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Theoretical and Experimental Study on Valve Flutter, Part II: Flutter Experiments and Similarity

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THEORETICAL AND EXPERIMENTAL STUDY ON VALVE FLUTTER

Part II: FLUTTER EXPERIMENTS AND SIMILARITY

5.FLUTTER EXPERIMENTS

5.1 Flutter experiments with an enlarged model

Extensive experiments have been carried out with the set up shown in Fig.1 in [6]. Table 1 gives some relevant data for this model.

TABLE 1 Enlarged model, essential data

working pressur	$P_{m12} \approx 1$ bar	W_2 = 0 to 20 m/s
air density	$\rho \approx 1.2$ kg/m³	$A_p \doteq 0.2$ m²
isentropic exponent	k=1.4	LC_D = 0.96 m
V_1; V_2	∞ ; 17.9 m³	X = 0 to 0.2m
nat.frequency of spring mass system	f_o = 1 to 5 Hz	J = 4.88
		$C_o \doteq$ 0 to 4(for C_3=1)
nat.frequency of acoustic sytem	$f_{g,o}$ = 5.71 Hz	C_2 = 0.2 to 1.7(for C_3=1)

In absence of an appreciable squeezing effect(as occuring with reeds) the damping of the spring mass system was very low and practically negligible($C_4 \approx 0.006$).

Experiments with this set up mainly where done to find out and confirm the correct basic equations. This shall be discussed in brief.

Non steady work exchange effect

The stability diagram for a model which does not consider the non steady work exchange is sketched in Fig.10a. The stability condition simply becomes: $C_2>2$(valid for $C_4=0$). The model allowed values $C_2<2$ only and therefore flutter should occur immediately when starting with small values ΔP_{12}. In spite of this the model was stable and reached stability limit at a certain value ΔP_{12} resp. W_2 or C_o,Fig.10b. The theoretical explanation for this behaviour was found in the non steady work exchange effect(see [6]). If this effect is considered in the equations the stability limit curve takes a form as indicated in Fig.10b or Fig.5,6,7, and explains quantitatively the measured results. Fig 10c shows some results indicating the transition from stability(3.record) to flutter (records 1 and 2). The spring constant in this experiment was c=1300N/m. The variable considered in Fig.10c was the oscillating pressure difference Δp_{12}. Proportionate oscillations occur with the other variables x, w_2, f_{pl}.

Fig.11 shows results from a state point at the stability limit i.e. oscillations with constant amplitudes. Results could be processed to get a vector diagram which considers relative amplitudes($\bar{x}$=1) and phase shifts. Though the amplitudes of the oscillations are about 20% of the (mean) equilibrium values, the agreement with theoretically calculated values is acceptable. It should be noted that theory of flutter assumes small oscillations as compared to the mean values.

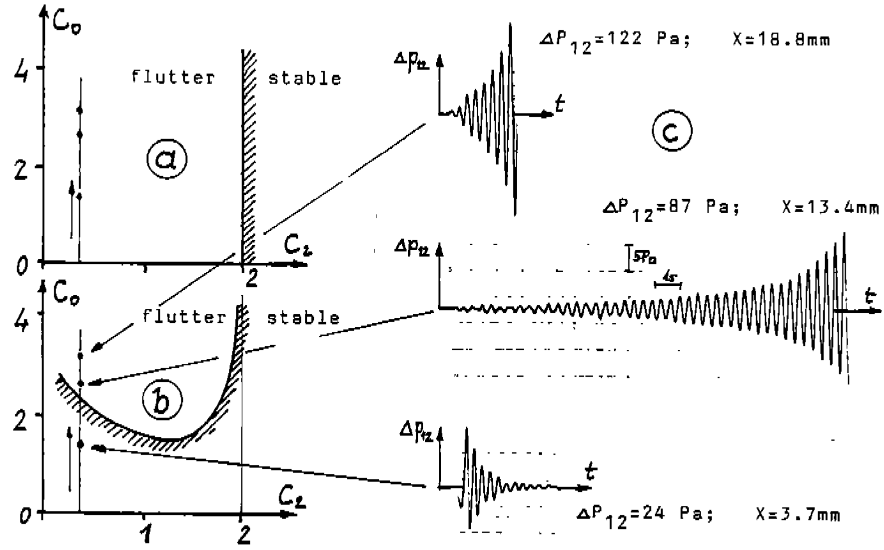
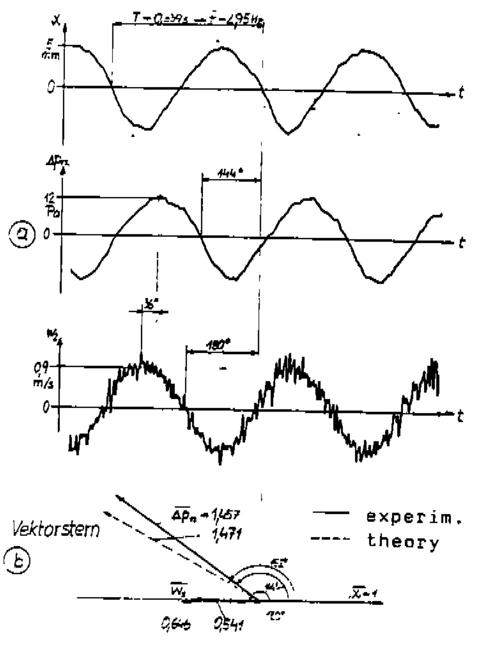


Fig.10 Experiment with respect to non steady work exchange effect. "a" stability diagram for a flutter model ignoring non steady work exchange. "b" model considering non steady work exchange. "c" recorded transients confirming the flutter model with non steady work exchange effect included.



