Dimensions of Cardan Shafts

The dimensions of the shaft depend on many factors. The rules below will give an approximate selection. In borderline cases, please consult us. The questionnaires in chapter 12 will help you. We would be pleased to give you advice.

8.1 Selection of Joint Size for Stationary Drives

The part of the propeller shaft which determines its useful life is normally the joint bearing. So the joint size should preferably be determined from the transferable torque of the bearing. The calculation below is based on the standard roller bearing calculation, where the oscillating movement is regarded as replaced by a rotational one.

The dimension for the transferability of the bearing is the joint load rating $T = C \cdot R$, where C is the dynamic transfer capacity of the bearing and R the distance of the bearing centre from the joint centre. The joint load rating is given in the data sheet for the shaft. T_{erf} can be determined using the same equation. It applies to uniform operation, i.e. when the torque M_d occurs throughout life L_h at rotation speed n and deflection angle ß.

$$T_{erf} = M \cdot K \cdot \frac{1}{2 \cdot \cos\beta} \cdot \left(\frac{Lh_{erf} \cdot n \cdot \beta}{46{,}8 \cdot 16667} \right)^{0{,}3} \text{Nm}$$

T_{erf} = necessary joint load rating in Nm
K = shock factor (see table)
ß = deflection angle of joint in ° (degrees). For angle < 3°, ß = 3° must always be used.
M = torque to be transferred in Nm
Lh_{erf} = necessary (required) life in h. This L_h at least is achieved by 90% of all shafts. The average L_h of all shafts is then 5 times as high.
n = rotation speed of shaft in rpm

Shock Factors

Drive Unit	K with rubber coupling	K without rubber coupling
Elec. motors	1	1
Motors with converter	1	1
Diesel engine 1-3 cylinders	2	2,5
4 or more cylinders	1,5	2,0
Petrol engine 1-3 cylinders	1,5	2,0
4 and more cylinders	1,25	1,75
Compressors 1-3 cylinders	1,25	1,75
4 and more cylinders	1,15	1,5

Example:

A working machine with a small mass moment of inertia, which assumes a torque of 1000 Nm at n = 1 450 rpm, should be driven by an electric motor via a shaft running under a deflection angle of 7°. The life should be 2000 h. What joint size is required?

Solution:

Electric motor and impact-free working machine gives an impact factor of 1.0. Then:

$$T_{erf} = 1000 \cdot 1 \cdot \frac{1}{2 \cdot \cos 7°} \cdot \left(\frac{2000 \cdot 1450 \cdot 7}{46,8 \cdot 16667} \right)^{0,3} = 1339 \text{ Nm}$$

So T_{erf} is found to be 1339 Nm. From the data sheet, we now select the shaft with the next highest value. If we are to use a shaft of design 008 for example, the type and joint size 008 195 are selected with a joint transfer capacity of 1460 Nm.

For the joint found, we now check that $\frac{M \cdot K}{\cos \beta} \leqq M_{max}$

1000 Nm · 1,0 < 1460 Nm · cos 7° = 1449,1 Nm.

The condition is fulfilled, and the shaft can be used. It will achieve a life of:

$$L_h = L_{h\,erf} \cdot \left(\frac{T}{T_{erf}} \right)^{3,33} = 2000 \text{ h} \cdot \left(\frac{1460 \text{ Nm}}{1339 \text{ Nm}} \right)^{3,33} = 2667 \text{ h}$$

In many applications, in particular in vehicles, the moment, the rotation speed and/or the deflection angle are not constant. We must then try to form classes to which moment, rotation speed and deflection angle can be allocated and determine their time proportions.

For an initial estimated joint size, the individual life can then be assessed for each class:

$$L_{hn} = \left(\frac{2 \cdot T_{vorh} \cdot \cos\beta_n}{M_n \cdot K} \right)^{3,33} \cdot \frac{16667 \cdot 46,8}{n_n \cdot \beta} \text{ h}$$

Where:

L_{hn} = individual life of class n, where n = 1,2,3...n
M_n = the moment allocated to class n
T_{vorh} = joint power factor of estimated joint size
n_n = rotation speed allocated to class n
$ß_n$ = deflection angle allocated to class n

See above for other symbols.

