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Gear Motor Selection

BAUER GEAR MOTOR
A REGAL REXNORD BRAND

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Information
Company: ____________________
Contact person: ____________________
Phone: ____________________
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Questionnaire for geared motor selection

Gearbox type

☐ BG
Helical gears

☐ BF
Parallel shaft gears

☐ BK
Bevel gears

☐ BS
Worm gears

☐ Hiflex
☐ Standard
☐ Stainless

Number of items: __________
Country of operation: __________

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Technical Data

Output shaft speed n2: __________ rpm
Torque M2: __________ Nm
Motor power: __________ kW
Efficiency class: __________
With pole-changing: __________

Voltage: __________ V
Connection: ☐ Y ☐ △
Temperature class: ☐ B ☐ F ☐ H
Frequency: ☐ 50 Hz ☐ 60 Hz
☐ Frequency inverter duty
Frequency range: __________ Hz

Operation

Service factor required: min __________ max __________

or Type of loads (conveyor, mixer, crusher, centrifuge, etc.): ____________________
Number of switching per hour: __________

Daily operating time: ☐ 8 hour ☐ 16 hour ☐ 24 hour
☐ Light shock load ☐ Medium shock load ☐ Heavy shock load
Operation Duty: ☐ S1 ☐ S2 ______ min ☐ other S ______ Duty = __________ %

Output shaft design

☐ Solid shaft on side V/H/VH: __________
☐ Hollow shaft with keyway
☐ Special shaft dimensions (DxL), ______ x ______ mm
☐ Second shaft end on motor (DxL), ____ x ____ mm
☐ Solid shaft without parallel key
☐ Hollow Shaft for shrink disk
☐ Shrink disk
☐ Second shaft end on motor with parallel key (ZW)
☐ Second shaft end on motor with square shaft (ZV)
☐ Other (sketch attached)
☐ Splined Shaft acc. to DIN 5480

Mounting position

☐ Foot with clearance hole
☐ Foot with tapped holes
☐ Torque arm with rubber buffers
☐ A-Flange
☐ C-Flange with tapped holes
☐ Foot plate
☐ rear ☐ bottom
☐ front ☐ top
☐ left ☐ right

Mounting (acc. to page. 2 - H1, H2, V1, V2, etc.): ____________________
Terminal box position (acc. to page. 3): ☐ I ☐ II ☐ III ☐ IV
other: ____

Painting
☐ Standard RAL 7031
☐ other RAL __________

Environment

IP prot. type per EN 60034: ☐ IP54 ☐ IP65 ☐ IP66 ☐ IP67 ☐ IP68 ☐ IP69K
☐ Indoor installation ☐ Outdoor installation ☐ Corrosive environment: __________
Ambient temperature range: from ______ °C to + ______ °C Relative humidity: ______ %

Motor Accessories

☐ Brake voltage: __________ V
☐ Backstop: ☐ left ☐ right
☐ Thermistor motor protection
☐ Temperatursensor KTY
☐ Encoder type __________
☐ Rain cover

Required braking torque: ______ Nm
☐ Brake wear/function monitoring
☐ Thermostats motor protection
☐ Temperatursensor PT100
No. of pulse: __________
☐ Forced ventilation

☐ Manual brake release
☐ Brake heater
☐ Anticondensation heater (Motor)
Supply voltage ☐ HTL \ ☐ TTL

Additional requirements may be specified in a freely written form.

Drive configuration

Drive configuration General

Motions are necessary in production plants and equipment for the manufacture of goods and products. Geared motors are used to implement these motions in stationary production equipment. The objective of drive configuration is to obtain the optimal motor for each type of motion.

Motions in machines and equipment vary considerably. Experienced design engineers reduce the necessary motions to a few standard types:

- continuous linear motion
- reciprocating linear motion
- horizontal linear motion
- vertical or oblique linear motion for lifting and lowering loads
- continuous rotary motion and reciprocating rotary motion

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All motions can be divided into:

- an acceleration phase
- a constant-velocity phase
- a braking (deceleration) phase

These motion phases must be examined separately when sizing a drive, in order to determine the phase with the highest load. After the maximum load has been determined, the drive system can be selected.
See our separate "Design Guide" publication for assistance with various use cases.

Required data for drive configuration

In addition to the data on (Specification of geared motors), the following data is necessary for drive configuration:

Designation	Description	Unit
Z	Cycle rate	[1/h]
t_d	Operating time per day	[h]
t_a	Deceleration time	[s]
n_2	Output speed	[rpm]
n	Rated rotor shaft speed	[rpm]
J	Moment of inertia	[kgm²]
J_{ext}	External moment of inertia	[kgm²]
J_{ext}	External moment of inertia referred to the rotor shaft	[kgm²]
J_{rot}	Rotor moment of inertia	[kgm²]
F	Force	[N]
m	Mass	[kg]
v	Velocity	[m/s]
a	Acceleration	[m/s²]
g	Earth gravitational constant	[m/s²]
P_{dyn}	Dynamic power	[kW]
P_s	Static power	[kW]
P	Power	[kW]
M_2	Output torque	[Nm]
M_{2erf}	Required drive torque	[Nm]
M_N	Rated torque at rotor shaft	[Nm]
M_a	Deceleration torque	[Nm]
M_L	Braking or driving load torque	[Nm]
M_{gr}	Specific limiting torque of gearbox at gear ratio i	[Nm]
M_{Br}	Rated braking torque	[Nm]
i	Gear reduction ratio	
FI	Inertia ratio	

Drive configuration process

Motor configuration

Determining the motor power

The required power can generally be calculated as follows:

$$P = \frac{F \times v}{\eta}$$

As previously described, all motions are divided into an acceleration phase (dynamic power), a constant-velocity phase (static power), and a braking (deceleration) phase.

