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DESIGN AND CONSTRUCTION ASPECTS OF A ZERO INERTIA CVT FOR PASSENGER CARS

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ABSTRACT–This paper concentrates on the design and construction aspects of a transmission for a mid-class passenger car with internal combustion engine. The transmission, consisting of a Continuously Variable Transmission (CVT) with a Van Doorne V-belt, a planetary gear set and a compact steel flywheel is used to prove the concept of mechanical torque assist. The design goal is to obtain a proof of concept transmission with maximal efficiency, using proven transmission technology. With the developed so called Zero Inertia CVT, the fuel economy of the car is improved by operating the engine at its fuel optimal operating line. To achieve a good vehicle acceleration response, the flywheel assists the powertrain mechanically.

KEY WORDS : Transmission design, CVT, Flywheel, Transmission efficiency

1. INTRODUCTION

The vehicle acceleration response to a driver's request of today's powertrains with a continuously variable transmission (CVT) is compromised when pursuing maximal fuel efficiency from the internal combustion engine. This compromise is the result of the small torque reserve at engine operating points with high fuel efficiency. An additional energy buffer and power assist source is needed to abandon the compromise between acceleration performance and fuel economy. This can be achieved by a mechanically driven flywheel. A new transmission concept is evolved in which a compact flywheel system is integrated in a powertrain with CVT. With the so called Zero Inertia (ZI) transmission there will never be a shortage of power to the wheels at the drivers request, not even when tracking the engine operating points with high fuel efficiency. The functional design and behaviour of the ZI transmission is described in Van Druten and Vroemen (1999). The steering and control of the driveline is described in Serrarens and Veldpaus (1999). The mechanically created power assist function has the ability to provide a high power density at low costs and can be constructed to have low energy losses because of the absence of energy conversions (i.e. chemical to electrical to mechanical). Furthermore the ZI transmission is composed of conventional components and the flywheel is rotating at speeds comparable to those of other transmission components (maximum speed of 8000 rpm). The kinematic layout of the ZI transmission is determined amongst others by the power losses in the flywheel system due to air drag and bearing friction. The CVT powertrain with integrated flywheel is optimized to achieve a minimum power loss in the total driveline. Also the dynamics of the flywheel system and, more especially of the rotor, the shaft and the bearings are part of the optimization. A prototype driveline is being developed and built into a passenger car. This so called "Proof of Concept" of the ZI powertrain will be under testing at the date of publication.

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2. PROOF OF CONCEPT TRANSMISSION

To achieve the development of the proof of concept ZI transmission within one year, the design uses many proven parts of the Van Doorne V-belt CVT. As a result the planetary gear set and the flywheel are positioned on the backside of the transmission. According to the functional design (Van Druten and Vroemen, 1999) two members of the planetary gear set each have to be connected to either the primary or the secondary pulley of the CVT.
The flywheel has to be connected to the third member

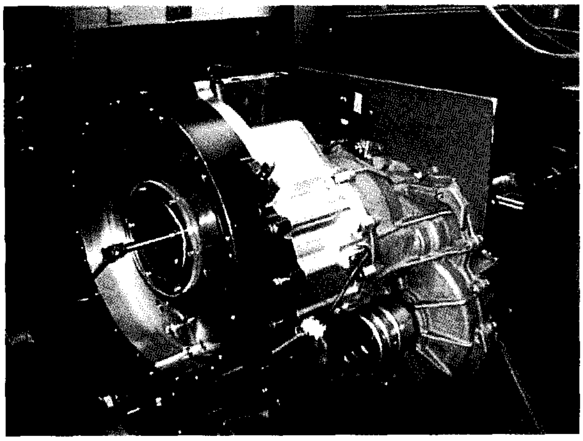

Figure 1. Proof of concept transmission on a test-rig.

