Sizing criteria for cylinders and servocylinders

1 **ON-LINE CONFIGURATOR**

Accessible directly from Atos website, the **configurator** leads users through the definition of desired cylinder code, selecting step by step the characteristics and options required.The configurator guarantees free access to technical documentation and 3D view of the selected cylinders.
Users registered in MyAtos area have free access to 3D models export, which can be used to complete mechanical assembly drawings of hydraulic machineries and systems.

Main configurator features :
• Visualisation and export of 3D models in STEP format
• Technical documentation of products and spare parts
• Configuration summary in PDF format
• Configurations storing within the trolley to create parts lists and quotation requests.

Register in **MyAtos** area to have full access to configurator functionalities and contents.

DISCOVER THE ATOS CYLINDERS CONFIGURATOR

2 **HYDRAULIC FORCES AND DYNAMIC LIMITS**

2.1 Hydraulic forces

To ensure the correct cylinder functioning it is necessary to check that the hydraulic force F_p is upper than the algebraic sum of all the counteracting forces acting on the cylinder:

$$F_p \geq m \cdot a + F_f + m \cdot g$$

F_f are the friction forces of the system, $m \cdot a$ the inertial forces and $m \cdot g$ the weight force (only for vertical loads). For gravity acceleration consider g = 9,8 m/s[2].
For F_p values refers to section 3 , otherwise F_p, A_1, A_2 and speed V can be calculated as follow:

Hydraulic force

$$F_p = \left| p_1 \cdot A_1 - p_2 \cdot A_2 \right| \cdot 10 \quad [N]$$

Cylinder speed

$$V = \frac{10 \cdot Q}{A \cdot 60} \quad \left[\frac{m}{sec} \right]$$

Pushing area

$$A_1 = \frac{\pi \cdot D^2}{4 \cdot 100} \quad [cm^2]$$

Pulling area

$$A_2 = \frac{\pi \cdot (D^2 - d^2)}{4 \cdot 100} \quad [cm^2]$$

2.2 Dynamic limits due to oil elasticity

The calculation of the pulsing value ω_0 of the cylinder-mass system allows to define the minimum accleration/deceleration time t_{min}, the max. speed V_{max} and the min. acceleration/deceleration space S_{min} to not affect the functional stability of the system. Calculate ω_0, t_{min}, V_{max} and S_{min} with the below formulas. Flexible piping or long distances between the directional valve and the cylinder may affect the stiffness of the system, thus the calculated values may not be reliable.

$$\omega_0 = \sqrt{\frac{40 \cdot E \cdot A_1}{c \cdot m} \cdot \frac{1 + \sqrt{\frac{A_2}{A_1}}}{2}} \quad \left[\frac{rad}{s} \right]$$
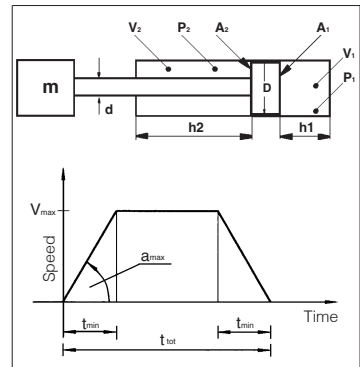
$$t_{min} = \frac{35}{\omega_0} \quad [s]$$

$$V_{max} = \frac{c}{t_{tot} - t_{min}} \quad [mm/s]$$

$$S_{min} = \frac{V_{max} \cdot t_{min}}{2} \quad [mm]$$

Note: for mineral oil consider E = 1,4•10[7] kg/cm·s[2]

Symbols

Quantity	Unit	Symbol
Force	N	F_p
Pressure	bar	p
Section	cm²	A
Bore size	mm	D
Rod diameter	mm	d
Cylinder stroke	mm	c
Flow rate	l/min	Q
Speed	m/s	V
Acceleration	m/s²	a
Load mass	kg	m
Oil modulus of elasticity	kg/cm·s²	E
Total time at disposal	s	t_{tot}

3 **SIZING**

The table below reports the push/pull sections and forces for three different working pressures.
Once the push/pull forces are known, the size of the hydraulic cylinder can be choosen from the table below. The values have been determined using the formulas in section 2 .
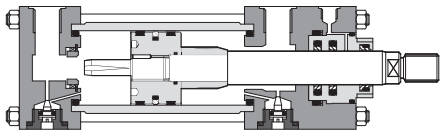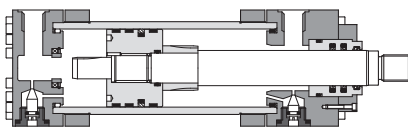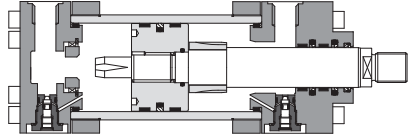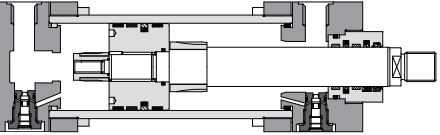
PULL FORCE [kN]

