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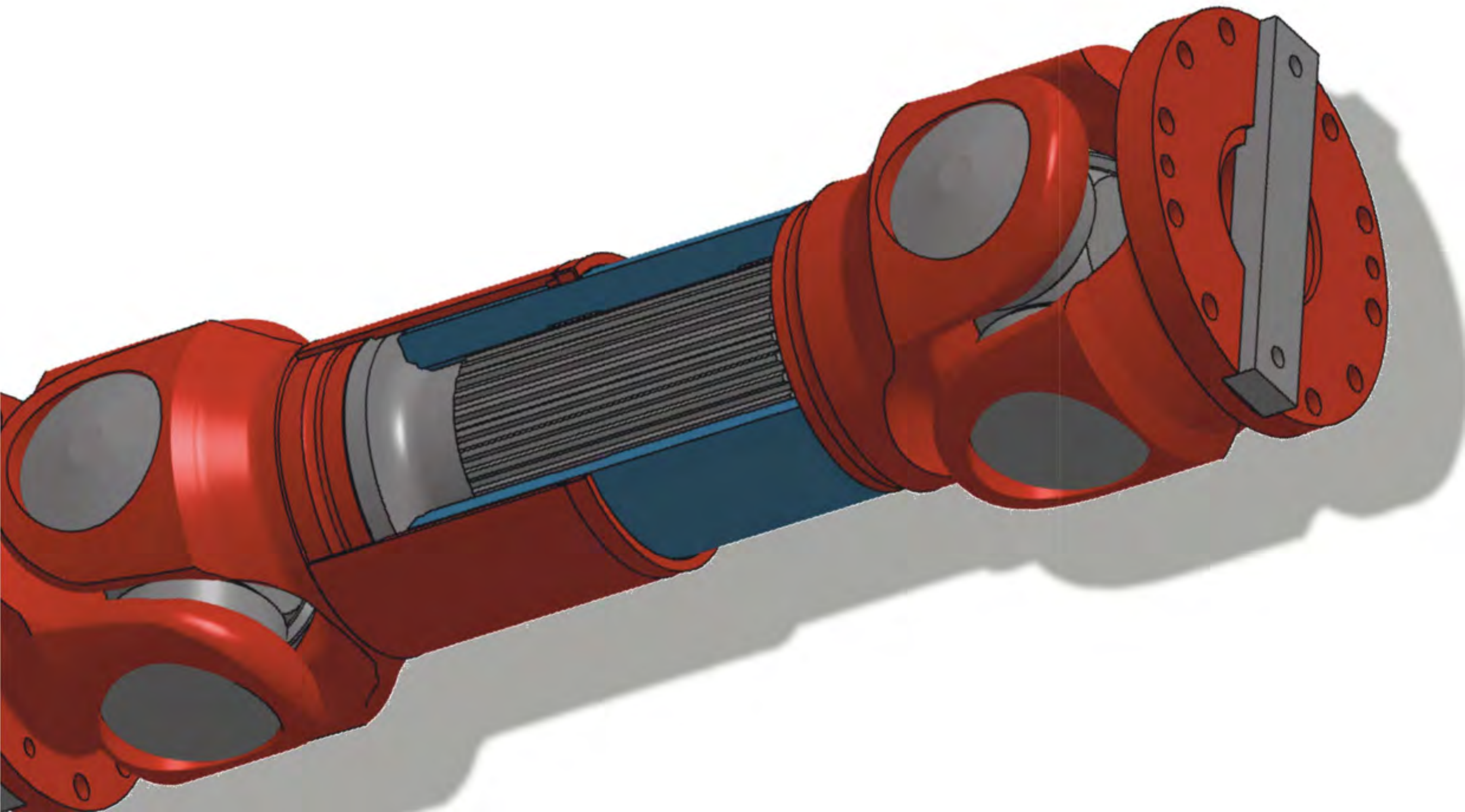
CARDANWORKS BY Sicit

CARDANWORKS Manual

Layout & Design



CARDANWORKS Manual

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1. Designing of cardan shafts for stationary applications

Cardan shafts transmit torques between non aligned shafts varying in position, limited by the strength of the materials. The driving engine and also the driven machine determine the load affecting the cardan shaft. The individual shock and load factors of the connected aggregates must be considered selecting a cardan shaft size. Besides the torque load and capacity, the geometric conditions like lengths and operating deflection angles, the speed, the environmental conditions, vibrations and much more have to be considered by selection procedure. The lifetime of the U.-joint bearing and the max. component strength are the two independent criteria as basis for the **CARDAN**WORKS shaft selection. These two values must be considered separately, with the aim to find an ideal balance between both.

