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REPORT

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**Code of practice for the safe operation of  
work-holding chucks used on lathes**

*Lignes directrices pour l'utilisation sûre des mandrins porte-pièce de tour*

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Reference number  
ISO/TR 13618:1993(E)

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## Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

- type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;
- type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;
- type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 13618, which is a Technical Report of type 2, was prepared by Technical Committee ISO/TC 39, *Machine tools*, Sub-Committee SC 8, *Chucks*.

This document is being issued in the type 2 Technical Report series of publications (according to subclause G.4.2.2. of part 1 of the ISO/IEC Directives, 1992) as a "prospective standard for provisional application" in the field of work-holding chucks for machine tools because there is an urgent need for guidance on how standards in this field should be used to meet an identified need. This Technical Report reproduces practically verbatim British Standard BS 1983-5:1989 and implements it as an ISO Technical Report. For the user's convenience, where possible, references to national standards have been changed to refer to International Standards.

This document is not to be regarded as an "International Standard". It is proposed for provisional application so that information and experience of its use in practice may be gathered. Comments on the content of this document should be sent to the ISO Central Secretariat.

A review of this type 2 Technical Report will be carried out not later than two years after its publication with the options of: extension for another two years; conversion into an International Standard; or withdrawal.

Appendices A to E of this Technical Report are for information only.

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## Introduction

Lathe chucks operated at any speed are potentially very dangerous. They have to be suitably guarded in order to ensure that personnel do not come into contact with a moving chuck and that parts released from the chuck (for whatever reason) cannot be thrown at personnel either directly or after a ricochet. Power chuck controls also have to be suitably interlocked such that workpieces are not inadvertently released. These safety aspects are covered in ISO 13046.

However, because of the versatility of lathe chucks, it follows that chuck designers and manufacturers cannot know the full range of uses to which their chucks will be put (i.e. range of machines on which a chuck may be mounted, type of jaws to be fitted, type of workpiece to be held). It is essential, therefore, for the user to take some responsibility for the application of a chuck. Further, in order that such duties can reasonably be undertaken by the user, it is essential that sufficient design data are available and that methods of calculation and/or of testing are specified. The machine tool manufacturer will also be involved in certain aspects of these problems.

This Technical Report attempts to outline the duties of, and to provide some of the necessary information needed by:

- a) the machine tool manufacturer;
- b) the chuck manufacturer;
- c) the chuck user.

However, because of the large number of chucks already in use, it is necessary also to attempt to recommend the proper course of action regarding the application of existing chucks for which the required design data were not, in fact, transmitted from manufacturer to user and which are now unobtainable.

# Code of practice for the safe operation of work-holding chucks used on lathes

## 1 Scope

This Technical Report identifies and describes safe practices for design and operation of workholding chucks used on turning machines.

The technical aspects covered by this code concern:

- (a) the adequacy of the gripping force in the chuck;
- (b) the fact that at excessive speed there may be failure of chuck components (fracture or excessive yielding);
- (c) acceptable degrees of lack of balance and consequent vibration;
- (d) the inertia loading imposed on the machine drive both by the chuck and by the workpiece;
- (e) gravitational forces arising from the mass of the chuck and workpiece, together in some circumstances with cutting forces, and their effect on the machine;
- (f) other aspects concerning the safe operation of lathe chucks.

Whilst primarily intended for application to lever and wedge type power chucks, including centrifugally compensated types, this code of practice can and should also be applied to manual chucks, but in such cases it is necessary to know the input torque.

NOTE 1. It should be recognized that even when a torque wrench or power driver is used, the grip is known to a lesser accuracy than, say, that of a power chuck having an hydraulically operated drawbar.

NOTE 2. Publications referred to in this Technical Report are listed in Appendix E.

## 2 Chuck grip

### 2.1 General

It should be recognized that there will be change of grip as the rotational speed increases even when the chuck has centrifugal compensation.

In the case of uncompensated, or only partially compensated, chucks set up for external grip, i.e. the jaws move inwards radially as the chuck is tightened, then an increase in rotational speed causes a loss of grip. However, when set up for internal grip an increase in rotational speed causes an increase in grip. Over-compensation has the opposite effect, i.e. an external grip increases with speed. However over-compensation is not recommended in general because it may lead to progressive tightening if the speed is cycled up and down repeatedly.

It is essential that the chuck gripping condition is evaluated by the user or by tooling experts employed by him.

### 2.2 Forces applied to the chuck

2.2.1 *General.* The forces and torques applied via the workpiece to the jaws of the chuck can be represented by four terms:

- $\Sigma F_{ax}$  the total axial thrust;
- $\Sigma F_r$  the total radial force;
- $\Sigma M_d$  the total torque (about the spindle axis);
- $\Sigma M_k$  the total (tilting) moment (about an axis perpendicular to the spindle in the transverse centre plane of the jaws).

Each cutting tool, deadweight force and out-of-balance force and torque makes a contribution, usually to two or more of these total forces and torques, hence each contribution has to be calculated or measured.

Evaluation of mass induced forces requires values of density (see table 1) unless components can be weighed. Evaluation of dynamic forces involves also the eccentricity,  $e$  (see clause 4).

Table 1. Typical value of density,  $\rho$

	kg/m <sup>3</sup>
Magnesium alloy	1800
Aluminium alloy	2750
Iron	7500
Steel	7850
Zinc	7000
Tin	7290
Copper	8780
Nickel	8800
Brass	8280 (on average)

2.2.2 *Cutting forces and torques.* There are many elaborate methods of calculating cutting forces and these methods are not precluded. Nevertheless the following simple methods are deemed to be sufficiently accurate.

(a) For turning, facing and boring:

(1) Estimate the tangential cutting force,  $F_s$  (in N), as:

$$F_s = \text{depth of cut (in mm)} \\ \times \text{feed (in mm)} \\ \times \text{specific cutting force (in N/mm}^2\text{)}$$

where the specific cutting force is taken from table 2.

Table 2. Specific cutting forces, $k_s$ , for turning, facing and boring							
Material		Tensile strength	Brinell hardness number*	Specific cutting force, $k_s$			
				Feed per revolution			
				0.1 mm	0.2 mm	0.4 mm	0.8 mm
		N/mm <sup>2</sup>	HB	N/mm <sup>2</sup>	N/mm <sup>2</sup>	N/mm <sup>2</sup>	N/mm <sup>2</sup>
Carbon steels	low carbon (0.15 % C)	up to 490	up to 150	3600	2600	1900	1360
	low carbon (0.25 % C)	490 to 580	150 to 200	4000	2900	2100	1520
	medium carbon (0.4 % C)	580 to 680	180 to 250	4200	3000	2200	1560
	high carbon (0.55 % C)	680 to 830	200 to 300	4400	3150	2300	1640
Cast steel		290 to 490		3200	2300	1700	1240
		490 to 680		3600	2600	1900	1360
		680+		3900	2850	2050	1500
Alloy steels		680 to 830		4700	3400	2450	1760
		830 to 970		5000	3600	2600	1850
		970 to 1370		5300	3800	2750	2000
		1390 to 1750		5700	4100	3000	2150
Stainless steel		580 to 680		5200	3750	2700	1920
Tool steel		1460 to 1750		5700	4100	3000	2150
Manganese hardened steel				6600	4800	3500	2520
Cast iron			up to 200	1900	1360	1000	720
Cast iron			200 to 250	2900	2080	1500	1080
Cast iron, alloy			250 to 400	3200	2300	1700	1200
Tempered cast iron				2400	1750	1250	920
Copper				2100	1520	1100	800
Copper with commutator mica (collectors)				1900	1360	1000	720
Brass			80 to 120	1600	1150	850	600
Cast copper				1400	1000	700	520
Cast bronze				3400	2450	1800	1280
Zinc alloy Zn-Al 10-Cu2				940	700	560	430
Pure aluminium				1050	760	550	400
Aluminium alloy with high Si content (11 % to 13 % Si)				1400	1000	700	520
Piston alloy Al, Si (toughened) G Al-Si				1400	1000	700	520
				1250	900	650	480
Other aluminium castings		up to 290		1150	840	600	430
		290 to 420		1400	1000	700	520
Wrought aluminium alloys		420 to 579		1700	1220	850	640
Magnesium alloys				580	420	300	220
Hard rubber, ebonite				480	350	250	180
Rubber free insulating compound Novotex, Bakelite, Pertinaz				480	350	250	180
Hard paper, cardboard				380	280	200	140
Hard graphite (nuclear)				—	—	90	—

\*See ISO 4964.

NOTE. When surfacing on a lathe, the depth of cut is measured radially and the feed axially but when facing the depth of cut is measured axially and the feed radially.

Alternatively estimate the power,  $P$  (in W), available as in appendix A and derive the cutting force as follows:

$$\begin{aligned} &\text{Cutting speed,} \\ V \text{ (in m/s)} &= \pi \times \text{cutting diameter (in m)} \\ &\quad \times \text{spindle speed (in r/s)} \end{aligned}$$

$$\text{Tangential cutting force, } F_s = \frac{P}{V}$$

(2) Increase  $F_s$  by 1 % for each degree of top rake less than  $10^\circ$ , add 10 % to allow for tool wear.

(3) Usually, feed force  $\approx 0.6F_s$ . (For difficult materials at slow speed, e.g. titanium, feed force =  $F_s$ .)

The feed force lies parallel to the spindle axis when cylindrical turning or boring, i.e.  $F_v$  in figures 1 to 3. It lies perpendicular to the spindle axis when facing, i.e.  $F_p$  in figures 1 to 3.

(4) Separating force  $\approx 0.25F_s$  and may usually be neglected. The separating force lies perpendicular to the spindle axis when cylindrical turning or boring, i.e.  $F_p$  in figures 1 to 3. It lies parallel to the spindle axis when facing, i.e.  $F_v$  in figures 1 to 3.

(b) For drilling (and, approximately, for deep-hole boring):

(1) Estimate the drilling torque,  $M$  (in N·m), as:

$$M = 1.2k \times C_s$$

where:

$k$  is the work material factor taken from table 3;

$C_s$  is the torque factor, taken from figure 4, for the drill diameter and feed rate in use.

(2) Estimate the feed force,  $F_a$  (in N), as:

$$F = k_f \times F_{s1}$$

where:

$k_f$  is a work material factor taken from table 4;

$F_{s1}$  is a force factor taken either from figure 5 (for drills of all sizes in brass and aluminium and for drills up to 12 mm diameter in steel and cast iron) or from figure 6 (for drills of 16 mm and over in steel and cast iron).

NOTE 1. The information given in table 4 and figures 5 and 6 is based on two separate series of tests and does, therefore, show small discrepancies in the region of 12 mm to 16 mm drill diameter.

NOTE 2. This calculation may be omitted if the workpiece is axially located by the chuck.

Table 3. Work material factor,  $k$ , for drilling (and deep hole boring) torque

Description	Typical specifications	$k$
Steels:		
Low carbon sulphurized (0.1 % C)	220 M 07 (En 1a) 240 M 07 (En 1b)	4 to 4.5
Low carbon low sulphur (0.2 % C) (0.25 % C) (0.3 % C) (0.35 % C) (0.1 % C)	080 A 22 (En 3) 070 M 20 (En 4) 080 M 30 (En 5) 070 M 26 (En 6) 045 M 10 (En 32)	5 to 5.5
Medium carbon (0.4 % C)	080 M 40 (En 8)	4 to 4.5
High carbon (0.55 % C)	070 M 55 (En 9)	
Alloy steels	709 M 40 (En 19) 817 M 40 (En 24) 826 M 40 (En 26)	6 to 6.5
Brass		2.0
Aluminium alloy (cast)		1.6
Cast iron: grey		
Feed rate > 0.7 mm/r < 0.6 mm/r		2.0 3.0
Malleable iron		
Feed rate > 0.7 mm/r < 0.6 mm/r		2.7 3.5

(c) For tapping:

(1) Estimate the torque,  $M$  (in N·m), as:

$$M = k \times C_t \times C_d \times C_m$$

where:

$k$  is the work material factor from table 5;

$C_t$  is the tap factor from table 6;

$C_d$  is the thread depth factor from table 7;

$C_m$  is the thread factor from table 8.

Increase by 50 % to allow for tap wear.

(2) Feed forces when tapping are not easy to estimate and are, in general small enough to be ignored.

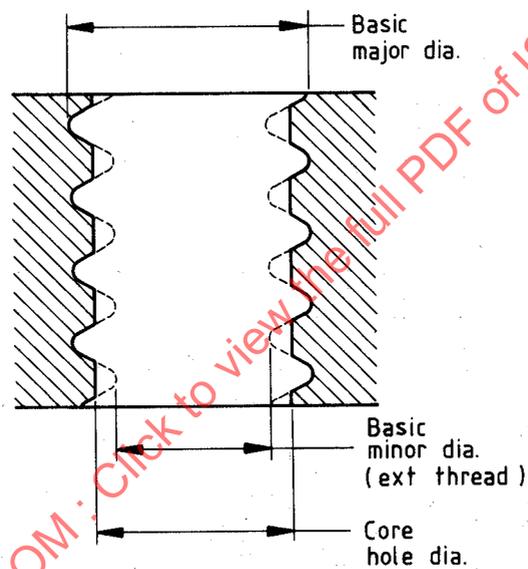
Material	$k_f$	
	Drill diameter	
	12 mm	16 mm
Low carbon steel (up to 0.25 % C) (all feed rates)	800	180
Medium carbon steel (over 0.3 % C) (all feed rates)	1100	200
Grey cast iron feed rate > 0.7 mm/r < 0.6 mm/r	460	80
	640	100
Malleable cast iron feed rate > 0.7 mm/r < 0.6 mm/r	380	90
	500	120
Brass feed rate > 0.7 mm/r < 0.6 mm/r	170	
	300	
Aluminium feed rate > 0.7 mm/r < 0.6 mm/r	400	
	600	

Material	Brinell hardness number*	$k$
Grey cast iron	HB	
	200	1.8
	300	1.9
Malleable cast iron	150	1.6
	250	1.8
Steels:		
low carbon (0.15 % C)	150	2.0
low carbon (0.25 % C)	200	2.4
high carbon (0.55 % C)	300	3.1
typical alloy steel	400	3.5
Aluminium alloys		0.7
Magnesium alloys		0.4
Brass		1.4
Leaded brass		0.7
Phosphor bronze		0.8
*See ISO 4964.		

Tap type	$C_t$
Spiral-point	1.0
Helical flute RH	1.3
Straight flute: in general but over 40 mm dia. length of thread less than one diameter	1.7
	1.3

**Table 7. Values of thread depth factor,  $C_d$ , for tapping**

Depth of thread	$C_d$
%	
55	0.57
65	0.75
75	0.9
80	1.0
85	1.1



$$\text{Percentage depth of thread} = \frac{A}{B} \times 100$$

where:

$A$  = basic major diameter - core hole diameter

$B$  = basic major diameter - basic minor diameter

**Table 8. Values of thread factor,  $C_m$ , for tapping**

(a) ISO metric coarse pitch series

Diameter $\times$ pitch	$C_m$
M3 $\times$ 0.5	0.1
M4 $\times$ 0.7	0.22
M5 $\times$ 0.8	0.35
M6 $\times$ 1	0.63
M8 $\times$ 1.25	1.3
M10 $\times$ 1.5	2.2
M12 $\times$ 1.75	3.5
M16 $\times$ 2	6.0
M20 $\times$ 2.5	11
M24 $\times$ 3	19
M30 $\times$ 3.5	31
M36 $\times$ 4	48
M42 $\times$ 4.5	69
M48 $\times$ 5	96
M56 $\times$ 5.5	133
M64 $\times$ 6	179

(b) ISO metric constant pitch series

Diameter	Pitch (mm)						
	1	1.25	1.5	2	3	4	6
mm	$C_m$	$C_m$	$C_m$	$C_m$	$C_m$	$C_m$	$C_m$
8	0.9	1.3					
10	1.1	1.6	2.2				
12	1.3	2.0	2.7	4.4			
16	1.8		3.7	6.0			
20	2.3		4.7	7.7			
24	2.8		5.6	9.3	19		
30	3.5		7.1	11.8	24		
36			8.6	14.3	29	48	
42			10.0	16.7	34	56	
48			11.5	19	39	65	
56			13.5	22	46	76	
64			15.5	26	53	88	179
72			17.5	29	60	99	202
80			19.0	32	66	111	226
90				36	75	125	256
100				40	83	139	285
110				45	92	154	315
125				51	105	175	360
140				57	118	196	404
160				65	135	225	463
180				73	152	253	522

(c) B.S.W.