From the individual life, the total life can be determined as follows:

$$L = \frac{100\%}{\frac{q_1}{L_{h1}} + \frac{q_2}{L_{h2}} + \ldots + \frac{q_n}{L_{hn}}}$$

where:
q = time proportion in %
$L_{h1}...L_{hn}$ individual life in h.

8.2 Selection of Joint Sizes for Vehicle Drives

In this section, the following symbols are used:

M_{FG}	= function torque capacity (from data sheet)
M_X	= general dimensioning moment for a propeller shaft
M_A, M_B, M_C	= dimensioning moment for propeller shafts A, B, C
$M_{mot.}$	= general proportional engine torque on propeller shaft
$M_{mot\ max}$	= max. engine torque
$M_{Rad\ x}$	= general proportional wheel adhesion torque at propeller shaft
s	= joint bearing safety factor = 1,5 < s <2,0
k	= shock factor (see table above)
μ_R	= tyre coefficient of friction = 0,6 < μ < 1,0
η	= general gear efficiency
η_G	= efficiency of engine gear
η_V	= efficiency of transfer box
η_A	= efficiency of final drive
i_W	= theoretical value for converter ratio
i_{WF}	= converter brake conversion
$i_{G\ max}$	= max. engine gear ratio (1st gear)
$i_{G\ min}$	= min. engine gear ratio (1st gear)
$i_{V\ max}$	= transfer box ratio (1st gear)
$i_{V\ min}$	= transfer box ratio (nth gear)
i_A	= final drive ratio
V	= engine torque distribution ratio $T_{mot\ V} / T_{mot\ H}$
R_{dyn}	= dynamic rolling radius of tyre
G_V	= front axle load; total front axle load
G_{V1}	= front axle load, 1st axle
G_{V2}	= front axle load, 2nd axle
G_H	= rear axle load; total rear axle load
G_{H1}	= rear axle load, 1st axle
G_{H2}	= rear axle load, 2nd axle

The function torque capacity M_{FG} of the propeller shafts is given in the data sheets in this catalogue. This moment can be transferred by the propeller shaft for short periods at limited load frequency with 0° joint deflection angle.

With a joint deflection angle of ß°, the function limit moment is reduced by the factor cos ß°.

The function torque capacity M_{FG} must be sufficiently larger than the dimensioning moment M_x.

$$M_{FG} \geqq 1{,}5 \cdot M_x$$

The dimensioning moments M_x for the propeller shafts between the engine and the final drive are calculated approximately from the moments of the torque M_{motx} exerted by the engine and the adhesion moment M_{radx} exerted by the wheel, as follows:

$$M_x = ½ (M_{motx} + M_{radx})$$

For propeller shafts A between the engine and the gearbox, the influence of the high rotation speed part and the engine shock factor must be taken into account.

If a converter is fitted, some special features should be observed:

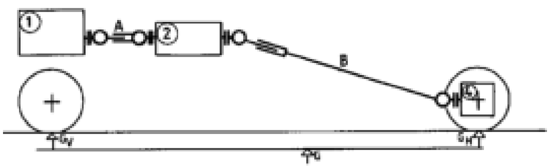
If the propeller shaft is installed between the engine with converter and the gearbox, the impact factor s = 1 must be used. If the propeller shaft is between the engine and gearbox with a converter in front, the effect of the wheel moment = 0.

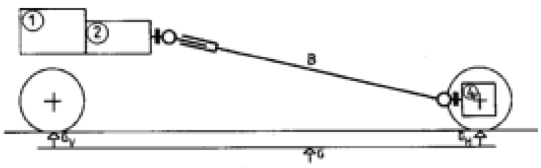
If the brake conversion $i_{WF} < 1{,}4$, its influence can be ignored, so $i_W = 1$.

If the brake conversion $i_{WF} > 1{,}4$, its influence must be allowed for by a factor of 0.76, so $i_W = 0{,}76 \cdot i_{WF}$.

8.3 Selection System for Propeller Shafts in Vehicles for Normal Use

Road Vehicle 4 x 2





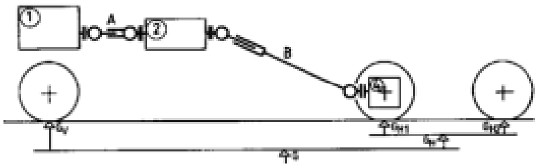
Selection torque for propeller shaft A between engine 1 and gearbox 2.