Depending on the type of motion, the force F necessary to overcome all opposing forces such as rolling friction, linear friction, gravitational force, acceleration and so on arising from the drive train has a strong influence on the required power and must be determined explicitly for each use case.

See Section 15 for assistance in selecting the right motor power.

Determining the required torque

After the motor power has been determined, the required gearbox output torque can be calculated with:

$$M_2 = \frac{P \times 9550}{n_2}$$

Determining the gear reduction ratio

The gear reduction ratio is the ratio of the rated speed of the motor (see the motor data in Section 15) to the desired output speed of the geared motor.

$$i = \frac{n}{n_2}$$

Gearbox size selection

Determining the factor of inertia

The inertia ratio is the ratio of the sum of the moments of inertia of all masses driven by the motor and converted to the motor speed, including the moment of inertia of the motor rotor, to the moment of inertia of the rotor:

$$FI = \frac{J_{ext'} + J_{rot}}{J_{rot}} \quad \text{where} \quad J_{ext'} = \frac{J_{ext}}{i^2}$$

Determining the shock load

The shock load (see Sections 6, 7, 8 and 9) is determined from the inertia factor, the type of transmission component and the relative moment of acceleration.

Determining the minimum service factor f_{Bmin}

Based on the operating time per day, the cycle rate and the ascertained shock load, the service factor f_{Bmin} can be taken from the tables in Sections 6, 7, 8 and 9.

Based on this minimum service factor f_{Bmin}, select a geared motor from the tables that has a higher service factor as well as the required output speed, output torque and motor power.

Note: The service factor relates solely to the required torque for static operation needed by the application, which should be covered by the output torque of the selected geared motor.
The dynamic portion is not taken into consideration here.

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The actual service factor of the geared motor with regard to required torque for static operation can therefore be calculated as follows:

$$f_B = \frac{M_{gr}}{M_{2erf}}$$

The final step is to specify the accessory options for the geared motor.

Brake specification

Essentially it is necessary to determine, based on the amount of friction energy to be dissipated by the brake, whether the brake is a holding brake or a service brake.
See Section 16 for the definitions of holding brakes and service brakes.

Once all the necessary data and requirements are known, the required braking torque can be calculated as follows:

$$M_{br} = M_a \pm M_L$$

$$M_a = \frac{J \times n}{9{,}55 \times t_a}$$

If the specific application data is not known, for horizontally driven equipment we recommend selecting a braking torque that is 1.0 to 1.5 times the rated torque of the motor.

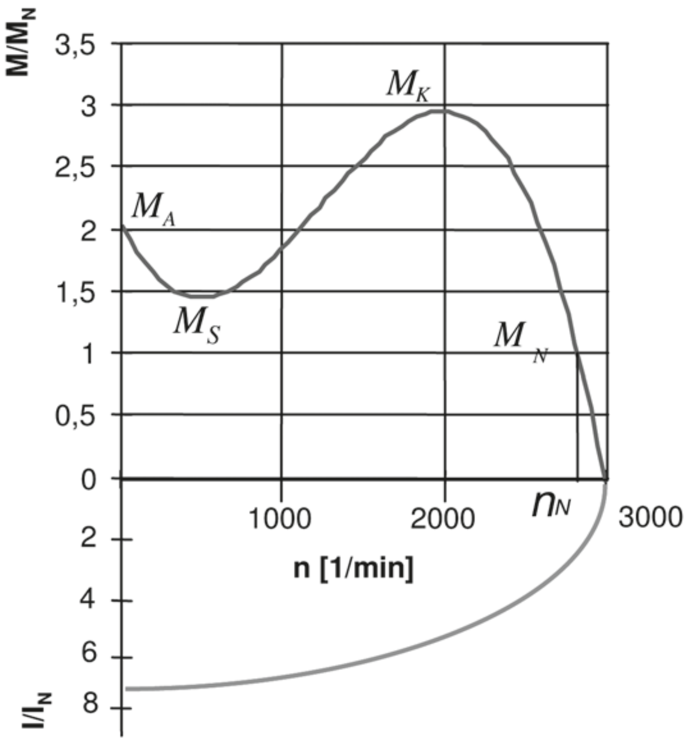
In the case of applications with significant external moments of inertia (FI greater than 2) and with operating cycles per hour, the brake size must always be selected on the basis of the thermally allowable braking energy. See Section 16 for detailed information on brake configuration.

In the case of lifting equipment, for safety reasons a braking torque twice as large as the rated torque of the motor should always be selected.

Torque–speed characteristic

The torque versus speed curve shows the operating characteristics of the asynchronous motor. The reference points shown schematically on the torque versus speed curve are significant criteria for motor selection.