$\Delta P_{12}=42$ Pa
$X=25.5$mm
$W_2=8.51$m/s
$c=1300$ N/m
$f_o=2.63$ Hz
$D=0.65$ m
$r=1.414$

$C_1=2.278$
$C_2=1.300$
$C_3=3.940$
$C_4=0.027$
$C_5=0.483$
$C_6=0.923$

	experi-mental	flutter model
$A_2:A_1$	1.0	0.98
$b=f/f_o$	1.12	1.15

Fig.11 Lift-, pressure-,and velocity-oscillations at stability limit. a records. b vector representation of"a" and comparision with results calculated with flutter model.

Gas spring effect

The following theoretical result,[1], may be the base for a crucial experiment for the gas spring effect: the flutter model predicts a flutter frequency f at the stability limit for undamped systems

$$f = f_o \cdot \sqrt{1 + \Delta c / c} \qquad (22)$$

Δc is the gas spring constant(see equ(13a) in [6]) which was calculated for the set up to Δc=429N/m. Two different sets of soft springs where used to give an appreciable effect in frequency increase. The following Table compares experimental and theoretical results calculated from flutter model. The agreement is good, especially if one considers that without gas spring effect there would be no frequency increase at all.

c	$\frac{\Delta c}{c}$	f_o	f exper.	f/f_o theory	f/f_o exper.	deviation	
N/m	–	Hz	Hz	-	–	%	
253	1.696	1.144	1.809	1.642	1.581	3.8	
549	0.781	1.685	2.198	1.335	1.304	2.3	

Gas inertia effect

To design an experiment with appreciable gas inertia effect, stiff springs have to be used, thus increasing the natural frequency f_o and C_5 (c=3704N/m; f_o=4.31Hz; f_{go}/f_o=1.325). The set up ran at its limit of ΔP_{12}=500Pa resulting in a measured value $A_2:A_1$=0.89(still below stability limit). With weak springs(c=253N/m) the stability limit was reached already at ΔP_{12}=41Pa! The following Table compares the results:

	experimental	flutter model (complete)	flutter model without gas inertia effect	flutter model without gas inertia and non st. work exchange effect
$A_2:A_1$	0.89	0.85	1.46	2.26
$b=f/f_o$	1.10	1.044	1.09	1.11

It could be seen that the (complete) flutter model gives good results while models without non steady effects give considerable deviations.

5.2 Experiments with reed valves

Obviously in real valves conditions are not as "pure" as in enlarged models but, however, the flutter model should describe at least the basic behaviour of compressor valves with respect to flutter. Quasisteady experiments with 2 reed valves have been performed, valve A and valve B, Fig.12. Fig.13 shows the set up. Valve B is from a commercial 2 cylinder refrigerant compressor and was tested mounted on cYlinder head,Fig.14.

For a certain valve operating in a set up with constant volumes V_1, V_2 the gas spring parameter C_2 is a constant. The main variable, adjustable with a control valve, is the pressure difference ΔP_{12}. A diagram $A_2.A_1 - \Delta P_{12}$ is suitable to present experimental results. Fig.15 gives a schematic sketch of such a diagram. For X<0.7mm the squeezing effect of the reed in many designs gives a considerable gain in stability. In designs with sharp seat edges an additional damping effect may occur which is due to periodic reattachment and separation of flow at the sealing ring of the valve.

