DEVELOPMENT OF A CALCULATION PROCEDURE INCLUDING FLUID STRUCTURE COUPLING TO ASSESS CAVITATION EFFECTS

Dr.-Ing. Gotthard Ph. Rainer, Head of "Structural Analysis",
D.I. Ernst Gschweitl, Manager "Structural Analysis/Applied Calculation",
D.I. Hans Christoph Raffel, Senior Engineer "Structural Analysis/Applied Calculation"

AVL - Company for Internal Combustion Engines and Instrumentation Ltd., Graz,Austria

ABSTRACT

Between cylinder liner and parent bore of internal combustion engines cooling water flow is existing.

Due to the operating conditions cavitation effected by

- dynamic acoustic excitations and
- fluid flow

may occur.

AVL is investigating both effects using the MSC/NASTRAN acoustic element to consider the dynamic acoustic effected cavitation and the AVL developed CFD-software FIRE for the fluid flow effects.

This paper considers only the dynamic acoustic effect. Here the calculational and experimental work is explained, which has been done till now to apply MSC/NASTRAN to cavitation calculation. In addition to this, the difficulties to perform measurements and to compare the results out of calculation and of measurements are described.

The MAC (Modal Accuracy Criterion) is used as a tool to compare measurement and calculation. The calculations are carried out as eigenmode analysis and as forced vibrations.

1. INTRODUCTION

In the field of static, thermal and dynamic-acoustic calculations AVL has been using the Finite Element Method (FEM) for developing engines for many years /1,2,3,4/.Based on experimental work a procedure to assess cavitation was defined years ago /5/. As a part of this procedure structural eigenmodes were used.

With the MSC/NASTRAN Version 67 the consideration of dynamic effects in the cooling water became possible. Therefore AVL started a development work to make the fluid-structure coupling for design optimization available to avoid cavitation in engines.

Cavitation is induced by a loss of pressure so that vapour bubbles arise. If these vapour bubbles collapse near the material surface, the occuring high pressure will destroy the material. The reasons for cavitation are

- vibration effects and
- fluid flow velocities.

Vibration effects lead to local structural distortions which cause an increase or decrease of pressure in the cooling water. This effect is addressed in the submitted paper.

Fluid flow velocities are influenced by the geometric shape of the structure and the flow velocity. AVL is investigating these effects with the inhouse-developed CFD program FIRE /6/. An example for CFD calculations in an unit injector hole is shown in Fig. 1.

