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Abstract— **In this article, an initial study of the potential fuel savings of a flywheel hybrid driveline is presented. A systematically developed steering management system controls the driveline in order to obtain the highest possible fuel savings. After explaining the layout and functionality of the flywheel hybrid driveline and the management system, results of numerical simulations are presented showing considerable fuel savings.**

Keywords— **Continuously Variable Transmission (CVT), Flywheel, Fuel Efficiency, Hybrid Driveline, Management System**

I. Introduction

FUEL consumption and exhaust emissions of vehicular systems are subject to global concern. The high levels of exhaust pollution during the late sixties and the oil crisis of the mid seventies initiated the technical search for improving the fuel economy of vehicles. Through the subsequent years, this drive for research has been intensified by environmental protection movements, increasing prices of gasoline and the awareness of limited oil resources.

To decrease the fuel consumption of passenger cars, many improvements in a wide variety of aspects have been made. The aerodynamical drag—generally quantified by the so called C_w-value—is greatly reduced by sophisticated vehicle body designs. The losses due to friction and heat production in transmission units have been reduced by improved gear, bearing and lubrication design. Revolutionary modifications to both diesel and Otto internal combustion (i.c.) engines advanced the fuel economy tremendously. The number of inlet and outlet valves is increased to enhance the distribution of the air/fuel mixture and to improve the removal of the combustion products in the cilinders. This technique both increases the combustion efficiency and decreases the exhaust emissions of the engine. Variable valve timing techniques (dependent on the driving situation) magnify these improvements even more.

Complete new concepts for vehicle propulsion have been merely based on electric components such as electric motors, ultra capacitors and electric accumulators. Electric propulsion of vehicles generates no exhaust emissions and the consumption of fossil fuels is shifted to electric power plants which are cleaner and more efficient per kWh generated energy. During braking of the vehicle, the electric motor can be utilized as a generator such that braking energy is buffered into the batteries instead of being dissipated as friction heat at the mechanical braking system. An unacceptable disadvantage of electric propulsion is the poor driveability as of the limited power exchange of the batteries (electric accumulators) and the limited operating time range of such vehicles. Moreover, recharging of the batteries takes an order of magnitude longer than refueling of an i.c. engine based driveline.

The advantages of electric propulsion and fossil fuel propulsion can be combined in a so-called hybrid driveline. In principle, the driveline consists of a prime mover matching the maximally desired power, *e.g.* an i.c. engine, and an additional energy storage device, *e.g.* an electric motor with batteries, able to generate short term power. The additional energy storage device can improve performance through assist of the i.c. engine, it can be employed to propulse the vehicle at low i.c. engine efficiency, and finally the battery can be recharged during braking or at high i.c. engine efficiencies (high torques).

Instead of electric energy storage, energy can also be conserved as, for instance, kinetic energy or hydraulic energy. The first alternative is generally embodied as a flywheel containing kinetic energy when accelerated to high speeds. The second one as oil reservoirs containing potential energy when put under pressure. In this paper, a hybrid driveline utilizing the i.c. engine as a prime mover and the flywheel as short term energy storage device is subject of investigation.

The advantage of a flywheel is its ability to deliver and absorb high peak powers. Moreover, braking energy can be re-used at a higher efficiency than in the case of electric based hybrids. The latter requires a mechanical/electric conversion and vice versa to make this regenerated energy available at the wheels again. For the flywheel this energy remains purely mechanical.

A disadvantage is that the flywheel is a leaky energy storage device. This "leak" is introduced by mechanical friction at the bearings and, more substantially, air friction at the flywheel surface. Use of modern bearing technology, *e.g.* magnetic bearings, and embedding the flywheel in a vaccuum chamber can greatly reduce this leakage.

In this paper, the potential fuel economy of such a flywheel hybrid driveline in comparison with a conventionally propulsed vehicle is computed by simulating the operation of the driveline along a standardized drive cycle.

The remainder of this paper is organized as follows. Section II explains the layout and functional design of the flywheel hybrid driveline. The management system to generate the control setpoints for the complete driveline is discussed in Section III. The fuel economy greatly depends on the losses in the various system components. Models for these system

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losses are elaborated in Section IV. Having all ingredients to analyse the driveline, in Section V the results of fuel consumption simulations are explained. Finally, in Section VI the paper is concluded with a discussion about the obtained results and future developments.

