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THE APPLICATION OF ADVANCED ANALYSIS METHODS TO THE REDUCTION OF NOISE FROM AIR COMPRESSORS

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ABSTRACT

This paper discusses the use of modern analysis techniques in the reduction of noise from mechanical, structural and gas flow sources of air compressors. Noise legislation, already existing in Europe, coupled with increasing environmental awareness is causing manufacturers to make a greater effort to reduce noise radiation from their products. To achieve this objective, engineers can call on such techniques as finite element analysis, advanced frequency-domain and time-domain based analysis and computer modelling of gas dynamics for muffler systems, in addition to the traditional methods. Each of the techniques will be discussed in turn, and its effectiveness on the noise reduction of air compressors assessed.

INTRODUCTION

The need to reduce compressor noise is driven by two forces - the market demands and legislation. Today's customer demands quieter and more refined products. Legislation aims to control environment noise and is increasingly being enforced. In order to meet both customer and statutory noise requirements, analysis and control techniques have had to be developed accordingly.

In the European Community (EC) many types of off-highway equipment are subject to legislation restricting their radiated noise levels. As long ago as 1963, construction sites were identified as major noise nuisances (Reference 1) and in 1974 local authorities were empowered to restrict noise emission from these sites under the Control of Pollution Act (Reference 2). In 1979, the EC issued a directive giving measurement procedures for assessing noise radiation from construction plant and equipment (Reference 3) and this was followed in 1984 by further directives giving noise limits at certain operating conditions for individual equipment classes, including compressors (Reference 4). Adopted by all member states this means that no manufacturer or importer may market, within the EC, any compressor for use on construction sites which exceeds the specified sound power level when operating at its nominal operating pressure and speed.

Although the directive refers to compressors intended for use on construction sites, in practice, since the manufacturers may not know where their products will be used, all compressors are generally tested and certified. Proposals have been put to the EC to further reduce legislative noise limits of compressors (along with other equipment classes) by 2 to 6 dBA. Although these reductions have not yet been finalized, it is almost certain that some further reductions will be enforced.

The compressor industry in Europe has long been producing machines with canopies or enclosures to provide weather protection as well as for appearance. It has, therefore, been a relatively simple step to enhance the canopies to the level of good acoustic enclosures. However, with the continuing trend towards lower noise limits, and increased customer demand for more refined products, the manufacturers of compressors, and their drive motors (often internal combustion engines), are increasing their focus on the fundamental sources of noise within their products.

ANALYSIS AND CONTROL TECHNIQUES

Finite Element Analysis

Over recent years the authors have undertaken a significant programme of research aimed at developing analytical techniques for the design of low noise I.C. engine and reciprocating compressor structures. Much attention has been paid to crankcase design for minimum radiated noise combined with acceptable levels of weight, operating stress and distortions. This has been due to the increasing demands of noise legislation coupled with commercial pressures for lighter weight, lower noise products.

In general, reduced radiated noise levels may be achieved by the addition of mass, either in terms of increased section thicknesses, or by the adoption of more radical features such as deep ribs or even underplate or bedplate designs. The challenge in this approach is to ensure low noise with minimum weight penalty, requiring optimum distribution of material to minimise noise radiation, compatible with requirements of operating loads and production.

Detailed understanding of vibration transfer paths and the relative importance of different excitation sources may only be achieved by the use of finite element analysis. This approach is complementary to any experimental work which may be carried out, but it also may be applied to designs for which no hardware yet exists.

The finite element method provides a ready and proven tool for the modelling of geometric effects. As with any tool, experience in use is required, and a step-by-step methodology has to be established in order to use the method effectively. As a result of numerous applications the authors have converged on an approach whose basic stages may be summarized as follows:

- Generation of finite element model and prediction of weight.
- Prediction of free vibration behaviour.
- Correlation of predicted and measured free vibration behaviour (if available), followed by refinement of the model in critical vibration areas to match measured results.
- Prediction of response of the structure to operating forces.
- Identification and ranking of excitations and vibration response mechanisms, in terms of contributions to the total radiated noise spectrum.
- Treatment of individual excitations and/or transfer mechanisms in decreasing order of importance.
- Repeat analyses of the last three steps until a satisfactory solution is found.