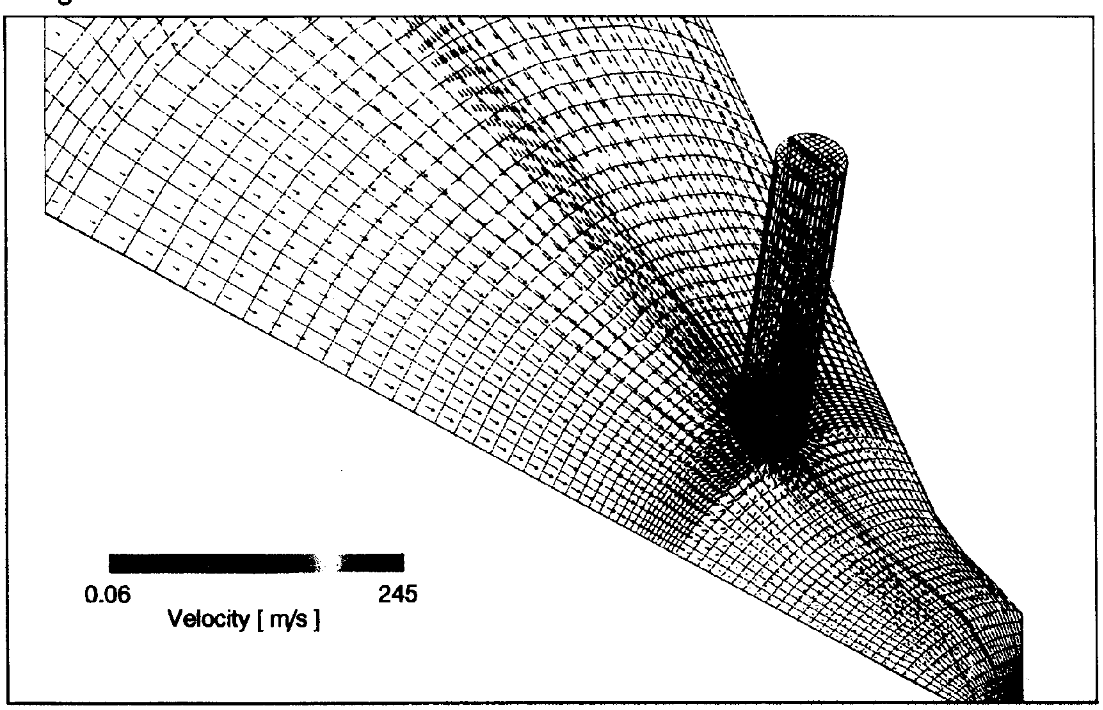


Fig. 1: CFD-calculation of the pressure distribution in an unit injector

The following topics are describing MSC/NASTRAN's capabilities to consider vibration effects on the cavitation.

2. CALCULATION PROCEDURE

The following chapter describes the calculation procedure shown in Fig. 2.

Using MSC/NASTRAN as a "Kernel System" AVL developed software to check the results and to evaluate the acoustic behaviour. Additionally to basic calculations experimental work is also necessary to check the numerical results (eigenmodes). Experimental and calculational eigenmodes are compared by MAC (**M**odal **A**ssurance **C**riterion). The acoustic behaviour is evaluated by special software modules.

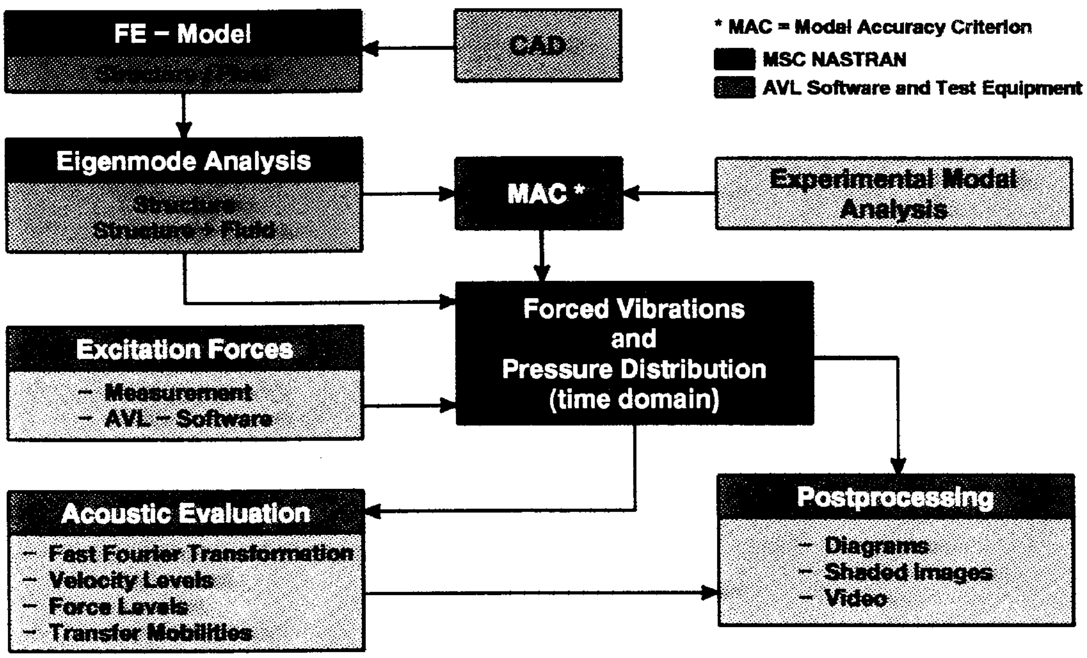


Fig. 2: Procedure to calculate fluid-structure interaction.

2.1 MODELLING THE FLUID-STRUCTURE-INTERACTION

The presence of a fluid may significantly influence the dynamic behaviour of a structure and vice versa the distortions of the structure may change the loads transmitted to the fluid.

To calculate these problems with MSC/NASTRAN /7/ the "Pressure Method" is used. Analogous to the displacements (3 degrees of freedom at each node) calculated with the "Displacement Method" in structural analysis pressure values are computed at the fluid points (1 degree of freedom at each node). Velocities and accelerations at the fluid points are analogous to forces in structural analysis.

The fluid elements can assume the properties of irrotational and compressible fluids governed by means of the three-dimensional and compressible wave equation. However, these elements do not simulate flow as there is no mass transport included. The available fluid elements are in the shape of hexahedra, pentahedra and tetrahedra.

Material data are the bulk modulus, the mass density and the speed of sound of the fluid.