At first it was tested if all trends in stability which could be

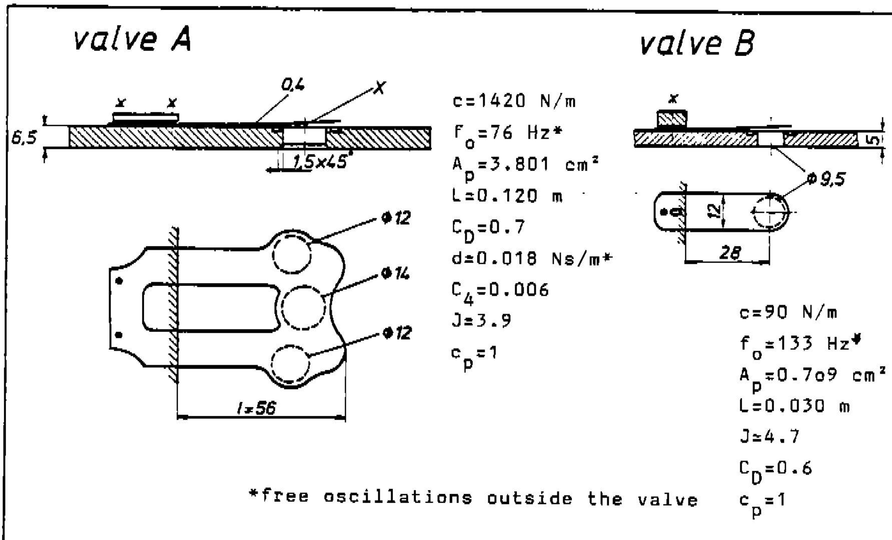


Fig.12 Reed valves used for flutter experiments

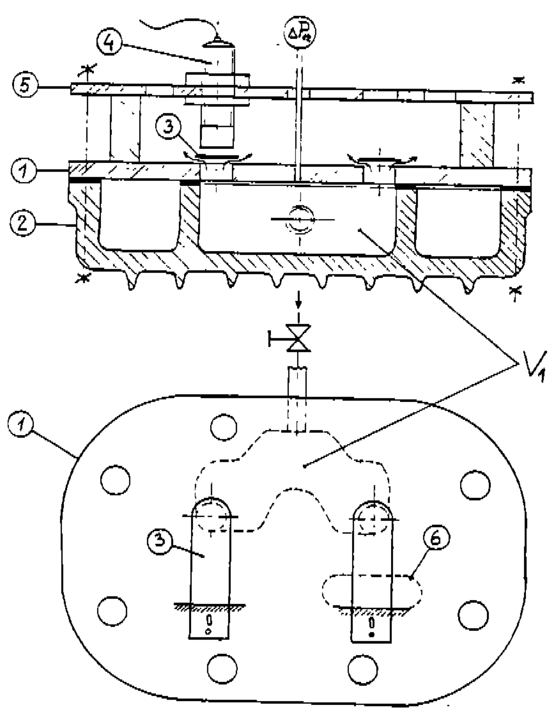


Fig.14 Set up for experiments with valve B. 1 valve seat plate; 2 cylinder head; 3 valve reed; 4 eddy current displacement transducer; 5 transducer mounting plate; 6 cavity machined in seat plate(.5mm deep)

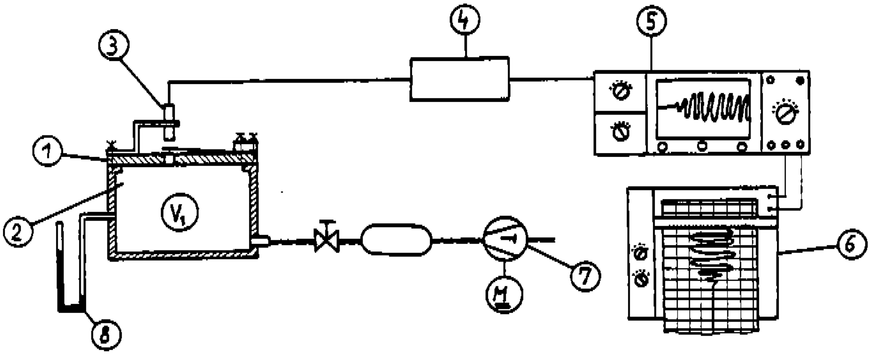


Fig.13 Set up for experiments with valve A. 1 valve seat plate; 2 chamber, V_1=1.5 l; 3 eddy current displacement transducer; 4 electronics to 3; 5 digital storage oscilloscope; 6 recorder.

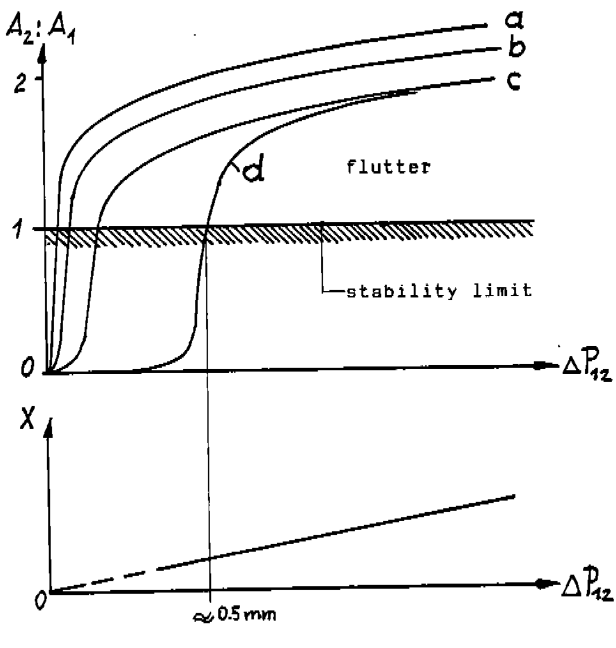


Fig.15 Typical characteristics of flutter model for V_1,V_2=const.;C_3=1. a gas inertia effect, non steady work exchange effect and damping neglected; b gas inertia effect and damping neglected; c damping neglected; d damping according to squeezing effect(reed valves).