Experience has shown that the best trade-off of accuracy vs. analysis time and cost for noise prediction work is achieved using finite element models constructed of a mixture of solid and shell element types. The authors have performed a significant amount of research into the rapid generation of these "solid-plus-shell" finite element models. This work has covered such aspects as the correct specification of properties at solid/shell element interfaces, and knowledge of acceptable approximations to be made in the modelling of such features as fillet radii and gasket joints. For a reciprocating compressor analysis such a model may comprise up to 50,000 degrees of freedom.

In practical applications it is usually sufficient to predict free mode shapes for natural frequencies up to approximately 3 kHz (dependent upon operating speeds). The prediction of the 40 or 50 modes in this frequency range for a model of this size is a significant computing task, but not beyond today's engineering computers. The reduction of higher-frequency noise sources is best achieved either by experimental techniques, or by the use of local finite element models.

It is important to note that the correlation of predicted and measured free mode shapes cannot be carried out by eye for anything other than very low frequency modes. Instead, one has to develop automated modal assurance criterion techniques, which yield a mathematical comparison of the free mode shapes, and so enable checking of "like-with-like" free modal frequencies. Once the free mode shapes and frequencies have been compared, the finite element model may be modified to improve either the frequency match, or the shape match, or both. It is unreasonable to expect a perfect shape match for all modes, not least because of the variations in performance of individual, nominally identical, structures. Such variations may be particularly pronounced in pre-production or prototype components.

The prediction of response due to operating forces may be divided into two main stages, the specification of operating forces, and the calculation and synthesis of the individual modal response to those forces. For most applications, we find that it is sufficient to use "representative" forces in the development of low noise structures. These forces comprise the frequency spectra of vertical and horizontal components of main bearing loads plus the spectra of gas pressure forces. These are calculated using standard analysis routines which do not include the interaction of the running gear (e.g. crankshaft) with the crankcase. Other excitation forces (such as piston slap) may not, at present, be calculated with any accuracy, and thus the response of the structure to such forces must be assessed on a comparative basis.

The main exception to this use of representative forces occurs when significant crankshaft/cylinder block interaction is suspected. These can be analysed using crankshaft analysis methods which include the dynamic interaction of the crankshaft, oil film and crankcase. If such coupled resonances are shown to be probable, the combined analysis would be performed to ensure correct specification of remedial measures. The first stage of such an analysis would be the calculation of the dynamic stiffness of the crankcase, using unit loads applied to each main bearing of the finite element model.

Such computer programs for calculation of forced vibration response have been developed by Ricardo, and use the free modal vibration data as input. The programs have been written to enable output of both the total noise or vibration spectra, and the contribution of individual free modes to that total. By this means it is possible to identify which free modes need to be either reduced in response, or changed in frequency to avoid significant excitation. Two main methods of calculation of radiated noise are currently in use. The Rayleigh approximation, which is equivalent to the assumption of a flat plate vibrating in an infinite baffle, yields good results for most work, and is very rapid. This approximation is used to calculate the sound radiated from each face of the structure. The results from each face are then simply summed to yield the total radiated noise. For detailed investigations, or for calculation of sound power and intensity in areas of complex geometry, a true solution of the general wave equation is used. A boundary element method has been developed for this purpose and is described in Reference 5. The application of the method is illustrated on an example in Figure 1.

The treatment of free modes identified as being problematic in the calculation of forced response is based on a study of the free mode shape, the corresponding strain energy density distribution and the frequency sensitivity of the mode to element thickness. Frequency sensitivity studies are particularly useful in the identification of those parts of the structure for which the mode is most affected by the addition of mass or stiffness.

The material re-distribution is carried out by experienced designers, to ensure that critical parameters such as ease of production and response to quasi-static (operating) loads are not forgotten. Individual modes (or their excitations) are treated in order: it is essential to reduce the noise generated by the most dominant free mode before any other. Modified designs are analysed using a repeat of the above procedure. For analyses based on representative forces, no re-calculation of those forces is necessary, such repeat analyses are generally performed in less than half the time of the original.

