A High Fidelity Starter Model for Engine Start Simulations

Qi Ma, Sai S V Rajagopalan, Stephen Yurkovich and Yann G Guezennec

Abstract— **Engine start is a very crucial phase in the operation of automotive engines, and the starter motor plays a vital role in this short transient period. However, a complete and exhaustive model of the combined engine-starter system has not appeared to date in the open literature. Although many researchers have modeled the starter system in engine models, the role of the roller one-way clutch is typically omitted. The roller one-way clutch not only protects the starter motor from damage due to high (transient) engine RPM, but it also enables the starter motor to deliver only positive torque to the engine crankshaft. As the trend for model-based calibration grows, the use of computer simulation in engine cold start and crank-to-run transition problems will demand high-fidelity simulations and modeling, which in turn will require the inclusion of an accurately modeled one-way clutch mechanism. In this paper, a novel roller one-way clutch model is developed based on the principles of the one-way clutch. Simulation results are presented for a cylinder pressure resolved engine model and results for starter current characteristics and engine RPM, are verified against actual engine data obtained during the starting phase.**

I. INTRODUCTION

Engine start is the initial phase, and a very crucial phase in the operation of automotive engines. The quality of engine start influences emission and also the drivability of the vehicle. The starter motor plays a vital role in this short transient period. The main function of the starter system is to supply cranking torque to the crankshaft of the I.C. engine until a sustainable RPM is achieved due to consecutive robust engine combustion events. The torque generated by the starter motor is amplified through two or three stages of gear reduction so as to be capable of cranking the engine shaft from zero to around 200 RPM. The cranking torque provided by the starter motor is a linear or quadratic function (depending on the stator winding configuration) of the current flowing through the rotor winding of the starter motor.

In order to protect the starter system from damage due to high (transient) engine RPM, a roller one-way clutch (OWC) is introduced. The internal structure of a production starter system is depicted in Figure 1. As shown, a planetary gear-train is used to amplify the torque generated. In older or heavy-duty applications, a conventional gear reduction scheme may be adopted instead. As high density magnets are readily available today, the Permanent Magnet motor is more common. It makes the starter system compact with very high torque capability.

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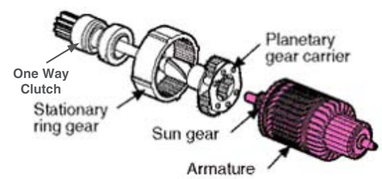

Fig. 1. The components of a starter system (From www.autoshop101.com)

A. Motivation

In recent years, Hardware in the Loop Simulation (HIL) has become the trend in engine cold start control development and calibration process. A higher fidelity engine simulation during crank to run transition is thus indispensable. It requires more accurate starter system models and engine friction models [11]. From the perspective of system identification, the starter motor torque and the engine friction torque are two coupled physical variables during engine start conditions. In order to identify one torque, the other must be known. Relatively speaking, the starter motor torque is easier to obtain due to accessability of measurements. On the other hand, the engine friction torque is more complicated [5] [6] [7]. Both modeling and identification are not always trivial [8] [9] [10]. In addition, the behavior of engine friction may become even more complicated during engine cold start. In order to have a high fidelity crank angle resolved instantaneous friction model, it is imperative to have a very accurate description of the starter motor torque. Based on this guideline, an algorithm described in [1] [2] [3], which derives the coupled engine and starter system friction torque, based on crank angle resolved cylinder pressure torque, inertia torque (reciprocating and rotating) and starter motor torque, is examined on a production 4-cylinder engine (model year 2002) to study the friction torque at engine start conditions.

However, an interesting point is raised: the starter motor cannot be permanently locked with the crankshaft during cranking phase. A simple mathematical deduction can show why this is true. The gear ratio of a production Permanent Magnet Gear Reduction (PMGR) starter system (from starter motor to crankshaft) is about 90. If we assume that the starter system is locked with the engine crankshaft permanently during cranking phase, the inertia of the rotor of the starter motor will be mapped to the engine crankshaft. It results in 90^2 times of the inertia of the starter motor being added to the engine crankshaft. Based on engine data, the total inertia "seen" by the crankshaft is typically about ten times greater. The inertia torque (rotational and reciprocating) can be calculated from engine angular

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acceleration [1] [2] [3]. It turns out that a huge amount of cylinder pressure torque, unfeasible to any automotive engine, is required to rotate such an inertia so as to get the same amount of angular velocity fluctuation as observed on the crankshaft. It is thus suspected that there is intermittent slip between the crank shaft and the starter system during engine cranking and the torque is transmitted not continuously but as short bursts. It can also be inferred that the torque transmitted from the starter system is bounded by an upper limit, which is definitely much higher than the torque actually supplied by the motor.

