N89-22902

INFLUENCE OF DIFFERENT TYPES OF SEALS ON THE STABILITY BEHAVIOR OF TURBOPUMPS

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One of the main problems in designing a centrifugal pump is to achieve a good efficiency while not neglecting the dynamic performance of the machine. The first aspect leads to the design of grooved seals in order to minimize the leakage flow. But the influence of these grooves to the dynamic behavior is not known very well. This paper presents experimental and theoretical results of the rotordynamic coefficients for different groove shapes and depths in seals. Finally the coefficients are applied to a simple pump model.

NOTATION

C	seal clearance
c_L	leakage coefficient = $[\dot{Q}/(2\pi R^2)][(\rho/2\Delta p)^{0.5}]$
D,d	direct and cross-coupled damping
d_{ij}	dynamic coefficients for damping
F_x,F_y	external forces
H	groove depth
K,k	direct and cross-coupled stiffness
k_{ij}	dynamic coefficients for stiffness
L	seal length
M,m	direct and cross-coupled inertia
m_{ij}	dynamic coefficients for inertia
Δp	pressure drop
R	shaft radius
$\dot{Q}$	volumetric flow rate
v	leakage velocity
x,y	coordinates for two orthogonal directions

α damping coefficient (real part of the eigenvalue)

ξ modal damping = $-\alpha/(\alpha^2 + \omega^2)^{-0.5}$

Ω rotational frequency

ω eigenfrequency (imaginary part of the eigenvalue)

ω_o eigenfrequency of the "dry" shaft

Ω rotational frequency

INTRODUCTION

Like all kinds of machines, centrifugal pumps must operate with a good efficiency. Because of the high rotational speeds, contactless seals have to be used to separate areas of different pressures in the pump. But the leakage flow through these seals reduces the efficiency of the machine (fig. 1).

As a result of this aspect, pump manufacturers very often use grooved seal surfaces to give more resistance to the fluid flow and so to reduce the leakage.

Besides the efficiency, also the dynamic behavior of the seals is an important aspect for today's turbopumps. The operational speed usually is much higher than the first critical speed of the "dry" shaft. But a resonance problem is avoided because during operation the seals introduce a great amount of stiffness to the system. Therefore they are also called wear rings. Not only stiffness, but also damping arises from the seals, and the physical mechanisms can cause instability for the centrifugal pump.

So the influence of seals to the dynamic behavior of turbopumps is very important and has to be taken into account (ref. 1). Especially the influence of grooved seal surfaces has to be investigated because these kind of seals are used for better effectiveness as mentioned before.

These influences are studied in this paper. For three different groove shapes with various depths (fig. 2) the dynamic coefficients of the seals are determined theoretically and experimentally. The results are discussed, and the influence to the rotordynamic behavior of a simple pump model is demonstrated.

THEORETICAL APPROACHES TO DETERMINE SEAL COEFFICIENTS

It is the aim of the theoretical approaches to describe the relations between external forces acting on a seal and the motions of the shaft by a mathematical expression. The results can be easily included in a finite element procedure for the overall behavior of a pump, if they are described in a matrix equation:

$$\begin{bmatrix} m_{11} & m_{12} \\ m_{21} & m_{22} \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} d_{11} & d_{12} \\ d_{21} & d_{22} \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} F_x \\ F_y \end{bmatrix} \qquad (1)$$

m_{ij}, d_{ij}, k_{ij} dynamic coefficients for inertia, damping and stiffness

x, y coordinates for two orthogonal directions

F_x, F_y external forces

In the following chapters two ways of calculating these dynamic seal coefficients are briefly described. Both theories come out with skewsymmetric matrices.

$M_{11} = m_{22} = M$ direct inertia

$m_{12} = -m_{21} = m$ cross-coupled inertia

$d_{11} = d_{22} = D$ direct damping

$d_{12} = -d_{21} = d$ cross-coupled damping

$k_{11} = k_{22} = K$ direct stiffness

$k_{12} = -k_{21} = k$ cross-coupled stiffness

Finite Length Theory

In 1982 CHILDS published a finite length theory (ref. 2,3) to calculate seal coefficients. It is based on a bulk-flow model (fig. 3a) for the fluid behavior in the seal. The force equilibrium equations together with continuity equations can be solved by a perturbation method. Integrating the resulting pressure distribution over the seal surface leads to the seal forces from which the dynamic coefficients can be extracted.

Contained in this calculation are empirical constants to include the friction at the rotor and stator surfaces, which have to be determined experimentally.

In regards to grooved seals it is possible to extend the model. The continuity equation in the circumferential direction is enlarged by the additional cross sections of the grooves, and the empirical friction factors are determined separately for the circumferential and axial directions. Also the friction of the circumferential fluid flow at the groove walls is taken into account.

This extended bulk-flow theory was developed by NORDMANN (ref. 4) and is applicable to grooved seals, but some empirical constants have to be measured in addition.

Finite Difference Theory

In 1986 DIETZEN and NORDMANN (ref. 5) published a method to calculate seal coefficients by means of a finite difference technique. The method is based on the Navier-Stokes Equations, continuity equation and energy equation, so that every fluid flow can be modelled.