II. Flywheel Hybrid Driveline

In Figure 1 the flywheel hybrid driveline considered here is pictured.

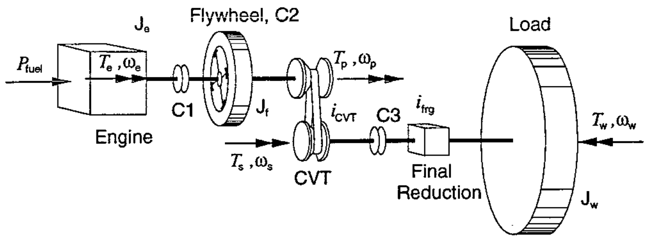

Fig. 1. Layout of the flywheel hybrid driveline

From left to right the i.c. engine is connected to the driveline via the clutch C_1. The driveline has a parallel branch to the flywheel via a clutch C_2 (integrated into the flywheel core). The flywheel system can have an internal gear ratio i_f to increase or decrease the flywheel speed relative to the engine speed. Subsequently, a continuously variable transmission (CVT) connects the flywheel and engine to the wheels (Load) via the clutch C_3 and the final reduction gear with ratio i_{frg}. The engine delivers a torque T_e and rotates at an angular speed ω_e. The CVT has a primary (engine) side and a secundary (wheel) side. In Fig.1, the torques and angular speeds at the primary and secundary sides are represented by T_p, T_s, ω_p and ω_s, respectively. The ratio i_{CVT} of the CVT is continuously variable in time and reduces or expands the secondary speed given the primary speed, according to the kinematic relation:

$$\omega_s(t) = i_{CVT}(t)\, \omega_p(t). \quad (1)$$

At the wheels a torque T_w and angular speed ω_w are defined. The engine, the flywheel and the load respectively have an inertia J_e, J_f and J_w.

After some elaboration on the differential equations for this driveline, the time derivative of the angular wheel speed becomes (independent time variable t omitted and '˙' stands for the first time derivative):

$$\dot{\omega}_w = \frac{i_{CVT}^2 i_{frg} T_e + \dot{i}_{CVT}(J_e + i_f^2\, J_f)\omega_w - i_{CVT}^3 i_{frg}^2 T_w}{i_{CVT}(i_{CVT}^2 i_{frg}^2 J_w + J_e + i_f^2\, J_f)}. \quad (2)$$

For given numerical values of the parameters i_{frg}, i_f J_e J_w and J_f, the external wheel torque T_w, the applied engine torque T_e and an implemented strategy for the CVT ratio i_{CVT}, the angular wheel speed ω_w can in principle be solved from (2).

The external wheel torque T_w generally is a function of air drag and roll resistance and is expressed as:

$$T_w = c_d\, \omega_w^2 + T_r, \quad (3)$$

where c_d is the air drag coefficient and T_r is the constant roll resistance torque at the wheels. Hill-climbing *etc.* is not considered here. Only prescribed patterns of $\omega_w(t)$ (the vehicle speed expressed in radians per sec) are considered. Substituting equation (3) into (2), the desired $\omega_w(t)$ can be realised by appropriately manipulating the engine torque $T_e(t)$ and the CVT ratio $i_{CVT}(t)$.

This system is a redundant one, as two inputs are controlling one output. In fact, the engine and the kinetic energy of the flywheel can in principle both contribute to the desired amount of power at the wheels. In the next section a management system is explained that controls the engine torque, the clutches and the CVT ratio such that this desired power at the wheels is available and such that optimal engine efficiency is guaranteed for a given or desired pattern of the power at the wheels.