MSC/NASTRAN Version 67 offers two ways to define the fluid-structural interface. The type of interface may either be matching or non-matching meshes where the fluid and structure contact each other.

2.2 SOLUTION METHODS FOR THE EIGENVALUE PROBLEM

The general form of the eigenvalue problem is stated as

$$\left(K - \lambda M\right)u = 0$$

K denoting the stiffness matrix, **M** the mass matrix and **u** the displacement/pressure vector.

If there is a pure structural model, it is known that both M and K are real, symmetric matrices. Methods for real eigenvalue extraction can be used.
In case of a fluid-structural model, both matrices M and K are not symmetric due to appearing coupling terms. Therefore methods for complex eigenvalue analysis have to be used.

2.3 TRANSIENT RESPONSE ANALYSIS

Transient response analysis is a process depending on time. So the underlying motion is stated as:

$$M\ddot{u}(t) + D\dot{u}(t) + Ku(t) = P(t)$$

with the mass matrix **M**, the damping matrix **D**, the stiffness matrix **K**, the displacement/pressure vector **u(t)** at time instant **t**, and **P(t)** as vector of applied loads at time t.
$(\dot{u})$ and $(\ddot{u})$ denote the first and second derivatives of u with regard to time. Let me make some remarks concerning the equation above:

- The matrices **M**, **D**, and **K** do not depend on time, and hence they are constant.
- Structural damping is included in the damping matrix **D**.
- The excitation **P(t)** is explicitely defined in the time domain. All of the applied forces are known at each instant in time.

The differential equation of motion is solved for the unknowns u(t) to get the vibrations of the model in the time domain.

MSC/NASTRAN provides the user with a direct (SOL 109) and a modal (SOL 112) transient response analysis.
In the direct method the numerical integration within MSC/NASTRAN is done by a Newmark-Beta type method similar to the classical one.
The results of transient response analysis are the displacement/pressure vectors at each time step and the corresponding velocity and acceleration vectors.

The direct transient response method is recommended to be used only in case of small models, few time steps or high frequency excitations.

The modal transient response analysis is much more efficient. The coordinates **u** are transformed to modal ones, using some of the modes found by real eigenvalue analysis. If, in addition, modal damping is used, i.e., each mode is related to a damping coefficient, the equations of motion become uncoupled, and can be solved analytically for pure structural models. However, in case of coupled fluid-structure systems, the equations of motion do not decouple totally, which is due to interaction terms. The problem is solved by using modal coordinates utilizing the direct transient response Newmark-Beta type numerical integration.

To generate time dependent transient loads, AVL uses inhouse-developed programs which are calculating loads for various engine excitations. These loads are transfered via an interface-software to MSC/NASTRAN.

2.4 EVALUATION OF THE ACOUSTIC TRANSMISSION BEHAVIOUR

Exciting a structure to vibrations causes the "structure borne noise" within the structure, Fig 3. These vibrations are transmitted to the human ear by the surrounding medium (e.g. air). Therefore, it is necessary to determine the transmission behaviour in order to reduce noise emission by changing the construction of the structure. In connection with internal combustion engines, the range of 1000 - 2000 Hz is of most interest, because of the fact that the excitation of such engines drops off rapidly above 2000 Hz.

