Modeling of power losses in poly-V belt transmissions: hysteresis phenomena (standard analysis)

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Abstract
A simulation software is being developed for predicting the power losses in poly-V belt transmissions. For this purpose, some theoretical models have been implemented considering different types of power losses taking place in a Front Engine Accessory Drive (FEAD). Even though these power losses come from different phenomena, the analysis in this paper is focused on the elastomeric belt hysteresis losses due to dynamic bending, stretching, shear, flank and radial compression of the belt rubber. Experimental tests are performed in order to determine the intrinsic loss modulus of the belt elastomeric constituent causing the loss of energy. This study permits determining a hysteresis power loss map of a belt transmission.

Keywords: FEAD, Accessory drive, Poly-V belt transmission, Power losses, Hysteresis loss

1. Introduction

Poly-V belts are nowadays widely used in the automotive and truck industries and are considered to be an essential part of the power transmission of vehicles.

In most cases, on the front engine of vehicles, power is delivered with a single poly-V (v-ribbed) belt from the crankshaft to the individual accessories such as the compressor, the alternator, the water and steering pump, etc.

Fig. 1 Example of a Poly-V belt transmission with focus on the poly-V belt (inset).
The belt transmission including all the individual accessories is commonly called Front Engine Accessory Drive (FEAD), see Fig. 1. In order to properly transmit power, poly-V belts are generally made of reinforced elastomer; more precisely, unidirectional reinforcing tension cords are inserted to support the tension in the longitudinal direction. The elastomeric matrix provides good compliance to bend easily when running on and off the pulleys and a high friction surface of contact which interacts directly with the pulleys.

Nowadays, new environmental regulations push truck and car manufacturers to reduce the power losses of their engines. Most of the works related to power losses in belt transmission were initiated by Gerbert (1999). However, only flat and V belts were considered. Furthermore, constant elastomeric properties of the belt rubber (storage and loss moduli) were used even though polymeric materials are known to exhibit complex time and temperature dependent hysteretic properties. Several authors (Chen et al. 1998, Greenberg et al. 1995) studied the global efficiency of belt transmission and therefore the global power loss. In the case of poly-V belts, experimental studies were carried out on the analysis of the pulley-belt slip (Manin et al. 2009) and on the pulley-belt friction coefficient identification (Cepon et al. 2010). Recently, the speed losses in poly-V belt drives with two equal-sized pulleys have been studied by Balta et al. (2015).

The present paper focuses on the power loss related to the hysteresis dissipation of the belt submitted to dynamic loading, following up the model developed by Manin et al. (2014) and implementing improvements.

2. Standard analysis

2.1. Hysteresis power losses of belts

Similarly to other power transmissions, the energy lost by a functioning FEAD has several origins: the hysteresis of the belt elastomer (rubber), the belt-pulley slip and losses from the mechanical components of the system: bearings and tensioner. The present paper focuses on the belt hysteresis phenomena: bending, stretching, shear, flank and radial compression. Indeed, energy is dissipated when the belt runs on and off the pulleys due to the hysteretic behavior of its constitutive elastomer. In addition to that, this energy and therefore power is lost by hysteresis when the belt is loaded in different modes: the belt hysteresis phenomena.

The power loss occurs, because the constitutive elastomeric material exhibits a viscoelastic yet dispersive behavior, which loses energy as heat when a load is applied, then removed, Fig. 2.

According to Meyers & Chawla (1999) the amount of energy lost is equal to \( \oint \sigma \, d\varepsilon \) or using some trigonometric relations from the triangles in blue, Fig. 2, the energy lost can be quantified by:

\[
W_h = \pi E^* \varepsilon_a^2
\]  

(1)

Where \( E^* \) is the loss modulus and \( \varepsilon_a \) a given strain value for which the power loss by hysteresis is intended.