Diameter	threads per inch	C <sub>m</sub>
in		
1/4	26	0.6
5/16	22	1.1
3/8	20	1.6
1/2	16	3.2
5/8	14	5.1
3/4	12	8.1
7/8	11	11
1	10	15
1 1/8	9	20
1 1/4	9	23
1 1/2	8	34
1 3/4	7	51
2	7	58
2 1/4	6	86
2 1/2	6	96
3	5	161
3 1/2	4 1/2	227
4	4 1/2	261

(e) B.S.P.

Nominal diameter	Outside diameter	threads per inch	C <sub>m</sub>
in	in		
1/8	0.383	28	0.9
1/4	0.518	19	2.5
3/8	0.656	19	3.2
1/2	0.825	14	6.8
5/8	0.902	14	7.5
3/4	1.041	14	8.7
7/8	1.189	14	10
1	1.309	11	17
1 1/4	1.650	11	22
1 1/2	1.882	11	25
1 3/4	2.116	11	28
2	2.347	11	31
2 1/4	2.587	11	34
2 1/2	2.960	11	39
2 3/4	3.210	11	43
3	3.460	11	46

(d) B.S.F.

Diameter	threads per inch	C <sub>m</sub>
in		
1/4	20	1
5/16	18	1.5
3/8	16	2.3
1/2	12	5.2
5/8	11	7.7
3/4	10	11
7/8	9	16
1	8	22
1 1/8	7	32
1 1/4	7 1/2	35
1 1/2	6	56
1 3/4	5	91
2	4 1/2	126
2 1/4	4	175
2 1/2	4	196
3	3 1/2	300
3 1/2	3 1/4	402
4	3	532

(f) Inch-based constant-pitch series

Diameter	Pitch (threads per inch)			
	8	12	16	20
in				
1		11	6.6	4.5
1 1/8		12.4	7.5	5
1 1/4		13.8	8.3	5.6
1 3/8		15.3	9.2	6.2
1 1/2	34	16.7	10.1	6.8
1 3/4	40	19.6	11.8	7.9
2	46	23	13.5	9.1
2 1/4	52	25	15.2	10.2
2 1/2	58	28	17	11.4
2 3/4	64	31	19	12.5
3	70	34	20	13.7
3 1/4	76			
3 1/2	82			
3 3/4	88			
4	94			

**2.2.3 Loading on the chuck: overhung workpiece, simple tooling.** The loading on the chuck for an overhung workpiece, using simple tooling, is the easiest case to analyse.

Referring to figure 1:

$$\text{Axially } \Sigma F_{ax} = F_v + F_{vax}$$

|  
axial thrust (drilling)  
|  
longitudinal feed force (turning)

$$\text{Torque } \Sigma M_d = F_s \frac{d_z}{2} + Wge + M_{dax}$$

|  
drilling torque  
|  
static unbalance torque  
(neglect if nominally symmetrical workpiece)  
|  
cutting torque (turning)

$$\text{Radial force } \Sigma F_r^* = (F_s - Wg)^2 + (F_p)^2 + W\omega^2 e$$

|  
Dynamic out-of-balance  
( $\omega = 2\pi N/60$  where  $N$  is the spindle speed in r/min)  
(neglect if nominally symmetrical workpiece)

r.m.s. of cutting forces + deadweight

$$\text{Tilting moment } \Sigma M_k^* = \sqrt{\left\{ (F_s/l_z - Wg/l_s)^2 + \left( F_v \frac{d_z}{2} - F_p/l_z \right)^2 \right\} + W\omega^2 e/l_s}$$

|  
Moments in  $F_s$  plane  
|  
Moments in  $F_p$  plane  
|  
Non-rotating

|  
Dynamic out-of-balance  
(neglect if nominally symmetrical workpiece)  
|  
Rotating

**2.2.4 Loading on the chuck: vertical spindle, simple tooling.** From figure 2:

$$\Sigma F_{ax} = F_v + F_{vax} + Wg$$

$$\Sigma M_d = F_s \frac{d_z}{2} + M_{dax}$$

$$\Sigma F_r^* = \sqrt{\{ (F_p^2 + F_s^2) + W\omega^2 e \}}$$

$$\Sigma M_k^* = \sqrt{\left\{ (F_s/l_z)^2 + \left( F_v \frac{d_z}{2} - F_p/l_z \right)^2 \right\} + W\omega^2 e/l_s + Wge}$$

**2.2.5 Inclined slides and multiple slides.** When an inclined slide is used the cutting forces act at a different point and in different directions, see figure 3, where they are denoted by  $F_{sj}$ ,  $F_{vj}$ ,  $F_{pj}$ , for a slide rotated by angle,  $\alpha$ , from the 'horizontal' position.

The forces, torques and moments  $F_s$ ,  $F_v$  and  $F_p$  then become:

$$\text{Axial force, } F_v = F_{vj}$$

Radial forces at the axis,

$$F_s = F_{sj} \cos \alpha + F_{pj} \sin \alpha$$

$$\text{and } F_p = F_{pj} \cos \alpha - F_{sj} \sin \alpha$$

NOTE 1. These terms replace  $F_s$  and  $F_p$  in the equations in 2.2.3.

$$\text{Torque about the axis} = F_{sj} \frac{d_z}{2}$$

NOTE 2. This term replaces  $F_s \frac{d_z}{2}$  in the equations in 2.2.3.

Moment in a vertical plane\*

$$= F_s/l_z - F_{vj} \frac{d_z}{2} \cos \alpha$$

NOTE 3. This term replaces  $F_s/l_z$  in the equations in 2.2.3.

Moment in a horizontal plane\*

$$= F_p/l_z - F_{vj} \frac{d_z}{2} \sin \alpha$$

NOTE 4. This term replaces  $F_p/l_z$  in the equations in 2.2.3.

In the case of multiple slides, each slide is treated as inclined and the resultant values are summed as follows, where the suffix  $j$  indicates the slide:

$$\text{Axially, } F_v = \Sigma F_{vij}$$

$$\text{Radially, } F_s = \Sigma F_{sj}$$

$$\text{and } F_p = \Sigma F_{pj}$$

$$\text{Torque} = \Sigma F_{sij} \left( \frac{d_{zj}}{2} \right)$$

Moment in vertical plane\*

$$= \Sigma \left\{ F_{sj} l_{zj} - F_{vij} \left( \frac{d_{zj}}{2} \right) \cos \alpha \right\}$$

Moment in horizontal plane\*

$$= \Sigma \left\{ F_{pj} l_{zj} - F_{vij} \left( \frac{d_{zj}}{2} \right) \sin \alpha \right\}$$

**2.2.6 Required grip.** The values of  $\Sigma F_{ax}$ ,  $\Sigma M_d$  are used as follows to establish the total grip,  $F_{sp}$ , needed to prevent slip:

$$\mu_{sp} F_{sp} \geq \sqrt{\left\{ \left( \frac{2 \Sigma M_d}{d_{sp}} \right)^2 + \left( \Sigma F_{ax} \right)^2 \right\}}$$

where  $\mu_{sp}$  is the coefficient of friction given in table 9.

NOTE. When the workpiece is axially located by the chuck  $\Sigma F_{ax}$  may be treated as zero provided it has a positive value initially.

\*These items can, at present, be used only subjectively in setting safety factors; no numerical criteria are available.

The choice between tangential and axial values in table 9 is somewhat arbitrary. When there is no positive axial location tangential values for  $\mu_{sp}$  should be used if the torque term ( $2 \Sigma M_d / d_{sp}$ ) is predominant, i.e. for most turning operations. For drilling however when the term  $\Sigma F_{ax}$  predominates it is acceptable to select a value of  $\mu_{sp}$  from the axial column of table 9.

The grip  $F_{sp}$  then has to be increased by a factor,  $S_z$ , in order to provide for:

- (a) a margin of safety to cater for values of  $\Sigma M_k$ ;
- (b) any further margin of safety.

(The force of  $\Sigma F_r$  will cause radial deflection of the workpiece but no criteria are available, currently, to establish limits.)

A minimum of  $S_z = 2$  is to be adopted, increased as necessary to cater for large values of  $\Sigma M_k$  (for which  $l_z > d_{sp}$  probably) and other adverse factors, and a factor  $S_{sp} = 1.5$  is used to provide a margin of safety when calculating the required total static grip, ( $F_{spo}$ ) given by:

$$F_{spo} = S_{sp} (F_{spz} S_z + F_c)$$

for external grip (jaws moving radially inwards to grip)

$$F_{spo} = S_{sp} (F_{spz} S_z - F_c)$$

for internal grip (jaws moving radially outwards to grip)

where  $F_c$  is the centrifugal force on jaws, see 2.3.

**2.2.7 Effect of a tailstock centre.** If a tailstock centre is used then the loading situation at the chuck becomes complex. Two approximate simplifications are possible.

(a) When the workpiece is not axially located by the chuck then an overestimate, and hence a safe estimate of the forces is obtained if the tailstock is ignored and the calculations made as for an overhung workpiece.

This approach is justified on the basis that should the workpiece slip in the chuck then it may well slip off the tailstock centre.

(b) When the workpiece is axially located by the chuck then:

$$\Sigma F_{ax} = 0;$$

$\Sigma F_r^*$  is evaluated after applying the multiplying factor given in figure 7 to each component;

$\Sigma M_d$  is evaluated as for an overhung workpiece;

$\Sigma M_k^*$  is evaluated after applying the multiplying factor given in figure 7 to each component.

Thus the effect of a tailstock centre is to modify the values of  $\Sigma F_v^*$  and  $\Sigma M_k^*$  thus leading to the subjective choice of lower values for the safety factor  $S_z$ .

NOTE. No guidelines are available to deal with this aspect; moreover the values of  $\Sigma F_r^*$  and of  $\Sigma M_k^*$  will usually be small. Hence, at present, it is recommended that  $\Sigma F_r^*$  and  $\Sigma M_k^*$  be neglected.

### 2.3 Change of grip at speed

It is essential that the chuck manufacturer provides graphs, figure 8 being an example, showing the change of grip at various speeds when the chuck is fitted with standard jaws positioned flush with the outside diameter, inwardly stepped (see figure 9(c)). Supplementary data for outwardly stepped jaws and for smaller radii would be acceptable, as additional curves or on separate graphs, as would comparable data for blank jaws. The information may be calculated or obtained experimentally using a stiff load transducer, e.g. one having a steel load path. Results obtained using a flexible load transducer, e.g. of the hydraulic type, are not acceptable.

NOTE 1. The transducer should, preferably, be some 10 times stiffer than the chuck.

The chuck manufacturer also has to state the masses of base jaws and any top jaws supplied and give the location of their centres of mass (both being marked, preferably, on the jaws).

The chuck user has to read off, from the graphs, the change in grip,  $F_c$  (in N), arising from the change in speed:

- (a) an increase for internal gripping;
- (b) a decrease for external gripping. (See 2.1.)

Unless an internal grip has to be limited by the need to avoid marking or distorting the workpiece, it is quicker and preferable to assume  $F_c = 0$ .

The loss of grip should not normally be allowed to exceed one half of its original value.

Where the conditions of use are not covered by the graphs available the chuck user has to calculate the change of grip,  $F_c$ , of uncompensated chucks as:

$$F_c = \omega^2 \Sigma (m_1 R_1)$$

where:

$m_{1,2}$  etc. are the masses of the jaw components (in kg);

$R_{1,2}$  etc. refer to the radii of their centres of mass (in m);

$\omega$  is the angular velocity (in rad/s) =  $2\pi N/60$  where  $N$  is the spindle speed in (r/min).

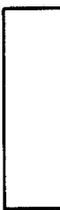
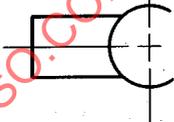
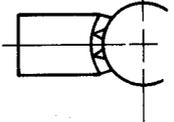
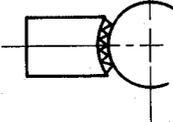
Figure 10 shows a log-log plot of  $F_c$  covering the range of chuck speeds, from 10 r/min to 10 000 r/min, and products of jaw mass (in kg)  $\times$  radius of centre of mass (in m) from 0.001 kg·m to 100 kg·m.

For example, a 5 kg jaw set at 250 mm radius will have an  $mr$  value of  $5 \times 0.25 = 1.25$  kg·m and when rotated at 750 r/min will cause a loss of grip per jaw of  $(2\pi \times 750/60)^2 \times 1.25 = 7711$  N.

NOTE 2. Where the jaw data are not available from the chuck manufacturer or where the user has designed and manufactured the jaw, the user has to determine the required information on the mass and the position of the centre of gravity by calculation or by measurement.

\*These items can, at present, be used only subjectively in setting safety factors; no numerical criteria are available.

**Table 9. Experimentally determined gripping coefficient,  $\mu_{sp}$  (mandrel chuck) (taken from VDI 3106)**

Gripping coefficient, $\mu_{sp}$	Direction of load	Workpiece material 0.32 % carbon steel							
		axial		Workpiece surface					
		tangential	axial						
jaw gripping surface (geometrical shape)   Smooth   Serrated   Fine serrated	 PS Type of chuck	0.05	0.06	0.07	0.10	0.12	0.13	Fine	
		0.08	0.10	0.11	0.16	0.20	0.22	0.22	Fine to rough
		0.11	0.14	0.15	0.22	0.28	0.31	0.31	Unmachined
	 PS Type of chuck	0.10	0.12	0.13	0.17	0.22	0.24	0.24	Fine
		0.16	0.20	0.22	0.29	0.36	0.40	0.40	Fine to rough
		0.22	0.28	0.31	0.41	0.50	0.55	0.55	Unmachined
	 PS Type of chuck	0.17	0.22	0.24	0.29	0.36	0.40	0.40	Fine
		0.29	0.36	0.40	0.48	0.60	0.66	0.66	Fine to rough
		0.41	0.50	0.55	0.67	0.84	0.92	0.92	Unmachined
	PS Scroll chuck PK Facecam chuck KS Wedge chuck		KS		PK		KS		Correction values Al Cu Mg Pb 0.97 Brass 0.92 Cast iron 0.80

Compensated chucks have to be operated strictly in accordance with manufacturers' instructions. The use of other jaws can lead to over or under compensation, both effects being dangerous; the former may lead to progressive tightening up as speeds change while the latter can lead to loss of grip as for an uncompensated chuck.

NOTE 3. The 'compensating' part of the chuck can be represented, in the preceding equation, by a negative  $mr$  term.

## 2.4 Achieving the required grip

It is essential that the chuck manufacturer provides a graph showing actuating torque (or drawbar pull) v. grip achieved for the chuck in the as new condition.

The chuck user has to ensure that the chuck is kept in good condition, i.e. cleaned and relubricated according to the instructions supplied with it and recalibrated as necessary, for example every 12 months. It is important that the correct lubricant is used.

The maximum operating torque or force (as appropriate) should normally be used for all rigid workpieces but with the following provisos and exceptions.

(a) If the required grip is less than 50 % of the available grip according to the manufacturer's graph then there are no provisos.

(b) If the required grip exceeds 75 % of the available grip but is not more than 90 % then the chuck has to have been serviced and recalibrated within, say, 3 months.

(c) If the required grip exceeds 90 % of the available grip then the chuck has to be rechecked before use.

(d) Where adequate grip cannot be achieved then cutting forces and/or spindle speed have to be reduced.

(e) If a grip less than the maximum is required in order to avoid surface markings then the provisos of (a), (b) and (c) apply but reading actual grip instead of available grip; the actuating torque being chosen accordingly.