Torque vs. Speed Curve



The **starting torque MA** with the rotor stationary, which is also called the locked-rotor torque, determines the acceleration of the equipment or system. If the motor is powered directly from the mains, bear in mind that the starting torque, usually listed in the motor data tables in the form of the ratio M_A/M_N, is a fixed and unalterable quantity. This means that the desired acceleration can only be approximated when the motor is operated directly from the mains. Operation from a frequency converter is discussed separately.

The **pull-up torque MS** is the least amount of torque developed by the motor while it is coming up to speed. It must always be greater than the effective load torque at the time when the pull-up torque occurs, as otherwise it will not be possible to accelerate the drive.

The **breakdown torque MK** is the maximum torque the motor is capable of producing. If the load increases above the rated torque M_n, the slip s increases, the speed n decreases, and the motor delivers more torque. This can rise to a maximum level M_K. After this point the motor stalls, which means that it suddenly stops running at this slip value (breakdown slip). If the breakdown torque is exceeded, either the load must be removed or the motor must be switched off immediately. Otherwise the motor will be destroyed as a result of overheating.

The **rated torque MN** is the torque available in continuous operation at the rated power P_N and rated speed n_N.

Gear Motor Selection

Motor configuration

Dynamic power

The dynamic power is the power that accelerates the entire system, which consists of the load, transmission components, gearbox and motor.

$$P_{dyn} = \frac{m \times a \times v}{\eta}$$

P_{dyn}	Dynamic power [W]
m	Mass [kg]
a	Acceleration [m/s²]
v	Velocity [m/s]
η	Efficiency

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Static power

The static power includes all forces present under zero-acceleration conditions. This includes rolling friction, linear friction, lifting force (with lifting) and wind force, among others.

$$P_S = \frac{F_F \times v}{\eta}$$

P_S	Static power [W]
F_F	Travel resistance [N]

Total power P_G

$$P_G = P_{dyn} + P_S$$

$$P_G = \frac{m \times a \times v}{\eta} + \frac{F_F \times v}{\eta}$$

Horizontal motion, rotary motion and vertical motion upwards	
Start-up time [s]	$t_A = \dfrac{\left(J_M + \dfrac{J_{ext}}{\eta}\right) \times n_M}{9{,}55 \times \left(M_A - \dfrac{M_L}{\eta}\right)}$
Cycle rate [c/h]	$Z = Z_0 \times \dfrac{1 - \left(\dfrac{M_L}{M_A \times \eta}\right)}{\left(\dfrac{J_S + \dfrac{J_{ext}}{\eta} + J_M}{J_M}\right)} \times K_L$
Vertical motion downwards	
Start-up time [s]	$t_A = \dfrac{\left(J_M + \dfrac{J_{ext}}{\eta}\right) \times n_M}{9{,}55 \times \left(M_A - (M_L \times \eta)\right)}$
Cycle rate [c/h]	$Z = Z_0 \times \dfrac{1 - \left(\dfrac{M_L \times \eta}{M_A}\right)}{\left(\dfrac{J_S + J_M + (J_{ext} \times \eta)}{J_M}\right)} \times K_L$

Motor selection

Example:

Required dynamic torque at motor (for acceleration):	126 Nm
Required static torque at motor	70.0 Nm
Total torque at motor:	196 Nm