Fig. 2  Typical Strain-Stress hysteretic behavior of a linearly viscoelastic material submitted to periodic loading.
The loss modulus $E''$ in Eq. (1) is the imaginary part of the complex modulus $E^*$ which usually depends on the strain amplitude $\varepsilon$, the frequency $\omega$ and the temperature $T$. The real part of $E^*$ is the viscoelastic material storage modulus $E'$. These moduli are represented in Eq. (2).

$$E^* = E' + iE'' = f(T, \omega, \varepsilon)$$  \hspace{1cm} (2)

Concerning the temperature effect on the complex $E^*$ modulus (Eq. 2), $E^*$ is strongly influenced by the temperature. Thus, it is important to use moduli ($E''$, $E'$, $E''$) at the appropriate temperature from belt elastomer characterization (DMA: Dynamical Mechanical Analysis). In this paper, the temperature of 25 and 60 degrees Celsius are considered.

### 2.2. Bending losses

Bending is the deformation mode that results from the belt running on and off pulleys, Fig. 3. This type of power loss does not depend on the belt tension $F$, and therefore, on the torque (or power) transmitted. The strain value equivalent to that in Eq. (1) can be calculated from the beam bending theory:

$$\varepsilon_{a_ben} = \frac{x}{R}$$  \hspace{1cm} (3)

Where $1/R$ is the curvature and $x$ the through-thickness distance from neutral axis of the belt, where the axial strain equals zero. Anyway, for a flat belt made of homogeneous isotropic material, the bending strain in a belt takes place as described in Fig. 3. Nevertheless, for the case of the serpentine belts as depicted in Fig. 4, the neutral axis coincides with the cord layer.

Equation (1) suggests that the elastomer loss modulus $E''$ has to be evaluated on the strain value $\varepsilon_a$. Then, the loss modulus of the constitutive materials and the strain resulting from the system geometry, are used to determine energy losses due to belt bending hysteresis following the equation derived hereafter, Eq. (4).

![Fig. 3  Bending theoretical aspect when a belt is running on pulleys.](image)

![Fig. 4  Poly-V belt cross section: location of the neutral axis and key geometric parameters.](image)
Where $B$ is the poly-V belt width, $H$ the different thicknesses of each belt layer, $\alpha$ the rib wedge angle, $n$ the number of ribs, Fig. 4, and $W_{h,ben}$ represents the energy loss per belt unit length.

Note that in Eq. (4), it was assumed that the loss modulus $E'$ depends heavily and exclusively on the temperature $T$ and the angular frequency $\omega$ ($2\pi$ frequency). Indeed, the loss modulus $E'$ of the elastomeric belt does not change much within the realistic range of strain $\varepsilon_a$ experienced by the belt, so its variation will be neglected in the following. In addition, concerning the belt ribs layer ($H_c$ thickness in Fig. 4), the equivalent belt width $B_{eq}$ is also considered when calculating losses, because of the lack of material between the ribs. Since empty areas do not contribute to any hysteresis energy loss $W_h$.

Also, it is worth noting that the Eq. (1) is demonstrated for the strain magnitude varying from $-\varepsilon_a$ to $+\varepsilon_a$ (traction-compression), but e.g. when the belt is lying inside in the top layer the strain varies from $0$ to $+\varepsilon_{H_t}$ (traction only), if this strain range would be changed to an equivalent interval, from $-\varepsilon_{H_t}$ to $+\varepsilon_{H_t}$, the strain magnitude used to calculate the amount of energy lost has also to be changed to $+\varepsilon_{H_t}/2$. Thus, in Eq. (4) the $1/4$ factor appears from the squared term.

2.3. Stretching losses

The belt is subjected to tension variation $\Delta F$ as it runs on a pulley, see Fig. 5. In the case of stretching, the strain value equivalent to that in Eq. (1) can be calculated from the Hooke’s law:

$$\varepsilon_{a, st} = \frac{\Delta F}{E \ A} = \frac{F_T - F_S}{E \ A}$$

(5)

Where, as said before, $\Delta F$ is the effective pull transmitted between the pulleys via the free spans of the belt, $F_T$ (tight) and $F_S$ (slack) are the belt span tensions around a pulley and $E \ A$ the strain stiffness along the belt, with $E'$ the belt storage modulus in the longitudinal direction multiplied by its cross section area $A$.