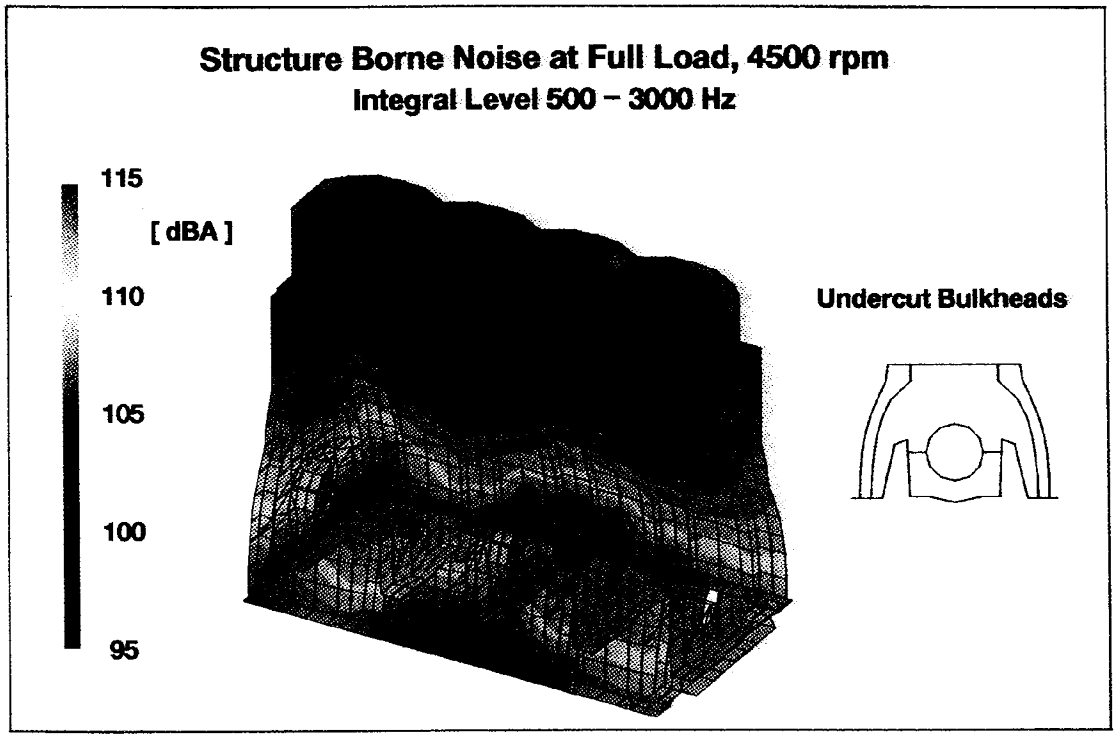
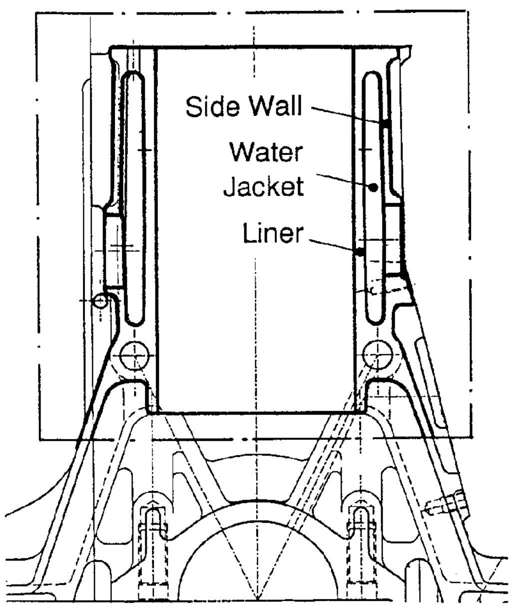


Fig. 3 Structure borne noise on an engine surface.

The transient response analysis computes diplacements, velocities, and accelerations for each grid point at each time. Especially the velocities are used to assess the transmission behaviour. Therefore, the velocities and the acting forces are converted from the time domain into the frequency domain by means of the Fast Fourier-Transformation (FFT). Then acoustic values like transfer mobility, velocity levels and integral (velocity) levels are calculated.

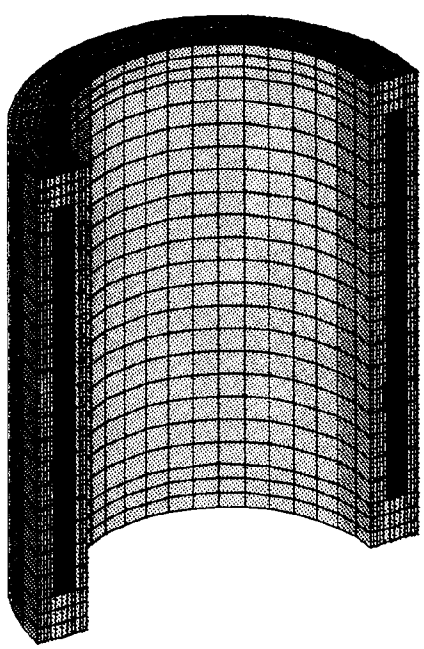
3. APPLICATION OF THE CALCULATION FOR A CYLINDER LINER



The calculation of fluid-structure interaction is carried out for a cylinder liner of an internal combustion engine with waterjacket and cylinder sidewall, Fig. 4.

Fig. 4: Cross section of an engine block with cylinder liner and cooling water jacket

3.1 FE-MODEL



In order to save DOFs, the finite element model of the cylinder liner is meshed as a half model, Fig. 5.

