Abstract—The applicability area of the SPA and SPZ V-belt transmissions is referring on the transmissions which are loaded with small and medium loads and on the transmissions where the accuracy is not an important parameter. Due to design calculus, there are many variants in order to choose the design parameters; the aim is to obtain small distances between the V-belts wheels axes and a small number of belts. An important problem is represented by the cases when the designer, according to the design data, may choose both variants of the V-belts: SPA or SPZ. The paper presents the influence of the V-belts type on the axis distance and belts number values. A finite element analysis is made in order to find out which variants of belts – SPA or SPZ – are reliable to be chosen, according to the equivalent stresses.

Keywords—ANSYS, axis distance, design, finite element modelling, V-belt.

I. INTRODUCTION

The V-belt are used in mechanical transmissions which are working with a power up to 1200 (kW), with a maximum peripheral speed of 50 (m/s) and with an axis distance up to 3 (m); the average efficiency of them is between 95% ... 96% [1]. The V-belt transmissions are used in mechanical transmissions as links between the motors and the speed reducers and in friction transmissions, where the accuracy is not an important parameter.

![Fig. 1. Mechanical transmission with v-belt](image)

Fig. 1 presents an application of a v-belts transmission in a mechanical transmission composed by an electric motor 1, a speed reducer 3 and a lifting machine 4.

A problem is represented by the cases when, according to the input data, the designer may chose two variants of V-belts. The question is which of these variants is reliable according to the number of the belts, to the axis distance and to the equivalent stresses.

The paper presents some practical aspects of the design calculus of a V-belt transmission which is used as a link between an electric motor and a speed reducer and identifies the influences of the main parameters on the belts number, on the axis distance and on the equivalent stresses.

II. THE DESIGN PROCESS

The design process is presented for a V-belt transmission which convey a power \( P = 5 \) (kW) and an input rotation \( n_1 = 500 \) (rpm), from a three phase induction motor [2]; the output rotation is \( n_2 = 350 \) (rpm) and represents the input rotation for a speed reducer which is acting a lifting machine; the lifting machine is working 16 hours a day.

The type of the V-belt is chosen according to the power and the input rotation; for the design data, the V-belt type is on the border between the SPZ and SPA type [3]. Due to this, the design calculus will be achieved for both variants with the values for the primitive diameter of the driver wheel \( D_{p1} = 100 \) (mm) - SPZ belt and \( D_{p1} = 112 \) (mm) - SPA belt [3].

The transmission’s ratio is defined as

\[
i = \frac{n_1}{n_2}.
\] (1)

According to the input data, \( i = 1.42 \); the driven wheel primitive diameter is obtained with

\[
D_{p2} = iD_{p1}.
\] (2)

the values are chosen from [4]: the driver wheel primitive diameter \( D_{p2} = 140 \) (mm) and the driven wheel primitive diameter \( D_{p2} = 160 \) (mm). The medium diameter of the wheels is

\[
D_{pm} = \frac{(D_{p1} + D_{p2})}{2},
\] (3)

with the values 120 (mm) and 136 (mm). The tensioning wheel’s diameter is [5]
\[
D_{p0} = (1...1.5)D_{p1}
\] (4)

with the values 110 (mm) and 123 (mm).

The preliminary axis distance is established between the limits of [4]

\[
0.7(D_{p1} + D_{p2}) \leq A \leq 2(D_{p1} + D_{p2})
\] (5)

with the intervals \( A \in [168, 480] \) (mm) for SPZ belt and \( A \in [190.4, 544] \) (mm) for SPA belt.

The angle of the V-belt’s branches is

\[
\gamma = 2\arcsin \frac{D_{p2} - D_{p1}}{2A}
\] (6)

and the winding angles on the driver and the driven wheels

\[
\beta_1 = 180^\circ - \gamma
\] (7)

and

\[
\beta_2 = 180^\circ + \gamma
\] (8)

The primary length of the belt is [4]

\[
L_p = 2A + \pi D_{pm} + \frac{(D_{p2} - D_{p1})^2}{4A}
\] (9)

Considering the variation of the axis distance as \( A \in [168, 480] \) (mm) for SPZ belt and \( A \in [190.4, 544] \) (mm) for SPA belt, the primary length of the belt is calculated for the two types of belts; Fig. 2 shows the variation the primary length with the axis distance. The values of the primary length are \( L_p \in [715.37, 1337.82] \) (mm) for SPZ belt and \( L_p \in [760.81, 1466.05] \) (mm) for SPA belt. From the standards, are adopted the values for the primary length \( L_p \); \( L_p \in [710, 800, 900, 1000, 1120, 1250, 1400] \) (mm) for SPZ belt and \( L_p \in [800, 900, 1000, 1120, 1250, 1400] \) (mm) for SPA belt [5].

Fig. 3 presents the chosen possible variants of the primary length; the smallest value of 710 mm is accepted only for the SPZ belt.