This type of power loss only takes place on pulleys transmitting power. Stretching loss does not take place on idler or tensioner pulleys, because tensioners are designed to do not impose a resisting torque, so the strain in Eq. (5) is consequently null since $\Delta F = 0$.

Fig. 5  Stretching theoretical aspect when a belt is running on pulleys.
Thus, the equation of the stretching hysteresis energy lost per belt unit length is:

$$W_{h, st} = \frac{1}{4} \pi E' \varepsilon_{a, st}^2 A \tag{6}$$

Where $E'$ is the loss modulus from the belt material when it is stretched, $\varepsilon_{a, st}$ the strain value from the system operating conditions and belt strain stiffness, $A$ the belt cross section area from the belt geometry, Eq. (7).

$$A = B (H_t + H_b) + n \left[ B_d - H_c \tan\left(\frac{\alpha}{2}\right) \right] H_c \tag{7}$$

All geometric parameters in Eq. (7) are represented and can be better understood in Fig. 4.

2.4. Flank compression losses

Considering the compression of the poly-V belt flanks, Fig. 6. The strain value equivalent to that in Eq. (1) can be calculated from the Hooke’s law:

$$\varepsilon_{a, fc} = \frac{P_z}{E'} \tag{8}$$

Where $E'$ represents the storage modulus when the viscoelastic material (rubber) is compressed, $P_z$ is the horizontal component of the pulley-belt contact pressure $P_c$, and they are related as follows:

$$P_z = P_c \cos(\alpha/2) \tag{9}$$

The flank compression forces are represented in Fig. 7. Moreover, the belt span tension profile on pulleys (Fig. 8) implies that the contact pressure $P_c$ either is constant or depends on the arc of contact $\phi$ (Gerbert, 1999).
Fig. 8  Poly-V belt span tension profile along the arc of contact of pulleys when transmitting power.

On the driver (DR) pulley, in the sliding arc $\phi_{RS}$, Fig. 8, the contact pressure $P_c(\phi)$ can be estimated by:

$$P_c(\phi) = \frac{(F_t - F_z) e^{-f \phi}}{2 n H_c R \tan(\frac{\alpha_2}{2})}$$

Similarly, for a driven (DN) pulley, in the sliding arc $\phi_{NS}$:

$$P_c(\phi) = \frac{(F_z - F_t) e^{+f \phi}}{2 n H_c R \tan(\frac{\alpha_2}{2})}$$

Where, in addition to the belt geometric parameters and the belt span tensions, $F_c$ is the centrifugal action, $f$ the belt-pulley friction coefficient. The constant pressure $P_c$ then $P_z$ acting in the adhesion arc $\phi_a$ of both DN and DR pulleys is obtained when $\phi = 0$ in equations 10 and 11. Also, these equations (10, 11) are valid for $\phi$ varying from 0 to $\phi_s$ as shown in Fig. 8. The sliding angle $\phi_s$ is found using the well-known capstan equation (Euler, 1762), then the adhesion angle $\phi_a$ comes from the relation $\phi = \phi_s + \phi_a$. The belt-pulley wrap angle $\phi$ must be a priori known and it comes from the FEAD geometry. Finally, the equation governing the loss of energy by flank compression hysteresis is given:

$$W_{h,fc} = \frac{1}{4} \pi E^- \varepsilon_{a,fc}^2 n Acs_{one-V}$$

Where $E^-$ is the loss modulus from the rib material, when it is compressed, $\varepsilon_{a,fc}$ the strain value from the system operating conditions and geometry Eq. (8) and $n$ the number of ribs. The $Acs_{one-V}$ is the cross section area of only one V and it can be estimated by Eq. (13).

