Considerations about the friction inside on a transversal coupling

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Abstract. The paper presents, starting from some kinematic aspects as rotations and translations between the Oldham coupling parts, some kinematic equations useful in determining of intermediary element position depending by the transversal misalignments. Then, using the inside translational movements between parts, the friction coefficient between adequate materials to be used are studied. The friction which appears between the coupling parts has an important influence on their dynamic behaviour, wear and lifetime. The most significant friction and also wear is given by the translational movements between parts. Due to this, the study of the friction between some materials to be used in coupling manufacturing is required, but also difficult because of some particularities, as reduced parts translational movements for reduced transversal misalignments between shafts, or the alternative translational movement in translational joints. In the paper final part the results and conclusions are presented.

1. Introduction

Transversal couplings (figure 1) are used in movement and torque mechanical transmission, between two shafts with parallel axis [1, 2]. The shafts are connected by a kinematical linkage with the possibility to have translations in the transversal plane, named transversal movements, or eccentricities. According to the mechanisms theory, if the shafts are connected with joints to the basis, it will result the coupling associated mechanism, which is a plane mechanism [1-3].

The simplest known coupling that may realise a translation between two semicouplings, each placed on rotational shaft, are the Oldham coupling. As structure, the mentioned coupling consists in three elements: the input part, the intermediary element and the output part as is presented in figure 1. Between them, there are translational joints along some straight shaped slots, disposed on the intermediary element, particularly at 90 degrees. Due to the structural simplicity, the main inconvenient of this coupling type is the friction along the translational profiles between the elements. So, during the time, in the transversal couplings field, were developed many other structural solutions; at most of them, the linkage with translational joints is replaced with parallelogram or anti parallelogram linkages with rotational joints disposed between, so the structural complexity is increased.

In present days, many researches are focused to reduce the transversal couplings structural complexity and also to reduce the frictional loses between elements [4-6]. So, for the Oldham coupling, the steel intermediary element is usually replaced by some plastic element in order to reduce friction, to protect the transmission when is overloaded (as safety coupling), or to realise the electrical
isolation between the involved shafts. Another solution to reduce the friction inside coupling is to lubricate the inside translational joints.

![Figure 1. The Oldham coupling main parts [1, 2].](image)

In figure 2, there are presented two examples of modern times Oldham type couplings, a coupling with the intermediary element made by plastic material [6] and a full plastic coupling used in a copying machine, on a drive shaft connector [7]. In most of the cases, these couplings have reduced dimensions, to be used for low torques and rotations.

![Figure 2. Oldham coupling examples [6, 7].](image)

2. The coupling kinematical model

For the studied coupling shown in figure 1, the geometrical and kinematical model is presented in figure 3. The parts are defined in the reference system $O_{x'y'}$. For each reference system the origin is considered in the gravity centre, $G_i$, the reference axis orientation is convenient chosen [2].

For geometrical restrictions definition, is necessary to know the point’s coordinates, relative to the local coordinate system and also relative to the fixed coordinates system. The general relation of these coordinates is [2]:

$$
\begin{bmatrix}
x_{ui}
\end{bmatrix} = \begin{bmatrix}
x_i \\
y_i
\end{bmatrix} + \begin{bmatrix}
\cos \varphi & -\sin \varphi \\
\sin \varphi & \cos \varphi
\end{bmatrix} \begin{bmatrix}
r_x \\
r_y
\end{bmatrix},
$$

where $i = 1, 2, 3$.

Generally, for the rotation between two bodies, $i$ and $j$, the condition is $P_i = P_j$. For the considered rotation between the input semicoupling and fixed reference element and, respectively, the output semicoupling and fixed reference element; from relation (1) result

$$x_{o1} = 0, y_{o1} = 0,$$

respectively

$$x_{o3} = e_x, y_{o3} = e_y.$$
For the rotation between two bodies $i$ and $j$, the general condition is $\Delta_1 = \Delta_j$. For the translation between the input semicoupling and intermediary element, respectively, the output semicoupling and intermediary element, result:

\[ x_i \sin \phi_i - y_i \cos \phi_i + r_i \sin(\phi_i - \phi_j) = 0 \] (4)