Fig. 5: FE-Model of cylinder liner (gray structure) and cooling water (blue fluid)

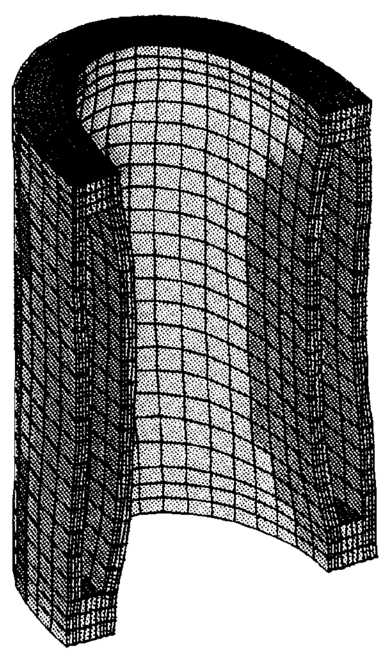
The structural part of the liner was modelled by means of CHEXA volume elements with 8 edges. At each grid point, three translational degrees of freedom exist. Altogether, the structural part of the model consists of 4160 elements, 5544 grids and 16632 degrees of freedom.

The hollow space (water-jacket) is filled with fluid elements of the type CHEXA of the same size as the structural elements. So, each structural grid point on the wetted area has a corresponding fluid grid point.
Altogether, the fluid part of the model consists of 1360 elements and 1890 grids. The complete coupled fluid-structural model has 18522 DOFs.

As we consider only half a model of the cylinder liner, boundary conditions are used to simulate the behaviour of the whole liner.

3.2 EIGENVALUE ANALYSIS OF THE CYLINDER LINER

3.2.1 Cylinder liner without fluid



First of all, an eigenfrequency analysis of the cylinder liner without fluid (SOL 3) was carried out.

Except for rigid body motions the first eigenfrequency appears at 2036 Hz, Fig. 6.

It is remarkable that distortions of the outer pipe already occur at lower frequencies than deformations of the inner one.
This phenomena can be explained by taking into account that, first, the outer pipe is not as thick as the inner one, and seond, a pipe looses its stiffness as its diameter increases.

The results were checked by measurements

Fig. 6: First eigenmode of the cylinder liner without fluid.

Fig. 7 shows a comparison between the first six calculated and nine measured eigenmodes using AVL's software for the evaluation of the MAC (= Modal Accuracy Criterion). The values coincide quite well. The mode shapes obtained by measurements could also be related to the calculated ones without any problems (except the fifth what is due to unavoidable inaccuracies in measurement).

Calculated Eigenmodes: Mode / Hz	Measured Eigenmodes: 1 / 1943	2 / 2686	3 / 3237	4 / 3269	5 / 4299	6 / 4335	7 / 4398	8 / 4642	9 / 4717
1 / 2036	0.98	0	0.18	0	0	0.02	0	0.15	0.17
2 / 2861	0.02	0.97	0.01	0	0	0	0	0	0.01
3 / 3491	0	0	0.96	0.95	0.05	0.05	0.12	0	0
4 / 4542	0.01	0	0.18	0.01	0.61	0.12	0.44	0.17	0.20
5 / 4823	0.01	0	0.22	0.02	0.06	0.41	0.12	0.01	0.01
6 / 4825	0	0	0	0	0.07	0.04	0	0.81	0.03

Fig. 7: Comparison of calculated and measured results using MAC

Additionally Fig. 8 shows a comparison between measurement and calculation for eigenmode No. 2.

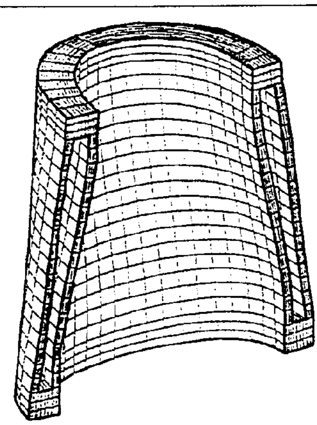
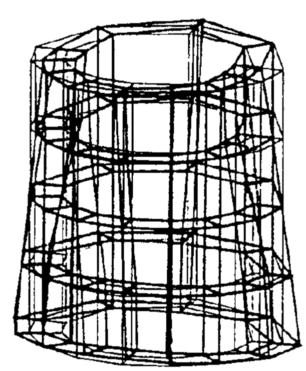
Calculation	Measurement
Mode 2, 2861 Hz, MAC = 0.97	Mode 2, 2686 Hz
	

Fig. 8: Comparison of calculated and measured results for eigenmode No. 2.

