The calculation of stable cutting conditions in turning

THE CALCULATION OF STABLE CUTTING CONDITIONS IN TURNING

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Nomenclature

b_g	*Limit width of cut*	m
c_{i1}	*Specific process damping coefficient for* $\kappa = 90^o$	Ns/m^2
c'_{i1}	*Specific process damping coefficient for* $\kappa \neq 90^o$	Ns/m^2
$\vec{c}_i$	*Resultant specific process damping*	Ns/m^2
F_1	*Amplitude of the excitation force*	N
h_o	*Nominal undeformed chip thickness*	m
k_e	*Equivalent stiffness of the clamped shaft*	N/m
k_{i1}	*Specific process stiffness for* $\kappa = 90^o$	N/m^2
k'_{i1}	*Specific process stiffness for* $\kappa \neq 90^o$	N/m^2
$\vec{k}_i$	*Resultant specific process stiffness*	N/m^2
s	*Feed*	mm/rev
X_o	*Amplitude of displacement at natural frequency*	m
X_1, X_2	*Apmlitude of displacement*	m
α	*Angle between the principal direction of motion and the direction of the chip thickness modulation; Clearance angle*	o
β_o	*Angle between* $\vec{k}_i$ *and the direction perpendicular to the cut surface*	o
γ_o	*Angle between* $\vec{c}_i$ *and the direction perpendicular to the cut surface*	o
ζ	*Damping ratio of the clamped shaft*	-
κ	*Cutting edge angle*	o
ω_o	*Natural angular frequency*	rad/s

The calculation of stable cutting conditions when turning compliant shafts.

1. Introduction

Chatter caused by the workpiece itself can often be observed when turning compliant shafts. For this the dynamic stiffness of the machine tool must exceed the stiffness of the workpiece. The vibration of the shaft can completely be described by motions in any two different directions. When the two motions are taken perpendicular to each other, there is no significant cross compliance. The two principal directions of motion are preferably chosen as shown in the situation of Fig. (1).

2. Theory

Assuming that the vertical component fo vibration does not affect the dynamic cutting force, this force will only be generated by the horizontal component. When also neglecting the influence of "mode coupling", which experimentally has been proved to be less important, the shaft may be considered as a one-degree-of freedom system. In this respect the situation is rather analogous to the one dealt with in (1), where a special tool holder is applied with $\alpha = 0^o$.

Then according to eq. (34) in (1), the stability limit is defined by

$$b_g = \frac{2\ \zeta\ k_e}{k_{i1}} \cdot \frac{1}{1 - \frac{c_{i1}}{k_{i1}}\ \omega_o} \qquad (1)$$

As the dynamic behaviour of the lathe is of minor importance, the dynamic quantities of the clamped shaft can be derived relatively simple by measuring the dynamic compliance $\frac{X_o}{F_1}$ at natural frequency. Still assuming a one-degree-of-freedom system it gives

$$2\ \zeta\ k_e = \frac{F_1}{X_o} \qquad (2)$$

With different cutting edge angles the related cutting geometry must be accounted for, as the undeformed chip thickness is considered to be a basic quantity. Equivalent cutting conditions are achieved for

$$h_o = s \cos \kappa \qquad (3)$$

When taking the dynamic cutting force ΔF perpendicular to the cutting edge, one can define

$$k'_{i1} = k_{i1} \cos \kappa \tag{4}$$

$$c'_{i1} = c_{i1} \cos \kappa \tag{5}$$

The effective dynamic stiffness of the shaft is

$$\frac{F_1}{X_o} \cdot \frac{1}{\cos \kappa} = \frac{2 \zeta k_e}{\cos \kappa} \tag{6}$$

From the foregoing the stability equation for $\kappa \neq 90^o$ can be derived

$$b_g = \frac{2 \zeta k_e}{k_{i1} \cos^2 \kappa} \cdot \frac{1}{1 - \frac{c_{i1}}{k_{i1}} \omega_o} \tag{7}$$