III. Management System

A management system for a vehicular driveline has to translate the pedal actuations of the driver into a desired behaviour of the vehicle under given constraints. In Fig. 2. this is explained using blockscheme terminology. The

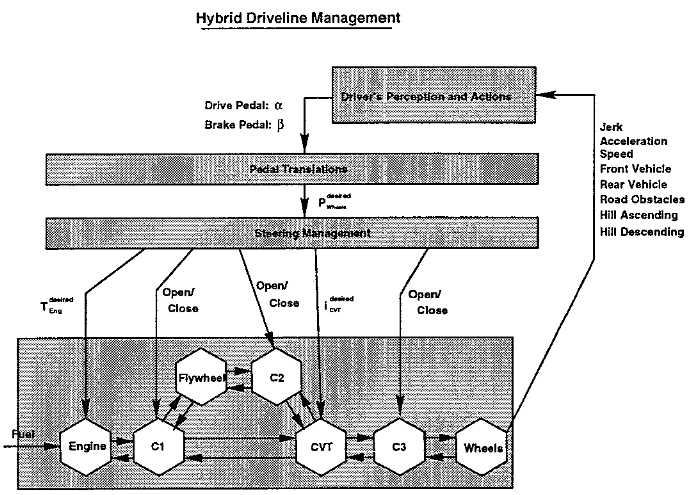

Fig. 2. Management of the flywheel hybrid driveline

driver percepts his environment and reacts on this by actuating the drive pedal α or the brake pedal β. These actions are considered—by the "Pedal Translations"— as a desired *power* at the wheels $P_{wheels}^{desired}$. This desired power serves as an input value for the "Steering Management" block. In this block the setpoints for the local driveline components are generated, aiming at minimization of fuel consumption and maintenance of driveability.

The operation of the clutches brings the driveline in different modes. These modes are listed in the following truth table, where $C_i = 1$ implies a closed clutch and $C_i = 0$ implies an open clutch:

mode	C_1	C_2	C_3
idle 1	0	0	0
idle 2	0	0	1
idle 3	0	1	0
idle 4	1	0	0
flywheel	0	1	1
charging	1	1	0
direct	1	0	1
hybrid	1	1	1

- **idle 1:** The engine and flywheel may be rotating, but there is no connection to the other driveline components and the vehicle
- **idle 2:** Same as idle 1 but with closed C_3
- **idle 3:** Same as idle 1 but with closed C_2
- **idle 4:** Same as idle 1 but with closed C_1
- **flywheel:** The flywheel is solely coupled to the wheels and propulses the vehicle.
- **charging:** The engine is coupled to the flywheel but there is no connection to the wheels. The engine can charge the flywheel at any desired or feasible engine output torque.
- **direct:** The engine is solely coupled to the wheels and propulses the vehicle. The flywheel is not assisting the propulsion.
- **hybrid:** The flywheel and engine together propulse the vehicle. In general the engine can deliver more power than required for vehicle propulsion according to the driver's wishes and in this case it will charge the flywheel to a higher speed.

First, a few assumptions about the engine and flywheel operation will be discussed from which the appropriate management strategy is derived.

Engine

The fuel efficiency (defined as the mass of fuel necessary for 1 kWh of mechanical output energy) of the engine is generally presented in a so-called *engine map.* In such a map, the engine efficiency is plotted as a function of the engine torque T_e and engine speed ω_e. Lines of constant engine power, $P_e = T_e\ \omega_e$, appear as hyperbolic curves in this map. In Figure 3., four of these *iso-power* lines are drawn. In the engine map, a collection of engine operation

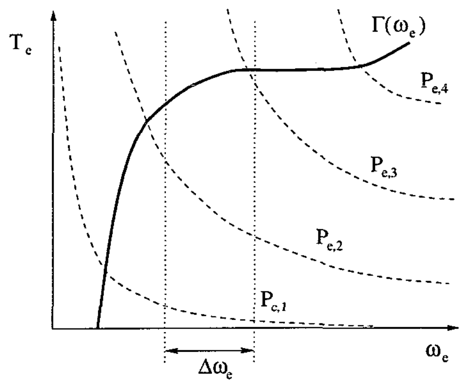

Fig. 3. Optimal efficiency curve in the Engine Map

points $\{(\omega_e,\ T_e)\}$ can be defined in which the engine efficiency is maximal for every feasible engine output power P_e. In Figure 3., the set of these optimal operating points is represented by the line $\Gamma(\omega_e)$. The idea behind nearly every hybrid driveline is to keep the engine at $\Gamma(\omega_e)$ as much as possible. In the present case, this is also persued. The speed interval indicated by $\Delta\,\omega_e$ is the most efficient speed range within the engine map. This range will be addressed further in the sequel.