NOTE. Soft and/or wrap round jaws can be used in order to reduce surface marking on the workpiece.

(f) Where a reduced grip is acceptable by virtue of the flexibility of the workpiece then the required grip should be calculated according to 2.4 not 2.1.

## 2.5 Flexible workpieces

When flexible workpieces are gripped in a multi-jaw chuck, the workpiece distorts and the change in grip at speed is much less.

The equations in appendix B enable the distortion and the diametral stiffness to be calculated.

NOTE. The appendix also illustrates the fact that six equally loaded jaws cause much less distortion than three.

Several situations can arise which need slightly different treatment.

(a) *Internal grip.* Initially do not rely on any increase in grip being available (i.e. in 2.2,  $F_c = 0$ ) although in fact a small increase may occur.

(b) *External grip.* Initially calculate the required grip (i.e.  $F_{sp0}$ ) as in 2.2. If this grip is acceptable from the point of view of geometric distortion, apply it, or an even greater grip, as in 2.4(e) using distortion as the criterion instead of indentation, otherwise treat as item (c).

(c) Where it is required to take advantage of the flexibility of the workpiece in reducing the loss of external grip it is necessary to know the flexibility of the chuck, obtained preferably from the chuck manufacturer, by direct measurement or by loss of grip on a grip transducer of known stiffness. The loss of grip can then be calculated as in figure 11 illustrated in figure 12.

## 3 Maximum speed of the chuck

It is essential that the chuck manufacturer states the maximum speed permissible for the chuck. This speed has to be supported by overspeed type-testing results such that the chuck type will have been run up to speeds 50 % above the maximum speed when fitted with soft jaw blanks of specified mass set to maximum radius as in figure 9.

NOTE. Tests using standard hard jaws can only be deemed sufficient in the case where one-piece jaws are used and when facilities for fitting soft jaws are not provided.

All chuck bodies have to be subjected to inspection for cracks.

The chuck user is responsible for conducting overspeed tests if jaws heavier or at greater radius than the standard conditions described above are to be used at speeds exceeding 75 % of the maximum speed quoted by the chuck manufacturer.

Where it proves impossible to obtain the necessary data from the chuck manufacturer the maximum peripheral speed has to be limited to:

- (a) 18 m/s for chuck bodies made of cast iron;
- (b) 24 m/s for chuck bodies made of ductile iron;
- (c) 32 m/s for chuck bodies made of steel.

The maximum peripheral speeds in revolutions per minute for various diameters of chuck bodies are given in table 10.

## 4 Balancing

### 4.1 Spindle and chuck

It is essential that the machine tool manufacturer and chuck manufacturer state the grade of balance of their respective products in accordance with ISO 1940-1.

NOTE 1. The specific unbalance,  $e$ , is defined as the permissible residual unbalance,  $U$ , divided by the mass,  $m$ . Because the unit of  $U$  is g·mm and the mass is in kg, then it is convenient for  $e$  to be in  $\mu\text{m}$ , and it represents the radial displacement of the centre of mass from the axis of rotation. The quality grade is the resultant circular velocity, that is  $e\omega$ , in mm/s, and when curves of constant quality are plotted on axes of  $U/m$  (or  $e$ ), v.  $\omega$ , the graph in figure 13 is obtained.

**Table 10. Maximum peripheral speeds for various diameters of chuck bodies**

Body diameter	Maximum peripheral speed		
	Cast iron	Ductile iron	Steel
mm	r/min	r/min	r/min
80	4 297	9 730	7 639
100	3 438	4 984	6 112
125	2 750	3 667	4 889
160	2 149	2 865	3 820
200	1 719	2 292	3 056
250	1 375	1 833	2 447
315	1 091	1 455	1 940
400	859	1 146	1 528
500	686	917	1 222
630	547	728	970

Four of the many recommended grades are specifically related to machine tools:

- (a) G 6,3 machine tools and general machinery;
- (b) G 2,5 machine tool drives;
- (c) G 1,0 grinding machine drives;
- (d) G 0,4 armatures of precision grinders.

For general purposes G 2,5 is recommended both for the spindle and for the chuck when mounted on a reference spindle, that is one having a very good quality of balance (i.e. G 0,4) and trueness of mounting faces. For example a 3000 r/min spindle of quality G 2,5 has to have an  $e$  value of 8  $\mu$ m to carry a chuck and maintain quality to G 2,5. An  $e$  value of 8  $\mu$ m implies that run out and wobble of the critical surfaces, at the nose, should be of this order or better.

NOTE 2. Spindles and chuck when graded in this way will usually perform to the same grade when mounted together.

#### 4.2 Workpiece balance

When one considers the workpiece as well as the spindle and chuck, it is not practicable to recommend a quality of balance as in 4.1. However, vibrations can also be assessed on the basis of the r.m.s. amplitude of the velocity they produce, see for example ISO 2372, where ranges from 0.071 mm/s to 71 mm/s are proposed and their appropriate applications listed. If one examines this table and considers class II application, machine tool experience is that group A (i.e. velocities less than 1.12 mm/s) is nearly always acceptable and group D (i.e. velocities exceeding 7.1 mm/s) is rarely acceptable while the area covered by groups B and C (i.e. velocities from 1.12 mm/s to 7.1 mm/s) is uncertain.

#### 4.3 Dynamic behaviour

In the case of lathes in particular, where there is often a rocking mode at a frequency of around 20 Hz to 40 Hz, i.e. 1200 r/min to 2400 r/min, and where there is also a machine fundamental corresponding to a speed usually above the maximum spindle speed, then the relationship between out-of-balance and resulting velocity is rather complicated. Evaluation of this relationship needs a measurement, or calculation, of effective mass and out-of-balance. Moreover, at speeds around the rocking mode natural frequency, it is also necessary to know the damping coefficient.

The problem of specifying permitted unbalance can best be handled by measurement of the velocity of the consequent vibration. Figure 14 shows a plot of such measurements. The off-peak readings are essentially a measurement of general vibrations at the natural frequency of around 26 Hz. The minor peaks are probably where harmonics of the spindle rotational frequency coincide with the natural frequency. The major peak is the effect of out-of-balance and is at spindle rotational frequency. A simple velocity measurement would show this type of response but could not identify the actual frequency components.

#### 4.4 Dealing with out-of-balance workpieces

The following procedures are recommended.

- (a) The machine manufacturer should:

- (1) ensure that normal installation practice does not result in a natural frequency below 10 Hz;

- (2) fit a velocity transducer (or other suitable instrumentation) on to the housing of the front spindle bearing with its axis normal to the spindle and lying usually in a horizontal plane (unless tests have shown that more severe vibration occurs in other directions); the transducer should respond to frequencies above 8 Hz, the output being set to trigger a warning or cut-out device;

- (3) provide upon request, a graph showing the maximum permitted 'mass  $\times$  eccentricity' v. 'spindle speed'.

- (b) It is essential that the user ensures that the installer measures the lowest natural frequency of the machine and that a label is affixed warning the operator never to run unbalanced workpieces at speeds within 25 % of this frequency.

- (c) When the workpiece is obviously unbalanced the user should first consider the use of a process in which the workpiece does not rotate, or he should balance the workpiece (any balance weights added have to be securely fastened). Secondly, he should consider using a face plate (and balancing that if possible; if not item (2) below applies). Only thirdly should he permit the use of chuck provided certain precautions are taken as follows.

(1) When an unbalanced workpiece is held in a chuck, it may be possible to balance the assembly by adding masses to the chuck. Even so, the out-of-balance terms of the equations in 2.2.3 are still applicable as far as chuck grip is concerned and so the grip has to be calculated, accordingly, after choosing the lowest practicable cutting speed.

(2) Calculate, when balancing is not possible, the 'mass  $\times$  eccentricity' and refer to the graph supplied by the machine manufacturer to ascertain the maximum permitted speed. The actual speed selected has to be below this permitted speed. In some cases the resulting speed may be well below a normal speed for the operation.

## 5 Inertia loading imposed on the drive

It is necessary for the machine tool manufacturer to state the total maximum inertia permitted for the chuck and workpiece together. A single inertia value applicable to all

spindle speeds may be given, or a separate value may be given for each spindle speed.

NOTE. The values quoted will usually be chosen to suit the most severe limiting feature whether it be motor, clutches, brake or any other component.

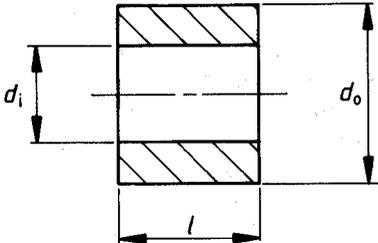
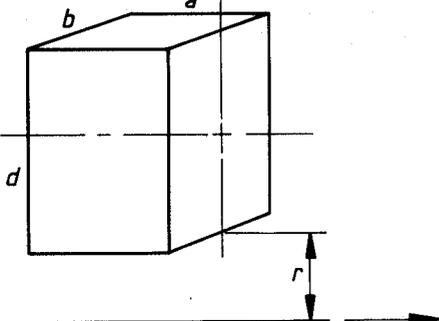
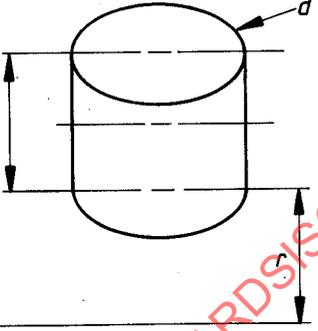
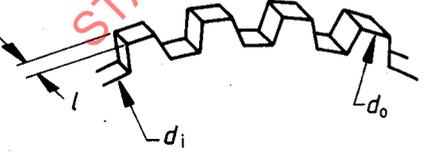
At the discretion of the machine tool manufacturer the value(s) quoted may be qualified according to the frequency of starting and/or stopping.

The chuck manufacturer has to state the inertia of the chuck. Normally the inertia of the chuck fitted with standard jaws and set to maximum gripping diameter should be given and, preferably, this inertia should be marked on the body of the chuck.

The chuck user should ensure that the total chuck and workpiece inertia does not lead to the permitted values being exceeded. Methods of calculating the inertia, (i.e. mass  $\times$  (radius of gyration)<sup>2</sup>) of the simple shapes are shown in table 11, and a practical method of measuring the inertia of complicated shapes is given in appendix C. Typical values of inertia,  $J$ , which apply to chucks wherein the jaws are outwardly offset and lie flush against the external diameter,  $d_1$ , are given in table 12.

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Table 11. Radii of gyration and moments of inertia

Shape and axis of rotation	$k^2$ (Inertia = $mk^2$ )	Inertia (in terms of $\rho$ )
<p>Hollow cylinder</p> 	$\frac{d_o^2 + d_i^2}{8}$	$\rho \frac{\pi l (d_o^4 - d_i^4)}{32}$
<p>Rectangular prism</p> 	$\frac{d^2 + b^2}{12}$ $\frac{4d^2 + b^2 + 12rd + 12r^2}{12}$	$\rho \frac{abd (b^2 + d^2)}{32}$ $\rho \frac{abd (4d^2 + b^2 + 12rd + 12r^2)}{12}$
<p>Cylinder</p> 	$\frac{4l^2 + 3d^2}{48}$ $\frac{16l^2 + 3d^2 + 48rl + 48r^2}{48}$	$\rho \frac{\pi d^2 (4l^2 + 3d^2)}{192}$ $\rho \frac{\pi d^2 l (16l^2 + 3d^2 + 48rl + 48r^2)}{192}$
<p>Toothed profile</p> 	<p>—</p>	$\frac{0.4 \rho l (d_o^4 - d_i^4)}{32} \text{ (approx.)}$

**Table 12. Typical values of inertia for chucks where jaws are outwardly offset and lie flush against the outside diameter**

$d_1$	$J$	
	Hand	Power
mm	kg·m <sup>2</sup>	kg·m <sup>2</sup>
80	0.0012	—
100	0.003	—
125	0.012	—
160	0.03	0.04
200	0.10	0.10
250	0.25	0.3
315	0.70	0.8
400	2.10	2.0
500	5.20	5.0
630	11.0	14.0

NOTE. These figures are taken from DIN 6386 : Parts 1 and 2.

## 6 Gravitational and cutting forces: effect on the machine

It is essential that the **machine tool manufacturer** specifies values which define both the loading permitted at the spindle nose and the axial positions of the bearing centre-lines with respect to the nose generally.

It is essential that the **chuck manufacturer** states the mass of the chuck and, in the case of chucks likely to be used on horizontal spindles, the position of the centre of mass. Normally the mass of the chuck as fitted with standard jaws should be given and, preferably, this mass, together with the axial position of the centre of mass, should be marked on the body of the chuck. Changes in mass arising from the use of non-standard jaws can be neglected.

It is essential that the **chuck user** ensures that the total loading of the spindle nose does not lead to the permitted values being exceeded. A method of assessing the combined effect of mass and cutting force is shown in figure 15 which defines shear force,  $S_F$ , radial reaction,  $R_F$ , and moment,  $M_F$ , at the front bearing, together with axial reaction,  $R_a$ .

## 7 Other aspects of the safe operation of lathe chucks

### 7.1 Chuck keys

A chuck key, if left in the socket when the spindle is started, is an acknowledged hazard.

In the interests of maximum safety, consideration should be given to using:

- a spring to eject the key as soon as it is released (it is both acceptable and preferable for the spring to be fitted to the key); or
- an interlock to prevent spindle rotation under power unless the key is returned to a holder fitted with a detector (a magnetic code, if possible, as this is less easily disabled by the operator than a switch); or
- any other similar device.

### 7.2 Gross overspeeding

In the interests of maximum safety, consideration should be given, not only to fitting warning-plates, but also to positive methods of preventing gross overspeeding such as:

- high-speed spindle noses containing special features, such as pins, grooves and cut-outs, which permit a matching high-speed chuck to be mounted but inhibit the mounting of a non-matching low-speed chuck;

NOTE. Some manufacturers vary the locking arrangement according to the speed range.

- a mechanical guard, trip or interlock which inhibits high-speed selection when a chuck having too large a diameter is mounted or, alternatively, inhibits mounting a large diameter chuck when high-speed is selected;
- where appropriate, provision of a separate high-speed inhibit in addition to any programmed speed selection;
- means to prevent an irregular and, hence, unbalanced workpiece being run at speeds acceptable to the chuck and size of workpiece but dangerous in view of the unbalance (see also 4.2).

### 7.3 Adaptors

The practice of user design, manufacture and application of adaptor/mounting plates is recognized as necessary. Such plates should be manufactured to a formal drawing (not an informal sketch), inspected and marked with a reference number. It is essential that the use of adaptors not so authorized is prohibited.

### 7.4 Mounting bolts for chuck body

It is essential that the chuck manufacturer specifies the tightening torque on the nuts, in the literature supplied with the chuck, together with a specification of nut and bolt material (generally, it is essential that only high tensile steel bolts are used).

Where adaptor plates are used, it is essential that bolts of a similar quality are specified.

It is necessary for the user to ensure that only authorized components are used.

### 7.5 Mounting bolts for jaws

The criteria for mounting bolts for jaws are as for chuck body mounting bolts.

### 7.6 Jaw materials

Chuck users who manufacture their own loose jaws should take care over the choice of material, its heat treatment and detail design.

From BS 970 : Part 1, steels 080 M15 case hardened or 080 A35 or 070 M55 toughened are usually suitable, provided precautions are taken to avoid embrittlement.

It is essential that such components are manufactured and authorized as for adaptor plates (see 7.3).

### 7.7 Dissipation of kinetic energy

In the interests of safety, consideration should be given to the problem of the kinetic energy of the drive, most of which usually resides in the motor. In the event of severe 'collision', e.g. a tool post being hit by a chuck jaw, then the whole of the kinetic energy of the drive is absorbed in 'winding up' the drive to a torque which may be well above normal torques and which is limited sometimes by yielding or fracture of a component. Under such circumstances the normal safety margin outlined in clause 2 could well be inadequate.

Machine tool manufacturers should fit some kind of torque limiting or ensure that 'safe' components such as the primary V-belts can take the same role.