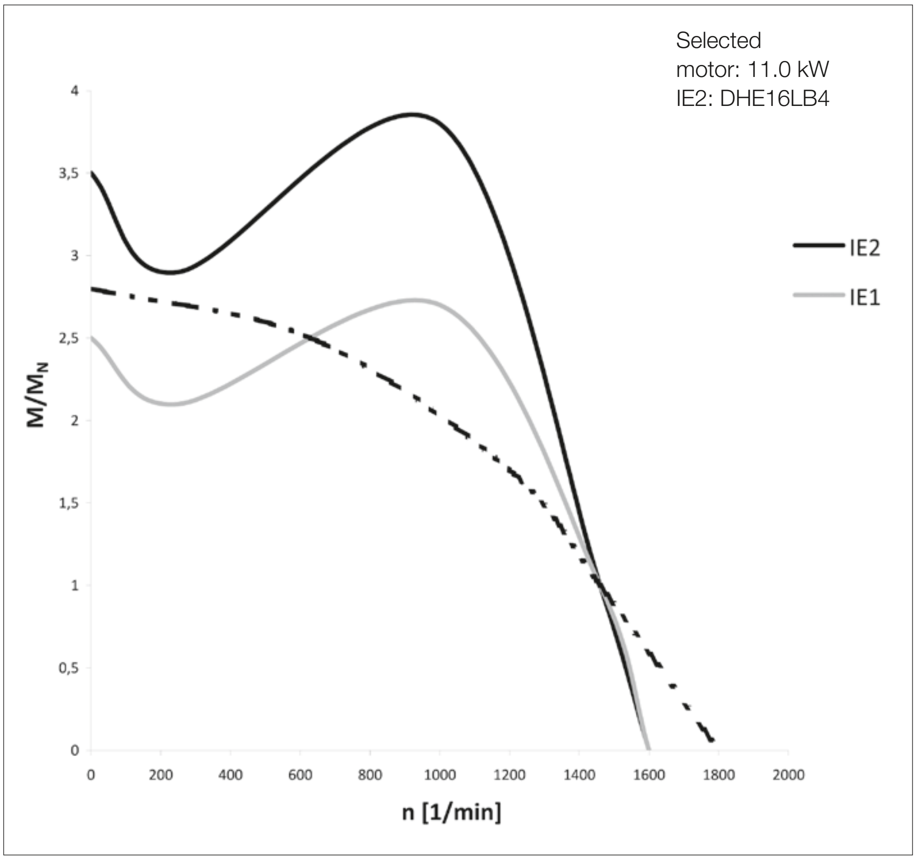
IE2

P_N [kW]	Type	n_N [rpm]	M_N [Nm]	I_N 400 V [A]	cos φ	η (100% load) [%]	η (75% load) [%]	η (50% load) [%]	I_A/I_N	M_A/M_N	M_S/M_N	M_K/M_N	J_{rot} [kgm^2]
7.5	DHE13LA4	1460	49	15.1	0.81	88.9	89.2	87.9	7.0	3.3	3.0	3.5	0.0345
9.5	DHE16MB4	1470	62	19.7	0.78	89.4	89.4	86.5	6.8	2.9	2.5	3.2	0.057
11	DHE16LB4	1470	71	22.5	0.78	90.3	90.0	88.3	7.9	3.5	2.9	3.8	0.076
15	DHE16XB4	1470	97	31	0.77	90.6	90.8	88.8	7.2	3.2	2.8	3.5	0.087
18.5	DHE18LB4	1470	120	35	0.83	91.5	91.7	90.0	7.9	3.6	3.0	3.3	0.160

IE1

P_N [kW]	Type	n_N [rpm]	M_N [Nm]	I_N 400 V [A]	cos φ	η (100% load) [%]	η (75% load) [%]	η (50% load) [%]	I_A/I_N	M_A/M_N	M_S/M_N	M_K/M_N	J_{rot} [kgm^2]
7.5	DSE13MA4	1440	50	15.3	0.81	87.5	87.8	87.1	6.2	2.8	2.5	3.2	0.02900
9.5	DSE13LA4	1440	63	19.2	0.82	87.1	87.5	87.5	6.0	2.9	2.6	3.0	0.03450
11	DSE16MB4	1460	72	22.6	0.81	87.7	88.0	87.3	6.0	2.5	2.1	2.7	0.05700
15	DSE16LB4	1460	98	29.5	0.83	88.9	89.2	88.9	6.1	2.5	2.1	2.8	0.07600
18.5	DSE16XB4	1460	121	37.5	0.81	89.3	89.9	88.5	6.1	2.6	2.2	2.8	0.08700

Due to the significantly higher starting torque (M_A) of IE2 motors (M_A/M_N 3.5) compared to IE1 motors (M_A/M_N 2.5), an 11 kW with an IE2 (DHE16LB4) motor can be used in this case. Otherwise the 15 kW IE1 (DSE16LB) should be selected.



No-load cycle rate Z_0

If the cycle rate is greater than normal (typically around 60 cycles per hour), the additional thermal load and, depending on the type of power transmission, the additional mechanical load must be taken into account in motor selection.

The no-load cycle rate Z_0 is the number of start cycles per hour with the motor running under no load (no external moments of inertia) in which the allowable winding temperature for the insulating material class F is reached.

No-load cycle rate Z_0:

PN [kW]	Type	Z0 [c/h]
0.12	DPE05LA4	65000
0.12	DPE06LA4	65000
0.18	DPE07LA4	47000
0.25	DPE08MA4	36000
0.37	DPE08LA4	27000
0.55	DPE08XA4	19000
0.75	DPE09LA4	15000
1.1	DPE09XA4	11000
1.5	DPE09XA4C	8700
2.2	DPE11MA4	6400
3	DPE11LA4	5000
4	DPE11LA4C	4000
5.5	DPE13LA4	3100
7.5	DPE13XA4	2400
9.5	DPE16LB4	2000
11	DPE16LB4	1800
15	DPE16XB4	1400
18.5	DPE18LB4	1200
22	DPE18XB4	1000

As a result of external loads, the no-load cycle rate is reduced to the allowable service cycle rate. The effect of the load is expressed by the inertia ratio FI and the load factor K_L.

Load factor K_L

The load factor reflects the relative load *P/PN* and the duty cycle of the motor in operation between the cycles.
The relative load has a quadratic effect on the allowable cycle rate. The effect of the duty cycle depends on the circumstances. With little or no load, the stress on the motor decreases due to the relatively long cooling periods, while at rated load or heavy loading the stress on the motor increases due to load losses.

