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BUSH CHAINS DESIGN PROCESS

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Abstract: The main application area of the bush chains is the distribution transmission from the combustion engines of the cars. An important issue of the chain drive transmissions is represented by the chains pitch (in order to reduce the vertical oscillations of the chain) and the axis distance (in order to reduce the overall dimensions of the transmission). The paper presents the design process for a bush chain, in order to obtain small pitches and small values for the axis distance.

1. Introduction

Chain drives are used in high loaded power transmissions and one of the important advantages is that they can be use in hard working conditions (dust, humidity, high temperatures). One of the application areas of the chain drives is the cars combustion engines transmissions (figure 1) [2].

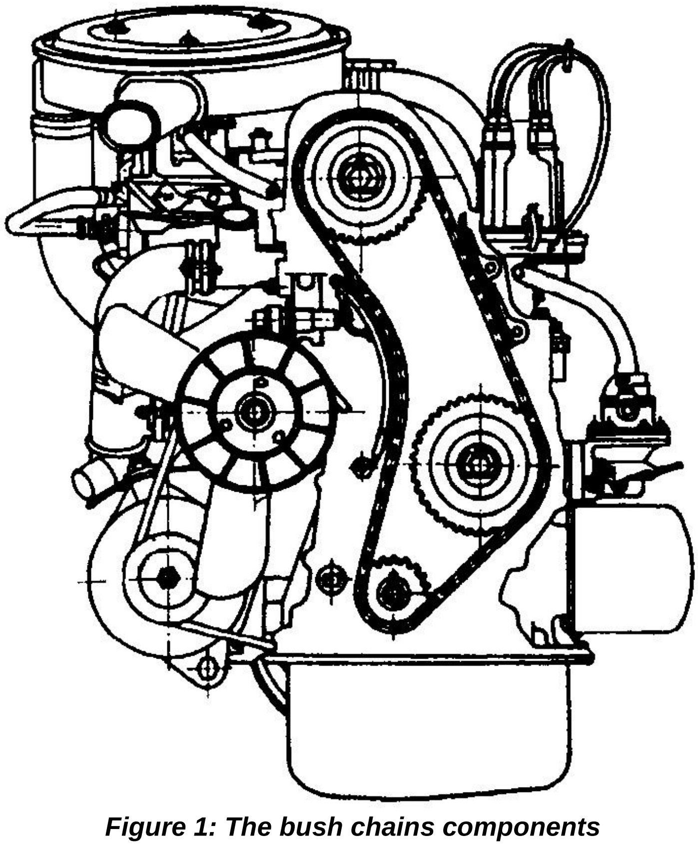

Figure 1: The bush chains components

The main disadvantage of the chain drives is that the motion at the driven wheel is non uniform (due to the polygonal shape of the contact line between the chain and the wheel). Due to that, supplementary dynamic loads, vibrations and noises are produced in the transmissions [1, 3]. According to these, there are two directions regarding the chain drive transmissions design: one has the purpose to reduce the chains pitch (in order to reduce the supplementary dynamic loads) and the second one has the purpose to reduce the axis distance (in order to obtain small overall dimensions).

The paper presents specific aspects of the bush chains design process by considering a given case for a car's combustion engine transmission.

2. The design process

The design process is presented for a bush chain drive used in a Seat Leon 265 HP / 195 kW combustion engines transmission. The chain drive ratio is i=2, the rotation of the driver wheel is n_1=1500 rpm and the chain drive should transmit the mechanical power due to inertias and the friction from the system which represents almost 7% of the engine's power; according to that, the power which should be transmitted is P=13.65 kW. Depending on the chain drive ratio, the number of teeth for the driver wheel is chosen z_1=27.

The torque at the driver wheel is

$$M_{t1} = 9.55 \cdot 10^6 \frac{P}{n_1} \tag{1}$$

and it is obtained M_{t1}=86905 Nmm.

The teeth number for the drive wheel is z_2=54

$$z_2 = i z_1. \tag{2}$$

The maximum pitch is obtained by using [1, 2, 3, 4]

$$p_{max} = \frac{5800}{\sqrt[3]{n_1^2 z_1}} \tag{3}$$

and is p_{max}=14.75 m.

Figure 2 shows the variation of the maximum pitch with the teeth number z_1. Depending on the chain drive ratio i, there are given recommendations in order to choose z_1. To obtain small dynamic loads, a small value for the maximum pitch is needed, which correspond to a high number of teeth number z_1. The values of z_1 are presented in STAS.