### 7.8 Stroke detectors

It is recommended that stroke detectors are fitted on all automatically operated chucks (collar and jaw type) to detect component contact.

### 7.9 End of bar detectors

It is recommended that end of bar detectors are fitted to prevent chucks gripping an insufficient length of material.

## 8 Summary of the responsibilities of machine tool manufacturer, chuck manufacturer and user

### 8.1 Summary of actions to be taken and information to be provided by the machine tool manufacturer

The machine tool manufacturer should:

- (a) state the grade of balance of the spindle and provide chuck mounting faces of accuracy appropriate to the grade (see 4.1);
- (b) ensure that the natural frequency is normally not below 10 Hz. Fit a velocity transducer adjacent to the front spindle bearing together with a warning display or cut out circuits. Provide a graph of mass  $\times$  eccentricity versus spindle speed. Check that the rocking mode is at a frequency above 10 Hz (if installer) (see 4.4);
- (c) state the maximum permitted inertia of the chuck plus workpiece (see clause 5);

(d) state the maximum permitted loading on the spindle with respect to the centre planes of bearings (see clause 6);

(e) consider, and fit where possible, devices to prevent overspeeding of chucks, (i.e. selecting a spindle speed  $>$  chuck permitted speed (see 7.2);

(f) consider and fit suitable arrangements to absorb the kinetic energy of the drive in the event of a collision between the chuck and a fixture (see 7.7).

### 8.2 Summary of actions to be taken and information to be provided by the chuck manufacturer

The chuck manufacturer should:

(a) provide data showing the change of grip with speed. State the masses of the jaws and give the location of the centre of mass (see 2.3);

(b) provide data showing the relationship between operating force and resultant static grip together with instructions for cleaning and relubricating (see 2.4);

(c) provide data describing chuck flexibility (see 2.5);

(d) state the maximum chuck speed (backed by appropriate type tests) (see clause 3);

(e) state the grade of balance of the chuck (the mounting faces will have to be of an accuracy appropriate to the grade) (see 4.1);

(f) state the inertia of the chuck (see clause 5);

(g) state the mass of the chuck and the axial position of the centre of mass (see clause 6);

(h) fit chuck keys and interlocks having safety aspects in mind (see 7.1);

(i) consider and fit, where possible, devices to prevent overspeeding of chucks (see 7.2);

(j) specify tightening torques and material specification for chuck mounting bolts (see 7.4);

(k) specify tightening torques and material specification for jaw mounting bolts (see 7.5).

### 8.3 Summary of actions to be taken by the chuck user

The chuck user should:

(a) calculate the forces applied to the chuck and the required grip (see 2.2);

(b) calculate the change of grip at speed and/or measure it using a dynamic grip meter (repeat tests at intervals in the case of a compensated chuck) (see 2.3);

(c) calculate the actuating torque (or drawbar pull) and ensure that it is available in practice (see 2.4);

(d) calculate, if applicable, the conditions relevant to a flexible workpiece (see 2.5);

(e) conduct overspeed tests if non-standard large or heavy jaw configurations are to be used (see clause 3);

(f) ensure that the installer measures the lowest natural frequency of the machine and affixes a warning label (see clause 4);

(g) ensure that unbalanced workpieces are either not run, balanced or run slowly, as appropriate (see clause 4);

(h) ensure that permitted inertia loadings are not exceeded (see clause 5);

(i) ensure that permitted external loadings are not exceeded (see clause 6);

(j) ensure that only authorized adaptor plates are used (see 7.3);

(k) ensure that only authorized mounting bolts are used (see 7.4);

(l) ensure that only authorized jaw mounting bolts are used (see 7.5);

(m) ensure that only authorized jaws are used (see 7.6).

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## Appendices

### Appendix A. Estimation of power available at the cutting zone

The maximum cutting power available at the spindle nose for short periods depends upon the overload setting of the motor contactor, the losses in the transmission and the aerodynamic drag on the chuck and workpiece. In the absence of specific information assume that  $1.75 \times$  rated power of the motor is available. In fact a more accurate figure may well be about two-thirds of this power calculated as follows.

(a) It is necessary for the machine tool manufacturer to state the power available when using the recommended overload contactor settings, and either:

- (1) the efficiency of the drive (when it is warmed up) for each of the speeds available from a stepped ratio drive or for a selected range of speeds from an infinitely-variable drive; or
- (2) the losses in the drive, the loss under no load conditions generally being sufficiently accurate.

(b) It is necessary for the chuck manufacturer to state, for high-speed chucks (i.e. those running above, say, 50 % of the limiting speeds shown in clause 3), the drag coefficient,  $\alpha$ , for jaw configurations in figures 9(a) and 9(c) such that:

$$\text{power loss (in } W) = \alpha N^3$$

where:

$N$  is the running speed (in r/min).

(c) The chuck user then has to calculate the power available as:

$$\text{power available} = \text{motor overload power} - \text{transmission losses} - \text{drag (workpiece drag has to be ignored)}$$

If the chuck user does not have the above information available, then the following procedure applies.

(i) The motor manufacturer can, and will, usually supply information of current,  $I_L$ , and power factor,  $\cos \phi$ , v. output torque.

(ii) The no-load losses can be measured and the power available deduced as follows.

Connect an ammeter into the motor circuit.

Run the spindle under no load, chuck jaws in the required position, at the required speed to warm up.

Take a meter reading when conditions stabilize (30 min to 1 h typically).

(iii) The no-load input can be calculated from the meter reading,  $I_L$ , using the power factor,  $\cos \phi$ , as:

$$\text{input} = \sqrt{3} V_L I_L \cos \phi$$

(iv) The overload input is calculated from the overload settings using the same formula.

(v) The difference is a close approximation to the cutting power available.

### Appendix B. Radial stiffness and out-of-roundness of ring held in jaws

A general solution to the deflection,  $\delta$ , at each of any number of equally-spaced load points (see figure 16) is given by Roark and Young\* as:

$$\delta = \frac{WR^3}{EI} \left\{ \frac{1}{4s^2} (\theta k_1 + sck_3) - \frac{1}{2\theta} \right\}$$

where:

$W$  is the load per point;

$\theta$  is half the angle between load points;

$s$  is the  $\sin \theta$ ;

$c$  is  $\cos \theta$ ;

$R$  is the radius of centroid of the cross section;

$$\left. \begin{aligned} k_1 &= 1 + \alpha + \beta \\ k_3 &= 1 + \alpha - \beta \end{aligned} \right\} \text{ where } \alpha = \frac{I}{ARE} \text{ (hoop stress factor);}$$

$$\beta = \frac{FEI}{GAR^2} \text{ (shear stress factor);}$$

$A$  is the area of the cross section (=  $bd$  for a rectangular section);

$I$  is the second moment of area of the cross section (=  $bd^3/12$  for a rectangular section);

$E$  is the modulus of elasticity;

$G$  is the shear modulus;

$F$  is the shape factor of the cross section (=  $6/s$  for a rectangular section).

The 'diametral' stiffness is given by  $(nW/2\delta)$  where  $n$  is the number of jaws.

It is sometimes convenient to know the out-of-roundness caused by the grip. The deflection midway between the load points is given by:

$$\delta' = - \frac{Wn^3}{EI} \left( \frac{1}{2\theta} - \frac{1}{4s^2} (sk_3 + \theta ck_1) \right)$$

Hence the difference in radius between the load points and the midway points ( $\delta - \delta'$ ) is given by:

$$\delta - \delta' = \frac{WR^3}{EI} \left( \frac{1-c}{4s^2} \right) (\theta k_1 - sk_3)$$

In most cases the values of  $\alpha$  and  $\beta$  are negligible hence  $k_1 = k_3 = 1$ , hence:

$$\delta = \frac{WR^3}{EI} \left\{ \frac{1}{4s^2} (\theta + 1sc) - \frac{1}{2\theta} \right\}$$

and

$$\delta - \delta' = \frac{WR^3}{EI} \left\{ \frac{1}{4s^2} (1-c) (\theta - s) \right\}$$

If  $\delta = (WR^3/EI) k_\delta$  and  $\delta - \delta' = (WR^3/EI) k_{\delta-\delta'}$  then  $k_\delta$  and  $k_{\delta-\delta'}$  may be tabulated as shown in table 13.

\*Roark and Young. Formulae for stress and strain. McGraw Hill Inc., 1965, p. 226.

Table 13. Values of  $k_\delta$  and  $k_{\delta-\delta'}$ 

Number of jaws	$\theta$	$k_\delta$	$k_{\delta-\delta'}$
	degrees	m	m
2	90	0.07439	0.14270
3	60	0.01594	0.03020
4	45	0.00608	0.01147
6	30	0.00168	0.00316

By writing  $b d^3/12$  for  $I$  and rearranging one obtains:

$$\delta = \frac{W}{b} \left(\frac{R}{d}\right)^3 \frac{12}{E} k_\delta$$

and

$$\delta - \delta' = \frac{W}{b} \left(\frac{R}{d}\right)^3 \frac{12}{E} k_{\delta-\delta'}$$

Taking a typical case by way of example, let:

$$W = 10\,000 \text{ N per jaw}$$

$$b = 0.01 \text{ m length of ring}$$

$$R/d = 10$$

$$E = 207 \times 10^9 \text{ N}\cdot\text{m}^{-2}$$

$$\delta = \frac{10^4}{10^{-2}} (10)^3 \frac{12}{207 \times 10^9} k_\delta = 0.058 k_\delta$$

$$\delta - \delta' = 0.058 k_{\delta-\delta'}$$

Thus the out-of-roundness would be some 8.2 mm for a 2-jaw chuck, 1.7 mm for a 3-jaw chuck, but only 0.18 mm for a 6-jaw chuck. The roundness would improve rapidly in the case of thicker sections.

These equations can be applied to rings whose axial length is contained wholly under the jaw and would give accurate results applicable:

- (a) to evaluating distortion on gripping the workpiece and hence geometric errors on release of the workpiece;

- (b) for comparison with chuck stiffness and hence lead to a more accurate evaluation of loss of grip at speed.

For longer, thin tube-type workpieces the rigidity of the part not under the jaw would cause errors which could be resolved only by more complex analysis.

## Appendix C. Measurement of the inertia of irregular components

A trifilar suspension, as shown in figure 17, acts like a rotary pendulum. The frequency of oscillation,  $\omega$ , is given by:

$$\omega = \frac{r}{k} \sqrt{\left(\frac{g}{l}\right)}$$

where:

$r$  and  $l$  are the dimensions of the rig indicated in figure 17;

$k$  is the radius of gyration of the table;

$g$  is the acceleration due to gravity.

The inertia of a simple circular plate table can be calculated, or deduced from the torsional natural frequency of the table  $\omega_0$  as:

$$I_{\text{table}} = \text{table mass} \times \frac{r^2}{\omega_0^2} \left(\frac{g}{l}\right)$$

A mass of unknown inertia placed on the table such that its centre of mass is at the centre of the table will lower the natural frequency to  $\omega_1$ , whence:

$$I_{\text{table} + \text{workpiece}} =$$

$$= (\text{mass of table} + \text{workpiece}) \frac{r^2}{\omega_1^2} \left(\frac{g}{l}\right)$$

Hence the inertia of the workpiece can readily be deduced.

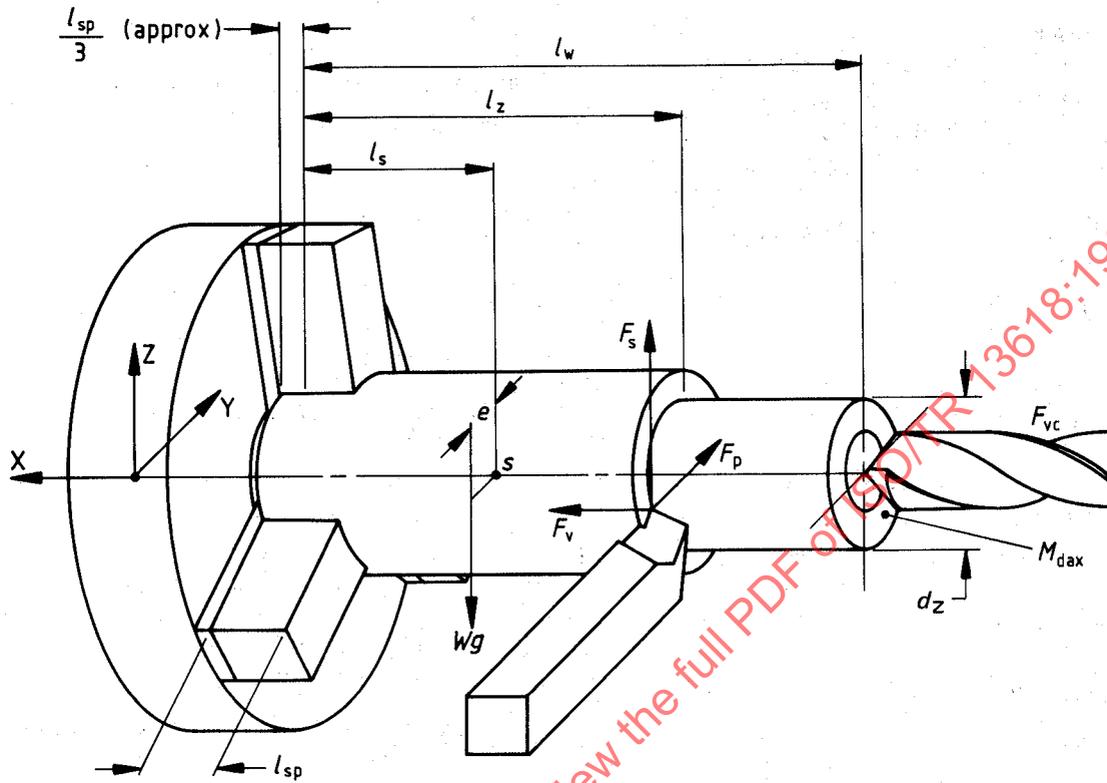


Figure 1. External forces: horizontal spindle, overhung workpiece

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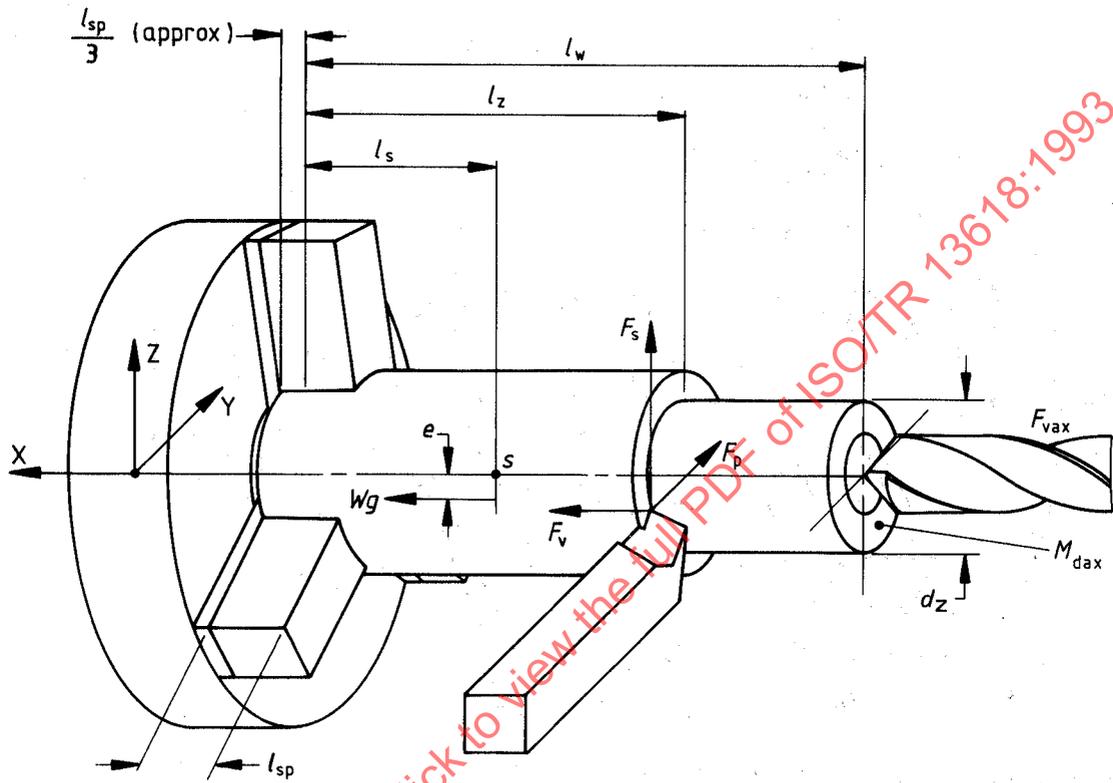