The load factor K_L for 4-pole motors is determined as follows:

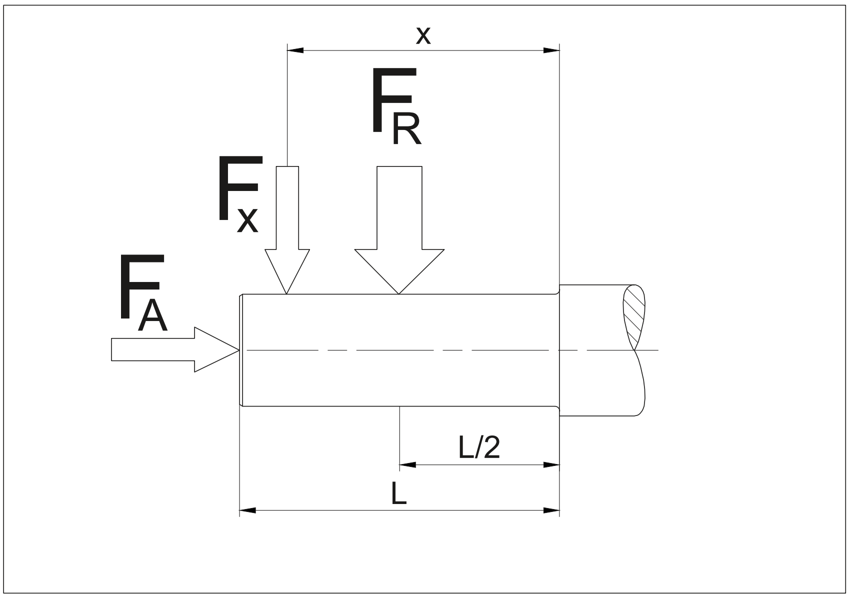
$$K_{L100} = 1 - \left(\frac{P}{P_n}\right)^{1,5}$$

$$K_L = 0{,}35 + (K_{L100} - 0{,}25) \times ED$$

Radial and axial forces on the output shaft

For each geared motor with a solid shaft, the allowable radial force $F_{R\,(N,V)}$ referred to the centre of the output shaft, x = l/2, is listed in the selection tables. The listed data applies to both foot-mounted and flange-mounted versions. If the force application point F_X is off centre, the allowable radial force must be recalculated taking into account the bearing lifetime and the shaft strength.

Maximum allowable radial force at force application point X



$F_{R(N,\,V)}$	Allowable radial force (x = l/2) according to the selection tables [N]
X	Distance from shaft junction to the force application point [mm]
F_A	Axial force [N]

To evaluate the radial force present at the force application point X, the allowable radial forces at position X must be determined with respect to the load limits of the bearings and the shaft strength.

If the calculated allowable radial forces at the force application point X are greater than the radial force that is present, the gearbox may be selected for the application.
If the calculated values are not sufficient or the force application point X is not within the stub shaft length l, please consult us.

Bearing load limit

$$F_{XL1} = F_q \times \frac{0,5 + b}{\left(\frac{X}{l} + b\right)}$$

$$F_{XL2} = F_q \times \frac{0,5 + a}{\left(\frac{X}{l} + a\right)}$$

Radial and axial forces on the output shaft

Shaft strength

$$F_{XW1} = F_{qmax} \times \frac{0,5}{\left(\frac{X}{l}\right)}$$

$$F_{XW2} = F_{qmax} \times \frac{0,5 + c}{\left(\frac{X}{l} + c\right)}$$

For the selected gear ratio and bearing type (normal or reinforced), F_q is the allowable perpendicular force F_{RN} or F_{RV} from the geared motor selection tables.

F_{qmax} is the maximum allowable perpendicular force for the selected gearbox size as listed in the geared motor selection tables, independent of the bearing type (normal or reinforced).

The factors a, b and c for the individual gearbox types are listed in the following tables.

Helical gear unit BG series

Taille	Paliers	Arbre Code	l	a	b	c
BG04	normaux	-.1	24	0.5625	1.5000	-
BG05	normaux	-.1	28	0.5893	1.3929	-
BG06	normaux	-.1	30	0.6667	1.4167	-
BG10	normaux	-.1	40	0.7125	1.6750	-
		-.7		1.1000	2.0625	-
BG20	normaux	-.1	50	0.6100	2.2500	-
		-.7		0.9400	2.5800	-
BG30	normaux	-.1	60	0.5917	2.1750	-
		-.7		0.9417	2.5250	-
BG40	normaux	-.1	60	0.6917	2.3667	-
		-.7		1.0083	2.6833	-
BG50	normaux	-.1	80	0.5625	2.0000	-
		-.7		0.8563	2.2938	-
BG60	normaux	-.1	100	0.5300	2.0200	-
		-.7		0.7650	2.2550	-
BG70	normaux	-.1	120	0.4750	1.7292	-
		-.7		0.7292	1.9833	-
BG80	normaux	-.1	140	0.4286	1.7000	-
		-.7		0.6000	1.8714	-
BG90	normaux	-.1	200	0.3675	1.5300	-
		-.7		0.5825	1.7450	-
BG100	normaux	-.1	220	0.3477	1.4341	-
		-.7		0.5386	1.6250	-