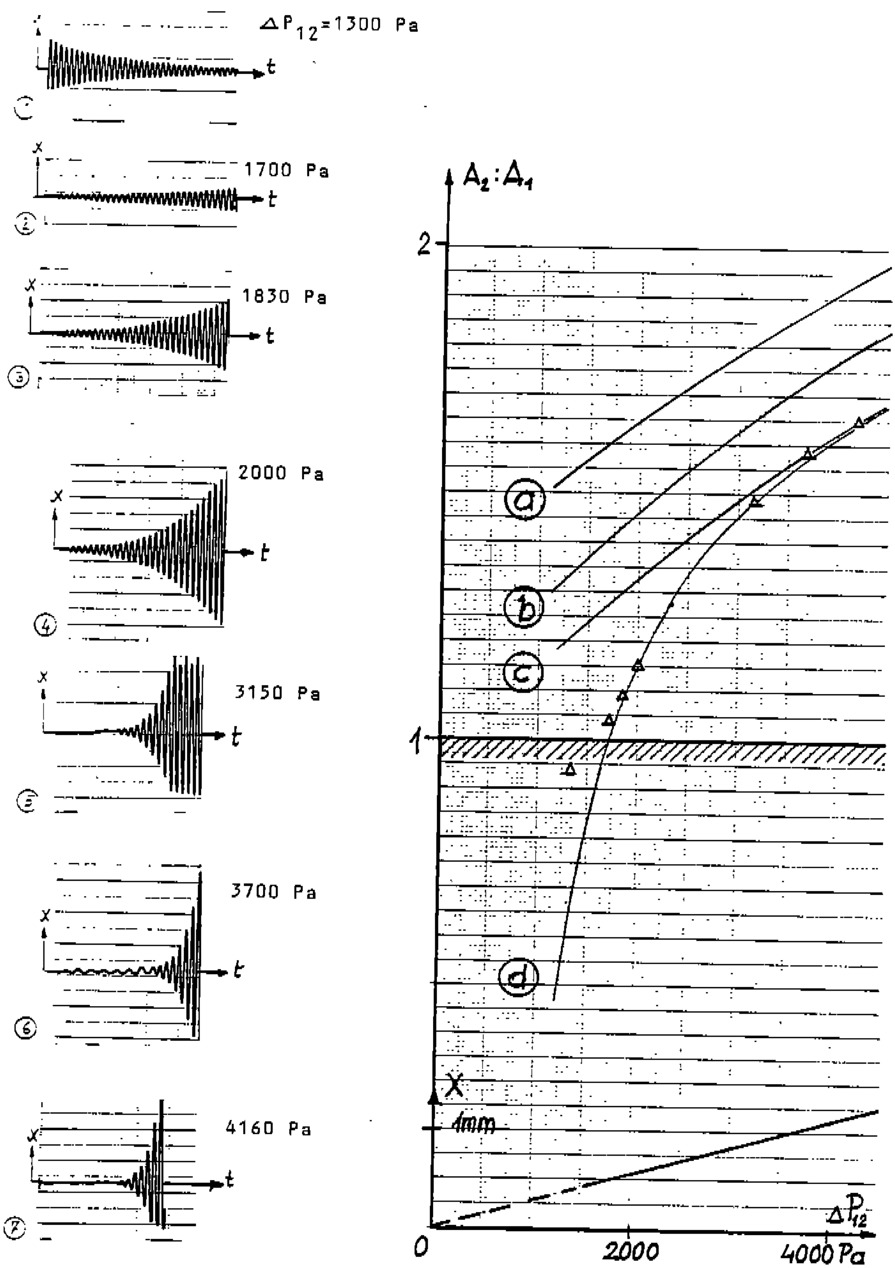


Fig.16 Experimental results, valve A. Left: x-t-records; right: evaluation of the records and comparision with theoretical results: a flutter model, gas inertia, non steady work exchange,damping neglected; b flutter model, gas inertia and damping neglected; c flutter model, damping neglected; d damping according to squeezing effect(K_d=14) Δ experimental results.

observed with compressor valves are predicted by the flutter model, especially:

parameter change	stability increase	stability decrease	comment
increasing ΔP_{12}		*	
decreasing V_1	*	*	depending on C_2, see Fig.7
reed natural fr. decreases		*	
2 valves operate instead of one	*		
reduced damping		*	
spring constant increases	*		

All observations confirmed the model.

Fig.16 shows quantitative results with valve A in comparision with results calculated from flutter model. The following adjustments have been used for calculations

- the natural frequency of the reed was measured outside the valve and multiplied by a factor 1.25(see Fig.7 in [6]).
- the damping force was calculated with equ(14) in [6] using a value $k_d=14$.

Fig.16 shows that for X>0.7mm the simple flutter model with non steady effects included and without damping gives a good approach to experimental results. For X<0.7mm the squeezing effect becomes appreciable and consideration of this describes the flutter behaviour till X=0.3mm.

Experimental results in Fig.16 correspond to a chamber volume V_1=1.5 l. A steel cylinder was put into the chamber reducing the free volume to 1 l. Stability was decreased considerably by this way. The agreement between experimental results and model predictions was similiar as in Fig.16. In an additional experiment a small mass was glued on the reed thus reducing its natural frequency from 76 Hz to 56 Hz. The measured reduction in stability again was in agreement with the flutter model.

Valve B

Stroboscopic observations with both valves operating showed, that both where moving(fluttering) completely in phase for unlimited time.Changing the natural frequency of one of the valves up to about 5% did not change this synchronous movement. The flutter model explains this behaviour in an easy way: the two reed oscillations are coupled by the pressure oscillation in the chamber. Greater deviations in natural frequency of the reeds resulted in noisy irregular oscillations.

To test the ideas concerning the squeezing effect, a cavity,0.5mm in depth, was machined into the seat plate just at the clamping edge of one of the two valves as indicated in Fig.14(marked"6"). The flutter frequency of this valve showed a reduction of about 25% as compared to the other valve, confirming the explanations given in [6],Fig.7. At the same time this reed experienced an enormous loss in stability for small lifts. This is explained by the fact, that the squeezing effect in the cavity area becomes ineffective because the gap is to wide. Fig.17 gives some results.