Figure 2. External forces: vertical spindle

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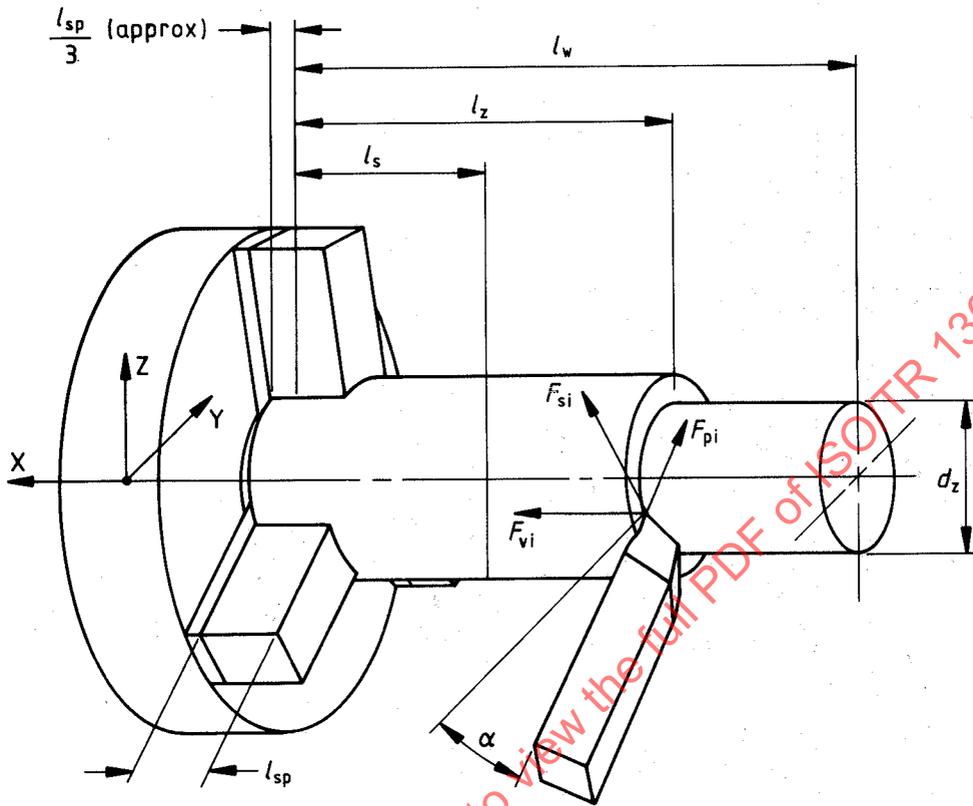


Figure 3. External forces: inclined slide

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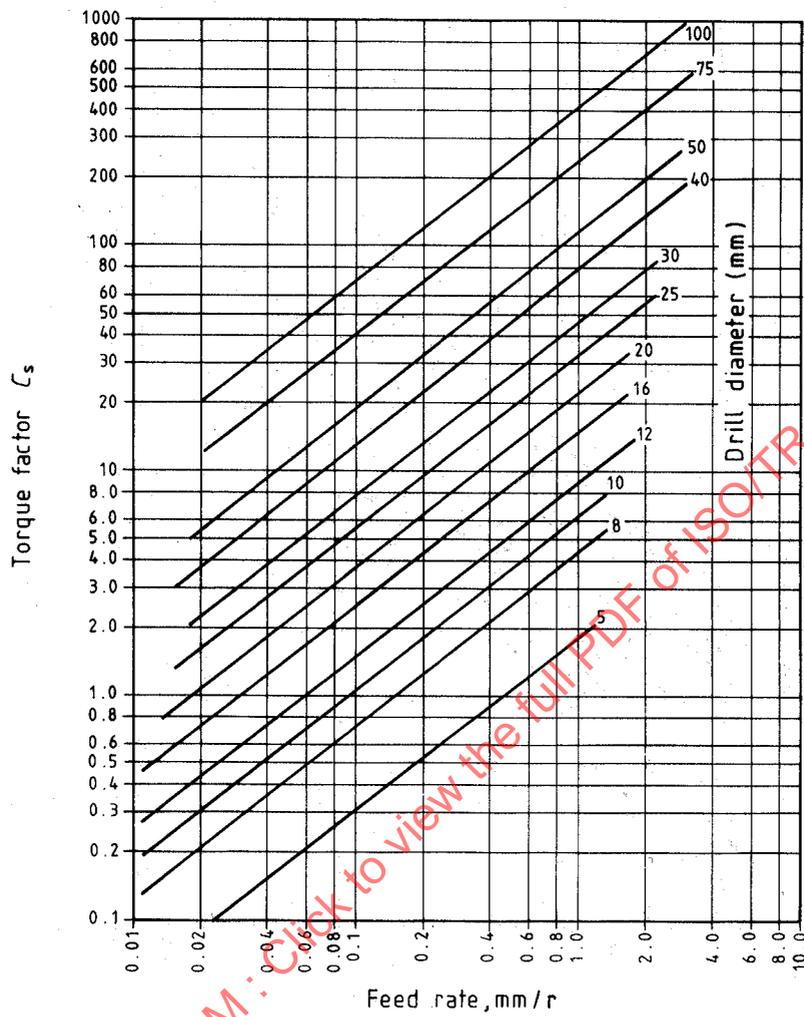


Figure 4. Feed rate v. torque factor  $C_s$ : drilling

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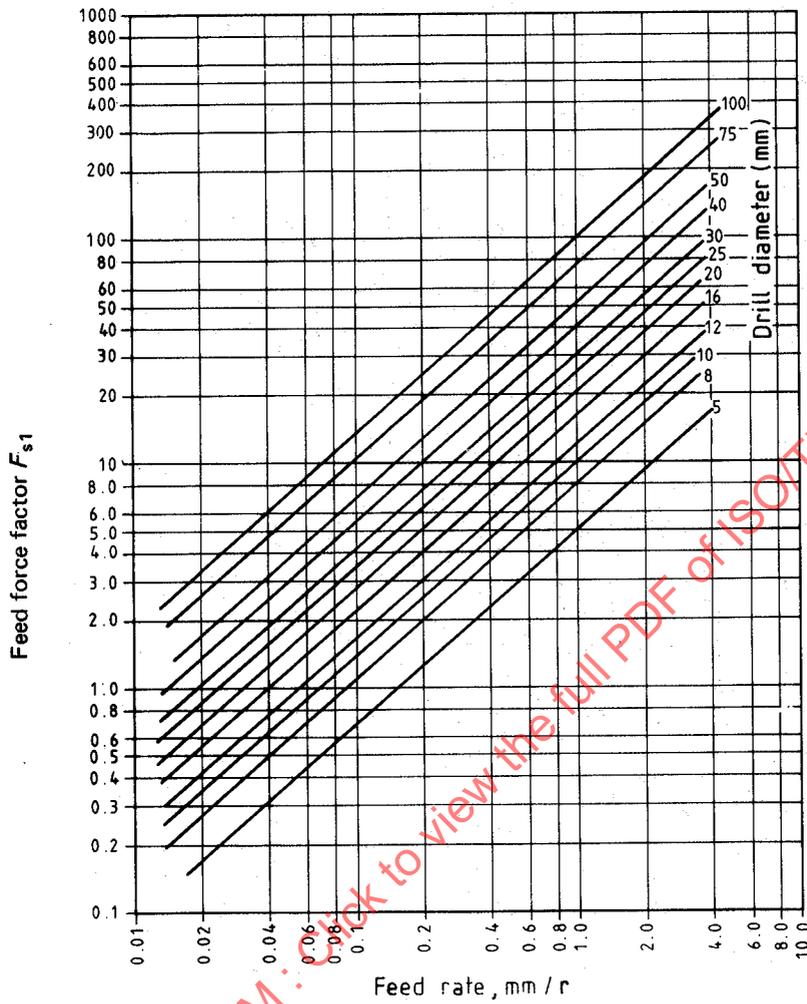


Figure 5. Feed rate v. feed force factor  $F_{s1}$  (for drills up to 16 mm diameter in steel and cast iron and for all drill sizes in brass and aluminium)

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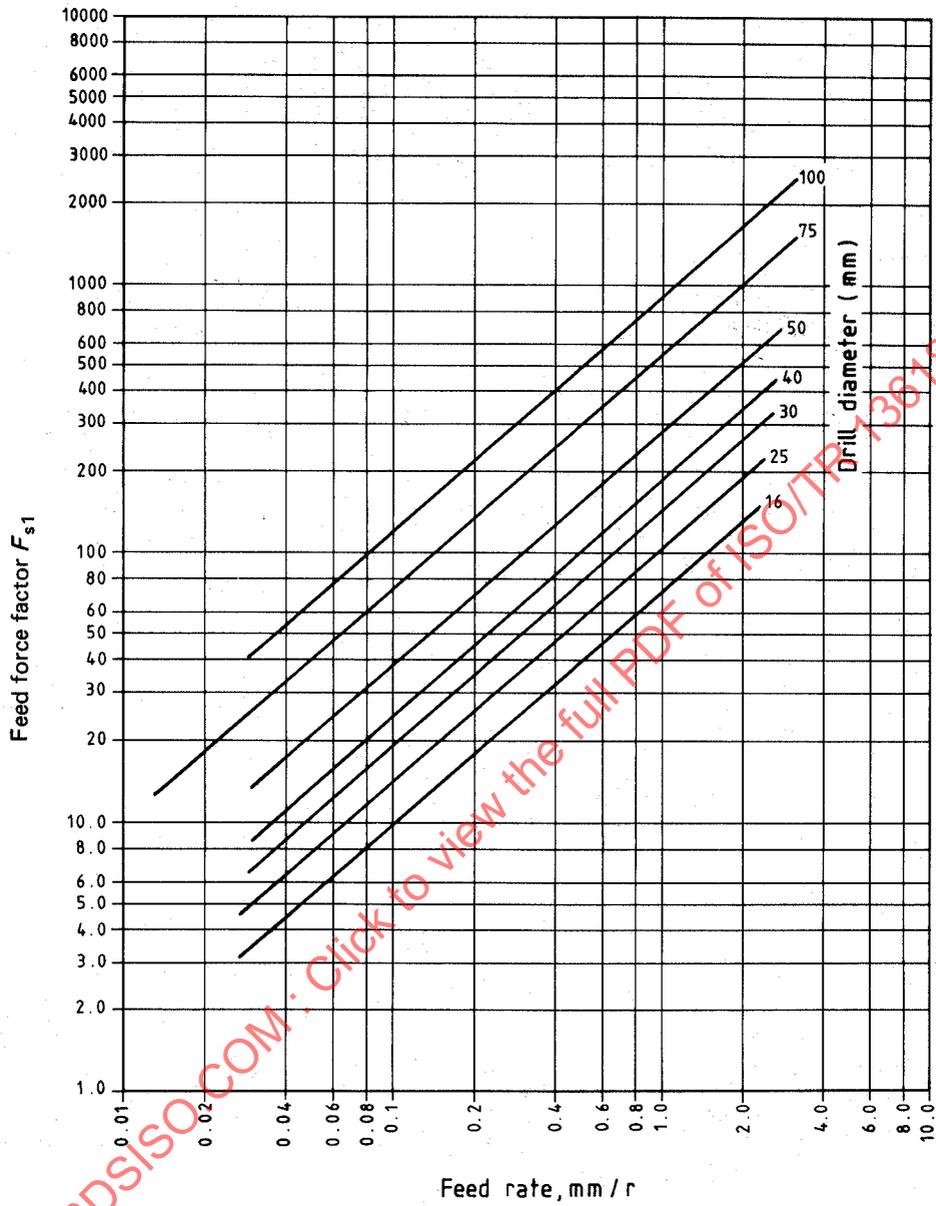


Figure 6. Feed rate v. feed force factor  $F_{s1}$  (for drills 16 mm diameter and larger, in steel and cast iron)

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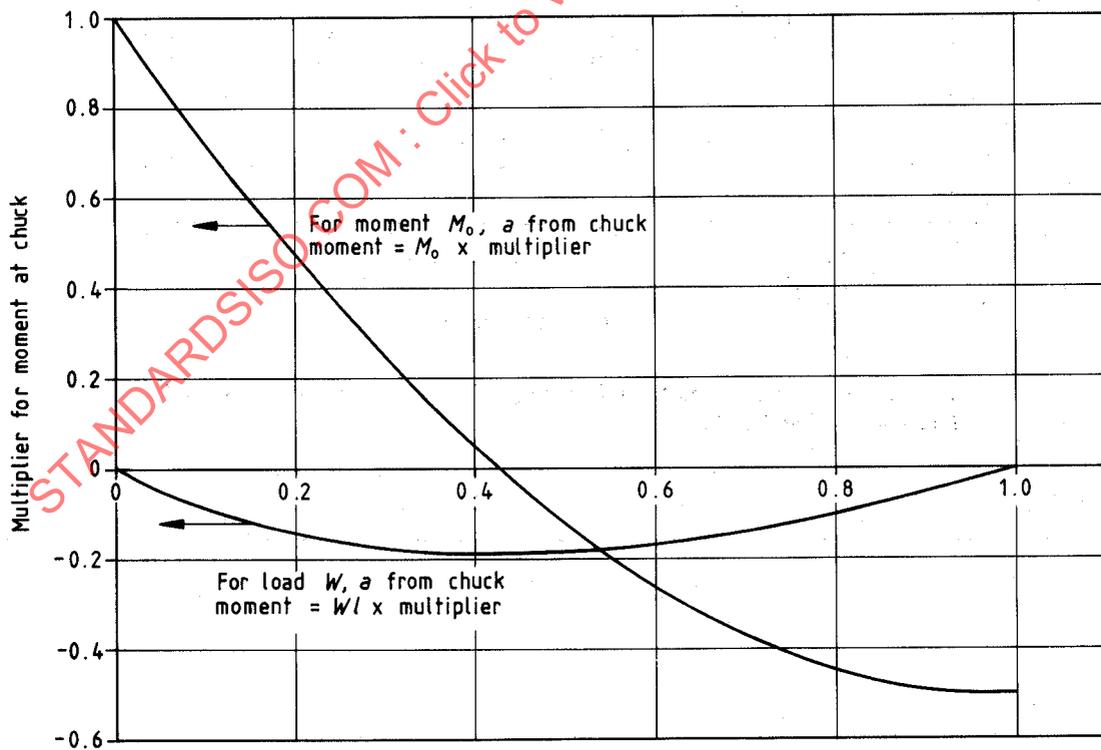
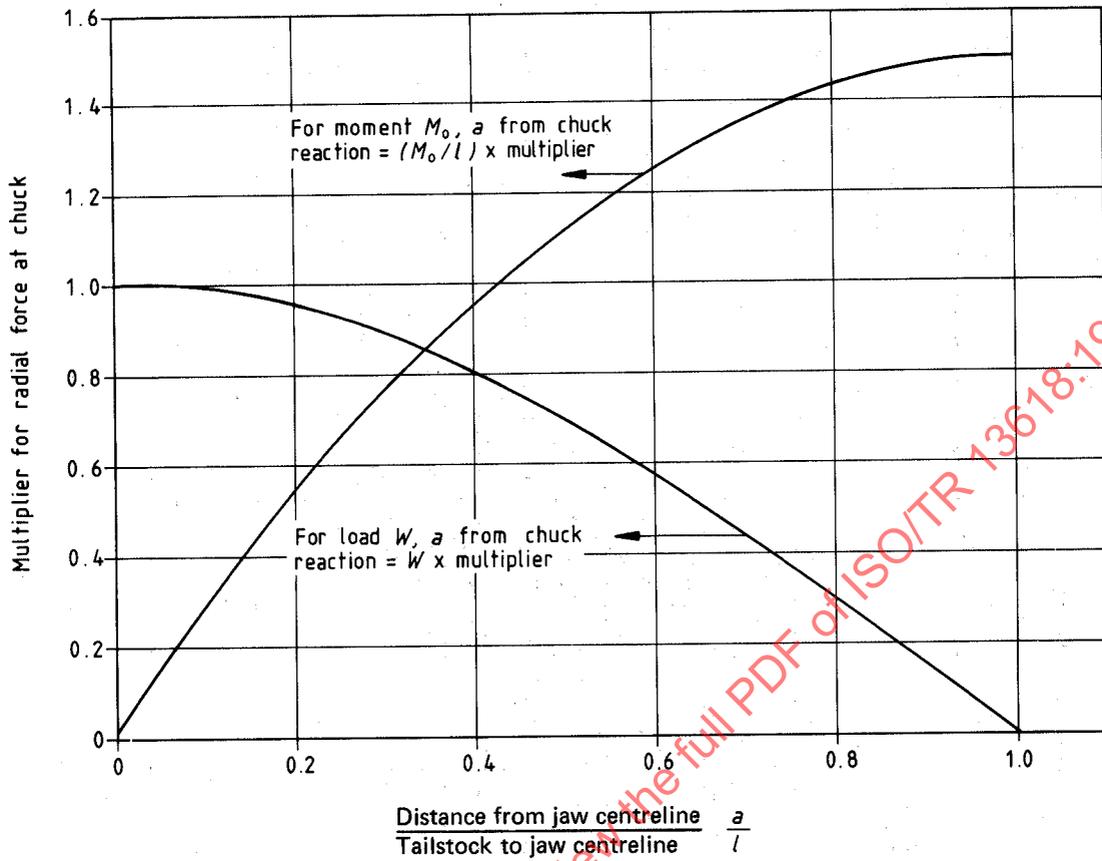