Shaft-mounted gear unit BF series

Frame size	Bearings	Output shaft code	l	a	b	c
BF06	normal	-.1	50	0.4500	1.4100	-
BF10	normal	-.1	60	0.5083	1.4833	-
		-.2		0.6500	1.6250	-
BF20	normal	-.1	70	0.4286	1.3571	-
		-.2		0.5571	1.4857	-
BF30	normal	-.1	80	0.3875	1.2563	-
		-.2		0.5688	1.4375	-
BF40	normal	-.1	100	0.4050	1.2250	-
		-.2		0.5250	1.3450	-
BF50	normal	-.1	120	0.3125	1.0625	-
		-.2		0.3959	1.1458	-
BF60	normal	-.1	140	0.3286	1.0821	-
		-.2		0.4036	1.1571	-
	reinforced	-.1		-	-	0.2750
		-.2		-	-	0.3643
BF70	normal	-.1	180	0.2722	1.0566	-
		-.2		0.3056	1.0889	-
	reinforced	-.1		-	-	0.2194
		-.2		-	-	0.2639
BF80	normal	-.1	220	0.2878	1.3536	-
		-.2		0.2873	1.3518	-
	reinforced	-.1		-	-	0.2364
		-.2		-	-	0.2268
BF90	normal	-.1	260	0.2500	1.4231	-
		-.2		0.2500	1.4231	-
	reinforced	-.1		-	-	0.2027
		-.2		-	-	0.1950

Radial and axial forces on the output shaft

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Bevel gear unit BK series

Frame size	Bearings	Output shaft code	l	a	b	c
BK06	normal	-.1	50	0.4375	1.9875	-
		-.2		0.4375	1.9875	-
		-.7		0.9125	2.4625	-
		-.8		0.9125	2.4625	-
BK10	normal	-.1	60	0.5917	2.2417	-
		-.2		0.5917	2.2417	-
BK20	normal	-.1	70	0.5071	2.2357	-
		-.2		0.5071	2.2357	-
	reinforced	-.1		-	-	0.3929
		-.2		-	-	0.3929
BK30	normal	-.1	80	0.5250	2.2750	-
		-.2		0.5250	2.2750	-
	reinforced	-.1		-	-	0.4125
		-.2		-	-	0.4125
BK40	normal	-.1	100	0.4300	2.1700	-
		-.2		0.4300	2.1700	-
	reinforced	-.1		-	-	0.3400
		-.2		-	-	0.3400
BK50	normal	-.1	120	0.4083	1.9417	-
		-.2		0.4083	1.417	-
	reinforced	-.1		-	-	0.3250
		-.2		-	-	0.3250
BK60	normal	-.1	140	0.3536	1.8036	-
		-.2		0.3536	1.0836	-
	reinforced	-.1		-	-	0.3121
		-.2		-	-	0.2979
BK70	normal	-.1	180	0.2861	1.6694	-
		-.2		0.2861	1.6694	-
	reinforced	-.1		-	-	0.2428
		-.2		-	-	0.2317
BK80	normal	-.1	220	0.2818	1.5545	-
		-.2		0.2818	1.5545	-
	reinforced	-.1		-	-	0.2305
		-.2		-	-	0.2214
BK90	normal	-.1	260	0.2519	1.6096	-
		-.2		0.2519	1.6096	-
	reinforced	-.1		-	-	0.1989
		-.2		-	-	0.1912

Worm gear unit BS series

Frame size	Bearings	Output shaft code	l	a	b	c
BS02	normal	-.1	30	0.6000	2.1000	-
		-.2		-	-	-
		-.7		1.3333	2.8333	-
		-.8		-	-	-
BS03	normal	-.1	40	0.4375	1.9875	-
		-.2		-	-	-
		-.7		0.9125	2.4625	-
		-.8		-	-	-
BS04	normal	-.1	40	0.5375	1.7875	-
		-.2		-	-	-
BS06	normal	-.1	50	0.4800	1.9400	-
		-.2		-	-	-
BS10	normal	-.1	60	0.5917	2.3083	-
		-.2		-	-	-
BS20	normal	-.1	70	0.5500	2.4357	-
		-.2		-	-	-
BS30	normal	-.1	80	0.5312	2.4313	-
		-.2		-	-	-
BS40	normal	-.1	120	0.4292	1.7042	-
		-.2		-	-	-

Transmission components

If a transmission component is used (gearwheels, chainwheels, V-belt, etc.), the resulting radial forces can be determined as follows.

$$F_R = \frac{2000 \times M}{D_T} \times f_Z \leq F_{R(N,V)}$$

F_R	Radial force [N]
M	Torque [Nm]
D_T	Pitch radius of the transmission component [mm]
f_Z	Safety factor

A safety factor f_Z depending on the type of transmission component attached to the output shaft must be included when determining the value of the radial force F_R that is present.

Factor f_z for the type of transmission component

Transmission component	Safety factor f_z	Note
Gearwheel	1	=> 17 teeth
Gearwheel	1,15	< 17 teeth
Chainwheel	1	=> 17 teeth
Chainwheel	1,25	< 17 teeth
Toothed rack	1,15	< 17 teeth (pinion)
V-belt	2……2,5	From tensioning force
Flat belt	2….3	From tensioning force
Friction wheel	3….4	

Axial force

The following specification applies to the allowable axial force F_A on the output shaft (either tension or compression) for all Bauer geared motors and for foot, flange or hollow-shaft versions:

$$F_A = 0,5 \times F_{R(N,V)}$$

Please consult us in case of larger axial forces.