The flow regime in valve B was more complex as compared to valve A. Obviously this is due to the sharp edges of the hole in the seat plate which may produce flow separation bubbles and hysteresis effects if the reed flutters. Values A_2:A_1 calculated from flutter model showed up to

50% higher values as compared to experiments thus indicating further damping effects in the real flow.

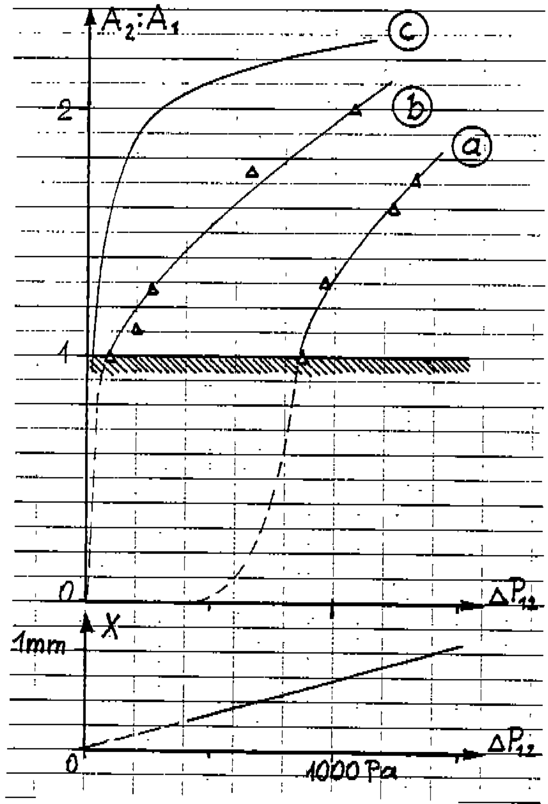


Fig.17 Experimental results, valve B. a valve without cavity according to Fig.14(marked "6"); b valve with cavity; c flutter model results corresponding to "b", damping neglected.

6.SIMILARITY THEORY

Having understood the non steady effects in the compressor process enables us to develop a similarity theory. In connection with such a theory the following questions may be raised:

- Given an existing(or projected) valve, working under certain conditions in a compressor. What are the design rules for an enlarged model for studying flutter phenomena under quasisteady conditions (with a fan) ?
- What are the design rules for an enlarged model for studying dynamic and flutter phenomena with simulation of varying piston displacement ?
- Given an existing compressor designed for gas A(e.g. air). Is it possible to use this compressor for a gas B(e.g.R 22) having similiar pressure time and valve lift-time histories ? If this is possible, for which speed, working pressure etc. similarity is achieved ? If this is not possible, are there minor changes in the compressor that make similarity possible ?

The author has already discussed similarity in a previous paper[4].Inasmuch as non steady effects are concerned this paper is superseded by the present paper. As has been pointed out in the previous paper the Reynolds number is of minor importance for the modelling of valve flow. Furthermore compressibility of the gas with respect to mass flow equation may be neglected. Compressibility with respect to isentropic compressions and expansions in the volumes upstream and downstream the valve is essential and of course has to be included in similarity.

6.1 Similarity rules for quasisteady flutter experiments

Let us suppose that -corresponding to a compressor valve- a scaled model is made, linear scaling factor M. Quasisteady flutter experiments shall be carried out as outlined in [6],Fig.1. We have already found the basic equations for flutter and transformed them into a non dimensional form: equ(7)...(11). In this case we may treat the problem of similarity in a completely analytical way: similarity is achieved if the six constants $C_1...C_6$ for the compressor arrangement and for the enlarged model are identical. Then identical equations describe both systems. In this purely analytical interpretation of similarity(see e.g.[8]) it is not necessary that the model shows geometrical similarity; the values of the six constants have to be identical only.

If we use scaled models of the valve, values like C_D,c_e,r,J,β may be assumed to be identical for valve and model. Concerning the volumes V_1, V_2 upstream and downstream the valve the following rules apply:

- flutter phenomena depend on content of volume only, not on shape.
- theoretically, flutter phenomena depend on volume function v only (equ(6)), not on specific values of V_1,V_2. Nevertheless there is some influence from non steady plenum chamber inflow(suction valve) and outflow(delivery valve) into piping. Hence it would be wise to keep the partition between V_1,V_2 the same in compressor and model. So we may define a "volume scale" M_V and write

$$V_{1,mo}=M_V\cdot V_{1,or} \qquad V_{2,mo}=M_V\cdot V_{2,or} \tag{23}$$

The indices "or" and "mo" denote the original(compressor) and model situation. Using the above mentioned arguments, the similarity law of identical values $C_1...C_6$ may be condensed to rules using quantities more familiar to the practitioner as given in the Table next page. The first relation includes identical Strouhal numbers for valve and model.

An example shall demonstrate the application of the rules:
With a model, scale factor M=20, quasisteady flutter experiments shall be done.