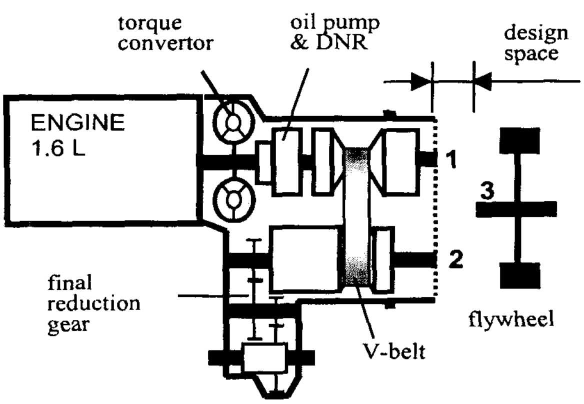

Figure 2. Conventional CVT layout with to be connected flywheel.

of the planetary gear set.

In Figure 2 the layout of the conventional CVT is illustrated. The torque convertor, used for smooth acceleration at low speeds, will be closed above 20 km/h with a lock-up clutch. The roller-vane oil pump can be switched either single or double sided according to the pressure demand. The V-belt is a standard Van Doorne 30/12 type, with a ratio-coverage of 5.3. The final two-stage reduction gear has a total ratio of 0.21. Within the given design space indicated in Figure 2 of approximately 70 mm, the flywheel including a planetary gear connecting the shafts 1, 2 and 3 is designed.

3. PLANETARY GEAR DESIGN

According to the functional design of the ZI transmission (Van Druten and Vroemen, 1999) any type of planetary gear can be used to connect the shafts 1, 2 and 3. For a compact construction it is preferable to position the planetary gear in parallel with the primary and secondary pulley so the flywheel can be positioned close to the primary pulley in axial direction.

In figure 3 an optimized gear layout is shown to have minimum power losses within the system. The optimization procedure according to (Van Druten and Kok, 1999) is used for different planetary gear layout designs, resulting in a basic three element planetary gear. The planetary gear speeds are both reduced from primary and secondary side with a reduction step of approximately 0.3. Figure 4 shows a cross section of the planetary gear set, the bearings of the carrier are fitted left and right (not shown).

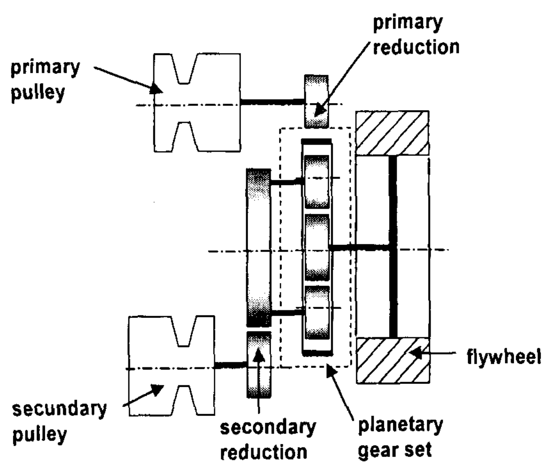

Figure 3. Planetary gear lay-out of ZI-CVT.

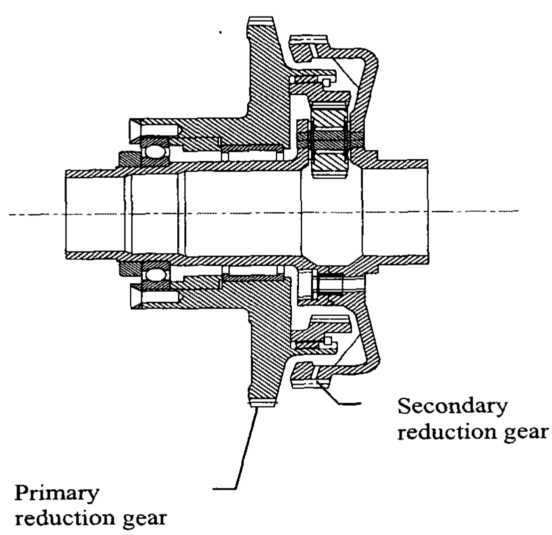

Figure 4. Cross section of planetary gear set.

