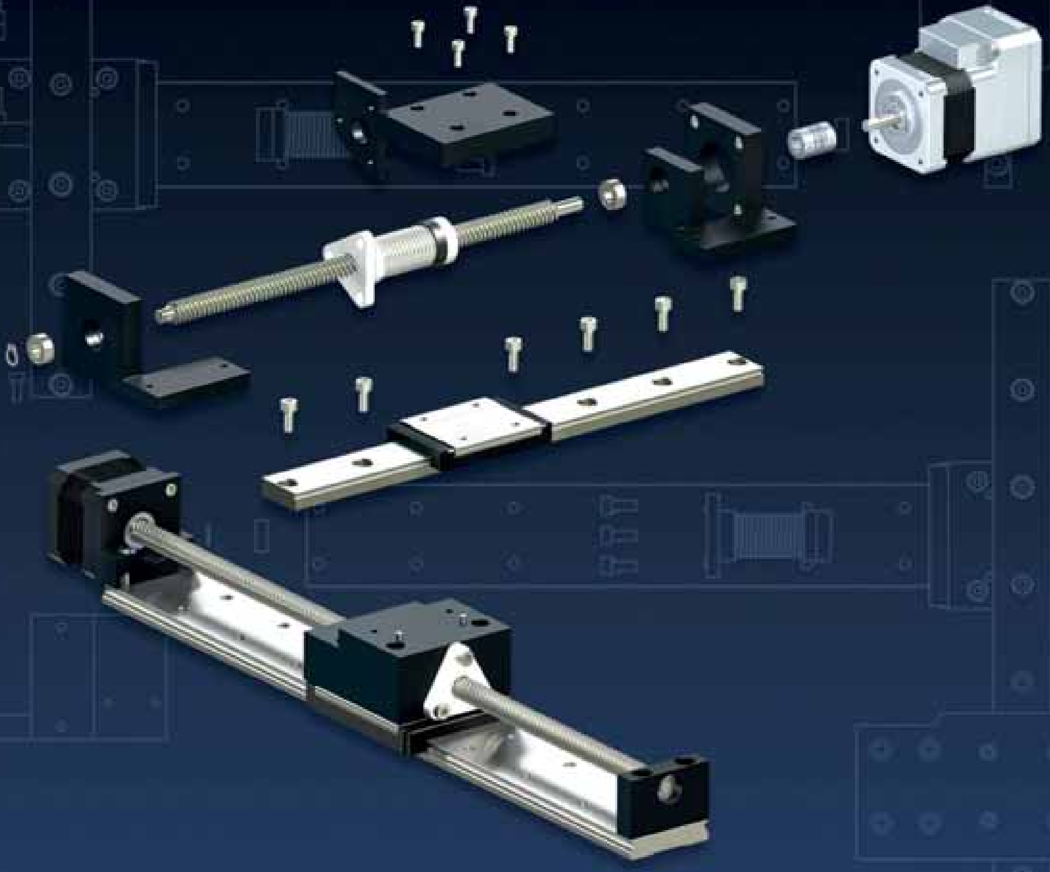
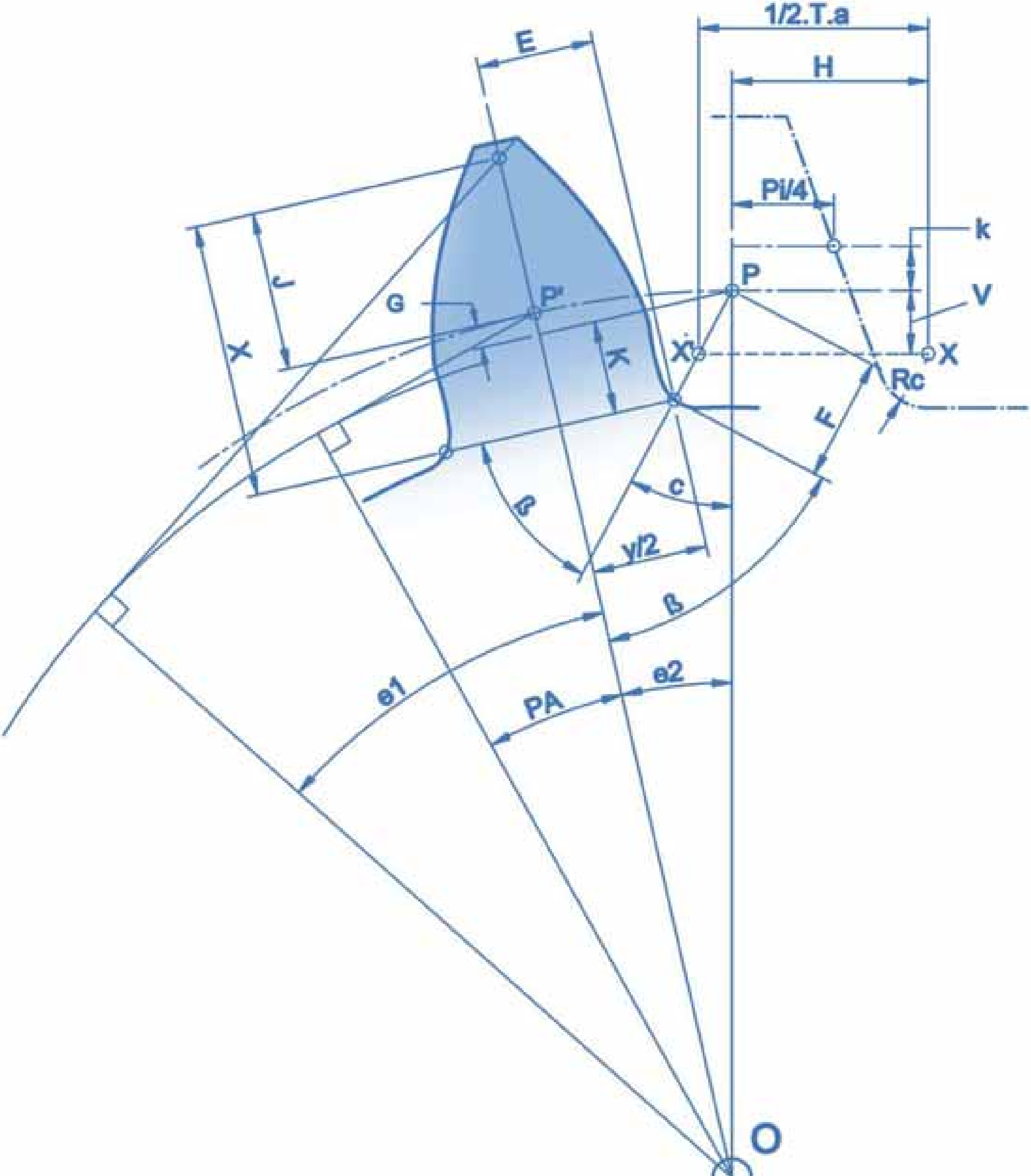
Reliance Precision Limited



Precise Motion Control Solutions

Technical Information

Contents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Section Contents

Standard Conversion Factors

Length	1mm = 0.03937 in	1in = 25.4 mm
Area	$1mm^2$ = 0.00155 in^2	$1in^2$ = 645.16 mm^2
Volume	$1mm^3$ = 61.02 x 10^{-6} in^3 1 litre = 1 x 10^6 mm^3	$1in^3$ = 16387 mm^3 1ml = 1 cm^3
Force and mass	1N = 0.101972 kgf 1kgf = 9.80665 N 1kg = 2.2046 lb	1lbf = 4.4482 N 1N = 0.2248 lbf 1lb = 0.4536 kg
Torque	1Nm = 8.8507 lbfin 1Nm = 141.612 ozfin 1Nm = 10197.16 gfcm	1lbfin = 0.1130 Nm 1ozin = 0.00706 Nm
Power	1kW = 1.360 PS (metric hp) 1kW = 1.341 hp	1hp = 0.7457 kW
Moment of inertia	$1kgm^2$ = 54674.75 $ozins^2$ $1gcm^2$ = 5.467 x 10^{-3} $ozin^2$ $1kgm^2$ = 23.73 $lbft^2$	$1ozins^2$ = 18.29 x 10^{-6} kgm^2 $1ozin^2$ = 182.9 gcm^2 $1lb.ft^2$ = 0.0421 kgm^2
Pressure and stress	$1N/m^2$ = 145 x 10^{-6} lbf/in^2 $1N/m^2$ = 64.75 x 10^{-9} $tonf/in^2$	$1lbf/in^2$ = 6.895 x 10^3 N/m^2 $1tonf/in^2$ = 15.44 x 10^6 N/m^2
Temperature	ºC = (ºF-32)*5/9 K = ºC+273.15	ºF = (ºC*9/5)+32 ºR = ºF+459.67
Angles	1rad = 180/π degrees 1mrad = 10.8/π arcmins	1 degree = π/180 rad 1 arcmin = π/10.8 mrads

S.I. Multiples

Prefix name	Symbol	Factor
tera	T	10^{12}
giga	G	10^9
mega	M	10^6
kilo	k	10^3
hecto	h	10^2
deca	da	10^1
deci	d	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	µ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}
femto	f	10^{-15}
atto	a	10^{-18}

Angular Resolution

Bits	Counts	arcmins	mrads
7	128	168.75	49.087
8	256	84.375	24.544
9	512	42.188	12.272
10	1024	21.094	6.1359
11	2048	10.547	3.0680
12	4096	5.2734	1.5340
13	8192	2.6367	0.76699
14	16384	1.3184	0.38350
15	32768	0.65918	0.19175
16	65536	0.32959	0.095874
17	131072	0.16479	0.047937
18	262144	0.082397	0.023968
19	524288	0.041199	0.011984
20	1048576	0.020599	0.0059921

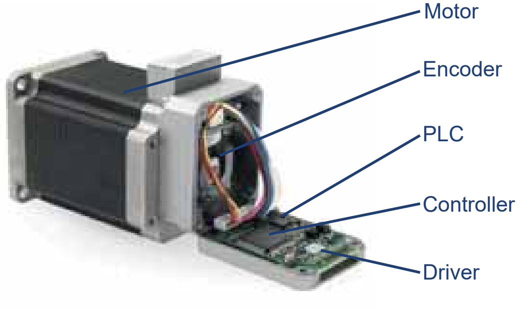
Contents

RELIANCE COOL MUSCLE FEATURES

The Reliance Cool Muscle (RCM) is packed with features that help you reduce the size and cost of your machine while reducing development time.

Simple and Compact

An intelligent driver with a 32 bit CPU based motion controller, driver amplifier, magnetic encoder and power management are all built on to the motor.

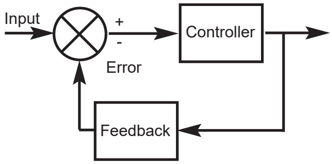


FULL CLOSED LOOP SYSTEM

RCM is a fully closed loop system. With a high resolution magnetic encoder and the intelligent driver board mounted on the back, RCM constantly monitors every aspect of its operation, eliminating any missed steps.

Closed Loop System

By receiving position input from the sensor, the RCM knows its position and can correct itself.

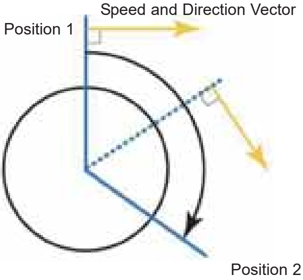


H-Infinity

Using the newest control technology, the RCM goes beyond static PID control by utilising the robust H∞ control system. H∞ responds to dynamic loads across the entire speed range, reduces the need to tune gains and increases the allowable inertia mismatch.

SMOOTH AND ACCURATE MOVEMENTS

The RCM's high resolution encoder gives you exceptionally fine positioning of 50,000 units per rotation. The RCM uses Vector Drive Control to produce extremely smooth motion, even at low speeds, not possible with micro-stepping drivers.

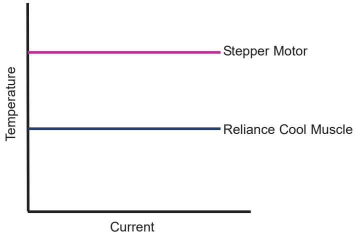


Vector Drive Control

Vector Drive is a control technique used in servo systems; it is a completely different technique from micro-stepping. Unlike micro-stepping, Vector Drive Control is not subject to resonance problems, produces smooth movements, increases torque and increases efficiency.

COOL OPERATION

RCM's power management monitors and provides the optimum current based on load, keeping the motor cool. In addition, the RCM generates high torque at low speeds.



RCM applies optimum current to produce motion, whereas an open loop stepper always uses maximum current.

RCM has high torque even at low speeds and excels at both smooth motion and slow speeds.

RCM only draws power for what it needs, making the RCM power efficient and increasing motor life.

CONTROL

P-type RCM is a drop-in step loss free replacement for a step/direction or CW/CCW pulse drive.
C-type RCM takes ASCII commands or MODBUS RTU from your PC, PLC or can use analogue (joystick) control.

	Control	Variations
P Type	Pulses	CW/CCW Step/Direction
C Type	PC Embedded Computer PLC Switch	Pre-programmed Dynamic Command
	Analogue Input	Position Speed

FULLY USER PROGRAMMABLE

Program the RCM to create the motion you need. Define motion profiles and create programs using easy-to-understand RCM Language (RCML). Motion programs you create can be downloaded to the RCM. The programs can be executed via PC, embedded computer or simply using hardware inputs.

RCML

Reliance Cool Muscle Language allows easy creation of motion programs. Programs can be downloaded to the RCM using free Control Room software available from:

www.reliance.co.uk/en/downloads-motors-and-cables

```
P1=1000
P2=2000
S1=200
S2=300
A1=50
A2=150
T1=20
```

Define motion profiles such as speed, acceleration position and timer.

```
B1
A1,S1,P1
S2,P2,P1
C2
B2
A2,S1,P3
```

Define motion programs using the motion parameters defined above.

USER DEFINABLE PARAMETERS

Define the character of your RCM to suit your needs. The RCM gives you over 60 parameters which can easily be set using RCML.

K48=10000	Origin offset distance set to 10000 pulses.
K58=200000	Software limit + side set to 200000 pulses.
K37=3	Motor resolution set to 1000.
K46=1	Automatic home routine using mechanical stopper.
K38=0	Sets analogue interface to speed control mode.

PARAMETER EXAMPLES:

Home Search Method

The home search parameter lets you select a home search method. Home position can be determined using a hard-stop/bumper instead of a home switch. The RCM hits a bumper at low speed and torque and keeps pushing until it reaches a specified current level at which the motor determines that it has reached home. This method eliminates the need for a home switch and wiring.

Software Limit

Set software limits using RCM parameters. Set limits on both CW and CCW sides, to eliminate switches.

These two software features will save you the cost of three sensors and the time needed to install wiring and calibrate them.

SOFTWARE INTERFACE

Serial Protocol

The PC interface to the Cool Muscle 'Y' Cable is RS232. Cool Muscle serial communications use the ASCII character set. Characters are transmitted with 8 data bits, no parity bit, and one stop bit. There is no hardware or software flow control. The baud rate is selectable from 9600, 19200, 38400 (default) and 57600 baud. The command separator is carriage return or comma. Line feeds are optional and are ignored.

Register Model

The data memory of the Cool Muscle is divided into families of registers. Each register is labelled by its family (letter) and register number. General parameters and settings are in the K-family. For example register K58 holds the software limit for maximum travel in the + direction in units of 1000 steps. To read the limit, simply send the register name K58 to the motor and it will respond with K58.1=247 if the limit is 247000 steps. To change the limit to 352000 steps, send command K58=352. The K-parameters are non-volatile. If multiple motors are daisy chained together it is necessary to add the motor number to the command, so K58.2 refers to register K58 in motor 2. Each motor can store 25 positions in P-registers P1 to P25, fifteen speeds in S1 to S15, and so on. There are registers for eight accelerations, seven maximum torques, eight dwell timers, and fifteen unassigned registers for general use. M6.4 is torque limit register number six in motor 4, for example. These registers are volatile, but can be saved into the Cool Muscle controller's EEPROM. The saved values are automatically reloaded into RAM when the motor is switched on.

Program Banks

Up to 15 programs can be stored in the controller EEPROM. Each is a sequence of commands.

```
B2.5               This is program 2 in motor 5
A3.5,S5.5,M1.5     Load motion parameters from registers
F2.5               Reset OUTPUT 2
X7.5               Start of loop, loop 7 times
 P1.5              Go to position in register P1
 T1.5              Dwell for time in register T1
 P2.5              Go to position in register P2
 T1.5              Dwell for time in register T1
X.5-               End of loop
C3.5               Call program B3.5 as a subroutine
O2.5               Assert OUTPUT 2
END.5              End of program
$.5                Save to EEPROM, motor 5
```

This program can be started by sending the short command [2.5.

Logic Banks (PLC Function)

Logic banks are similar to bank programs but are run periodically with a maximum frequency of once every 1ms.

```
L1.1               This is logic bank 1 in motor 1
I3.1,J2.1          Test INPUT 3, if set jump to bank 2
END.1

L2.1               Logic bank 2
S.1=S2.1           Load speed from register S2
A.1=A1.1           Load acceleration from register A1
^.1                Activate new speed and acceleration
J3.1               Jump to bank 3
END.1

L3.1               Logic bank 3
I3.1,T0.1,J4.1     Test INPUT 3, if set ignore (T0.1)
END.1              Otherwise jump to bank 4

L4.1               Logic bank 2
S.1=S3.1           Load speed from register S3
^.1                Activate new speed
J1.1               Jump to bank 1
END.1

$.1                Save to EEPROM, motor 1
```

Execution starts in logic bank L1.1. If INPUT 3 is not set, nothing happens until the next periodic run. Then L1.1 runs again.
If INPUT 3 is ever set, a jump occurs. Logic bank L2.1 makes a speed change and control jumps immediately to logic bank L3.1.

Control now remains with L3.1 until INPUT 3 is reset. Then L4.1 makes another speed change and control goes back to the beginning.
The effect is that motor speed is selected using INPUT 3. The speed change is smooth, using the acceleration in register A1.1 and S-curve shaping if parameter K69 is set.
Bank programs and logic banks can both run at the same time, so an ordinary bank program can initiate a motor move and then a logic bank can modify the speed en route.

More Information

A quick reference card listing all registers and commands is on our website, together with a more detailed programming manual: www.reliance.co.uk/en/downloads-motors-and-cables

ELECTRICAL INTERFACE

The RCM has 4 inputs and 2 outputs that can be used as digital, analogue, serial or pulse counter (input only). RCM lets you assign a different function to each edge and level of a signal.