The final axis distance is established with [4]

\[
A = p + \sqrt{p^2 - q}
\] (10)

where:

\[
p = 0.25L_p - 0.393(D_{p1} - D_{p2})
\] (11)

and

\[
q = 0.125(D_{p2} - D_{p1})^2
\] (12)

The values of the axis distances are presented in the Fig. 4; the SPA belts variants assures small axis distances (relative to the values obtained for the SPZ belts) do to the bigger values of the primitive diameters of the wheels.

The preliminary number of belts is established with [4]

\[
z_0 = \frac{c_p P}{c_1 c_p^2 P_0}
\] (13)

where: \( c_f \) is the functioning conditions coefficient, \( c_p = 1.2 \) [4]; \( c_1 \) is the coefficient of the length, \( c_1 \in \{0.84, 0.86, 0.88, 0.90, 0.93, 0.94, 0.96\} \) for SPZ belts and \( c_1 \in \{0.81, 0.83, 0.85, 0.87, 0.89, 0.91\} \) for SPA belts [5]; \( c_p \) is the winding coefficient, \( c_p \in \{0.965, 0.97, 0.975, 0.98, 0.988, \)}
0.992, 0.993} for SPZ belts and \(c_z \in \{0.965, 0.973, 0.976, 0.985, 0.991, 0.9925\}\) for SPA belts [5]; \(P_0\) is the nominal power transmitted by one belt, \(P_0 = 1\) (kW) for SPZ belts and \(P_0 = 1.58\) (kW) for SPA belts.

Fig. 5 presents the values of the preliminary belts number; due to the higher values of the nominal power which can be transmitted by one belt, the SPA type belt assures small values for the preliminary number of the belts (relative to the values obtained for the SPZ belts).

\[ z = z_0 / c_z, \]  

(14)

where \(c_z\) represents the belts number coefficient, \(c_z = 0.85\) for SPZ belt and \(c_z = 0.90\) for SPA belt [4].

Fig. 6 shows the final number of the belts \(z\) which is established by rounding \(z\) to the most appropriate integer value; small belts number are obtained for the SPA belts, with the primary length variants from 3 to 7 (which means \(L_p \in \{1120, 1250, 1400\}\) (mm) – see Fig. 3).

An important parameter which influences the durability of the belt is the bending frequency \(v\). Depending on the belts number \(z\), on the velocity of the belt \(v\) and on the primary length \(L_p\), the bending frequency is defined as [6]-[8]

\[ v = v z / L_p, \]  

(15)

where the velocity of the belt is

\[ v = \pi D_{p1} n_1 = \pi D_{p2} n_2. \]  

(16)

According to the calculus, the value of bending frequency – for all the cases – is situated bellow the allowable value of 60 Hz [6]-[8] – see Fig. 7. Small values of the bending frequency assure higher durability; in this case the variants with high values of the primary length, by considering the SPA type of V-belt, are preferable to be used.

An important criterion which influences the V-belts usage is represented by the stress produced from the external load. There are analyzed one variant of SPZ belt and one variant of SPA belt, by using the finite element method, with the software ANSYS 14.0 [9]. It is made a static structural analysis by considering the external load as the traction force \(F_t\) which is acting on one belt. The boundary conditions are defined as considering the belt fixed on the wheel – the sliding phenomenon is not considered.

For the modelling is considered the variant 4 of belts with the following input data: for the SPZ V- belt – the primitive diameter of the driver wheel \(D_{p1} = 100\) (mm) and the number of belts \(z = 8\) and for the SPA V- belt – the primitive diameter of the driver wheel \(D_{p1} = 112\) (mm) and the number of belts \(z = 5\).

The external traction force which is acting on one belt is determined by [6]-[8]

\[ F_t = 2.2 \frac{M_{tt}}{D_{p1}}, \]  

(17)

where the torque is established with

\[ M_{tt} = 9.55 \cdot 10^6 \frac{P}{n_1}. \]  

(18)

By considering the power \(P=5\) (kW) and an input rotation \(n_1=500\) (rpm), it results the value of the force \(F_t=262.62\) (N) for the SPZ belt and \(F_t=375.17\) (N) for
the SPA belt; according to the calculated values, it is noticed that the SPA belt is higher loaded than the SPZ type.

The distribution of the equivalent stresses is presented in the Fig. 8 for the SPZ belt and in the Fig. 9 for the SPA belt.

An interesting observation is that the SPA belt has a small value for the maximum equivalent stress 33.45 (MPa) even is higher loaded than the SPZ belt which has a big value for the equivalent stress 46.65 (MPa); so, by considering the maximum equivalent stress, the SPA belt type is preferable to be used.

III. CONCLUSION

The studies presented in the paper are referring on a particular case of the V-belts design process: according to the values of the transmitted power and of the input rotation, the designer can choose both of the V-belts variants – SPZ belts and SPA belts. The V-belt type is influencing the axis distance (and indirectly the V-belts primary length) and the number of the belts; generally, the aim of the design process is to obtain small primary lengths and a small number of belts in order to reduce the costs and the overall dimensions.

According to the results presented in the Fig. 10, the SPA variant of V-belt assures a small number of belts; in order to obtain a small value of the of the axis distances (Fig. 4) there are chosen SPA belts, also.

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