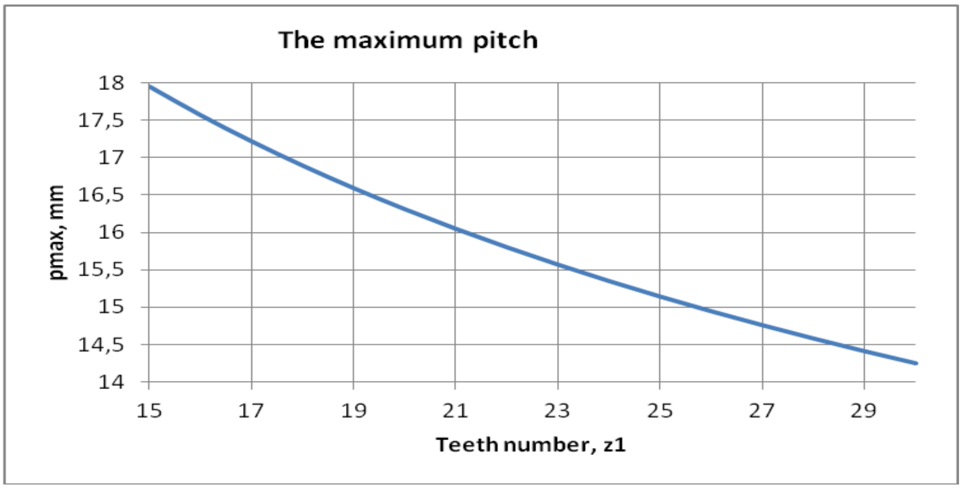

Figure 2: The maximum pitch

According to the figure 2, the values chosen from STAS 5174 for the pitch are: p=8 mm, p=9.525 mm, p=12.7 mm and p=15.875 mm. The corresponding bolt-bush crushing area has the values: A_s=11.01 mm^2, A_s=27.97 mm^2, A_s=50.28 mm^2 and A_s=67.45 mm^2.

The allowed loading force on the chain is established with [1, 2, 3, 4]

$$F_{ua} = A_s \frac{p_a}{K_e}, \tag{4}$$

where K_e=1.625 is the global correction coefficient depending on the functioning conditions and p_a=19 MPa is the allowable pressure of the lubricant.

The medium linear speed is

$$v_m = \frac{z_1 p n_1}{60 \cdot 1000}. \tag{5}$$

The power which can be transmitted by the chain drive is

$$P_{ua} = \frac{F_{ua} v_m}{1000} \tag{6}$$

and the variation of it is presented in the figure 3.

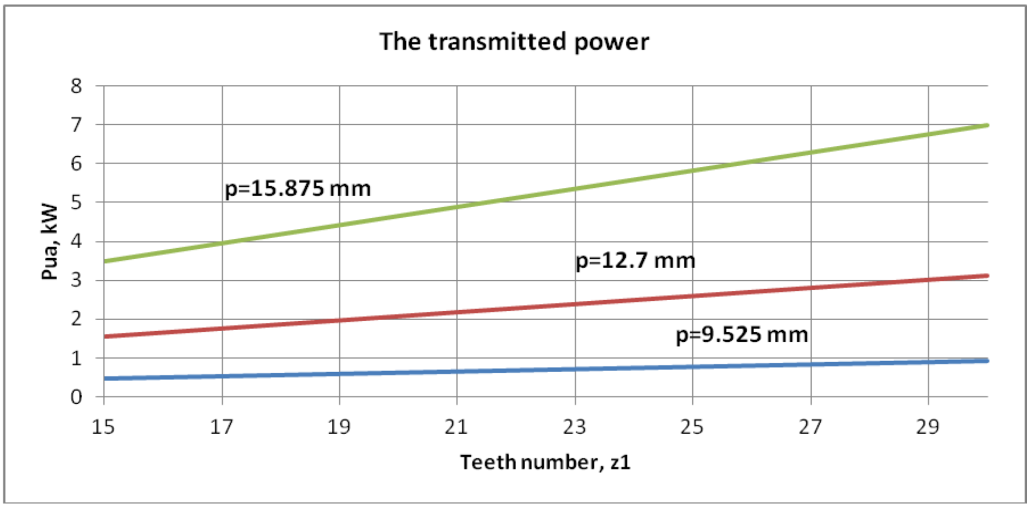

Figure 3: The transmitted power

The number of the chains is obtained with

$$z_r = \frac{P}{P_{ua}} \tag{7}$$

and the variation of it is presented in the figure 4.