Pin Layout

Pin #			
1	+24 V DC IN	Motor power	+24 V±10%
2	0V	Power ground	Note 7
3	INPUT 2-	Return for pin 9	Notes 2, 8
4	OUTPUT 2+	Digital/analogue output, serial Tx	Note 5
5	OUTPUT 1+	Digital/analogue output, serial Tx	Notes 1, 5
6	INPUT 4+	Digital/analogue input	Notes 3, 4
7	INPUT 3+	Digital input	Note 3
8	INPUT 1-	Return for pin 10	Notes 1, 8
9	INPUT 2+	Digital/counter input, serial Rx	Notes 2, 8
10	INPUT 1+	Digital/counter input, serial Rx	Notes 1, 8
11	0V	Signal ground	Note 7
12	+5V DC OUT	Power out	Note 8

Notes

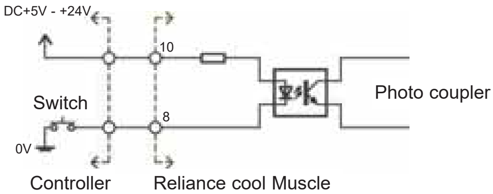
1. Normally used for serial communication with the host PC via an accessory ‘Y’ or USB cable. In a daisy chain system with multiple motors, used for serial communication with the next upstream, slave or master motor. If the Cool Muscle is being used stand-alone INPUT 1 and OUTPUT 1 can be assigned other functions. These functions are activated when the ‘Y’ Cable is detached (power off before disconnecting).
2. In a daisy chain system with multiple motors, used for serial communication with the next downstream, slave motor. Otherwise INPUT 2 and OUTPUT 2 can be assigned other functions.
3. When programmed as a digital input, INPUT 3 and INPUT 4 logic levels are:-
   HIGH > +3 V (minimum 7 mA)
   LOW < +0.8 V
4. Analogue input range is 0 V to +4.8 V. Resolution is 10-bit (0 – 1023).
5. When programmed as a digital output this signal is NPN, open collector. When programmed as an analogue output the signal range is 0V to 5V. Resolution is 8-bit (0 – 255).
6. Total output current maximum 50 mA.
7. Pins 2 and 11 are internally connected.
8. When used for STEP/DIRECTION pulse control, INPUT 1 is the step input and INPUT 2 is the direction input. When used for CW/CCW pulse control INPUT 1 steps the motor clockwise and

INPUT 2 steps anti-clockwise.
Maximum pulse frequency: 500 K pulse/s
Minimum pulse width; 0.8 μs
Pulse level high > +3 V (minimum 7 mA)
Pulse level low < +0.8 V

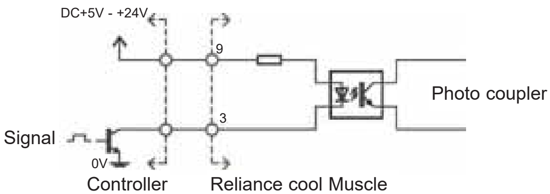
Wiring for INPUT 1 and INPUT 2

These inputs are opto-isolated inputs, minimum 5 V, maximum 24 V. Examples:-

Input 1



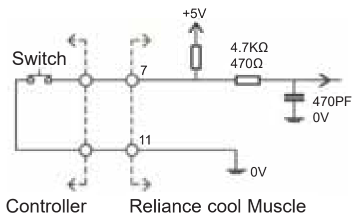
Input 2



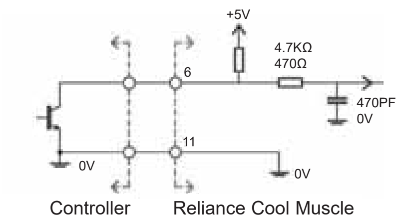
Wiring for INPUT 3 and INPUT 4

These inputs are internally pulled up to +5 V. To operate them connect to 0 V through a switch or open collector (NPN) output. Examples:-

Input 3



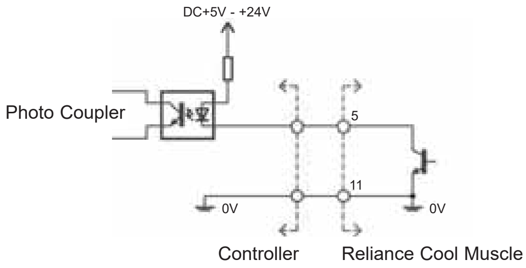
Input 4



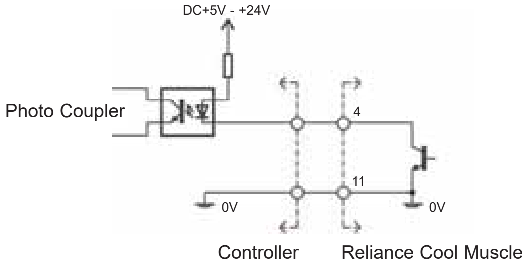
Wiring for OUTPUT 1 and OUTPUT 2

Outputs 1 & 2 can work across a range of voltages from 5 V to 24 V. The collector current of the transistor must be limited to a maximum of 100mA.

Output 1



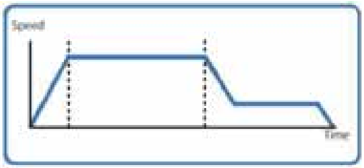
Output 2



ADVANCED MOTION

Speeds and accelerations can be changed whilst the motor is in operation. RCM supports a range of advanced motion features such as PTP motion incorporating changing accelerations and variable torque control. The powerful push mode is also standard allowing for electric emulation of common pneumatic operations.

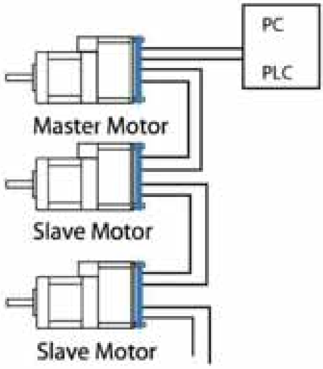
Continuous PTP: There are no stops in motion between origin and P3. Speed and acceleration are changed at each point.



Push Mode: Mimics a typical pneumatic motion. It keeps pushing for a given time and at a set current level when a motor encounters a resistance such as a bumper or stopper.

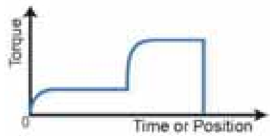
Contact us for product information, design support and custom solutions
sales@reliance.co.uk +44 (0) 1484 601002 www.reliance.co.uk

NETWORK



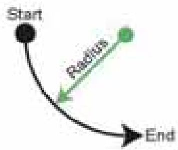
RCM provides you with different networking solutions to suit your needs. When multiple RCMs are connected in a daisy chain network, any RCM can tell other motors to activate programs as well as receive commands from a computer or embedded controller.
In fact, after programming, RCMs can operate without any PC, PLC or HMI control.

TORQUE CONTROL AND FEEDBACK



The RCM controller uses the integrated current and position sensors to maintain sophisticated torque control during operation. Peak running torque can be easily set within motion programs, or the built in Push Mode function can be quickly implemented to mimic pneumatic cylinder operations.

2-AXIS CO-ORDINATED MOTION



The RCM servo provides 2 axes contouring utilizing a 2+ motor daisy chain network. Additional linear axes can be implemented on the same motor for applications such as dispensing, cutting, or inspection. Programs can be run directly from the motor without the need for a host controller, or can be streamed from a PC for greater flexibility.

LOGIC PROGRAMMING AND PLC FUNCTIONALITY

The RCM real time operating system precisely controls I/O timing allowing for PLC style I/O operation. Logic banks provide a flexible logical and mathematical capability analogous to that offered by traditional ladder logic. User defined actions can be triggered by external inputs or by internal motor conditions such as speed, torque, or position.

RACK ACTUATOR

Installation

Each Racktuator™ is supplied pre-assembled with carefully set clearances and alignments. Disassembly may result in reduced performance and accuracy. The rack should not be removed from the housing to avoid possible damage to teeth when it is re-inserted.

Axial load rating

The axial load rating is dependent upon the rack, pinion and motor. As a general guide the load ratings in the product data pages (see pages 2-15 to 2-17) can be used to determine the allowable rack thrust.

Basic Ratings:

- For axial thrust loads of up to 3 N use a RCMRA17-6-250 tubular rack (see page 2-15) with a PEEK pinion
- For axial thrust loads from 3 N to 20 N use a stainless steel pinion in lieu of PEEK pinion
- For axial thrust loads from 20 N to 90 N use the RCMRA23L (see page 2-16), which utilises a 17-4 PH stainless steel pinion and hardened stainless steel rack

Position accuracy and side wobble

Positioning accuracy depends on the resolution of the motor and the drive system. For the RRA series the full step size of the motor is 1.8° which translates to 0.2 mm of linear motion of the rack. Finer positioning may be achievable with a half-stepping or micro-stepping drive. For the RCMRA series where the angular resolution is 0.0072° the linear resolution is 0.0008 mm, depending on load and dynamic conditions.

Side wobble is dependent upon initial clearance between rack and bearing bore, the length of rack and wear between the rack and plain bearings. For RRA17-6-250 (see page 2-15) the maximum side wobble is ±0.2 mm at end of the rack with maximum protrusion from the housing.

Backlash

This is currently set on assembly between 0.020 mm and 0.060 mm axial clearance of the rack. It is possible to improve this on assembly and also reduce the rotation of the rack in the clearance, please contact us. It is not advisable to reduce backlash to zero as pinion eccentricities and temperature variations could cause binding. A temperature rise of approx. 35°C would be needed to cause possible binding of rack and pinion when a backlash of 0.010 mm is set.

Lubrication

PEEK pinion and stainless steel racks require no lubrication. Stainless steel racks and pinions require a smear of a lithium based grease on to the rack teeth for periodic lubrication.

RACKTUATOR™ STEPPER MOTORS

Stepper motors operate by rotating the motor shaft at discrete intervals (1.8º for our steppers) as they receive electrical input pulses. This basic characteristic distinguishes stepper motors from other motors and makes them ideal for applications where accurate positioning and control is required, without the need for expensive feedback hardware.

Features of stepper motors

Position holding - (Detent torque)	Even with no power applied to the windings, stepper motors will resist rotation, which may be useful in applications that would normally experience ‘drift’. If power is applied, this holding torque is significant.
High acceleration -	They have excellent acceleration performance that allows a start, stop and reverse to be performed at relatively high speed.
Good reliability -	The only components subject to wear are the bearings, as there are no brushes or commutators.
Low component count -	Stepper motors permit open-loop, high precision positioning control, therefore feedback hardware for control is not required, which leads to low cost system design.

Drive methods

There are three main modes of driving a stepper motor - Full Step, Half Step and Micro Step. With Full Step, the angular movement is the basic step angle, ie 1.8º. By manipulating the energisation sequence applied to the motor, it is possible to reduce the basic step angle by half. By manipulating the shape of the pulses applied to the motor, as well as the energisation sequence, the basic step angle can be split into several hundred micro steps. This large increase in resolution can only be achieved by using more complex drive electronics.

There are a number of methods of driving stepper motors, that basically divide into two groups; unipolar and bipolar drives. In unipolar drives the current always travels through the windings in the same direction. This is often achieved by attaching one end of each winding to a fixed voltage supply rail. With bipolar drives the current travels in both directions, which can give benefits in performance, although it usually requires more switching components. In both cases additional components such as resistors are often used to adapt a drive to a specific motor or to modify the characteristics of the motor-load system.

Our Racktuator™ stepper motors have 6 wires, which allow the user to chose between unipolar and bipolar drives. Steppers with less wires do not.

The design and implementation of a suitable drive for a specific application can be quite an involved process and is outside the scope of this technical section. For further advice please contact Reliance Technical Sales.

Technical information

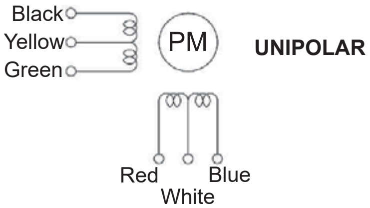
Insulation parameters	Dielectric Strength: 500 VAC Insulation Class: Class B
Insulation resistance	100 MΩ min. (at 500 V DC)
Dielectric strength	500 V AC (1 minute)
Operating temperature range	-20ºC to +50ºC
Permissible temperature rise	80ºC max. (resistance method)

Note: Do not allow the surface temperature of the motor case to rise above 90°C during operation.

Installation - connections

Size 17 steppers have 200mm cables with a EHR-6 connector (JST). Mating parts are available from RS components; top entry (stock no 515-1434) or side entry (stock no 515-1349).

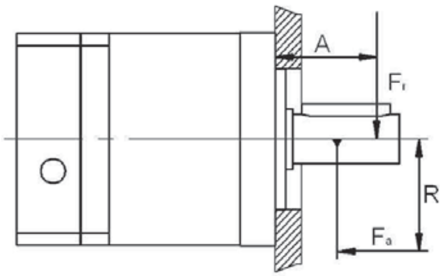
Installation - wiring diagram



CW rotation mounting end

Step	Black	Red	Green	Blue	Yellow	White
0	ON	ON			COM	COM
1		ON	ON		COM	COM
2			ON	ON	COM	COM
3	ON			ON	COM	COM
0	ON	ON			COM	COM

OUTPUT SHAFT BEARING LIFE



1. Calculate F_{rL} with the following formula

$$F_{rL} = \frac{F_a \times R + F_r \times (A + C_2)}{C_1}$$

2. Calculate the force-proportion

$$e = \frac{F_a}{F_{rL}}$$

Please contact us if e>0.22

3. Calculate F_L with the following formula

$$L_h = \frac{16667}{n} x \left(\frac{C_L}{F_{rL}}\right)^3$$

FORMULA SYMBOLS

L_h	h	lifetime
F_a	N	axial load at the output shaft
F_r	N	radial load at the output shaft
R	mm	distance, axial load to centre of the gearbox
A	mm	distance, radial load to flange plane
n	min^{-1}	output shaft speed
C_x	-	gearbox constants from following table

		RGP40	RGP60	RGPN70
C_1	mm	10.5	11.5	13.5
C_2	mm	12.9	15.5	23
C_L	N	2250	6050	9950

MAXIMUM LOAD IN CENTRE OF THE OUTPUT SHAFT

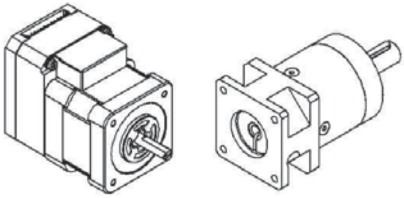
		RGP40	RGP60	RGPN70
F_r	N	200	500	1000
F_a	N	200	600	1200

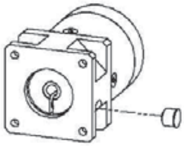
PLANETARY GEARBOX INSTALLATION MOUNTING

RGP40 NEMA 17

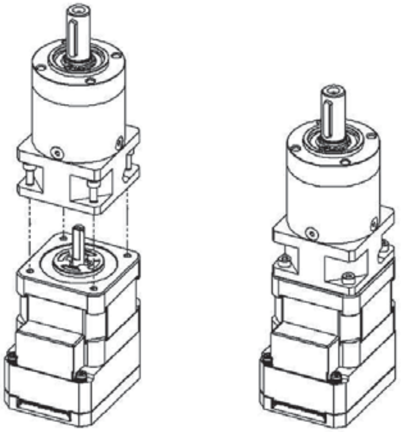
Make sure the gearbox has the correct mounting features for the selected motor.

Clean the Cool Muscle and the RGP gearbox so they are grease free, make sure not to get cleaning fluid into either the motor or gearbox.

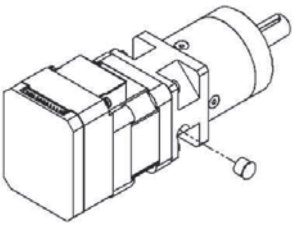




Remove the cover cap, adjust the position of the clamp to be in line with the mounting hole and open the clamp so that clamp diameter is greater than the motor shaft diameter.



The preferred method for mounting is in a vertical orientation as shown, mount the gearbox so it is flush with the motor, secure the gearbox and motor together with 4 off S-M3-8 screws and torque them up to T_{Mount} Nm.



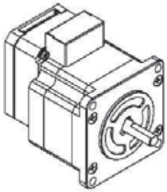
Tighten the clamp shaft onto the motor to T_{Clamp} Nm and re-attach cover cap.

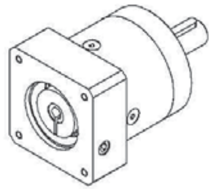
NEMA 17 Mounting Screw Torque	
Socket head cap screw order code	S-M3-8
T_{Clamp} (Nm)	1.1

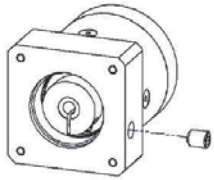
RGP40, RGP60 and RGPN70 NEMA 23

Make sure the gearbox has the correct mounting features for the selected motor.

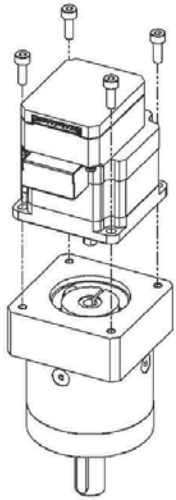
Clean the Cool Muscle and the RGP gearbox so they are grease free, make sure not to get cleaning fluid into either the motor or gearbox.

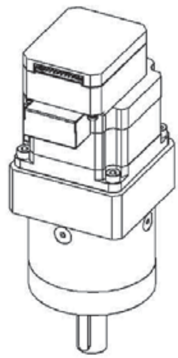




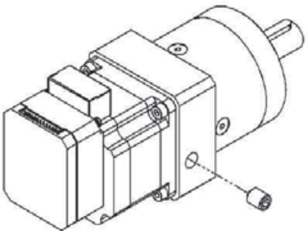


Remove the cover screw, adjust the position of the clamp to be in line with the mounting hole and open the clamp so that clamp diameter is greater than the motor shaft diameter





The preferred method for mounting is in a vertical orientation as shown, mount the gearbox so it is flush with the motor secure, bolt the two together with 4 off S-M4-12 screws and torque them up to T_{Mount} Nm.

