TECHNICAL REPORT

ISO/TR 16806

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Pneumatic fluid power — Cylinders — Load capacity of pneumatic slides and their presentation method

Transmissions pneumatiques — Vérins — Capacité de charge des unités de guidage pneumatique et leur méthode de présentation

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

In exceptional circumstances, 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), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO/TR 16806 was prepared by Technical Committee ISO/TC 131, *Fluid power systems*, Subcommittee SC 3, *Cylinders*.

Introduction

In pneumatic fluid power systems, power is transmitted and controlled through a gas under pressure within a circuit. A pneumatic slide consists of a mounting surface for attaching a load, which is moved by an air cylinder and guided by stiff shafts to maintain alignment. There are limits to the amount of load that can be attached to a pneumatic slide, and these limits should be described as shown in this Technical Report.

Pneumatic fluid power — Cylinders — Load capacity of pneumatic slides and their presentation method

1 Scope

1.1 This Technical Report describes how to calculate the loading limits for a pneumatic slide based upon:

— external forces applied in the three principle planes of a tool plate, and applied at any point;

— external torque applied in the three principle planes of a tool plate;

— bearing limits determined by the slide manufacturer in conjunction with the bearing supplier.

1.2 This Technical Report also describes how to calculate tool plate deflections due to the loads.

1.3 This Technical Report describes how to present the rating information in technical documentation for application by a user.

1.4 This Technical Report assumes that all of the applied loads and torque will be absorbed by the guide rods and not by the piston rod. Only the axial thrust load (but not the resulting moments) will be absorbed by the piston rod.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 5598:1985, *Fluid power systems and components — Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 5598 and the following apply.

3.1
pneumatic slide
slide
mechanism containing a movable loading plate with guide rods, operated by an air cylinder

3.2
guide rod
shaft, passing through a set of bearings, which controls the deflection and twist of the loading plate

3.3
loading plate
plate onto which is placed a load to be moved

3.4
tool plate
loading plate attached at the end of the piston rod and guide rods

3.5
carriage plate
loading plate attached in the middle of the slide, containing the guide rod bearings

NOTE In this design, mounting plates are attached at both ends of the guide rods for mounting the slide, allowing the carriage to move.

3.6
housing
portion of the slide containing the bearings, when there is no carriage plate, and used for mounting the slide

4 Rating factors

4.1 Pressure containing capability

The manufacturer shall determine the maximum pressure that the pressure containing envelope is capable of sustaining if there is no load attached.

4.2 Maximum axial load

The manufacturer shall determine the maximum load for both push and pull directions, when the load reactions pass through the centre of the piston rod. Describe the limitations for any column buckling.

4.3 Maximum combined loading for a tool plate, and its deflections

The manufacturer shall determine the following coefficients:

$$A = 2l_1 / f$$

$$B = 2l_1 / f\,(l_1 + l_2)$$

$$C = l_2\,(l_1 + l_2)\ /\ (2l_1 + 3l_2)$$

$$D = (3l_2^2 + l_1 l_2 - l_1^2) / (2l_1 + 3l_2)$$

$$H = 12EI / l_2\,(2l_1 + 3l_2)$$

$$W = w\,(l_1 + l_2)$$

where

l_1 is the distance between the two bearing centrelines on one guide rod (this may vary with stroke); if there is only one bearing on a guide rod, then l_1 is the length of the bearing;

l_2 is the distance from the outer edge of the tool plate to the centreline of the closest bearing (this may vary with stroke);

f is the scaling factor chosen by the manufacturer to bring calculated numbers into convenient size for tabulation;

E is the modulus of elasticity for the guide rods;

I is the plane moment of inertia for two guide rods;