Gear Motor Selection

Sizing based on efficiency

Drive configuration based on efficiency

With the IEC 60034-30-1 standard and the ErP 2009/125/EC EU directive, utilisation of the potential energy savings in industrial environments has been given increased urgency and made legally mandatory.
In the industrial applications area, electric motors consume the vast majority of electrical energy (approximately 70 %). They are used in all areas and in many applications, such as fans, pumps, grinders, rolling mills, lifts, transport and conveying equipment, household appliances, and office machines.
Due to this broad range of applications, electrical drive systems are a primary target for energy saving policies. As electric motors consume a large amount of electrical energy, even small improvements in efficiency lead to significant savings.
In many cases, especially in transport and conveying equipment, it is necessary to reduce the speed of a three-phase squirrel-cage motor. This can be done by using external traction gearboxes or by using external or integrated reduction gearboxes. With regard to energy savings, the efficiency of the gear unit and transmission components must not be ignored.

The overall efficiency of a system is calculated as follows:

$$\eta_{Installation} = \eta_{Motor} \times \eta_{Reducer} \times \eta_{Machine}$$

Savings potential Motor: η_{motor}

In accordance with the Motor Regulation 16640/2009/EC, the legally binding EU ErP directive 2009/125/EC specifies IE3 (Premium Efficiency) as the minimum efficiency for new motors operating in continuous running duty (S1) ≥ 0,75 kW, effective 1. January 2017.

The right motor frame size and motor type should be selected based on environmental and economical aspects based on the new motor regulations for the IE3 series.

The new Regulation (EU) 2019/1781 extends the scope of the affected motors in the performance and product spectrum and sets new binding implementation dates of 1 July 2021 and 1 July 2023.

Calculation of the efficiency under partial load

The motor data sheets list motor efficiency figures according to Motor Regulation (EU) 2019/1781 for operation at several load levels (50 %, 75 % and 100 %).
The efficiency at any partial load point can be calculated approximately from the efficiency figures for 75 % and 100 % load, and the energy balance of the application can be evaluated accordingly.

$$R_{VL} = \frac{\left(\frac{100}{\eta_{100}} - 1\right) - 0{,}75 \times \left(\frac{100}{\eta_{75}} - 1\right)}{0{,}4375}$$

$$R_{VO} = \left(\frac{100}{\eta_{100}} - 1\right) - R_{VL}$$

$$\eta_P = \frac{100}{\left(1 + \frac{R_{VO}}{p}\right) + R_{VL} \times p}$$

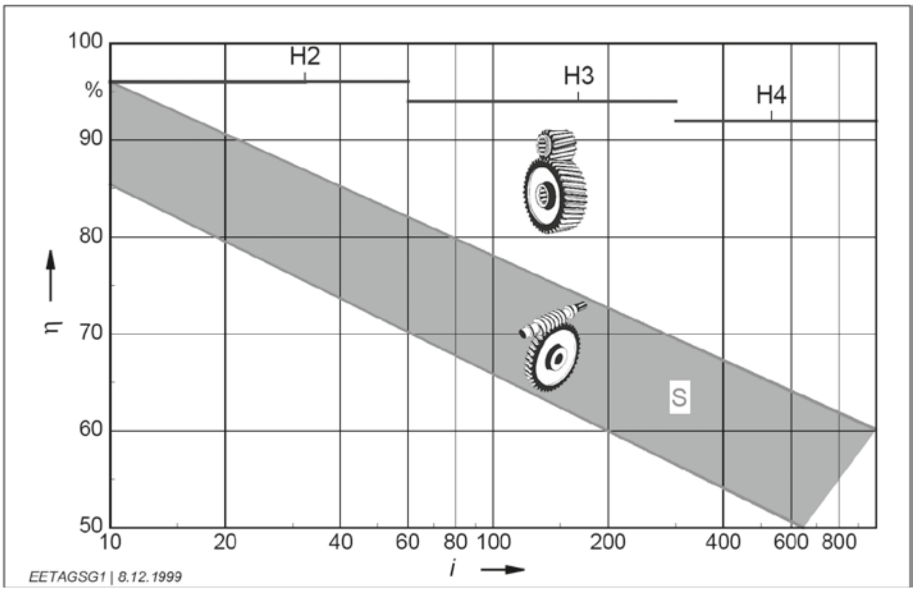
with

η_{100}	Efficiency at 100 % load
η_{75}	Efficiency at 75 % load
RVL, RVO	Intermediate results
p	Partial load (value range: 0 to 1 or overload)
η_p	Efficiency at partial load point p

Gear efficiency η_{gear}

Comparison of the general savings potential of gearboxes and motors in continuous running duty (S1) shows that the energy savings potential of gearboxes is significantly higher than that of motors. The efficiency of gearboxes is predominantly dependent on the tooth geometry and the friction values of the bearings and seals. At high input speeds and with vertical designs in which the first stage rotates fully immersed in oil, splash losses cannot be neglected. Vertical designs should generally be avoided.

The efficiency of worm gear drives is highly speed dependent (see illustration). Bauer worm gear units are available as two-stage worm gear units for frame sizes BS04 and larger. This enables very high reduction ratios and significantly higher efficiency than with pure worm gear units. A loss of 2 % per stage can be assumed for two-stage worm gear units.



