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Shafts with interference-fit collars Part I: some factors affecting the design of an assembly

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Data Item was monitored and guided by the Stress Analysis and Strength of Components Committee which first met in 1964 and now has the following membership:

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SHAFTS WITH INTERFERENCE-FIT COLLARS

PART I: SOME FACTORS AFFECTING THE DESIGN OF AN ASSEMBLY

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SHAFTS WITH INTERFERENCE-FIT COLLARS

PART I: SOME FACTORS AFFECTING THE DESIGN OF AN ASSEMBLY

1. INTRODUCTION

This document, Part I of a four-part series, outlines the main aspects of the design of interference-fit collars on rotating shafts insofar as the design is affected by the interference fit. The other documents in the series are:

- Part II: nominal elastic interface pressures and stresses, Item No. 68003 including ESDUpac A6803,
- Part III: procedure for calculation of nominal elastic interface pressures and stresses in assemblies of three or more elements, Item No. 68004 including ESDUpac A6804, and
- Part IV: fatigue strength of plain shafts, Item No. 68005.

A collar is here defined as a flat-sided disc or cylinder of constant axial thickness, retained on the shaft by a uniform initial interference; typical configurations are shown in Sketch 8.1. The effects of departures from these shapes near the interface are discussed where relevant. Geometrical features remote from the interface do not affect conditions on the interface, so this document is equally applicable whether the collar is, for example, a gear wheel or a v-grooved pulley. The document is, however, not valid for assemblies incorporating dowels or keyways in the shaft, which alter the stress distribution.

The interference fit may be used to transmit load or merely to retain the collar on the shaft. In the latter case, steady working loads may be negligible but inertial loads, such as starting torques or run-down torques, must still be considered. This document is concerned mainly with the following design criteria:

- (i) maximum stresses not to exceed specified allowable values for the particular materials chosen,
- (ii) freedom from general slip under specified working conditions, and
- (iii) a specified minimum fatigue strength.

While particular applications may impose a variety of other requirements, these three are commonly predominant.

2. NOTATION

Both SI and British units are quoted but any coherent system of units may be used.

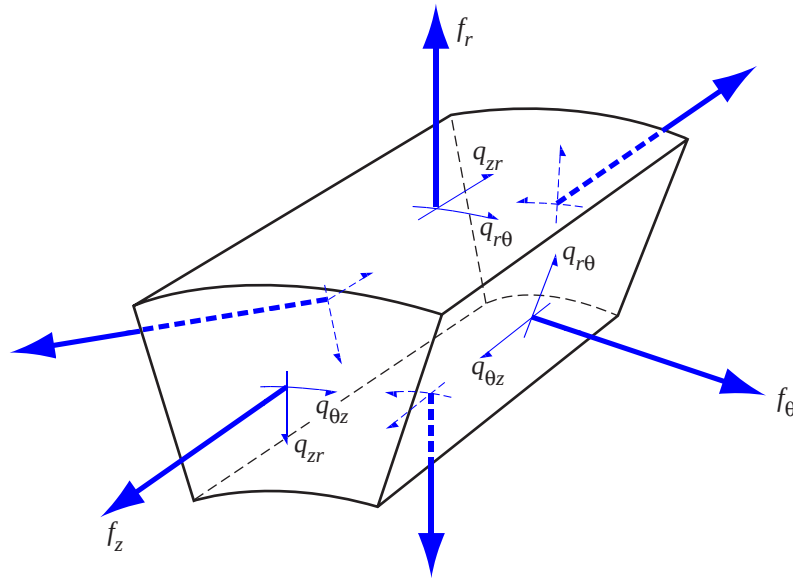
d	mean diameter of shaft or seating under collar	m	in
D	outside diameter of collar	m	in
f	circumferential stress at bore of collar due to interference plus rotation	N/m ²	lbf/in ²

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f_e	von Mises-Hencky equivalent stress $\frac{1}{\sqrt{2}} \left[(f_r - f_\theta)^2 + (f_\theta - f_z)^2 + (f_z - f_r)^2 + 6(q_{r\theta}^2 + q_{\theta z}^2 + q_{zr}^2) \right]^{1/2}$ (see Sketch 2.1 and Table 8.2)	N/m ²	lbf/in ²
f_t	tensile strength of material	N/m ²	lbf/in ²
f_0	circumferential stress at bore of collar due to interference alone	N/m ²	lbf/in ²
l	axial length of shaft under collar	m	in
p	interface pressure due to interference plus rotation	N/m ²	lbf/in ²
p_0	interface pressure due to interference alone	N/m ²	lbf/in ²
P	total applied axial force (see Section 3.1)	N	lbf
P_{sl}	axial force required to produce slip, if applied alone (see Section 3.1)	N	lbf
T	total applied torque in radial plane (see Section 3.1)	N m	lbf in
T_{sl}	value of T required to produce slip if applied alone (see Section 3.1)	N m	lbf in
α	half angle of tapered shaft	deg	deg
μ	coefficient of friction between collar and shaft		
ν_s	Poisson's ratio for material of shaft		
ρ	density of material	kg/m ³	lbm/in ³ *
ω	angular velocity	rad/s	rad/s

Convention: tensile stresses are positive, compressive stresses are negative; pressure is positive.