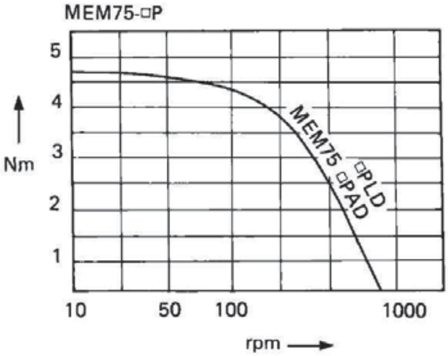


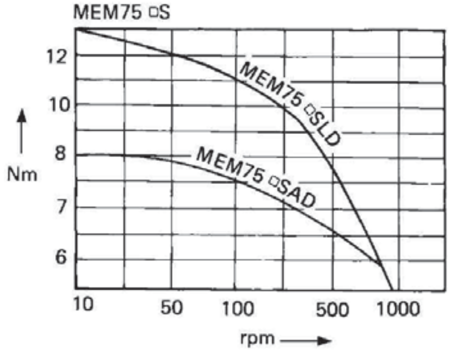
Tighten the clamp shaft onto the motor to T_{Clamp} Nm and re-attach cover screw.

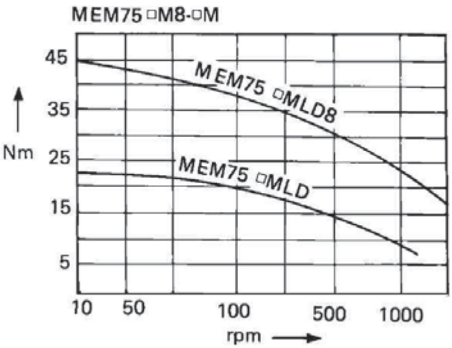
NEMA 23 Mounting Screw Torque	
Socket head cap screw order code	S-M4-12
T_{Clamp} (Nm)	1.1

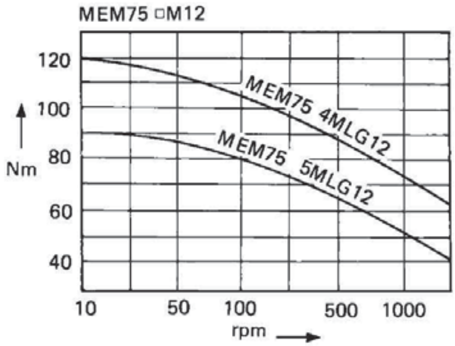
Clamping Screw Torque			
Socket width across flats (mm)	2	4.5	9.5
T_{Clamp} (Nm)	2.5	3	4

TORQUE CURVES









OUTPUT SHAFT TORQUE

This is derived from an eight hour day, continuous unidirectional drive and no impact fluctuating load.

PEAK TORQUE CAPABILITY

Momentarily allowable torque is 250% of rated torque (under the same conditions as output torque).

DYNAMIC LOAD FACTOR

The rated torque has been derated in accordance with the following table:

Dynamic Load Factor (Sf)

Drive	Driven	Load Type		
	Hrs/day	Uniform Sf	Medium Impact Sf	Heavy Impact Sf
Electric Motor	<3 3 - 10 24	1.0	1.0 1.25 1.5	1.5 1.75 2.0

$$\text{Allowable Torque} = \frac{\text{Rated Torque}}{\text{Sf}}$$

REVERSING MOTION

Should MEMs be used in a reversing drive (eg. servo application) the units should be derated to 80%.

TEMPERATURE RANGE

The unit will operate satisfactorily between -10º and +65ºC. For the all metal units, maximum temperature is 75ºC.

MOUNTING POSITION

The standard position is horizontal. For other planes, please contact our sales team for more information.

REQUIREMENTS FOR ASSEMBLING UNITS

1. Alignment - Radial alignment errors, after fitting the input and output shafts, should be within 0.15mm.

2. Location of Internal Gear - A unit should be located in the manner in which the torque distribution is uniform in the internal gear.

3. Clearance - Axial clearance between the unit's revolving parts (Carrier A & B) and casing should be 2mm to 4mm.

4. Thrust Support when Mounting Vertically - When mounting the units with shafts vertically, care must be taken to ensure that the mass of the module is supported by the shaft bearings and not the planet disks containing the internal gear. If the unit is mounted with the output shaft uppermost, then a shoulder will be required on the input shaft and vice versa.

5. Lubrication - For grease lubrication the casing should be filled with grease to between 50% and 80% of the volume and for oil lubrication to between 30% and 50% of the volume.

OVERHANG LOAD (OHL) - kg

The overhang load is a bending force acting on the shaft generated by external forces.

Calculate the OHL according to following equation and select an appropriate bearing:

$$\text{O.H.L. Capacity} = T \times Ef/R$$

T : Driving torque
R : Pitch circle radius of gear or sprocket
Ef : Element factor:

Gear	1.1 - 1.25
Sprocket	1.25
Flat Belt	2.5 - 3.0
V Belt	1.5 - 2.0

NOTES FOR HANDLING

1. Plastic Unit P - Lubricated with grease when assembled. (Units without grease lubrication are special to order).

2. Sintering Alloy Unit S - Not lubricated with grease when assembled. (Units with grease lubrication are special to order).

3. Metal Unit M - Not lubricated with grease or oil.

4. Do not mix strong acid or oil additives and thinners to the lubricant of the plastic units.

5. Do not allow rapid temperature variations. This will generate moisture.

6. Store the MEM units in a dark room below 40ºC and keep in a dry, dust-free atmosphere.

7. If a unit is mounted on a surface which acts as a sounding-board, the noise will be amplified above the inherent noise level of the unit. Take care when mounting the unit.

HOUSED UNITS (LGH)

MEM modules can be supplied mounted in an aluminium housing complete with output shaft and support bearings. The complete unit is rated at 10Nm output torque, and can have either one, two or three modules. Maximum reduction ratio is 125:1 with 3x5:1 ratio modules. The accompanying motor must have a ‘D’ shaped shaft of 8mm diameter and 7mm over the flat.

Also included is the MEM26. This is a housed unit complete with input and output shaft. Actual ratio is 91.125:1 and the unit is capable of handling 2 Nm output torque.

Larger modules are available up to 1000 Nm output torque. Please enquire.

GEAR MANUFACTURE

Reliance's precision instrumentation gears are manufactured using high accuracy gearcutting equipment. Standard gears are produced in stainless steel, hardened stainless steel, aluminium alloy and brass (wormwheels only). Alternative materials such as PEEK polymer or Delrin are available on request.

GEAR TOLERANCES

Gears are generally offered as Quality 10 (see the individual product pages). Higher qualities are available as shown in the table below. Most gears in the catalogue can be produced in these qualities to order.
Reliance standard tolerances are largely based on AGMA 390-03 backlash.

Reliance Standard Gear Qualities — Table values in 0.001mm (0.0001")

Quality Class	Modular Range	Total Composite Error	Tooth to Tooth Composite Error	Indicator Limits (Gauge zeroed at std. pitch rad.) Max	Indicator Limits (Gauge zeroed at std. pitch rad.) Min	Gear Quality Code
AQ9	1.5mod	26 (10)	18 (7)	-18 (-7)	-69 (-27)	-
AQ10	0.8 to 0.5 mod	26 (10)	13 (5)	-18 (-7)	-61 (-24)	-
AQ11		18 (7)	10 (4)		-53 (-21)	C
AQ12		13 (5)	8 (3)		-48 (-19)	B
AQ14		7 (2.7)	3.6 (1.4)		-41 (-16)	A
AQ10	0.4 to 0.2 mod	26 (10)	13 (5)	-13 (-5)	-51 (-20)	-
AQ11		18 (7)	10 (4)		-43 (-17)	C
AQ12		13 (5)	8 (3)		-38 (-15)	B
AQ14		7 (2.7)	3.6 (1.4)		-33 (-13)	A

Values in the above table refer to measurements obtained by means of the dual flank tester.

To specify a gear, other than the standard quality, add the quality code to the gear part number.
Example of a quality 12 gear - **P05S1B10F6A-100 B**
Quality code

Comparison of National Gear Quality Standards

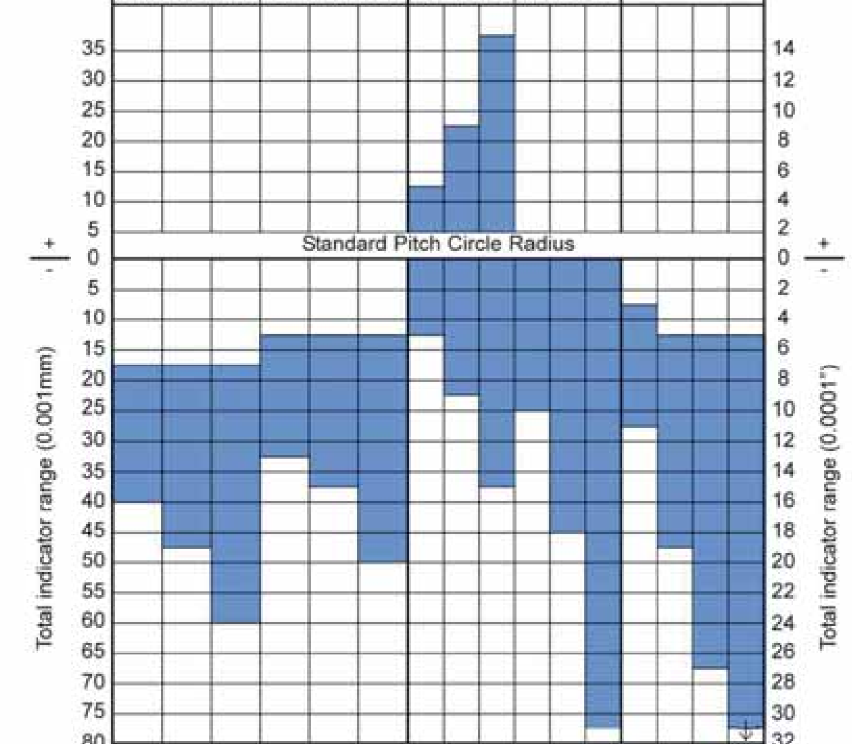
Reliance Quality Class	American AGMA 390.03 (1980)	British BS. 4582 (pt.1 : 1970)	German DIN. 867 & 3963	International ISO	Japan JIS	Admiralty BR.6001
AQ9	Q 9	Class C	Q 8	8	4	Class 3
AQ10	Q 10	Class B	Q 7	7	3	Class 2
AQ11	Q 11	Class A	Q 6	6	2	Class 1
AQ12	Q 12	Class A	Q 5	5	1	†
AQ14	Q 14	†	Q 3	3	0	†

† Reliance quality higher than any equivalent in this specification.
Table applies to gears up to 50 mm diameter.

RELIANCE GEAR STANDARDS FOR FINE PITCH GEARS

The table below is a comparison between Reliance (AGMA) and equivalent UK specifications:

AGMA 390.03 1980 Backlash Designation "C"						Admiralty Standard BR 6001(1)						BS 4582 Pt1 (BS 978 Pt1)			
Quality Number						Class						Class			
14	12	10	14	12	10	1	2	3	1	2	3	A	B	C	D
0.8 to 0.5 mod			0.4 to 0.2 mod			Method 'A'			Method 'B'						



	*Total Composite Error															†	
(0.001mm)	7	13	26	7	13	26	18	31	48	18	31	48	15	25	40	100	
	2.7	5	10	2.7	5	10	7	12	19	7	12	19	6	10	16	40	(0.0001")
	*Tooth to Tooth Composite Error																
(0.001mm)	3.6	8	10	3.6	8	10	8	13	23	8	13	23	5	13	23	58	
	1.4	3	4	1.4	3	4	3	5	9	3	5	9	2	5	9	23	(0.0001")

*AGMA values quoted are for over 20T up to 50 mm (2") diameter. Admiralty and B.S. tooth to tooth errors are for over 30T.
† Minimum indicator level 0.006" or 0.15 mm.

For numbers of teeth outside the range consult the relevant specification.

STANDARD MODULES AND CIRCULAR PITCHES - METRIC

Reliance's precision instrumentation spur gears are available as standard in the following modules and circular pitches, being those most commonly used in the design and manufacture of gear control mechanisms and instruments:

Module 0.2, 0.25, 0.3, 0.4, 0.5, 0.6, 0.8, 1.0, 1.25, 1.5
Circular pitch 1, 2, 2.5, 3

Pressure Angle and Rack Form

Except where stated otherwise, gears in this catalogue are cut to 20º pressure angle involute form teeth. Reliance standard gears will mesh satisfactorily with gears of the same module cut to the following standards:

(i) BS 4582 (1970) Part 1, Figure 1.

(ii) DIN 867 and 58412.

(iii) AGMA 207.06 (Assuming the pitch is cut to an equivalent module).

The gears will not mesh satisfactorily with gears cut to DIN 58400 unless the outside diameter of the latter is reduced to PCD + (2 x module).

DIN 58400 tooth proportions are:

Addendum	1.1 x Module
Dedendum	1.5 x Module for pitch 0.1 to 0.6 Module

Metric Tooth Proportions (Dimensions in mm)

Module	Circular Pitch	Addendum	Dedendum	Working Depth	Whole Depth	Equivalent Inch Diametral Pitch
1.5	4.712	1.5	1.875	3.0	3.375	16.933
1.25	3.927	1.25	1.563	2.5	2.8125	20.320
1.0	3.142	1.0	1.400	2.0	2.4	25.400
0.8	2.513	0.8	1.120	1.6	1.92	31.750
0.6	1.885	0.6	0.840	1.2	1.44	42.333
0.5	1.571	0.5	0.700	1.0	1.2	50.800
0.4	1.257	0.4	0.560	0.8	0.96	63.500
0.3	0.942	0.3	0.420	0.6	0.72	84.667
0.25	0.785	0.25	0.350	0.5	0.6	101.600
0.2	0.628	0.2	0.280	0.4	0.48	127.000
0.318	1.0	0.318	0.446	0.637	0.764	79.796
0.637	2.0	0.637	0.891	1.273	1.528	39.898
0.796	2.5	0.796	1.114	1.592	1.910	31.919
0.955	3.0	0.955	1.337	1.910	2.292	26.599

The above list is by no means exhaustive. Please enquire if you require a special module as Reliance holds a large stock of non-standard cutters.

MATERIALS AND SPECIFICATIONS

Reliance's precision instrumentation gears are manufactured from the materials listed below. We reserve the right to change the actual material to an equivalent specification without notice depending on availability.

Reliance Precision Gear Materials				
Material	**Specification**		**Used on**	**Material Code**
Stainless steel	303S31 (303S21) or 303S42 (303S42) or 302S31 (302S25) or 303 to MIL QQ-S-764	BS 970	Pin hub gears Clamp hub gears Hubless gears Worms Gear clamp & hubs	S1
Stainless steel	316S31 (316S16)	BS 970	Rack pinions	S2
Stainless steel (hardened)	17-4PH1025 Hardened to 34-42 HRc		Hardened pin hub gears	S8
Aluminium alloy	L168 or HE 15-TF or 2024-T4 to MIL QQ-A-225/6	BS 1474	Pin hub gears Clamp hub gears Hubless gears Gear clamp & hubs	A1
Phosphor bronze	PB 102	BS 2874	Worm wheels	B1
Brass	CZ121	BS 2874	Spur gears	B2
Brass (Naval)	Alloy 464 to MIL QQ-B-637		Worm wheels	B3
Acetal	Delrin 150		Hubless gears	D1

Finishes

Stainless steel, bronze and brass gears remain in their natural condition. Passivation to DEF STAN 03-2, process M can be carried out if required. Aluminium components are anodised to specification DEF STAN 03-24 (chromic acid process) or DEF STAN 03-25 (sulphuric acid process). Gear teeth are not normally anodised.

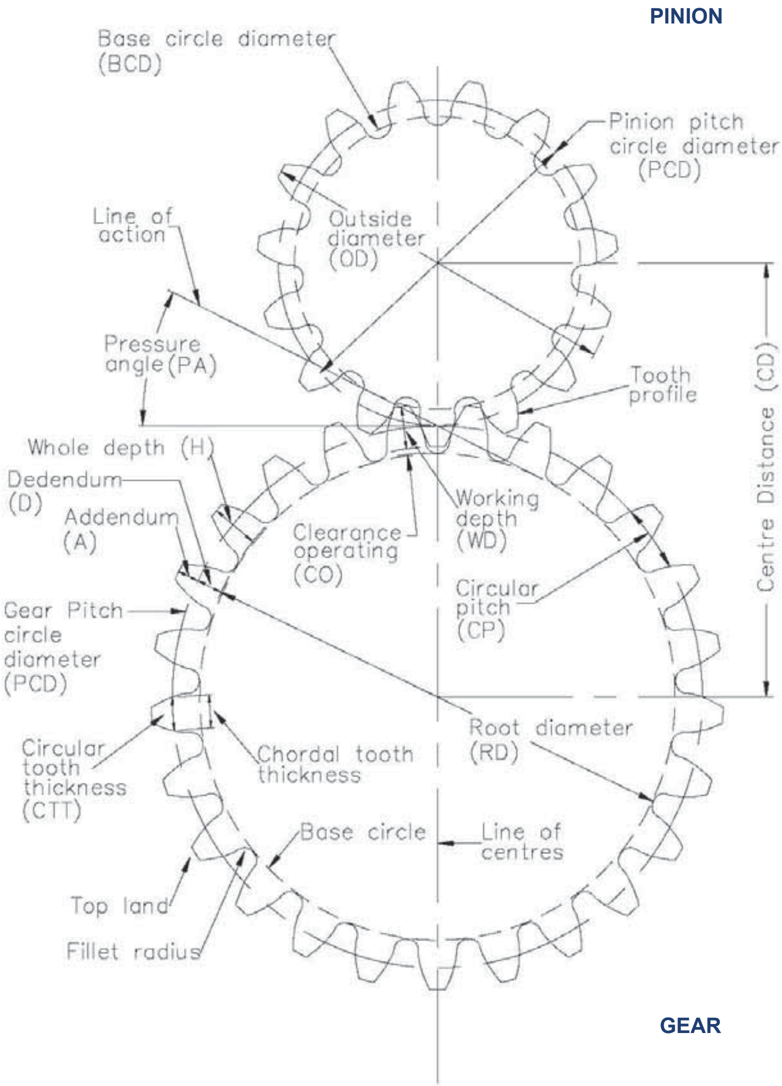
Anti-backlash Gears

Materials and finishes of standard anti-backlash gear components.

Where possible circlips, anti-backlash springs, shims and set screws will be stainless steel. However, some smaller pinions may have beryllium copper or zinc plated carbon steel circlips as standard.

SPUR GEAR GEOMETRY

A basic description of gear tooth terms is shown below. General formulae to enable correct understanding of spur gear geometry is shown opposite.