and

\[ x_i \sin \phi_i - y_i \cos \phi_i - r_i \sin(\phi_i - \phi_j) = 0, \] (5)

respectively

\[ x_i \sin \phi_i - x_i \sin \phi_j + y_i \cos \phi_i - y_i \cos \phi_j + r_i \sin\left(\phi_i + \alpha\right) - \phi_j = 0 \] (6)

and

\[ x_i \sin \phi_i - x_i \sin \phi_j + y_i \cos \phi_i - y_i \cos \phi_j - r_i \sin\left(\phi_i + \alpha\right) - \phi_j = 0. \] (7)

The kinematic restriction equation is

\[ \rho_i - f(t) = 0. \] (8)

Based on the previously relations, from (4) to (8), result:

\[ \sin(\phi_i - \phi_j) = 0 \] (9)

and
\[
\sin[(\varphi_2 + \alpha) - \varphi_1] = 0.
\]  

(10)

From relation (9) result \( \varphi_2 = \varphi_1 + 2K\pi \), where \( K \in Z \), for \( \forall \varphi_1, \varphi_2 \in R \). For \( K = 0 \), result \( \varphi_2 = \varphi_1 \).

Also, from relation (10) result \( \varphi_2 + \alpha = \varphi_1 + 2K\pi \), where \( K \in Z \), for \( \forall (\varphi_1 + \alpha), \varphi_2 \in R \). As previously, \( K = 0 \) and result \( \varphi_2 = \varphi_1 + \alpha \) where \( \alpha = \text{const} \).

For the real Oldham coupling, the translation directions on the intermediary element are orthogonal, so \( \alpha = \frac{\pi}{2} \); result

\[
\varphi_2 = \varphi_1 + \frac{\pi}{2}.
\]  

(11)

Also, from relations (6) and (7), result:

\[
x_{in} = x_0 \left( \sin \varphi_1 \cos \varphi_1 \frac{1}{tg\alpha} + \cos^2 \varphi_1 \right) - y_{in} \left( \cos^2 \varphi_1 \frac{1}{tg\alpha} - \sin \varphi_1 \cos \varphi_1 \right);
\]

(12)

\[
y_{in} = x_0 \left( \cos^2 \varphi_1 \frac{1}{tg\alpha} + \cos \varphi_1 \sin \varphi_1 \right) - y_{in} \left( \cos \varphi_1 \sin \varphi_1 \frac{1}{tg\alpha} - \sin^2 \varphi_1 \right).
\]

(13)

Based on relations (11), (12) and (13), for the studied Oldham coupling \( (\alpha = 90^\circ) \), there is presented in figure 4 the graphic representation of the intermediary element coordinates values \( x_{o2} \) and \( y_{o2} \) variation, for a complete input shaft rotation, \( \varphi_1 \in [0, 2\pi] \).

![Figure 4. The intermediary element coordinates values variation.](image)

From figure 4 it can be observed the curves periodicity, each repeated at \( \pi \). So, for a complete shaft rotation, the coupling intermediary element is rotating itself two times [2].

3. The friction in the translational joints

According to the literature [8], for Oldham coupling, the friction force between input, respectively, the output semicoupling and intermediary element, is a rotating vector which follows the direction of the translational joint and is given by general relation

\[
F_{\theta j} = \mu \frac{T}{R_{o2}},
\]

(14)

where \( j = i+1 \) and \( i = 1, 2 \).
In previous relation, $\mu_{ij}$ is the friction coefficient between the materials of the coupling elements; $T$ represents the torque applied on input shaft, and $r_{ij}$ the radius where the normal force, $N_{ij}$, loads the considered elements on the translation direction. In figure 5 are detailed these loads, for the particular situation when eccentricity between the input semicoupling 1 and intermediary element 2 is $e_x = e$ and between the intermediary element 2 and output semicoupling 3, $e_y = 0$.

![Figure 5. The loads inside the coupling.](image)

Taking account by the intermediary element rotation, each translational joint friction changes the direction twice on a coupling rotation. Considering the coupling constructive characteristics and the orthogonal displacement of the translation joints on intermediary element, the resultant friction force is given by

$$F_x = \sqrt{F_{11}^2 + F_{23}^2}.$$  (15)

Following the previously considerations, the resultant friction changes the direction for four times during a single coupling rotation [2], with the stroke limits given by the coupling transversal misalignment.

