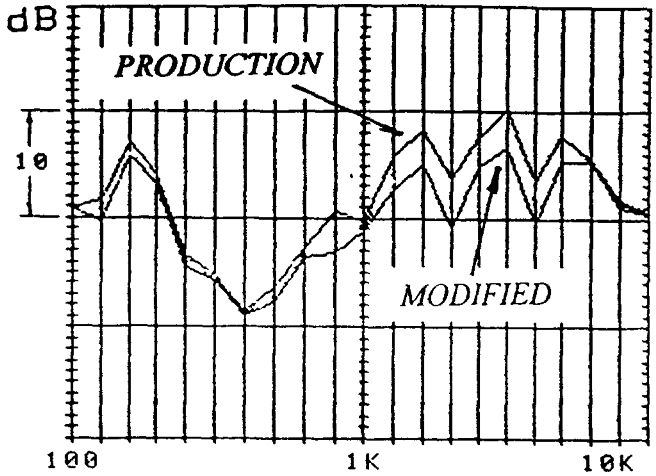

FIG.5. EFFECT OF THE SOUND AND VIBRATION ABSORBING DAMPER ON THE ROTARY COMPRESSOR NOISE

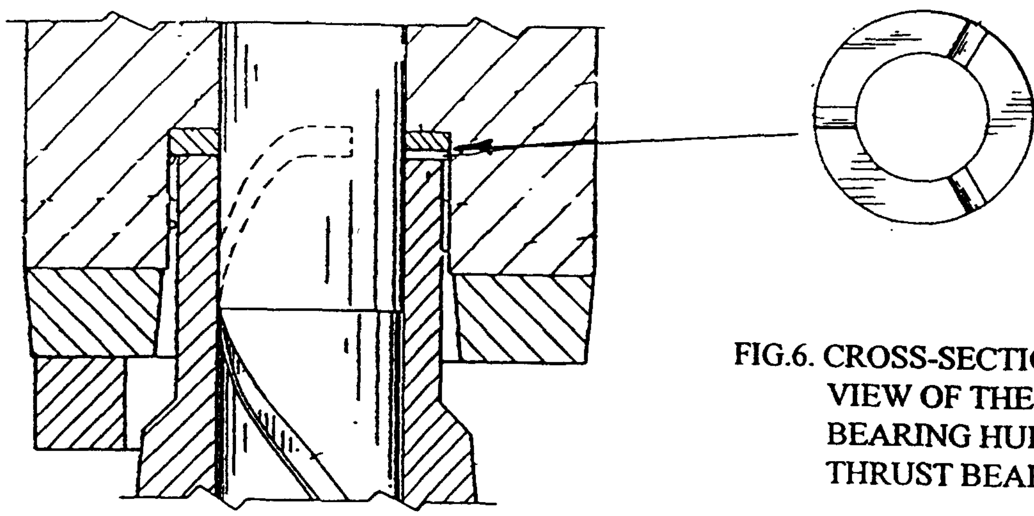

FIG.6. CROSS-SECTIONAL VIEW OF THE ROTOR, BEARING HUB AND THRUST BEARING

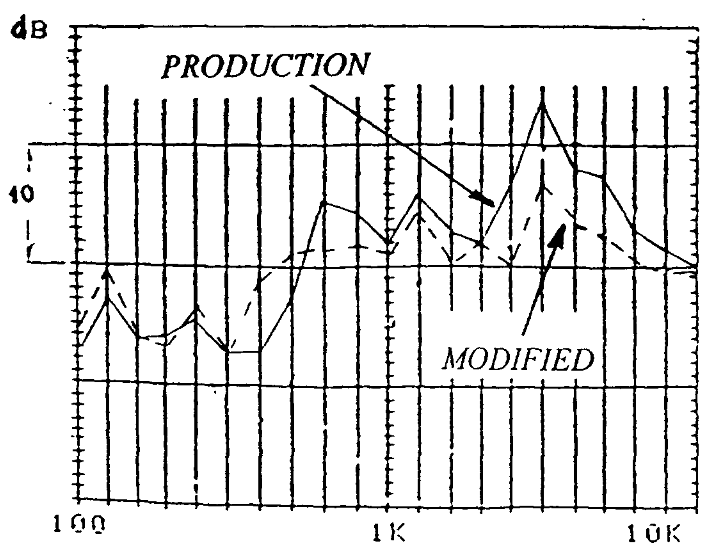

FIG.7. EFFECT OF THE VESPEL THRUST BEARING ON THE ROTARY COMPRESSOR NOISE