SIMILARITY RULES FOR QUASISTEADY FLUTTER EXPERIMENTS		
$f_{o,mo} = f_{o,or} \cdot \frac{1}{M} \frac{W_{2,mo}}{W_{2,or}}$	natural frequency of valve plate	
$c_{mo} = c_{or} M \cdot \frac{(\rho W_2^2)_{mo}}{(\rho W_2^2)_{or}}$	spring constant	(24)
$M_v = M^3 \cdot \left(\frac{\rho W_2^2}{k P_{m12}}\right)_{or} \cdot \left(\frac{k P_{m12}}{\rho W_2^2}\right)_{mo}$	volume scale $\sqrt{kP_{m12}/\rho}$ = a velocity of sound	
$C_{3,mo} = C_{3,or}$	spring characteristic parameter	
$d_{mo} = d_{or} M \cdot \frac{f_{o,or}}{f_{o,mo}} \cdot \frac{(\rho W_2^2)_{mo}}{(\rho W_2^2)_{or}}$	damping parameter	

Intending to use ambient air and a low pressure fan we choose:

$P_{m12} = 1 bar = 10^5 Pa$; $\rho_{mo} = 1.2 kg/m^3$ $k_{mo} = 1.4$; $W_{2,mo} = 15 m/s$

Fig.18 gives relevant data for the compressor valve and the model. The fan should have a pressure at least five times ΔP_{12} to allow for an adequate flow resistance in the control valve.

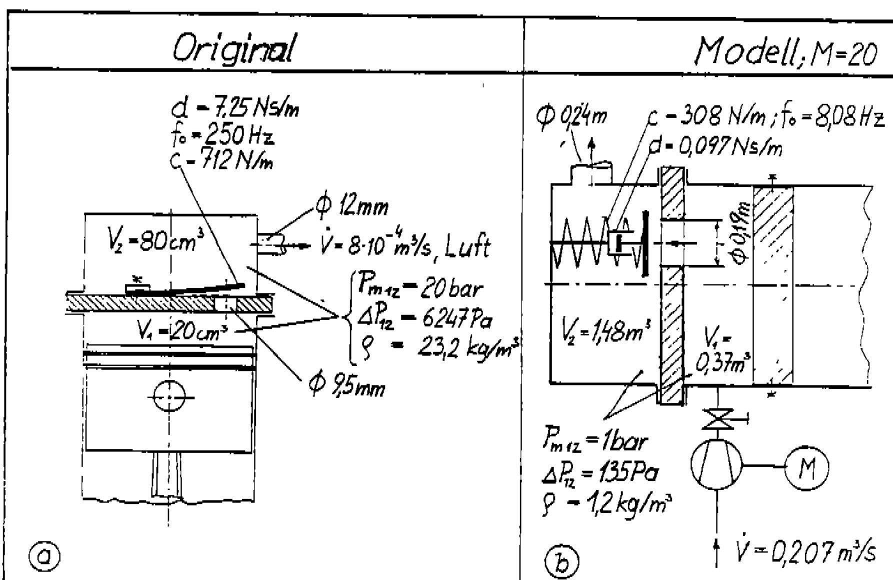


Fig.18 Quasisteady flutter experiments with an enlarged model.
a compressor data; b data for enlarged model according to similarity rules.

6.2 Similarity rules for a set up modelling a non steady compressor cycle

Instead of a fan the volume displacement of a large scale piston, driven by a hydraulic cylinder may be used, Fig.19. Suction and discharge valve have to be investigated separately. On command the piston makes a single and controlled stroke. Now contrary to section 6.1 we do not distinguish between steady state pressure difference ΔP_{12} and small oscillations $\Delta p_{12}(t)$ and between X, x(t) etc.

As we have no analytical representation of the (non linear) compressor cycle process the analytical concept of similarity is not applicable. In this case we may use the concept of force ratios: the valve plate moves under the action of various forces. If the ratio of these forces for compressor valve and model are equal the valve plate and the model valve plate experience similiar time histories. These force ratios are varying with time according to piston movement and has to be kept the same for valve and model. The forces acting on valve plate are the same as we have found in the flutter model, namely:

- spring force($\sim cX$)
- (quasi) steady flow force
- force due to non steady flow
- force due to gas spring effect($\sim \Delta cX$)
- force due to mass transfer effect causing isentropic compressions and expansions in the volumes upstream and downstream the valve
- damping force($\sim d.\dot{X}$)

A new parameter has to be introduced: T_K, being the time during the valve is open. Similarity with respect to time calls for a fixed ratio between T_K and period for one oscillation of natural frequency of the valve plate:

$$(T_K:1/f_o)_{mo} = (T_K:1/f_o)_{or} \tag{25}$$

Accepting furthermore the linearization of the isentropic equation enables us again to use low pressure devices. For the concept outlined the following similarity rules may be derived

SIMILARITY RULES FOR MODELLING NON STEADY COMPRESSOR CYCLE	
$f_{o,mo} = f_{o,or} \cdot \frac{T_{K,or}}{T_{K,mo}}$	natural frequency of valve plate
$C_{mo} = C_{or} M^3 \cdot \frac{\varrho_{mo}}{\varrho_{or}} \cdot \frac{T_{K,or}^2}{T_{K,mo}^2}$	spring constant
$P_{m12,mo} = P_{m12,or} \frac{k_{or}}{k_{mo}} \frac{\varrho_{mo}}{\varrho_{or}} M^2 \frac{T_{K,or}^2}{T_{K,mo}^2}$	working pressure (26)
$C_{3,or}(X) = C_{3,mo}(M \cdot X)$	spring characteristic parameter
$d_{mo} = d_{or} M^3 \cdot \frac{\varrho_{mo}}{\varrho_{or}} \frac{T_{K,or}}{T_{K,mo}}$	damping parameter