TERMINOLOGY FOR METRIC SPUR GEARS

TERM	DEFINITION	FORMULAE
Addendum (A)	The radial distance between the pitch circle and the outside diameter.	A=M
Addendum modification (K)	A method of modifying low tooth number gears to avoid undercutting and altering gear size to allow use of non standard centres.	See page T4-8
Backlash (B)	The circumferential clearance between mating gear teeth.	See page T4-15
Base circle diameter (BCD)	The diameter of the base cylinder from which the involute is generated.	BCD = N·Mcos PA
Base pitch (BP)	The pitch along the base circle or line of action.	BP = π Mcos PA
Basic rack	The straight sided rack shape from which teeth are generated.	See BS 4582.
Centre distance (CD)	Distance between the axes of rotation of mating spur gears.	$CD = \frac{PCD_{pinion}}{2} + \frac{PCD_{gear}}{2}$
Circular pitch (CP)	The distance along the pitch circle between corresponding points on adjacent teeth.	CP = π M
Circular tooth thickness (CTT)	The distance between opposite faces on the same tooth measured at the pitch circle diameter.	$CTT = \frac{\pi M}{2}$
Clearance operating (CO)	The amount by which the dedendum in a given gear exceeds the addendum of the mating gear.	CO = D - A
Dedendum (D)	The radial distance between the pitch circle and the root diameter.	D = 1.4M (BS4582) = 1.25M (BS436)
Diametral pitch (DP)	The size of the tooth expressed in teeth per inch of pitch diameter.	
Face width	The width of the tooth in an axial plane.	
Fillet radius	The radius of the fillet curve at the base of the gear tooth.	
Length of action	The distance on an involute line of action through which the point of contact moves during the action of the tooth profiles.	

Standard Gears

Indicator limits	The size band of manufacture for the gear measured radially from the PCD.	
Module (M)	The size of the tooth expressed in mm of pitch diameter.	
Number of teeth (N)	Number of teeth on the gear.	
Outside diameter (OD)	The diameter over the tops of the teeth.	OD = PCD + (2M)
Pitch circle diameter (PCD)	An imaginary circle whose diameter is formed by meshing gears so that the circles actually touch each other, as if gears were driven purely by the friction of the circles.	PCD = (N+2K)·M Note: for unmodified gears K=0
Pressure angle (PA)	The angle between a line tangential to the pitch circles and a line perpendicular to the tooth profiles at the point of contact. (Equal to the side angle of the basic rack for standard gears).	Standard = 20º
Root diameter (RD)	The diameter of the base of the teeth.	RD = OD - (2H)
Total composite error (TCE)	The total error in the gear measured by the dual flank gear test. TTCE and pitch line runout are included.	
Tooth to tooth composite error (TTCE)	The change in error of each tooth on the gear measured by the dual flank tester.	
Undercut	The loss of profile in the vicinity of the involute start at the base circle due to tool cutter action generating gears with low tooth numbers. (N_{min} = minimum teeth for no undercut)	$N_{min} = \dfrac{2}{\text{Sin}^2\,\text{PA}}$
Whole depth (H)	The total depth of a tooth space.	H = A + D
Working depth (WD)	The depth of engagement between mating gear teeth.	WD = 2A

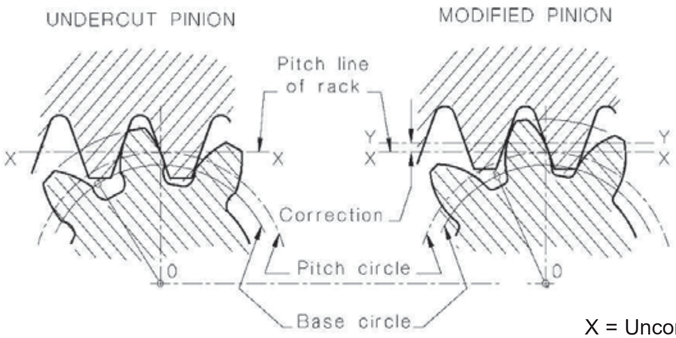
Note: for imperial gears to BS 978 Part 1, Equivalent Module = $\dfrac{25.4}{DP}$

GEARS WITH SMALL NUMBER OF TEETH

Unless otherwise requested, all gears in this catalogue having 16 teeth or fewer will be enlarged by applying addendum modification in accordance with BS4582 Part 1 (metric) as shown in the table below. These gears are indicated (†) against the appropriate tooth numbers on the product pages.

A small amount of backlash will be introduced between corrected pinions and mating gears when the modification sum is other than zero and the nominal centre distance is adjusted only by an amount equal to the modification sum.

For minimum backlash it will be necessary to either reduce the centre distance further, or to apply a secondary correction to the pinion or wheel. See the above B.S specification for details.



X = Uncorrected PCD
Y = Corrected PCD

Data for Addendum Modified Gears of Unit Module and Unit DP

No. of Teeth	Addendum Modification	Enlarged PCD	Enlarged OD (PCD+2)
10	0.4151	10.8302	12.8302
11	0.3566	11.7132	13.7132
12	0.2982	12.5964	14.5964
13	0.2397	13.4794	15.4794
14	0.1812	14.3624	16.3624
15	0.1227	15.2454	17.2454
16	0.0642	16.1284	18.1284

Example (Module)

Find P.C.D. and O.D. of enlarged gear having 13 teeth, 0.6 module

P.C.D. =13.4794 (from table) x 0.6 module =8.088 mm
(Standard P.C.D. would be 13 x 0.6 =7.800 mm)

Similarly,

O.D. =15.4794 (from table) x 0.6 module = 9.288 mm
(Standard O.D. would be 7.8 + (2 x 0.6) = 9.00 mm)

Note:

For Imperial (diametral pitch) gears, divide the PCD or OD value in the table by the diametral pitch. The answer will be in inches.

ENGINEERING DATA

For instrumentation Reliance normally recommend stainless steel pinions mating with aluminium alloy gears. Generally the pinion is subjected to most wear since it experiences a higher number of stress cycles than the wheel. This combination of materials tends to balance the wear between the pinion and the gear.

1. Gear Materials

Stainless steel
The 300 series stainless steels are used for gears when maximum corrosion resistance is required. They are 'true' stainless steels containing 18% chromium and 8% nickel.

Gears made from 303 stainless steel are essentially nonmagnetic and cannot be hardened by heat treatment. They are recommended for low torque applications as their mechanical properties and resistance are low.

Hardened stainless steel
17-4PH is a precipitation hardening stainless steel that offers a remarkable combination of high strength and hardness. Its high chromium content (15-17.5%) makes it an excellent material for arduous environments.

Aluminium alloy
Gears made from aluminium alloy are widely used in measuring applications. Its light weight offers reduced inertia. The inertia of an aluminium alloy gear is approximately 35% that of a steel gear. In particular, aluminium alloy L168 offers excellent corrosion resistance when anodised, moderately good mechanical properties and good stability.

Phosphor bronze
As a gear material phosphor bronze has a fine grain and good resistance to tooth sliding wear hence its use as a worm wheel material.

2. Installation

Gears in this catalogue are designed to be a slide fit on the shafts. The gears are available with four fixing methods: standard clamp, pins, set screws and Reli-a-Grip™ clamp.

Traditional clamp hub style gears have a gear hub with a relatively thin wall partially split. The clamp is a close fit on the hub and is compressed when the clamp screw is tightened. Clamping gears onto the shaft offers extremely easy assembly with the best assembled accuracy. However, as the fastening depends upon friction it can only be used in low torque applications.

Pin type gears are supplied as standard with a set screw and a sub-drilled hole. The set screw should be used to position the gear on the shaft during the drilling and pinning operation and can be removed once the gear is secure.

The sub-drilled hole provides a lead in for the drilling operation. It is recommended that drilling and pinning is completed outside the gearbox and the gear is thoroughly cleaned afterwards.

In less critical applications the set screw may be used to retain the gear on the shaft. To avoid damaging the shaft and to make removal of the gear easier the set screw should seat on a small flat, or dimple on the shaft.

3. Lubrication

All gears should be lubricated, but there are variations in degree.

Highly loaded precision gears should be in enclosed assemblies with complete lubrication to obtain the best possible hydrodynamic film. The system can be splash, spray or force fed, depending on the application. Moderately loaded precision gears, such as fractional horsepower systems, should be enclosed with oil or grease lubrication which can be spread by splash or dip lubrication.

Lightly loaded gears in instrumentation systems only need to have a marginal boundary lubrication as provided by periodically wiped on oils or grease. In many instances a light coat of Rocol MT LM or similar molybdenum disulphide grease will suffice for the life of the system. Anti-backlash gears should not be directly lubricated except via a very light application on the mating pinion.

Negligibly loaded fine instrument gears only need a brushed on film of light oil as a simple means of reducing friction.

4. Speed

The maximum pitch line velocity for stainless steel meshing with aluminium alloy with boundary lubrication is approximately 5,300 mm/sec (for a pair of meshing actuation gears correctly lubricated, this rises to approximately 8,000 mm/sec). This represents 5,000 rpm on measurement gears of 20 mm diameter (and 7,500 rpm on actuation gears of 20 mm diameter).

For speeds in excess of this and other material combinations please consult Reliance technical sales.

5. Gear Loading

The gears in this catalogue can be used for both feedback and actuation systems. The loads and material selection will depend on the application. In general a feedback system is designed to maintain accuracy and an actuation system is designed to transmit power.

5.1. Actuation Gears

The following analysis is intended to give a guide to the load capacity of a pair of spur gears. To simplify the calculations, a number of assumptions have been made. It must be noted that in many applications this will give a conservative estimate of the gear capacity, therefore, in critical applications an exact analysis must be completed.
Please consult the relevant gear standards or Reliance Technical Sales.

The analysis is based on AGMA 2001-B88 and assumes the following:

1. The gears are simply supported in rolling element bearings.
2. Pinion revolutions $>10^7$.
3. Gears are grease lubricated.
4. Reliability of 1 failure in 100 is acceptable.
5. Gear material is 17-4PH hardened.

The basic load capacity (F_b) of a pair of spur gears is defined as the maximum tangential force at which they can operate indefinitely.

F_b has two values: one calculated from tooth root strength (F_{bs}), and one for tooth flank pitting (F_{bw}). The useful or transmitted load capacity, F_t, is usually less than F_b due to transient or dynamic loads generated within the mechanism.

For tooth root strength	$F_{ts} = F_{bs}/K_a$	K_a & C_a = Application factors
For tooth flank pitting (wear)	$F_{tw} = F_{bw}/C_a$	

Both calculations should be made and the lower value used.

The application factors K_a and C_a make allowance for any externally applied loads in excess of the nominal tangential force F_b and they are most accurately determined by direct measurement. In determining application factors, consideration should be given to the fact that many prime movers develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered including system vibrations, acceleration torques, overspeeds, variations in system operation and changes in process load. Impact loads due to reversing across backlash can be significant in servo systems.

As a general guide application factors for a motor gear system range from 1.0 for uniform loads up to 1.75 where heavy shock loads are anticipated.

For strength $F_{bs} = 177.7 \times J \times F \times M \times K_v$ [N]

For wear $F_{bw} = 14.64 \times N \times I \times F \times M \times K_v$ [N]

F_b = Basic load capacity (F_{bs} & F_{bw})
N = Number of teeth
J = Geometry factor, strength
I = Geometry factor, wear
F = Face width of smallest gear
M = Module
K_v = Dynamic factor

(i) Number of teeth
This is the number of teeth in the gear being analysed.

(ii) Geometry factors, I and J
These factors take account of the effect of tooth proportions on stress. The bending strength geometry factor, (J) takes account of the shape of the tooth. The wear resistance geometry factor, (I) takes account of the radii of curvature of the contacting tooth profiles. Please see the graphs on page T4-13.

(iii) Face width, F
This is the face width of the smallest gear in mm. (Face width in contact).

(iv) Module, M
This is the gear module expressed as shown on the respective gear pages.

(v) Dynamic factor, Kv
This accounts for internally generated gear tooth loads which are induced by the non-conjugate meshing action of the gear teeth.

$$K_v = \left(\frac{84}{84 + \sqrt{200V_t}} \right)^{0.4}$$

For quality 10 gears only
V_t = Pitch line velocity (m/s)

Example calculation to find the theoretical load capacity of a 5:1 pass of 17-4PH spur gears as follows:

Pinion - P06S8B6F4A-25
Gear - P06S8B8F6A-125

Pinion speed is 500 rpm.

(i) Number of teeth from part number = 25

(ii) Geometry factors from graph

J = 0.37
I = 0.118

(iii) Smallest gear face width from part number

F = 4

(iv) Gear module from part number

M = 0.6

(v) Dynamic factor from equation

$$K_v = \left(\frac{84}{84 + \sqrt{200V_t}} \right)^{0.4} = \left(\frac{84}{84 + \sqrt{200 \times 0.393}} \right)^{0.4} = 0.96$$

$$\text{where} : V_t = \frac{\text{rpm} \times \pi \times N \times M}{60000} \left[\text{m/s} \right]$$

F_{bs} = 177.7 x 0.37 x 4 x 0.6 x 0.96 = 151.5 N

F_{bw} = 14.64 x 25 x 0.118 x 4 x 0.6 x 0.96 = 99.5 N

For alternative materials the above values need to be modified as shown below:

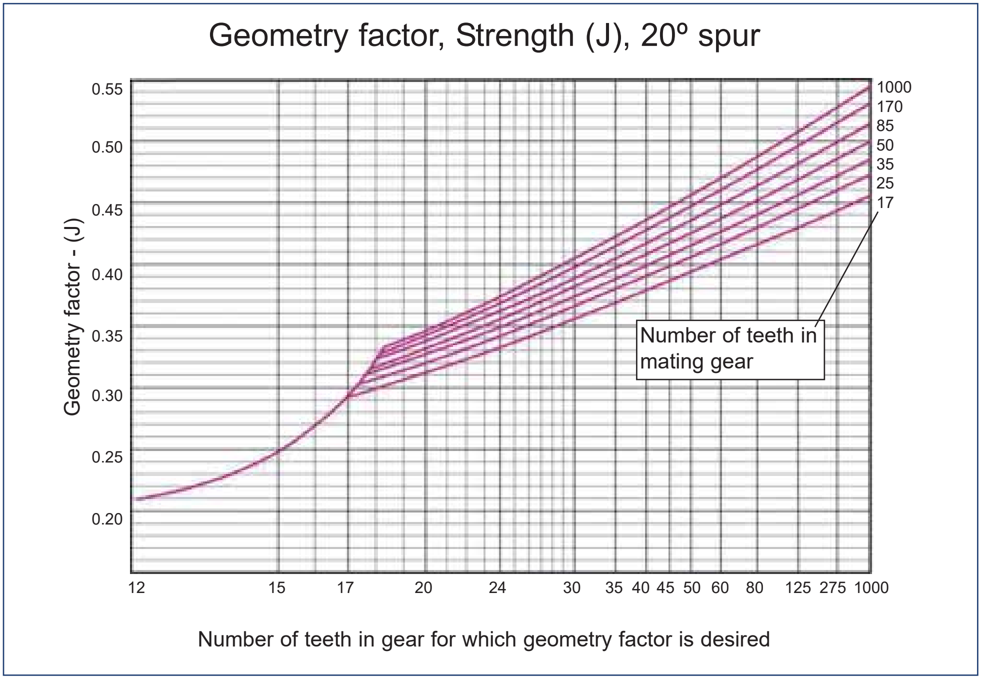
Gear Material Modification Factors

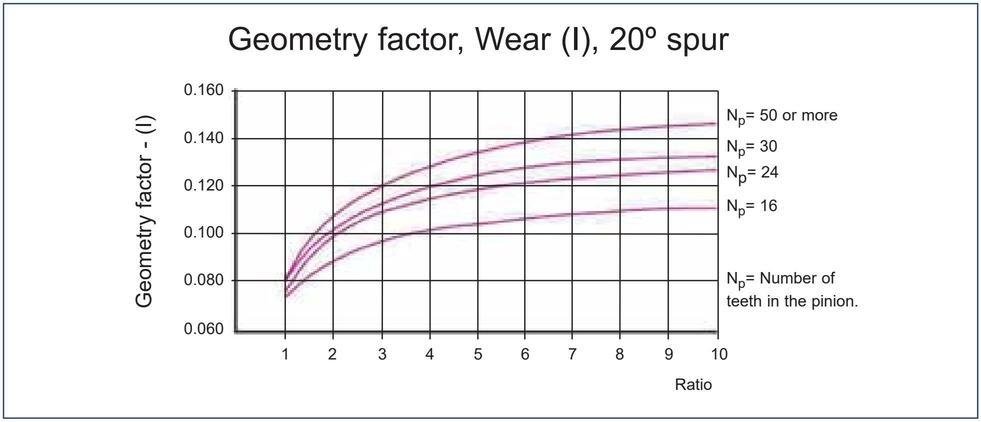
Material	Specification	Strength	Wear
Hardened stainless steel	17-4PH	1.00	1.00
Stainless steel	303S31	0.43	0.15
Stainless steel	316S31	0.47	0.20
Aluminium alloy	L168	0.37	0.10
Brass	CZ121	0.35	0.13

Example:
If the gears in the example on the previous page were made from 303S31

Fbs = 151.5 x 0.43 = 65.1 N
Fbw = 99.5 x 0.15 = 14.9 N

The application factors should be applied after the reduction for material.





5.2. Instrumentation and Feedback Systems

Gears and components designed for the precise transmission of angular position generally work at the low torque levels normally associated with servo components such as synchros, resolvers, optical encoders etc. Tooth loads of 1.2 N per mm face width should result in an adequate accurate life. Higher loads will tend to increase deflections of gear teeth, shafts, bearings etc, resulting in significant values of lost motion and a decrease in life.

Example: To find the maximum advisable torque on a gear 40 mm diameter x 3 mm face width.

Torque = force x radius = 1.2 x 3 x 0.04/2 = 0.072 Nm (10 oz.in.)

5.3. Anti-backlash Gear Spring Tension

In order for anti-backlash gears to function as anti-backlash devices, it is necessary to ensure that the spring tension will provide sufficient torque to overcome the friction and acceleration torque in the system, ie the spring torque must be capable of driving/accelerating the gear train and any associated components.

The spring tension capability of anti-backlash gears listed in this catalogue will adequately cope with the low torques normally encountered.

As a general guide, torque settings on anti-backlash gears of 1.059 to 1.765Ncm (1.5 to 2.5 oz.in.) will suffice in most applications. Ideally the spring torque should be set to the minimum at which the anti-backlash gear performs satisfactorily, thus avoiding unnecessary high preload on the gear teeth and premature wear.