$I = \pi\ (d_G{}^4) / 32$

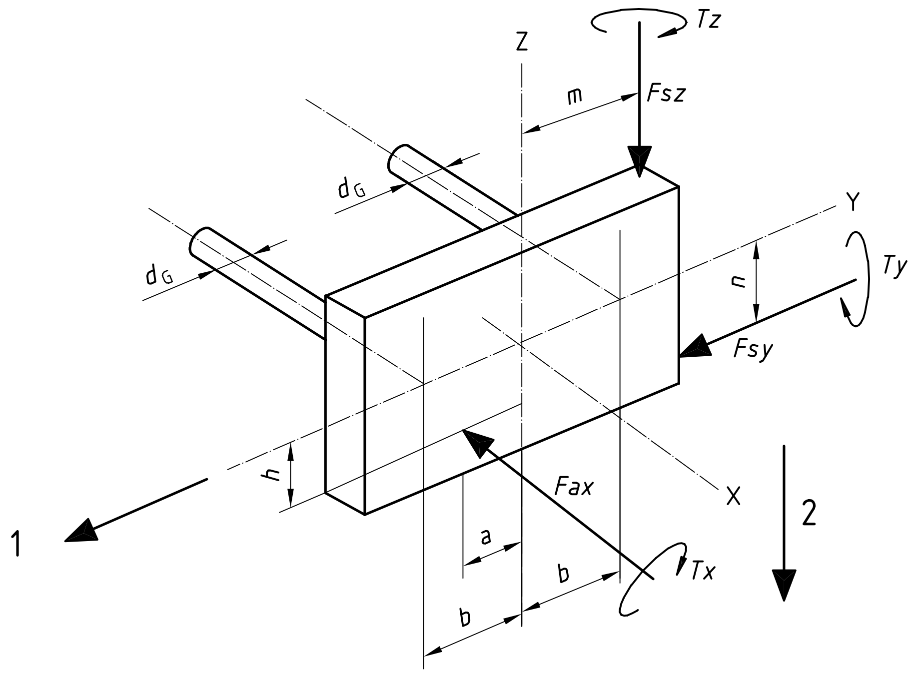
d_G is the diameter of guide rod;

w is the weight of guide rod per unit of length.

5 Presentation of ratings

5.1 Sketch of loading on tool plate

See Figure 1.



Key

1 inline deflections

2 parallel deflections

Figure 1 — Tool plate identifications

5.2 Tabulations

Coefficient *A*

STROKE	BORE SIZE			

Coefficient *B*

STROKE	BORE SIZE			

Coefficient *C*

STROKE	BORE SIZE			

Coefficient *D*

STROKE	BORE SIZE			

Coefficient *W*

STROKE	BORE SIZE		

Coefficient *H*

STROKE	BORE SIZE					

5.3 Graph

See Figure 2.

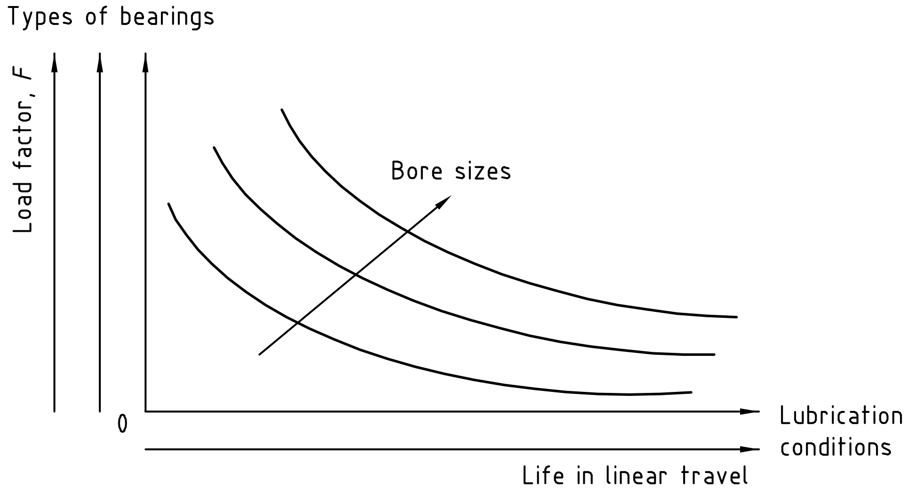


Figure 2 — Graph

5.4 Calculation formulas

5.4.1 Maximum tool plate load capacity

$$F^2 = \{(Fsy + W)/B + [(a)Fax + Tz]/A\}^2 + \{[Fsz\ (1 + m/b) + W]/B + [(h)Fax + Ty]/A + [Tx + (n)Fsy]/bB\}^2$$

where

A, B, W are coefficients determined in 4.3 and tabulated in 5.2;

a, b, h, m, n are dimensions on the tool plate as shown in Figure 1;

Fax, Fsy, Fsz, Tx, Ty, Tz are the applied forces and moments as shown in Figure 1.