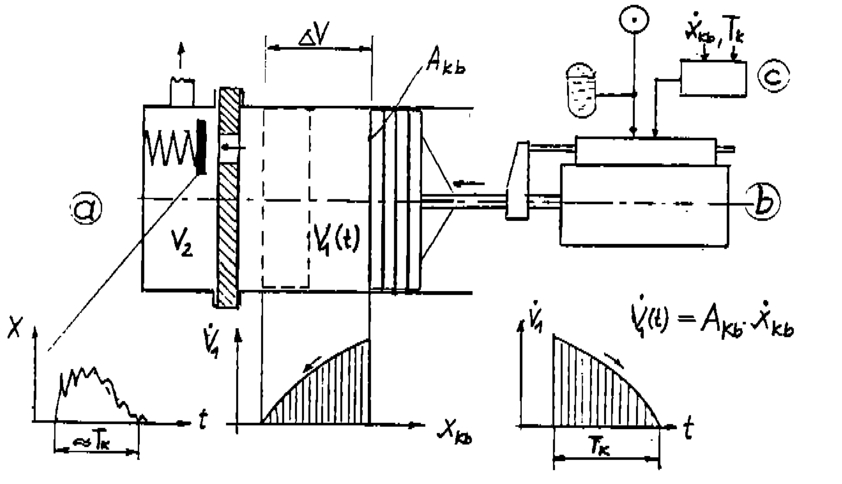


Fig.19 Arrangement for modelling non steady flow and pressure changes in a compressor cycle. a enlarged compressor model; b hydraulic cylinder c control valce for b.

A separate volume scale M_V now is excluded and therefore $M_V=M^3$. It should be noted that similarity is independent from a specific valve design. The valve may be e.g. a ring plate valve with nonlinear spring. The only condition is that the model is scaled.

6.3 Similarity experiments with a compressor with different working gases and operation parameters.

In principle the rules of section 6.2 may be used together with M=1 (similarity experiments with the same compressor!). Some specialities shall be discussed in more detail.

Correct similarity calls for a fixed ratio of reed natural frequency to frequency of rotation(speed), which in turn calls for unchanges speed. If the speed is unchanged the general velocity level in valve remains unchanged. Identical valve lift X calls for identical density of gas. Identical pressure oscillations call for identical velocities of sound. Hence the rules become

SIMILARITY RULES FOR EXPERIMENTS WITH A COMPRESSOR WITH DIFFERENT GASES		
$n_{mo} = n_{or}$	equal speed	
$\rho_{mo} = \rho_{or}$	equal density of gas	(27)
$a_{mo} = a_{or}$	equal velocity of sound	

A minor deviation from ful similarity may be caused by different isentropic exponents causing sligthly different indicator diagrams.Another deviation results from different viscosities if squeezing effects are appreciable.

It comes out that the rules (27) are very restrictive. The only parameter leaving some freedom is the temperature of the model gas which influences a and ρ.

Finally the following question is discussed: a compressor designed for a gas A is operated with excellent valve dynamics at a certain speed. If we want to operate this compressor with similiar valve dynamics with gas B and allow for changes in valve spring constant and in speed, what are the similarity rules in this case ?

Besides the assumption $X_{max,or} = X_{max,mo}$ the following relations must hold

$$M=1 \quad C_{3,Or}=C_{3,mo} \quad Str_{or}=Str_{mo} \quad (f_o:n)_{or} = (f_o:n)_{mo}$$

If we demand equal ratios of forces on valve plate the following rules result

SIMILARITY RULES FOR A COMPRESSOR OPERATED WITH DIFFERENT GASES, SPEEDS AND VALVE SPRINGS				
$n_{mo} = n_{or} \cdot \frac{a_{mo}}{a_{or}}$	compressor speed	$f_{o,mo} = f_{o,or} \cdot \frac{n_{mo}}{n_{or}}$	natural frequency of valve	(28)
$c_{mo} = c_{or} \cdot \frac{(\rho n^2)_{mo}}{(\rho n^2)_{or}}$	spring constant	$d_{mo} = d_{or} \cdot \frac{n_{mo}}{n_{or}}$	damping constant	

While valve lift time histories are identical in amplitude, pressure-change-time histories transform according to

$$\Delta P_{mo} = \Delta P_{or} \cdot \frac{(\rho n^2)_{mo}}{(\rho n^2)_{or}} \tag{29}$$

An example shall make clear the application: the delivery valve of an air compressor shows excellent valve dynamics at n=1800 min^{-1}, P_{m12}=4bar, 180°C, ρ=3.08kg/m³. We want to operate an adapted version of this compressor with the refrigerant R 22 at 5bar, 37°C, ρ=17.9kg/m³. Which are speed and spring-constant, enabling similiar valve dynamics ?