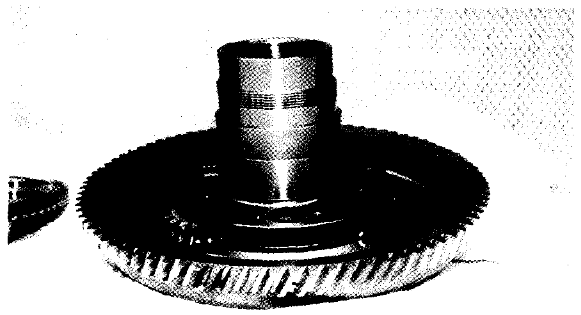

Figure 5. Carrier to be driven by the secondary pulley.

Figure 5 shows the carrier of the planetary gear set of Figure 4 with the planets mounted.

4. FLYWHEEL DESIGN

The flywheel design is based on an optimization to reduce air drag and bearing losses as described in (Van Druten and Kok, 1999). For this application the optimization prescribes a ring shaped steel flywheel with inertia J=0.4 [kg.m^2], rotating at a maximum speed of ω = 800 [Rad/s]. Such a choice immediately eliminates common flywheel problems like extreme stresses and the need for a vacuum system to reduce the air drag. With these specs, a steel flywheel is evident because of the high mass density, low cost and good manufacturing properties.
The flywheel construction as shown in Figure 7 in cross section can be divided into three separate parts: the shaft, the rotor and the bearings. The shaft and rotor are built separately to simplify the manufacturing process and are bolted together. On the left hand a cylindrical roller bearing is applied and on the right hand a pre-tensioned pair of spindle bearings. In this set up the shaft is allowed to grow thermally and all axial forces are supported by the spindle bearing.

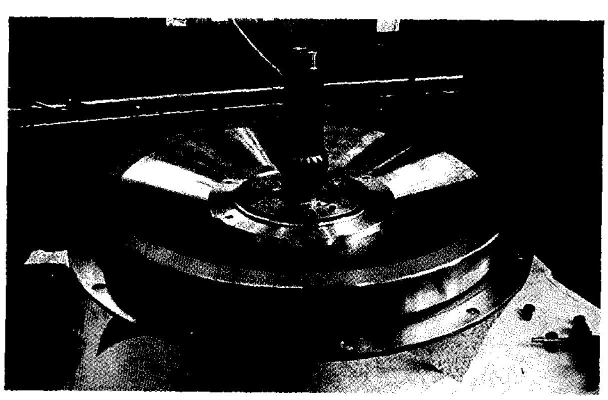

Figure 6. Steel flywheel for ZI-CVT.

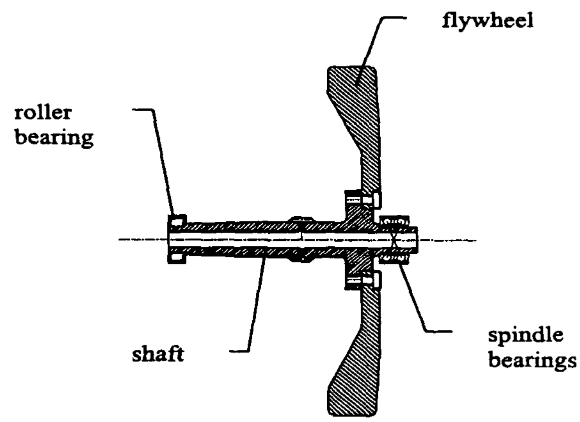

Figure 7. Cross section of flywheel, shaft and bearings.