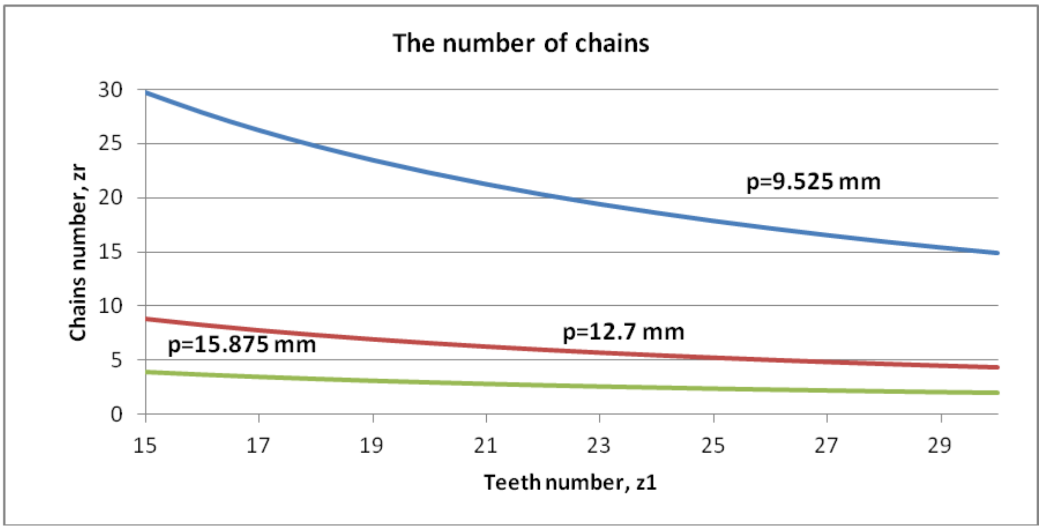

Figure 4: The number of chains

Due to the constructive reasons, there are allowed maximum 3 chains; according to the figure 4 only a pitch of p=15.875 mm assures that. If the teeth number for the driver wheel is z_1=30, there are chosen 2 chains; if z_1=20 … 29, there are chosen 3 chains. This conclusion is interesting according to the literatures data, where for a ratio of i=2, a teeth number z_1=27 is indicated.

The preliminary axis distance is obtained with

$$A_{prel} = (30 \ldots 50)p. \qquad (8)$$

The number of chain mails is [1, 2, 3, 4]

$$W = \frac{z_1 + z_2}{2} + \frac{2A_{prel}}{p} + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \frac{p}{A_{prel}}. \qquad (9)$$

In order to obtain small overall dimensions, it is chosen A=30p; so the number of chain mails will be

$$W = \frac{z_1 + z_2}{2} + 60 + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \frac{1}{30}. \qquad (10)$$

Figure 5 shows the variation of the chain mails number, according to the determined number of teeth z_1.

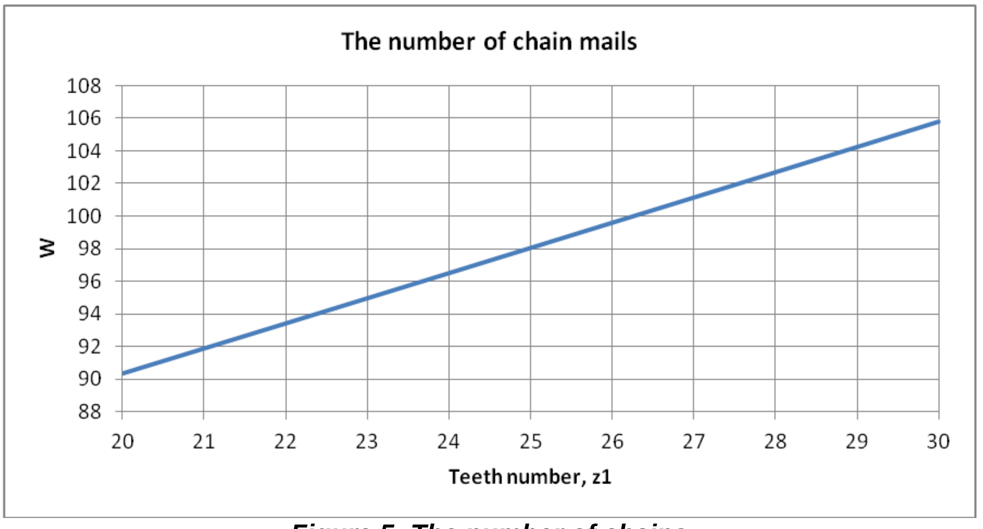

Figure 5: The number of chains

The recalculated axis distance is established with [1, 2, 3, 4]

$$A_r = \frac{p}{4}\left[W - \frac{z_1 + z_2}{2} + \sqrt{\left(W - \frac{z_1 + z_2}{2}\right)^2 - 8\left(\frac{z_2 - z_1}{2\pi}\right)^2}\right]. \qquad (11)$$

Due to the mounting conditions (a mounting deflection should be assured), the final axis distance is obtained by reducing the calculated axis distance with (0.002 … 0.004)A_{rec}. The final value is A=475 mm (independent on the teeth number z_1).

3. Conclusions

The aim of the chains drives design process is oriented in order to obtain, for given data, a chain solution which is characterized by a small pitch and small overall dimensions. The small pitches assure small internal dynamic loads due to the polygonal shape of the contact area between the chain and the chains wheels.

The literature indicates given single values for the teeth number of the driver chain wheel, depending on the transmissions ratios.

According to the studies presented in the paper, a high teeth number of the driver chain wheels is offering the possibility to obtain small values for the maximum pitch and also the calculus based on these principle is giving as result a small number of chains.

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