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Hydraulic fluid power — Method for evaluating the buckling load of a hydraulic cylinder

Transmissions hydrauliques — Méthode d'évaluation du flambage d'un vérin

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Foreword

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The committee responsible for this document is ISO/TC 131, *Fluid power systems*, Subcommittee SC 3, *Cylinders*.

This second edition cancels and replaces the first edition (ISO/TS 13725:2001), which has been technically revised.

Introduction

Historically, cylinder manufacturers in the fluid power industry have experienced very few rod buckling failures, most likely due to the use of adequately conservative design factors employed during cylinder design and to the recommendation of factors of safety to the users. Many countries and some large companies have developed their own methods for evaluating buckling load.

The method presented in this Technical Specification has been developed to comply with the requirements formulated by ISO/TC 131.

Hydraulic fluid power — Method for evaluating the buckling load of a hydraulic cylinder

1 Scope

This document specifies a method for the evaluation of the buckling load which

a) takes into account a geometric model of the hydraulic cylinder, meaning it does not treat the hydraulic cylinder as an equivalent column,

b) can be used for all types of cylinder mounting and rod end connection specified in Table 2,

c) includes a factor of safety, k, to be set by the person performing the calculations and reported with the results of the calculations,

d) takes into account possible off-axis loading,

e) takes into account the weight of the hydraulic cylinder, meaning it does not neglect all transverse loads applied on the hydraulic cylinder,

f) can be implemented as a simple computer program, and

g) considers the cylinder fully extended.

The method specified is based on the elastic buckling theory and is applicable to single and double acting cylinders that conform to ISO 6020 (all parts), ISO 6022 and ISO 10762. If necessary, finite element analyses can be used to verify as well as to determine the buckling load.

The method is not developed for thin-walled cylinders, double-rods or plunger cylinders.

The method is not developed for internal (rod) buckling.

The friction of spherical bearings is not taken into account.

NOTE This method is based mainly on original work by Fred Hoblit.[2] This method has been established in reference to the standard NF PA/T3.6.37.[1]

2 Symbols and units

2.1 General

The symbols and units used in this document are given in Table 1. See Figures 1 and 2 for labels of dimensions and other characteristics.

Table 1 — Symbols and units

Symbol	Meaning	Unit
C	stiffness of a possible transverse support at the free end of the piston rod	N/mm
D_{1e}	outside diameter of the cylinder tube	mm
D_{1i}	inside diameter of the cylinder tube	mm
D_2	outside diameter of the piston rod	mm
e_a, e_d	distance where the loading of an eccentrically loaded column is equivalent to a concentric axial force F and end moment $M = F [x] e$	mm
E_1	modulus of elasticity of cylinder tube material	N/mm^2

Table 1 *(continued)*

Symbol	Meaning	Unit
E_2	modulus of elasticity of piston rod material	N/mm^2
F	maximum allowable compressive axial load; modified by the factor of safety, (see k below), it creates in the piston rod a maximum stress equal to the yield stress of the piston rod material	N
$F_{critical}$	Euler buckling load of the cylinder	N
I_1	moment of inertia of the cylinder tube	mm^4
I_2	moment of inertia of the piston rod	mm^4
k	factor of safety [see Clause 1, c)]	—
L_1	cylinder tube length (in accordance with Figure 1)	mm
L_2	piston rod length (in accordance with Figure 1)	mm
L_3	length of the portion of rod situated inside the cylinder tube, i.e. the distance between the centre points of the piston and the piston rod bearing (in accordance with Figure 1) with the rod fully extended	mm
Lp	length of the piston	mm
M_a	fixed-end moment at the beginning of the cylinder tube of a fixed hydraulic cylinder	N·mm
M_{bc}	moment at the junction of cylinder tube and piston rod	N·mm
M_d	fixed-end moment at the end of the piston rod of a fixed hydraulic cylinder	N·mm
M_{max}	maximum moment in the piston rod	N·mm
R_a	reaction at the beginning of the cylinder tube	N
R_d	reaction at the end of the piston rod	N
R_{bc}	reaction between cylinder tube and position rod	N
X	distance from the end of a beam	mm
Y	deflection of a slender beam at distance x	mm
G	gravitational acceleration	mm/s^2
Δ	elongation of the possible transverse support at the free end of the piston rod	mm
θ	angle (crookedness) between the deflection curve of the cylinder tube and the deflection curve of the piston rod (see Figure 2)	rad
ρ_1	mass per unit volume of cylinder tube material	kg/mm^3
ρ_2	mass per unit volume of piston rod material	kg/mm^3
σ	stress	N/mm^2
σ_e	yield point of a material	N/mm^2
σ_{max}	maximum compressive stress	N/mm^2
φ_a	angle of the deflection curve at the beginning of the cylinder tube	rad
φ_b	angle of the deflection curve at the end of the cylinder tube	rad
φ_c	angle of the deflection curve at the beginning of the piston rod	rad
φ_d	angle of the deflection curve at the end of the piston rod	rad
ψ_a	angle at the beginning of the cylinder tube (see Figure 2)	rad
ψ_d	angle at the end of the piston rod (see Figure 2)	rad