To calculate the fluid flow in a seal, the seal gap is described with a grid of calculation points (fig. 3b). Using also a disturbance method the fluid velocities and pressure values are calculated for all nodal points as a function of the values of the surrounding grid points.

The wall stresses are modelled by a logarithmical law, which only influences the first grid line at the wall. So it is possible to calculate the fluid flow for all seal geometries without using any empirical values. Again the resulting pressure distribution is integrated yielding the forces and finally the desired seal coefficients.

A disadvantage for this finite difference theory is the high computational time it takes, but the results are very good compared to measurements as the next chapter will show.

MEASUREMENTS OF SEAL COEFFICIENTS

Test Rig

A test rig to measure rotordynamic seal coefficients was built at the University of Kaiserslautern by MASSMANN and NORDMANN (fig. 4), (ref. 6).

Between a stiff shaft which is rigidly supported and a stiff housing, two symmetrical seal inlets are situated. During operation the shaft rotates and water is pumped through the seals. Their dynamic behavior is measured by impacting the housing with a hammer and recording the input (force) and output signal (motion of the housing in relation to the shaft). A FFT-Analyser calculates the transfer function from which the dynamic coefficients are extracted.

Because of the large mass of the housing, which is moved during measurements, the inertia coefficients cannot be evaluated by this test rig.

Leakage

A first result for the seal geometries shown in fig. 2 is the leakage performance. Fig. 5 presents the dimensionless leakage coefficient c_L for the different seal geometries versus groove depth.

It is obvious that the smooth seal has the highest leakage. For increasing groove depths, the coefficient goes down in general. Only the rectangular groove shape shows an increasing leakage for larger depths. Overall, rectangular and sawshaped grooves show the best performance.

Dynamic Coefficients

The resulting seal coefficients for one operational point are given in fig. 6. They are compared with theoretical results coming from the two theories mentioned before. The inertia coefficients are not shown in this figure, because they could not be measured as mentioned before.

For all geometries the values for the direct stiffness term are influenced very much by the various groove depths; they decrease. Cross-coupled stiffnesses also decrease slightly, while cross-coupled damping stays nearly constant. Especially for the rectangular groove shapes direct damping also goes down with increasing depths.

The comparison with the calculated data shows very good agreement for the finite difference theory in all parameters. Finite length theory predicts higher direct stiffness terms, but shows good agreement for the other coefficients.

The results for other operational points are qualitatively similar.

Discussion

Using the finite difference technique it is possible to calculate accurate dynamic seal coefficients for all seal geometries, but it is very time consuming, and so it is also of interest to find out the case in which the finite length method works satisfactorily.

Direct damping is calculated pretty well by finite length theory, but this may arise from the fact that measured friction factors have to be included in the calculation.

Also finite length theory shows reasonable results for rectangular groove shapes, but is very wrong for other geometries especially for the very important direct stiffness terms. This can be explained regarding the fluid flow in the seals calculated by finite difference technique (fig. 7). One can see a recirculation inside rectangular grooves which does not affect the main flow very much. So the bulk flow model, with an extended circumferential continuity equation can be applied successfully. For other groove geometries the main flow direction is very much influenced by the grooves, so that the bulk flow model is not applicable.

INFLUENCE OF EIGENVALUES

For real machines not the single dynamic coefficients are important but the resulting overall behavior, for example critical speed, damping and stability behavior. These data are given by the eigenvalues of a centrifugal pump which consist of eigenfrequencies and damping values.

In order to demonstrate the effect of different seal geometries on the rotordynamics of turbopumps, a simple rotor model is chosen. Fig. 8 shows a Jeffcott rotor as a model for a double suction feed pump.

The coefficients for the different seal geometries are applied to this model. The results at operational speed are shown in fig. 9. A smooth seal increases the first eigenfrequency of the dry shaft by 96% and all grooved seals have less stiffening effect, getting smaller with increasing groove depths. The best groove geometry to introduce stiffness to the system is the sawshaped groove. Values for modal damping are improving for grooved seals, but a general characteristic cannot be observed. It has to be pointed out that modal damping is determined by both the real and imaginary part of the eigenvalue. The real parts of the eigenvalues decrease for grooved seals (fig. 10).

In fig. 10 eigenvalues and unbalance responses are shown versus running speed for rectangular groove shapes. For a constant pressure drop versus speed, which is not realistic for a pump, eigenfrequencies don't change very much, damping slightly decreases and the unbalance responses show no dramatic differences for various groove depths. The pressure drop in a real machine would be nearly proportional to running speed squared. In this case the eigenfrequencies and damping values increase with running speed. The unbalance responses for grooved seals are now much worse compared to plain seals.

Finally the calculated eigenvalues of a multistage boiler feed pump are shown using the data for grooved seals and plain seals respectively (fig. 11). With one exception all eigenfrequencies are increased using plain seals. The results for modal damping show no uniform characteristics.

So from the dynamic point of view it is hard to tell whether one should use plain or grooved seals. The special application has to be investigated.

CONCLUSIONS

The design of grooved seals for turbopumps reduces the leakage flow and so increases the efficiency.

The influence of these kinds of seals to the dynamic performance is investigated for different types of seals. Rotordynamic seal coefficients are evaluated experimentally and the results are compared to calculated ones coming from finite length theory and finite difference technique respectively. The latter gives the best results for all seal geometries, while finite length theory seems to be applicable for rectangular grooves only.