The pair of spindle bearings was chosen because, under pre-tension, it guarantees stiffness and has no play in radial and axial direction. The size of the bearings is chosen as small as possible, with respect to the life span requirements, in order to reduce the bearing losses. To drive the rotor, the sun gear of the planetary set is milled directly into the shaft.
In the ZI transmission, the flywheel will have to be able to rotate at speeds between 0 and 800 [Rad/s] without resonance appearing. This requires a first eigen-frequency of the construction well above 800 [Rad/s]. In order to meet this demand, a study has been done on the eigenmodes and frequencies of different flywheel constructions in which the construction is simplified to a serial connection of three stiffness: The shaft, bearings and hub stiffness, which will be discussed in this order.
For the analysis of the shaft stiffness, a steel shaft with length l=170 [mm], diameter d=30 [mm] and a rotor with J_r=0.4 [kg.m^2], J_t=0.2 [kg.m^2] and mass m=20 [kg] is assumed. A second order model, as described in (Van Tilborg, 1999) predicts an angular displacement of the rotor at the first, and a vertical displacement at the second eigen- frequency as shown in Figure 8.

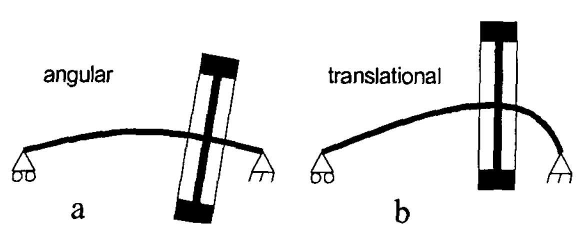

Figure 8. First and second eigenfrequency of flywheel shaft.

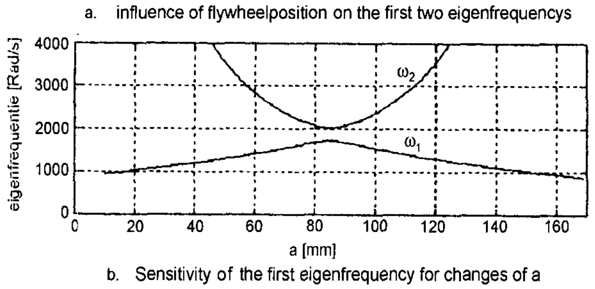

Figure 9. Influence of flywheel position to the first and second eigenfrequency of the flywheel shaft.

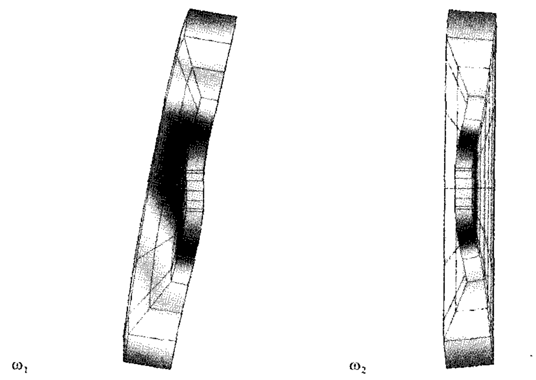

Figure 10. First and second eigenmode of the flywheel hub.

From the eye of the rotor, the two eigenfrequencies can be translated into an angular stiffness $k_{\varphi,\ shaft}$, and a radial stiffness $k_{r,\ shaft}$. With this knowledge the best position of the rotor on the shaft is not obvious. In Figure 9 the first two eigenfrequencies are plotted as a function of the distance, a, between the right bearing and the rotor.

Figure 9 shows that the first eigenfrequency is the highest when the rotor is positioned in the middle of the shaft. Due to the limited available space, the flywheel could not be moved to the middle of the shaft, but is positioned near the right bearing as shown in Figure 7.

Considering the shaft as rigid and the bearings as flexible, one can also distinguish an angular stiffness $k_{\varphi,bear}$, and a radial stiffness $k_{r,bear}$, caused by the bearing stiffness. For this particular construction these are defined by:

$$k_{\varphi,bear} = k_1 \cdot (l-a)^2 + k_2 \cdot a^2 \tag{1}$$

and

$$k_{r,bear} \approx k_2 \tag{2}$$

With k_1 and k_2 the radial stiffness of the left and right bearing. As the hub of this flywheel is relatively thin, it has a non-negligible contribution to the first (angular) eigenfrequency. A FEM analysis calculated for the first two eigenfrequencies: ω_1=2110 [Rad/s] and ω_2=4120 [Rad/s].