A promising development in this area would be an automatic optimisation of designs for low noise or weight. Although many workers have reported successful automatic optimisation of simple components (such as ancillary component brackets or suspension arms), we are not aware of any reliable automated system for components as complex as a compressor crankcase. The authors have spent considerable effort in the application of an optimisation program based on the use of static analysis. The conclusion from this work was that any optimisation procedure must be based on dynamic analysis, to include the effects of mass distribution. As a follow-on, Ricardo have now embarked on a joint research programme with academic establishments and finite element code firms to develop an automatic design optimisation system, based on dynamic analysis of the "reduced" problem. The work is divided into two broad areas: mathematical derivation of the mapping between the modal stiffness and mass matrices and the real structure, and the specification of design limits (such as minimum casting thickness). However, due to the scale of the problem, no results are expected for several years.

Other areas of analytical noise prediction research include the use of coupled fluid/structure modelling for use in the design of shielding systems, and the use of metal-matrix composites to reduce reciprocating mass (and hence vibration and noise).

Experimental Modal Analysis

Experimental Modal Analysis (EMA) is widely used to quickly and reliably identify mode shapes and frequencies in a resonant structure - typically an engine or compressor crankcase - but equally applicable to components such as transmission cases, oil pans and covers.

Provided hardware is available, EMA can be a preferred alternative to Finite Element Analysis on the grounds of speed and accuracy although EMA is limited in its modelling scope to "small" changes to the structure. Finite Element Analysis, as described above, is a considerably more versatile approach to modal analysis and structural modelling. EMA therefore should be viewed as complementary to analytical methods.

Time/Frequency Domain Analysis

This section describes the application of digital signal processing techniques used primarily to identify the sources of noise from rotating and reciprocating machinery in order to reduce noise levels and improve noise quality. Whilst these techniques have been developed principally for the identification and reduction of internal combustion engine and transmission noise, they are equally applicable to reciprocating and rotary compressors.

Measurement of overall noise levels. Overall noise levels are traditionally obtained using a number of farfield microphone measurements. For accurately quantifying the overall noise of a compressor it is preferable to calculate the sound power by making a large number of sound pressure measurements ($\geq$ 20) on a hemispherical surface enclosing the machine. Today it is often quicker and more convenient to measure the acoustic intensity over all faces of a control volume containing the machine (Reference 6).

Time/frequency domain measurements. In order to reduce the overall noise of a product and to improve the quality of the remaining noise it is necessary to identify the individual noise sources. Frequency domain techniques have been much publicised and used routinely for some time to study noise emission from rotating and reciprocating equipment. The recent availability of high speed digital acquisition and analysis equipment has resulted in significant advances in the understanding of impulsive noise sources by analysing noise, vibration and the positions of components simultaneously in the time domain.

Reciprocating piston, rotating vane and rolling piston compressors all produce impulsive sound pressure time histories due to the opening and closing of discharge and, in some cases, suction valves. Examination of the time histories over the operational cycle of the compressor will reveal a series of individual events indicating that compressor noise is caused by excitation from various sources. The noise levels will usually be highest whilst the discharge valve is open since the compressed gas is abruptly discharged into a port and various passages. However, in addition to the noise due to gas pulsations, mechanical impacts occur due to impacts of the valves with the stop and seat. Other mechanical impacts such as piston slap, which is a common noise source in reciprocating engines have not received much attention although rotor chatter is known to occur in screw compressors. The extent to which these impacts are translated into radiated noise depends upon the vibration and radiation characteristics of the structures. Much work has been done in reducing the transmission and radiation of compressor noise, particularly with hermetically sealed compressors where the isolation and structural dynamics of the shell have a leading effect upon the radiated noise.