3.2.2 CYLINDER LINER WITH FLUID

The modal complex eigenvalue solution (SOL 110) was used to calculate the eigenfrequencies and modes of the cylinder liner with fluid.

The fluid causes an increase of mass of about 7.5% (from 2.51 kg to 2.71 kg). This indicates that the coupled fluid-structure eigenfrequencies decrease in comparison to the structural eigenfrequencies. In fact, the first eigenfrequency already occurs at 1820 Hz,Fig. 9.

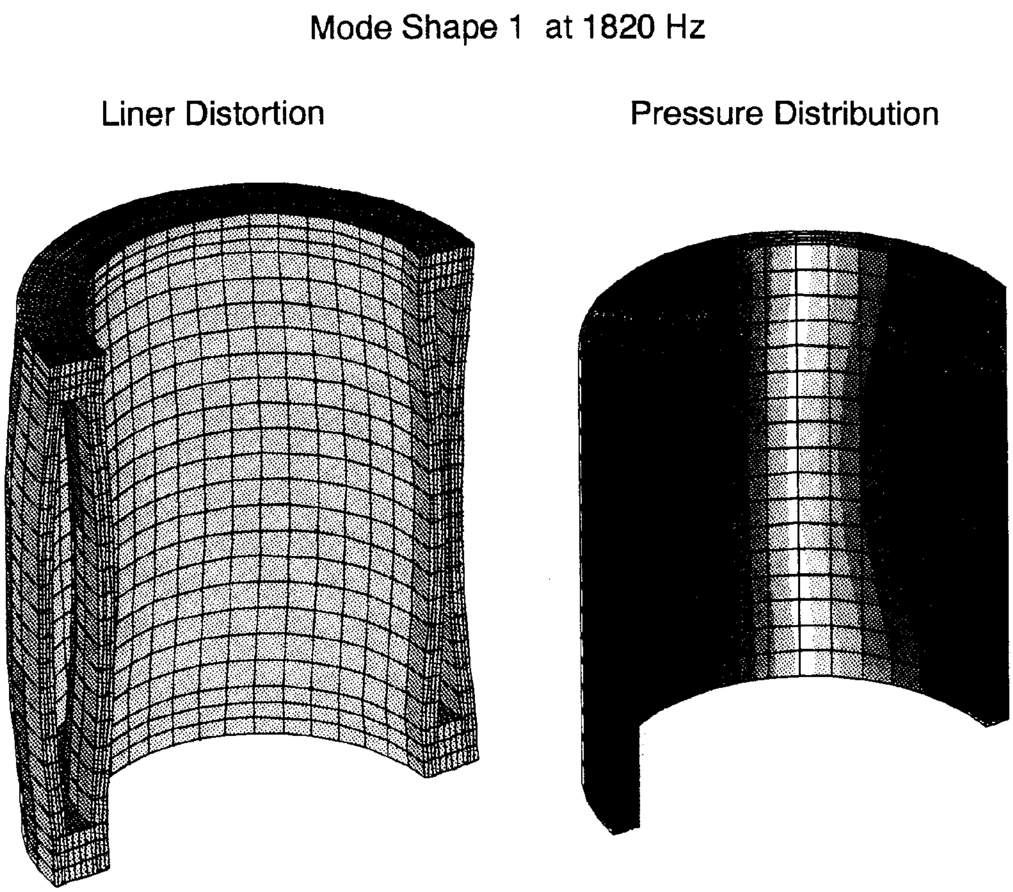


Fig. 9: Cylinder liner with fluid - 1st mode shape with the corresponding pressure distribution.

The presence of the fluid essentially influences the stiffness of the model, therefore the first mode shape is locally deformed (both pipes) to ellipses. This is also true for most of the other mode shapes.

The couple modes structure and fluid cannot be related to the structural ones easily. Generally, the number of modes found in the frequency range up to 5000 Hz nearly doubles (from 6 to 11), and the modes exhibit local deformations of higher order than the ones of the liner without fluid.

3.3 TRANSIENT RESPONSE ANALYSIS OF THE CYLINDER LINER

To perform the transient response analysis, the structure is supported by springs, which do not essentially influence the eigenfrequencies and modes of the model. The cylinder liner is excited at the top deck in the direction of the positive x-axis in order to simulate a piston impact. The load is applied at a grid point which lies on the surface of the intersection plane.