* A density value expressed in familiar British units as pounds mass per cubic inch has to be divided by 386 before it can be used in the formulae in this Data Item. (A force of 1 lbf acting on a mass of 1 lbm produces an acceleration of $12 \times 32.174 = 386 \text{ in/s}^2$.)



Sketch 2.1 Element of Shaft or Collar Showing Notation for Stresses

3. LOAD TRANSMISSION

3.1 Basic Equations for Parallel Shafts

The axial slipping force for an interference-fit collar on a circular parallel shaft may be estimated from

$$P_{sl} = \pi \mu p d l. \quad (3.1)$$

Similarly, the slipping torque may be estimated from

$$T_{sl} = \frac{1}{2} \pi \mu p d^2 l. \quad (3.2)$$

Under a combined load system, the slip condition can be taken as

$$\left(\frac{P}{P_{sl}} \right)^2 + \left(\frac{T}{T_{sl}} \right)^2 = 1, \quad (3.3)$$

where P and T are maximum service loads, *i.e.* combinations of steady, dynamic and vibratory loadings as appropriate and, in general, each is the sum of the maximum value of each of these components.

3.2 Basic Equations for Tapered Shafts

The axial slipping force for an interference-fit collar on a circular tapered shaft may be estimated from

$$P_{sl} = \pi p d l \left(\frac{\mu + \tan \alpha}{1 - \mu \tan \alpha} \right) \quad (3.4)$$

when the force is directed up the taper and from

$$P_{sl} = \pi p d l \left(\frac{\mu - \tan \alpha}{1 + \mu \tan \alpha} \right) \quad (3.5)$$

when the force is directed down the taper.

The slipping torque may be estimated from

$$T_{sl} = \frac{1}{2} \pi p d^2 l \cos \alpha (\mu^2 - \tan^2 \alpha)^{1/2} \quad (3.6)$$

for $\mu > \tan \alpha$.

Under a combined load system, the slip condition is given by Equation (3.3) using values of P_{sl} calculated from Equations (3.4) or (3.5) and T_{sl} from Equation (3.6).

3.3 Estimation of Interface Pressure p

For a collar of uniform radial thickness, of the type shown in Sketch 8.1, the nominal interface pressure including the effect of rotational speed can be obtained from Item No. 68003 and the program that accompanies it. For bushed collars and similar assemblies comprising more than two components, Item No. 68004, which is also accompanied by a program, gives a procedure for the calculation of the nominal interface pressures and stresses. The actual interface pressure where the shaft protrudes from the collar is locally higher than the nominal value but this effect should be neglected in calculating the load transmission properties.

In many practical cases where the collar is not of uniform thickness or length, satisfactory estimates of the interface pressure may be made by the use of suitable idealisations of the actual geometry. Some examples of suggested idealisations are given in Sketch 8.2. In cases difficult to visualise, under-estimates and over-estimates should be made and compared; in cases where stresses appear critical, particularly those involving high peripheral speeds, a full stress analysis would be required.

3.4 Coefficient of Friction and Behaviour after Slipping

The effective coefficient of friction between collar and shaft depends on several factors, the two most important being the surface conditions and the geometrical accuracy of the joint. Surface condition, such as oxidation, lubrication and surface finish, affects the coefficient of friction between the surfaces. Geometrical accuracy in the machining of the mating components, *e.g.* ovality, errors in parallelism and tolerances, affects the actual interferences achieved and thus the friction between the mating components. **High accuracy and a smooth finish are both essential in practice in order to achieve the interface pressures calculated from Item No. 68003.**

Table 8.1 gives suggested design values of μ which are 0.02 below the minimum published test results from a series of accurately made steel assemblies. These values can only be a guide to what may obtain in particular practical conditions. The values are based on tests under an axial load or torque and apply at the instant of first slip. The load required to sustain slip between the surfaces is typically 50 to 80 per cent of the load at first slip, except of course in the case of an axial load tending to push the collar up or down a taper.