Figure 7. Multiplying factors when tailstock centre is used

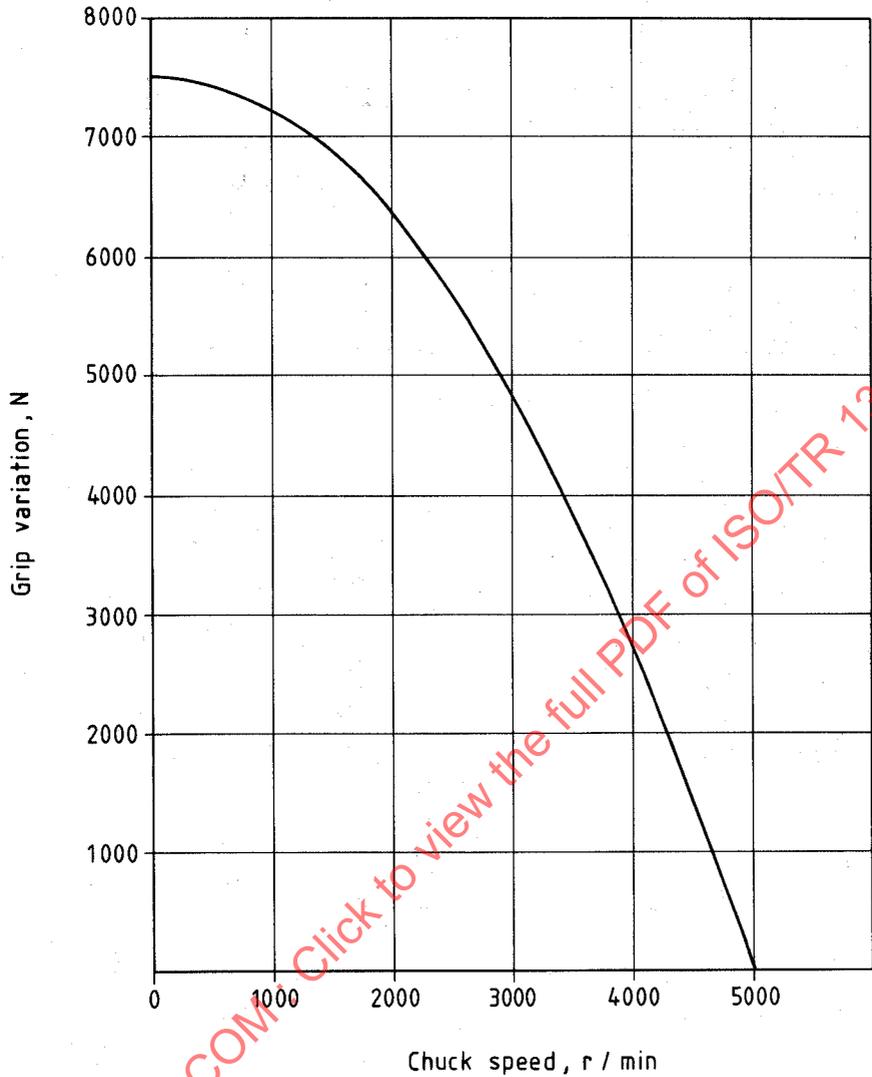


Figure 8. Typical curve showing change of grip with speed

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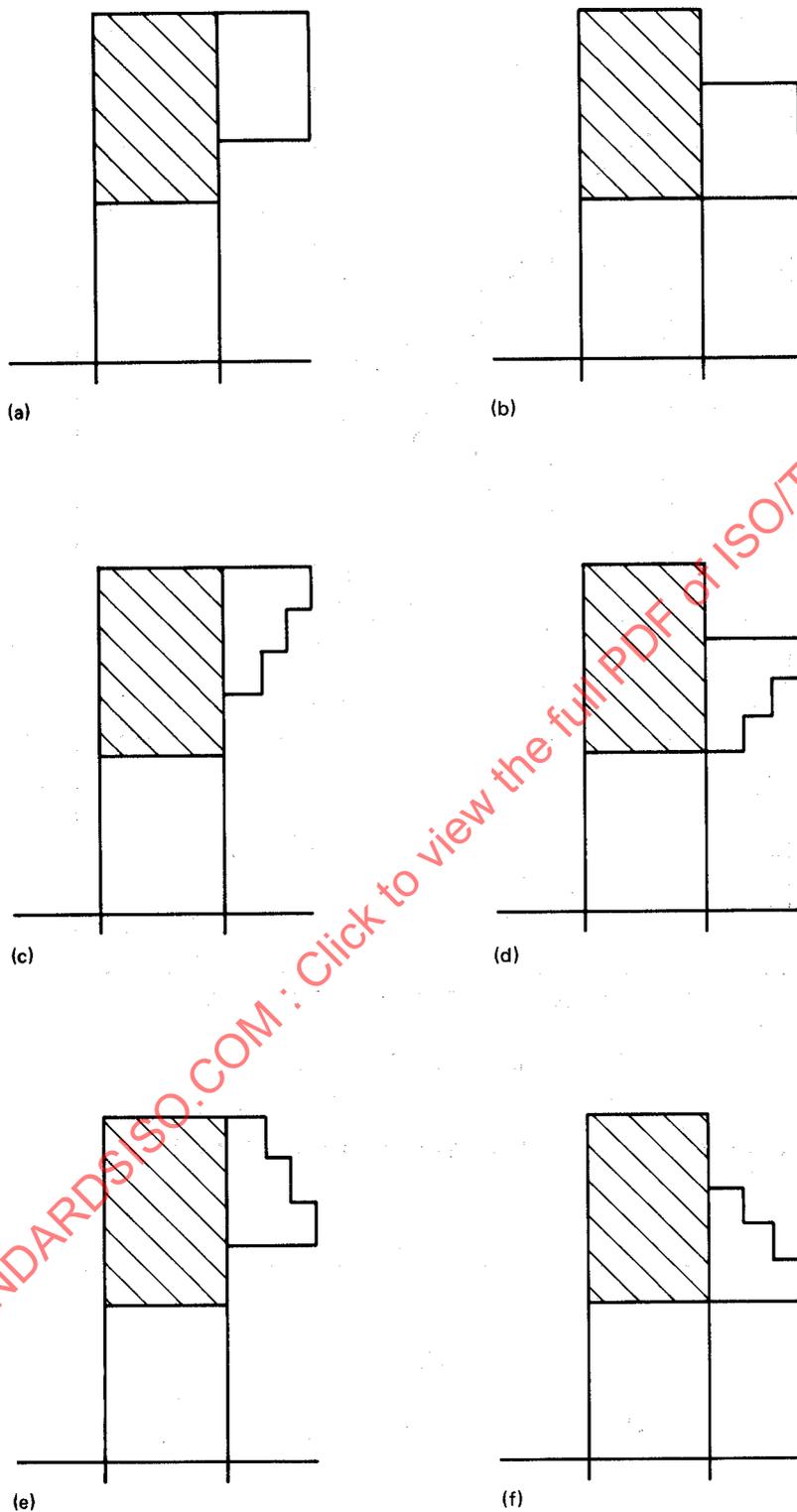


Figure 9. 'Standard' jaw positions

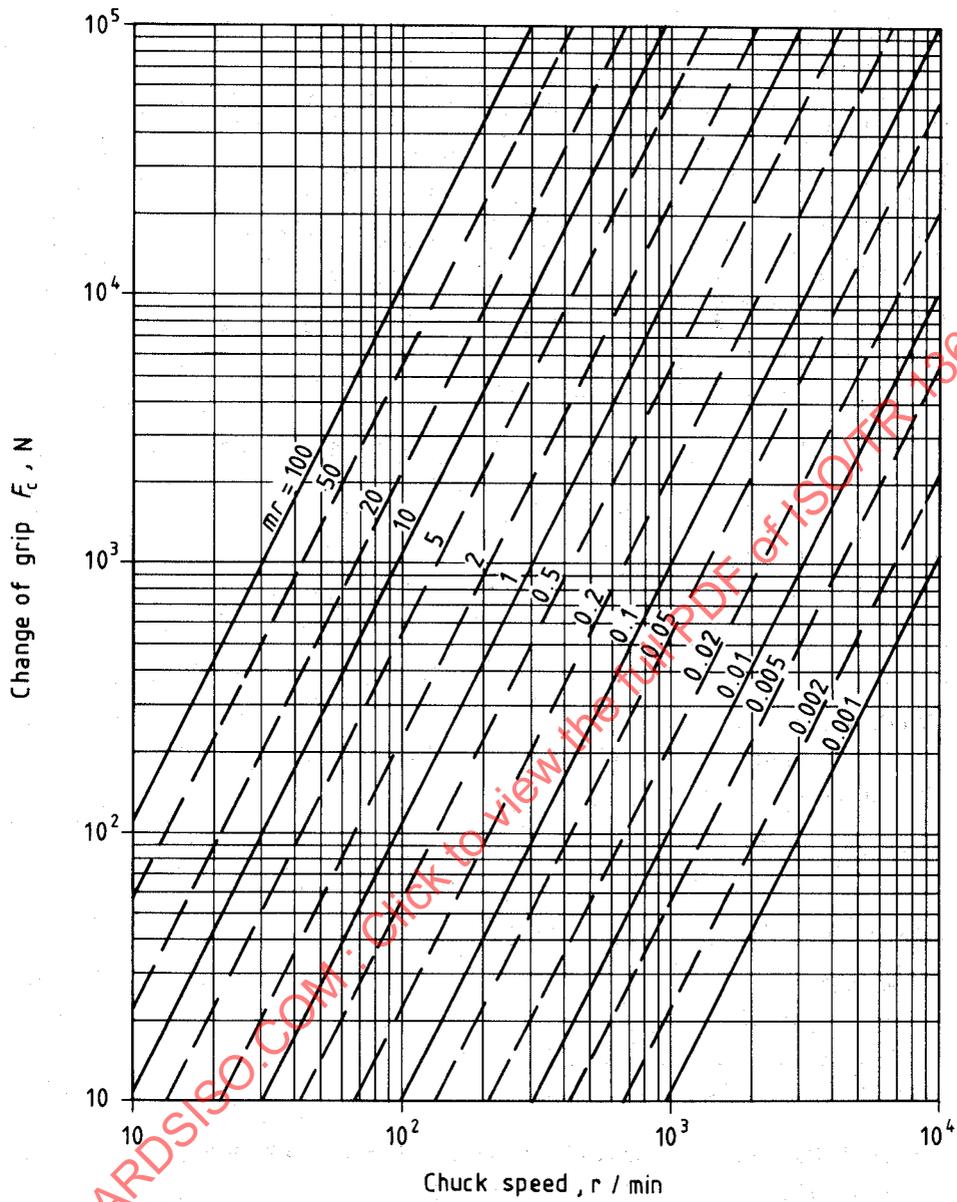
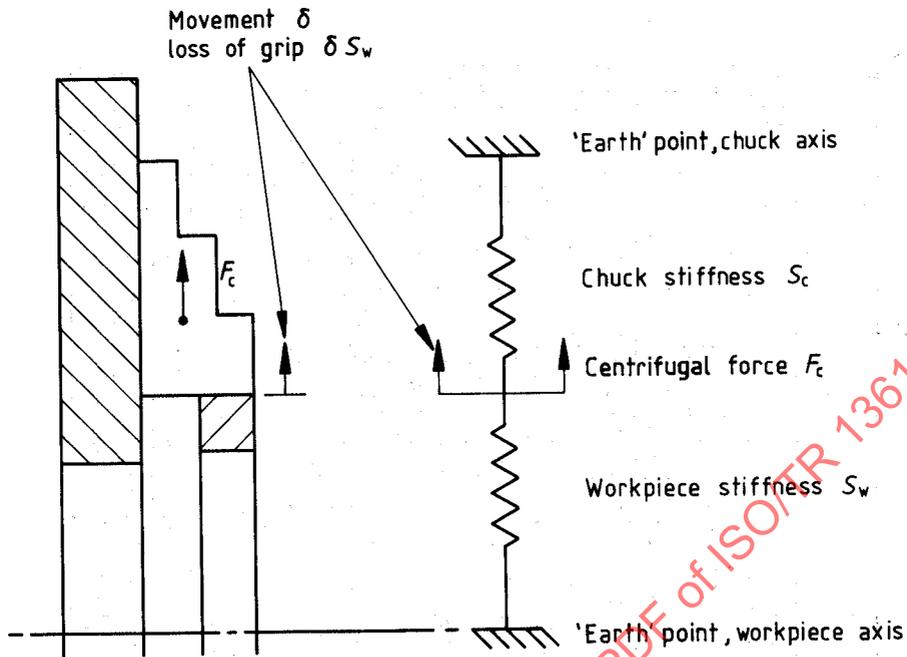


Figure 10. Calculation of change of grip at speed



**Chuck section**

Stiffness is defined as total jaw load/change of diameter

Overall stiffness  $S_o = S_c + S_w$

Deflection  $\delta = \frac{F_c}{S_o} = \frac{F_c}{S_c + S_w}$

Loss of grip  $= \delta S_w = \frac{S_w F_c}{S_c + S_w} = \frac{F_c}{1 + (S_c/S_w)}$

Showing that:

- (a) for rigid workpiece ( $S_w = \infty$ ), loss of grip  $= F_c$ ;
- (b) for very flexible workpiece ( $S_w = 0$ ) loss of grip  $= 0$ ;
- (c) for  $S_w = S_c$ , loss of grip  $= F_c/2$ .