The selection procedure outlined in this manual is only a general guidance. The selection and the operation of a cardan shaft must consider the safety of people and working surrounding (see **CARDAN**WORKS Manual Operating, service and safety). Please take in mind that the selection of the cardan shaft will be carried out according to the data provided by the customer to **CARDAN**WORKS. Accurate and complete information will help to avoid errors and misapplication. **CARDAN**WORKS will not be responsible for errors due to inaccurate or incomplete information supplied as well as for inappropriate handling and operation of the cardan shaft. It is the responsibility of the customer to ensure the applicability of the attached components to carry the maximum occurring load. CARDANWORKS reserve the right for technical changes and modification.

1.1 Cardan shaft size selection based on maximum bearing life

The working life of a cardan shaft is mainly restricted by the performance of the U.-joints. The service life is limited in case of an optimal lubrication by fatigue of the material after millions of operations, and on the other hand in case of inadequate sealing by corrosion and by abrasive wear as result of contamination.

The assumption for the lifetime calculation is the first case.

The testing of a cardan shaft for suitability in a specific application is not applicable for cost and time reasons. For this reason the selection of a cardan shaft follows a procedure based on the modified roller bearing theory. The calculation of the bearing life is based on the DIN/ISO 281 guidelines. The B10 lifetime determined by applying the DIN/ISO 281 recommendation is a theoretical value achieved or exceeded by 90% of similar bearings.

In practise the average service life is often approximately five times higher than the calculated B10 life. An impact on the durability of the Universal Joint has: the quality of the bearing itself, the quality of the cross body (steel, heat treatment, finishing, precision, etc.), lubrication, quality of the seals and static overload.

A major influence on the life of the cardan shaft in operation has the deflection angle. Every increase of the deflection angle by 5° is reducing the life of the Universal Joint bearing by half. Therefore it is advisable to operate the cardan shaft at rather small deflection angles, considering that a minimum angle of app. 3° is recommended.

Typical bearing life-oriented applications are locomotive primary drives, ventilators, pumps and paper machines. The torque peaks occur rather rarely, infrequent and of short duration – like a motor start-up.

The U.-joint life torque capacity **U_{LT}** as an engineering data characteristic of an specific cardan shaft size is established by multiplying C, the dynamic service life C_{dyn} (see DIN/ISO 281) of the bearing caps and the load arm depending on the geometry of the cross body.

The desired U.-joint life torque capacity **U_{LTd}** for the selection of a certain cardan shaft size is calculated with the formula shown below for uniform load and operation, assuming that the torque **T** occurring at a life span **L_h** as well as the speed **n** and the deflection angle β are constant.

$$U_{LTd} = T \cdot SF \cdot ß \cdot \left(\frac{Lh_d \cdot n}{1{,}5E+07}\right)^{\frac{3}{10}}$$

U_{LTd} desired UJ life torque capacity [Nm]
T applied, nominal torque [Nm]
SF shock factor (see chart)
β deflection angle of the joint [°], for a deflection of < 3°, set β = 3°
L_{hd} desired life span [h]
n rotational speed of the cardan shaft [rpm]

The applied, nominal torque **T** is calculated from the rated power and speed:

$$T = \frac{9550 \cdot P}{n} \text{ [Nm]}$$

P power of the driving machine [kW]
n speed of the shaft [rpm]

Shock factors SF

Torque peaks as they may occur in drivelines with internal combustion engines must be considered by the shock factor **SF**. The factor may be higher in cases without the use of a flexible coupling.