Bore [mm]		25		32		40			50			63			80			100		
Rod [mm]		12	18	14	22	18	22	28	22	28	36	28	36	45	36	45	56	45	56	70
A_2 Pulling area [cm²]		3,8	2,4	6,5	4,2	10,0	8,8	6,4	15,8	13,5	9,5	25,0	21,0	15,3	40,1	34,4	25,6	62,6	53,9	40,1
Pull force [kN]	p=100 bar	3,8	2,4	6,5	4,2	10,0	8,8	6,4	15,8	13,5	9,5	25,0	21,0	15,3	40,1	34,4	25,6	62,6	53,9	40,1
	p=160 bar	6,0	3,8	10,4	6,8	16,0	14,0	10,3	25,3	21,6	15,1	40,0	33,6	24,4	64,1	55,0	41,0	100,2	86,3	64,1
	p=250 bar	9,4	5,9	16,3	10,6	25,1	21,9	16	39,6	33,7	23,6	62,5	52,5	38,2	100,2	85,9	64,1	156,6	134,8	100,1

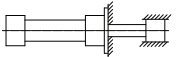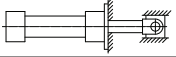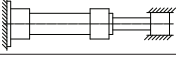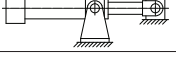| Bore [mm]		125			140	160			180	200			250		320		400	
Rod [mm]		56	70	90	90	70	90	110	110	90	110	140	140	180	180	220	220	280
A_2 Pulling area [cm²]		98,1	84,2	59,1	90,3	162,6	137,4	106,0	159,4	250,5	219,1	160,2	336,9	236,4	549,8	424,1	876,5	640,9
Pull force [kN]	p=100 bar	98,1	84,2	59,1	90,3	162,6	137,4	106,0	159,4	250,5	219,1	160,2	336,9	236,4	549,8	424,1	876,5	640,9
	p=160 bar	156,9	134,8	94,6	144,5	260,1	219,9	169,6	255,1	400,9	350,6	256,4	539,1	378,2	879,6	678,6	1.402,4	1.025,4
	p=250 bar	245,2	210,6	147,8	225,8	406,4	343,6	265,1	398,6	626,4	547,8	400,6	842,3	591,0	1.374,4	1.060,3	2.191,3	1.602,2

PUSH FORCE [kN]
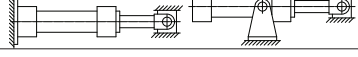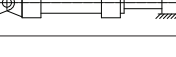
Bore [mm]		25	32	40	50	63	80	100	125	140	160	180	200	250	320	400
A_1 Pushing area [cm²]		4,9	8,0	12,6	19,6	31,2	50,3	78,5	122,7	153,9	201,1	254,5	314,2	490,9	804,2	1.256,6
Push force [kN]	p=100 bar	4,9	8,0	12,6	19,6	31,2	50,3	78,5	122,7	153,9	201,1	254,5	314,2	490,9	804,2	1.256,6
	p=160 bar	7,9	12,9	20,1	31,4	49,9	80,4	125,7	196,3	246,3	321,7	407,2	502,7	785,4	1.286,8	2.010,6
	p=250 bar	12,3	20,1	31,4	49,1	77,9	125,7	196,3	306,8	384,8	502,7	636,2	785,4	1.227,2	2.010,6	3.141,6

SERIES CK/CH - tab. B137 - B140 to ISO 6020-2	**SERIES CH BIG BORE SIZE - tab. B160 to ISO 6020-3**
- Nominal pressure **16** MPa (160 bar) - max. **25** MPa (250 bar) - Bore sizes from **25** to **200** mm - Rod diameters from **12** to **140** mm	- Nominal pressure **16** MPa (160 bar) - max. **25** MPa (250 bar) - Bore sizes from **250** to **400** mm - Rod diameters from **140** to **220** mm
SERIES CN - tab. B180 to ISO 6020-1	**SERIES CC - tab. B241 to ISO 6022**
- Nominal pressure **16** MPa (160 bar) - max. **25** MPa (250 bar) - Bore sizes from **50** to **200** mm - Rod diameters from **28** to **140** mm	- Nominal pressure **25** MPa (250 bar) - max. **32** MPa (320 bar) - Bore sizes from **50** to **320** mm - Rod diameters from **36** to **220** mm

5 CHECK OF THE BUCKLING LOAD

5.1 Calculation of the ideal lenght

Style	Rod end connection	Type of mounting	Fc
A, E, K, N, T, W, Y, Z	Fixed and rigidly guided		0,5
A, E, K, N, T, W, Y, Z	Pivoted and rigidly guided		0,7
B, P, V	Fixed and rigidly guided		1,0
G	Pivoted and rigidly guided		1,0
B, P, V, L	Pivoted and rigidly guided		1,5
A, E, K, N, T, W, Y, Z	Supported but not rigidly guided		2,0
C, D, H, S	Pivoted and rigidly guided		2,0
B, P, V	Supported but not rigidly guided		4,0
C, D, H, S	Supported but not rigidly guided		4,0

For cylinders working with push loads, the buckling load's checking has to be considered before choosing the rod size. This check is performed considering the fully extended cylinder as a bar having the same diameter of the cylinder rod (safety criteria):

1. determine the stroke factor "Fc" depending to the mounting style and to the rod end connection, see table at side

2. calculate the "ideal lenght" from the equation:

ideal length = Fc x stroke [mm]

If a spacer has been selected, the spacer's length must be added to the stroke

3. calculate the F_P push force as indicated in section 3 or using the formulae indicated in section 2

4. obtain the point of intersection between the push force and the ideal length using the rod selection chart 5.2

5. obtain the minimum rod diameter from the curved line above the point of intersection

