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Motor/Gearhead Selection

Selecting Keyless vs Keyed Connections for Motor to Gear Reducer

Knowing the differences between keyless and keyed-shafted connections is vital in distinguishing the advantages and disadvantages of both types of specifications when choosing a gearbox for your motor. In today's marketplace, the need for speed, precision and small size are not only dictating the standard for shaft locking devices but also challenging motion component manufacturers to evaluate methods of keyless shaft locking for dynamic loads. Motor and drive manufacturers are making products in smaller and smaller packages which are increasingly capable of rapid acceleration and rotary positioning accuracy. These industry changes are causing backlash, stress distribution and balance to all be addressed in shaft locking devises, in many cases rending the shaft key obsolete.

In the examples that follow, both types of connections are compared in terms of torque transmission. A shaft with a diameter of 16mm is used for both cases. The key and keyway dimensions are standard according to DIN 6885. Calculations show the maximum torque that can be transmitted through both keyed and keyless shafts. Also, there are calculations for maximum transmissible torque for the key only. Material for the shaft and key is steel 35S20.

I. Keyless shaft

With the assumption that slipping will not occur, the torque transmitted through the shaft is:

$$T = \frac{\tau . J}{r}, \text{ where } J = \frac{\pi . r^4}{2}$$

T - Max. Transmitted Torque, (N.m)
J - Polar Moment of Inertia, (m^4)
τ - Yield Stress for Steel 35S20, ($380x10^6$ N/m^2)
r - Radius of the shaft, (8mm)

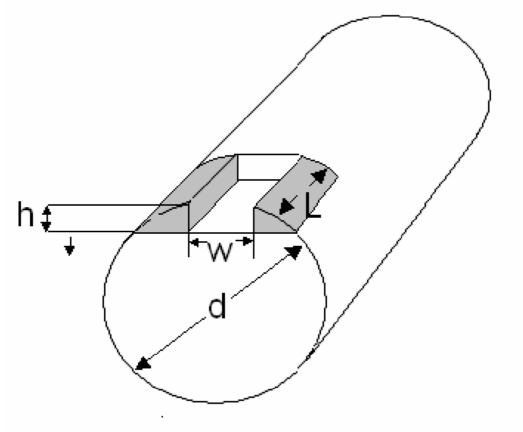
$$J = \frac{\pi . 0.008^4 m}{2} = 6x10^{-9} m^4$$

$$T = \frac{380x10^6 N/m^2 . 6x10^{-9} m^4}{0.008m} = 285 N.m$$

This is the maximum torque that can be transmitted through the keyless shaft before any plastic deformations take a place.

II. Keyed shaft

The stress level imposed on the keyway sides is critical in evaluating keyed applications. With the assumption that the key will not fail before the shaft, torque which the shaft can hold is:



$$T = \frac{\tau . d . l . h}{2}$$

T - Max. Transmitted Torque, (N.m)
τ - Yield Stress for Steel 35S20, (380x10^{6} N/m^{2})
d - Diameter of the shaft, (16mm)
l - Effective length of the keyway, (25mm)
h - depth of the keyway, (3mm)

$$T = \frac{380x10^{6}\, N/m^{2}.0.016m.0.025m.0.003m}{2} = 228N.m$$

The value 228Nm is the maximum torque that can be transmitted through the keyed shaft with diameter of 16mm before any plastic deformations occur.

III. Key

The most commonly used type of key is the flat key. With this type, there are two modes of failure likely: shear failure and crushing failure. In the calculations the total value of the yield stress is used and the factor of safety is not included.

Shear failure of the flat keys

The keyway of the shaft and hub exert equally and opposite forces on the key, resulting in shear deformation. This equates to the forces attempting to shear the key at the radius of the shaft. Shear stress can be noted as follows:

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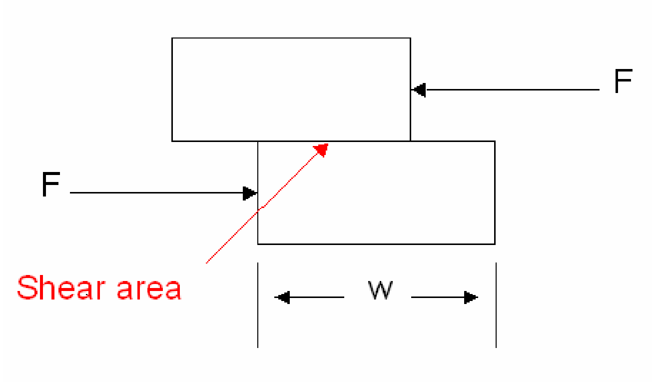


$$\tau = \frac{F}{w.l} \Rightarrow F = \tau.w.l$$

$$T = F.r$$

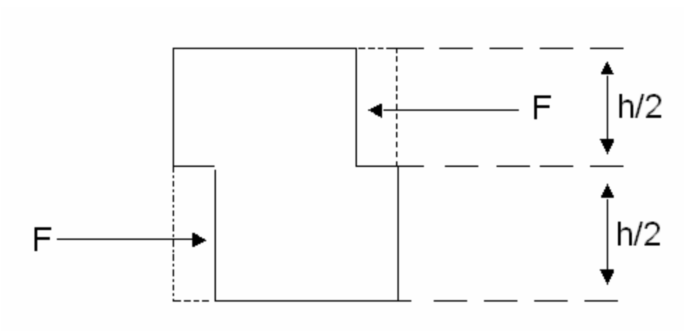
T - Max. Transmitted Torque, (N.m)
F - shear force acting on the key, (N)
τ - Yield Stress for Steel 35S20, ($380x10^6$ N/m^2)
w - width of the key, (5mm)
l - Effective length of the keyway, (25mm)
r - Radius of the shaft, (8mm)

$F = 380x10^6 N/m^2.0.005m.0.025$ m = 47500N

T = 47500N.0.008m = 380N.m

Crashing failure of the shaft keys

Contact of the keyways in the shaft and the hub can result in permanent compressive deformation. This crushing stress is a result of compressive stress that equals the applied force divided by the area of contact.



$$\tau = \frac{F}{l.\frac{h}{2}} \Rightarrow F = \frac{\tau.l.h}{2}$$
$$T = F.r$$

T - Max. Transmitted Torque, (N.m)

F - Compressive force, (N)

τ - Yield Stress for Steel 35S20, (380×10^6 N/m^2)

r - Radius of the shaft, (8mm)

l - Effective length of the keyway, (25mm)

h - Height of the key

$$F = \frac{380x10^6.0.025m.0.005m}{2} = 23750N$$

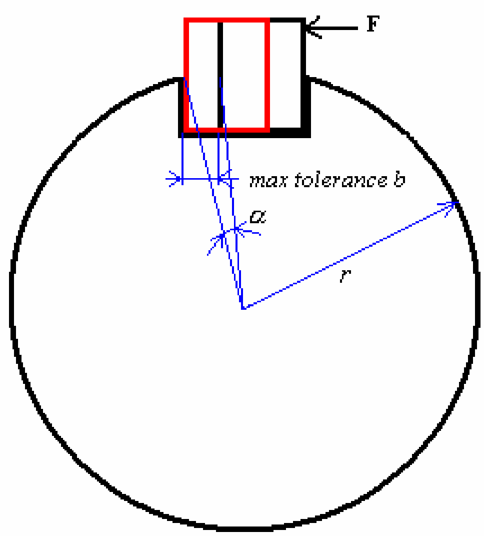
$$T = 23750N.0.008m = 190N.m$$

The key's torque handling capacity is 190N.m. This equates to the torque that the key can hold before plastic deformation.

Another factor to consider is having the optimal fit when mating the key to the keyway. According to ISO JS9 for parallel key with normal fit, the tolerance for the keyway with a width of 5mm is +0.015 /–0.015mm. The key's tolerance per DIN 6885 is +0.05/0. The clearance between the shaft's keyway and the key is up to 0.015mm, allowing angular movement of the shaft up to 0.1074°. This relates to a 6.445 arcminutes potential backlash. In addition, the keyway tolerances allow for keyways that may be 0.015mm off the shaft centerline.