Flywheel

A flywheel is a kinetic energy buffer. A flywheel, which can either be a disc or a rim, contains energy once it is brought into a rotational movement, say with rotational speed ω_f. The energy contents of the flywheel is then described by

$$E_f = \frac{1}{2}\, J_f\, \omega_f^2, \tag{4}$$

where J_f is the relevant moment of inertia. Thus, the energy contents increases linearly whith J_f, and quadratically with ω_f. It is possible to withdraw power from the flywheel by lowering its speed. This flywheel power is then described by

$$P_f = -J_f\, \omega_f\, \dot{\omega}_f \tag{5}$$

The power withdrawal is feasible only until a certain minimum speed $\omega_f^{\min}$ has reached. Below this speed sustained power withdrawal is nearly impossible because there is simply not enough energy left in the flywheel.

Regarding Figure 1., the flywheel is rigidly connected via the gear ratio i_f to the driveline whenever C_2 is closed. Hence, at the primary side of the CVT, the flywheel forces a rotational speed ω_p. If C_1 is closed, this will also be the case for the engine speed ω_e. For convenience, the flywheel speed and the engine speed are expressed in the primary speed ω_p, *i.e.* $\omega_e := \omega_p$ (if $C_1 = 1$) and $\omega_f := i_f\ \omega_p$ (if $C_2 = 1$).

Fuel Saving

It was perceived that there exists a most optimal operating speed range $\Delta\,\omega_p$ of the engine. The gear ratio i_f is chosen such that $\omega_f^{\min}$ coincides with the beginning of this operating range. If the flywheel needs to be recharged, that is if it reaches $\omega_f^{\min}$, then the engine is started and connected to the driveline via C_1. Along the optimal efficiency curve $\Gamma(\omega_p)$, the flywheel is recharged until $\omega_p = \omega_f^{\min}/i_f + \Delta\omega_p$ is reached. Then the engine is shut off again. This recharging can occur during vehicle propulsion and coasting, (*hybrid* mode) or during standstills (*charging* mode). At sufficient energy contents, the flywheel can solely propulse the vehicle (*flywheel* mode). Intermediate charges of the flywheel can occur once the driver wants to brake the vehicle. The braking power is then guided back into the flywheel (coasting), which will consequently accelerate.

Concluding this subsection it is stated that fuel can be saved by intermittently operating the engine along its most optimal efficiency speed range $\Delta\omega_p$ on the curve Γ and by regenerating braking energy into flywheel energy instead of dissipating it at the braking system. The next subsection explains how the flywheel, charging, direct and hybrid mode can all be casted into one single steering formulation.

Management Strategy

Suppose the driver actuates the drive pedal and brake pedal since he or she desires a (positive or negative) power pattern in time at the wheels,

$$P^{\text{desired}}_{\text{wheels}}(t) := P_{\text{w,d}}(t) = f(\alpha(\tau), \beta(\tau)|\tau \leq t). \qquad (6)$$

The interpretation of the drive and brake pedal actions as a desired power at the wheels is not unique, but provides an intuitively "right" quantity for designing the steering management [2]. The exact formulation of $f(\alpha(\tau), \beta(\tau)|\tau \leq t)$ is still subject of investigation and will not be further discussed here.

Assuming 100 % driveline efficiency it is seen that,

$$P_{\text{w,d}} = T_{\text{p}}\, \omega_{\text{p}} \qquad (7)$$

Moreover,

$$C_3\, T_{\text{p}} = C_1\, T_{\text{e}} - C_2\, J_{\text{f}}\, i_f^2\, \dot{\omega}_{\text{p}}, \qquad (8)$$

where, again, $C_{\text{i}} = \{0, 1\}$.

Only the case $C_1 = C_2 = C_3 = 1$ will be elaborated here as the others can be easily derived from this most general case. Whenever the engine is operating, the optimal fuel efficiency line has to be tracked, thus

$$T_{\text{e}} = \Gamma(\omega_{\text{p}}). \qquad (9)$$

This can in principle be achieved by appropriate control of the engine throttle valve. Substitution of equation (9) in equations (8) and (7) results in the following nonlinear differential equation:

$$\left(\Gamma(\omega_{\text{p,d}}) - J_{\text{f}}\, i_f^2\, \dot{\omega}_{\text{p,d}}\right) \omega_{\text{p,d}} = P_{\text{w,d}}, \qquad (10)$$