$$Acs_{one-V} = \left[B_d - H_c \tan(\frac{\alpha}{2})\right] H_c$$

2.5. Radial compression losses

The radial compression of the poly-V belt middle and top layers are considered, Fig. 9. In this case, the strain value equivalent to that in Eq. (1) can also be calculated from the Hooke’s law, Eq. (14).

$$\varepsilon_{a,rc} = \frac{P_v}{E H_b}$$
Fig. 9  Radial compression theoretical aspect when a belt is lying inside and outside pulleys.

When the belt is lying inside, i.e. with its ribs in contact with the pulley (Fig. 9), the pressure $P_v$ responsible for compressing and decompressing the rubber causing the energy lost by hysteresis is given by Eq. (15).

$$P_v = \frac{2 P_c H_c \tan(\frac{\alpha}{2})}{B_d} \tag{15}$$

When the belt is lying outside, i.e. with its back side in contact with the pulley, normally in FEADs, this is an idler pulley with no resisting torque. Therefore, no sliding arc exists and $\phi = \phi_a$. Then, the pressure $P_v$ in Eq. (14) must be replaced by a flat belt-pulley contact pressure $P_c$ which is constant and equal to Eq. (16).

$$P_c = \frac{F - F_c}{R B} \tag{16}$$

Where $F$ can be $F_t$ or $F_s$, since in the vicinity of an idler pulley the belt span tension ideally does not change, $R$ is the pulley pitch radius and $B$ the belt width. Thus, the equation of the radial compression hysteresis energy lost per unit length on a pulley is:

$$W_{h,rc} = \frac{1}{4} \pi E_{H_b}^* \varepsilon_{a(rc)}^2 Acs_{H_b} \tag{17}$$

Where $E_{H_b}^*$ is an appropriate loss modulus from the belt middle material, $\varepsilon_{a(rc)}$ the compression strain value Eq. (14), $Acs_{H_b}$ the cross section area of the belt middle layer, which is:

$$Acs_{H_b} = B \ H_b \tag{18}$$

Where $B$ is the belt width and $H_b$ is the thickness of the belt middle layer, Fig. 4. In addition to that, in the previous paragraph the word “appropriate” was used to designate the modulus, because we here have to consider that the belt is thin and wide so plain strain conditions prevail rather than plain stress. This implies that the classical (loss or storage) modulus divided by $(1 - \nu^2)$ looks appropriate leading to:

$$E_{H_b}^* = \frac{E^*}{(1 - \nu^2)} \tag{19}$$

Where $E^*$ is e.g. the classical loss modulus from a DMA test (from $H_b$ or $H_t$ layer) and $\nu$ the Poisson ratio.

Note that in the case of belt lying outside, in Eqs. (14), (17), (18) and (19) the dimension $H_b$ must be replaced by $H_t$. The storage modulus in Eq. (14) is also affected by the plain strain condition that is why the notation $E_{H_b}^*$ instead of $E^*$. 

2.6. Shear losses

When the belt is running on pulleys subjected to resisting torques there exists a sheared belt layer $H_{sh}$, Fig. 10. That can be $H_t$ or $H_b$ depending on the belt side in contact with the pulley. For poly-V belts lying inside, the sheared layer $H_{sh}$ in Fig. 11 can be calculated as follows:

$$H_{sh} = H_b + H_c^*$$  \hspace{1cm} (20)

With

$$H_c^* = \frac{H_c^2 \tan\left(\frac{\alpha}{2}\right)}{B_d}$$  \hspace{1cm} (21)

Note that the distance $H_c^*$ is obtained by doing the equivalence between the areas $A1$ and $A2$ in Fig. 11. Thus, the equation of the shear hysteresis energy lost per unit length when the belt is lying inside is:

$$W_{h,sh} = \frac{1}{4} \pi G^* \gamma_{a,sh}^2 Ac_{sheared}$$  \hspace{1cm} (22)

Where $G^*$ is the shear loss modulus of the belt rib and middle layers material, these are very often the same, and if the material is considered homogeneous and isotropic the following relation is valid:

$$G^* = \frac{E^*}{2 (1 + \nu)}$$  \hspace{1cm} (23)

![Fig. 10](image1.png) Sheared belt layer from the belt tension cords to the contact surface of the pulley theoretical aspect.