5.2 Rod selection chart
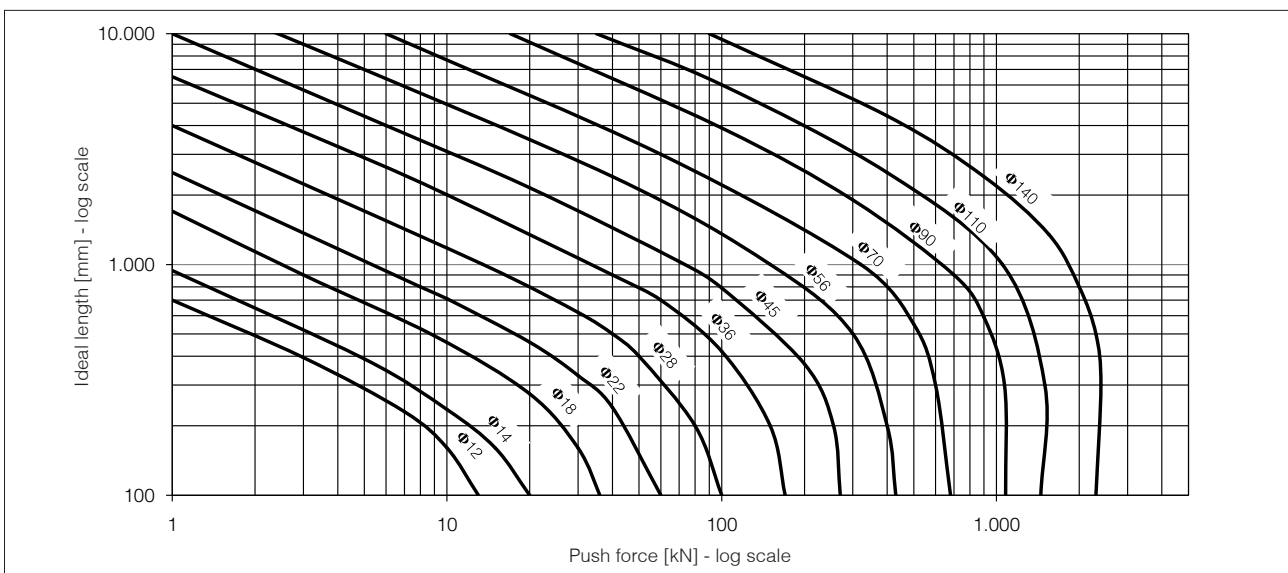
6 PREDICTION OF THE EXPECTED CYLINDER'S MECHANICAL WORKING LIFE

The rod thread is the cylinder's max critical part, thus the expected cylinder's working life can be evaluated by the prediction of the expected rod thread fatigue life. The fatigue rod fractures take place suddenly and without any warning, thus it is always recommended to check if the rod is subject to fatigue stress (not necessary if the cylinder works with push loads) and thus if the expected rod threads fatigue life may become an issue in relation to the required cylinder working life. The charts below do not include the rods which are fatigue-free for working pressures over 250 bar. The curves are referred to ideal working conditions and do not take into account misalignments and transversal loads that could decrease the predicted life cycles. The charts are intended valids for all the cylinders and servocylinders series with standard materials and sizes (section 6.2) or option **K** "Nickel and chrome plating" rods (section 6.3). For the evaluation of the expected fatigue life of stainless steel rods (CNX series), contact our technical office. For double rod executions the mechanical working life calculation does not apply to secondary rods since the thread is weaker than the primary rods.

6.1 Mechanical working life calculation procedure

1. Identify the curve of proper rods fatigue life graph according to the selected bore/rod size and rod treatment. Fatigue-free bore/rod couplings are not included in the graphs.

2. Intersect the working pressure with the curve corresponding to the rod under investigation and determine the expected rod life cycles. If the calculated rod fatigue life is lower than 500.000 cycles a careful analysis of our technical office is suggested.