3. Experimental and calculated results

Experiments to measure the limit width of cut b_g have been carried out with the work material SKF 1550. In orthogonal cutting the feeds were 0.07 and 0.22 mm/rev and in the case of $\kappa = 45^o$ respectively 0.10 and 0.30 mm/rev. The corresponding claculated stability charts were derived with the aid of eqs. (1) and (7), whilst the specific cutting data of the work material were taken from table 1 (1). In the figs. (2) and (3), orthogonal cutting results are shown for a shaft having a dynamic stiffness of $\sim 0.6 \times 10^6$ N/m and a natural frequency of $\sim$ 150 Hz, the influence of the nose radius being eliminated by radial feed. In the figs. (4) and (5), cutting results for $\kappa = 45^o$ are shown in which the influence of the nose radius has been reduced by shortening the shaft, thereby increasing its dynamic stiffness and thus b_g up to three times its initial value. By this the natural frequency shifts to $\sim$ 190 Hz. In reference to experiments by Van Brussel and Vanherck (2), which show the overall dynamic cutting force to be unaffected by the modulating frequency between 130 and 180 Hz, it can be assumed that the values of k_i respectively $c_i \omega_o$ are identical for both situations.

4. Conclusion

Generally the agreement between calculated and experimental results is fair, which confirms the usefullness of the method to obtain the dynamic quantities of the cutting process, as initially described in (3).

About the relevant influence of process damping there are no doubts. In calculating stable cutting conditions in the case of turning compliant shafts, the system may be reduced to a one-degree-of-freedom system, its direction of vibration coinciding with the direction of chip thickness modulation. Considering the undeformed chip thickness as a basic quantity, the results show that orthogonal cutting data can be applied in cases where $\kappa \neq 90^o$.
Finally the assumption has been proved that the vibrational component in cutting speed direction does not affect the dynamic cutting force.
It should be mentioned that, in unstable conditions, experiments showed this component to exceed the component in the direction of chip thickness modulation.

REFERENCES

1. Kals, H.J.J., Fertigung 5 (1971) 165
2. Van Brussel, H., Vanherck, P., Measurement of the dynamic cutting coefficient and prediction of stability. Report 70c16, University of Louvain, presented to C.I.R.P. Technical Committee Ma, Tirrenia (1970)
3. Kals, H.J.J., C.I.R.P. Ann.19 (1971) 297

SUBSCRIPTION OF THE FIGURES

Fig. 1 *An analysis of the vibration of the shaft*
Fig. 2 *The limit width of cut* b_g *as a function of cutting speed v applying a radial feed of 0.07* mm/rev
Fig. 3 *The limit width of cut* b_g *as a function of cutting speed v applying a radial feed of 0.22* mm/rev
Fig. 4 *The limit width of cut* b_g *as a function of cutting speed v applying a longitudinal feed of 0.10* mm/rev
Fig. 5 *The limit width of cut* b_g *as a function of cutting speed v applying a longitudinal feed of 0.30* mm/rev

