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STUDY OF THE TAF 2012 DRIVELINE (CV-CD)

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Abstract

TAF is the acronym for Articulated Forestry Tractor – a heavy vehicle uses in a logging operation for pulling cut trees out of a forest, in a process called “skidding”. TAF use a cardan shafts transmission from the gear box (CV) to distribution box (CD). The distribution box split the movement thru axels or winch. The study applied the cardan theory on a specific case and based on the numeric results a few design modifications are proposed for reducing the angular velocity pulse on the CD input shaft. Theoretical results may have a significant importance in practice, impacting the reduction of transmission noise and vibrations - already claimed in the case of the distribution box for the current constructive solution. An original AutoLISP program are made in order to analyze and optimize the transmission, with graphical results.

Key words: Driveline, Cardan shaft, Skidder, Pulse, Angular velocity

1. TAF driveline

TAF is the acronym for Tractor Articulat Forestier (Articulated Forestry Tractor) – trademark registered by the company IRUM Reghin [4]. TAF is designed for forestry operations and uses the winch as main tool. A diesel motor drives the machinery during travel or when working with the winch.

Figure 1 shows the diagram of the transmission of motion from the engine to the axles and winch. The motion transmission powertrain consists of the elements from Table 1, numbered as in fig. 1.

Table 1: TAF driveline elements

No.	Element	Chassis
1	Engine	Front
2	Gear box - CV	Front
3	Cardan	Front
4	Telescopic cardan	-
5	Distribution box - CD	Rear
6	Cardan	Rear
7	Winch	Rear
8	Telescopic cardan	-
9	Cardan	Front
10	Axle	Balancer-Front
11	Cardan	Rear
12	Axle	Rear

The A-A joint allows the relative rotation of the front chassis in relation to the rear chassis to ensure the steering of the machinery. The skidder cabin is attached to the front chassis and, to prevent the machinery from overturning on slopes transversal to the direction of movement, the front axle is mounted on a oscilating frame (balancer) to the front chassis (axis B-B).

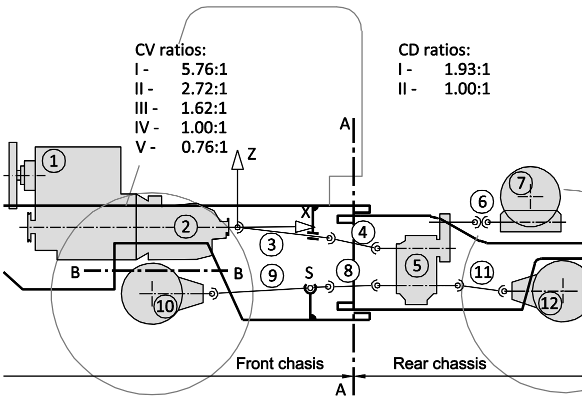


Fig. 1: TAF driveline elements

The motion is transmitted from the gear box to the transmission box via a universal double joint, consisting of cardans 3 and 4, the latter being telescopic. It is associated to an axes system located in the centre of the universal joint at the output of the gear box (fig. 1). Consistent with the numbering of the kinematic elements in fig. 1, the following notation will be considered:

- φ_2 - the rotation angle of the output shaft from the gear box 2

- ω_2 - the angular speed of the output shaft from the gear box 2
- φ_3 - the rotation angle of cardan 3
- $\Delta\varphi_3$ - the phase difference introduced by the cardan 3; $\Delta\varphi_3=0°$ if the output fork of cardan 3 is rotated by 0° relative to his input fork; in fig. 2, $\Delta\varphi_3=90°$ and $\Delta\varphi_4=0°$
- γ_3 - the angle between the cardan 3 axis versor and X axis versor; the direction of the cardan versors corresponds to the direction of transmitting the motion from the engine to the driven final elements (winch and axles, via the transmission box)
- $\gamma_{2,3}$ - the angle between the axes of cardans 2 and 3
- etc.

The goal of this study is to minimise the vibrations and noise in distribution box (CD), by optimizing the cardans shaft alignments, especially in steering position of the skidder.

2. Angular speed at the CD input

At first, there is extracted the transmission powertrain towards the transmission box (fig. 2). The question that arises is the determination of the variation of the angular speed for driving the transmission box, in relation to the rotation angle of the gear box output shaft. In this regard, Table 2 prepares the required numerical data. All axes are deemed to be in the same plane: the XZ plane. Therefore, the machinery is not steering and there are no errors in the alignment of axes.