6.2 Rods fatigue life charts for standard rod
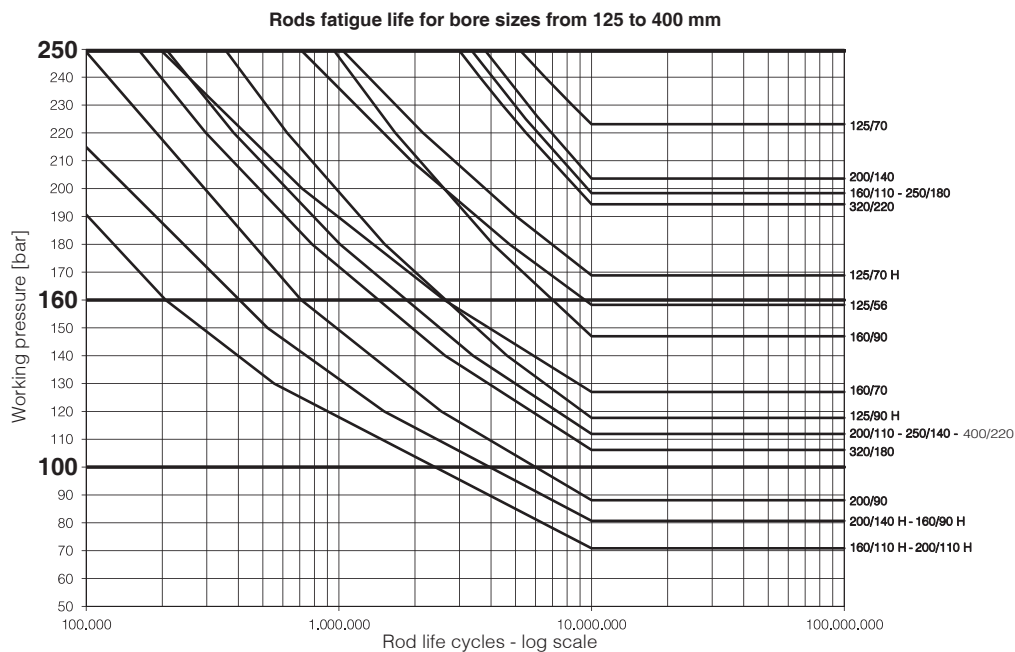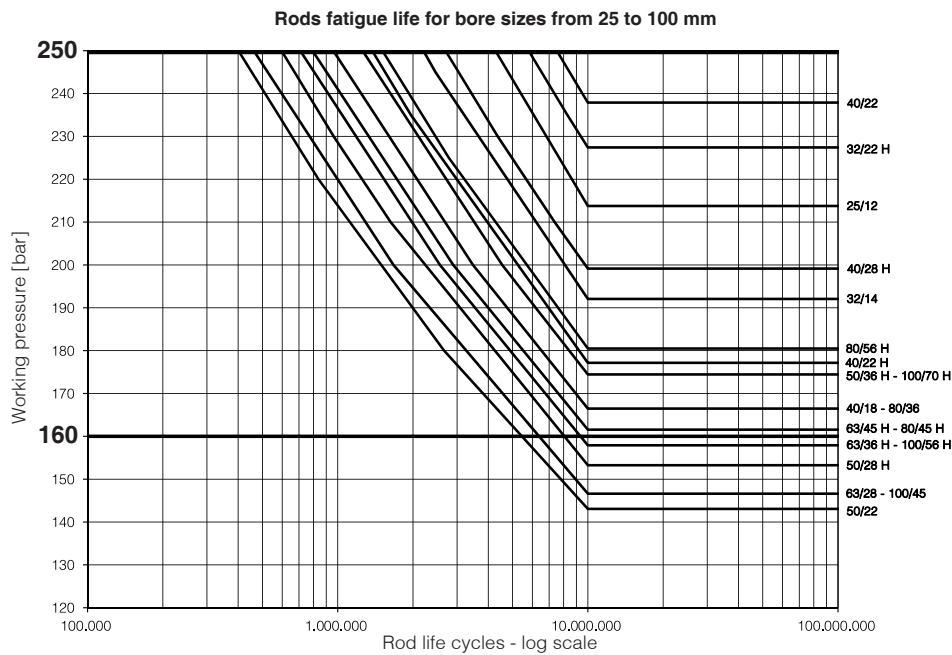
Rods fatigue life for bore sizes from 25 to 100 mm

Rods fatigue life for bore sizes from 125 to 400 mm

Note: the curves are labelled according to the bore/rod size. The light male thread (option **H**) is indicated by the "H" after the rod
Example: label **125/90 H** means bore = 125 mm, rod = 90 mm and rod with option **H**

6.3 Rods fatigue life charts for Nickel and Chrome plating rod (option K)
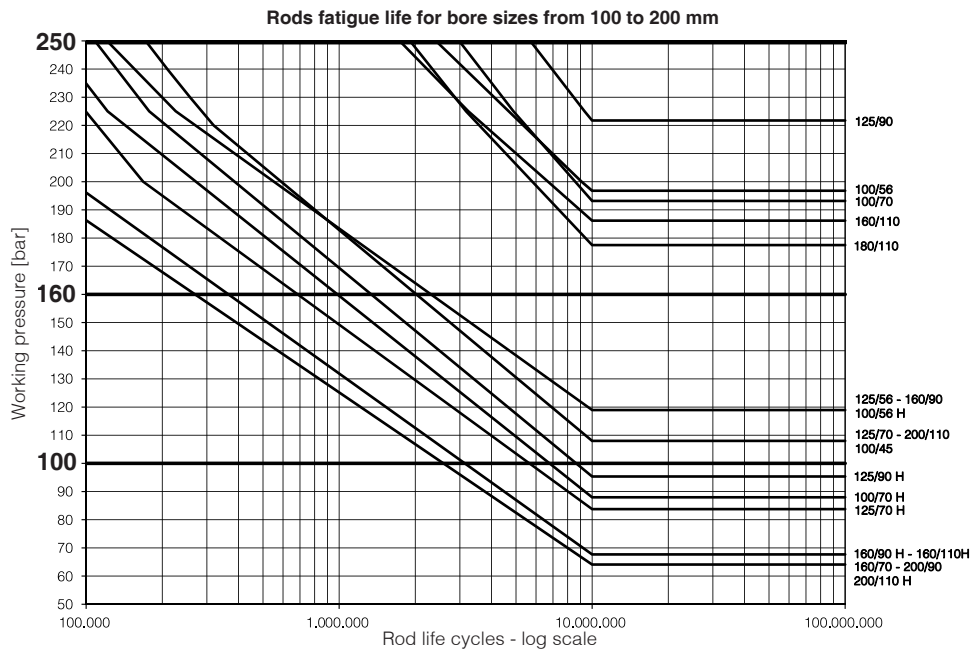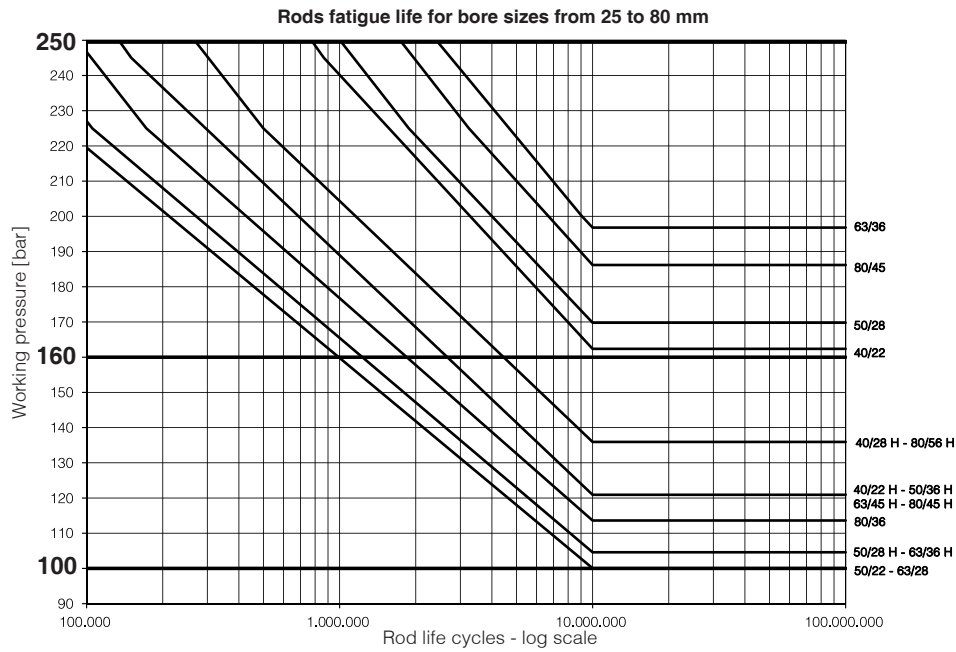
Rods fatigue life for bore sizes from 25 to 80 mm