The influence to the eigenvalues of a simple pump model shows less stiffening effect of grooved seals compared to plain seals. Whether, from a dynamic point of view, grooved seals are better or worse compared to plain seals cannot be stated generally. An analysis of the special application seems to be necessary.

LITERATURE

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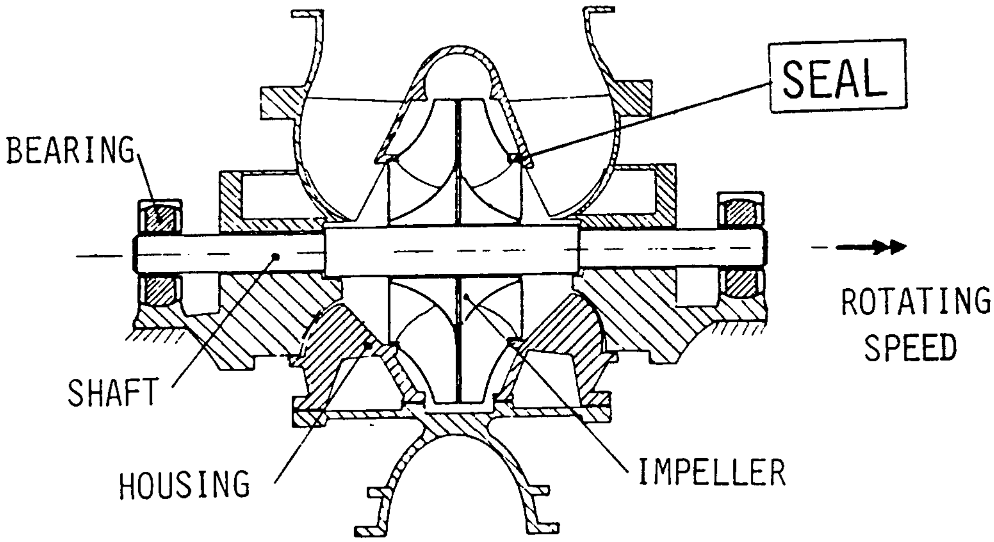