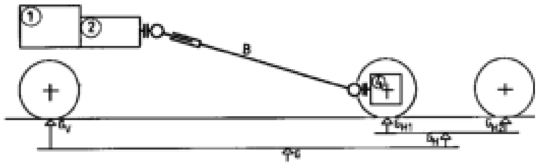
$$M_A = (M_{mot\,max} \cdot i_W \cdot s \cdot k \cdot + G_H \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{Gmin}} \cdot \eta_G \cdot \eta_A) \cdot \frac{1}{2}$$

Selection torque for propeller shaft or multiple joint shaft B between gearbox 2 and differential 4.

$$M_B = (M_{mot\,max} \cdot i_W \cdot i_{G\,max} \cdot \eta_G + G_H \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A) \cdot \frac{1}{2}$$

Road Vehicle 6 x 2





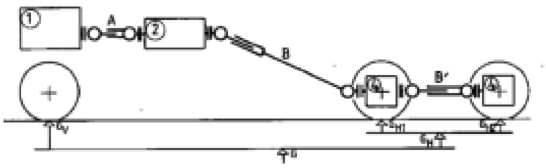
Selection torque for cardan shaft A between engine 1 and gearbox 2.

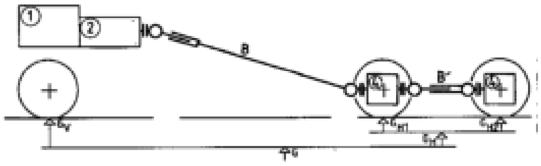
$$M_A = (M_{mot\,max} \cdot i_w \cdot s \cdot k \ + G_{H1} \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{Gmin}} \cdot \eta_G \cdot \eta_A) \cdot \frac{1}{2}$$

Selection torque for propeller shaft or multiple joint shaft B between gearbox 2 and differential 4.

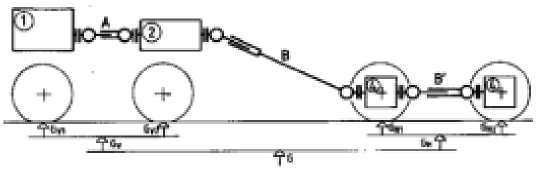
$$M_B = (M_{mot\,max} \cdot i_W \cdot i_{G\,max} \cdot \eta_G + G_{H1} \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A) \cdot \frac{1}{2}$$

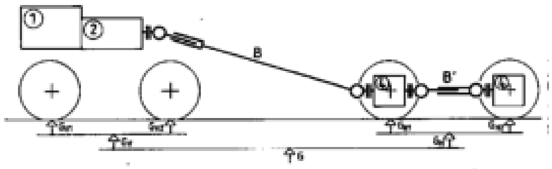
Road Vehicle 6 x 4





and Road Vehicle 8 x 4





Selection torque M_A for propeller shaft A between engine 1 and gearbox 2

$$M_A = \left(M_{mot\,max}\, i_w \cdot s \cdot k + G_H \cdot R_{yn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{G\,min}} \cdot \eta_G \cdot \eta_A\right) \cdot \frac{1}{2}$$

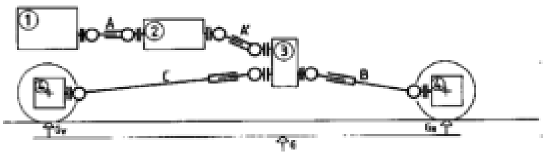
Selection torque M_B for propeller shaft or multiple joint shaft B between gearbox 2 and differential 4

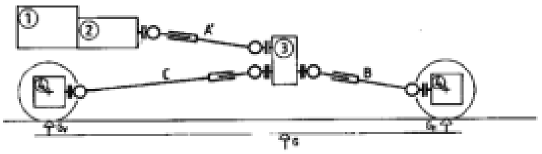
$$M_B = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot \eta_G + G_H \cdot R_{dyn} \cdot \mu_R \frac{1}{i_A} \cdot \eta_A\right) \cdot \frac{1}{2}$$

Selection torque for $M_{B'}$ for propeller shaft B' between differential gears 4

$$M_{B'} = \left(M_{mot\,max}\, i_w \cdot i_{G\,max} \cdot \eta_G + G_{H2} \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A\right) \cdot \frac{1}{2}$$