With the rules above and calculated speeds of sound we get

$$n_{mo}=1800 \cdot \frac{179}{427} = 755 \ min^{-1} \qquad c_{mo}=c_{or} \cdot \frac{17.9 \cdot 755^2}{3.08 \cdot 1800^2} = c_{or} \cdot 1.022$$

$$f_{o,mo}=f_{o,or} \cdot \frac{755}{1800} = 0.42 \cdot f_{o,or}$$

Pressure changes transform with

$$\Delta P_{mo} = \Delta P_{or} \frac{c_{mo}}{c_{or}} = 1.022 \cdot \Delta P_{or}$$

The suction valve has to be investigated separately. Not exactly modelled are the following processes:

- different values k cause sligthly different indicator diagrams and hence opening and closing of valves at different crank angles.
- oil stiction effects of valve plate
- details of damping

Damping effects according to gas squeezing may be important with reed valves, but usually are of minor importance in designs like ring plate valves. Another point is the follwing: with reduced speed the valve plate impact velocity is also reduced. If we want to maintain impact velocity, maximum lift may be increased. This calls for further similarity considerations.

7.SOME APPLICATIONS

This section shall demonstrate some ideas how to use the valve flutter theory for the design process.

Starting with adequate basic valve dimensions and operation data the designer may calculate a quasisteady valve-lift time curve(prior to traditional computer simulation) and amplitude ratio $A_2:A_1$ according to valve flutter theory,Fig.20, thus enabling an easy estimation of the flutter situation.A computer can do this calculations and plot the curves. Instead if this,corresponding points of state may be plotted in a stability diagram,Fig.20b.

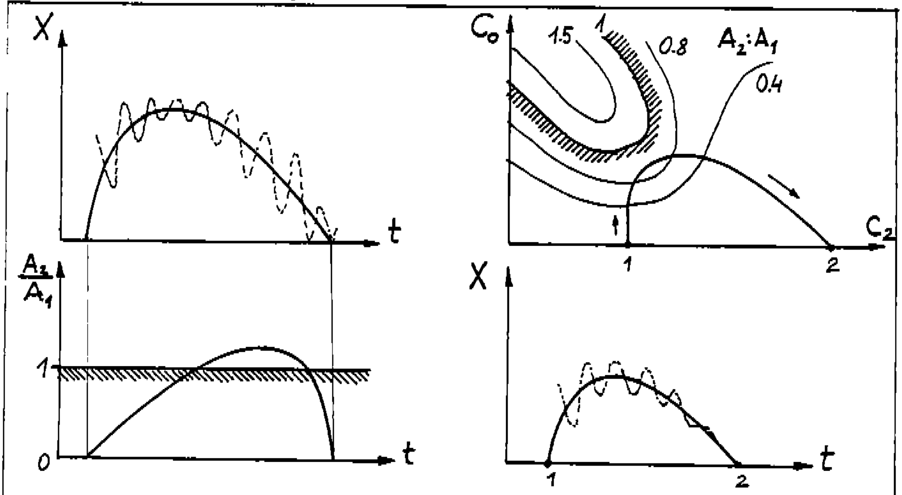


Fig.20 Estimation of the flutter situation.

A method to avoid flutter by "hard ware" -inspired by the theory of valve flutter- is sketched in Fig.21. As the gas spring parameter C_2 acts in a damping way on the valve plate and prohibits flutter at all for $C_2>2$, special damping chambers may be designed as indicated in Fig.21. This chambers have holes to the plenum chambers with effective cross sections in the same order of magnitude as the effective flow area of the valves. The volumes of these chambers shall have values which results in $C_2\approx1-2$ (when formed with cylinder volume and damping chamber volume, ignoring plenum chamber volume).At the expense of additional flow resistance flutter will be avoided under all conditions. Preliminary experiments have been carried out with valve A,Fig.12, with a damping chamber volume of 15 cm^3($C_2\approx1$). An appreciable increase in stability was achieved.

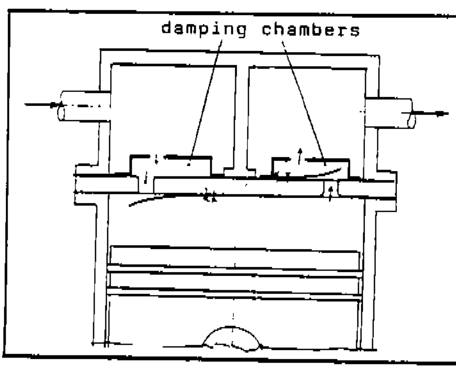


Fig.21 Concept of damping chambers to suppress valve flutter

8.CONCLUSIONS

- The theory of valve flutter outlined in this paper may become a useful tool for the valve designer. It opens a new and more systematic way to avoid valve flutter by choosing adequate design and operation parameters.

- Mathematical description of flutter phenomena in agreement with experiments calls for some refinements in the basic equations as compared to traditional computer simulation models. The author expects that by the introduction of these improvements into simulation models the results achieved will be far more precise.

- Having understood the mechanisms of valve flutter enables the valve designer to search for new concepts to avoid flutter.

- The theoretical background of the flutter theory makes it possible to draw up a similarity theory for the non steady behaviour of valves and pressures in the compressor cycle. Such a similarity theory is a useful tool for the researcher and also for the practitioner who wants to apply well stablished experience to new designs.

- This paper concerns simple valve configurations like reed valves. With some skill the whole argumentation and theory may be extended to more complex valve configurations e.g. ring plate valves as often found in large gas compressors.

9.REFERENCES

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