The angular stiffness of the hub can then be approximated by:

$$k_{\varphi,hub} = \omega_1^2 J_t \tag{3}$$

Having analyzed the three components separately the total angular stiffness $k_{\varphi,tot}$ is defined by:

$$\frac{1}{k_{\varphi,tot}} = \frac{1}{k_{\varphi,shaft}} + \frac{1}{k_{\varphi,bear}} + \frac{1}{k_{\varphi,hub}} \tag{4}$$

For the construction of flywheel, hub and bearings as shown in Figure 7, the first eigenfrequency is 1100 [Rad/s]. To support the flywheel construction, the bearing on the left is mounted directly into the transmission housing. The spindle bearing is connected to this housing in radial direction by a steel membrane and in axial direction by a steel rod through the flywheel shaft, as shown in Figure 11. In this way the flywheel has a lightweight, stiff support in both directions.

To avoid noise production by the membrane, the pair of spindle bearings is set in a "o"-configuration to provide angular stiffness to the center of the membrane (Van Tilborg, 1999). Figure 12 shows the first eigenmode of the membrane.

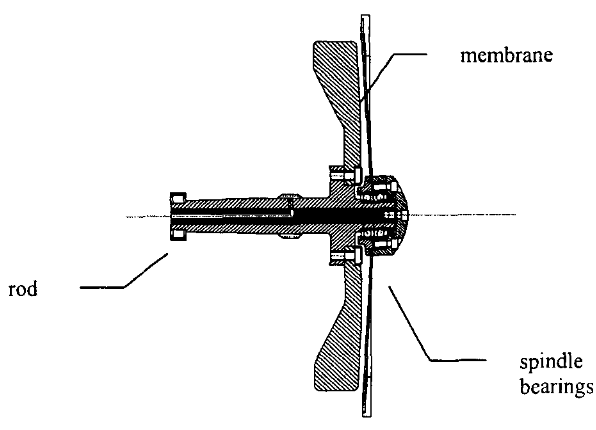

Figure 11. Spindle bearing, radial supported by a membrane and axial supported by a rod.

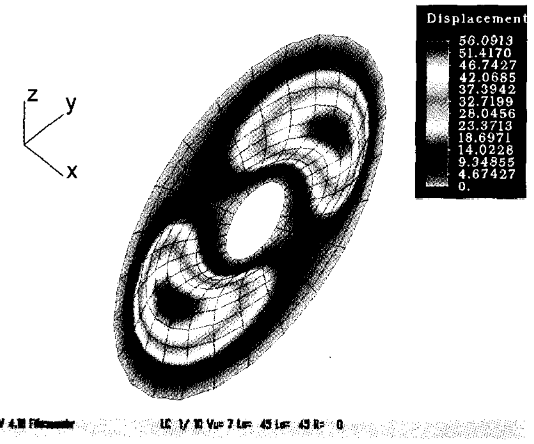

Figure 12. Membrane designed using FEM analysis.

To lubricate the gears and bearings a lubrication path is designed which is fed by the oil channel in the rod. From this part, the oil will be rotated outwards.