All-Wheel Drive 4 x 4





Selection torque M_A for propeller shaft A between engine 1 and gearbox 2

$$M_A = \left(M_{mot\,max} \cdot i_w \cdot s \cdot k + G \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{G\,min} \cdot i_V} \cdot \eta_A \cdot \eta_G \cdot \eta_V \right) \cdot \frac{1}{2}$$

Selection torque $M_{A'}$ for propeller shaft A' between gearbox 2 and transfer box 3

$$M_{A'} = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot \eta_G + G \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{V_{min}}} \cdot \eta_A \cdot \eta_V \right) \cdot \frac{1}{2}$$

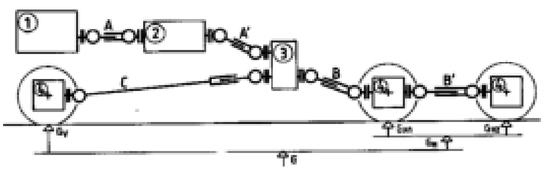
Selection torque M_B for propeller shaft or multiple joint shaft B between transfer box 3 and differential gears 4

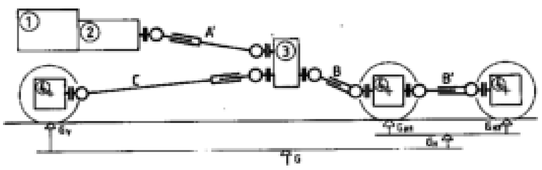
$$M_B = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot i_{V_{max}} \cdot \eta_G \cdot \eta_V \cdot \frac{V}{1+V} + G_H \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A \right) \cdot \frac{1}{2}$$

Selection torque M_C for propeller shaft C between transfer box 3 and differential gears 4

$$M_C = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot i_{V_{max}} \cdot \eta_G \cdot \eta_V \cdot \frac{1}{1+V} + G_V \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A \right) \cdot \frac{1}{2}$$

All-Wheel Drive 6 x 6





Selection torque M_A for propeller shaft A between engine 1 and gearbox 2

$$M_A = \left(M_{mot\,max} \cdot i_w \cdot s \cdot k + G \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{G_{min}} \cdot i_{V_{min}}} \cdot \eta_A \cdot \eta_G \cdot \eta_V \right) \cdot \frac{1}{2}$$

Selection torque $M_{A'}$ for propeller shaft A' between gearbox 2 and transfer box 3

$$M_{A'} = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot \eta_G \cdot + G \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A \cdot i_{V_{min}}} \cdot \eta_A \cdot \eta_V \right) \cdot \frac{1}{2}$$

Selection torque M_B for propeller shaft or multiple joint shaft B between transfer box 3 and differential gears 4

$$M_B = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot i_{V_{max}} \cdot \eta_G \cdot \eta_V \cdot \frac{V}{1+V} + G_H \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A \right) \cdot \frac{1}{2}$$

Selection torque $M_{B'}$ for propeller shaft B' between differential gears 4

$$M_{B'} = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot i_{V_{max}} \cdot \eta_G \cdot \eta_V \cdot \frac{V}{1+V} \cdot \frac{1}{2} + G_{H2} \cdot R_{dyn} \cdot \mu_R \cdot \frac{1}{i_A} \cdot \eta_A \right) \cdot \frac{1}{2}$$

Selection torque M_C for propeller shaft C between transfer box 3 and differential gears 4

$$M_C = \left(M_{mot\,max} \cdot i_w \cdot i_{G\,max} \cdot i_{V_{max}} \cdot \eta_G \cdot \eta_V \cdot \frac{1}{1+V} + G_V \cdot R_{dyn} \cdot \mu_R \frac{1}{i_A} \cdot \eta_A \right) \cdot \frac{1}{2}$$

These selections will avoid major dimensioning errors. However, they disregard important influences on the useful life such as deflection angle, rotation speed, loading, effect of dirt, temperature etc. For example, halving the deflection angle doubles the life, as 9.1 shows.

Please therefore use our questionnaire in chapter 12. We recommend the correct joint size using our computer program.

8.4 Critical Rotation Speed

The propeller shaft found from dimensioning specifications 8.1, 8.2 or 8.3 must now be checked for bending-critical rotation speed.