Working pressure [bar] vs Rod life cycles - log scale

Curve labels (right side):
63/36
80/45
50/28
40/22
40/28 H - 80/56 H
40/22 H - 50/36 H
63/45 H - 80/45 H
80/36
50/28 H - 63/36 H
50/22 - 63/28

Rods fatigue life for bore sizes from 100 to 200 mm

Working pressure [bar] vs Rod life cycles - log scale

Curve labels (right side):
125/90
100/56
100/70
160/110
180/110
125/56 - 160/90
100/56 H
125/70 - 200/110
100/45
125/90 H
100/70 H
125/70 H
160/90 H - 160/110H
160/70 - 200/90
200/110 H

Note: the curves are labelled according to the bore/rod size. The light male thread (option **H**) is indicated by the "H" after the rod
Example: label **125/90 H** means bore = 125 mm, rod = 90 mm and rod with option **H**
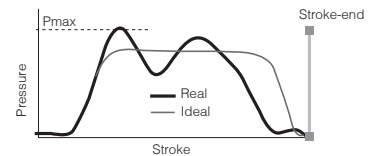
7 | CHECK OF THE HYDRAULIC CUSHIONING

7.1 Functioning features

Hydraulic cushioning act as "dumpers" to dissipate the energy of a mass connected to the rod and directed towards the cylinder stroke-ends, reducing its velocity before the mechanical contact, thus avoiding mechanical shocks that could reduce the average life of the cylinder and of the entire system.
Cushioning proves to be effective as much as the pressure inside the cushioning chamber gets close to the ideal profile described in the diagram at side. The diagram compares the ideal profile with typical cylinders real pressure profile.

7.2 Application features

The following guidelines refer to CK, CH, CN and CC cylinders: for CH big bore sizes, contact our technical office. In order to optimize the performances of cushioning in different applications, three different cushioning versions have been developed:

- slow version, with cushioning adjustment, for speed $V \leq 0,5 \cdot Vmax$
- fast version, without adjustment, for speed $V > 0,5 \cdot Vmax$
- fast version, with cushioning adjustment, for speed $V > 0,5 \cdot Vmax$

Adjustable cushioning are provided with needle valve to optimize the cushioning performances. The maximum permitted speed value Vmax depends to the cylinder size, see table below.

ø Bore [mm]	25	32	40	50	63	80	100	125	160	200
Vmax [m/s]	1	1	1	1	0,8	0,8	0,6	0,6	0,5	0,5

Pressure in the cushioning chamber

Speed during cushioning

7.3 Max energy calculation procedure

Check the max energy that can be absorbed by the selected cushioning as follow:

1. calculate the energy to be dissipated **E** by the algrebraic sum of the kinetic energy E_c and the potential energy E_P (for horizontal applications the potential energy is: $E_P = 0$)

$$E = E_c + E_P$$

- E_c (kinetic energy) due to the mass speed

$$E_c = 1/2 \cdot M \cdot V^2 \quad \text{[Joule]}$$

- E_P (potential energy) due to the gravity and related to the cylinder inclination angle α as shown at side

For front cushioning:

$$E_P = -Lf \cdot \frac{M \cdot g \cdot sen\, \alpha}{1000} \quad \text{[Joule]}$$