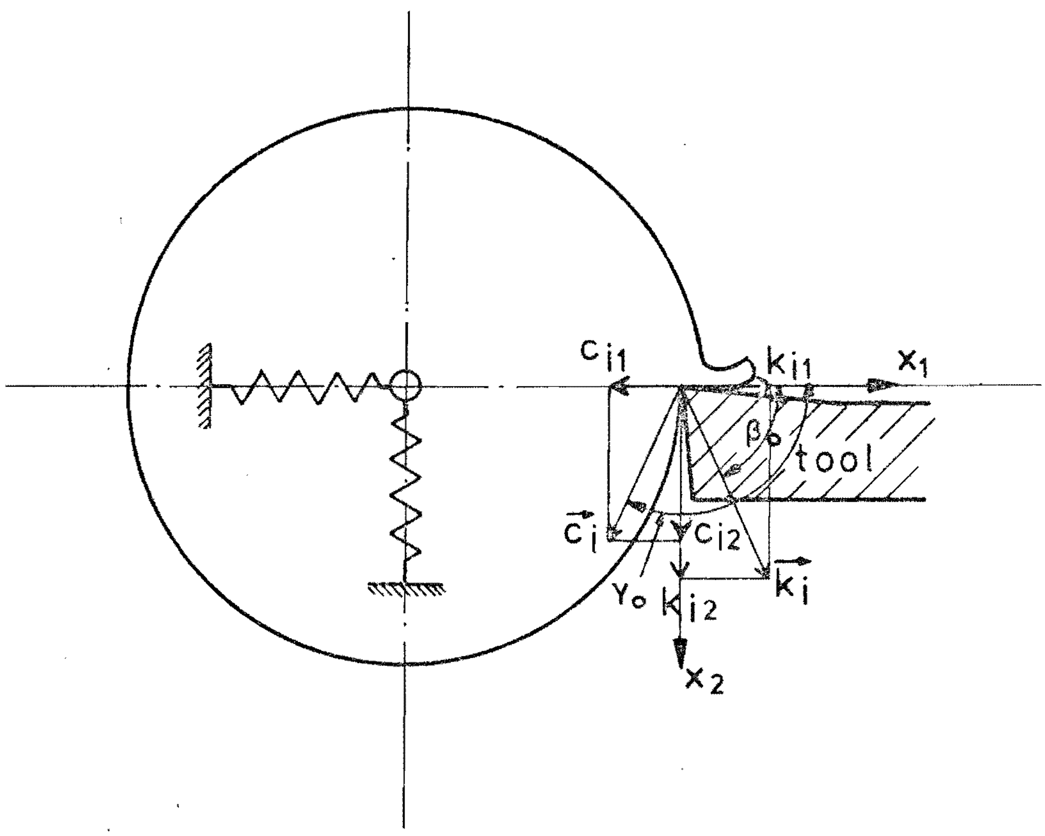