In general, propeller shafts run uncritically, i.e. their operating speed is below the critical speed. The critical speed for propeller shafts with steel tube is calculated from the equation:

$$n_{crit\,tube} = 1{,}22 \cdot 10^8 \cdot \frac{1}{l_o^2} \cdot \sqrt{D^2 + d^2}\ \text{min}^{-1}$$

where D = tube external diameter, d = internal diameter and l_0 = free length between the joints or centre bearing assemblies all in mm.

If special propeller shafts are produced with steel rotating rod, calculate the critical rotation speed as

$$n_{crit\,rod} = 1{,}22 \cdot 10^8 \cdot \frac{D}{l_o^2}\ \text{min}^{-1}$$

where D = rod diameter and l_0 = free length, all in mm.

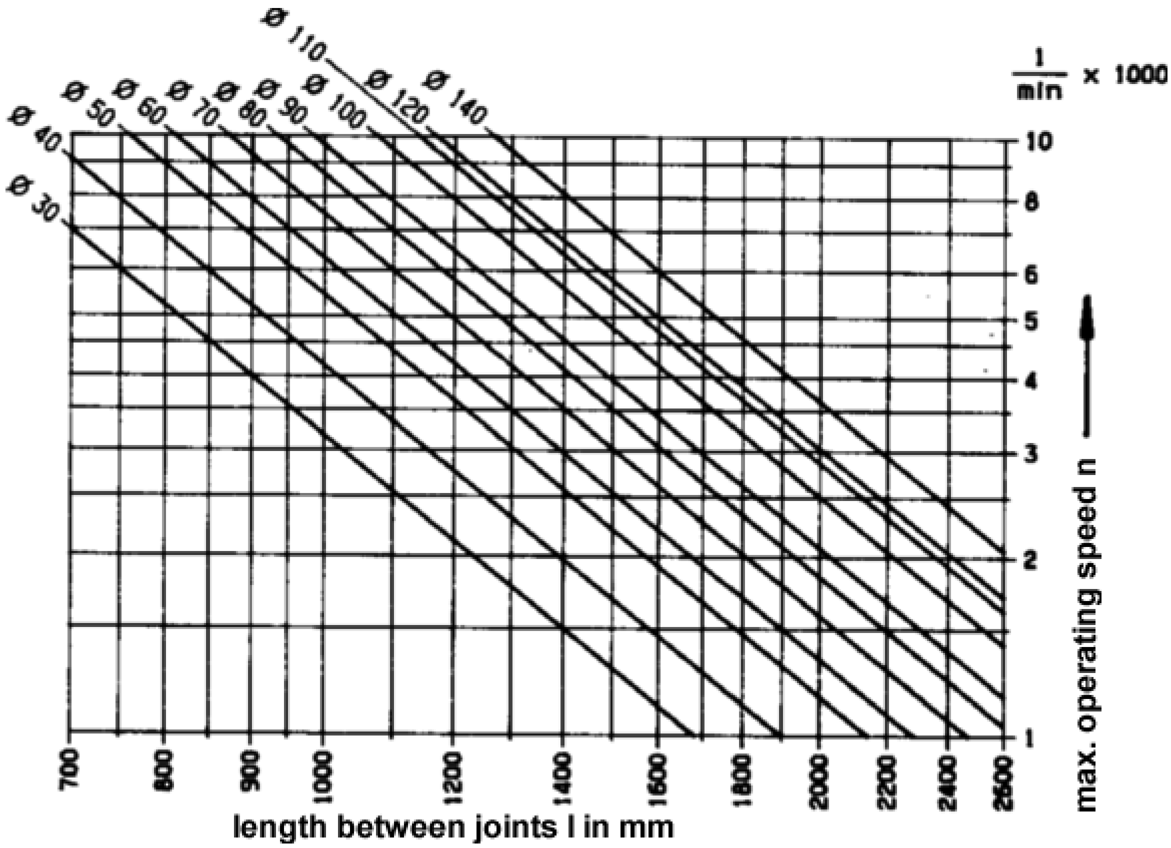
These equations apply for smooth tubes or rods. propeller shafts only achieve around 80-90% of this speed because of play in bearings and sliding pieces and additional dimensions. As the max. operating speed should lie 10-20% below this critical speed, the operating rotation speed selected is:

$n_{operation} \leqq 0{,}6 \ldots 0{,}7\ n_{krit}$

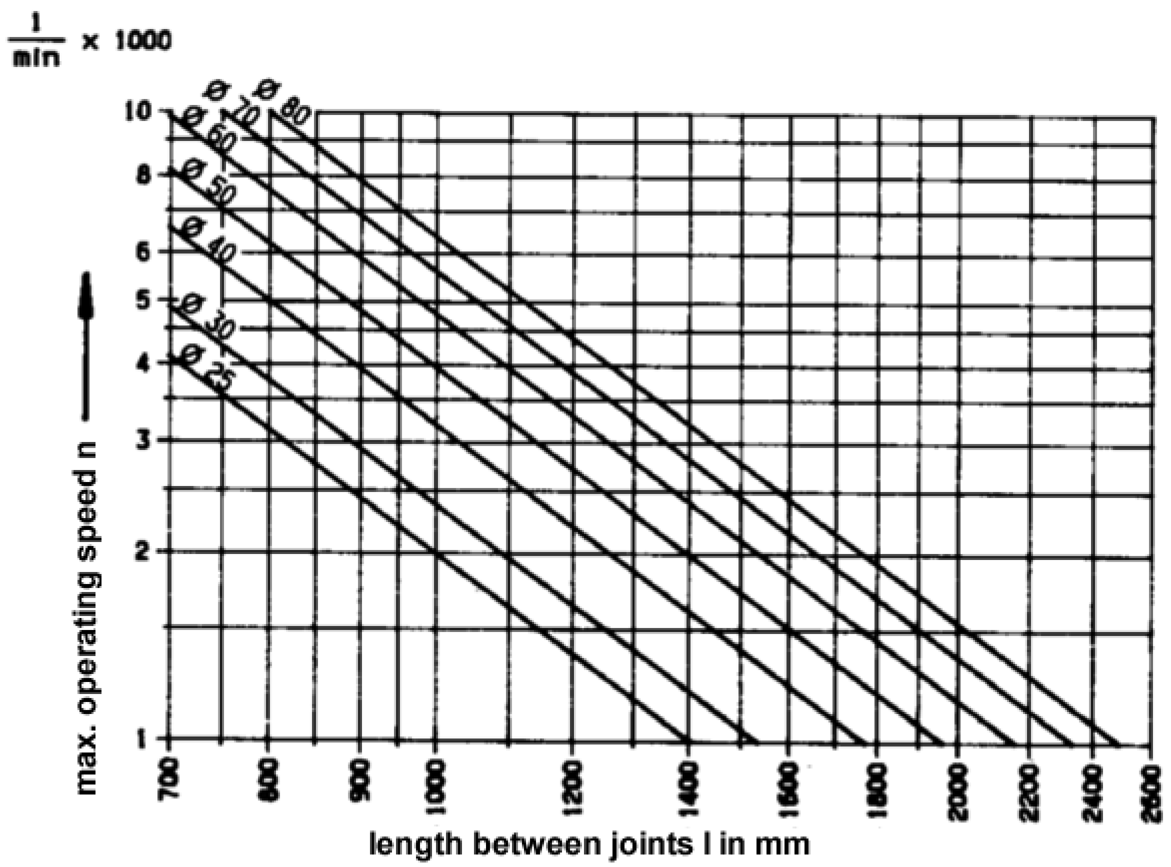
The maximum operating speed can be taken from the diagrams below.

Fig. 24:

Propeller shaft with steel tube



Propeller shaft with steel rotating road

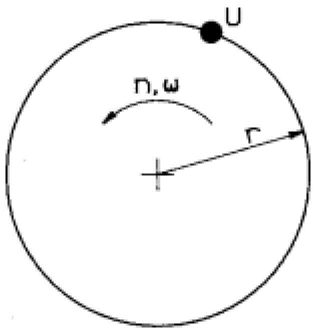


If the maximum operating speed is not sufficient, a larger tube diameter or rod design with centre bearings should be used.

8.5 Balancing Propeller Shafts

Propeller shafts for drive shafts in the automotive industry are dynamically balanced. Balancing is the equalization of weight of eccentrically running masses (Fig. 25) in the propeller shaft to achieve quiet running and reduce load on the joints and bearings in the connected assemblies.