from which $\omega_{\text{p,d}}$, the desired primary speed, can be solved provided the initial condition $\omega_{\text{p,0}}$. The desired speed $\omega_{\text{p,d}}$ can be controlled by the CVT according to equation (1). Therefore the desired setpoint for the CVT ratio is

$$i_{\text{CVT,d}}(t) = \frac{\omega_{\text{s}}(t)}{\omega_{\text{p,d}}(t)}, \qquad (11)$$

where $\omega_{\text{s}}(t)$ is the current secundary speed, which is directly related to the current wheel speed through $\omega_{\text{s}}(t) = \omega_{\text{w}}(t)/i_{\text{frg}}$. The steering (9)...(11) serves as a dynamic setpoint generation, which can in general not be followed by the driveline due to time delays, slugishness, flexibilities, component losses and other non-linear driveline behaviour such as the opening and closing of the cluthces. The errors introduced by these phenomena can be minimized by sophisticated component controllers. This is not subject of this paper but will be addressed in future publications.

IV. Component losses

To compute the fuel economy of the hybrid driveline the losses of the various components are of importance. The losses in the flywheel, the clutches, and the CVT appear to be the largest and will therefore be discussed here.

Flywheel losses

The losses in the flywheel are a result of:

- air drag at the total flywheel surface;
- lubricant and roll friction losses in the bearings;
- sealing losses.

Optimization of the flywheel dimensions, using material of a mass density as high as possible (within acceptable costs) [1], results in the following specifications of a disc-shaped flywheel

material	laminated steel
mass density	7800 [kg/m^3]
radius	0.237 [m]
thickness	0.029 [m]
mass	40 [kg]
inertia	1.12 [kgm^2]
minimal speed	400 [rad/s]
maximal speed	650 [rad/s]

The power loss of the flywheel—when put in a vaccuum chamber at 20 [mbar]—as a function of its rotational speed is nearly linear and measures about 130 [W] at 400 [rad/s] and 210 [W] at 650 [rad/s].

Clutch losses

Use is made of oil-drained wet plate clutches. The clutches have a constant loss torque whenever they are in disengaged mode. This torque is introduced by the drag of the oil films between the plates and measures about 0.5 [Nm] for every clutch. The energy loss dissipated as friction heat during the slipping phase of the clutch engagement can be computed to be half of the difference of the energy contents of the flywheel before and after the engagement [4], thus

$$E_{\text{loss,C}} = 0.5\left(\frac{1}{2} J(\omega^2_{\text{before}} - \omega^2_{\text{after}})\right). \qquad (12)$$

CVT losses

The CVT considered is of the transmatic type with a metal pushbelt, see Figure 4. The CVT consists of two pulleys,

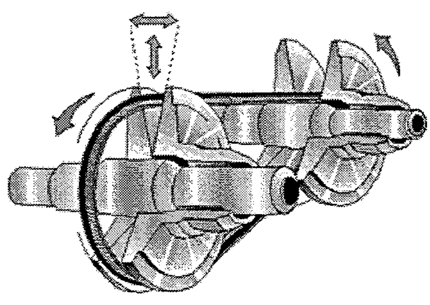

Fig. 4. Metal pushbelt CVT

each with one axially adjustable sheave, a metal pushbelt and a hydraulic actuator system. The metal pushbelt consists of a segment package held together by two sets of rims, one at each side of the segments. The ratio between the two pulleys can be adjusted continuously by guiding the metal belt up and down along the pulley sheaves. This can be established by hydraulically increasing or decreasing the pressure on the sheaves of the rotating pulleys. The CVT ratio i_{CVT} is kinematically limited between 0.4 and 2.5. The losses in the CVT are introduced by:

- mechanical losses due to slip and friction between segments, bands and pulley sheaves
- hydraulic losses in the pump and the hydraulic actuation system

The mechanical losses are discribed as a function of the primary torque T_{p} and the CVT ratio i_{CVT}. The rotational speed is of less influence and is therefore not included in a description of the CVT efficiency. In Figure 5. a surface plot of this efficiency is displayed as a function of T_{p} and i_{CVT}. The efficiency is typically low for high i_{CVT} and low T_{p} [3].