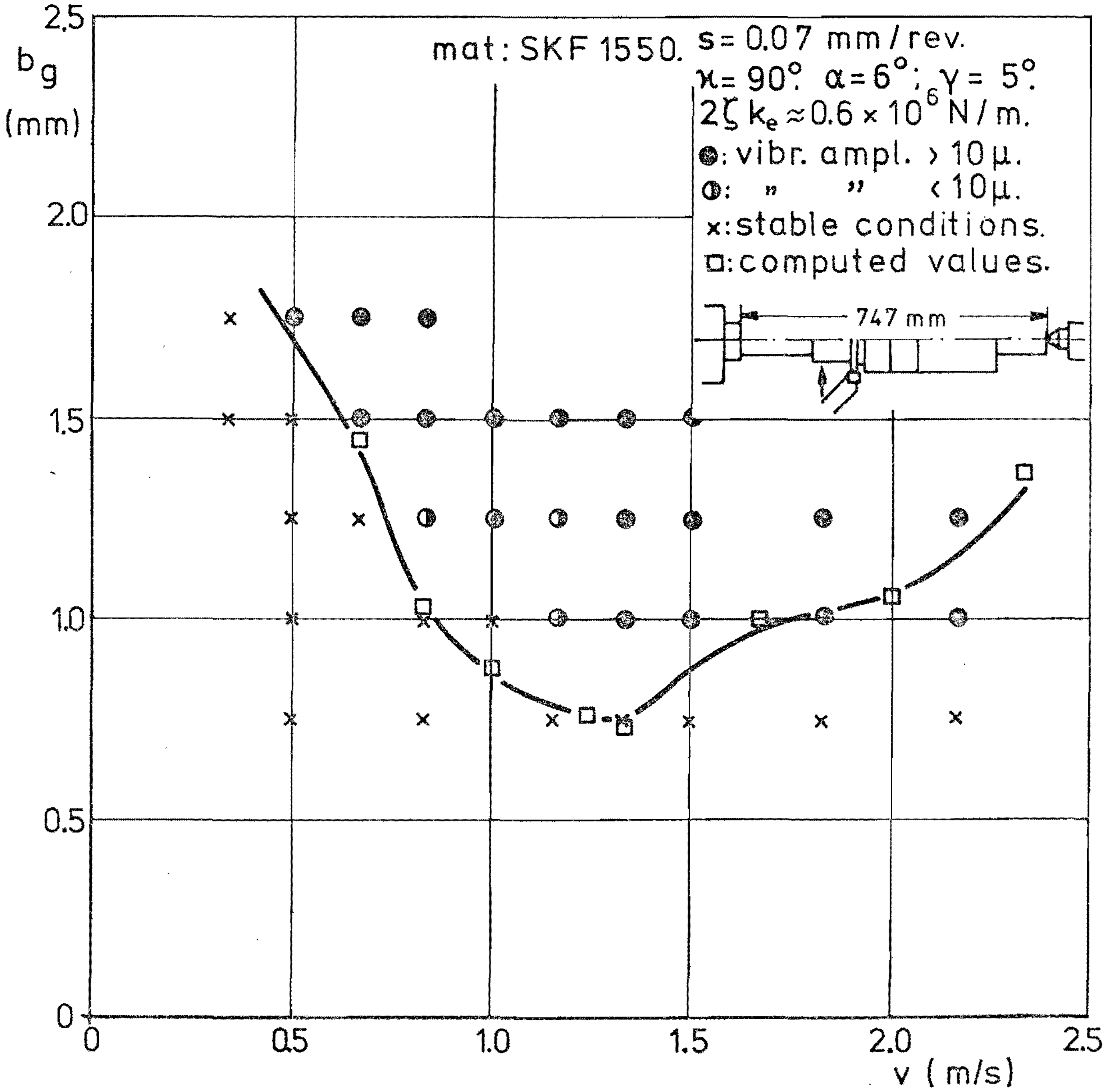