6. Lost Motion and Backlash Control

The following section deals with lost motion, which we know to be one of the basic problems in designing fine pitch gear trains. The accepted definition of lost motion is the amount by which the output shaft may be turned without turning the input shaft.

It may be thought that lost motion is a function of the gear cutting operation alone, but, in fact the teeth of the gears may contribute very little to the overall lost motion value. A complete understanding of all the elements which induce lost motion is essential in order to achieve a well designed gear train. The following factors must be individually considered for their own contribution to overall lost motion in the gear train:

(a) Nominal centre distance.

(b) Centre distance tolerance.

(c) Size and tolerance of mating gears.

(d) Total composite error of gears.

(e) Fits between bores, shafts and bearings.

(f) Bearing accuracy (class).

(g) Radial play of bearings.

(h) Shaft straightness and alignment.

(i) Fits between electrical and/or mechanical component spigot diameters, and housing bores.

(j) Eccentricity and radial play of electrical and/or mechanical component shafts.

(k) Torsional elasticity.

(l) Differential expansion.

Each of the previous, except nominal centre distance, tend to induce a change in centre distance which will push together or pull apart the mating gears. This push-pull action produces two backlash values, minimum at the point of the tightest mesh, and maximum at the point of loosest mesh.

(a) Calculation of Nominal Centre Distance

Nominal centre distance can be considered as the starting point in the calculation of overall backlash values. Nominal centre distance is calculated by taking half the sum of the (theoretical) pitch diameters of the mating gears.

i.e. $$CD = \frac{PCD_1 + PCD_2}{2}$$

(b) Centre Distance Tolerance

Centre distance tolerance is an extremely important area for consideration. Any increase in centre distance in excess of the nominal value will increase the backlash. A decrease in nominal centre distance will decrease the backlash. In this case caution must be exercised to avoid interference between mating gears as a result of this decrease.

The relationship between centre distance change to backlash for 20º PA spur gear is given by:

$B = 2 \tan ø \cdot \Delta C$ where

B = Circumferential backlash

$ø$ = Pressure angle ($\tan 20° = 0.36397$)

ΔC = Distance between theoretical nominal and actual centre distance

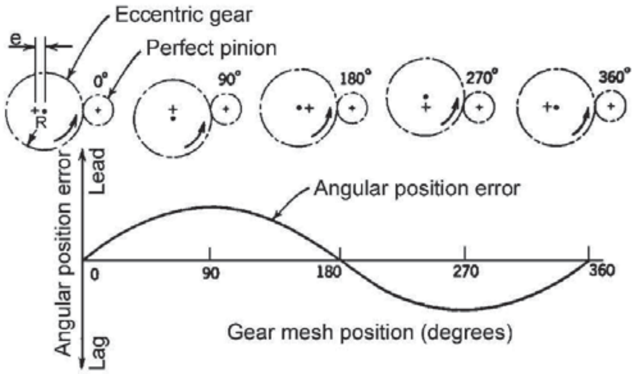
Note: Maximum Angular Backlash (minutes of arc) = $\dfrac{\text{Maximum Circumferential Backlash} \times 57.3 \times 60}{\text{Pitch Circle Radius}}$

7. Gear Error

The error function of a gear is approximately sinusoidal and for practical considerations can be assumed to be so. The first derivative of the time displacement curve yields the velocity function, therefore, the output velocity variable will also be an approximate sinusoid but the maximum velocity error will be displaced 90º from the maximum position error.

In summation, pitch circle runout will cause a sinusoidal error which is revealed as an output transmission error when meshed with a mating gear. The magnitude is given by the following example:

In the example on page T4-16, if the small pinion were not a perfect gear its error would be seen superimposed on the large gear error cycling at pinion frequency.



Angular position error $E_A = \dfrac{e}{R}\sin\theta$ Linear position error $E_L = e\sin\theta$

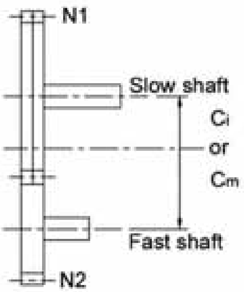
8. Transmission Accuracy of Gear Trains

The following section is based on work carried out by Reliance Gear Company (now known as Reliance Precision Limited) to provide some guidance in the design of accurate data transmission gearing.

The transmission error referred to by equations 1 and 2 below represent the maximum statistical point to point error during a forward and reverse cycle of a single pass of quality 14 anti-backlash gearing assembled in a data transmission gearbox.

For quality 10 or 12 gearing add 50% or 30% respectively to the error calculated for quality 14 gearing.

For average transmission error substitute the numbers 3.25 and 83 in the equations for the numbers 4.4 and 112 respectively.



$$\Sigma_1 = \left(1 + \frac{N_2}{N_1}\right) \cdot \left(\frac{112}{C_m} \text{ or } \frac{4.4}{C_i}\right) \text{------} (1)$$

$$\Sigma_2 = \left(1 + \frac{N_1}{N_2}\right) \cdot \left(\frac{112}{C_m} \text{ or } \frac{4.4}{C_i}\right) \text{------} (2)$$

C_m and C_i = Centre distance in mm and inches respectively.
N_2 and N_1 = Number of teeth in pinion and wheel respectively.
Σ_1 and Σ_2 = Maximum statistical transmission error in minutes of arc measured at the slow and fast shafts respectively.

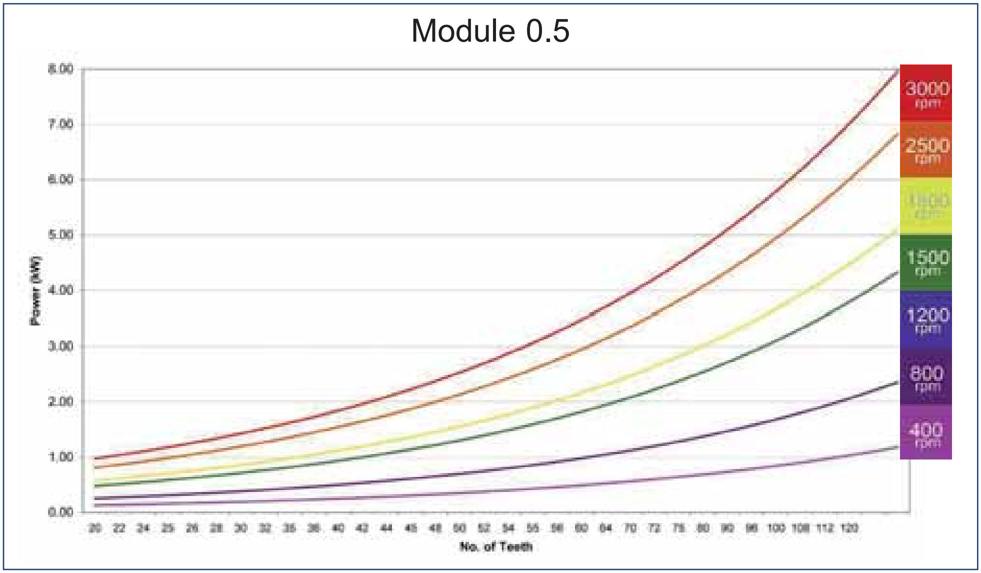
BACKLASH FOR STANDARD GEARS

The table below refers to the allowable backlash within the range of Spur, Bevel, and Worm gear pairs with a designated centre distance. The allowable backlash is necessary to absorb the deviations of noise and oscillation in order to maintain smooth operation.

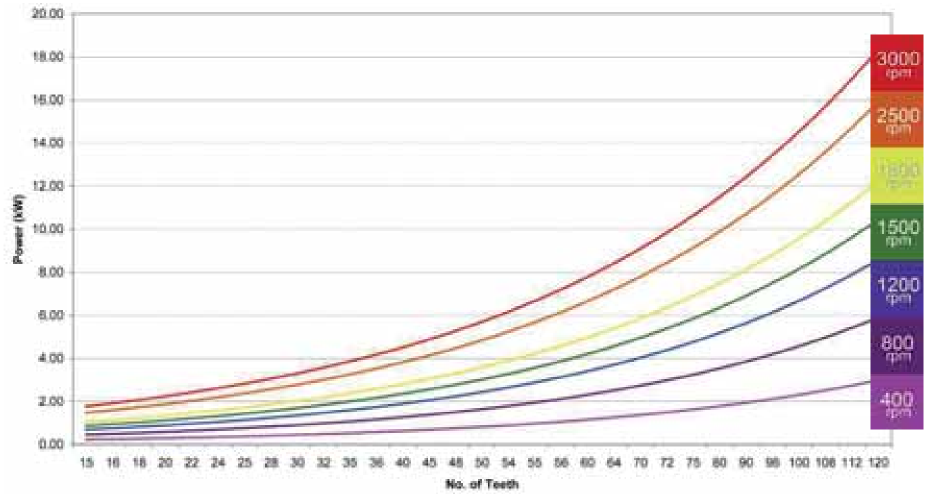
Gear Type	Condition	Module (m)	Backlash (mm)
Spur Gear	Brass/ Ground	< 0.9	0.02 - 0.06
	Brass	0.9 to 0.75	0.04 x m - 0.10 x m
	Ground	0.9 to 1.0	0.04 x m - 0.08 x m
Bevel Gear pair	Stainless steel or Brass	< 0.9	0.02 - 0.08
		0.9 to 1.5	0.05 - 0.12
Worm Gear pair Centres < 50 mm	Worm - Stainless steel Worm wheel - Brass	≤ 1.0	0.08 - 0.20

Note. These figures apply to the standard gear range only.
To convert Circumferential Backlash to Angular Backlash see page T4-15

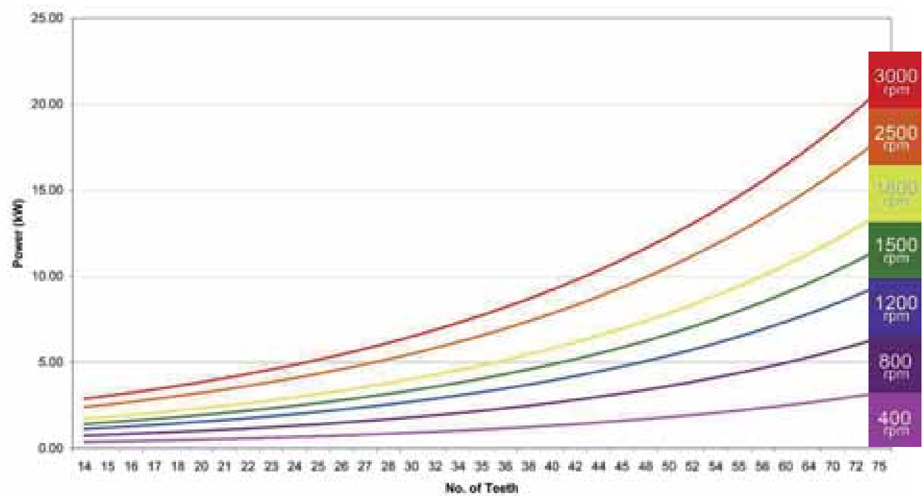
TYPICAL GROUND GEARS ALLOWABLE TRANSMISSION CAPACITY



Module 0.8



Module 1.0



WORMS AND WHEELS FORMULAE

Ratio (R) $= \dfrac{\text{No. of teeth on wormwheel (T)}}{\text{No. of starts on worm (t)}}$

Centre Distance (CD) $= \dfrac{\text{PCD worm}}{2} + \dfrac{\text{PCD wheel}}{2}$

Lead (L) = The axial distance by which a thread advances in one revolution $= \pi \times t \times m$

Where $m_{(metric)}$ = Axial module

$m_{(imperial)} = \dfrac{1}{DP}$

Actual outside diameter of worm **OD_w** $= PCD + (2 \times m)$

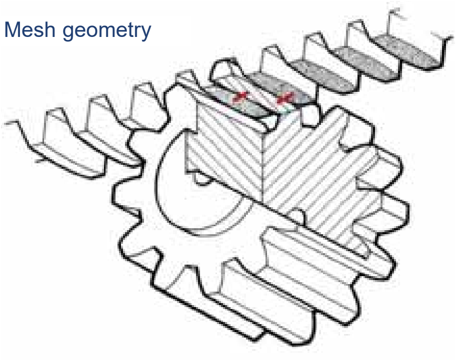
Typical outside diameter of wormwheel **OD_{ww}** $= PCD + (3 \times m)$

RACK MANUFACTURE

Reliance standard precision racks are produced by a thread grinding process, which generates teeth of helicoidal form. This provides two distinct advantages: very good pitch accuracy and sufficient tolerance of meshing conditions (within 0.25º) to make high precision alignment of the pinion unnecessary.

This feature will be appreciated from the diagram below. Slight misalignment of the straight-tooth pinion, in terms of deviation from a true right-angle between the axis and rack in either plane, results merely in a change of position of the contact points across the face.

Mesh geometry



Points of contact.
Standard pressure angle is 20º.
25º pressure angle available on request.

RACK STANDARDS AND TOLERANCES

Reliance precision racks are offered in four basic grades of accuracy through most of the range, please see the individual product pages for details. Grade 4b has been introduced to offer a lower cost grade 4 where a single rack is to be used in a non-butting application.
The tooth form is generally in accordance with BS 4582 part 1. fig 1. for metric racks.

Rack Grade	5	4	4b	3	2	1
Max pitch error between any two points per 300 mm of rack	0.005	0.008	0.008	0.015	0.025	0.050
Max end to end pitch error up to 300 mm of track*	±0.004	±0.004	±0.008	±0.008	±0.013	±0.025
Adjacent tooth error	0.0025	0.0025	0.0025	0.005	0.010	0.013
Pitch height variation	+0 -0.013	+0 -0.013	+0 -0.013	+0 -0.013	+0 -0.018	+0 -0.025

* Applies pro rata to length >300 mm

All dimensions in mm

ENGINEERING DATA

1. Linear Speed

Linear speeds of up to 10 m/s can be achieved with correctly installed rack and pinion systems. When specifying a system, care needs to be taken to ensure that the transducer count rates are not exceeded. With grease lubrication, care should be taken to ensure that the lubrication is not thrown off the pinion

2. Load Capacity

The following analysis is intended to give a guide to the load capacity of a rack system. To simplify the calculation a number of assumptions have to be made. In many applications this will give a conservative estimate of the gear capacity, therefore in critical applications an exact analysis must be completed. Please consult the relevant gear standards or Reliance Technical Sales.

The basic load capacity (F_b) of a rack and pinion is defined as the maximum linear force at which they can operate indefinitely.

F_b has two values: one calculated from tooth strength (F_{bs}) and one for tooth flank wear (F_{bw}). The useful or transmitted load capacity, F_t, is usually less than F_b due to transient or dynamic loads generated within the mechanism.

For tooth root strength $F_{ts} = F_{bs}/K_a$ K_a&C_a = application factors

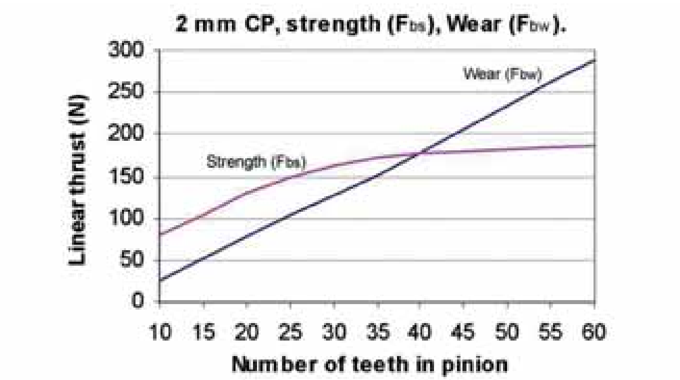
For tooth flank pitting (wear) $F_{tw} = F_{bw}/C_a$

Both calculations should be made and the lower value used.

The application factors K_a and C_a make allowance for any externally applied loads in excess of the nominal linear force F_b. These are most accurately determined by direct measurement. In determining application factors, consideration should be given to the fact that many prime movers develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment.

There are many possible sources of overload which should be considered, including system vibrations, acceleration torques, overspeeds, variations in system operation and changes in process load conditions. Impact loads due to reversing across backlash can be significant in servo systems.

As a general guide application factors for a motor gear system range from 1.0 for uniform loads up to 1.75 where heavy shock loads are anticipated.



Technical Information

The previous graph has been calculated in accordance with AGMA 2001-B88 for a life of at least 10^8 load cycles, and a rack hardness exceeding 50 HRc and pinion material 17-4PH. For alternative pitches and materials the graph values need to be modified as shown in the table below:

Pitch and Rack/Pinion Material Modification Factors

Rack	Pinion	Pitch (mm)	Strength	Wear
Hardened Round Rack (hardness>50 HRc)	17-4PH	1	0.50	0.50
	316	1	0.23	0.10
	PEEK polymer	1	0.04	0.01
Rectangular Rack (hardness 35-45 HRc)	17-4PH	1	0.38	0.28
		2	0.75	0.56
		2.5	0.94	0.70
	316	1	0.23	0.10
		2	0.47	0.20
		2.5	0.59	0.25
	PEEK polymer	1	0.04	0.01
Tubular and Round Rack	17-4PH	1	0.23	0.10
	316	1	0.23	0.10
	PEEK polymer	1	0.04	0.01

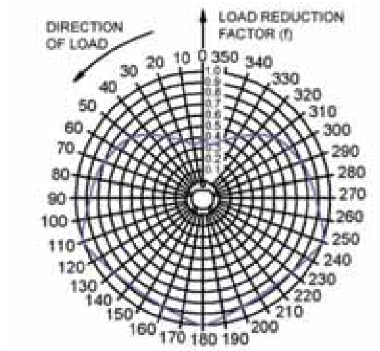
Example:
A 40 tooth, 1 mm CP pinion material 316 meshing with rack of hardness <50 HRc.
The application factors should be applied after the reduction for material and pitch.

$$F_{bw} = 175 \times 0.10 = 17.5 \text{ N}$$
$$F_{bs} = 170 \times 0.23 = 39.1 \text{ N}$$

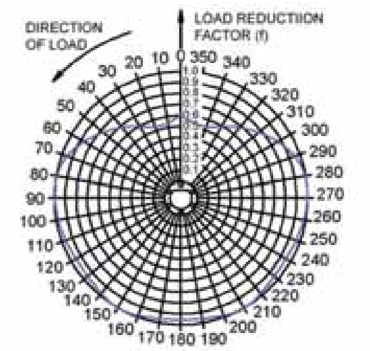
3. Bearing Capacity

When linear bearings are used with the hardened round bar racks the capacity of the support bearings needs to be considered. Where possible the bearings should be positioned with all the ball rows running on the rack shaft. However, it is important that the balls do not run on the edges of the teeth. If necessary the 5 and 6 row bearings can be used with 1 row above the teeth. In this scenario, the manufacturer’s ratings apply with a modification for the direction of the load application. The factors given in the following charts should be substituted for the bearing manufacturer’s load reduction.