2.2 Additional notations

The following additional notations are also used in this document:

$$s_1 = \sin(q_1 L_1) \quad (1)$$

$$c_1 = \cos(q_1 L_1) \quad (2)$$

$$s_2 = \sin(q_2 L_2) \quad (3)$$

$$c_2 = \cos(q_2 L_2) \quad (4)$$

$$q_1 = \sqrt{\frac{k \times F}{E_1 \times I_1}} \quad (5)$$

$$q_2 = \sqrt{\frac{k \times F}{E_2 \times I_2}} \quad (6)$$

NOTE The origin of these notations (used for calculation) comes from the original work of Hoblit (see Reference 2].

3 General principles

3.1 Purpose

The cylinder is a system consisting of three parts (Figure 2). Two parts, the cylinder tube and the rod outside of the tube, are considered as columns. This system is subject to compressive forces (F, -F). The third part is the connection between these two parts in the form of the small piece of the rod inside the tube and is modelled as a rotational spring. The purpose of this Technical Specification is to determine the maximum allowable force, F_{max}, that avoids reaching yield stress of the rod material, σ_e, as well as buckling.

3.2 Description

The cylinder is in static equilibrium. The cylinder is subjected to a deformation due to the compression forces (F, -F). This deformation is identified for each of the three parts of the cylinder by geometric unknowns (angles) and static unknowns (forces, moments) and a specific relation (Hoblit model) due to the rotational spring joining the cylinder tube and the rod.

Based on considerations of equilibrium and kinematics, a set of equations is formulated. The type of fixations (e.g. pin-mounted or fixed at the two ends) defines the number of unknown values (from 9 to 13). There are as many equations as unknown values. Six types of fixation are treated (Table 2).

The system of equations can be solved for an F value previously set. However, it is important to establish a particular value of F, noted $F_{critical}$. $F_{critical}$ cancels the determinant of the system of equations. This value should not be reached because it leads to an infinite value of the maximum stress of the rod (σ_{max}).

It is therefore necessary to find the value of F (F_{max}) between the zero value (in fact $\varepsilon \cdot F_{critical}$) and $F_{critical}$ (in fact $[1-\varepsilon] \cdot F_{critical}$) that leads the stress in the rod to reach the yield stress of the rod material (when $\sigma_{max} = \sigma_e$).

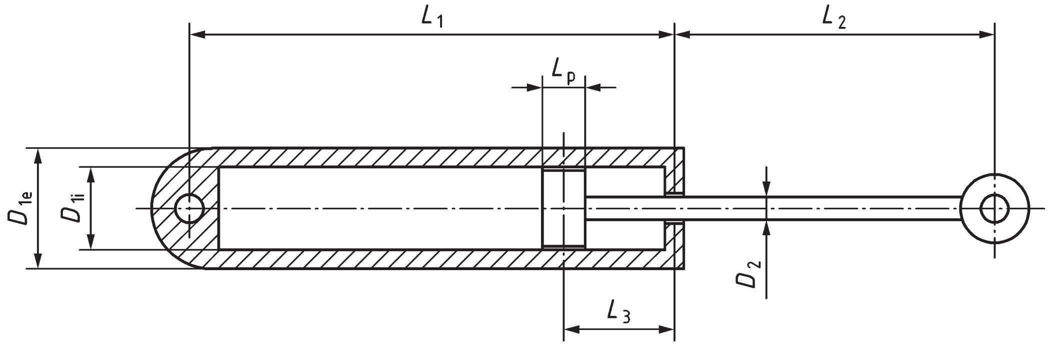
NOTE ε is a seed value used in the method of proportional parts to solve the set of equations.

3.3 Dimensional layout of hydraulic cylinder

Figures 1 and 2 depict the variables and principles used within this Technical Specification.