The work conducted in our laboratories to identify noise sources has involved acquiring up to 32 channels of data simultaneously with the digitisation 'locked' to the machine's rotational cycle using a phase-locked-loop device. These data typically consist of:

- the rotational speed
- a position signal within the rotational cycle
- nearfield and farfield sound pressure
- nearfield acoustic intensity
- vibration of the casing

- vibration of internal components
- position of internal components
- gas pressure at different positions.

Correlation of the motion of the individual components with their vibration, the gas pressures, the casing vibration and the radiated noise yields a powerful technique in the identification of the individual noise sources. An example of correlation study results is shown in Figure 2 in terms of the cumulative energy integral over the operating cycle of a machine. This integral indicates where in the cycle the contributions to the energy originate, as well as the total integrated value at the end. Figure 2 correlates the vibration of a casing with gear impacts deduced from gear speed measurements, and identifies cause-and-effect relationships.

Noise quality. The contribution of each impact to the overall noise level can be readily assessed by calculating the cumulative energy integral. The effect of each individual impact upon noise quality can also be assessed by digitally editing the signal in the time domain to remove an individual impulse and replaying the time history real-time through a digital to analogue converter to headphones or a purpose-built listening room. In this way the contribution of individual noise sources can be ranked both in terms of subjective and objective level without firstly having to identify the components or phenomena causing the noise and secondly obtaining a means of reducing or eliminating it. These techniques speed up development and reduce costs by concentrating development efforts on the noise sources most worthy of attention. Figure 3 shows the cumulative energy integrals before and after a design modification to the machine. Each abrupt rise seen in the curves corresponds to an impact. It is primarily the large steps (large impact) that makes the noise objectionable. The figure shows that a major impact source has been removed. In this case also the overall noise level (the integral over 360°) was reduced.

For most products in today's market the "quality" of the noise is almost as important as the level of the noise and so quantification of subjective effects becomes necessary. However, for reliable results to be obtained, large, well-designed, statistical panel tests are required which are both expensive and time consuming. The derivation of objective indices which correlate with the results of subjective tests provide the engineer with a powerful evaluation tool which can be used both for the development of new products and for making reliable comparisons with similar products from other manufacturers. For example, for evaluation of gear train sound, a "rattle index" was developed which correlated well with subjective impressions. An example of the application of this index is given in Figure 4, which shows the comparison of A-weighted sound pressure levels and of the rattle index for 10 test builds of a gear train. The figure demonstrates the usefulness of the index and the fact that A-weighted sound pressure level is not a sufficient descriptor for some noise quality problems.

Time domain analysis is particularly powerful for identifying the source of impact noises since their frequency content is relatively broad band. Nevertheless both time and frequency domain analysis should be used together to quantify and identify noise sources. Frequency domain techniques are particularly valuable for order related analysis or for identifying dominant spectral peaks caused by structural resonances.

Modeling of Compressor Flow and Acoustics

The compressors flow and pressure dynamics in its piping can be calculated using a cycle simulation, and its output used for prediction of intake and exhaust noise. An example of such a simulation is the code WAVE, described in References 7 and 8. This code uses one-dimensional computational fluid dynamics to solve for the flow and pressure dynamics in the manifold systems. The simulation can represent pressure wave dynamics of arbitrary amplitude -- i.e., it is not limited to acoustic linear waves. Another important feature is its detailed thermodynamics representing the behavior of gases of various composition. As shown in Reference 7, such a simulation can be used to predict the dynamics of the suction and discharge valves. Also, the effects of manifold geometry (pipe lengths, diameters and volumes) determine the amplitude and shape of the pressure pulsations in the system, and these effects can be dependably simulated by the full non-linear representation.

Although the solution of the manifold pressure dynamic is carried out using fluid-dynamic model (i.e., not specifically acoustic) the method is capable of dealing accurately with small amplitude waves as well. This can be seen in Figures 5a and 5b comparing the model predictions to data of Reference 9. The agreement in the predicted transmission loss by the present model obtained with speaker excitation (i.e., acoustic waves) with data and with the acoustic linear model is seen to be very good. It should be stressed that although the comparisons made in Figure 5 are with data taken in the linear acoustic range, the model is fully capable of handling high amplitude pressure waves and even shocks (Reference 8). Thus the model can serve as a design tool extending over a range of applications from performance, to valve impact predictions, to the acoustics of inlet and discharge systems.