Modal transient response analysis was carried out with the integration timestep $\Delta t = 0.000048825$ sec. The cut-off frequency for the modal transformation is 5000 Hz in the case of the model without fluid, whereas both 50 fluid and structural modes were used when solving the coupled problem.
Fig. 10 shows the distortion of the cylinder liner for the maximum force (5000 N).

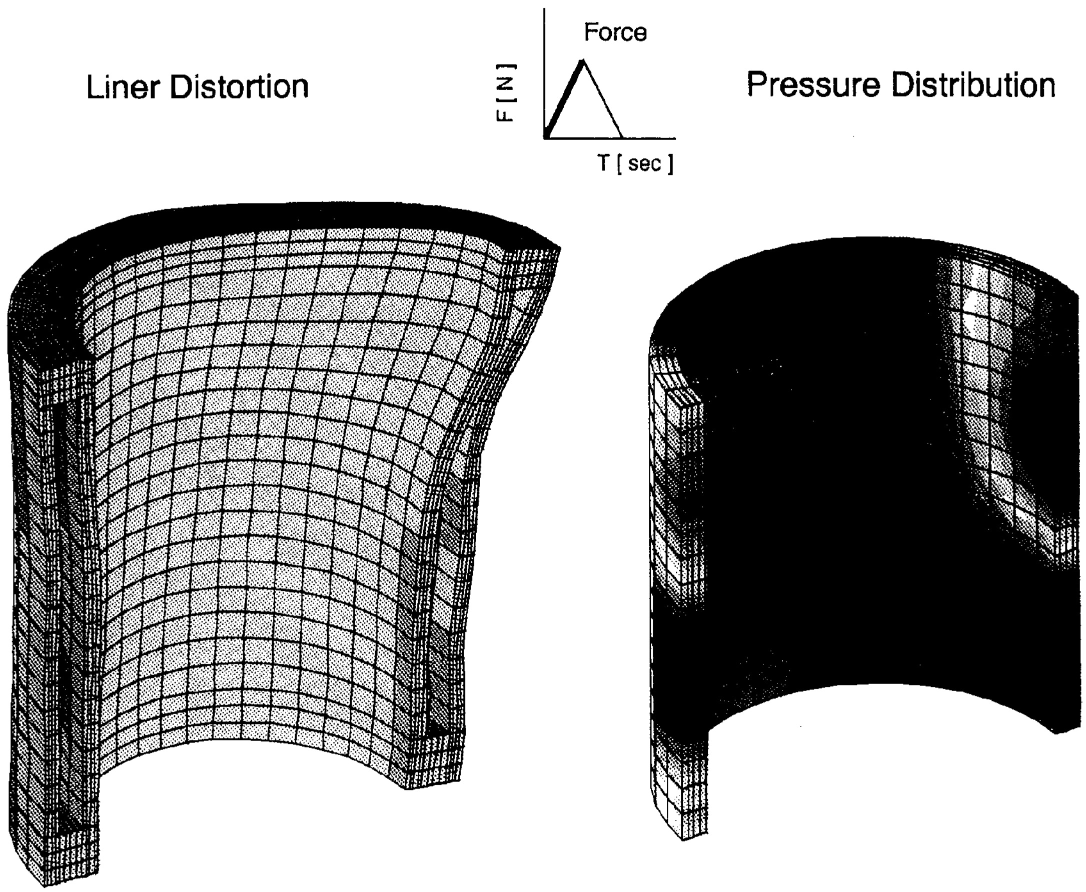


Fig. 10: Distortion of the structure and pressure in the fluid at maximum acting force.

Transient response analysis is carried out within MSC/NASTRAN, and then MODANS will evaluate the acoustic behaviour at predefined points on the outer cylinder wall, Fig. 11.