FIGURE 1: Cross section of a turbopump

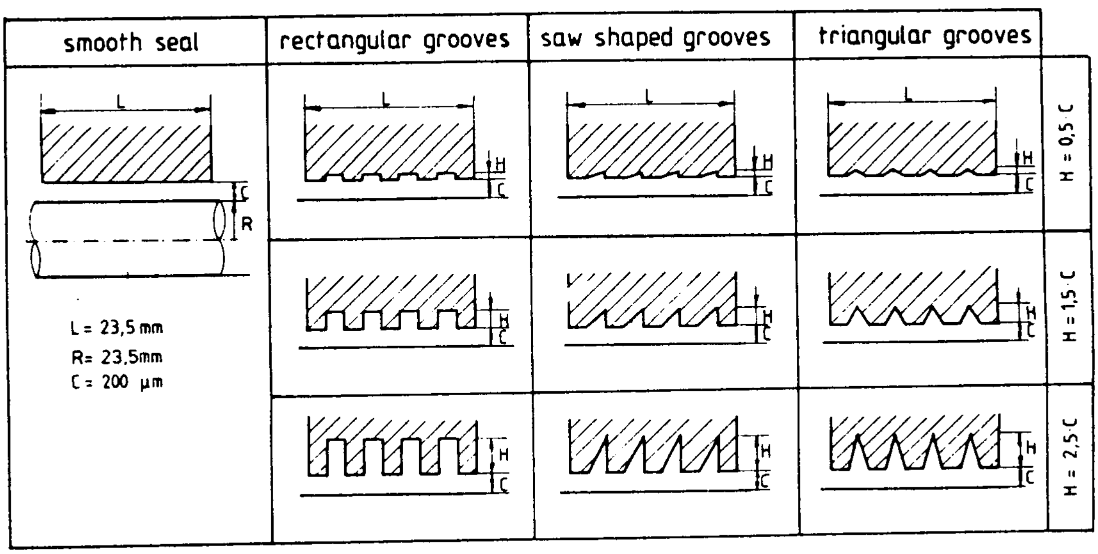

FIGURE 2: Seals with different groove shapes and depths

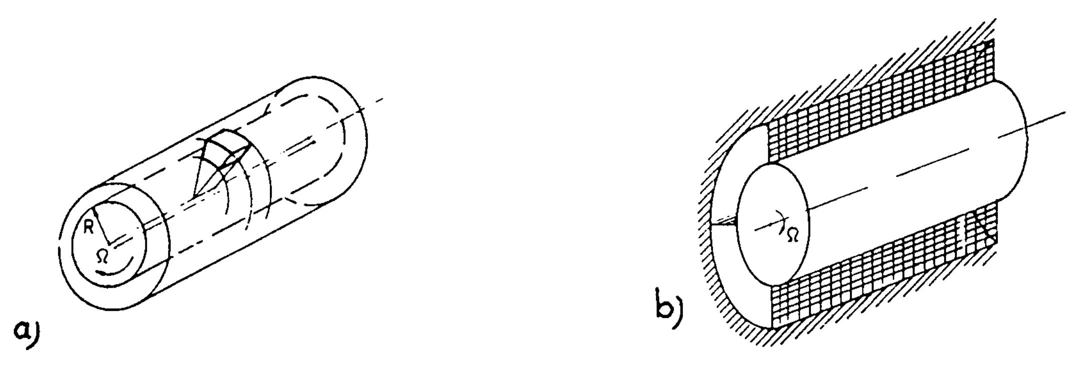

FIGURE 3: Models for a) finite length and b) finite difference theory