5. TRANSMISSION HOUSING DESIGN

For the Proof of Concept ZI-CVT design it is chosen to have a modular construction in which the transmission could be used with or without planetary gear and flywheel. To accomplish this, the assembly is made with two sub-assemblies; a flywheel housing in which the planetary gear set and secondary gear are fixed in axial direction and the flywheel itself with spindle bearings and membrane. Figure 13 shows the two sub-assemblies who can be mounted and removed easily

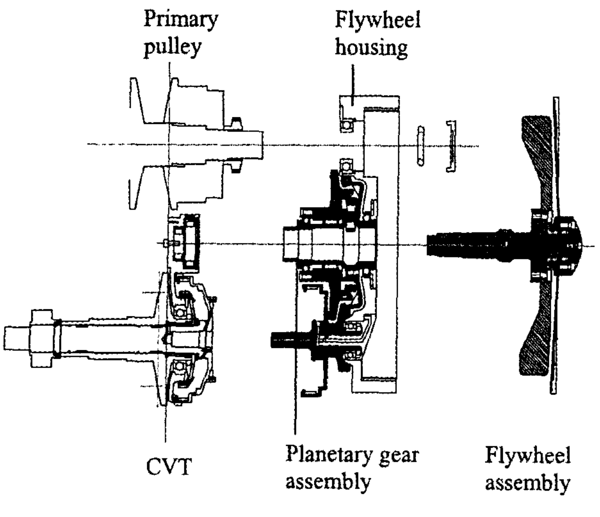

Figure 13. Modular assembly of planetary gear set and flywheel to the CVT.

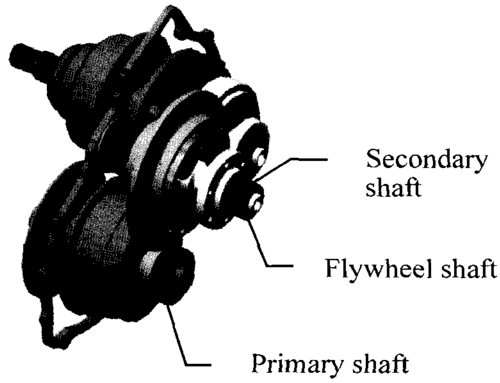

Figure 14. 3D view of primary, secondary and flywheel shaft.

without re-building the standard CVT. This makes it possible to investigate the functioning of the added assemblies separately maintaining the standard CVT behaviour as a reference.

When the flywheel assembly is mounted, the membrane covers the flywheel housing. In Figure 14 it is seen that the flywheel shaft is parallel but not in between the primary and secondary shaft.

The flywheel rotates in it's housing under ambient air pressure. A vacuum pump and a seal to reduce the air density within the housing are taken into consideration. As a result of the low flywheel operation speed (5000 rpm driving at 120 km/h), the decrease in air friction power loss is less than the increase in power loss due to the vacuum pump and the seal. For safety, a ring is mounted around the flywheel hub to secure bearing functioning of the flywheel in case the spindle bearings would fail. Due to the low energy in the flywheel, 1/14th compared to the car energy, minimal safety precautions are needed to avoid that the flywheel could damage the housing.

6. TEST-RIG & CAR IMPLEMENTATION

At first the Proof of Concept ZI transmission is tested separately (without internal combustion engine), on a test-rig. The tests comprise the functioning of the mechanical torque assist, the efficiency of the transmission and the functioning and optimization of the CVT control and driveline management. The second test program comprises the driveability aspects of the ZI-CVT in a mid-class passenger car with 1.6 L engine. Before testing the transmission on the test-rig, the ZI-CVT was built into the car to check the total assembly. In Figure 15 the engine compartment of the test-car is shown with built in CVT and flywheel housing.

CVT housing

1.6 L engine Flywheel housing

Figure 15. Engine compartment of the VW Bora test-car with built in ZI-CVT.

7. CONCLUSION

According to the design objective a proof of concept transmission with mechanical torque assist, called Zero Inertia CVT, is built and under testing. The low speed operated steel flywheel, driven through a planetary gear by the primary and secondary pulley of a conventional CVT, assists the engine up to 40 kW when needed. Proven, low cost transmission technology is used to upgrade the CVT design (only gears, bearings and a steel rotor). The test results, regarding the functionality and the efficiency of the transmission to be obtained before the congress date, will be presented at the congress.

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