The above formula describes the maximum combined loads that can be carried by the tool plate. If some of the loads do not exist in an application then it is possible to increase the other loads.

$F = fR_A$, the load factor presented in Figure 2

where

f is the arbitrary scaling factor described in 4.3;

R_A is the bearing capacity which the slide manufacturer establishes, in conjunction with a bearing supplier, taking into account the bearing design, materials, its life rating, and lubrication conditions. These are then reflected in Figure 2.

5.4.2 Linear deflections of the tool plate

— For inline deflections:

$$\delta = [4(C)Fsy + 2(a)Fax + 2Tz + W(D)]/H$$

— For parallel deflections:

$$\delta = [4(C)Fsz + 2(h)Fax + 2Ty + W(D)]/H$$

where

C, D, H, W are coefficients determined in 4.3 and tabulated in 5.2;

a and h are dimensions on the tool plate as shown in Figure 1;

Fax, Fsy, Fsz, Ty, Tz are the applied forces and moments as shown in Figure 1.

NOTE Tx is not a factor in the deflections.

Deflections are in the direction as indicated in Figure 1.

5.5 Nominal ratings

Show the following for the data determined in 4.1 and 4.2:

MAXIMUM PRESSURE CONTAINING CAPABILITY = ________

Table 1 — Maximum axial force (push - pull)

	BORE SIZE			
Maximum pull force				
Maximum push force				
Stroke limit for maximum push force [a]				

[a] This shall be the limit for column buckling of the piston rod, or it may be the maximum stroke offered by the manufacturer if it is less than the buckling limit.

6 Identification statement (Reference to this Technical Report)

It is strongly recommended that manufacturers use the following statement in test reports, catalogs and sales literature when electing to comply with this Technical Report:

"Rating of load capacities and deflections of pneumatic slides conforms to ISO/TR 16806:2003, *Pneumatic fluid power — Cylinders — Load capacity of pneumatic slides and their presentation method*."

Annex A
(informative)

Development of rating equations

Consider a tool plate with all possible force and torque loads applied as shown in Figure 1. These loads must be resolved into components that act on the rod guide bearings in order to determine if they are within the bearing capacities.

Begin with the side load forces Fsy and Fsz at their arbitrary locations and resolve these into equivalent side loads, and moments, on the principle axes (see Figure A.1).

Fsy is translated to axis Y with corresponding moment $nFsy$ acting clockwise around the X axis.

Fsz is translated to axis Z with corresponding moment $mFsz$ acting clockwise around the X axis.

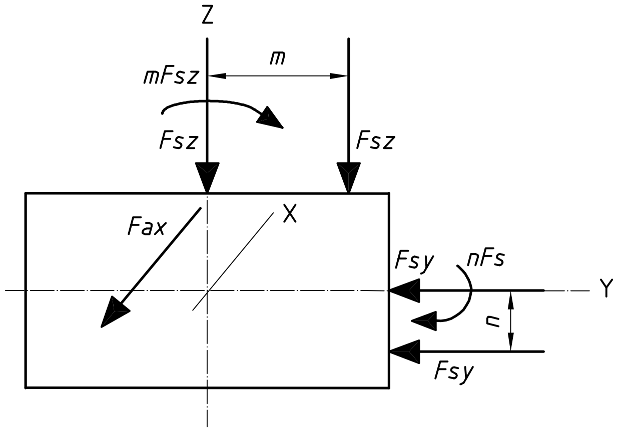


Figure A.1

Now consider how all of the forces and moments act on a free body of the tool plate and its guide rods. Figures A.2 to A.4 describe the resulting reaction loads on the bearings:

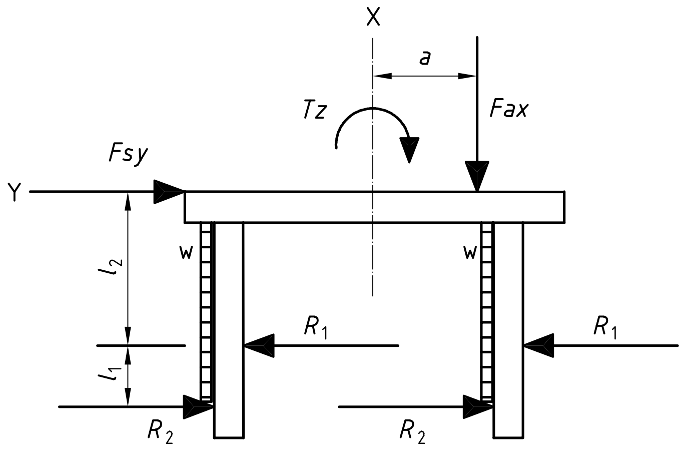


Figure A.2

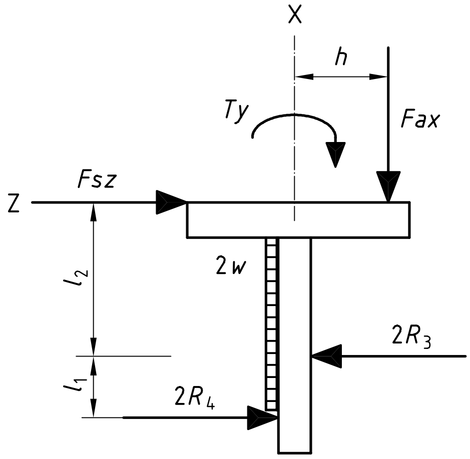


Figure A.3

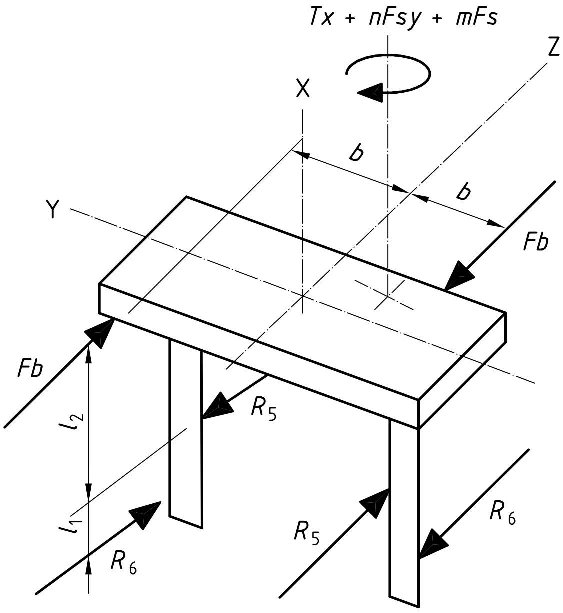


Figure A.4

From the assumptions stated in Clause 1, forces in the X axis will be absorbed by the piston rod. Therefore, these will not be considered in the balance of forces on the tool plate and guide rods. However, there must be a balance of forces in the Y and Z axes of each of the three sketches, plus a balance of moments from all three axes.

Consider Figures A.2 and A.3 and the balance of moments about the axis in the plane of the sketch.

NOTE Weight of the guide rods will only be effective if they are subject to gravity.

In Figure A.2, $\sum Mz = 0$

Consider moments about a line passing through the reactions R_2, at the intersection of the X axis:

$$2R_1(l_1) = Fsy(l_1 + l_2) + Fax(a) + Tz + 2w(l_1 + l_2)[(l_1 + l_2)/2]$$

$$R_1 = [Fsy(l_1 + l_2) + Fax(a) + Tz + w(l_1 + l_2)^2]/2l_1$$

Consider moments about a line passing through the reactions R_1, at the intersection of the X axis:

$$2R_2(l_1) + 2wl_1(l_1/2) = Fsy(l_2) + Fax(a) + Tz + 2wl_2(l_2/2)$$

$$R_2 = [Fsy(l_2) + Fax(a) + Tz + w(l_2^2 - l_1^2)]/2l_1$$

In Figure A.3, $\sum My = 0$

Consider moments about a line passing through the reactions R_4, at the intersection of the X axis:

$$2R_3(l_1) = Fsz(l_1 + l_2) + Fax(h) + Ty + 2w(l_1 + l_2)[(l_1 + l_2)/2]$$

$$R_3 = [Fsz(l_1 + l_2) + Fax(h) + Ty + w(l_1 + l_2)^2]/2l_1$$

Consider moments about a line passing through the reactions R_3, at the intersection of the X axis:

$$2R_4(l_1) + 2wl_1(l_1/2) = Fsz(l_2) + Fax(h) + Ty + 2wl_2(l_2/2)$$

$$R_4 = [Fsz(l_2) + Fax(h) + Ty + w(l_2^2 - l_1^2)]/2l_1$$

Now, consider Figure A.4 and resolve the moments in the plane perpendicular to the X axis, into a couple $Fb(2b)$ also acting in a plane perpendicular to the X axis. Then:

$$Fb(2b) = Tx + nFsy + mFsz$$

Twist of the tool plate is prevented by equal forces from each guide rod acting to oppose the loads Fb. Each guide rod, then, is subject to the load Fb plus reactions from the rod bearings as shown in Figure A.4. A balance of moments on each guide rod, perpendicular to the Y axis, can then be made:

Consider moments about a point on the guide rod at the intersection of the reaction R_6:

$$R_5\ (l_1) = Fb(l_1 + l_2)$$

$$R_5 = (l_1 + l_2)\ [Tx + nFsy + mFsz]/2bl_1$$

Consider moments about a point on the guide rod at the intersection of the reaction R_5:

$$R_6\ (l_1) = Fb(l_2)$$

$$R_6 = (l_2)\ [Tx + nFsy + mFsz]/2bl_1$$

All of the reaction loads on the guide rod bearings are now determined. However, they are applied in different directions as shown in Figure A.5 below, for each bearing:

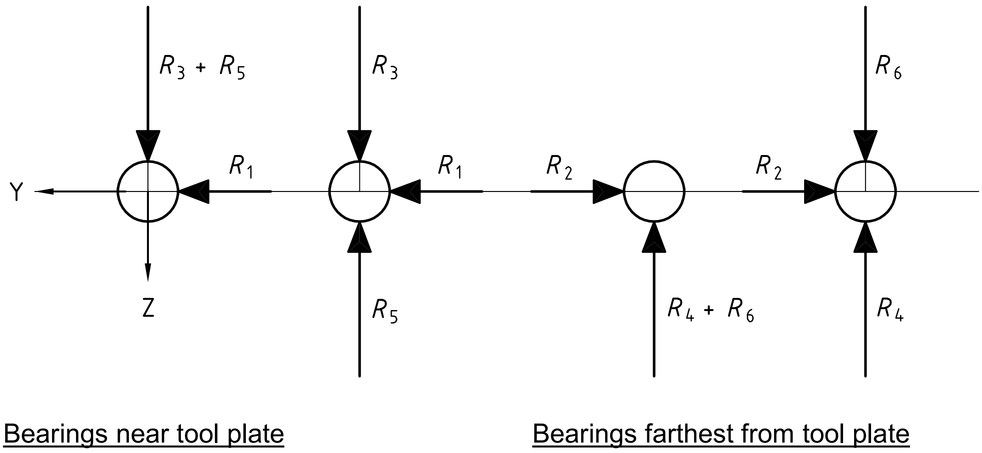


Figure A.5

The total load on each bearing is the vector sum of the several reactions as follows:

$$R_A{}^2 = R_1{}^2 + (R_3 \pm R_5)^2 \qquad\qquad R_B{}^2 = R_2{}^2 + (R_4 \pm R_6)^2$$

where

R_A is the total reaction on each bearing near the tool plate (use + sign for one bearing; – sign for the other bearing);

R_B is the total reaction on each bearing farthest from the tool plate.

If a slide design consists of only one bearing on each guide rod, then l_1 is the length of that bearing, and only the reaction R_A will exist.