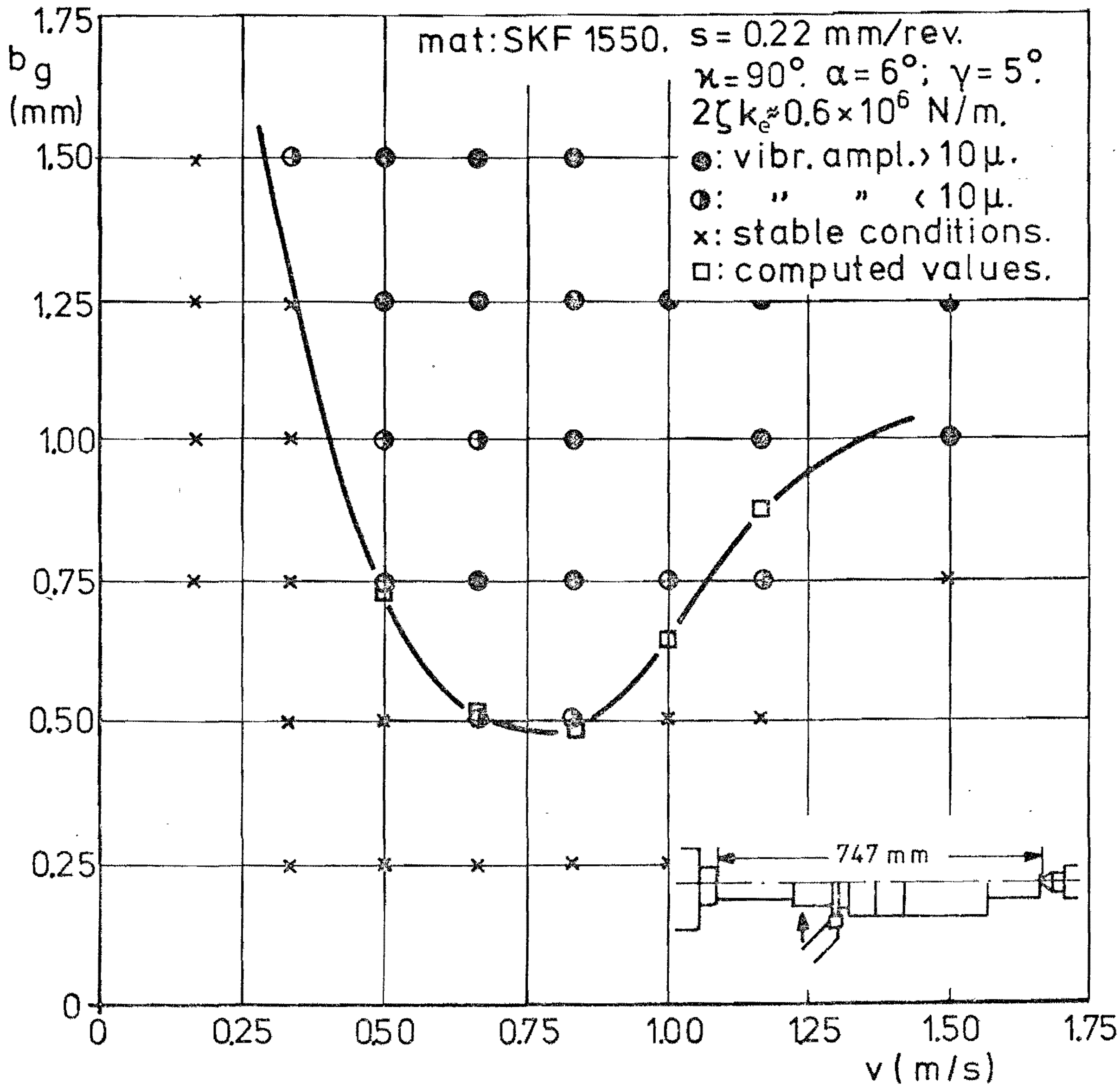