Closed bearing 5 rows of balls



Closed bearing 6 rows of balls



4. Lubrication

Lubrication is not required when using PEEK polymer pinions. For other combinations unlubricated systems are not recommended. Measurement applications should use a very thin coat of light oil, in many machine tool applications stray cutting oil is sufficient. Grease lubrication is recommended for higher loads, but care should be taken to ensure the lubrication is not thrown off the pinion at speed.

INSTALLATION

The installation techniques differ according to the type of rack. All racks should be mounted with teeth pointing downwards wherever possible so that dust etc cannot settle in them.

1. Soft Round and Tubular Rack

Plastic moulded bearings are recommended for use with soft round and tubular racks, these can be found in the Bearings and Spacers section of the Reliance catalogue. Round racks are not recommended for multi-section use.

2. Hardened Round Rack

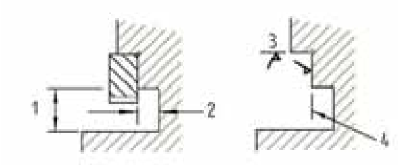
Bearings for the round bar rack should be fitted in accordance with the manufacturer's instructions. It is important that the balls do not run on the edge of the teeth. Round racks are not recommended for multi-section use.

3. Rectangular Rack

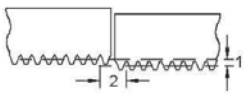
Reliance rectangular section rack is manufactured to enable butting to form infinite lengths. Socket head cap screws, plain washers and a thread locking adhesive are preferred for mounting. Dowels are not recommended. The pitch line of the rack in its constrained position must be straight to obtain maximum accuracy. To avoid distortion, racks should be screwed to a machined flat surface.

Machining requirements for rack location

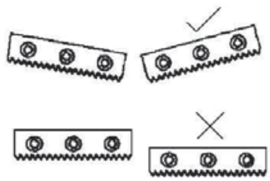
1. Pinion clearance
2. Clearance required if:
   a) anti-backlash pinions are used
   b) full face of rack is to be used
3. Abutment
4. Mounting face



To align racks, two adjustments need to be made, pitch line alignment and pitch adjustment. The pitch line straightness is not critical (see drawing below) but steps at the joints should be avoided as they can lead to excessive noise and wear.



1. Pitch line alignment
2. Pitch adjustment and error compensation



Pitch Line Alignment

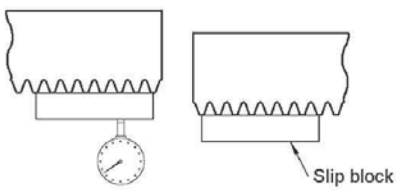
There are three methods of setting the pitch line at a joint. These are:

i) Setting the base of the racks against an abutment perpendicular to the mounting face. The misalignment is then governed by the rack pitch line to base tolerance.

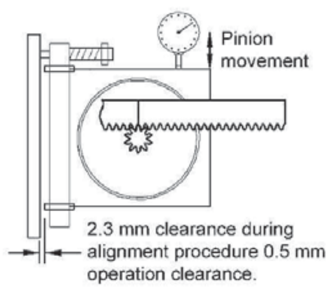
ii) Using the tops of the rack teeth as a reference. These are parallel to the pitch line within 0.008 mm. Use a short straight edge (eg. slip block) as shown below.

iii) The best measurement of the pitch line is with the pinion installed on a flexplate. A dial indicator fitted as shown gives a direct reading of the pitch line straightness.

Pitch line alignment using slip block



Dial indicator carried with flexplate

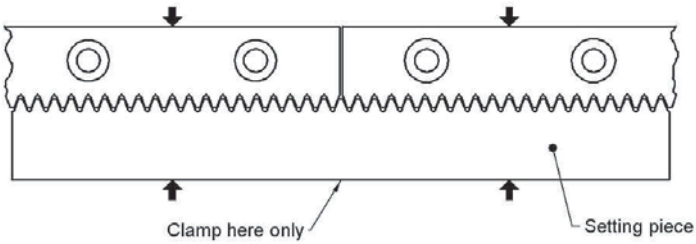


The flexplate spring loads the pinion into mesh on both flanks of the teeth, ensuring complete backlash elimination.

Pitch Adjustment and Error Correction

Pitch accuracy can be obtained by one of three methods depending upon accuracy required.

GRADE 1 (and for the initial setting of all grades)
For pitch accuracy across the joint of ±0.020 mm the Rack Setting Piece is the simplest method.



GRADE 2, 3 or 4
After initial setting and with the measuring system functioning, length bars may be used as references. Checks made against these allow adjustment to be made within the system resolution.

GRADE 3, 4 or 5
After initial setting and with the measuring system functioning, comparison should be made with a laser measuring system. This allows pitch adjustment and machine error compensation within the system resolution over the full travel of the axis.

RACK APPLICATIONS

Reliance precision racks are manufactured in both round and rectangular sections, and can be used for both measurement and actuation. In general the smaller pitches (1mm) are ideal for measurement, as the smaller pinion diameter gives higher linear resolutions.
The larger pitches (2 mm and 2.5 mm) allow a higher load capacity.

For most applications the rack can be used for both the feedback and the actuation. In very precise applications we recommend that an unused section of the actuation rack is used for feedback. Alternatively a separate rack can be used.

All Reliance racks are calibrated to measure correct at 20ºC using a temperature compensated laser. Calibration graphs can be supplied if required.

RACK ACTUATOR

Information about the Racktuator™ is provided on page T2-9.

Leadscrew and Nut Assemblies

FEATURES

Reliance's precision leadscrew assemblies are designed specifically for motion control applications where accuracy must be maintained. Rather than being adaptations of general purpose screws or nuts they have a precision rolled screw thread which has been designed for maximum life and quiet operation.

A further enhancement available on stainless steel leadscrews up to 2.4 metres long is a specially formulated TFE coating which can extend normal nut life by up to 300%.

Innovative anti-backlash nut designs provide assemblies which are wear compensating with low frictional drag torques and excellent positional repeatability.

Reliance stainless steel leadscrews offer the following:

1. High Accuracy
Precision thread rolling process provides a standard lead accuracy of 0.0006 mm/mm. Higher accuracies up to 0.0001 mm/mm can be provided.
The unloaded repeatability of anti-backlash assemblies is within 0.0013 mm.

2. Long Life
More than 7.5 million metres of travel can be expected.

3. Low Drag Torque
An anti-backlash nut design which does not require high spring forces to maintain bi-directional anti-backlash characteristics gives very low nut to screw friction.

4. Low Maintenance
Self lubricating and wear compensating nuts eliminate the need for repeated lubrication or adjustment.

5. Wide Range
Diameters from 3.2 mm to 24 mm.
Leads from 0.30 mm to 92 mm.
Lengths up to 4 metre.

6. Custom Thread Design
Unique thread form designed specifically for leadscrews in anti-backlash applications.

7. Smooth Quiet Operation
No recirculating ball noise or metal to metal contact.

8. Lower Cost
Less than comparable ball screws or ground leadscrews, while still providing high accuracy and long life.

9. Modifications
Special leadscrew ends and other leads are available on the stainless steel leadscrew range in selected sizes. Please contact Reliance Technical Sales or refer to the leadscrews modification section of this brochure.

ENGINEERING DATA

1. Lead

The lead of the screw is the amount of linear movement of the nut for one revolution of the leadscrew.

2. Drive Torque

The required motor torque to drive a leadscrew assembly is the sum of three components: inertial torque, static friction torque and torque to move the load. Additional torque associated with driving and supporting the leadscrew must also be considered.

Inertial torque: $T = I\alpha$

I = Inertia of leadscrew (kgm^2)

α = Angular acceleration ($rads/s^2$)

Static friction torque: Anti-backlash leadscrews are typically supplied with a static frictional torque of 0.007 - 0.05 Nm. Higher pre-load forces lead to higher frictional drag torques but better anti-backlash characteristics.

Torque to move load: The torque to move a certain load is a function of the lead and efficiency of the leadscrew assembly.

$$\text{Torque} = \frac{\text{Load x Lead}}{2\pi \text{ x Efficiency}}$$

Torque = Newton metres

Load = Newtons

Lead = Metres

(Note - efficiency of 70% would require 0.7 in these equations)

4. Backdriving

In general when the screw pitch is less than 1/3 its diameter and the screw is uncoated, backdriving will not occur. (Coated screws require to be 1/4 diameter). For higher leads where backdriving is likely, the torque required for holding a load is as follows:

$$\text{Backdrive torque} = \frac{\text{Load x Lead x Efficiency}}{2\pi}$$

Torque = Newton metres

Load = Newtons

Lead = Metres

Small vibrations in the system may break the static friction initiating backdriving, therefore, for small critical applications use smaller lead or an external locking device.

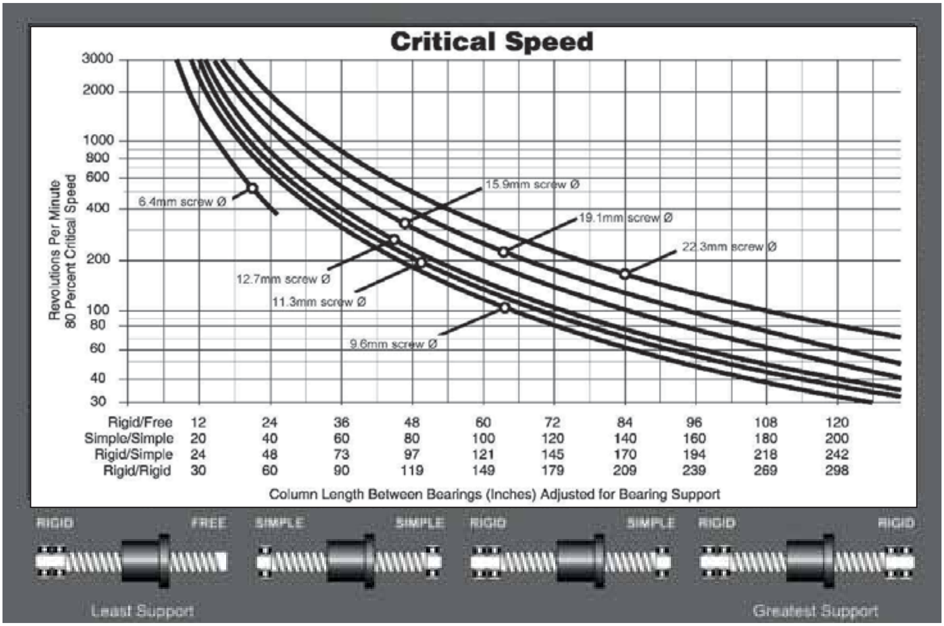
5. Traverse Speed

The polyacetal nut materials provide long wear-life over a wide variety of conditions, but very high loads and/or high speeds will accelerate nut wear. We recommend the following maximum linear traversing speeds for optimum life:

Lead	Maximum traverse speed
2.5 mm - 12 mm	100 mm/sec
12 mm - 25 mm	250 mm/sec
25 mm - 60 mm	760 mm/sec

6. Critical Speed

This is the rotational speed at which a leadscrew will experience vibration or other dynamic problems. See the critical speed chart below to determine if the application parameters result in speeds approaching critical.
To minimise critical speed problems use a longer lead, choose a larger diameter screw or increase the bearing mount support.



7. Maximum Load

Although the leadscrew assemblies are able to withstand relatively high loads without catastrophic failure, these units have been designed to operate with the loads shown on the product pages.

8. Efficiency

The efficiency of a leadscrew varies with the lead angle of the screw. The theoretical maximum efficiencies of all our leadscrews are given in the part number tables on the product pages. These have been calculated using the static coefficient of friction 0.08. For applications where the dynamic efficiency is critical please contact Reliance Technical Sales.

9. Leadscrew Inertia

Values of leadscrew inertia are given in the Typical Mechanical Properties chart on the next page.

10. Screw Straightness

Typical screw straightness is 0.25 mm/metre.

11. Leadscrew Interfacing

Examples of machined end options can be found on pages 7-27 to 7-35.

Physical Properties					
Leadscrew		Nuts		Assembly	
Material	Surface Finish	Material	Tensile Strength	Operating Temp. Range	Coefficient of Friction Nut to Screw
Stainless steel 303 series	Better than 0.4 µm	Polyacetal with lubricating additive	67 N/mm^2 9,700 psi	0 - 93°C	Static = 0.08 0.08# Dynamic = 0.15 0.09# # - with TFE coating

Typical Mechanical Properties				
Leadscrew Series	Static Frictional Drag Torque (Nm)	Screw Inertia kg m^2/m	Anti-backlash Life +	
			Plain Screw	TFE Coated Screw
LPX6 LPX10 LPX11 LPX13 LPX16 LPX19 LPX22 LPX24	Free wheeling	8.340×10^{-7} 4.170×10^{-6} 9.730×10^{-6} 1.446×10^{-5} 3.948×10^{-5} 8.479×10^{-5} 1.612×10^{-4} 2.030×10^{-4}	N/A Typical Backlash 0.076-0.25 mm	N/A Typical Backlash 0.076-0.25 mm
LNTG6 LNTG8 LNTG10	0.004-0.01 0.01-0.02 0.01-0.02	8.340×10^{-7} 1.390×10^{-6} 4.170×10^{-6}	0.12 - 0.25 million metres	0.38 - 1.0 million metres
LAB6 LAB10 LAB11 LAB13 LAB16 LAB19 LAB22 LAB24	0.004-0.01 0.01-0.02 0.015-0.03 0.015-0.03 0.02-0.05 0.03-0.063 0.03-0.063 0.03-0.063	8.340×10^{-7} 4.170×10^{-6} 9.730×10^{-6} 1.446×10^{-5} 3.948×10^{-5} 8.479×10^{-5} 1.612×10^{-4} 2.030×10^{-4}	0.12 - 0.25 million metres	0.38 - 1.0 million metres
LAF6 LAF8 LAF10 LAF11 LAF13 LAF16	0.004-0.02 0.01-0.03 0.01-0.03 0.015-0.04 0.02-0.05 0.03-0.055	8.340×10^{-7} 1.390×10^{-6} 4.170×10^{-6} 9.730×10^{-6} 1.446×10^{-5} 3.948×10^{-5}	1.0 - 1.5 million metres	3.8 - 5.0 million metres
LAK8 LAK10	0.01-0.02 0.01-0.02	1.390×10^{-6} 4.170×10^{-6}	2.0 - 2.5 million metres	4.5 - 5.8 million metres
LWD6 LWD8 LWD10 LWD11 LWD13	0.03 max 0.04 max 0.04 max 0.06 max 0.06 max	8.340×10^{-7} 1.390×10^{-6} 4.170×10^{-6} 9.730×10^{-6} 1.446×10^{-5}	2.5 - 3.15 million metres	5.0 - 6.35 million metres

Typical Mechanical Properties (continued)				
Leadscrew Series	Static Frictional Drag Torque (Nm)	Screw Inertia Kg m^2/m	Anti-backlash Life +	
			Plain Screw	TFE Coated Screw
LNTB6 LNTB8 LNTB10 LNTB11 LNTB13 LNTB16 LNTB19 LNTB22 LNTB24	0.004-0.01 0.01-0.02 0.01-0.02 0.01-0.02 0.015-0.04 0.015-0.04 0.02-0.05 0.03-0.06 0.03-0.06	8.340×10^{-7} 1.390×10^{-6} 4.170×10^{-6} 9.730×10^{-6} 1.446×10^{-5} 3.948×10^{-5} 8.479×10^{-5} 1.612×10^{-4} 2.030×10^{-4}	2.5 - 3.15 million metres	5.0 - 6.35 million metres
LCM6 LCM8 LCM10	0.03 0.04 0.04	8.340×10^{-7} 1.390×10^{-6} 4.170×10^{-6}	1.0 - 1.5 million metres	3.8 - 5.0 million metres
LAX13 LAX16	0.015-0.04 0.015-0.04	1.446×10^{-5} 3.948×10^{-5}	5.0 - 5.7 million metres	7.6 - 8.8 million metres

\+ Life will vary with loading, operating environment and duty cycle.
Longer screw leads generally give longer life.

TFE COATED LEADSCREW ASSEMBLIES

The TFE coating is designed to supply a more even distribution of lubricant than is normally achieved when using standard self lubricating plastics on steel. The entire screw surface is coated which gives an extremely even lubrication distribution and an expected increase in normal nut life of up to 300%. Lubrication to the screw/nut interface occurs by the nut picking up TFE particles from the coating as well as from migration of the internal lubricant from within the plastic nut.

Although care should be taken to ensure that chips and voids do not occur in the coating, small voids have been shown to have little effect on the system performance. The lubricant, although solid, has some of the “spreading” ability of fluid lubricants. When machining for bearing ends, soft fixtures are recommended.

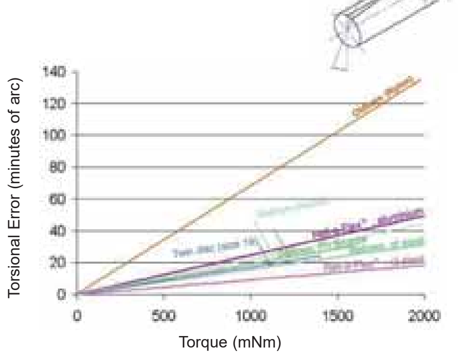
TFE coated screws provide the maximum level of self-lubrication and should not be additionally lubricated or used in environments where oils or other lubricant contamination is possible.

Technical Information

TORSIONAL STIFFNESS

This is the characteristic that describes the angular deflection when a torque is applied. High torsional stiffness contributes to increased accuracy and system response. It is essential for accurate feedback applications.
Applications that are subject to shock loads may require a less stiff coupling to reduce the peak torques and avoid premature failure or slipping clamps.