Comparison of typical efficiency (η) versus reduction ratio (i) for helical spur gear units (H) with two, three or four stages and two-stage worm gear units (S), relative to the rated power of the gear unit.

System efficiency η_{system}

The drive system provides the highest savings potential in the analysis of the overall efficiency. Designers and plant engineers should always strive to optimise the transmission components.

Transmission component	Conditions	Efficiency
Wire rope	Per full turn on the wire drum (with journal or roller bearings)	0.91–0.95
V-belt	Per full turn on the belt pulley (with normal belt tension)	0.88–0.93
Synthetic belts	For each full turn or roll, with roller bearings (normal belt tension)	0.81–0.85
Rubber belts	For each full turn or roll, with roller bearings (normal belt tension)	0.81–0.85
Toothed belts	For each full turn or roll, with roller bearings (normal belt tension)	0.90–0.96
Chains	For each full turn or chainwheel, with roller bearings (depending on chain size)	0.90–0.96
Spindles	Trapezoid-thread spindle Ballscrew spindle	0.30 – 0.70 0.70 – 0.95
Gear unit	With spur gears or bevel gears: 2 % per stage, with worm gears and other types of toothing, according to manufacturer's data	0.94–0.98

Gear Motor Selection

Shock loads of machinery

Shock loads for various types of machinery are listed in standards and guidelines as well as industry-specific documents and manufacturer's documents. If for example a crusher or a press is listed here with an shock load class of III, this is justified. On the other hand, under favourable conditions a belt conveyor could have an shock load class of I, but this could quickly change to III with on/off operation, high speed and overdrive due to a loose chain.
Consequently, the classifications in the following table should by no means be taken blindly. They provide a rough point of reference, but the ultimate classification of the shock load should always take into account the factors specified by Bauer, in particular the inertia ratio, the cycle rate and the transmission component(s).

Drive	Shock load		
Construction machinery			
Construction lifts		II	
Concrete mixers		II	
Road construction machinery		II	
Chemical industry			
Cooling drums		II	
Mixers		II	
Stirrers (light media)	I		
Stirrers (viscous media)		II	
Drying drums		II	
Centrifuges (light)	I		
Centrifuges (heavy)		II	
Transport and conveying systems			
Hauling winches		II	
Conveying machines			III
Apron conveyors		II	
Belt conveyors (bulk material)	I		
Belt conveyors (piece goods)		II	
Bucket belt conveyors		II	
Chain conveyors		II	
Circular conveyors		II	
Freight lifts		II	
Flour bucket conveyors	I		
Passenger lifts		II	
Flat belts		II	
Screw conveyors		II	
Gravel bucket conveyors		II	
Inclined lifts			III
Steel belt conveyors		II	
Chain conveyors		II	
Blowers and fans			
Roots blowers		II	
Blowers (axial and radial)	I		
Cooling tower fans		II	
Suction blowers		II	

Drive	Shock load		
Rubber			
Extruders			III
Calenders		II	
Kneaders			III
Mixers		II	
Rolling mills			III
Timber processing and woodworking			
Debarking drums			III
Planers		II	
Woodworking machinery	I		
Saw frames			III
Crane systems			
Luffing mechanisms	I		
Traversing mechanisms			III
Hoisting mechanisms	I		
Slewing mechanisms		II	
Jib mechanisms		II	
Plastics			
Extruders		II	
Calenders		II	
Mixers		II	
Grinders and pulverisers		II	
Metalworking			
Plate bending machines		II	
Plate straightening machines			III
Hammers			III
Planers			III
Presses			III
Shears		II	
Forging presses			III
Punches			III
Countershafts and driveshafts	I		
Machine tools (principal)		II	
Machine tools (ancillary)	I		

Drive	Shock load		
Food processing			
Filling machines	I		
Kneading machines		II	
Mashing machines		II	
Packaging machines	I		
Sugar cane cutters		II	
Sugar cane mills			III
Sugar beet cutters		II	
Sugar beet washers		II	
Paper			
Couching			III
Smoothing rolls			III
Hollander		II	
Pulp grinder			III
Calender		II	
Wet presses			III
Shredders			III
Suction presses			III
Suction rolls			III
Drying rolls			III
Stone and soil			
Crushers			III
Rotary kilns			III
Hammer mills			III
Tube mills			III
Beating mills			III
Tile and block presses			III
Fabrics			
Winders		II	
Printing and dying machines		II	
Tanning vats		II	
Shredders		II	
Looms		II	

Drive	Shock load		
Rolling mills			
Plate shears			III
Plate turners		II	
Billet presses			III
Billet and slab lines			III
Billet conveyors			III
Wire drawing machines		II	
Descaling machines			III
Sheet metal mills			III
Plate mills			III
Winders (strip and wire)		II	
Cold rolling mills			III
Chain transports		II	
Billet shears			III
Cooling beds		II	
Cross transports		II	
Roller tables (light)		II	
Roller tables (heavy)			III
Roll straighteners		II	
Tube welders			III
Trimming shears		II	
Cropping shears			III
Continuous casting machines			III
Roll adjustment devices		II	
Manipulators			III
Laundry			
Drum dryers		II	
Washing machines		II	
Water treatment			
Centrifugal aerators		II	
Archimedes screw		II	