CONCLUSIONS

Compressor noise represents a design issue of growing importance due to the increasing sophistication of the marketplace and pressure of legislation.

Advanced analysis methods are increasingly being applied to the solution of the noise problems. In many instances, the highly developed techniques used in the engine industry can equally be applied to compressor noise problem.

Among the methods in use for identification of noise sources and noise transmission paths are finite element analysis of compressor structure, experimental modal analysis and acoustic intensity measurements. From the point of view of pressure pulsation in inlet and discharge piping, fluid mechanics analysis of flow can be used to minimize excessive valve impact and limit undesirable pressure pulsations.

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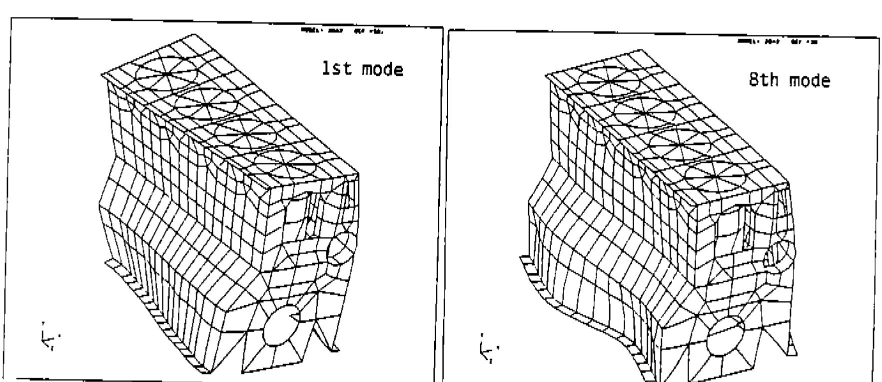

Figure 1. Dynamic finite element model of a crankcase structure showing its deformations in the 1st mode (525 Hz) and in the 8th mode (1120 Hz).

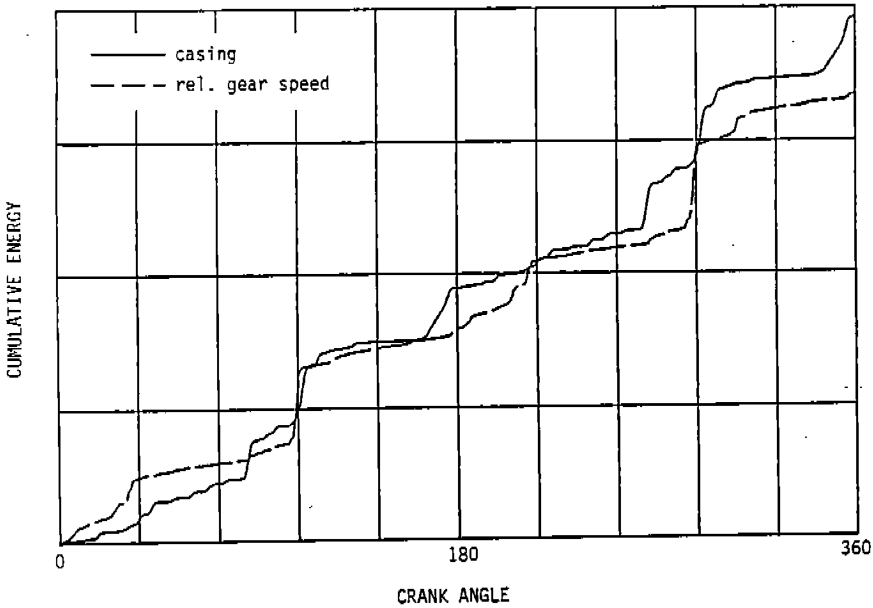

Figure 2. Average cumulative energy integrals of relative motion between gears and casing vibration, showing two vibration events correlating with gear impacts.