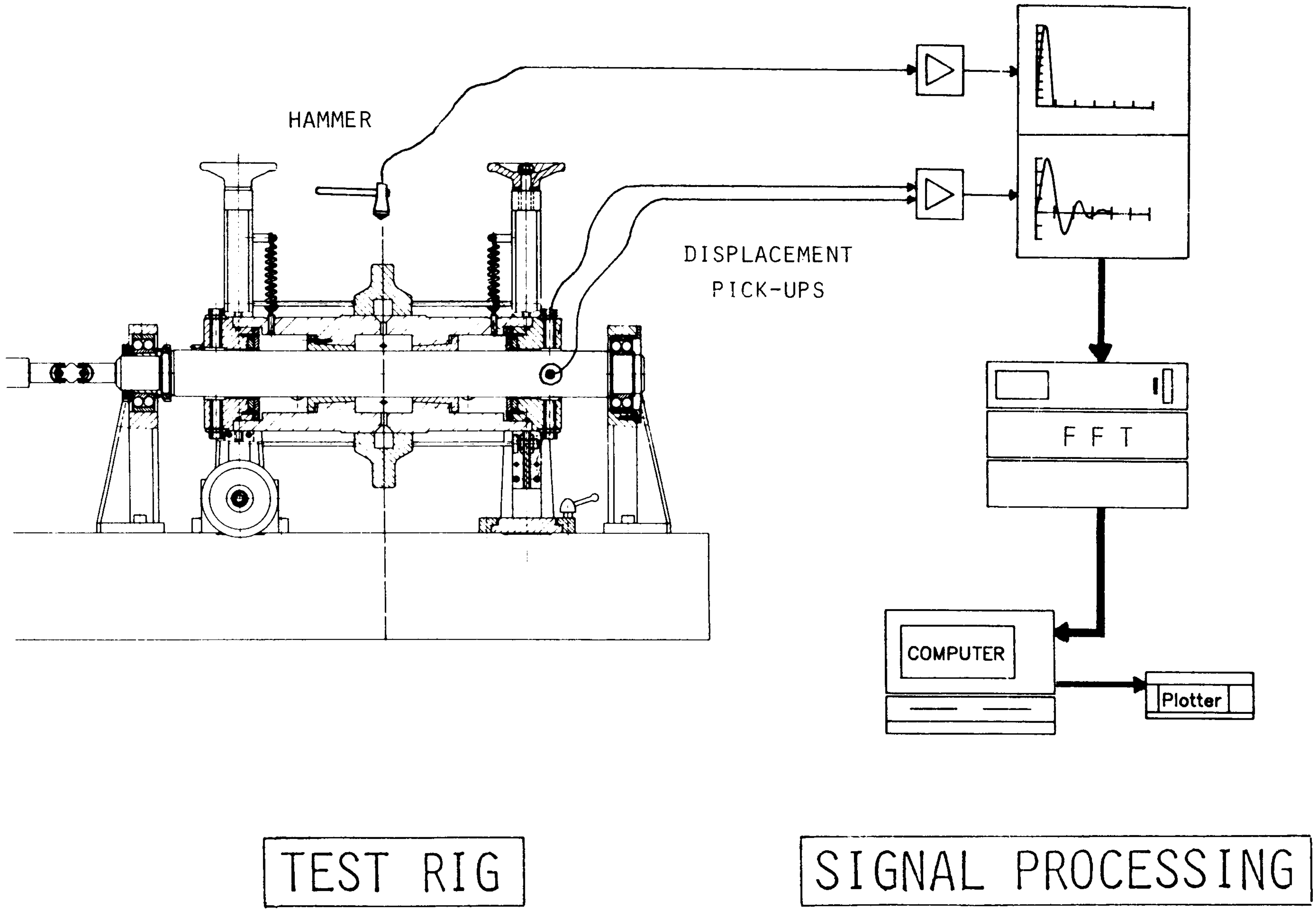

FIGURE 4: Test rig and signal processing

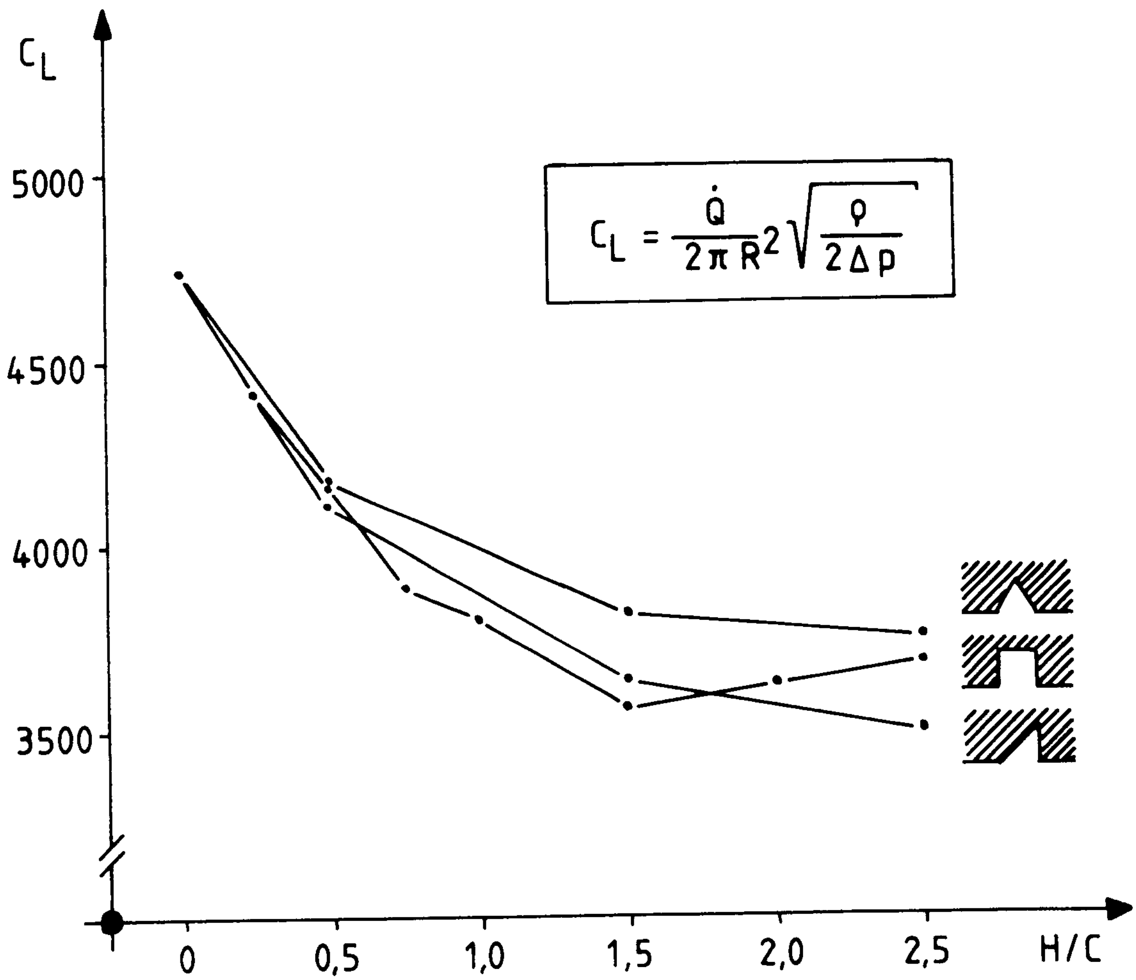

FIGURE 5: Leakage coefficients for different seal geometries versus groove depths