Fig. 25:



Definition of imbalance:

Imbalance U = u · r in gmm
where u = unequalized individual mass on radius r

Shifting of centre of gravity

$$\varepsilon = \frac{u \cdot r}{G} = \frac{U}{G} \text{ in gmm/kg,}$$

where G = weight of part to be balanced.

Sensible Values for Permitted Imbalance

Practical experience has shown that as the rotation speeds increase, a smaller shift in the centre of gravity can be permitted. It is therefore sensible to take the product of rotation speed x shift in centre of gravity as a value for the permitted imbalance. DIN ISO 1940 "Requirements for balance qualitites of rigid rotors" is also based on this concept. A table there gives "quality classes" for different components, where it has been assumed that there is no point in balancing the different elements (wheels, rims, wheel sets, crankshaft components, shafts etc.) of a closed machine group, e.g. a vehicle, to widely differing quality classes.

According to DIN ISO 1940, propeller shafts should comply with class G40 ($\varepsilon \cdot \omega$= 40 mm/s), and propeller shafts for special requirements, class G16 ($\varepsilon \cdot \omega$= 16 mm/s).

Unless the customer specifies otherwise, the shafts are balanced at the maximum rotation speed to quality class G16. The permitted residual imbalance is determined from the equation below:

$u = 99363 \cdot \frac{G}{n_{bal} \cdot d}$ in g per side

where:
u = permitted unequalized individual mass per side in g
G = shaft weight in kg
n_{bal} = balancing rotation speed in rpm
d = tube diameter in mm

Example:
Shaft of 44 kg, n_{bal} = 3500 rpm, Tube diameter 90:
u = 99363 · 44 / (3500 · 90) = 13,8 g unequalized individual mass per side

As repeated clamping gives different values due to play, the values of the equation only correspond 65% to the value permitted under DIN ISO 1940. In test runs with repeated clamping therefore, 135% of the value given in DIN ISO 1940 is permitted, i.e. approximately double the equation value.

8.6 Mass Acceleration Moments - Influence of Rotation Speed and Deflection Angle

In order to achieve adequate smooth running of the propeller shaft, the mass acceleration moment of the centre part between the joints must not be too large. The mass acceleration moment depends on the mass moment of inertia of the centre part, the rotation speed n and the deflection angle of the joint. The permitted size of the mass acceleration moment increases with the moment transferability of the joint, i.e. as the joint power factor T increases, the permitted mass acceleration moment M also increases.

For propeller shafts in goods vehicles, depending on requirements, installation conditions and sprung mass system, the specific mass acceleration moment

M_{spec} = 0,04 to 0,06 Nm/Nm

If sound radiation is taken into account (buses etc.), the specific mass moment of acceleration $M_{spec.}$ must be smaller; if humming noise is of secondary importance, M_{spec} can be larger.

The specific mass acceleration moment M_{spec} is the quotient of the mass acceleration moment of the centre part and the joint power factor T.

$M_{spec} = M / T$

where M = · J_m

and $\varepsilon = \left(\frac{n \cdot \pi}{30}\right)^2 \cdot \frac{\sin^2\beta \cdot \cos\beta \cdot \sin 2\alpha}{(1 - \sin^2\beta \cdot \sin^2\alpha)}$ in s^{-2}

with ß = deflection angle of joint,   = rotation angle position of propeller shaft ($_{max}$ at 45º),
n = rotation speed of shaft in rpm and J_m = mass moment of inertia of shaft centre part in Nms^2.

The table below was produced from these equations and gives the max. n x ß value for propeller shafts of centre length 1.5 m as approximate values.

Joint Size	n_{max} [rpm]	n x ß [rpm · degree]
196	5500	28000
200	5500	34000
253	5000	24000
375	4800	21000
376	4800	19000
411	4600	19000
490	4400	17500
491	4500	17500
590	4000	16000
600	4200	18000
610	4000	17000
620	4000	16000
680	3800	15000
700	3700	16000
710	3600	14000

How far these values can be exceeded depends on the requirements for smooth running and many peripheral conditions. With favourable sprung mass systems, the value can be exceeded up to 50 %.

8.7 Measures to Improve Smoothness of Running

To reduce the radiated noise (gears or axle noise), the propeller shaft can be fitted with a cardboard tube pressed into the shaft tube. This effectively damps the higher frequencies.