Drag phenomena within a torque converter driven automotive transmission - laminar flow approach

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Abstract. When discussing a torque converter driven, automotive transmission with respect to the vehicle’s coasting mode, automotive engineers have to take into account the slip between the converter’s propeller and turbine. If the turbine isn’t locked to the propellers during coasting process, drag phenomena within the converter’s fluid occur and they have to be properly assessed when computing the coasting process dynamics. The best way to make the needed evaluation is to have a separate torque converter and test it on a test bench, if the data provided by the manufacturer, in this respect, weren’t available. But there are several issues that could baffle this action. Among them, one could find the lack of information from the manufacturer, missing (bankrupted) manufacturer, classified information, old (out of date) products and so on. An even more challenging situation consists in dealing with a military special vehicle. Actually, the vehicle that would be subjected to the following topic is a military tracked, heavy vehicle (MBT) with a planetary driveline, driven by its engine via a hydraulic torque converter. In the attempt to assess its’ coasting dynamic performances, we faced the problem of the reverse rotation of the torque converter that strongly influences the general drag of the vehicle’s motion. Hence, this paper tries to provide a method to determine the transmission overall drag considering the torque converter as being its main contributor. The method is based on the experimental research our team has performed in the last several months. Using high-quality software and adjacent mathematics while assuming a certain sort of flow type within the torque converter, we aimed at determining the parameter of interest of the flow. The method can be successfully used for all type of hydrodynamic components of the transmission under the condition of developing the necessary experimental research. As far as the test were concerned, they were the typical ones designed to determine - on experimental basis - the mass inertial moments of the transmission components using the “falling weight” principle.

1. Introduction
Special vehicles require big amounts of money to have their experimental studies developed. Therefore, simulations would be a better second choice whenever comes about assessing a future product’s dynamic features. Our institution is being involved in several tests with respect to some special vehicles, aiming at determining their dynamic features. This paper focuses on assessing the coasting parameters of a special tracked vehicle that has a hydro torque converter in its driveline.
Generally speaking, the dynamic features of the vehicle can be analytically obtained when taking-off [1, 5]. Some of the parameters of the taking-off equations are also used in the coasting equations. We don’t hereby intend to promote some vehicle’s motion equations, they are already well known [5]. Within both these categories of equations, there is a term that estimates the inertia moments of the moving parts of the vehicle’s driveline. When designing the vehicle and manufacturing its transmission’s parts, these inertia moments can be easily acquired both from computation or measurements [1,5]. That is impossible for an already built-up vehicle. This is why we tried to figure out a way to compute the global moments of inertia of a vehicle’s transmission. The method itself isn’t new at all, it is well known among the automotive engineers (figure 1).

The vehicle is missing its tracks on both sides (unlike the one in the picture) so its sprockets, connected to the transmission, are both free. The sprocket on one side of the vehicle hosts a cable, which is coiled around its hub for about 5 to 6 meters long (depending on the crane’s height). At the end of the cable, falling down the crane’s arm, a weight is hanging. The test itself is quite simple. The sprocket acts like a spool. At the beginning, the weight is uppermost. The driver presses the clutch and depresses the brakes and the weight starts to fall. Knowing the mass of the weight, the spooling radius and the length of the fall, while measuring the falling time, one can find the global moment of inertia of the rotating parts of the driveline. Since the global moment of inertia depends on the gear ratio, the test should be performed for the each vehicle’s gear.

2. Particular features of the experiment. Paper’s goal.
The experiment is mainly suitable for the vehicles that have mechanical clutches. When disconnecting the clutch from the engine, the transmission parts are completely free to start rotating. No friction, except for the one from gearing, occurs. Moreover, this friction can be kept “under control” when evaluating the moments of inertia, since this is rather easy to assess it.

When a hydro torque converter is involved, beside the above mention sources of friction, another important source comes into being, i.e. the friction of the converter’s fluid. The impeller stands still (it is connected to the stopped engine) while the turbine starts to rotate, together with the rest of the transmission parts. Moreover, the angular speed of the turbine is different from gear to gear since the total ratio differs from gear to gear. Eventually, for the same gear, the turbine’s angular speed also varies according to the increasing speed of the accelerated weight on its way down.

We’ll try to provide a method to determining the global moment of inertia of a transmission that involves a torque converter, taking into account the liquid friction that occurs between the converter’s turbine and impeller, after properly processing the experimental data [4].

2.1. Mathematical model of the phenomenon
Figure 2 depicts the diagram of the mathematical model. The first assumption was that the whole transmission’s inertia is the same as of a ring that replaces the sprocket’s spool with the spooling radius, \( r \). The liquid friction of the torque converter is assumed to act along the ring’s one side face, at the same radius, \( r \). The rest of the transmission’s friction is considered to be constant, no mater the angular speed or gear ratio and contained by the converter’s frictional torque. Hence, the only friction
that varies is the one in the converter, referred as \( M_f R \). The moment of inertia \( M_i \) brakes the motion while the load \( m g \) accelerates it (with \( m \) the weight mass).