**Figure 11. Calculation of change of grip at speed: flexible workpieces**

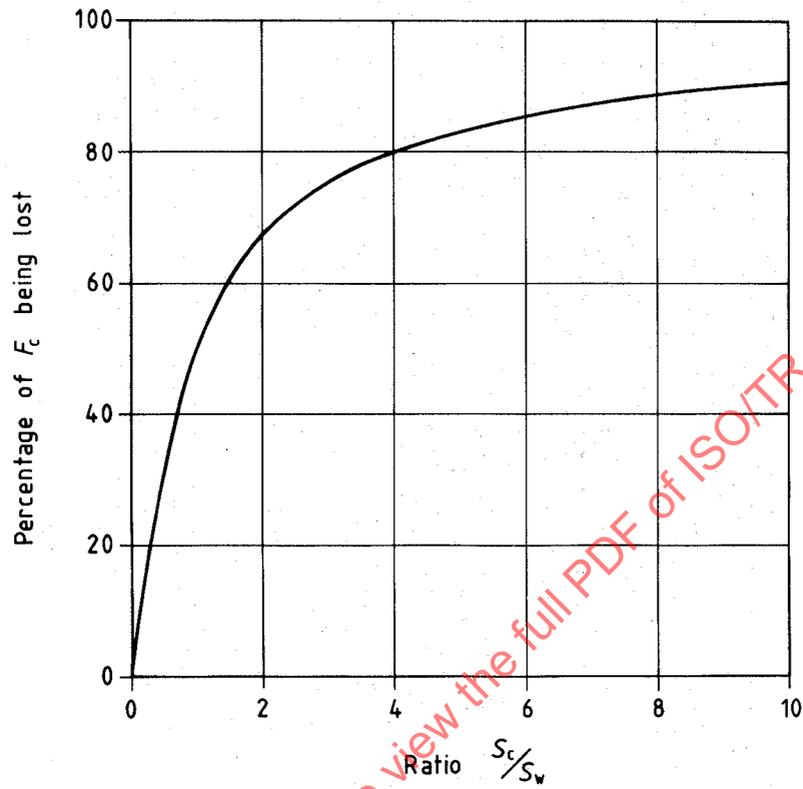


Figure 12. Graph of change of grip v. stiffness ratio for a flexible workpiece

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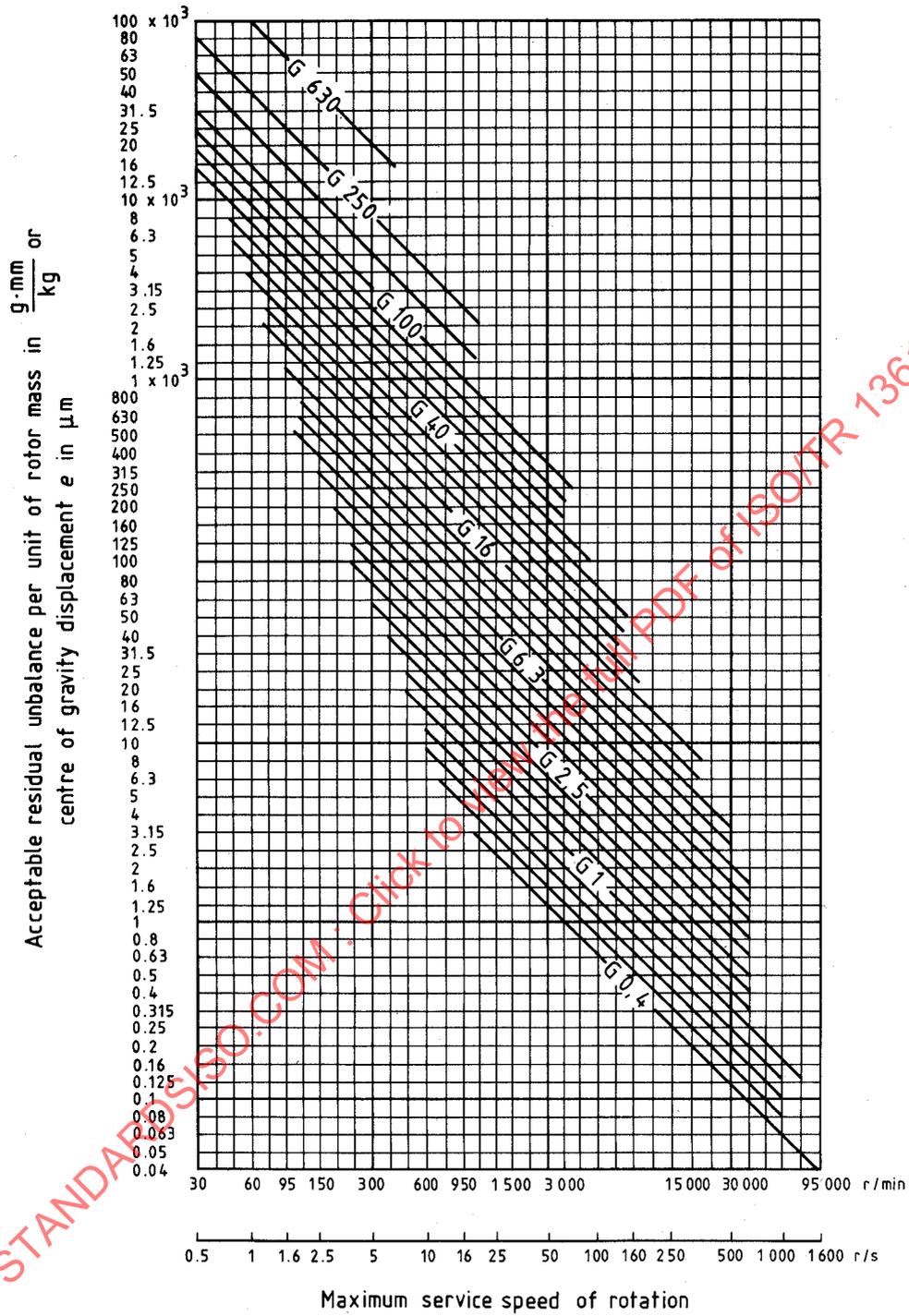


Figure 13. Maximum residual specific unbalance (see ISO 1940-1)

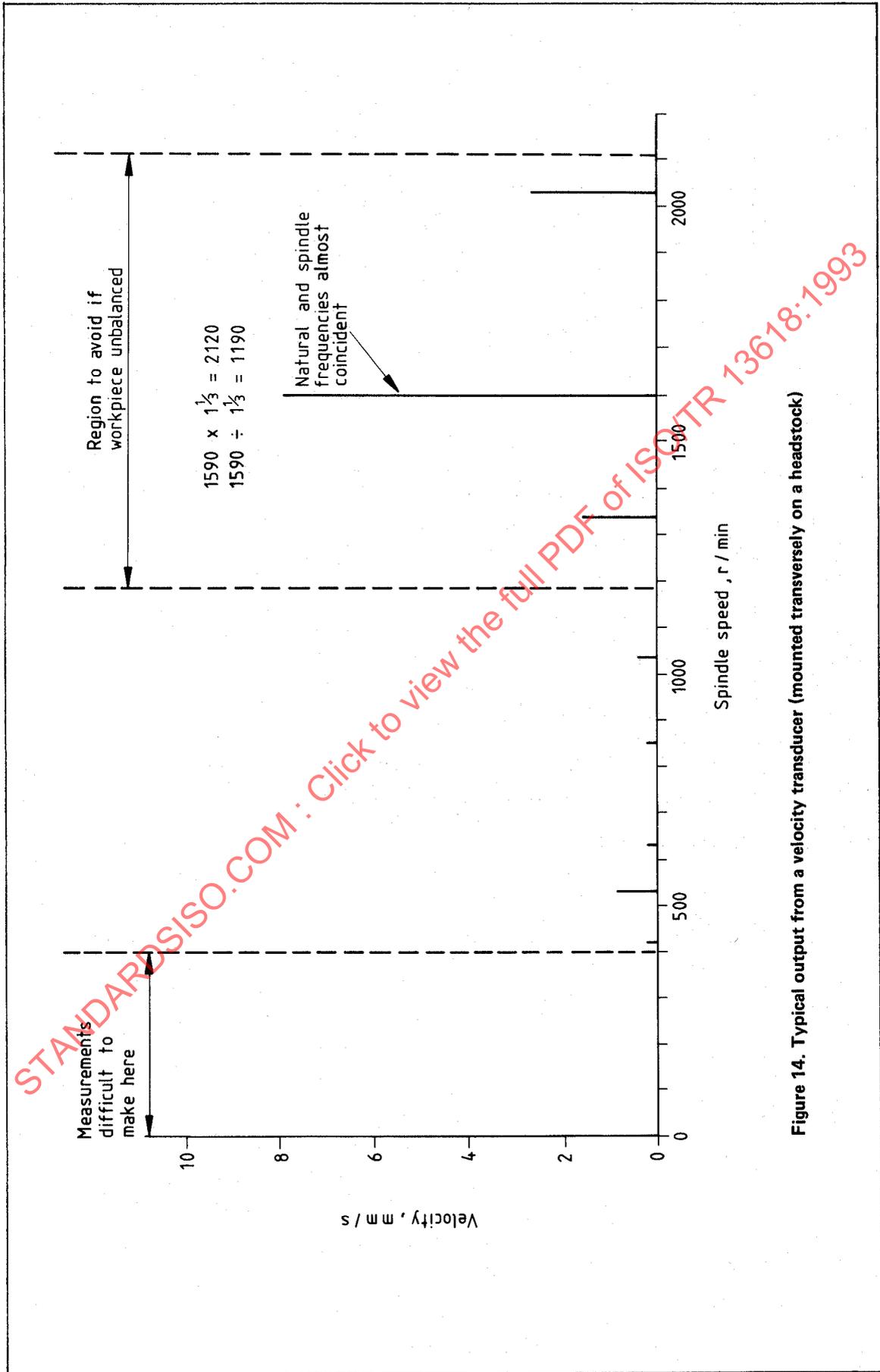
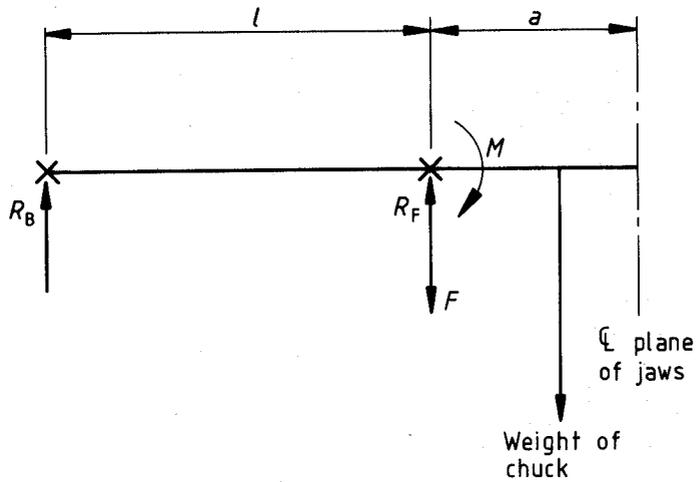


Figure 14. Typical output from a velocity transducer (mounted transversely on a headstock)



Evaluate load  $F$  and moment  $M$  at front bearing as in clause 2, but with all relevant overhang lengths increased by  $a$  and with the weight of the chuck included.

Then, at the front bearing, worst case assumed,

Shear force,  $S_F = F$  and  $(M/1)$  behind and beyond the reaction point respectively

Reaction,  $R_F = F + (M/1)$

Moment,  $M_F = M$

Axial thrust,  $R_a = \Sigma F_{ax}$

If  $F$  and  $M$  are in very different planes their effect can be evaluated separately and the reaction  $R_F$  obtained by adding  $F$  and  $M/1$  vectorially.

In the case of vertical spindles the weight of the chuck is added to the axial force instead.

NOTE.  $S_F \leq S_{Fperm}$  according to the shear stress in the spindle  
 $R_F \leq R_{Fperm}$  according to the radial load rating of the front bearing  
 $R_a \leq R_{aperm}$  according to the axial load rating of the thrust bearing  
 $M_F \leq M_{Fperm}$  according to the bending stress level in the spindle.

Figure 15. Loads applied to spindle

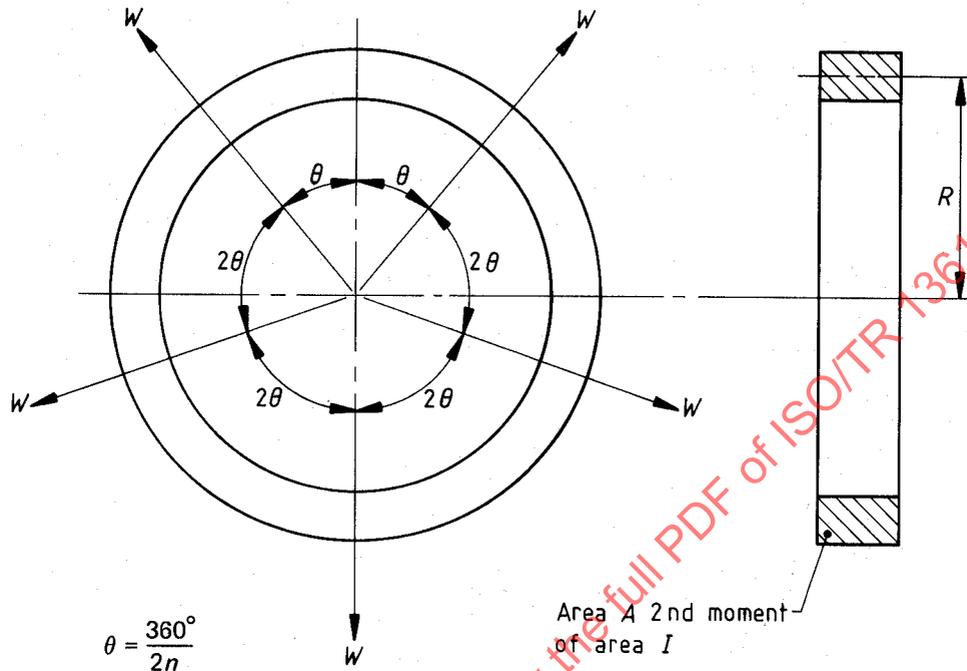
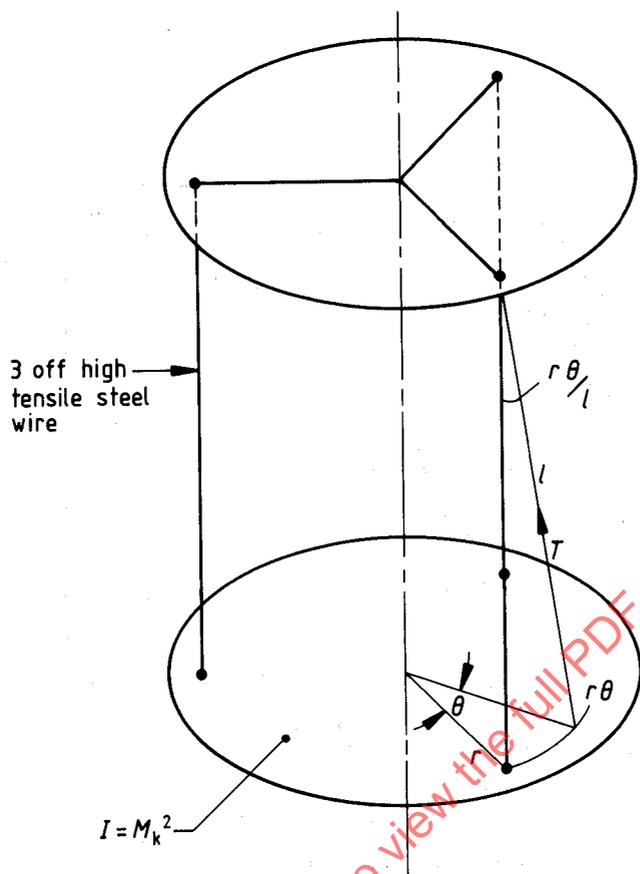


Figure 16. Clamping on to a thin ring: multiple jaws

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Equating the restoring and inertia torques

$$3T \sin\left(\frac{r\theta}{l}\right) r = -I\ddot{\theta}$$

where  $T = \frac{Mg}{3}$

$$\ddot{\theta} = \frac{-gr^2\theta}{lk^2} = -\omega^2\theta$$

$$\therefore \omega = \frac{r}{k} \sqrt{g/l}$$

Figure 17. Trifilar suspension

## Appendix D. Worked example

NOTE. This example has been chosen so as to illustrate the calculations, not to recommend practice.

