**Investigations into tensioning force as a factor for energy loss in chain drives**

Among the group which makes up the flexible drives, one major advantage of chain drives in comparison with the technically equivalent synchronous belt drives is the greater degree of efficiency. The degree of efficiency $\eta$ of the chain drive (0.97 to 0.98) is on average higher than with belt drives (0.93 to 0.97). This, at least, is the doctrine.

If it happens, however, for the most widely differing reasons which arise in actual practice (“sum errors, AT10/2000), that tensioning devices are required for the chain, a different situation comes about. Even chain drives which in the first instance are correctly dimensioned and fitted will already undergo an elongation in run-in within the first few hours of operation, and thereafter will incur wear as a function of time and loading. In any event, the course of wear will be several times slower if the chain drive is tensioned and damped against slack side sag, vibrations, and load shocks. Chains of the highest quality are tensioned in order to achieve complex function advantages with regard to the determination of purpose, for the transfer of moment force without energy losses, and with high reliability, long service life, and minimum environmental burden, in particular running noise.

However, as the tensioning force taking effect on the chain from the outside increases, so the degree of efficiency of the chain drive deteriorates. The proportion of the energy which arrives at the driven shaft becomes smaller and smaller. Particularly if a large number of chain drives are running in one machine, the focus is increasingly placed on the energy loss incurred by the tension principle, as well as the wear and lubrication state of the drives.

The joints of roller chains are multi-stage radial slide bearings. When the arc of contact of the chain wheel wrap runs through, or chain tensioners or offset deflection elements, limited fluctuation movements occur on only a few sectors between bolts, bushings, and rollers, in a manner similar to a multi-stage radial roller bearing. By contrast with conventional roller bearings with circumferential friction surfaces, in the chain links contact surfaces occur between bolts and bushings which are repeated and are limited to only a few degrees by the pivot angle at the chain wheel circumference. It is true that torsional rotation capability does pertain between the bushing and the roller, but it likewise exists on the contact surfaces limited by the pivot angle. The chain lubricant is constantly being pressed out of these contact surfaces, and in most cases is only partially replaced by the pivot movement. With chains that are drip-lubricated or lubricated by hand, a shortage of lubricant occurs if the intervals between lubrication are too great. It is the contact surfaces of the chain links which incur the main influencing factor for redundant work, wear, and running noise.

Using a test device which in practice is relatively easy to produce, investigations have been made into what influence the tensioning force taking effect on the chain drive has on energy losses on the slack side. Taking as a basis a type of chain drive which is often encountered in mechanical engineering practice, regarded hereafter as the “reference” chain drive, the intention is to show what direction of development will allow for advantages to be anticipated in respect of the degree of efficiency of real chain drives.
The reference chain drive is characterised by the following parameters: Chain 08B-1, number of teeth \(Z_1 = 19, Z_2 = 19\); distance between axes \(A = 501\) mm; ambient temperature \(21^\circ\)C; first lubrication carried out by the manufacturers. The chain was fitted with the necessary installation elongation. Arranged above the point of intersection of the inner tangents of the chain drive is a pneumatically adjustable tensioning unit, taking effect against the slack side. The driven shaft is rigidly coupled to a flywheel. The device and its variants with the chain tensioners are shown in Figures 1 to 3.

![Figure 1: Test device for the measurement of force and power losses with ROLL-RING](image1)

![Figure 2: Test device for the measurement of force and power losses with polyethylene slide chain tensioners](image2)
In one series of measurements, first the force values for the phase of the run-up of the chain drive were determined. It is known that, before the moment transfer through the chain, first the joint play and, if appropriate, any slack side sag are eliminated. In this situation, the adhesion friction in both sides is to be overcome.

The adhesion friction forces of the chain drive at different chain tensioning forces represent a relational measurement for the run-up resistance of the chain, up to the point at which the adhesion friction is overcome, without operating load.

To do this, a fine steel cable was hooked into the chain, running over a force measurement unit to a tensioning screw with a fine thread (0.75 mm pitch). The chain wheels were adjusted to the polygon minimum. By slow turning of the tensioning screw, whereby dynamic influencing values remain negligibly small, a tensioning force onto the chain was determined, up until the chain had overcome the adhesion friction.

The maximum value automatically displayed for overcoming the adhesion friction was then read off on the force measuring unit.

The reference chain drive used for the experiment required a tensile force of 4.75 N to overcome the adhesion friction.

In the continuation of the experiments, a conventional commercial ROLL-RING tensioning and damping element, type 108 030 01, was then inserted into the chain, in accordance with the installation instructions. With the arrangement of the tensioning and damping element, at the intersection point of the inner tangents of the chain wheels, a chain tensioning force took effect of 14.2 N. The force which was measured at this tensioning force to overcome the adhesion friction was 6.94 N, a value increased in comparison with the “reference” chain drive by 2.19 N, or a factor of 1.46.