Driving machine	SF with flexible coupling
Electric Motor	1
Engine with converter	1
Diesel Engine 1-3 cylinder	1,3
Diesel Engine 4 and more cylinder	1,15
Otto-Cycle Engine 1-3 cylinder	1,25
Otto-Cycle Engine 4 and more cylinder	1,1

The U.-joint life torque capacity **U_{LTd}**, calculated according to this procedure, can be taken as the indication for the selection of a cardan shaft size. The chosen cardan shaft size with a higher **ULTc** will outperform the requirements. The reachable life expectancy, the expected service life, is then computed by setting both rating values in relation by the following formula:

$$L_h = L_d \cdot \left(\frac{U_{LTc}}{U_{LTd}}\right)^{3{,}33} \text{ [h]}$$

U_{LTc} chosen UJ life torque capacity [Nm]
U_{LTd} desired UJ life torque capacity [Nm]

Life expectancy of a duty cycle with varying operating conditions

The assumption for the rating life formula is that the bearing load, the speed and the other operating conditions are constant. Otherwise the total operating period must be split of into time fractions (n=1,2,3.....) reflecting the individual operating conditions (n=1,2,3,....).

For a selected **CARDAN**WORKS shaft size with a specific **ULTc** rating the individual service life of each time fraction is calculated as following:

$$L_{hn} = \frac{1{,}5E+07}{n_n \cdot \beta_n \cdot SF} \cdot \left(\frac{U_{LTc}}{T_n}\right)^{3{,}33} \quad [h]$$

L_{hn} expected service life at time /operating fraction n, n = 1,2,3....
β_n deflection angle at time /operating fraction n [°]
T_n torque at time /operating fraction [Nm]
n_n speed [rpm] at time /operating fraction n

When the individual service life fractions (q_1,q_2,...q_n in percent) are established the following formula can be used to calculate the total life service life expectancy of the whole duty cycle:

$$L = \frac{100\%}{\frac{q_1}{L_{h1}} + \frac{q_2}{L_{h2}} + \frac{q_3}{L_{h3}} \ldots + \frac{q_n}{L_{hn}}} \quad [h]$$

1.2 Cardan shaft size selection based on maximum component strength

Numerous of cardan shafts are operated under non-constant conditions. Very dynamic applications are rolling mills, locomotive secondary drives, crane travel drives and so on were torque peaks, high shock loads or vibrations are a permanent challenge for the cardan shaft. Size selection based on the torque capacity means to compare the expected peak torques with the nominal torque capacity $\mathbf{T_{MAX}}$ and the reversing $\mathbf{T_{FAT}}$, respectively the pulsating torque capacity $\mathbf{T_P}$ based on the component strength.

T_{MAX} - the nominal torque

The rated torque of the chosen CARDANWORKS shaft size $\mathbf{T_{MAX}}$ must not be exceeded by the maximum expected operating torque. This maximum expected operating torque or peak torque is calculated by multiplying the nominal torque of the driving engine and the load factor **LF** of the driven machine.

$$T_{MAX} \geq T \cdot L_F \quad [Nm] \qquad \text{with} \qquad T = \frac{9550 \cdot P}{n} \quad [Nm]$$

The load factors shown in the following charts consider shocks and vibrations causing overloads and should be applied as approximate values only.

LF	Type of load	Driven machine
1,1 - 1,5	Continous load	Centrifugal pumps Generators Conveyors Small ventilators Machine tools Printing machines
1,5 - 2	Light shock load	Generators (non-cont. load) Small paper and textile mc/s Compressors (multi-cyl.) Wood handling machines Rod and bar mills Locomotive primary drives
2 - 2,5	Medium shock load	Cylinder pumps Marine drives Transport roller tables Heavy paper/textile machines Pumps Mixers

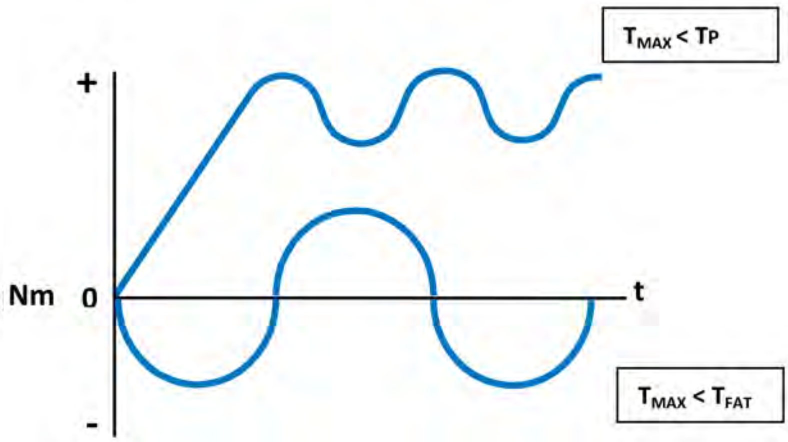
LF	Type of load	Driven machine
2,5 - 3	Heavy shock load	Building machinery Crane drives Locomotive secondary Presses
3 - 6	Extra heavy shock load	Reversing conveyors Straightening machines Cold rolling mills Rev. rolling mills
6 - 10	Extreme shock load	Feed roller drives Wrapper roller drives Plate-shears Reversing slabbing and blooming mills