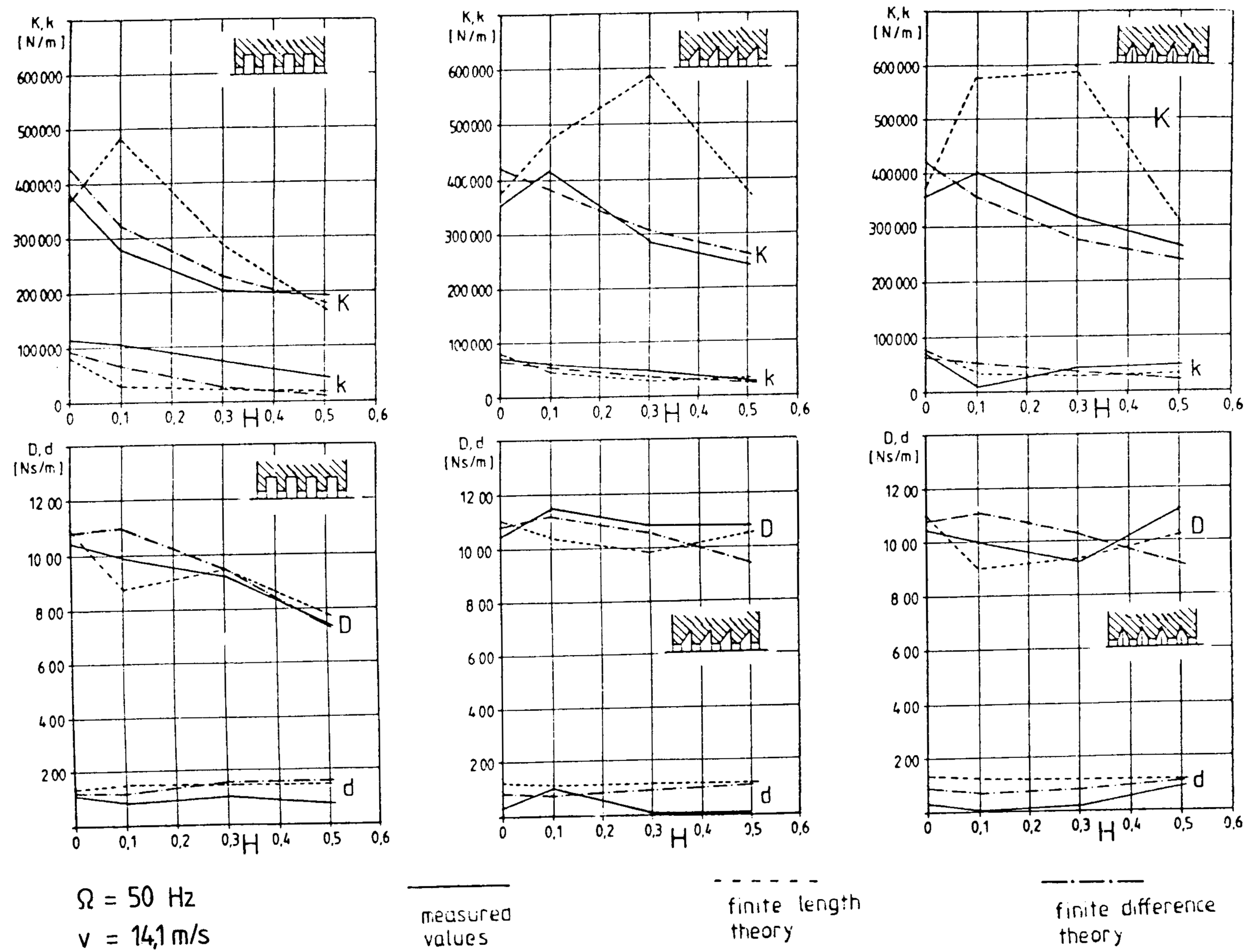

FIGURE 6: Dynamic stiffness (K,k) and damping (D,d) coefficients for different seal geometries and groove depths (H)