![Fig. 11](image2.png) Hypothetical poly-V belts sheared layer $H_{sh}$ (belt lying inside the pulley).
The cross section area subjected to shear loading $A_{cs,sheared}$ in Eq. (22) is equal to:

$$A_{cs,sheared} = (H_b + H_c) B$$  \hspace{1cm} (24)

Finally, in Eq. (22), similar to others, Hooke’s law can be applied and the shear angle $\gamma_{a,sh}$ (Fig. 12), enables the shear stress calculation, Eq. (25).

$$\tau_{sh} = G' \gamma_{a,sh}$$  \hspace{1cm} (25)

Where $G'$ is the shear storage modulus. Furthermore, $\tau_{sh}$ is the shear stress, which in FEADs, cyclically occurs and causes the loss of energy by (shear) hysteresis. Note that the Eq. (23) between shear and traction-compression moduli can also be applied to find $G$ for Eq. (25).

In parallel, if $\gamma_{a,sh}$ is considered to be small and constant from Fig. 12 the following relation can be established:

$$\tan(\gamma_{a,sh}) = \frac{\sigma}{H_{sh}} \rightarrow \sigma = H_{sh} \gamma_{a,sh}$$  \hspace{1cm} (26)

Where $\sigma$ is the shear deflection, and according to the book by Gerbert (1999), from the shear theory chapter (p. 91), this elongation of the cord have to take two cases into consideration, Fig. 13.

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**Fig. 12** Sheared surface area and shear deflection of a small poly-V belt element.

**Fig. 13** Shear deflection at driver (DR) and driven (DN) pulleys.
The shear stress $\tau_{sh}$ can be found by back substitution into Eqs. (25) and (26), taking into account the shear deflection in Fig. 13 (right) for driven (driver) pulleys, where the (pulley) belt runs faster than the (belt) pulleys:

$$\tau_{sh} = \frac{\sigma}{H_{sh}} G' = \frac{G'}{H_{sh}} \left( \frac{\Delta V_s}{\omega} \phi + y \right)$$

(27)

Where $\omega$ is the pulley angular velocity, $\phi$ the wrap angle at the contact point considered, $\Delta V_s = V - R \omega$ is the relative velocity, which moves the belt cord layer, and $y$ the sum of increments $(\epsilon - \epsilon_s) R \, d\phi$ along $\phi$ represented by the following integral:

$$y = \int_{0}^{\phi_a} (\epsilon - \epsilon_s) \, R \, d\phi = \int_{0}^{\phi_a} \frac{\phi_a F(\phi) - F_s}{E' \, A} \, R \, d\phi$$

(28)

Where $\epsilon_s$ is the belt (tensile) strain at the seating point, $F$ the belt span tension with the tensile modulus $E'$ and area $A$.

Thus, the shear deflection $\tau_{sh}$, for a driven pulley is expressed by:

$$\tau_{sh} = \frac{G'}{H_{sh}} \left( \frac{\Delta V_s}{\omega} \phi + \int_{0}^{\phi_a} \frac{\phi_a F(\phi) - F_s}{E' \, A} \, R \, d\phi \right)$$

(29)

Similarly, for driver pulleys one gets:

$$\tau_{sh} = \frac{G'}{H_{sh}} \left( \frac{\Delta V_T}{\omega} \phi + \int_{0}^{\phi_a} \frac{\phi_a F(\phi) - F_s}{E' \, A} \, R \, d\phi \right)$$