For the left side of the pulley, the motion equation can be written as \( m \frac{dv}{dt} = mg - T \), while for the left side of the pulley, we can write down \( Tr = l \frac{d\omega}{dt} + c\omega \), that assumes a laminar (linear dependency) flow within the converter. Adding the cinematic connection, \( \frac{dv}{dt} = r \frac{d\omega}{dt} ; \omega = r\omega \), we get the following system:

\[
\begin{align*}
  m \frac{dv}{dt} &= mg - T \\
  Tr &= l \frac{d\omega}{dt} + c\omega \\
  \frac{dv}{dt} &= r \frac{d\omega}{dt} \\
  \omega &= r\omega
\end{align*}
\]

(1)

In system (1) \( a = \frac{dv}{dt} \) is the linear acceleration acting over the right side of the pulley. On the both sides of the pulley \( T \) is the cable (thread) tension. The moment of inertia can be written as \( M_i = l \frac{d\omega}{dt} \) where \( l \) is the global kinetic moment of inertia of the transmission and \( \omega \) is the angular speed of the flywheel that stands for the whole transmission. Since the flow inside the torque converter was supposed to be laminar, the frictional moment of resistance (drag moment) can be expressed as \( M_f c R \omega \), assuming a linear dependence of the drag moment to the angular speed of the equivalent flywheel. In this expression, \( c \) is a constant (actually, a drag coefficient). The drag moment consists of a drag force placed at the \( r \) radius of the flywheel (a ring-like flywheel, with a single radius \( r \)). Eventually, \( t \) is the time (independent) variable of the actual process.
Extracting \( T = mg - m \frac{dv}{dt} \) from the first equation of system (1) and replacing it in the second one we get 
\[
\left( mg - m \frac{dv}{dt} \right) r = I \frac{d\omega}{dt} + c \omega
\]
we get \( \omega + \omega = \omega \). Yet, taking into account the first cinematic connection it leads to
\[
mgr - mr^2 \frac{d\omega}{dt} = I \frac{d\omega}{dt} + c \omega
\]
or grouping the terms according to \( \omega \) we get:
\[
mgr = \left( mr^2 + I \right) \frac{d\omega}{dt} + c \omega
\]  
(2)
This last equation can be further processed, to get some constant terms in front of the angular speed variable and its first derivative: 
\[
\frac{d\omega}{dt} = \frac{mgr}{mr^2 + I} - \frac{c}{mr^2 + I} \omega
\]  
Having \( \frac{mgr}{mr^2 + I} = b \) and \( \frac{c}{mr^2 + I} = a \) it yields to 
\[
\frac{d\omega}{dt} = b - a \omega
\]
Separating the variables, eventually delivers:
\[
dt = \frac{d\omega}{b - a \omega}
\]  
(3)

2.2. Integrating the equation of motion
The integration of equation (3) is rather simple. The method of integration involves the separation of the variables. Thus, equation (3) can be written as
\[
\int_0^t \frac{d\omega}{dt} = \int_0^\omega \frac{d\omega'}{b - a \omega'}
\]
For a definite integral it yields to 
\[
t = \int_0^\omega \frac{d\omega}{b - a \omega}
\]
Since for \( \omega = 0 \), \( ln1 = 0 \) (limit conditions) then for real, positive numbers, the solution can be written as 
\[
\frac{e^{-at} \omega}{b^t} = 1 - \frac{a \omega}{b}
\]
that eventually provides the solution 
\[
\omega = \frac{b}{a} \left( 1 - e^{-at} \right)
\]
Should be appropriate now to replace the previously defined constants, \( a \) and \( b \), 
\[
\frac{b}{a} = \frac{mgr}{mr^2 + I} = \frac{mgr}{c} \quad a = \frac{c}{mr^2 + I}
\]
so we can finally get 
\[
a = \frac{c}{mr^2 + I}
\]
and 
\[
b = \frac{mgr}{mr^2 + I}
\]
Therefore, the final solution of the equation, expressing the angular speed evolution of the sprockets’ angular speed when the weight is falling under the given circumstances (among we find laminar flow approach of the converters’ frictional torque) will be given by:
\[ \omega = \frac{mg r}{c} \left( 1 - e^{-\frac{c}{mr^2 + I} t} \right) \] (4)

Should be noticed that the above equation has, from the experimental data point of view, two unknown parameters, i.e. the global friction constant \( c \) (met in the moment of global friction expression as \( M_{fr} = c \omega \)) and the kinetic moment of inertia \( I \). One way to experimentally find constant \( c \) is to horizontally pull the cable of the sprocket at constant speed. Since \( M_{fr} = c \omega \), while measuring \( M_{fr} \) and knowing the angular speed, the constant will come up, as will be further seen. Eventually, we could compute the kinetic moment of inertia for every gear of the transmission.