Taking as a starting point the known fact that a tensioned chain is always more difficult to move than a non-tensioned one, in the further experiments the chain without the ROLL-RING tensioning and damping element was now increasingly loaded by pneumatic means with a chain tensioning force rising in increments of 10 N.
In concordance with the usual chain tensioner variants, a slide element made of polyethylene or a chain tensioning wheel with the teeth number $Z = 15$, mounted on a grooved ball bearing, were secured to the piston rod head of the pneumatic cylinder.

![Figure 4: Adhesion friction force of the chain as a function of the tensioning force conveyed by the chain tensioner.](image)

The summary diagram from these experiments shows a linear dependency of the adhesion force of the chain to the tensioning force of the reference chain drive, with the upper line in the diagram standing for the tensioning force conveyed by the slide element. The difference between the two lines is indeed identifiable, but may be less determinant with regard to practical application than the permissible chain speed in sustained operation.

A generalisation of the function appears permissible inasmuch as the energy losses increase with chain drives with rising tensioning force. The reference chain drive is, in our experience, an average chain drive in the mechanical engineering sector. Particularly worth observing would be the results with reversing chain drives and chain drives in interval operation, which, with every change of direction or run-up, run through the phase of full adhesion friction.

In addition, the tensioning force ranges of conventional chain tensioner types were also entered in the diagram in accordance with their catalogue data.

**Figure 4**: Adhesion friction force of the chain as a function of the tensioning force conveyed by the chain tensioner.

According to this, there is a basic evaluation with regard to the energy losses:

The load-free chain drive without chain tensioner does not run up in the ideal manner, but is optimum with regard to energy. A chain drive tensioned with a ROLL-RING chain tensioner comes close to the optimum in terms of energy (factor 1.46). Spring-loaded telescopic chain tensioners (factor 1.88 to 5.18) and vibrating chain tensioners with elastomer blocks (factor 4.72 to 6.92) run up, in accordance with the higher tensioning force, with tensioning force-related energy losses which are in part several times higher.
In order to investigate the dynamic effects of the chain tensioning force, a further series of experiments was carried out with the device shown in Figures 1 to 3. To do this, the chain drive was accelerated by a three-phase motor in each case to a chain speed of 2.65 m/s.

After the motor had been switched off, the chain run-on [m] was determined as a relationship value for the braking capacity in the chain operation, with tensioning forces increased in increments. The inertia of the system incurred by the various different tensioning force devices is regarded as low and negligible.

Because during the experiments, with the exception of the tensioning force determination, the same device was involved and the braking always took effect from the same chain speed to standstill, the masses and their kinetic energy before and after braking are likewise equalised. The smaller the chain run-on, the greater the energy losses which result, taking effect due to the tensioning force.

In the further experiments the chain was once again in succession increasing loaded by pneumatic means, by way of a slide element made of polyethylene or a chain tensioning wheel with the number of teeth \( Z = 15 \), mounted on a grooved ball bearing, with rising chain tensioning force and in increments of 10 N.

In order to produce a diagram, the chain run-on of the reference chain drive attained with the non-tensioned chain was set at 28.92 m as a starting value, and the difference between this starting value and the shortened values for the chain run-on, measured in each case with the tensioning forces increased in increments, was determined as a relation value.

The diagram shows a linear relationship between the shortening of the run-on and the tensioning force of the reference chain drive. The lower diagram line stands for the imposition of the tensioning force by the chain tensioning wheel, and the upper diagram line stands for the imposition of the tensioning force by the slide element.

It seems that a generalisation of the function is likewise permissible, in that the energy losses from chain drives in sustained operation rise as the tensioning force increases. For systems which are operated for several years and in sustained operation or with high shift exploitation levels, this is undoubtedly an energy factor worth noting.
And in respect of wear, too, unnecessarily high chain tensioning forces are also to be taken into account as a disadvantageous costs factor. The tensioning force ranges of conventional commercial types of chain tensioner were also entered in the diagram, in accordance with their catalogue details.

According to this, there is a further basic assessment with regard to the energy losses due to chain tensioning in sustained operation:
The load-free chain drive does not run in an ideal manner, but a chain drive without chain tensioner is optimum in terms of energy. Close to the optimum in terms of energy is a chain drive with ROLL-RING chain tensioners. Spring-loaded telescopic chain tensioners and vibration chain tensioners with elastomer blocks are, according to the higher tensioning force data provided by the manufacturers, associated with tensioning force-related energy losses which are in part several times higher in comparison with a chain drive without a tensioner.