(30)

By introducing the following non-dimensional notation:

$$G_s^2 = \frac{G' \, B \, R^2}{H_{sh} \, E \, A}$$

(31)

Also, considering an increase in tension over a small poly-V belt element $R \, d\phi$:

$$dF = \tau_{sh} \, B \, R \, d\phi$$

(32)

The terms (31) and (32) together with the shear deflection from a driven pulley, Fig. 13 (right):

$$\sigma = \frac{\Delta V}{\omega} \phi + y$$

(33)

Then, from the Eqs. (25), (26), (32), (33) and after differentiation to eliminate integral terms the following differential equation is valid for a driven pulley:

$$\frac{d^2 F}{d\phi^2} = G_s^2 \, F = G_s^2 \left( \frac{c \, \Delta V_s}{V} - F_s \right)$$

(34)

Where $c = \sum E_i \, A_i$ ( $i$ is equal to the number of individual members constituting the Poly-V belt) is the strain stiffness, $E'$ the tensile modulus, $A$ the belt cross section area and $V$ the belt linear velocity.
Next, if the equation 34 is solved using appropriate Boundary Conditions (BCs), i.e., no shear at the beginning of the adhesive arc $\phi_a (\gamma = \tau = dF/d\phi = 0)$ and where the belt seats on the pulley, at $\phi = 0$, the belt span tension $F = F_s$, its analytic solution for driven pulleys is:

$$F(\phi) = + \frac{c}{V} \Delta V_s \left( \cosh(G \phi) - 1 \right) + F_s \quad (35)$$

Similar analysis, the Eq. (32) and its BCs, can be made and the following equation demonstrated for driver pulleys:

$$F(\phi) = - \frac{c}{V} \Delta V_T \left( \cosh(G \phi) - 1 \right) + F_T \quad (36)$$

In addition to these formulations physical quantities of $\Delta V_s$ and $\Delta V_T = R \omega - V$ can be obtained from Gerbert (1999). Finally, with $F(\phi)$ provided by Eqs. (35, 36), the Eqs. (29, 30) can be supplied, then integrated, and by back substitution the energy lost by shear hysteresis can be calculated by Eq. (22).

2.7. Total belt hysteresis losses

Once all the energy losses per unit length equations have been established, i.e. Eqs. (4), (6), (12), (17) and (22), the hysteresis power loss due to bending, stretching, shear, flank and radial compression phenomena can also be determined according to the following equation:

$$P_h = W_h V \quad (37)$$

Where $P_h$ is the hysteresis power loss in watts and $V$ the belt linear velocity in meters per second.

3. Results

3.1. Power loss map

Results are presented, in the form of a power loss map (Fig. 15) for a FEAD example defined in table 1. The poly-V belt of simulation is detailed in table 2. The FEAD under consideration is a 3 pulleys system with an idler pulley as depicted in Fig. 14. The pulley 1 is assimilated to the driver pulley, the two pulleys 2 and 3 are accessory driven pulleys and an idler pulley is added as it is often the case for Front Engine Accessory Drives (FEADs). There are several interests of introducing an idler pulley, one of these is to increase $\phi$, and therefore, to decrease the setting tension $F_o$.

For the simulation case, here, $F_o = 1325$ Newtons, and concerning the operating conditions the work temperature equal to 25 and 60 Celsius and the engine is considered to be running at constant idling speed (600 RPM) with no engine speed fluctuation. In addition, from Manin (2010) in Fig. 14 the belt span tensions can be calculated (quasi-static loading): $F_1 = F_o + T_1/(2 r_1) = 1742$ , $F_2 = F_1 - T_2/r_2 = 908$ and $F_3 = F_2 - T_3/r_3 = 75$ Newtons.

Table 1  Belt transmission (example) geometric parameters and operating conditions.