For rear cushioning:

$$E_P = + Lf \cdot \frac{M \cdot g \cdot sen\, \alpha}{1000} \quad \text{[Joule]}$$

2. identify the proper cushioning chart depending to the rod type, the cushioning side (front or rear), and the cylinder series (section 7.4 for CK, CH, CN cylinders or section 7.5 for CC cylinders)
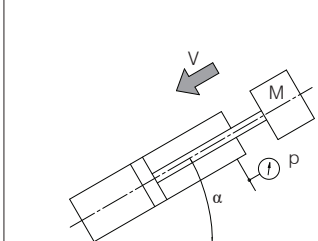
3. intersect the working pressure with the proper bore/rod size curve and extract the corresponding E_{max} value

4. compare the E_{max} value with the energy to be dissipated **E** and verify that:

$$E \le E_{max}$$

5. for critical applications with high speed and short cushioning strokes an accurate cushioning evaluation is warmly suggested, contact our technical office

Symbols

E	= energy to be dissipated	[J]
E$_{max}$	= energy max dissipable	[J]
M	= mass	[kg]
V	= rod speed	[m/s]
Lf	= cushioning length	[mm]
(see section 12 of tables B137, B140)		
g	= acceleration of gravity	[m/s²]
consider g=9,81 m/s²		
α	= inclination angle	[°]

7.4 Cushioning charts for CK - CH - CN cylinders

Notes:
- the front cushioning graphs are labelled according to the bore/rod size, the rear cushioning graph is labelled according to the bore size
- the curves are intended valid for mineral oil ISO 46 and a fluid temperature of 40-50 °C: the use of water or water-based fluids and higher/lower temperatures can affect the cushioning performance because of high viscosity variations respect to standard mineral oil
- for adjustable versions the E_{max} value is referred to cushioning cartridge fully closed, the max energy to be dissipated may be increased opening the cushioning cartridge, thus reducing the max pressure reached in the cushioning chamber
- the cushioning charts have been determined with 250 bar maximum pressure admitted in the cushioning chamber

Front cushioning - standard rods

Front cushioning - intermediate & differential rods
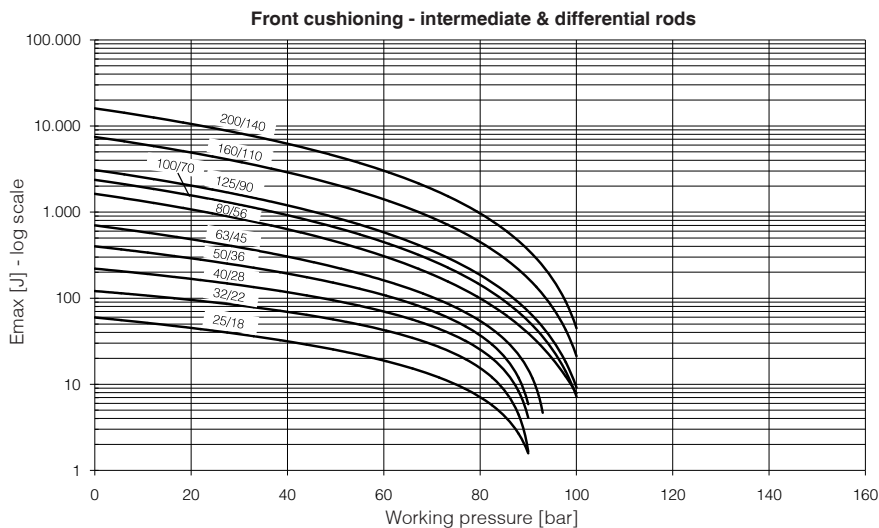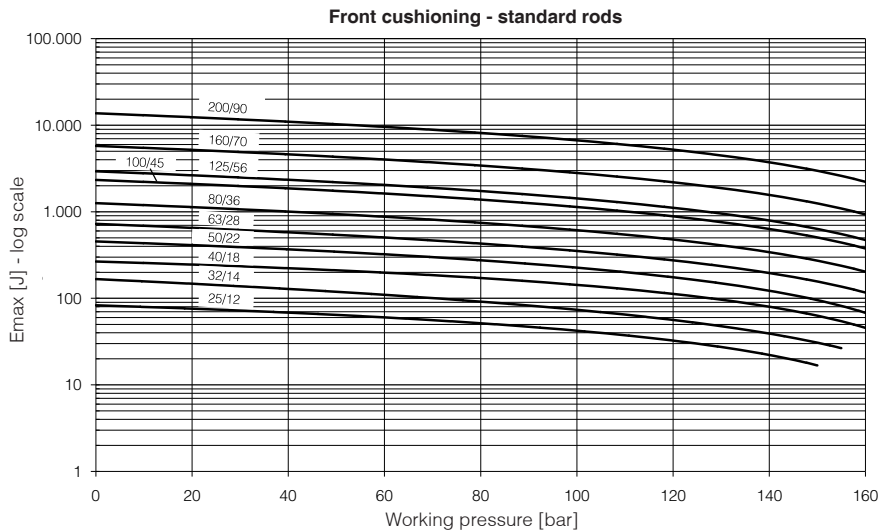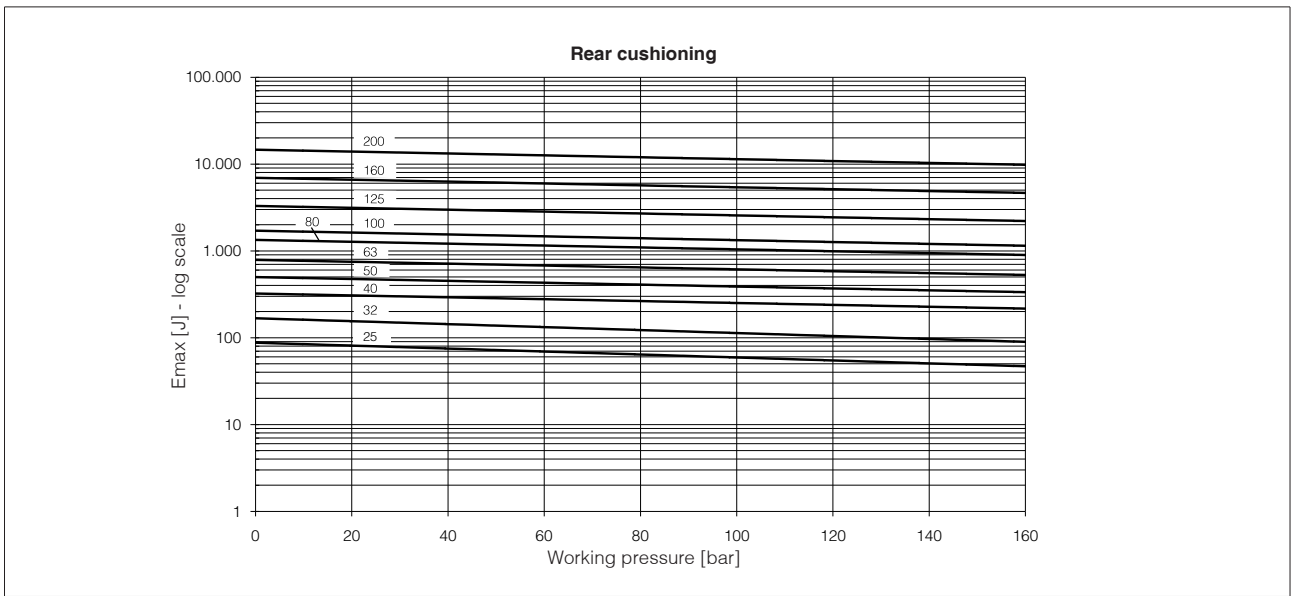
Rear cushioning