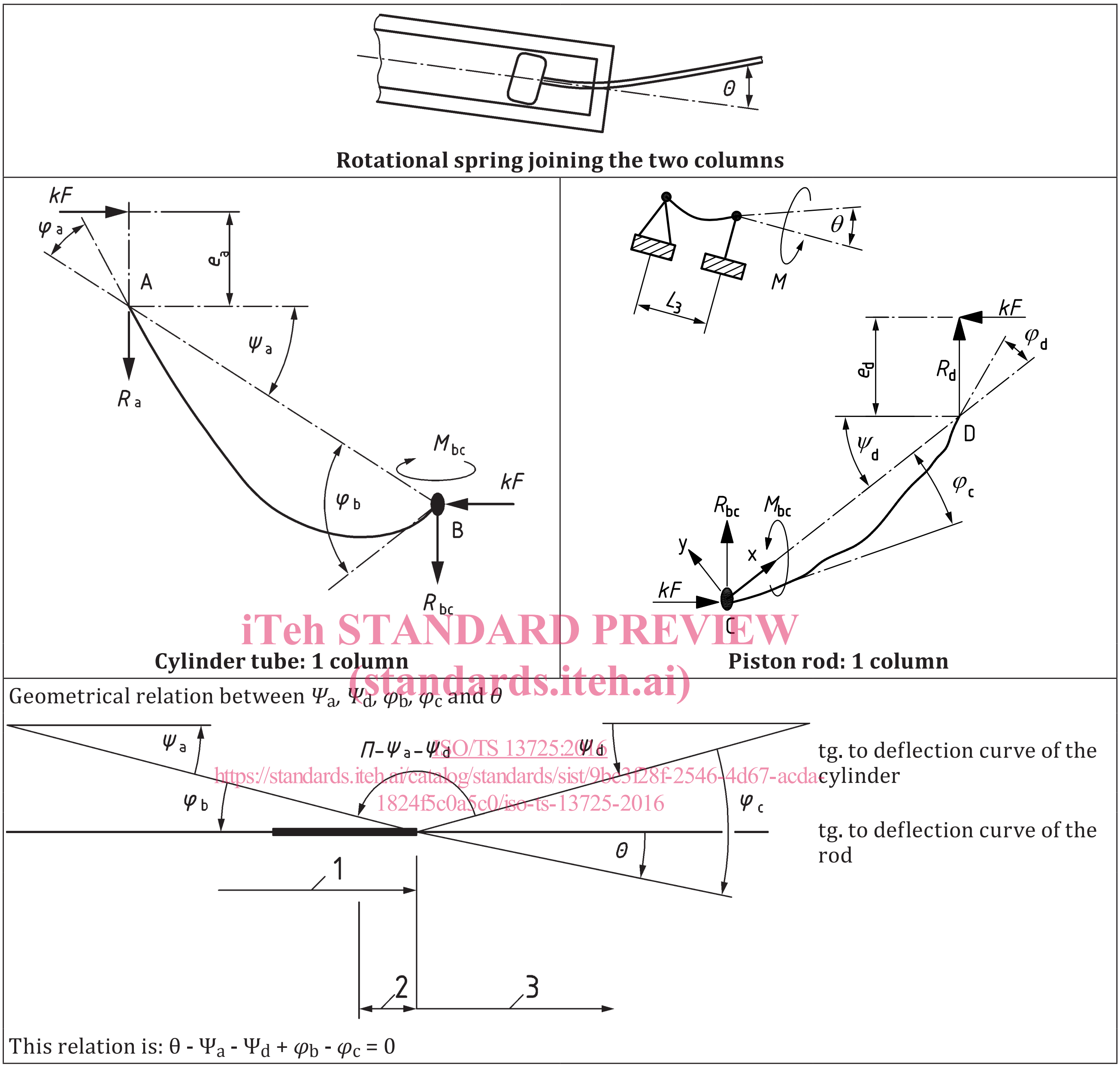
In the event that the external load F on the cylinder is at its maximum with the rod fully extended, the worst case occurs when the cylinder is in the horizontal position. In this case, the maximum allowable compressive load is at its lowest and creates the maximum stress in the piston rod. For this reason, and also considering the way of calculation where L_3 is insignificant compared with L_1 and L_2, L_3 is the shortest distance between the two centre points of the piston and the bearing.

When an almost retracted cylinder is loaded with a pushing force, there might be a risk of internal buckling of the rod. Therefore, the rod is to be calculated separately if this is regarded as a risk.



NOTE $L_3 = \dfrac{\left(L_p + \dfrac{(D_{1e} - D_{1i})}{2}\right.}{2}$ is a possible minimum value of L_3.

Figure 1 — Cylinder



Key

1 cylinder

2 rod portion inside the cylinder

3 rod

Figure 2 — Model of the hydraulic cylinder

3.4 Common calculation of maximum stress in the rod (for all mounting types) σ_{max}

The piston rod can be considered as the critical part of the cylinder if the thickness of the cylinder tube is sufficient. This condition should be verified before applying the generic method.