Calculations:

The assumption in this case is that the key slides instead of twisting.



$$\alpha = \frac{b.180}{r.\pi}$$

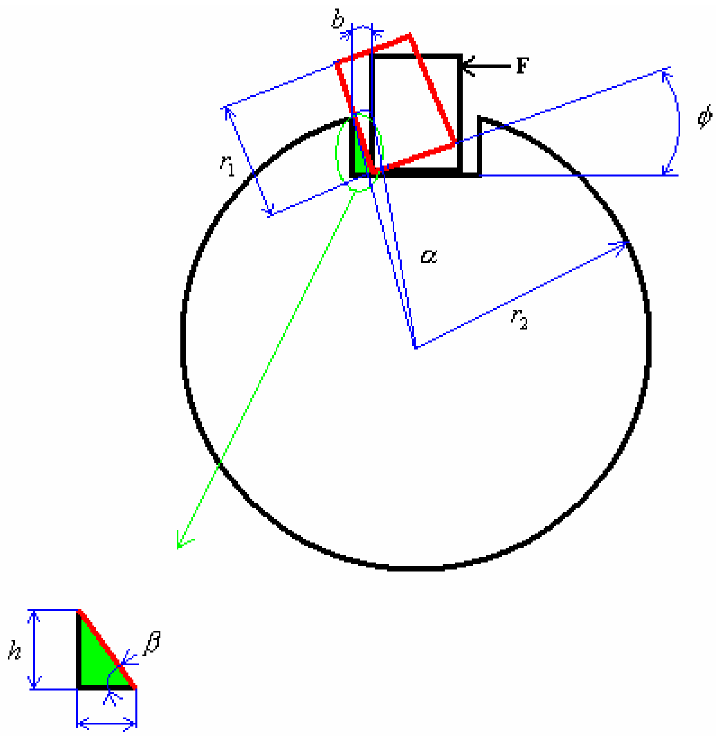
α **-** Backlash angle, (DEG/arcmin)

b **-** Arc length, (the tolerance value is been used since the angle is small, 0.015 mm.)

r **-** radius of the shaft, (8mm)

$$\alpha = \frac{0.015.180}{8.\pi} = 0.1074^{\circ}$$
$$\alpha = 0.1074.60 = 6.445 arc \min.$$

If it is assumed that the key is in the middle and twists, the backlash is as follows:



$$b = \frac{r_1.\pi.\phi}{180}$$, where $\phi = 90^{\circ} - \beta$, where $\beta = \tan^{-1}\left(\frac{h}{\frac{tolerance}{2}}\right)$

$$\alpha = \frac{b.180}{r_2.\pi}$$

α - Backlash angle, (DEG/arcmin)
ϕ - Angle of key twist (DEG)
β - Angle between keyway shaft bottom and key side
r_1 - Radius of the twist, (key height, 5mm)
r_2 - Radius of the shaft, (8mm)
h - Depth of the keyway, (3mm)

b - Arc length, (mm)

$$\beta = \tan^{-1}\left(\frac{3mm}{\frac{0.015mm}{2}}\right) = 89.85676^\circ \Rightarrow \phi = 90^\circ - 89.85676^\circ = 0.143^\circ$$

$$b = \frac{3mm.\pi.0.143^\circ}{180} = 0.00074875mm$$

$$\alpha = \frac{0.00074875.180}{8.\pi} = 0.00536^\circ = 0.32 arc\min.$$

The most vital aspect when addressing performance issues is the concern with backlash. Fully eliminating backlash is rarely possible; however, backlash can be reduced by precise fitting of the gearbox to motor and complicated machining. The increase in keyway wear is due to a variety of factors. Increased quantity of frequent machine starts, stops and load reversals are some common causes of key wear. With increasing acceleration and deceleration rates, wear is increased in terms of both the frequency and force of the impact between the key and keyway.

As time passes, backlash will also increase at an accelerated rate. Resulting from impact, material is compressed and removed from the keyway, widening the keyway and making the velocity at which the key impacts the keyway higher with each load change. Highly dynamic loading can result in keyways wearing down to the point of problematic backlash or even failure over a very short time span.

More over, having a keyway may pose problematic at the time of disassembly. Depending on the environment and duty cycle, oxidation and corrosion may occur and, as a result, can "weld" the key assembly components.

Eliminating the problems stated above requires eliminating the keyway method and using a frictional type of connection between the components of the assembly, one sample of which can be using a shrink disk. Shrink disk connection will result in a zero clearance and consequently no additional backlash on the system. Also, it leads to higher amount of transmissible torque according to the calculations above. Finally, eliminating the keyway doesn't require consideration of keyed shaft notch factors. That allows a smaller shaft diameter and smaller bearing sizes which leads to decreasing overall the cost of the system.