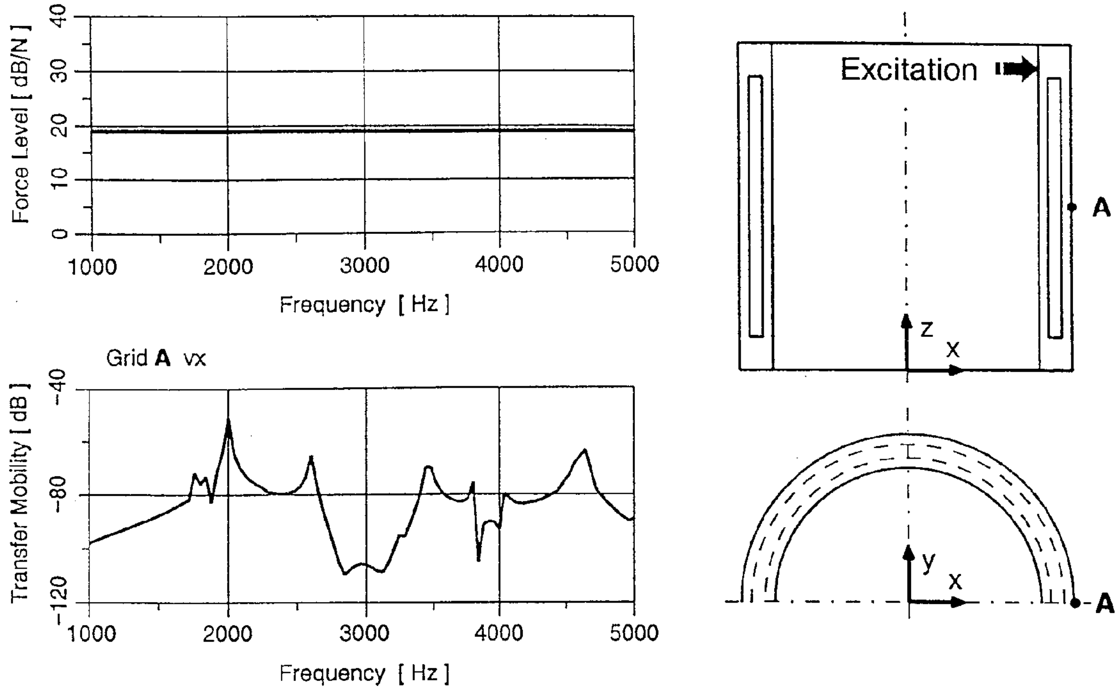


Fig. 11: Transfer mobility to the outer surface of the sidewall.

Additionally, calculations were carried out for the cylinder liner without fluid. The comparison showed that compared to the liner without fluid the cylinder liner with fluid may exhibit a worse acoustic transmission behaviour, since more resonance phenomena may occur in the relevant acoustic domain between 1000 and 2000 Hz.

3.4 CAVITATION ASSESSMENT

Based on the established calculation procedure, the assessment of cavitation effects can be done in the following way:

The "steady state" cooling water pressure created by the water pump is superposed to the dynamic pressure caused by the vibrations. The areas with pressure values near to zero or negative are endangered to cavitate. To avoid cavitation, geometric modifications have to be made usually.
In most of the cases it will not be possible to reduce excitation forces or the pressure created by the water pump. Therefore, as a next step parameter-studies for various geometric shapes of the cooling water jacket and of the surrounding structure have to be made.

4. COMPUTATIONAL ASPECTS

The finite element calculations were carried out on a IBM-RISC 6550 computer.

CPU-time demand and storage requirements increase with increasing complexity of the model, i.e., with the number of degrees of freedom (DOFs). Especially the calculation of the fluid-structural coupling terms required for the stiffness and mass matrices is a very time-consuming process.

As far as transient response analysis is concerned, the number of integration time steps is the determining factor for CPU time and storage demand. But it should be mentioned that output requests (i.e., the time history of vibration) may influence storage requirements to a large extend.

Cylinder liner without fluid: 16632 DOFs

- Eigenfrequency analysis using Lanczos-method (SOL 3):
  - CPU-time 22 min
  - Storage requirements 220 Mbyte

- Modal transient response analysis (SOL 112, 512 timesteps, output request for 36 DOFs):
  - CPU-time 90 min
  - Storage requirements 1260 Mbyte

Cylinder liner with fluid: 18522 DOFs

- Modal eigenfrequency analysis using Lanczos-method (SOL 110):
  - CPU-time 145 min
  - Storage requirements 350 Mbyte

- Modal transient response analysis (SOL 112, 512 timesteps, output request for 36 DOFs):
  - CPU-time 215 min
  - Storage requirements 1370 Mbyte

5. SUMMARY

The described calculation procedure to assess cavitation shows that a lot of points like mesh density, boundary conditions, CPU-time or disk-storage have to be considered. To perform calculations for large FE-structures and to evaluate the results for design optimization in an efficient way it has to be done additional work, e.g. the

- definition of optimization criteria for the geometry
- investigation of numerical accuracy (considered eigenmodes, etc.)
- minimization of hardware requirements like CPU or disk storage (data handling)

Provided that all these efforts are included, the described calculation procedure will also be applicable for other structural optimization work for pipes, tanks, etc.

The authors wish to thank Mr. J. Haslinger for his engagement in performing the calculations and checking the procedure.

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