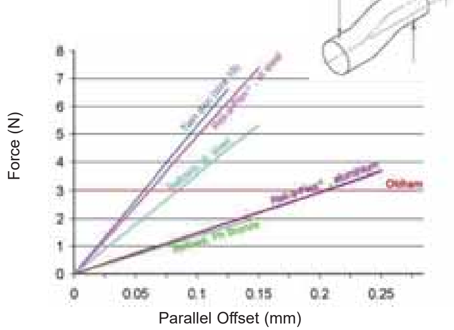
Test samples are standard products with a nominal O.D of 20 mm



RADIAL COMPLIANCE

This is the characteristic that describes the force the coupling applies on the support bearings when the shafts are misaligned. High radial compliance is essential to provide low bearing loads.

Test samples are standard products with a nominal O.D of 20 mm



TORQUE CAPACITY

In general, the rated torque figures are based on $>10^6$ torque reversals and the peak torque should not be applied for more than 1% of the duty cycle.

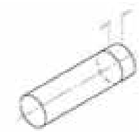
SHAFT MISALIGNMENT

The most common type of misalignment is a combination of angular, parallel and axial misalignment and occurs due to the build-up of tolerances as associated parts are assembled together. As these accumulate randomly, worst-case misalignment should be calculated and used to select the correct coupling to avoid premature failure.

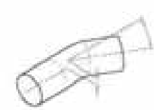
Angular

Parallel

Axial



Combined



TRANSMISSION ERROR

Often referred to as kinematic error, this is the total error in the driven shaft position with regardto the driving shaft position. In a system the following factors must be individually considered to determine their overall effect.

a. Backlash .... internal clearance related
b. Torsional wind up .... torsional stiffness related
c. Velocity error.. coupling design related

a. Backlash
Is the amount of free rotational movement inherent in the coupling under zero or near zero torsional loads. Only the Oldham coupling type in this catalogue is susceptible to slight backlash.

b. Torsional Wind Up
In applications where the resistance is frictional, the driven shaft will experience a position lag, which will double with direction reversal, proportional to the torsional stiffness.

During operating mode, the inertia and the torque will cause a momentary lag but this will not be seen at standstill.

c. Velocity Error
In general, couplings with double flexing elements (Reli-a-Flex®, Bellows and Twin disc couplings) will introduce negligible velocity errors.

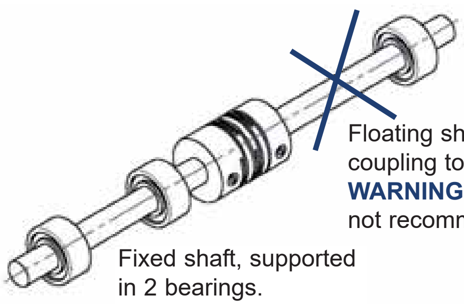
Velocity errors occur with angular misalignment and are proportional to shaft angle.
Only the Oldham coupling type in this catalogue is susceptible to this error.

LUBRICATION

This is not required on any of the couplings in this catalogue.

FLOATING SHAFTS

We do not recommend the use of couplings in this catalogue for floating shafts, where one or both ends of a shaft are supported by a coupling.



Technical Information

TORSIONAL RESONANCE

The torsional natural frequencies of a system are dependent on the mass/elastic characteristics of the various inertias and connecting shafts. Torsional resonance can occur under certain conditions when the natural frequency of the system is close to the excitation frequency of the driving system. It is most likely to occur when the load is predominantly inertial and can occur in closed loop position or velocity control systems, leading to torsional vibrations which in severe circumstances can destroy the coupling.

Choosing a coupling that operates well above or well below the operating frequencies can help to avoid premature failure.

The resonant frequency of a system can be calculated from the following:

$$F_R = 1/2\pi \times \sqrt{(1/J_M + 1/J_L) \times 10.8/\pi \times C_T}$$

where
F_R = Resonant frequency (Hz)
J_M = Motor inertia (Kgm^2)
J_L = Load inertia (Kgm^2)
C_T = Coupling torsional stiffness (mNm/min)

RELI-A-FLEX® INSTALLATION

Couplings are available with either clamp or set screw mounting. Clamp fastening, both Reli-a-Grip™ and traditional, allows repeated repositioning of the coupling on the shaft leaving the shaft unmarked. The effectiveness of the clamp is dependent on the diameter being a 'close' fit in the coupling bore. Use of Reliance components will ensure that the clamp works correctly.

Set screws provide an effective but non-adjustable means of connecting couplings and shafts. Ideally the shafts should have a small flat in the area of the screw, which allows the set screw to seat below the surface of the shaft.

OLDHAM COUPLING, SOLID COUPLING AND COLLAR INSTALLATION

Oldham Couplings
Ensure that the misalignment between shafts is within the coupling's ratings. Slide a hub onto each shaft to be joined with the drive tenons facing each other. Rotate the hubs on the shaft so the drive tenons are located 90° from each other. Place a torque disc so one groove fits over the drive tenons of a hub and centre the disc by hand.

Insert a shim with the thickness of the coupling's axial motion rating into the groove of the torque disc. Slide the tenons of the second hub into the mating groove in the disc until it touches the shim stock.

Fully tighten the screw(s) on each hub to their recommended seating torque. Remove the shim stock to leave a small gap between the top of the drive tenons and the torque disc to allow for axial movement.

Solid Couplings

Align the coupling on the two shafts to be connected. Tighten the Nypatch® clamp screws in two stages. Starting with the inside screws, tighten to half of the recommended seating torque. Repeat for the outside screws, again tightening to half of the recommended seating torque (on two-piece collars be sure to maintain the gap between the two halves of the coupling during installation). Tighten screws to the full recommended seating torque following the same pattern, beginning with the inside screws.

Collars

Use collars as they are received.
Wipe the bore clean and apply a thin coat of light oil to the shaft. Place collar in desired location on shaft and tighten the collar until a slight resistance is felt (on two-piece collars be sure to maintain the gap between the two halves of the collar during installation). Bring collar into final position and tighten screws to the full recommended seating torque.

FEATURES

The Reliance range of precision slides includes both ball and crossed roller units. Load capacities from 1.5 to 12580 N are available. Ballslides are available in both stainless steel and aluminium. Crossed roller slides are available in aluminium only.
These units offer the designer:

- Pre-assembled units allowing quick and simple assembly.
- Factory set preload to prevent side play and backlash and to control friction.
- Low particle production for use in clean/medical environments.
- Low inertia and light weight allowing low powered rapid traverse.
- High straight line accuracy of 0.0001 mm per mm travel.

1. Ballslides
Manufactured from aluminium, these slide units offer ultra low friction, high load capacity and long life. The base and slide are ready machined for mounting. Modifications may be made to suit special requirements. Complete special slides can also be supplied. Please contact us.

2. Crossed Roller Slides
When compared to ballslides these units offer equal size but higher load capacity and accuracy. They are also able to operate with high cycling rates and higher shock or cantilevered loads.

3. Rack Driven Ball Slides
The addition of a small high precision rack along the side of a ballslide offers the option either to drive, measure position, or both, at very high speeds and loads.

ENGINEERING DATA

For the highest accuracy, the load should be centred over the table or bed, allowing enough additional length to avoid reaching the maximum stroke length. To achieve the expected accuracy and life, the mating surfaces used to mount the slide should be flat. In extreme circumstances ‘potting’ of the base may be required.

Please refer to the product dimensions when selecting the fixings to avoid contact between screws and moving slide sections.

1. Vertical Applications
When using ball or crossed roller type slides in vertical applications, the position and manner of the load, and the effects of gravity should be given extra consideration. Limiting the travel with positive stops also extends life instead of relying on the ball or roller retainer to act as a stop.

2. Service Life
The theoretical service life of a slide based on L_{10} life is as follows:

Ballslides	Crossed roller slides
$L_{10} = (C/P)^3 \times 50\text{x}10^3$	$L_{10} = (C/P)^{10/3} \times 50\text{x}10^3$

Where : L_{10} = Life at 90% reliability (m)
C = Dynamic load rating (N)
P = Calculated load (N)

3. Lubrication

All types of slides can use similar lubricants but require them under different conditions.

Recommended Lubricants	General Application High quality turbine oil Lithium soap based grease (NLGI No. 2)
	Clean Environments Kluber Isoflex Topas NCA 52

4. Temperature Limits

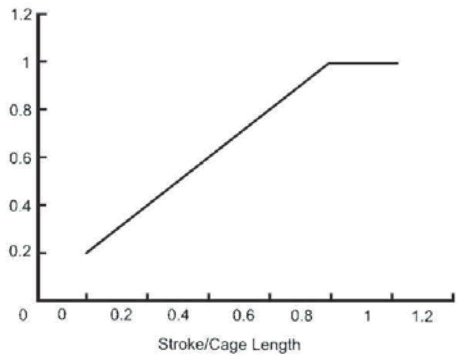
The maximum temperature is limited to 65ºC (150ºF) by the rolling element retainers. High temperature retainers can be supplied to operate up to 100ºC (212ºF) and although the slides can operate at higher temperatures this will reduce their life. Please contact us for further details.

MINIATURE STROKE SLIDES

1. Rating Life Calculation and Short Stroke Factor Diagram

$$L = K_{st} \left(\frac{C_{100B}}{P}\right)^3 \cdot 10^5$$

$$L_h = \frac{L}{2 \cdot s \cdot n \cdot 60} = K_{st} \frac{L}{V_m} \cdot \left(\frac{C_{100B}}{P}\right)^3$$



2. Rating Life L

The rating life of the RST miniature stroke slide series can be calculated by using the formulae above, in accordance with ISO 14728-1.

3. Lubrication

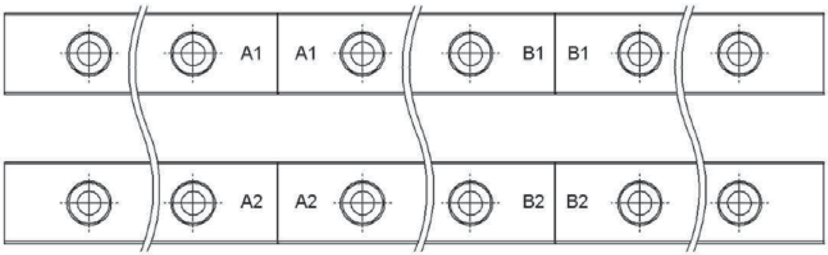
The lubrication of the RST miniature stroke slide series can be fullfilled by directly adding the lubricant onto the raceway of the rail.

MINIATURE LINEAR GUIDES

1. Rail Butt-Jointing

When a longer rail is required than the maximum standard length available, two or more rails can be butt-jointed to create the desired length. When ordering add a -J to the end of the part number.

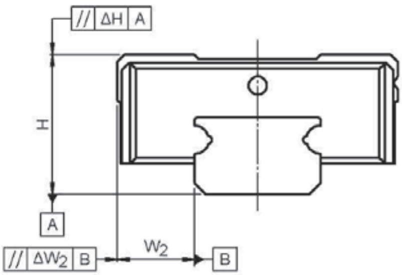
To ensure that the rails are mounted in an accurate and consistent manner they will be marked up as a matching pair when manufactured. The marking system for a rail that has has been ordered as a -J to be butt-jointed is shown below, where matching pairs have the same marking.



2. Accuracy

Miniature Linear Guides are available in three accuracy grades P, H and N.

Accuracy classes (μm)		Precision **P**	High **H**	Normal **N**
Tolerance of dimension height H	H	± 10	± 20	± 40
Variation of height for different runner block on the same position of rail	$\triangle H$	7	15	25
Tolerance of dimension width W	W_2	± 15	± 25	± 40
Variation of width for different runner block on the same position of rail	$\triangle W_2$	10	20	30



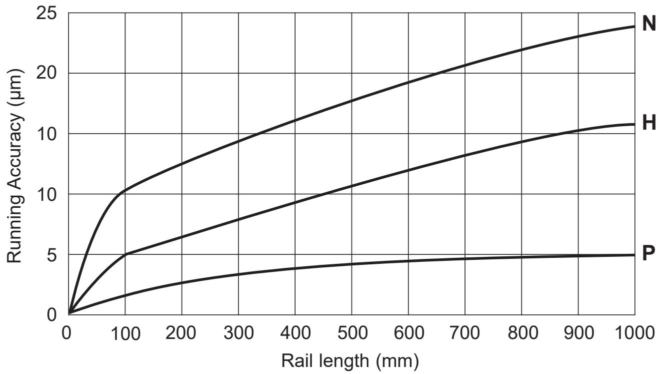
3. Speed

For the SS/ZZ variant:

Maximum speed: v_{max} = 3 m/s
Maximum acceleration: a_{max} = 250 m/s2 (If preload V0, maximum acceleration is 40 m/s2)

For the EE/EU/UZ variant:

Maximum speed: v_{max} = >5 m/s
Maximum acceleration: a_{max} = 300 m/s2 (If preload V0, maximum acceleration is 60 m/s2)



4. Pre-load

Miniature Linear Guides are available in three different grades of pre-load V0, VS and V1. The amount of pre-load can enhance stiffness, precision and torsional resistance but affects life and friction.

Pre-load Type	Model Code	Clearance (µm)						Application
		3	5	7	9	12	15	
Clearance	V0	+3 to 0	+3 to 0	+4 to 0	+4 to 0	+5 to 0	+6 to 0	Very smooth
Standard	VS	+1 to 0	+1 to 0	+2 to 0	+2 to 0	+2 to 0	+3 to 0	Smooth Precision
Light Pre-load	V1	0 to -0.5	0 to -1	0 to -3	0 to -4	0 to -5	0 to -6	High rigidity Minimal vibration High precision Load balance

5. Operating Temperature

Miniature Linear Guides can operate in temperatures ranging from -40°C to +80°C. Temperatures of 100°C can be reached for short term operation.

INTRODUCTION

Timing belts are endless toothed belt systems available in 2.5 mm and 5 mm pitch; intended for applications requiring a level of power transmission.

ENGINEERING DATA

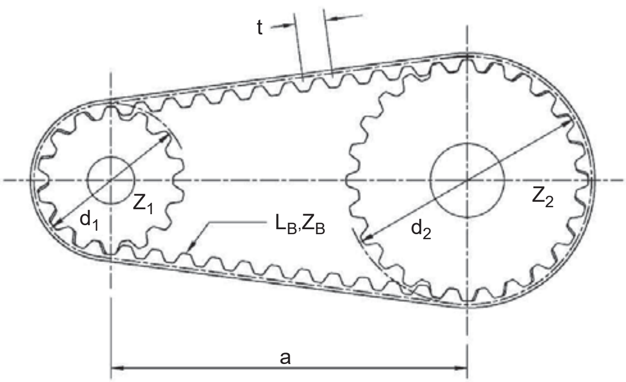
1. Belt and Chain Length

Knowing the centre distance, the belt length can be calculated from the following:

For ratios = 1:1 $$L_B = Z_1 \times t + 2a$$ [mm]

For ratios ≠ 1:1 (approximate formula) $$L_B \approx \frac{t}{2}\left(Z_2 + Z_1\right) + 2a + \frac{1}{4a}\left[\frac{\left(Z_2 - Z\right)t}{\pi}\right]^2$$ [mm]

a	=	Centre distance			
L_B	=	Belt length	t	=	Belt pitch
d_1	=	Pitch diameter small pulley	d_2	=	Pitch circle diameter large pulley
Z_1	=	No of teeth, small pulley	Z_2	=	No of teeth, large pulley
Z_B	=	No of teeth in belt	Z_e	=	No of teeth in mesh



Technical Information

2. Centre Distance Calculation

Knowing the belt length, the centre distance can be calculated from the following;

For ratios = 1:1

$$a = \frac{(Z_B - Z_1)t}{2} \quad \text{[mm]}$$

For ratios ≠ 1:1
(approximate formula)

$$a \approx \frac{L_B - \frac{\pi}{2} \times (d_2 + d_1)}{4} + \sqrt{\left(\frac{L_B - \frac{\pi}{2} \times (d_2 + d_1)}{4}\right)^2 - \frac{(d_2 - d_1)^2}{8}} \quad \text{[mm]}$$

$$d_1 = \frac{Z_1 \times t}{\pi} \quad \text{[mm]} \qquad d_2 = \frac{Z_2 \times t}{\pi} \quad \text{[mm]}$$

3. Design Guidelines

Timing belt efficiency ranges from 95 to 98%, better than flat vee belts which rely on friction to transmit power. The 2.5 mm and 5 mm pitch timing belts are manufactured in wear resistant polyurethane with high grade steel wire tension members, therefore any elongation due to load and pre-tension will follow Hookes' law. The manufacturing process for these timing belts produces the 'classical' trapeziodal tooth form to close tolerances. This ensures an even distribution of load during use and the transmission of high torques. These belts are suitable for indexing, positioning and conveying drives.

It is possible to design drives with fixed centres but generally the drive centres should be adjustable or have idler pulleys. This is particularly important in multi-shaft or high power drives. The idler pulleys should be fitted to the slack side of the drive and must not be spring loaded. Timing belt drives do not require as much tension as other belt drives which depend on friction to transmit load. The belt should be installed with a snug fit, neither taut nor loose. As a general guide the correct level of tension can be determined by measuring the force necessary to deflect the belt an amount equal to 1/64th of the span centres "a". Values for the measuring force recorded on a spring balance applied mid-span should be within 20% of the values shown below.

2.5 mm - 0.07kg   5 mm - 0.30kg

The belts must be rigidly mounted. Variation in centre distance can lead to premature wear. The belt and pulley system must be assembled loose to prevent over stretching. The belts are guided on the pulleys by flanges. One pulley should be flanged on both sides, or two alternative flanges provided, one on each pulley. For drives with vertical shafts, both pulleys should be flanged on both sides.

For a belt to transmit full power, a minimum of 6 teeth must be in mesh on each pulley.
The number of teeth in mesh can be determined from the following formula:

$$Ze = \frac{Z_1}{180} * \text{arc cos} \frac{(Z_2 - Z_1) \times t}{2 \pi a}$$

Number of teeth in mesh calculation is always based on the smallest pulley.

To minimise belt fatigue, pulleys with a minimum of 20 teeth are recommended. As a general guide larger pulleys reduce the amount of belt flexing and therefore improve belt life.

SPECIFICATION

The first step in choosing the correct bearing for an application is to determine the forces which it will support in service. The forces will depend on the exact configuration of the system and will probably include some, or all, of the following:

- The weight of the shaft, including gears and other shaft attachments.
- Gear mesh reaction forces, due to torque transmission (see below).
- Gear separation due to anti-backlash forces.
- Forces due to belt or pulley tensions.
- Axial pre-load forces.

GEAR MESH REACTIONS

In order to calculate the loads which will be applied to the bearings in the simply supported spur gear pass arrangement shown on the next page, it is first necessary to calculate the forces at the gear mesh.