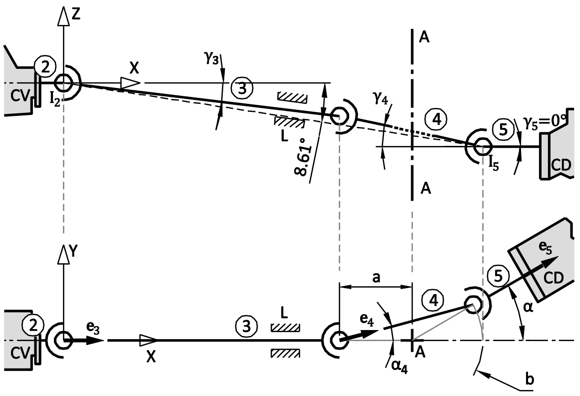


Fig. 2: TAF driveline from CV to CD

Based on these assumptions, there can be obtained the following known relations for the calculation of the angular speed of cardan 3 and of its rotation angle in relation to those of the output shaft of gear box 2 [1]:

$$\omega_3 = \frac{cos\gamma_3 \cdot \omega_2}{sin^2\varphi_2 + cos^2\gamma_3 \cdot cos^2\varphi_2}$$

$$cos^2\varphi_3 = \frac{cos^2\varphi_2 \cdot cos^2\gamma_3}{sin^2\varphi_2 + cos^2\gamma_3 \cdot cos^2\varphi_2}$$

In the second step, for a $\Delta\varphi_3$ phase difference of the connection flanges located on cardan 3, the relations which allow the determination of the angular speed and the rotation angle of cardan 4 are as follows:

$$\omega_4 = \frac{cos\gamma_{4,3} \cdot \omega_3}{cos^2(\varphi_3 + \Delta\varphi_3) + cos^2\gamma_{4,3} \cdot sin^2(\varphi_3 + \Delta\varphi_3)}$$

$$cos^2\varphi_4 = \frac{sin^2(\varphi_3 + \Delta\varphi_3) \cdot cos^2\gamma_{4,3}}{cos^2(\varphi_3 + \Delta\varphi_3) + cos^2\gamma_{4,3} \cdot sin^2(\varphi_3 + \Delta\varphi_3)}$$

where, $\gamma_{4,3} = \gamma_4 - \gamma_3$

In the end, the angular speed at the transmission box input and the rotation angle of its input shaft are obtained using the following relations:

$$\omega_5 = \frac{\cos\gamma_{5,4} \cdot \omega_4}{cos^2(\varphi_4 + \Delta\varphi_4) + cos^2\gamma_{5,4} \cdot sin^2(\varphi_4 + \Delta\varphi_4)}$$

$$cos^2\varphi_5 = \frac{sin^2(\varphi_4 + \Delta\varphi_4) \cdot cos^2\gamma_{5,4}}{cos^2(\varphi_4 + \Delta\varphi_4) + cos^2\gamma_{5,4} \cdot sin^2(\varphi_4 + \Delta\varphi_4)}$$

where, $\gamma_{5,4} = \gamma_5 - \gamma_4$

For the initial data shown in Table 2, there was drawn a number of diagrams for shafts 2, 3, 4 and 5. There was selected an engine rotation of 800 rpm and the second step - corresponding to a transmission ratio of 2.72. Of main interest is the variation of the angular speed of shaft 5 - CD input - in relation to the rotation angle of the CV output shaft 2, at a constant speed of the motor.

Table 2: Study cases for α=0°

Notation	Value		
	Case 1	**Case 2**	**Case 3**
n_1	800 rpm		
i_{II}	2.72:1		
φ_2	0÷360°		
α	**0°**		
γ_2	0°	0°	0°
γ_3	6.82°	6.82°	8.61°
γ_4	12°	12°	8.61°
γ_5	0°	0°	0°
$\Delta\varphi_3$	0°	90°	90°
$\Delta\varphi_4$	90°	0°	0°
$\Delta\omega_5$	**1**	**0.56**	**0**

In this regard, the diagrams in fig. 3 and 4 has angle φ_2 on the abscissa and the differences from the angular speeds of shafts 3, 4, 5 and the constant angular speed of shaft 2 on the ordinate, $\Delta\omega_i = \omega_i - \omega_2$, $i = 3, 4, 5$. There were considered three cases:

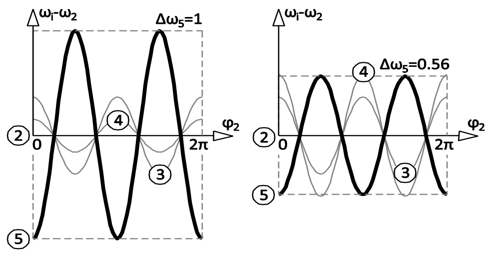


Fig. 3: Angular speed pulse: **Case 1 and 2**

- **Case 1**, corresponding to the current state of tilting and phase difference of the cardan axes. For comparisons not linked to the drive shaft rotation, it was considered that in this case the magnitude of the angular speed pulsations at the CD input is unitary ($\Delta\omega_5$=1 rad/sec).
- **Case 2**, corresponding to the situation where the phase difference of the flanges associated to a cardan was applied to cardan 4 instead of cardan 3, without modifying the tilting angles of the cardans. This minor modification alone led to halving the angular speed pulsations as compared to case 1!
- **Case 3**, corresponding to the situation where the relative position of CD towards CV is maintained, but the tilting angles of cardans 3 and 4 are equalized at a value of 8.61° - corresponding to the tilting of the straight line joining the articulation centres I_2 and I_5 (see the notations in fig. 2). The result is the expected one, of annulling the pulsations of the angular speed at CD input.

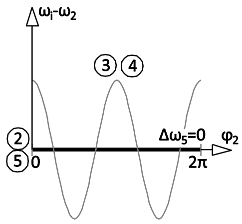


Fig. 4: Angular speed pulse: **Case 3**

Case 3 involves a modification of the construction at the **L** bearing, posing no technical complications with the advantage of annulling the pulsations of the angular speed. The results above are only valid if the angles of the chassis is null (α=0°).

3. The effect of articulating the chassis

A relative rotation of the two chassis leads to changing the angles between the axes of cardans 3 and 4, as well as between cardans 4 and 5. The new angles will depend on the α rotation – as angle made by the CD input axis with the XZ plane. Figure 2-botom show a top view of the transmission, at the relative articulation of the chassis with angle α; there should be noted point A where the articulation axis intersects the top view plane.

In order to determine the new angle between cardans 4/3 and 5/4, the projections of the versors of the axes of these cardans in the XYZ system are considered (fig. 2):

$$\begin{cases} e_{3x} = cos\gamma_3 \\ e_{3y} = 0 \\ e_{3z} = sin\gamma_3 \end{cases},$$

$$\begin{cases} e_{4x} = cos\gamma_4 \cdot cos\alpha_4 \\ e_{4y} = cos\gamma_4 \cdot sin\alpha_4 \\ e_{4z} = sin\gamma_4 \end{cases},$$

$$\begin{cases} e_{5x} = cos\gamma_5 \cdot cos\alpha \\ e_{5y} = cos\gamma_5 \cdot sin\alpha \\ e_{5z} = sin\gamma_5 \end{cases}$$

Based on the scalar product of versors, the new angles between cardans 4/3 and 5/4 are:

$$\gamma_{4,3} = \arccos(e_{3x} \cdot e_{4x} + e_{3y} \cdot e_{4y} + e_{3z} \cdot e_{4z})$$

$$\gamma_{5,4} = \arccos(e_{4x} \cdot e_{5x} + e_{4y} \cdot e_{5y} + e_{4z} \cdot e_{5z})$$

Where, using the notations from fig. 2, the tilting angle of cardan 4 in relation to the XZ plane, noted α_4, can be calculated according to the rotation angle α of the chassis and the constructive dimensions a, b:

$$sin\alpha_4 = \frac{b \cdot sin\alpha}{\sqrt[2]{a^2 + b^2 + 2abcos\alpha}}$$

Figures 5a, b shows the three cases, whose initial data is shown in Table 3. These cases refer to a relative rotation of the two chassis by **α=20°**. There is monitored the variation of the angular speed for the CD input shaft in relation to the rotation angle of the CV output shaft. The magnitude of the pulsations $\Delta\omega_5=\omega_5-\omega_2$ is compared to value 1 assigned to case 1.