3. Experimental data

We have performed several tests with the same falling weight for each gear of the gearbox, except for the 4th gear (a failing transducer prevented in getting reliable results for the last gear [2]; nevertheless, while the remaining three sets of results were good and reliable). Another test has been also performed. It consisted in horizontally pulling the cable from the spool at constant speed while measuring needed force. This test has been performed for each gear and it aimed to experimentally determine the \( c \) constant of the friction moment expression \( M_{fr} = c \omega \). Since we knew the spooling radius, the pulling speed and force, it was easy to determine the above-mentioned coefficient. The values of the constant \( c \) are given in table 1, along with other data.

As an example, Figure 3 depicts the time histories of the measured angular speed for three different tests developed for the third gear of the transmission (due to resolution reasons, only three curves have been plotted). Figure 3 depicts the approximation curves of the experimental data plotted in Figure 3 (after properly processing [4]).

Using the Curve Fitting module of the Matlab programming environment [3] and considering the motion as developed according the motion law provided by equation (4), we computed the most accurate global kinetic moment of inertia of the transmission, \( I \), for each researched gear ratio of the transmission. Hence, given equation (4) and its known, contained parameters, the module provided the global kinetic moment of inertia \( I \) for each gear. The data are given in table 1.

Analyzing the data in the table one can notice that the constant affecting the angular speed within the drag moment has a rather small distribution of its values around 110...120 [s/kg\(^2\)/m]. That could be a proof of the quality of the measurements since it has to keep itself around a certain value [4].

Another way to check for the accuracy of both the measuring procedures and the mathematical model was to check for the terminal velocity (angular speed of the sprocket). Of course, we couldn’t measure it for real since the cable was too short, but using the extended time history of the modeled curves (using equation (4)) we have got the results depicted in figure 5.
We overlapped several curves to determine the value of the terminal angular speed of the sprocket wheel. It came out that the value of the angular speed is about 15.8 s\(^{-1}\). It corresponds to a value of about 150 revolutions per minute and it is reached sooner or later, depending on the gear ratio of each gear of the transmission. What is more important is that the vehicle’s speed that corresponds to this value is about 15 km/h. This leads to a very important conclusion: the engine brake is extremely effective for this vehicle and, as known, for all the tracked vehicles. It occurs whenever the vehicle’s speed exceeds the average values of 12 to 15 km/h. The “Error” column in the table below provides the overlapping error between the approximation curves in figure 4 and the time histories in Figure 3.

| Article | Test no. | m [kg] | r [m] | c \[s/kg\(^2\)m\] | I \[kg\cdot m^2\] | Error [%] |
|---------|----------|--------|-------|----------------|----------------|-----------|
| I       | 1        | 635    | 0.254 | 110            | 8100           | 7.8       |
|         | 2        |        |       | 140            | 7900           | 6.5       |
|         | 3        |        |       | 140            | 8000           | 5.7       |
|         | Average values » |       |       | 130            | 8000           | 6.7       |
| II      | 1        | 635    | 0.254 | 100            | 4400           | 8.5       |
|         | 2        |        |       | 120            | 4200           | 6.8       |
|         | 3        |        |       | 105            | 4400           | 7.6       |
|         | Average values » |       |       | 108            | 4333           | 7.6       |
| III     | 1        | 635    | 0.254 | 100            | 3300           | 5.1       |
|         | 2        |        |       | 100            | 3200           | 6.4       |
|         | 3        |        |       | 100            | 3000           | 5.9       |
|         | Average values » |       |       | 100            | 3167           | 5.8       |

4. Conclusions

Tracked vehicles are rarely braked since their propulsion system is featured by a low efficiency. Nevertheless, for this kind of vehicles would be useful to learn what the coasting regime leads to. We tried to experimentally reveal the dragging behavior of the torque converter (within a military vehicle’s transmission) and determining the global kinetic moments of the transmission for each gear, hence the coasting regime could be studied.
The dragging phenomenon starts whenever the terminal speed is overtaken, and that means about 15 km/h for the actual vehicle. Small fluctuation around this terminal velocity may occur due to the different ratios of the transmission. As a matter of fact, it varies between about 12 km/h in the first gear to about 15 km/h in the third one.

The mathematical model of the angular speed variation has been confirmed with enough accuracy (in the terms provided by equation (4)). That allowed us to determine the terminal speed.

The model allowed us to compute the global kinetic moments of the transmission. Their values are quite similar to the theoretical ones, validating our theoretical model.

We consider that the model provided in the paper hereby can be further developed using turbulent flow models for the torque converter.

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