<table>
<thead>
<tr>
<th>No</th>
<th>Type</th>
<th>Friction $f$ coefficient</th>
<th>Radius $r_i$ [mm]</th>
<th>$x_i$ : $y_i$ [mm]</th>
<th>Constant Torques $T_i$ [N.m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>DR</td>
<td>1</td>
<td>100</td>
<td>0 : 0</td>
<td>83.3</td>
</tr>
<tr>
<td>2</td>
<td>DN</td>
<td>1</td>
<td>60</td>
<td>100 : 250</td>
<td>50</td>
</tr>
<tr>
<td>3</td>
<td>DN</td>
<td>1</td>
<td>60</td>
<td>-100 : 250</td>
<td>50</td>
</tr>
<tr>
<td>4</td>
<td>DN</td>
<td>0.5</td>
<td>35</td>
<td>-70 : 145</td>
<td>0</td>
</tr>
</tbody>
</table>
Moreover, when the engine is running and the belt is winding accessory drives, the frequency $\Omega_{ii}$ which each phenomenon of hysteresis takes place can be calculated as follows for bending, flank and radial compression:

$$\Omega_{ii.ben} = \Omega_{ii.fe} = \Omega_{ii.re} = \frac{V}{\phi_{ii} R_{ii}}$$  \hfill (38)

Or for stretching and shear hysteresis losses:

$$\Omega_{ii.st} = \Omega_{ii.sh} = \frac{V}{L}$$ \hfill (39)

Where $V$ is the belt linear velocity, $L$ the belt length and $\phi_{ii} R_{ii}$ the arc of contact of the pulley "ii".

Indeed, Eqs. (38) and (39) are the frequencies which the hysteresis loop in Fig. 2 for each phenomenon is completed: one loading-unloading cycle. According to these frequencies the belt material must be tested with DMA, to guarantee an accurate representation of the system and the FEAD power loss quantities.

Also, in order to supply power loss models, e.g. from the poly-V belt top $H_t$, the middle $H_b$ and rib $H_c$ layer materials, the empirical equations describing the elastomer complex modulus in MPa as a function of the bending frequency $\Omega_{ii.ben}$ in Hertz were obtained by Dynamical Mechanical Analysis (strain of 0.2%), Eqs. (40, 41).

### DMA data on material properties at 25 degrees:

$$E^*_{H_t} = E^*_{H_b} = E^*_{H_c} = E^* + i E'' = 48.44 \left(\Omega_{ii.ben}\right)^{0.0565} + i \left[ 6.1 \left(\Omega_{ii.ben}\right)^{0.064} + 0.22 \right]$$ \hfill (40)

### DMA data on material properties at 60 degrees:

$$E^*_{H_t} = E^*_{H_b} = E^*_{H_c} = E^* + i E'' = 30.95 \left(\Omega_{ii.ben}\right)^{0.0565} + i \left[ 3.3 \left(\Omega_{ii.ben}\right)^{0.068} + 0.58 \right]$$ \hfill (41)
Later, the aforementioned fits were combined with the energy loss equations to predict the FEAD power loss coming from all hysteresis phenomena, Fig. 15.

3.2. Prediction of power loss

To predict the power losses, fits such as they in Eq. 40, for bending, stretching, shear, flank and radial compression were combined with energy loss equations Eqs. (4), (6), (12), (17) and (22), to predict the FEAD power losses coming from all hysteresis phenomena. The table 3 provides simulation results of the belt transmission example in Fig. 15 (left).

Thanks to the power loss map (Fig. 15, left) of the table 3, for example, we notice that, of the total amount lost by bending (34.2 W) only 2.14 W is lost on the crankshaft (pulley 1). Also, we quickly notice that the FEAD amount lost by bending is divided among all pulleys, which would be consistent, since the belt always bends when passing by FEAD components. Moreover, the (bending) loss of energy is proportional to the degree of bending: i.e. 56.7% (19.4 W) of bending is lost on pulley 4 (Fig. 15, left), it is the biggest bending amount surely, since the pulley 4 has the smallest radius, then it imposes the biggest bending strain value $\varepsilon_{a, b e n}$, which is proportional to the energy lost by bending, Eq.(4).