The maximum expected operating torque or peak torque will then be compared with either T_P or T_{FAT} ,the rated capacity of the cardan shaft, depending whether it is a pulsating /non-reversing or a reversing load which is acting on the cardan shaft. The rating data of the selected shaft must exceed the peak loads.

T_P - the pulsating one way fatigue torque

Strength capacity limit of the cardan shaft for unlimited fatigue life of one-way pulsating load.

$$T_P \approx T_{max} \cdot 0{,}7 \text{ [Nm]}$$



T_{FAT} - the reversing fatigue torque

Strength capacity limit of the cardan shaft for unlimited fatigue life of alternating load.

$$T_{FAT} \approx T_{max} \cdot 0{,}5 \text{ [Nm]}$$

Cardan shafts for the following specific type of applications may require additional consideration and are often subject to the approval of classification societies or authorities. If you need more information please contact us.

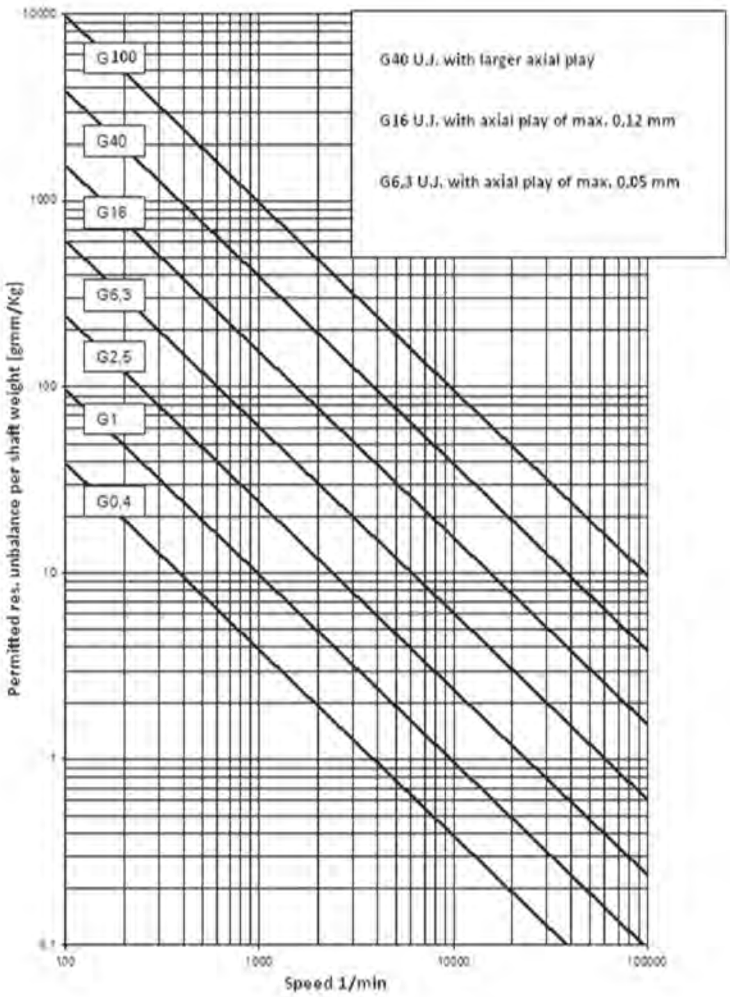
- **Railway drives**
- **Marine transmissions**
- **Explosive environments (ATEX)**
- **Crane travel drives**
- **Military use**
- **Passenger transportation**

2. Balancing of the cardan shaft

A cardan shaft must be balanced in two planes considering the operational speed to ensure a vibration free operation and to reduce forces acting on connected equipment. The precise balancing compensates forces of eccentric rotating masses of the cardan shaft according to the application requirements. It is a quality issue of a **CARDAN**WORKS shafts and a major prerequisite for a sufficient service life.