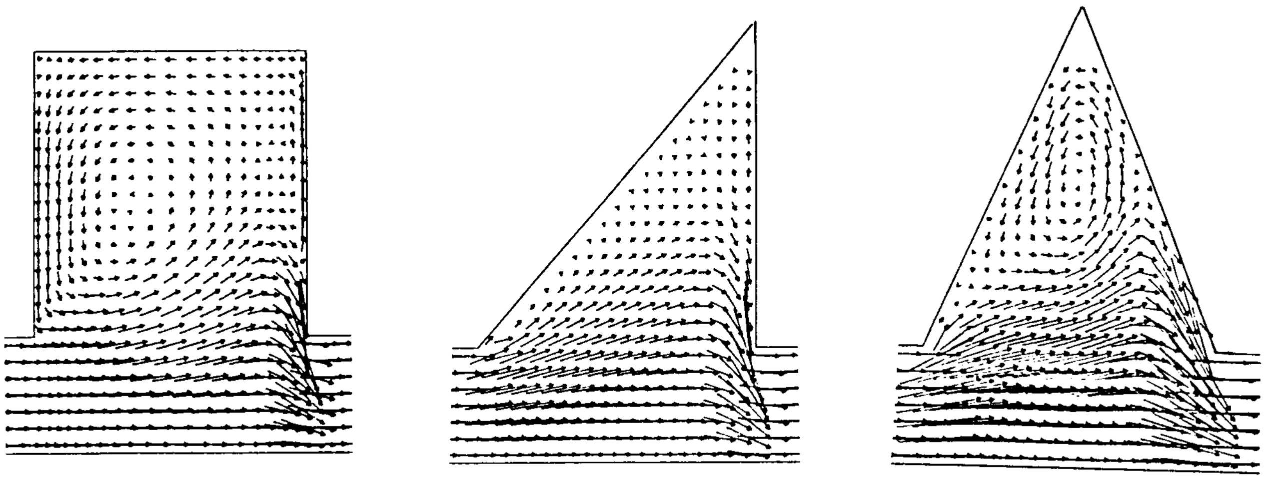

FIGURE 7: Flow fields for different groove shapes

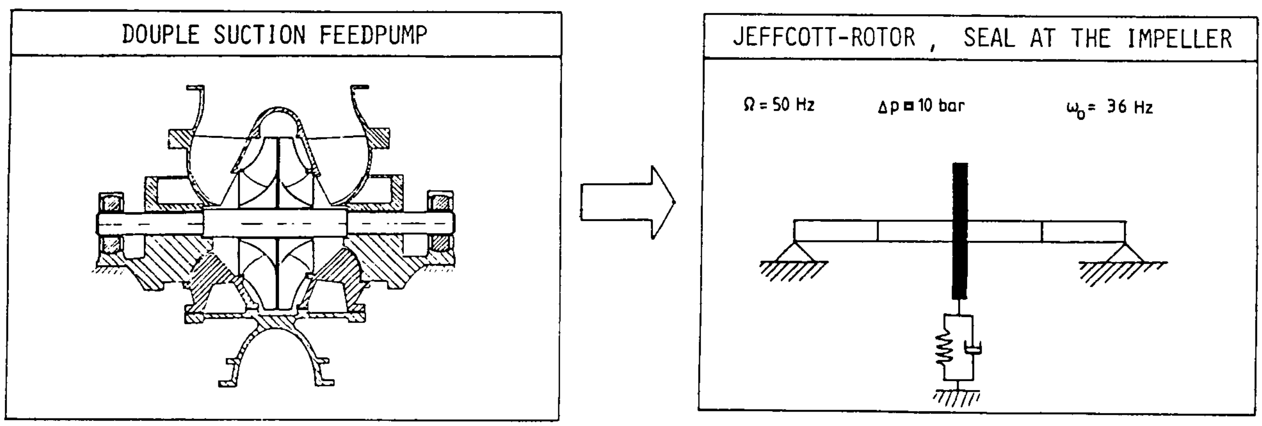

FIGURE 8: Simple model of a double suction feed pump

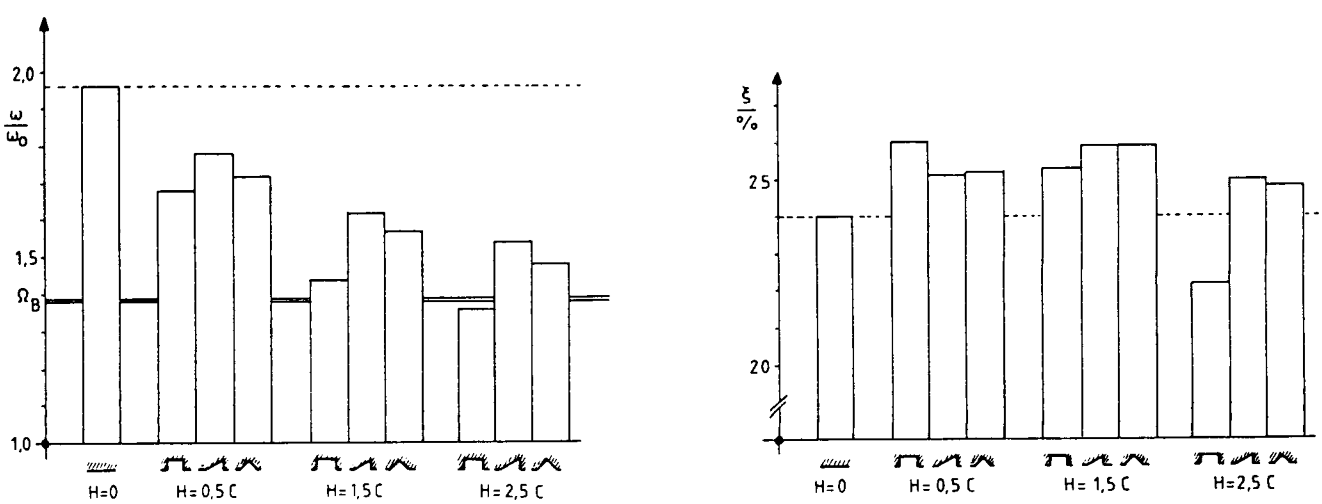

FIGURE 9: Eigenvalues of the pump model versus groove depths at operational speed for different seal geometries