7.5 Cushioning charts for CC cylinders

Notes:
- the front cushioning graphs are labelled according to the bore/rod size, the rear cushioning graph is labelled according to the bore size
- the curves are intended valid for mineral oil ISO 46 and a fluid temperature of 40-50 °C: the use of water or water-based fluids and higher/lower temperatures can affect the cushioning performance because of high viscosity variations respect to standard mineral oil
- for adjustable versions the E_{max} value is referred to cushioning cartridge fully closed, the max energy to be dissipated may be increased opening the cushioning cartridge, thus reducing the max pressure reached in the cushioning chamber
- the cushioning charts have been determined with 320 bar maximum pressure admitted in the cushioning chamber
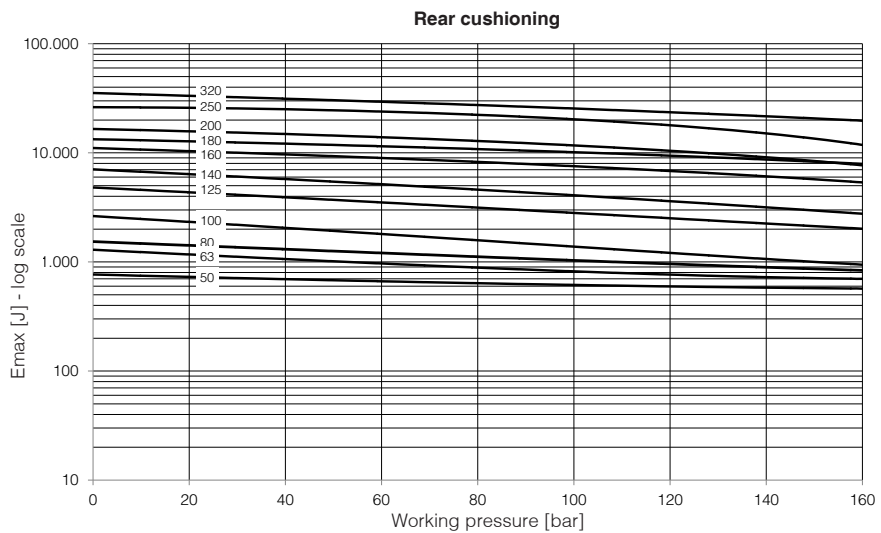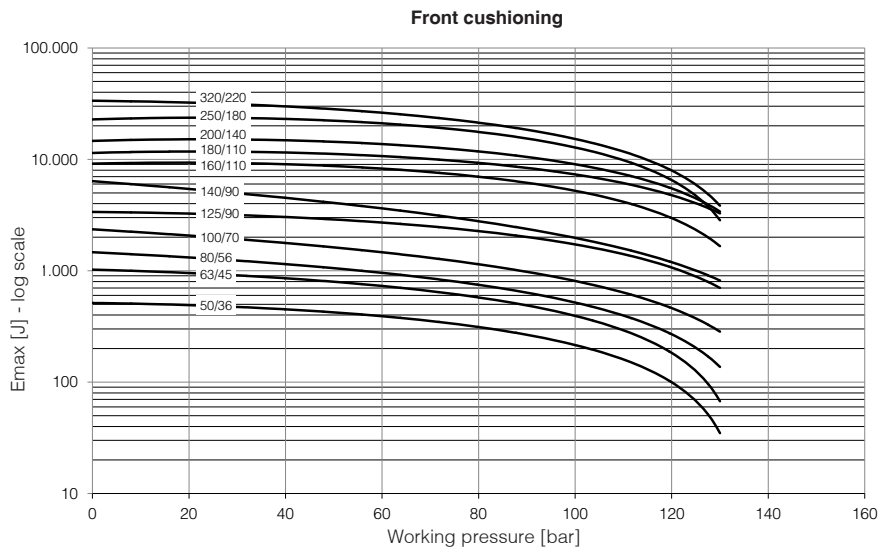
Front cushioning