Fig. 1



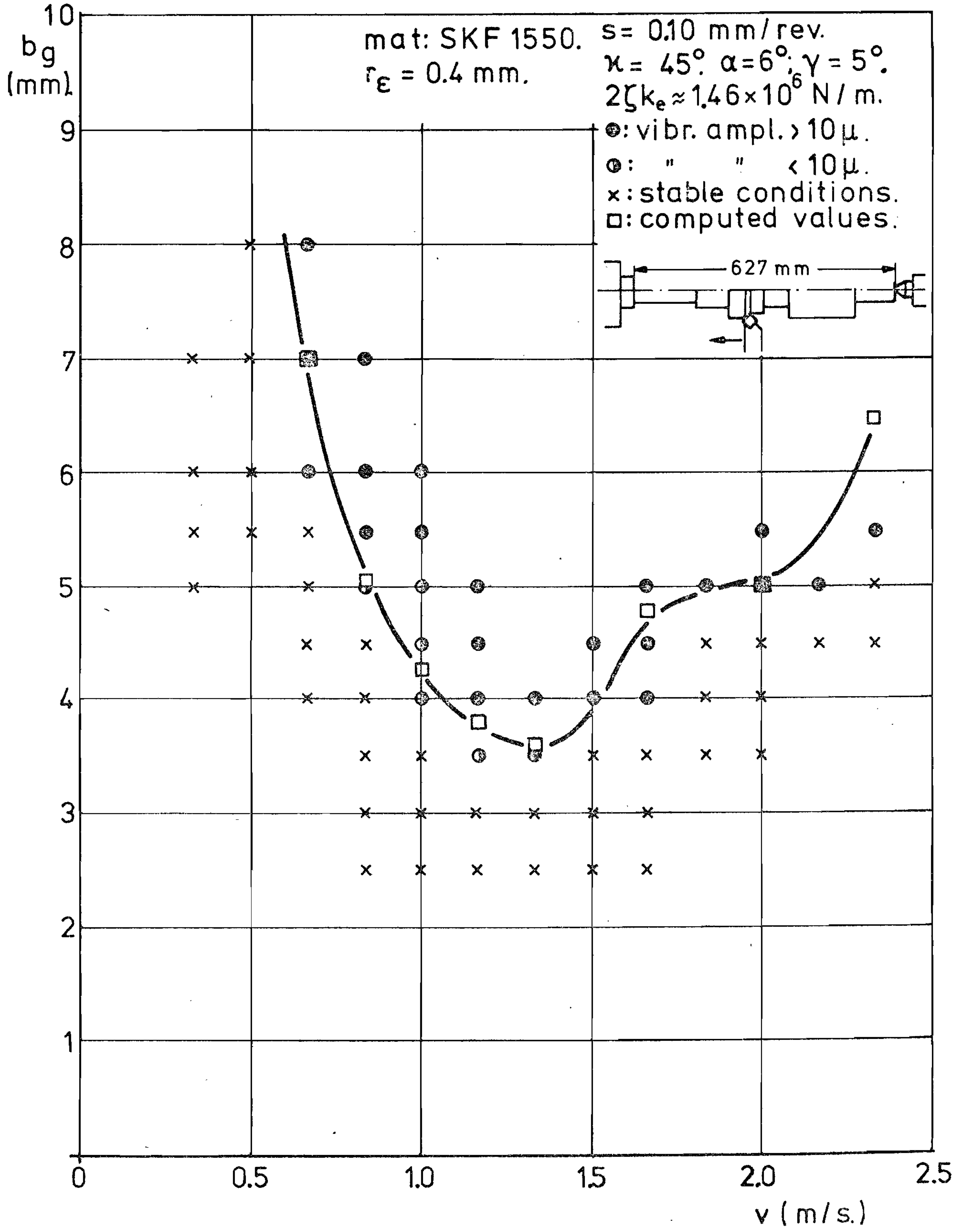
mat: SKF 1550.
s = 0.07 mm/rev.
ϰ = 90°. α = 6°; γ = 5°.
2ζ k_e ≈ 0.6 × 10^6 N/m.
●: vibr. ampl. > 10 μ.
◐: " " < 10 μ.
×: stable conditions.
□: computed values.
747 mm
b_g (mm)
v (m/s)
2.5
2.0
1.5
1.0
0.5
0