Table 3: Study cases for α=20°

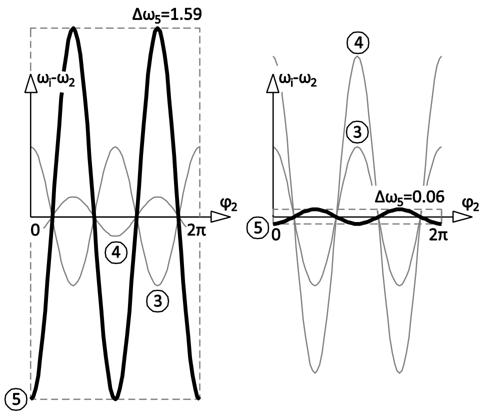
Param.	Value		
	Case 4	**Case 5**	**Case 6**
n_1	800 rpm		
i_{II}	2.72:1		
φ_2	0÷360°		
α	**20°**		
γ_2	0°	0°	0°
γ_3	6.82°	8.61°	8.61°
γ_4	12°	8.61°	8.61°
γ_5	0°	0°	0°
$\Delta\varphi_3$	0°	0°	90°
$\Delta\varphi_4$	90°	90°	0°
$\Delta\omega_5$	**2.62**	**1.59**	**0.06**

The analysis of the diagrams from fig. 5a shows a significant amplification of the pulsations for the current constructive solution, concomitant with the steering of the machinery, by 2.62 times!

There should be noted the influence of the site where the phase difference of the connection flanges is applied in the cardan transmission. In the case of a zero relative rotation of the chassis and of an alignment of the axes of cardans 3 and 4 ($\gamma_3=\gamma_4$=8.61°), the $\Delta\omega_5$ pulsations remain null regardless of where the phase difference is applied – on cardan 3 ($\Delta\varphi_3$=90°), or on cardan 4 ($\Delta\varphi_4$=90°).

On the other hand, together with the relative rotation of the chassis, the alignment of cardans 3 and 4 is no longer possible and the results are significantly better if the phase difference is applied to cardan 4. The relative rotation α of the chassis is limited to ±45°.

Fig. 5a: Angular speed pulse: **Case 4**



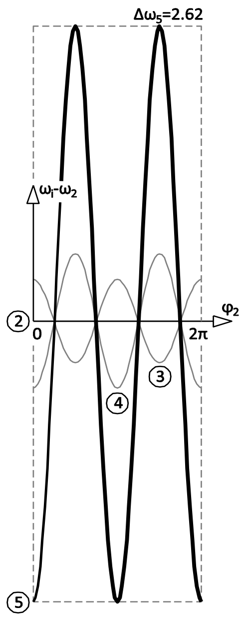


Fig. 5b: Angular speed pulse: **Case 5 and 6**

4. Conclusions

Figures 1, 2, 3 and 4 are at scale. The variation diagrams of the angular variations were drawn by writing and running in the AutoCAD environment of an AutoLISP software developed by the author [2, 5].

Theoretical results may have a significant importance in practice, impacting the reduction of transmission noise and vibrations - already claimed in the case of the transmission box for the current constructive solution.

There is proposed to verify in practice a constructive solution involving the modification of the **L** supporting bearing of cardan 3 – to align cardans 3 and 4 ($\gamma_3=\gamma_4=8.61°$) – as well as shifting the phase difference to cardan 4 ($\Delta\varphi_3=90°$, $\Delta\varphi_4=0°$).

The research will continue on the following directions:

- Performance of tests on a prototype where cardans 3 and 4 are aligned, with phase difference applied on cardan 4 - to verify in practice the theoretical results. There will be
- recorded data regarding two physical quantities: (i) vibration; (ii) noise.
- The study of the variation of angular speed for the input shafts in the two axles and winches, where CD works with two transmission ratios during travel and one transmission ratio while standing still and working with the winch.
- The study of the influence of the rotation of the balancer on the angular speed pulsations for the input shafts in the two axles, when two disarticulations are combined: α (chassis articulation), and β (balancer articulation).

Reference

[1] Dudiţă F. (1966), Cardans Transmissions, Technical Ed., Bucharest.

[2] Dudiţă F. (1977), *Mechanisms*, University of Braşov, Department of Mechanisms and Machine Elements.

[3] Pozdîrcă A. (2010), *Calculation and representation of curves and surfaces*, the “Petru Maior” University Publishing House.

[4] ***TAF 2012, *TAF Owner's Manual*, S.C. IRUM S.A., Reghin, 2016.

[5] ***AutoCAD 2015, Autodesk Inc., 2014