Rear cushioning

8 SEALING FRICTION AND IN / OUT SPEED RATIO

Basic sealing performances reported in the cylinders technical tables are not sufficient for a comprehensive evaluation of the sealing system, the following sections report additional verifications about minimum in/out rod speed ratio, static and dynamic sealing friction.

8.1 In / out speed ratio

Applications with low in/out rod speed ratio may involve leakages caused by partial "back pumping" of the oil trapped between the rod seals, thus it is recommended to check the correct back pumping with the diagram reported below.

1. Determine the in/out speed ratio **R** of the cylinder

$$R = \frac{V_{in}}{V_{out}} = \frac{Q_2 \cdot A_1}{A_2 \cdot Q_1}$$

2. Intersect the working pressure with the curve below and extract the corresponding R_{min} value admitted
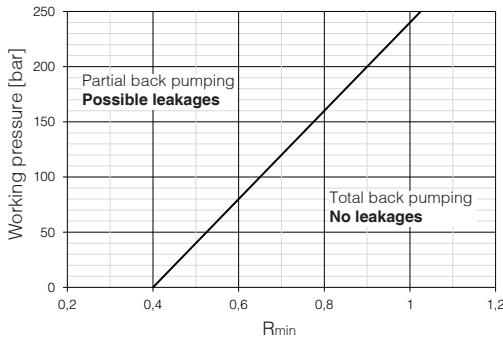
3. Verify that

$$\boxed{\mathbf{R} \geq \mathbf{R_{min}}}$$

If the equation above is not verified contact our technical office

8.2 Static and dynamic sealing friction

Sealing systems may affect the smooth rod motion, thus the assessment of the sealing friction forces is recommended in several applications like :

- Servoactuators with closed loop control
- Servocylinders where high accuracy in rod positioning is required
- Cylinders with low speeds (<0,05 m/s)
- Low pressure hydraulic systems (<10 bar) where sealing friction forces may have significant influence

The following sections allow to calculate both static and dynamic sealing friction according to the sealing system selected for CK, CH and CK* servocylinders.

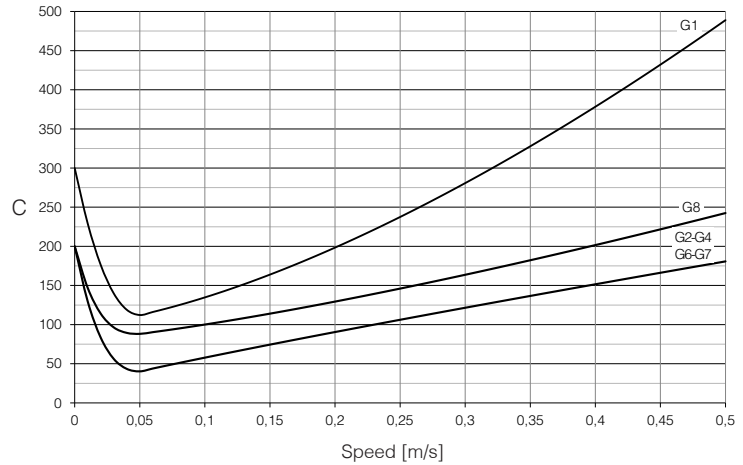### 8.3 Sealing friction calculation procedure

Calculate the **dynamic** sealing friction as follow:

1. Intersect the speed with the proper curve depending to the sealing system from the chart in section 8.4.

2. Extract the corresponding **C** value

3. Identify the proper diagram according to the sealing system (section 8.5)

4. Intersect the working pressure with the curve depending to the Bore size.

5. Extract the corresponding **A** value

6. $F_{sf} = \mathbf{A} \cdot (D + d) + \mathbf{C}$ [N]
considering D= Bore size [mm]; d= Rod size [mm]

Calculate the **static** sealing friction as follow:

1. Extract the **C** value corresponding to speed **V = 0 m/s** in the chart in section 8.4

2. Identify the proper diagram according to the sealing system (section 8.5)

3. Intersect the working pressure with the curve depending to the Bore size.

4. Extract the corresponding **A** value

5. $F_{sf} = \mathbf{A} \cdot (D + d) + \mathbf{C}$ [N]
considering D= Bore size [mm]; d= Rod size [mm]

8.4 Friction charts - C parameter vs speed

8.5 Friction charts - A parameter vs pressure

SEALING G1

SEALING G2 - G4 - G6 - G7

SEALING G8