Fig. 2



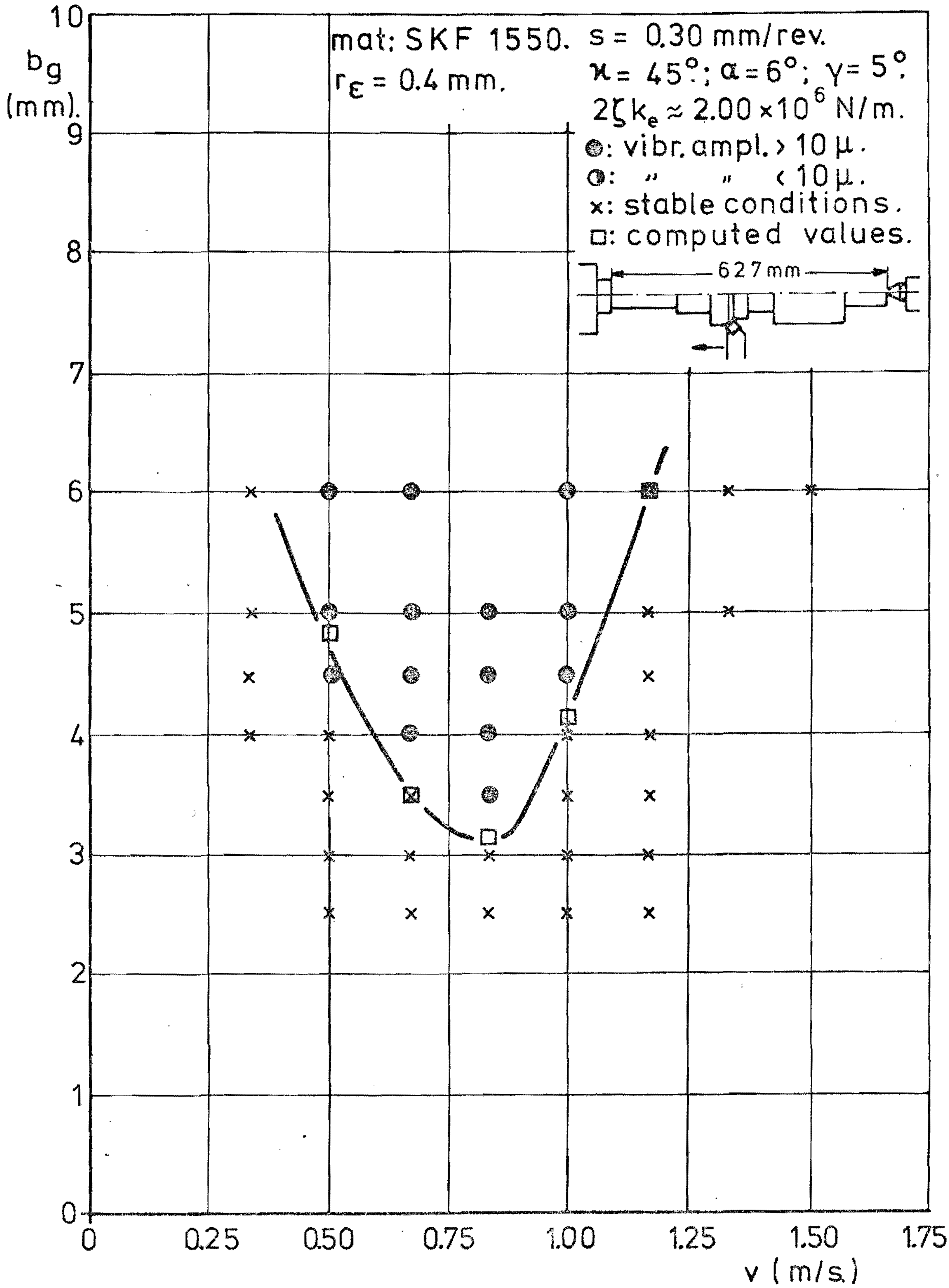
mat: SKF 1550. s = 0.22 mm/rev.
ϰ = 90°. α = 6°; γ = 5°.
2ζ k_e ≈ 0.6 × 10^6 N/m.
●: vibr. ampl. > 10 μ.
◑: " " < 10 μ.
x: stable conditions.
□: computed values.
747 mm
b_g (mm)
v (m/s)
0
0.25
0.50
0.75
1.00
1.25
1.50
1.75

Fig. 3



10
b_g
(mm).
9
8
7
6
5
4
3
2
1
0
0.5
1.0
1.5
2.0
2.5
v (m/s.)
mat: SKF 1550.
r_ε = 0.4 mm.
s= 0.10 mm/rev.
$\varkappa$ = 45°. α=6°; γ = 5°.
$2\zeta k_e \approx 1.46\times10^6$ N/m.
●: vibr. ampl. > 10µ.
◐: " " < 10µ.
×: stable conditions.
□: computed values.
627 mm

Fig. 4



mat: SKF 1550.
rε = 0.4 mm.
s = 0.30 mm/rev.
κ = 45°; α = 6°; γ = 5°.
2ζke ≈ 2.00×10^6 N/m.
●: vibr. ampl. > 10 μ.
◐: " " < 10 μ.
×: stable conditions.
□: computed values.
627 mm
bg (mm).
v (m/s)
0
1
2
3
4
5
6
7
8
9
10
0
0.25
0.50
0.75
1.00
1.25
1.50
1.75

Fig. 5