### D.1 Data

Consider a capstan lathe fitted with a 3-jaw power-operated chuck producing a component as shown in figure 18.

#### (a) General data

Lathe:

maximum spindle speed	3500 r/min
main motor power	25 kW

Chuck: 3-jaw, lever type, cast iron body:

outside diameter	300 mm
------------------	--------

Workpiece blank: steel black bar:

diameter	150 mm
length	200 mm

1st operation: spindle speed

150 r/min

Turn outside diameter to shoulder centre and drill bore (50 mm diameter)

Face end

2nd operation: spindle speed

150 r/min

Turn outside diameter of flange

Face flange (soft jaws to be used)

spindle speed	1500 r/min
---------------	------------

drill bore (5 mm diameter)	
----------------------------	--

#### (b) Data provided by the lathe manufacturer

Spindle balanced to ISO 1940-1 (see 4.1)

Grade 2,5

Spindle nose runout (see 4.1):

radial (FIM)*	10 µm
---------------	-------

wobble at flange (FIM)*	10 µm
-------------------------	-------

Typical natural frequency (rocking mode) (see 4.4) on recommended foundations  
(confirmation on installation)

16.7 Hz

16.7 Hz

Velocity transducer and vibration read-out not fitted.

Permitted mass  $\times$  eccentricity at spindle nose v. speed

see figure 19

Maximum inertia of chuck and workpiece (see clause 5)

see figure 20

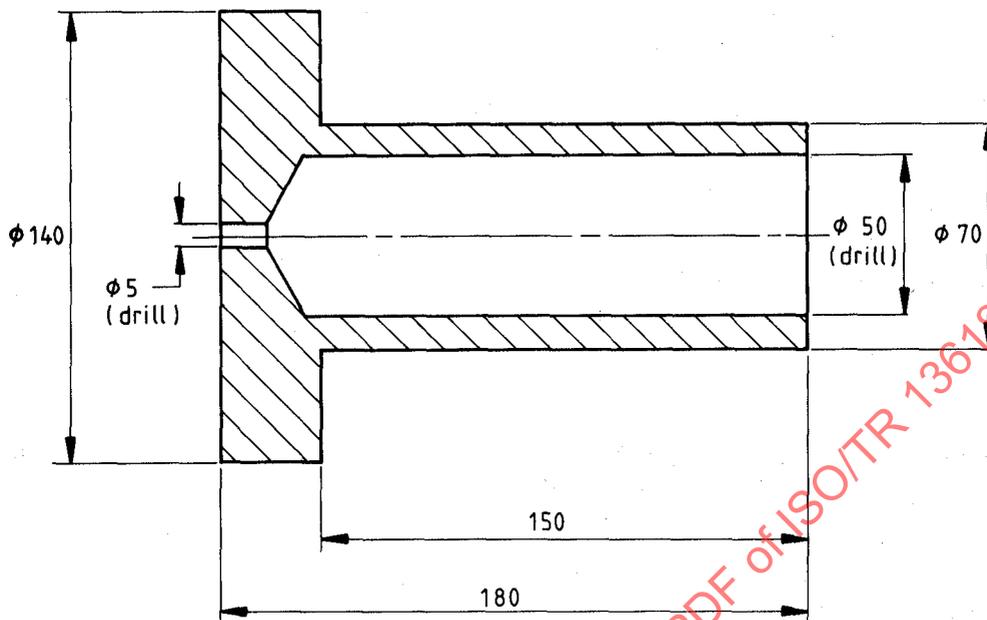
Permitted external load on spindle bearing (see clause 6)

see figure 21

Chuck diameter/spindle speed interlocking available (see 7.2)

Forward/reverse clutches slip at 200 % overload when adjusted according to the instructions  
in the service manual (see 7.7)

\*FIM = Full indicator movement.



Material : steel

All dimensions are in millimetres.

Figure 18. Sketch of component

(c) Data provided by the chuck manufacturer

Standard hard jaws (see 2.3):

mass per set  
centre of mass  
grip v. speed

2.82 kg  
see figure 22  
see figure 23

Total static grip v. drawbar pull (see 2.4) (serviced according to instruction manual)

see figure 24

Change in diameter v. total grip (see 2.5)

see figure 25

Maximum recommended speed of chuck body (see clause 3)

2500 r/min

Chuck balanced to ISO 1940-1 (see clause 4)

Grade 2,5

Inertia of chuck (fitted with standard jaws at maximum diameter) (see clause 5)

0.508 kg·m<sup>2</sup>

Mass of chuck (fitted with standard jaws) (see clause 6)

30 kg

Centre of mass

see figure 22

No overspeed devices fitted (see 7.2).

Tightening torque:

Chuck mounting bolts (see 7.4)  
Jaw mounting bolts (see 7.5)

60 N·m  
20 N·m

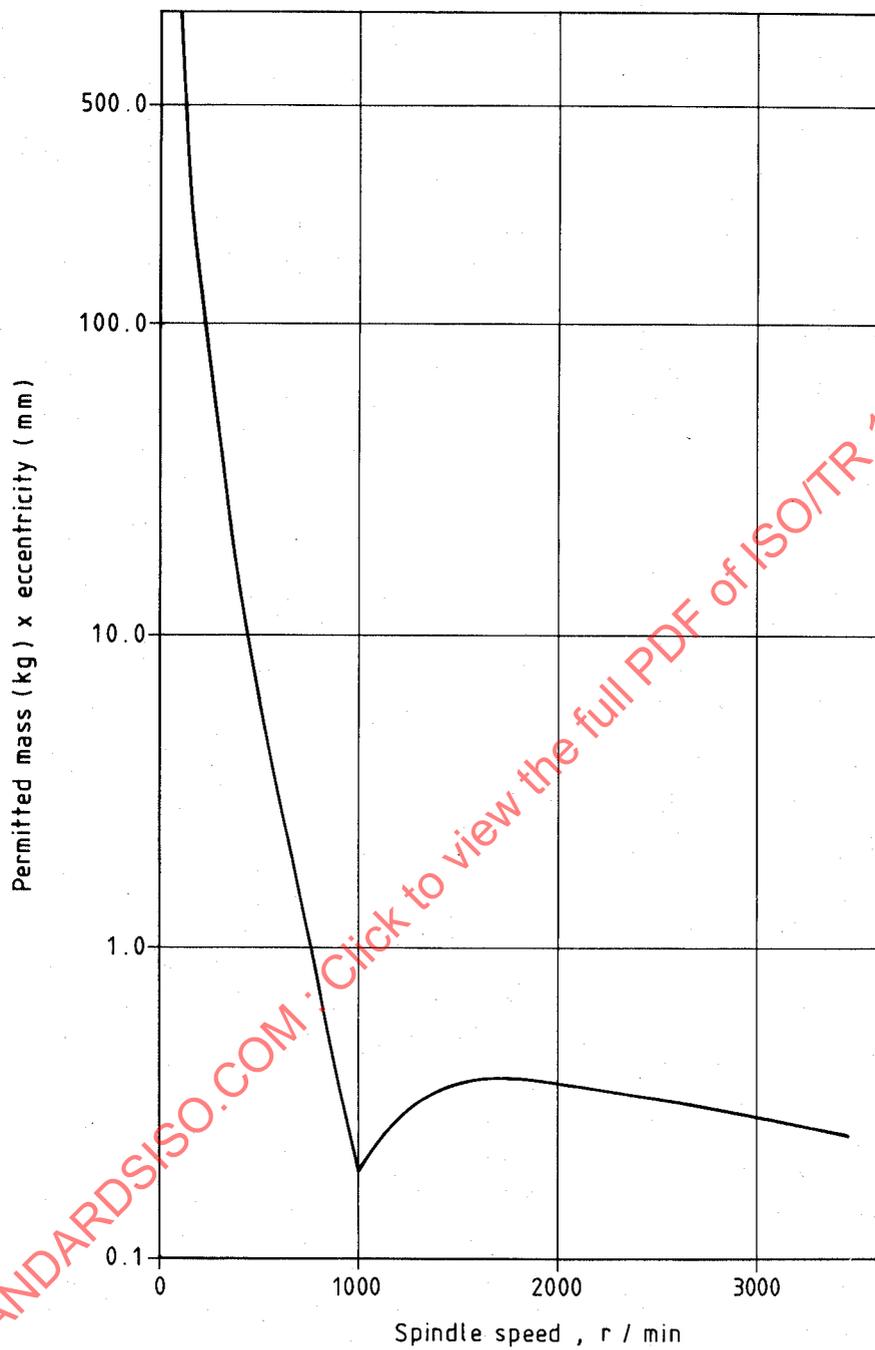
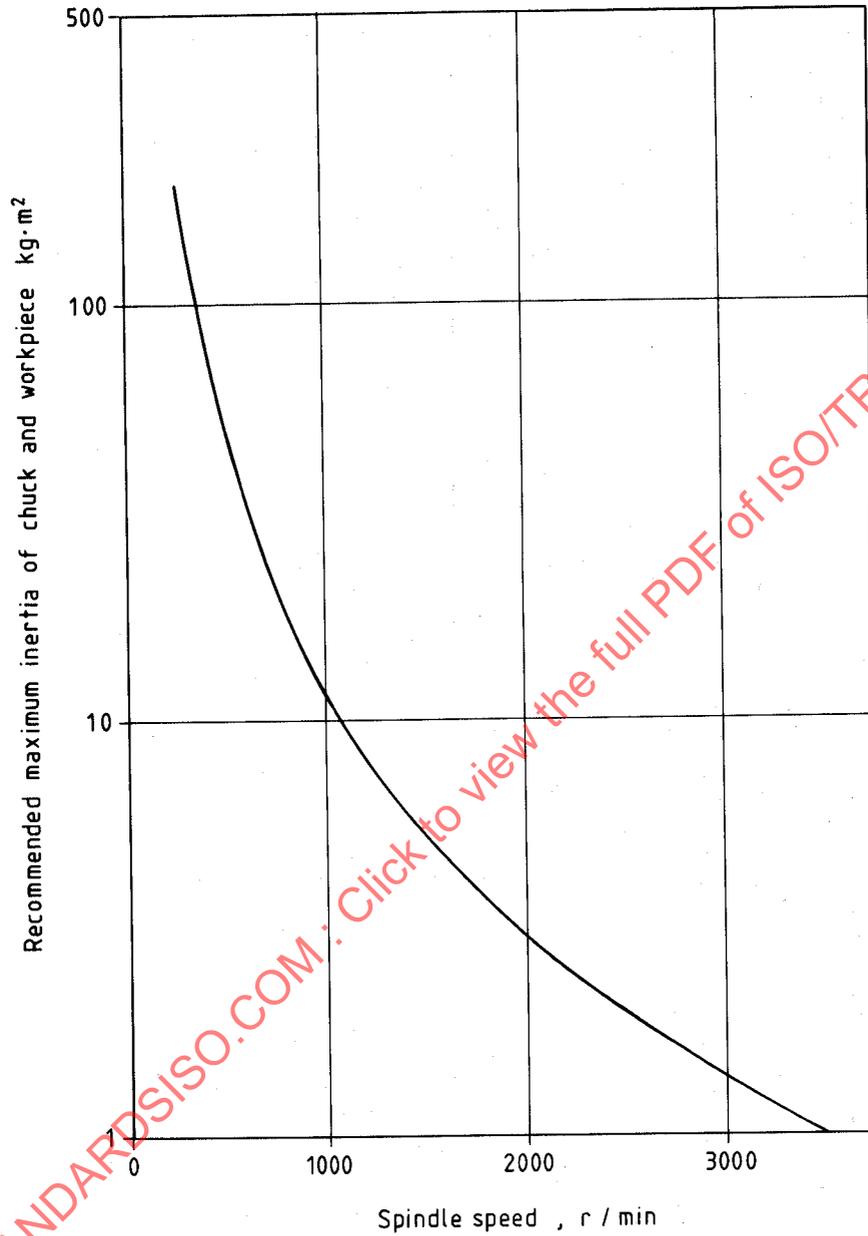
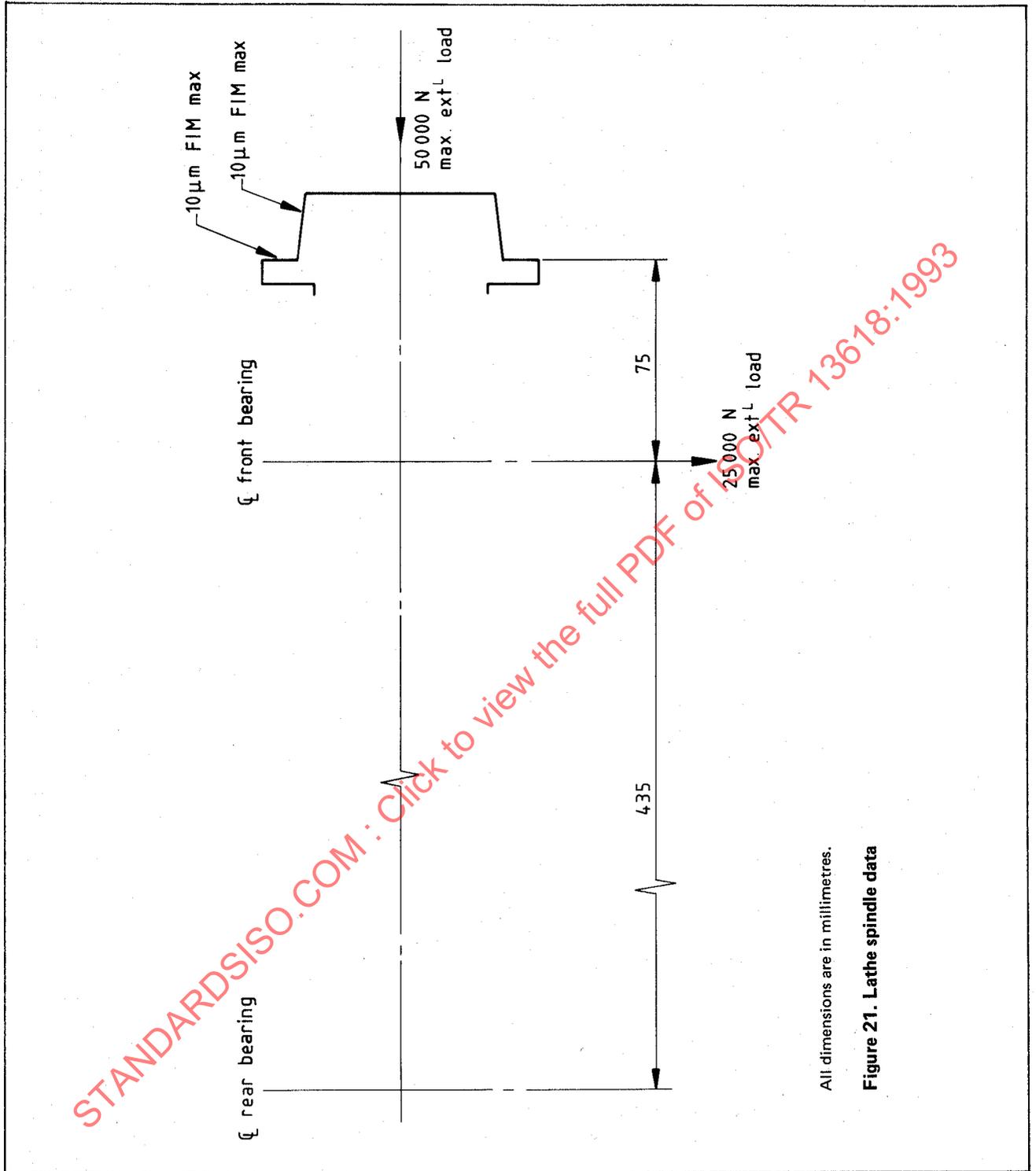


Figure 19. Permitted mass  $\times$  eccentricity v. spindle speed



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Figure 20. Inertia load permitted



All dimensions are in millimetres.

Figure 21. Lathe spindle data