In accordance with the ISO Standard 1940 – Balance quality of rotating rigid bodies -cardan shafts are normally balanced to a balancing grade G 40 (ω x ε= 40 mm/s). Cardan shafts with higher requirements should be balanced to the balancing accuracy G 16 (ω x ε= 16 mm/s). As a standard **CARDAN**WORKS shafts are balanced for maximum operating speed, if this speed is exceeding 300 rpm, according to G 16. The more precise the cardan shaft is in terms of component precision and assembly, the closer the shaft can be operated at the critical bending speed and a better result can be achieved in the balancing process.

Values for the permissible residual unbalance for the quality grades can be taken from the graph. It is necessary to consider the reproducibility levels which can be achieved. Influence on this may have the tolerances of the cardan shaft connections, accuracy of the measuring system, the play of the universal joints and the precision of the splines. Due to this, the diagram displays 65% of the admissible values according to DIN ISO 1940, with a tolerable transgression to 135% of the results - which corresponds to twice the data of the diagram - for verification testing



3. Axial forces of the cardan shaft length compensation

Generally no axial forces result from the kinematic motion of the cardan shaft but from the friction and the pressure build up by the spline during length variation. It must be taken into account that these axial forces must be absorbed by appropriate bearings of the connected devices and that for extreme values it may lead to a premature failure of the universal joint bearings.

The frictional force **F_{FR}** is resulting from the input torque and at the same time the compression of the splines.

$$F_{FR} = T \cdot \mu \cdot 1/r_m \cdot \cos\beta \ [N]$$

μ	coefficient of friction for hardened, nitrited and/or phosphated parts: **μ**= 0,12 for nylon-coated parts: **μ**= 0,065
T	applied torque [Nm]
r_m	spline pitch circle radius [mm]
β	deflection angle [°]

The axial force component **F_P** results additionally by a pressure built-up in the lubrication grooves of the length compensation. This force depends on the lubrication pressure which must not exceed 15 bar and can be reduced by an installed relief valve.

The maximum axial force **F_{AX}** is a combination of two components, the frictional force **F_{FR}** and a pressure force **F_P**.

$$F_{AX} = F_{FR} + F_P \ [N]$$

4. Critical bending speed

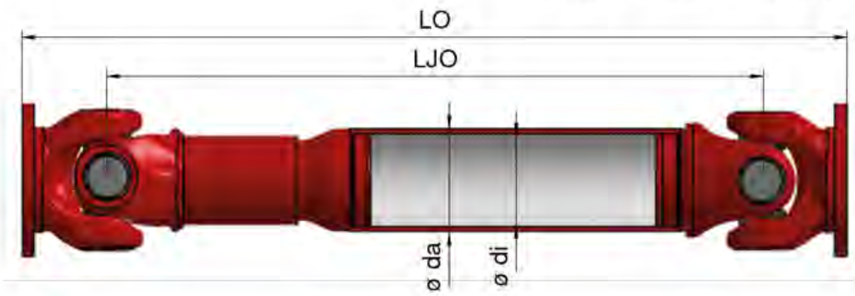
Except for short and rigid designs, cardan shafts are flexible components with critical bending speeds and flexural vibrations. It is essential to have this checked for safe operation with a sufficient distance to the critical bending speed.

The section between the joints of a deflected cardan shaft is exposed to acceleration and deceleration twice per rotation, creating vibrations and buckling play of the length compensation. This results in an increased noise level, eventual premature destruction of the shaft and connected units and at last it may culminate in the bending and breaking of the shaft.

The critical bending speed **n_{krit}** for a particular shaft configuration is determined by the length and the tube dimensions and can be calculated with the following formula:

$$n_{krit} = 1{,}22 \cdot 10^8 \cdot \frac{1}{L_{JO}^2} \sqrt{d_a^2 + d_i^2} \ \ min^{-1}$$

d_a	tube outside diameter [mm]
d_i	tube inner diameter (**d_a** – 2 x wall thickness) [mm]
L_{JO}	length of section between joints in operation (**L_O** – flange yoke height) [mm]
L_O	operating length [mm]



With the influence of the free play of the bearings in the joints as well as the free play in the splines cardan shafts cannot reach the speed n_{krit}. As for the safety reasons mentioned before the maximum permitted operating speed n_P must be kept at a certain distance below to the critical speed. In order to meet both requirements a corrective factor is applied as following:

$$n_P = 0{,}65 \cdot n_{krit}$$

For greater length dimensions the tube diameter, if not limited by the shafts size, has to be increased. It has to be check if other factors like increasing masses are no obstruction for the operation of the shaft. If certain limits are exceeded multi piece drivelines with intermediate bearing supports have to be installed. It has to be taken care of that some installations, e.g. with elastic mounting bearings, require even lower permitted operational speeds.