The results of table 3 are valid under the following conditions: the temperature equal to 25 degrees Celsius, the frequencies, Eqs. (38, 39), corresponding to the engine idling speed (600 RPM) and the dynamic strains around 0.4%.

It is worth noting that around 1% ($\frac{PL}{AP}$, table 3) of the power on input shaft of the belt transmission (Fig. 14) is lost by hysteresis of the belt. The amount of energy lost can vary depending on the belt transmission and its characteristics.

![Power loss map of the belt transmission example at 25°C (left) and 60°C (right).](image)

Table 3  Power losses in Watts (W) by hysteresis phenomena and characteristic angles in Degrees (deg).

<table>
<thead>
<tr>
<th></th>
<th>AP : Available engine Power at 600 rpm (W):</th>
<th>5236</th>
</tr>
</thead>
<tbody>
<tr>
<td>PL : Total amount of Power Losses (W):</td>
<td>46.4</td>
<td></td>
</tr>
<tr>
<td>$\phi$ (deg)</td>
<td>$\phi_s$ (deg)</td>
<td>$\phi_k$ (deg)</td>
</tr>
<tr>
<td>Pulley 1</td>
<td>197.9</td>
<td>12.9</td>
</tr>
<tr>
<td>Pulley 2</td>
<td>103.3</td>
<td>65.8</td>
</tr>
<tr>
<td>Pulley 3</td>
<td>166.4</td>
<td>18.8</td>
</tr>
<tr>
<td>Idler</td>
<td>107.6</td>
<td>107.6</td>
</tr>
</tbody>
</table>
For example, considering the same belt elastomer, the same FEAD geometry and the same engine work conditions, a thick belt would dissipate more energy than a thin belt, since a thick belt has more elastomer to be loaded and unloaded.

According to DMA experiments, Eqs. (40, 41), and data from Gerbert (1999), it can be noticed that storage $E'$ and loss $E''$ moduli decrease with temperature and increase with frequency. In addition to that, as described in Eq. (1), the loss modulus of the belt elastomer is directly proportional to the energy (power) loss. Thus, for engine work temperature higher than 25 Celsius, smaller amounts of energy lost (Eq. 1) should be expected and vice-versa, (e.g. Bending, Fig. 15). However, in the belt longitudinal direction the storage modulus of the belt-cords is also affected by decreasing temperature. So, sometimes increasing temperature does not necessarily mean decreasing hysteresis losses, Fig 15.

4. Conclusions

The power losses due to the belt bending, stretching, shear, flank and radial compression hysteresis have been investigated in this paper. The research was done in the frame of the belt Front Engine Accessory Drive used in trucks.

Theoretical models have been developed and implemented in order to quantify an accurate amount of energy lost by hysteresis. In this paper, it was highlighted that the system (FEAD) can lose energy by belt bending, stretching, shear, flank and radial compression hysteresis. The hysteresis phenomena are internal losses (friction inside the belt). These losses are supplementary heat; they are energy (power) lost as heat, in addition to the dissipation at the surface of the belt due to friction and sliding between the belt and the pulley groove (external losses).

In our modelling the belt elastomers intrinsic properties were accounted for. More precisely, the belt elastomer viscoelastic properties were determined by DMA experiments. The dependency on frequency has been highlighted and implemented. Then, a power loss map for a quasi-static application was provided, Fig. 15.

The power loss map represents an easy and practical way to do a diagrammatic representation of power losses. It also permits determining the phenomenon (e.g. bending, stretching, shear, flank or radial compression), the location (pulley 1, 2, 3 or idler) and the quantity (amount in Watts) from which power is lost in a belt transmission, Fig. 15.

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References


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