To estimate the friction inside the coupling translational joints, all the coupling parts are considered made from medium-carbon steel, greased with lubricant. For the friction tests, two test pieces made from the same material are used. To achieve the tests, is used a test rig which has as components a tribometer and a computer used for the acquisition of the measured data [9, 10], as is presented in figure 6.

On the top part, the tribometer has a holder which is mounted on a force sensor; this sensor allows measurements for forces about two directions, in a range of $F = 0.1...1000$ N, and the resolution $R_r = 50$ mN. The vertical motion of the holder is up to a distance $d = 150$ mm, with the speed of $v_v = 0.001...10$ m/s. Its lateral motion is up to $l = 75$ mm, with the speed of $v_l = 0.01...10$ m/s and the resolution $R_r = 2 \mu$m. In the bottom part, the tribometer has the unit which allows the reciprocating motion. This unit has the one direction force sensor which allows measurements up to $F = 1000$ N with a resolution of $R_r = 1\mu$N. The frequency of the motions is between $v = 0.1...60$ Hz and the stroke is $s = 0.05...25$ mm [10]. Inside the reciprocating motion unit is mounted an oil bath where is placed the adequate test steel piece. On this piece is acting the top holder which has fitted on it another steel test part. Inside the bath oil, the lubricant is Würth HHS Lube type, which operates between -25 to 150°C, and is compatible with metal surfaces [11].

The tests were performed with the following input parameters: the motion frequency $v \in \{1, 10, 20\}$ Hz; the normal force $F \in \{5, 10, 20\}$ N; the temperature of the oil bath $t = 20^\circ$C. The stroke of the oscillatory motion was established depending by the mentioned coupling eccentricity, $e = 4$ mm.
The variation of the friction coefficient during the oscillatory motion is presented in figure 7. It can be observed that, during the oscillatory motions, the contact between the two steel parts is characterised by two types of friction coefficients: the static friction coefficient (at the ends of the stroke, when the motion changes the direction) and the dynamic friction coefficient (during the translational motion).

For the friction force calculus in a coupling translational joint, the torque value is $T=10$ kNmm, eccentricity value is $e=4$ mm, the coupling exterior diameter is $D=120$ mm; also, the radius $r_x$ values result from figure 5. The obtained friction coefficient values and the relation (14) led to the friction force variation, as is presented in figure 8.
Considering the orthogonal displacement of the translation joints on intermediary element and the relation (15), the resultant friction force variation is presented in figure 9.

Figure 9. The resultant friction force variation.

### 4. Conclusions
During the oscillatory motions, the contact between the tested two steel parts is characterised by two types of friction coefficients: the static friction coefficient (at the ends of the stroke, when the motion changes the direction) and the dynamic friction coefficient (during the translational motion).

The static friction coefficient inside the translational joints appears in coupling real functioning conditions when the movement between parts are changing their direction; each translational joint friction changes the direction twice on a coupling rotation. The dynamic friction coefficient in the translational joints appears in real functioning conditions when the movement is maintaining the direction; these strokes depend by the coupling eccentricity. An increased eccentricity led to increased strokes in translational joints.

The friction forces that appear are increasing with the friction coefficient increasing; the maximum values are recorded for the ends of the stroke, when the motion changes the direction. Also, the friction force depends by the radius where the normal force is applied. With increasing the radius (which depends by the coupling eccentricity), decreasing the friction force. Despite the radius influence on the friction force, the diagram pattern from figure 7 is maintained also in figure 8.

Considering the translational joints orthogonal displacement on the intermediary element, the figure 9 presents a modified pattern; also, the resultant friction forces values are increased, having a major influence on the transversal coupling dynamic behaviour.

The lubricating of the translational joints has a positive influence to reduce the friction coefficients and also the surfaces wear.

To evaluate the friction, testing some materials patterns in standard condition instead of the entire coupling testing, could lead to the research reduced costs.

### 5. References
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