The tangential force at the gear mesh can be calculated from the following equation:

$$W_t = T/r$$

where T = Torque
and r = Radius

and the separating force at the gear mesh can be calculated from:

$$W_r' = W_t \tan\phi_t$$

where ϕ_t = transverse pressure angle
= normal pressure angle for spur gears
= 20º for our standard gears

$$W_r' = 0.364W_t$$

(for 20º pressure angle spur gear)

If required, the total radial load at the gear mesh can be calculated from the final equation:

$$W_r = \sqrt{(W_t)^2 + (W_r')^2}$$

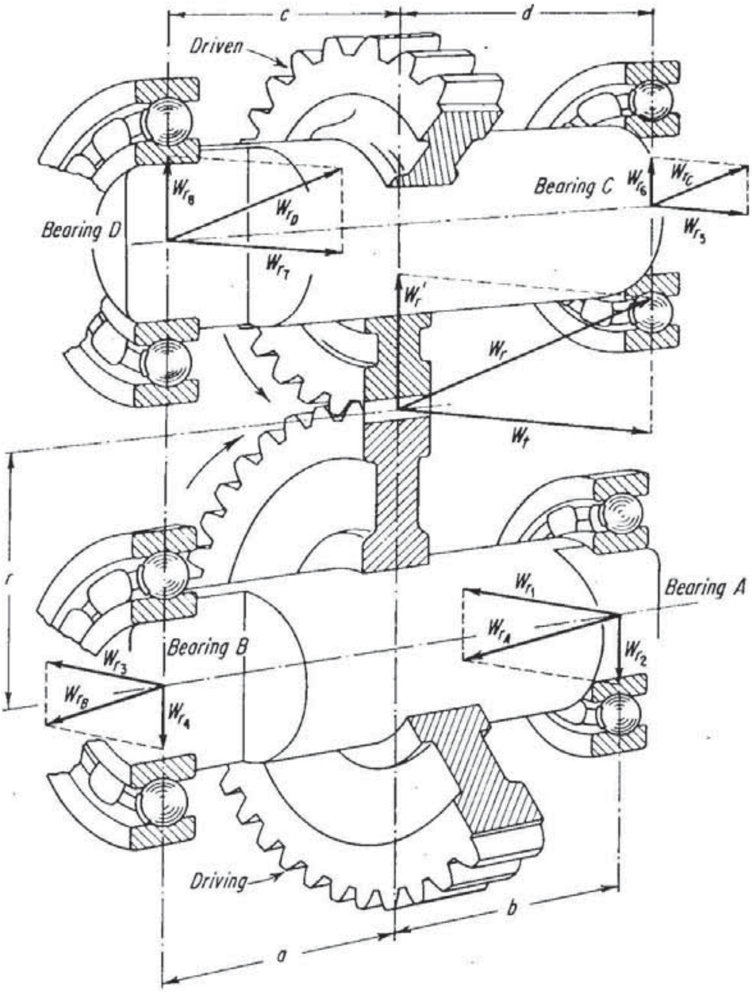
Bearing loads

Position	Forces At These Positions		
	Tangential Force	Separating Force	Total Radial Load
Gear Mesh	W_t	W_r'	W_r
Bearing A	$W_{r1} = \frac{W_t a}{a+b}$	$W_{r2} = \frac{W_r' a}{a+b}$	$W_{rA} = \sqrt{(W_{r1})^2 + (W_{r2})^2}$
Bearing B	$W_{r3} = \frac{W_t b}{a+b}$	$W_{r4} = \frac{W_r' b}{a+b}$	$W_{rB} = \sqrt{(W_{r3})^2 + (W_{r4})^2}$
Bearing C	$W_{r5} = \frac{W_t c}{c+d}$	$W_{r6} = \frac{W_r' c}{c+d}$	$W_{rC} = \sqrt{(W_{r5})^2 + (W_{r6})^2}$
Bearing D	$W_{r7} = \frac{W_t d}{c+d}$	$W_{r8} = \frac{W_r' d}{c+d}$	$W_{rD} = \sqrt{(W_{r7})^2 + (W_{r8})^2}$

For bearing life calculations based on these radial loads see page T12-3.

Note - These equations can only be used for spur gear calculations, because they are not affected by self-generated axial forces.

Bearing loads and gear mesh forces diagram

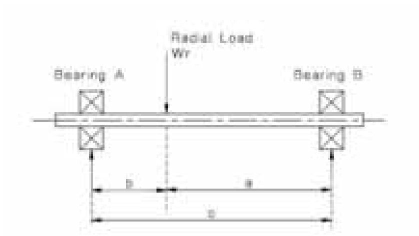


FORCE SHARING

To determine how the forces are shared between a pair of bearings, use the equations below for these two most frequently occurring configurations:

1. Radial Shaft Load Between Two Bearings

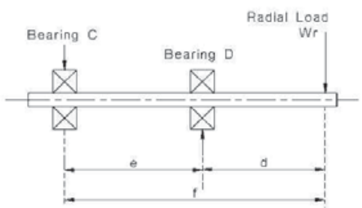
Loads are in constant units



Radial load on bearing $A = \frac{Wr * a}{c}$

Radial load on bearing $B = \frac{Wr * b}{c}$

2. Overhung Radial Load



Radial load on bearing $C = \frac{Wr * d}{e}$

Radial load on bearing $D = \frac{Wr * f}{e}$

The individual bearing loads can then be used to predict the bearing life.

BEARING LIFE

The life of a bearing is defined as the length of time a bearing will operate satisfactorily in the application at its operating speed under applied load. Life predictions depend on a careful definition of failure criteria and consideration of operating environment, mounting practice, lubrication, operating speed and loading. As a guide, the relationship between actual applied load and bearing fatigue life is given below:

$$L_H = \frac{16667}{N}\left(\frac{C}{P}\right)^3$$

L_H = Rated life in hours
N = Speed in rpm
P = Bearing load (e.g. N)
C = Bearing capacity (e.g. N)

INSTALLATION AND HOUSING CONSIDERATIONS

The installation of a bearing will usually be determined by how it fits with its mating components. Interference or transition fits provide the most positive location of the bearing, however, they will require pressing during installation. Clearance fits allow the bearing to be assembled very easily, but could potentially lead to problems depending on the operating conditions. If a press fit is required, it is essential that no appreciable force is transferred through the rolling elements of the bearing during installation.

Special care must be taken when using bearings in aluminium housings, especially when wide temperature variations are expected. It is possible for the contraction of the housing to squash the bearing raceway and remove the radial clearance required for the bearing to operate.

Potential problems with clearance fits:
Fretting - Wearing away of the surface due to rubbing of the components.
Accuracy - Accuracy can be compromised due to unpredictable movement.

Potential problems with interference fits:
Assembly - Can be difficult or impossible without damaging the bearing.
Radial clearance - Can be reduced if the interference is too great.

ISO METRIC SCREW THREADS: LIMITS AND TOLERANCES

Thread	Pitch	Internal/ External Tol. Class	Major Diameter			Pitch Diameter			Minor Diameter		
			max	tol	min	max	tol	min	max	tol	min
M1.6	0.35	6g (screw)	1.581	0.085	1.496	1.354	0.063	1.291	-	-	1.075
		6H (nut)	-	-	1.600	1.458	0.085	1.373	1.321	0.100	1.221
M2	0.40	6g	1.981	0.095	1.886	1.721	0.067	1.654	-	-	1.408
		6H	-	-	2.000	1.830	0.090	1.740	1.679	0.112	1.567
M2.5	0.45	6g	2.480	0.100	2.380	2.188	0.071	2.117	-	-	1.839
		6H	-	-	2.500	2.303	0.095	2.208	2.138	0.125	2.013
M3	0.50	6g	2.980	0.106	2.874	2.655	0.075	2.580	-	-	2.272
		6H	-	-	3.000	2.775	0.100	2.675	2.599	0.140	2.459
M4	0.70	6g	3.978	0.140	3.838	3.523	0.090	3.433	-	-	3.002
		6H	-	-	4.000	3.663	0.118	3.545	3.422	0.180	3.242
M5	0.80	6g	4.976	0.150	4.826	4.456	0.095	4.361	-	-	3.868
		6H	-	-	5.000	4.605	0.125	4.480	4.334	0.200	4.134
M6	1.00	6g	5.974	0.180	5.794	5.324	0.112	5.212	-	-	4.597
		6H	-	-	6.000	5.500	0.150	5.350	5.153	0.236	4.917
M8	1.25	6g	7.972	0.212	7.760	7.160	0.118	7.042	-	-	6.272
		6H	-	-	8.000	7.348	0.160	7.188	6.912	0.265	6.647
M10	1.50	6g	9.968	0.236	9.732	8.994	0.132	8.862	-	-	7.938
		6H	-	-	10.000	9.206	0.180	9.026	8.676	0.300	8.376

Reference: BS3643 Pt 2, 2007.

TORQUE AND TENSION GUIDELINES

The usual method for specifying and measuring fastener installation is tightening torque, as this is relatively easy to measure with a torque wrench. Unfortunately, a torque wrench does not give an accurate indication of bolt tension because it does not take friction into account. The friction is dependent on the bolt, nut and washer materials, surface smoothness, machining accuracy, degree of lubrication (including uncured retaining products) and the number of times a fastener has been installed. The torque values provided for the screws in the table below are, therefore, to be used only as a guide, the friction factors mentioned should be considered for each application.

Aluminium structural components assumed, as this is typical of the applications for which the Reliance standard products range is designed for.

Screw Size	Tightening Torque for 700 MPa Socket Head Cap Screw (Nm)	Tightening Torque for Stainless Steel Set Screws (Nm)
M1.6	0.13	0.05
M2	0.26	0.05
M2.5	0.52	0.18
M3	0.92	0.32
M4	2.10	0.75
M5	4.20	1.50

RELIANCE STANDARD MATERIALS

Reliance Precision screws are manufactured from the materials listed below. Where the product material is specified, we reserve the right to change the actual material to an equivalent specification without notice depending on availability.

Material Grade	Treatment	Grain Structure	Tensile Strength MPa (min)	Corrosion Resistance
A2-70	n/a	Austenitic	700	Excellent
A4-70	n/a	Austenitic	700	Excellent
303	n/a	Austenitic	585	Excellent
416	Hardened to 26/32 HRc	Martensitic	880	Good

General Properties of Austenitic Stainless Steels

- Excellent resistance to oxidation and corrosion.
- Essentially non-magnetic.
- Cannot be hardened by heat-treatment.
- Work hardened very easily.
- Relatively high coefficient of thermal expansion of 18 microns/metre/°C, close to that of aluminium.

General Properties of Martensitic Stainless Steels

- Good resistance to oxidation and corrosion.
- Magnetic.
- Readily heat treatable to high strength condition.
- Potential for tensile strengths > 1200 MPa.
- Coefficient of thermal expansion of approximately 11 microns/metre/°C.


Reliance
Components and Assemblies
Precision Components & Online Store
Precision Assemblies

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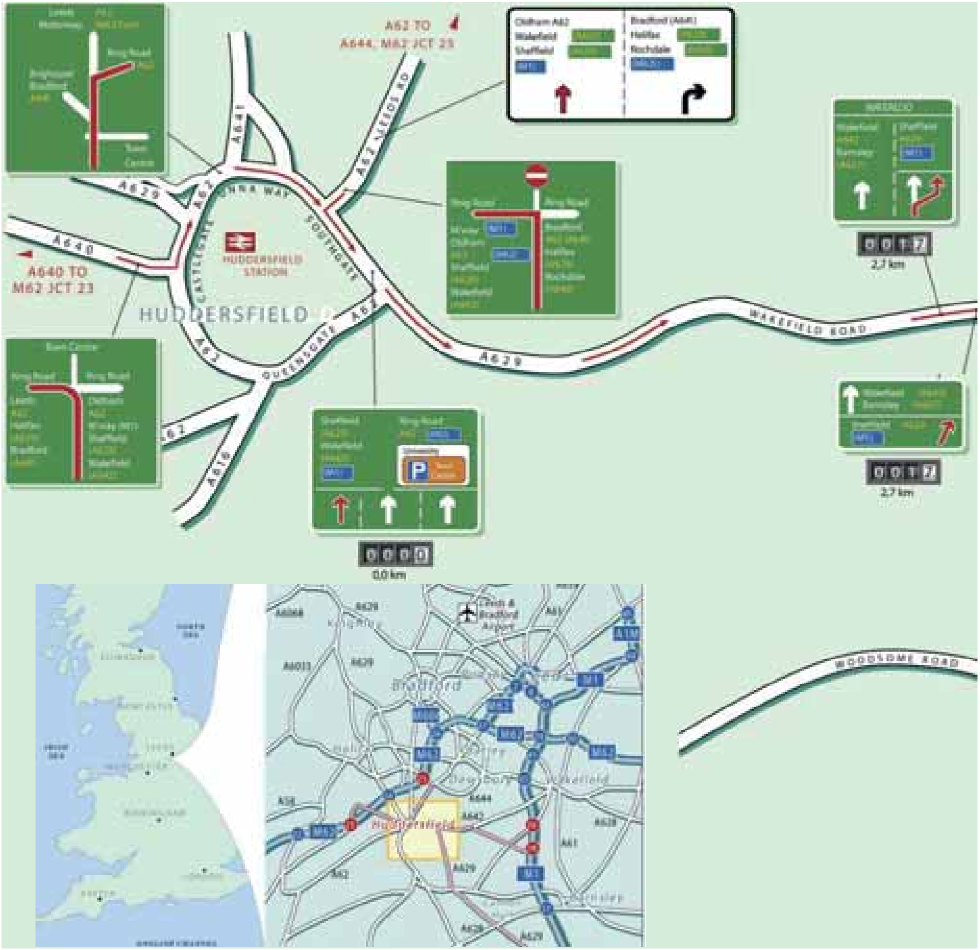

NORTH & SOUTH APPROACHES

From the North: Junction 39 of M1 (Denby Dale turn off), follow the A636 for 3 miles and then turn right on to the A637 for 2 miles. At the next roundabout turn left on to the A642 to Huddersfield. After 1 mile on the A642 turn left through the village of Lepton, to join the A629. Turn left and Reliance is 100 yards on the left.

From the South: Juntion 38 of the M1, follow the A637 for 4.5 miles passing Yorkshire Sculpture Park and straight accross at the second roundabout on to the A642 to Huddersfield.

or

Junction 35A M1, follow the A616 for 3 miles, turn off onto the A629 towards Huddersfield. Follow this road for approximately 15 miles, passing through Thurgoland, Ingbirchworth and Shepley. Reliance is on the right.

Location Map

EAST & WEST APPROACHES

From the East: Junction 25 of M62, follow the A644 for 2 miles and turn right at the next roundabout on to the A62. Follow the A62 for 5 miles until it joins the Huddersfield ring road. Leave the ring road following the A629 for Sheffield (Sainburys supermarket on your left). Follow the A629 for 3.4 miles. Reliance is on the left.

From the West: Junction 23 of the M62 (or J25 and follow as above), follow the A640 for 3 miles to Huddersfield ring road. Turn right on to the ring road. Follow the ring road for approximately 3/4 miles ignoring the signs for Sheffield (A616) and Manchester (A62) until you pass the University. Turn right at the next roundabout onto the A629, sign posted to Sheffield and Wakefield. Follow the A629 for 3.4 miles. Reliance is on the left.

Public Transport

Wakefield Station
25 minutes by taxi
Huddersfield Station
20 minutes by taxi

Manchester Airport
1 hour by road
Leeds/Bradford Airport
1 hour by road

Making a purchase on-line

Information about Reliance's products and services are available on our website www.reliance.co.uk/shop. Many of our catalogue components are available to purchase on-line including:

- Gears
- Couplings
- Bearings and spacers
- Screws and hardware
- Belts and pulleys
- Shafts and accessories

The on-line store is available to customers in the UK, Europe and the US and accepts payments in sterling, euro and dollars, using a secure checkout.

Creating a custom gear on-line

The on-line custom gear builder gives simple, efficient access to a range of over 300,000 precision gears. Not only does it allow a customised gear to be purchased quickly but it also reduces the risk of the wrong gear being selected. At the end of the on-line selection process all the required technical and commercial information is provided appropriate to the gear specification, including a 3D image, drawing, part description and item number, together with the price and delivery lead time.

There are an extensive number of possible combinations offered, with a wide choice of gears and associated variables in specification - including choice of material, gear module, number of teeth, bore diameter and face width. The on-line gear builder uses a filtering process to seamlessly remove redundant options to enable a swift selection of the required gear. Selections can be amended at any point simply by navigating backwards and forwards through the various steps.

3D models available to download

A selection of 3D models is available to download directly from the on-line store, alternatively please contact us and our sales and applications engineering team will be happy to provide them. We are able to provide Step and IGES models.

Conditions of Sale

In addition to these conditions of sale, our standard Conditions of Sale also apply. A copy of these is available on request and from our website www.reliance.co.uk/en/help

Minimum order charge - Orders are subject to a minimum order charge of £250.00.

Carriage and packing - Additional charges are made for carriage and packing.

Payment - Payment terms are 30 days. New customers are requested to complete an application form for a credit account. Customers who do not have a credit account with Reliance are requested to supply cheque with order. In addition, orders may be paid for by Visa and Mastercard.

Telephone orders - An order number must be quoted by the customer. We reserve the right to supply parts against a telephone order. All telephone orders are accepted subject to these conditions of sale and those detailed on the acknowledgement of order. An acknowledgement will normally be sent by Reliance and goods will be supplied in accordance with the order acknowledgement.

Certificates of Conformance - Reliance's quality management system is certified to AS9100 and ISO 9001. A Certificate of Conformance can be supplied at an additional charge of £10.00 per delivery. Alternatively, a Certificate with full material traceability can be supplied at a charge of £20.00 per delivery.

Confirmation - All orders, other than telephone orders with a value of less than £500 and orders placed through our website, are subject to acceptance in writing by Reliance Precision Limited.

Order amendments - Order amendments are subject to our approval and a charge will be made for reasonable compensation for any costs incurred.

Returns - Unused items may, solely at our discretion, be accepted for credit within 90 days of delivery. Any parts so accepted will be subject to a 20% service charge for re-inspection and handling. No credit can be allowed after the above period, or for any used or modified part, or for parts manufactured to a customer's specification.

Additional charges - Reliance reserves the right to charge for all additional expenses and taxes incurred over and above published prices (including without limitation duty, VAT, exchange rate fluctuations etc.)

Alterations - As a result of continuous product development, Reliance reserves the right to alter prices and other details without prior notice and to change dimensions where this does not affect the function of the item.

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