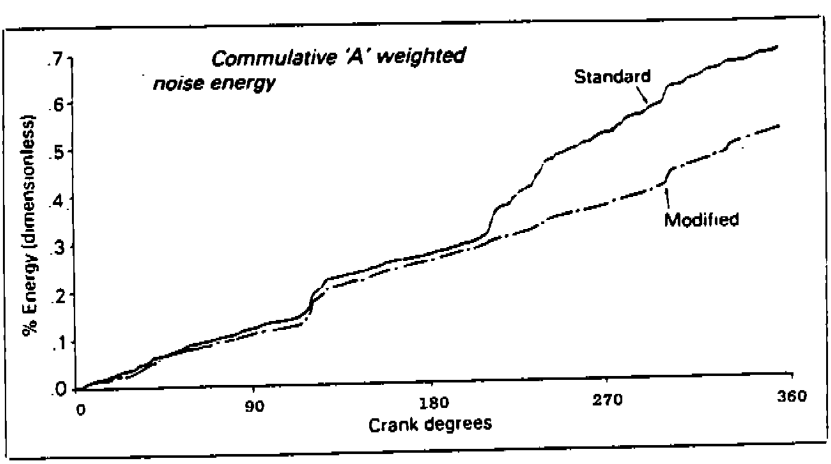

Figure 3. Comparison of cumulative energy integrals for a machine before and after component modification.

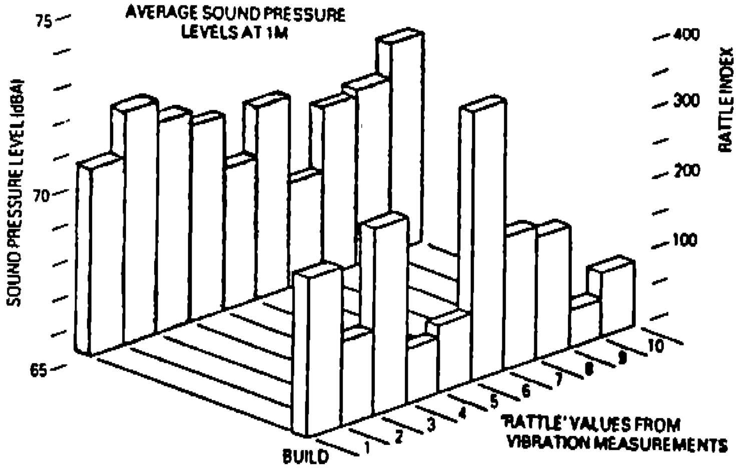

Figure 4. Comparison of an A-weighted sound pressure level to an objective index of gear rattle for 10 builds of a gear train.

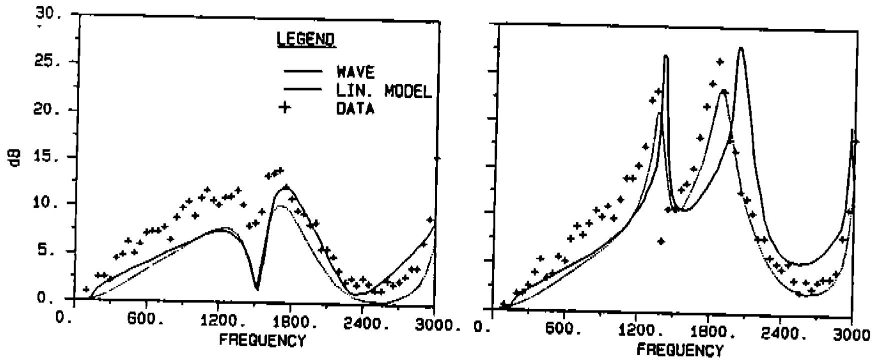

Figure 5. Transmission loss (sound attenuation) produced by concentric tube resonators with a perforated central tube element. ____ WAVE prediction; ..... linear model, + data (the latter two from Ref. 9). Both resonators had central tube diameter of 51 mm, outer shell diameter 76 mm and shell length of 127 mm. Part of the central tube (60%) was perforated with porosity of 3.7%, and the two silencers differed by the location of the perforated section.