5. Operating limitation by the relation between speed and deflection angle

For the reason of the kinematic of a cardan shaft the deflection angle must be limited in relation to the speed. It is not assured that the shaft can be operated at the maximum possible deflection angle provided by the yoke geometry and at the same instant be operated at the maximum permitted speed. Certain mass accelerations torques of the centre section of the cardan shaft must be undercut to provide smooth running at a low noise level and also low wear. This acceleration torque is related to the product of speed and deflection angle, **n x ß**.

The limit of the product of speed and deflection angle for cardan shaft sizes and length with a weight **G [kg]** which are not listed in the chart can be determined with this formula:

$$n \cdot \beta \leq \frac{36000}{\sqrt[6]{G}}$$

The maximum permissible values for each cardan shaft size with an average length L_c of 1500 mm are shown in the charts.

CARDAN SHAFT SIZE	limit of n x ß	nmax 1/min
X.078.	27000	6000
X.100.	25000	6000
X.113.	23000	5500
X.127.	21000	5500
X.142.	21000	5500
X.158.	20000	5500
X.172.	19000	5500
X.178.	19000	5000
X.204.	17000	4500
X.200.	16000	4000
X.225.	16000	4000
X.250.	15000	3500

CARDAN SHAFT SIZE	limit of n x ß	nmax 1/min
X.265.	15000	3500
X.285.	14000	3500
X.315.	14000	3500
X.350	13000	3500
X.390.	12000	3000
X.440.	11000	3000
X.490.	11000	3000
X.550.	10000	2500
X.620.	10000	2500
X.680.	9000	2000
X.750.	8000	2000
X.840.	8000	2000

6. Length dimensioning

The operating length **Lo** of a cardan shaft is determined by the distance between the flanges of the driven and the driving equipment and the length compensation during the operation. Cardan shafts with a fixed length **LF** like tube shafts, intermediate shafts or flange joints have a constant, non changing operating length. For the use of one of these types it is a prerequisite that one connecting side is equipped with a floating bearing configuration to compensate length variations due to temperature influences as well as fitting clearance.

The optimum working length of a cardan shaft or a short couple shaft with length compensation is if the length compensation is approximately extracted by one-third. As for the maximum compressed shaft with a length of $L_{C\,min}$ it is always the necessary to observe the fitting clearance.

$$L_O = L_C + \frac{1}{3} L_S \text{ [mm]}$$

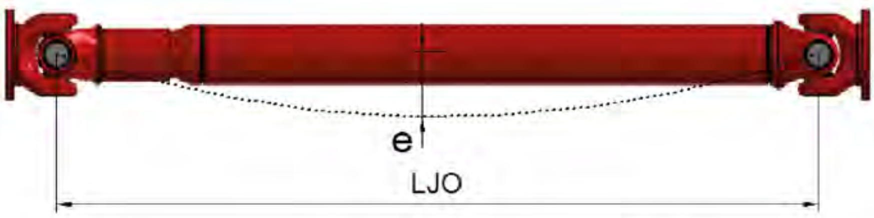
L_O operating length [mm]

L_C compressed length [mm], a further compression is not possible and would destroy the shaft

L_S length compensation [mm], an extension beyond this is permitted

7. Deflection under weight

Depending on the length and the weight of the centre section between the joints the cardan shaft will deflect at the distance **e** [mm] from the centre line. It has to be checked if this is exceeding certain limits and requirements.



$$e = \frac{5 \cdot G \cdot 9{,}81 \cdot L_{JO}^3}{384 \cdot E \cdot I} \text{ [mm]}$$

G	weight of the cardan shaft [kg]
L_{JO}	length of section between joints in operation (**LO** – flange yoke height) [mm]
$E = 210 \cdot 10^3$	elastic modulus [N/mm^2]
$I = \frac{\pi}{64} \cdot (da^4 - di^4)$	moment of inertia of tube [mm^4]