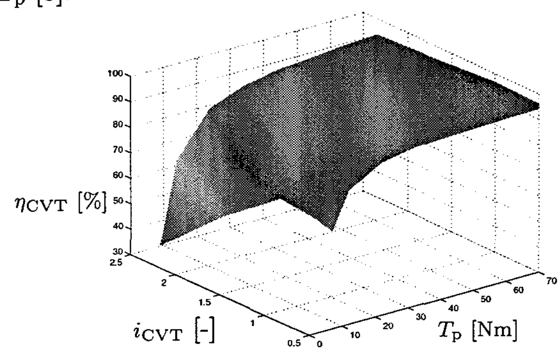

Fig. 5. CVT efficiency surface

The final contribution to CVT losses is made by the hydraulic pump which claims an almost constant torque at the primary pulley of 2 [Nm].

V. Fuel Consumption Simulations

The strategy proposed in the Section III is now applied in a simulation of a stylistic standardized European drive cycle known as the *ECE* drive cycle, [5]. The results are plotted in Figure 6.

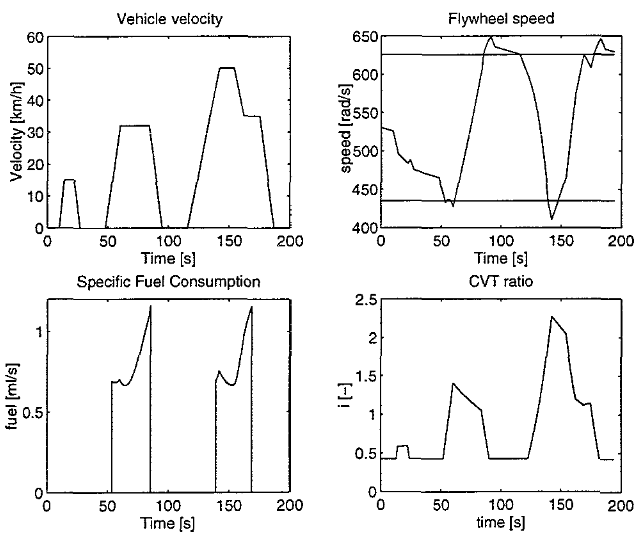

Fig. 6. Results on the ECE cycle

In this figure the upper left plot shows the stylistic ECE cycle as a vehicle speed vs. time plot. The upper right plot shows the flywheel speed as a function of time. It can be seen that the engine has started up two times for recharging the flywheel from its minimum speed (400 [rad/s]) towards its maximum speed (650 [rad/s]). This engine operation can also be derived from the lower left plot in which the fuel consumption as a function of time is displayed. The two intervals in the fuel consumption indicate the two start-ups necessary for recharging the flywheel. The lower right plot shows the CVT-ratio as a function of time that resulted from the strategy given by equation (11).

To validate the effectiveness of the combination of the proposed flywheel hybrid driveline and the management system, the total fuel consumption in the ECE cycle is compared to that of a driveline that is equal to that of Figure 1, but then without the flywheel. The strategy is the same as (9)...(11) with $C_1 = 1$, $C_2 = 0$ and $C_3 = 1$, *i.e.* the direct mode.

It appeared that the flywheel driveline requires 5 [l] gasoline per 100 [km] and the direct driveline 6.7 [l] per 100 [km], both calculated on the ECE drive cycle. This implies that the flywheel hybrid can potentially decrease the fuel consumption up to 25%.

VI. Discussion & Conclusions

It is shown that using a flywheel for short term energy storage can greatly decrease the fuel consumption. This improved fuel economy is caused by utilizing the i.c. engine in its most optimal operation range and by regeneration of braking energy. The optimal operation of the engine is controlled by an appropriate generation of dynamic setpoints for the engine throttle valve and the CVT ratio.

A point of concern is the following. To substract power from the flywheel, its speed has to be decreased. Consequently, the speed of the engine, when connected to the driveline ($C_1 = 1$) will also decrease or at least remain limited between the optimal operation interval. When high acceleration power is desired (high driveability demands), the engine speed will then be too low, and in fact the flywheel has to be decoupled from the driveline to circumvent driveability problems. Because this is not eligible in a control, comfort and fuel economy point of view, a new driveline is currently being developed using an epicyclic gearset with flywheel branched parallel to the CVT. This driveline will partly meet the driveability problem.

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