From a summation of forces on each guide rod in Figures A.2, A.3, A.4, it can be observed that forces R_1, R_3, R_5 are larger than any of the other forces acting on a rod. Therefore, the bearing closest to the tool plate has the maximum load at all times. Furthermore, the bearing where forces R_3 and R_5 are additive, is the highest loaded bearing and can be used for the capacity rating.

Then

$$(R_A)_{max}{}^2 = R_1{}^2 + (R_3 + R_5)^2$$

Substituting values into the equation for $(R_A)_{max}{}^2$ yields:

$$(R_A)_{max}{}^2 = \{[Fsy(l_1 + l_2) + w(l_1 + l_2)^2 + Fax(a) + Tz]/2l_1\}^2 +$$

$$\{[Fsz(l_1 + l_2) + w(l_1 + l_2)^2 + Fax(h) + Ty]/2l_1 + (l_1 + l_2)[Tx + nFsy + mFsz]/2bl_1\}^2$$

Multiplying both sides by the scaling factor, f, and substituting the following terms:

$$F = f\,(R_A)_{max} \qquad A = 2l_1/f \qquad B = 2l_1/f\,(l_1 + l_2) \qquad W = w(l_1 + l_2)$$

the following is obtained:

$$F^2 = \{(Fsy + W)/B + [(a)Fax + Tz]/A\}^2 + \{[Fsz(1 + m/b) + W]/B + [(h)Fax + Ty]/A + [nFsy + Tx]/bB\}^2$$

This is the equation described in 5.4.1.

DEFLECTION EQUATIONS

Consider two possible directions of deflections as described in Figure 1, where the loads causing each type of deflection are described in Figures A.2 and A.3. The deflection is different for these two cases because the system of connected guide rods resist the bending in different ways. But, for either of these cases a guide rod can be represented by an equivalent beam loaded as shown in Figure A.6:

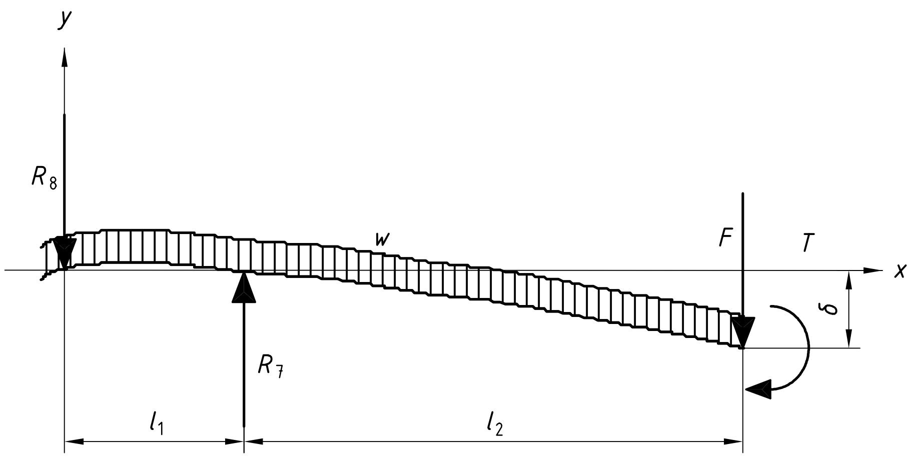


Figure A.6

Here all of the loads and torque are represented by the symbols F and T.

Taking successive balances of moments around R_7 and R_8, the reactions are determined as follows:

$$R_7 = \left[F(l_1 + l_2) + T + w(l_1 + l_2)^2\right] / l_1 \qquad R_8 = \left[Fl_2 + T + wl_2{}^2 - wl_1{}^2\right] / l_1$$

The double integration method is now used to determine the equation of the deflected beam:

For $0 \leqslant x \leqslant l_1$

$$\frac{d^2y}{dx^2} = \frac{M}{EI} = -\frac{-R_8x - 2wx\left(\frac{x}{2}\right)}{EI}$$

$$\frac{dy}{dx} = -\frac{wx^3}{3EI} - \frac{R_8x^2}{2EI} + C_1$$

$$y = -\frac{wx^4}{12EI} - \frac{R_8x^3}{6EI} + C_1x + C_2$$

At $x = 0$; $y = 0$ and $C_2 = 0$

At $x = l_1$; $y = 0$ and $C_1 = \frac{wl_1^3}{12EI} + \frac{R_8l_1}{6EI}$

For $l_1 \leqslant x \leqslant (l_1 + l_2)$

$$\frac{d^2y}{dx^2} = \frac{M}{EI} = -\frac{-R_8x + R_7\left(x - l_1\right) - 2wx\left(\frac{x}{2}\right)}{EI}$$

$$\frac{dy}{dx} = -\frac{wx^3}{3EI} - \frac{(R_7 - R_8)x^2}{2EI} - \frac{R_7l_1x}{EI} + C_3$$

$$y = -\frac{wx^4}{12EI} + \frac{(R_7 - R_8)x^3}{6EI} - \frac{R_7l_1x^2}{2EI} + C_3x + C_2$$

At $x = l_1$; dy/dx for $x \leqslant l_1 = dy/dx$ for $x \geqslant l_1$

and $C_3 = \frac{wl_1^3}{12E} + \frac{R_8l_1^2}{6EI} + \frac{R_7l_1^2}{2EI}$

Now, the desired result is an expression for maximum deflection. Therefore, use the equation where:

$l_1 \leqslant x \leqslant (l_1 + l_2)$

Which is

$$y = -\frac{w}{12EI}x^4 + \frac{(R_7 - R_8)}{6EI}x^3 - \frac{R_7l_1}{2EI}x^2 + \frac{2l_1^2(3R_7 + R_8) + wl_1^3}{12EI}x - \frac{R_7l_1^3}{6EI}$$

Substituting

$y = -\delta$ at $x = (l_1 + l_2)$

into the above equation, which yields

$$-\delta = \frac{R_7 l_2^3}{6EI} - \frac{R_8 l_2 \ (2l_1 + l_2) \ (l_1 + l_2)}{6EI} + \frac{w(l_1 + l_2)}{12EI} \left[l_1^3 - (l_1 + l_2)^3 \right]$$

Now, substituting the expressions for R_7 and R_8 into this yields

$$-\delta = \frac{l_2^3}{6EI\,l_1}\left[F(l_1 + l_2) + T + w(l_1 + l_2)^2\right] - \frac{l_2(2l_1 + l_2)(l_1 + l_2)}{6EI\,l_1}\left[Fl_2 + T + w\left(l_2^2 - l_1^2\right)\right] + \frac{w(l_1 + l_2)}{12EI}\left[l_1^3 - (l_1 + l_2)^3\right]$$

Combining terms and substituting $W = w\ (l_1 + l_2)$ yields

$$-\delta = \frac{-F\left[2l_2^2(l_1 + l_2)\right]}{6EI} - \frac{Tl_2(2l_1 + 3l_2)}{6EI} - \frac{Wl_2}{12EI}\left(3l_2^2 + l_1 l_2 - l_1^2\right)$$

Which can be simplified to

$$\delta = \frac{\left[4F(C) + 2T + W(D)\right]}{12EI / l_2(2l_1 + 3l_2)}$$

where

$$C = \frac{l_2(l_1 + l_2)}{(2l_1 + 3l_2)}$$

$$D = \frac{3l_2^2 + l_1 l_2 - l_1^2}{(2l_1 + 3l_2)}$$

Now the values for F and T will depend upon the orientation of the slide, either inline or parallel as shown in Figure 1.

Inline deflections

(also see Figure A.2)

$F = Fsy$ and $T = (a)\ Fax + Tz$

$\delta = [4Fsy\ (C) + 2(a)\ Fax + 2Tz + W(D)]/H$

Parallel deflections

(also see Figure A.3)

$F = Fsz$ and $T = (h)\ Fax + Ty$

$\delta = [4Fsz(C) + 2(h)\ Fax + 2Ty + W(D)]/H$

These are the equations described in 5.4.2, where:

$$H = \frac{12EI}{l_2(2l_1 + 3l_2)}$$

as described in 4.3.

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