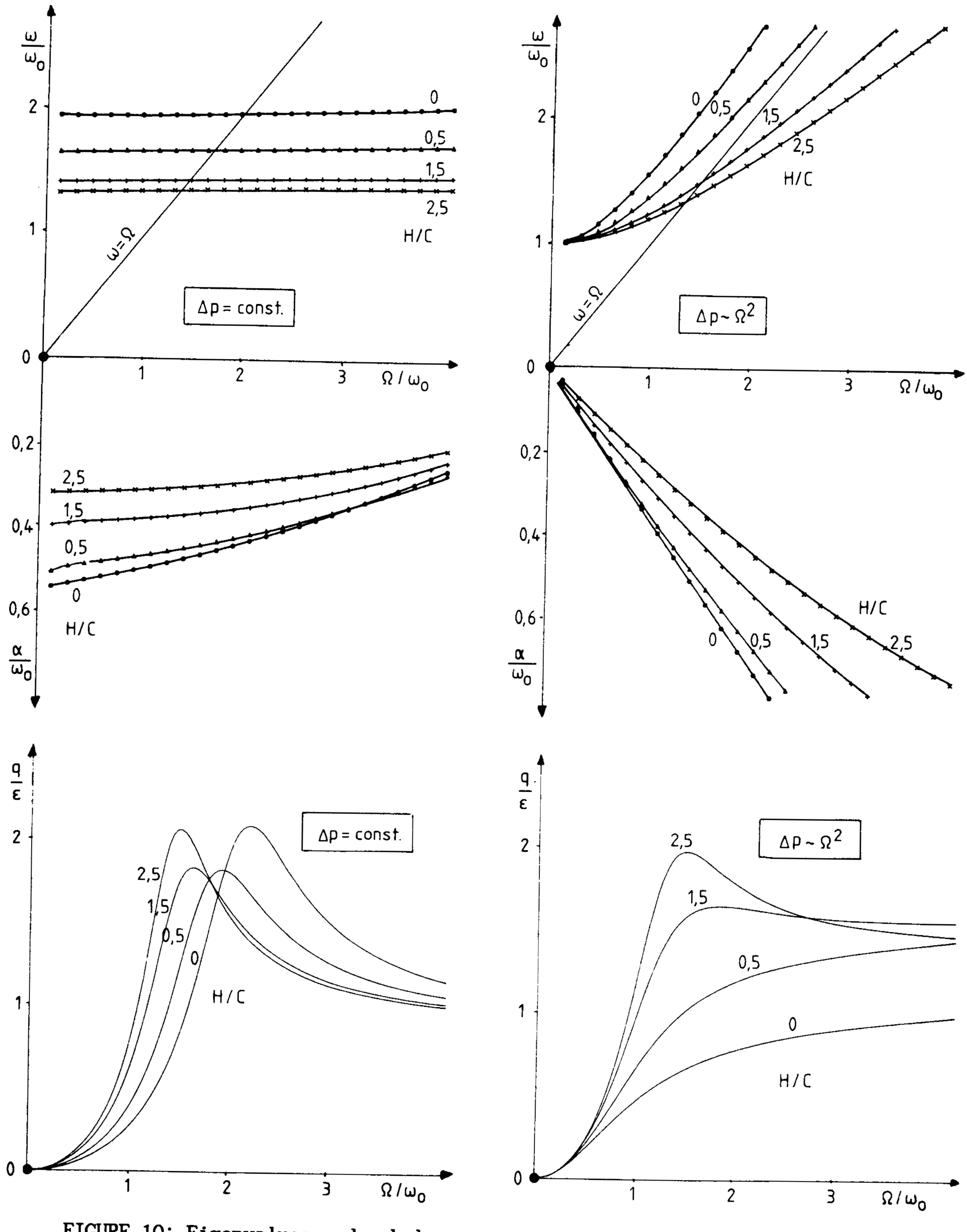

FIGURE 10: Eigenvalues and unbalance responses for rectangular groove shapes

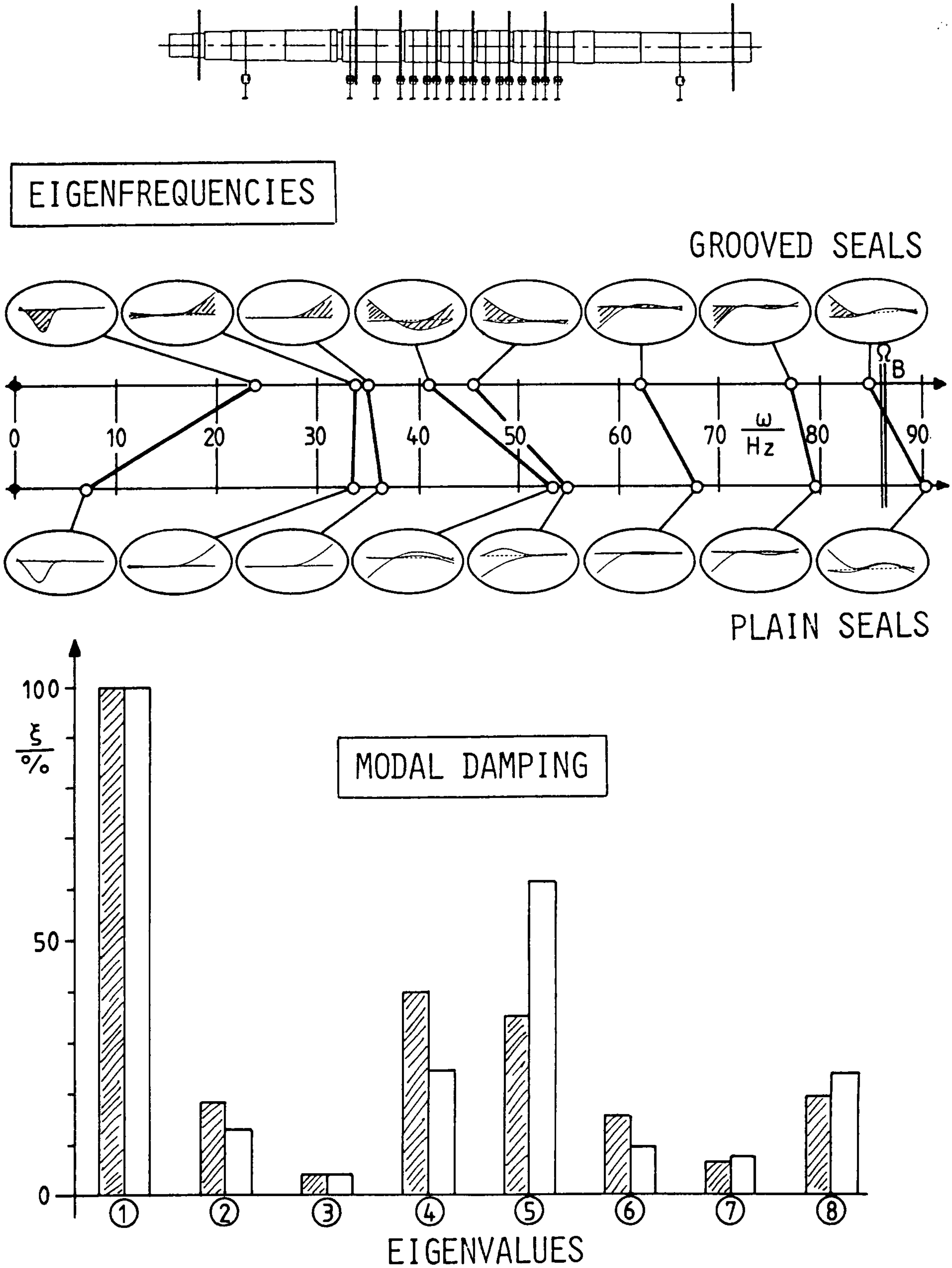

FIGURE 11: Eigenvalues of a multistage boiler feedpump