B. Experimental Investigation

In order to validate the above assumptions, cranking tests were conducted at 90°C, 25°C, 10°C and -20°C. Given the current and voltage measurements from the battery, the speed of the starter motor at the OWC input shaft can be estimated using motor dynamics in the crank angle domain. The estimated OWC input shaft speed vs. the measured OWC output shaft speed at -20°C is shown in Figure 2. The OWC input shaft speed corresponds to ω_1 in Figure 3. It is clear that the starter motor is not locked permanently with the engine crankshaft, the OWC is either in free spinning or slip phase. At hotter temperatures such as 90°C, 25°C and 10°C, the duration of free spinning/slipping phase is even shorter but the trend is quite similar. This could be attributed to lesser energy loss due to friction at higher temperatures.

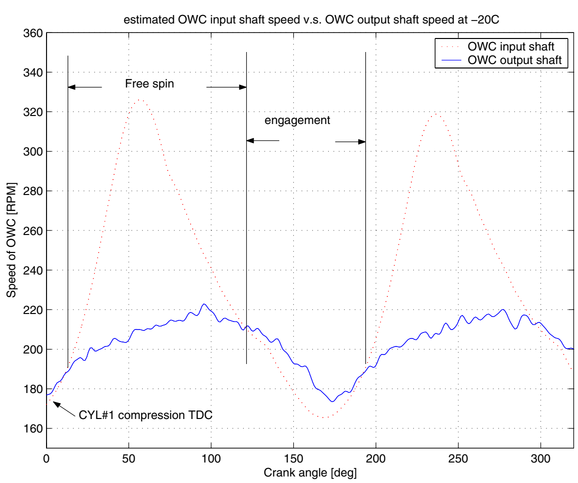

Fig. 2. Estimated OWC input shaft speed v.s. measured OWC output shaft speed at $-20°C$.

Based on the fundamental principle of the operation of the OWC, it becomes very clear that the starter system transmits the cranking torque in a discontinuous fashion. The mechanism of maintaining the engine cranking speed is by periodically releasing the kinetic energy stored in the starter system during free spin (unlocked) to the engine crankshaft during the locking or slipping phase of the OWC. Recent publications [4] [14] shed light on this phenomenon. It indicates that during slipping, a torque much greater than what the starter motor can deliver, is transmitted to the engine crankshaft by the starter system. Based on this indication, a novel model of the starter system, including the role of the roller OWC, is developed.

This paper is organized as follows: the development of the starter system model is presented in Section II; simulation results are presented in Section III; conclusions and future work are described in Section IV.

II. Model

The block diagram of the complete starter system is shown in Figure 3. It consists of four individual sub-models which are: the battery model, the starter current model, the starter angular velocity model and the OWC torque model.

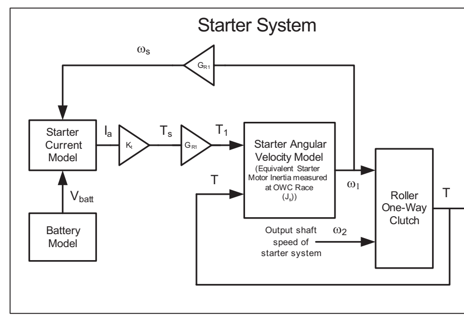

Fig. 3. Block diagram of the starter system

The cross section of a roller OWC is shown in Figure 4. ω_s, ω_1 and ω_2 in Figure 3 and 4, correspond to the speed

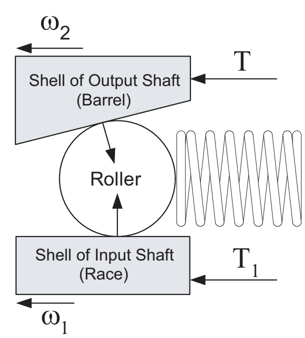

Fig. 4. Cross Section of Roller One-Way Clutch

of the starter motor shaft, the race shaft and the barrel shaft respectively. T_s is the torque measured at the starter motor shaft and T is the torque transmitted to the engine crankshaft. T_1 is the torque at the input shaft of the OWC. G_{r1} is the gear ratio of the internal gear reduction, G_{r2} is the gear ratio of the external gear reduction between the engine flywheel and the starter pinion gear and T_{CRK} is the cranking torque.

A. Starter motor model

Because it is in the most common with passenger cars today, the Permanent Magnet Gear Reduction (PMGR) starter system is mainly discussed here.

A typical PMGR starter system model without considering the OWC is shown in equation set 1.

$$
\begin{aligned}
L\frac{di_a}{dt} &= -R_a i_a - k_e G_{r1}\omega_1 + V_{batt} \\
J_{owc}\frac{\omega_1}{dt} &= G_{r1}T_s - T = G_{r1}k_t i_a - T
\end{aligned} \qquad (1)
$$

where L, R_a, k_e, k_t, J_{owc}, i_a and V_{batt} correspond to the starter rotor inductance, the rotor winding resistance, the back emf constant, the torque constant, the equivalent inertia, the starter motor current and the battery voltage respectively.

B. Roller One-Way Clutch model

The roller OWC is widely used in the starter system due to its compact design and high torque capacity. A free body diagram of the roller OWC is shown in Figure 5. Given a fixed clutch geometry, the torque capacity of a roller OWC can be estimated by Equation 2 [4]. It gives a good estimate of the torque capacity at steady state conditions. However, in transient situations, the OWC is capable of transmitting much higher torque.

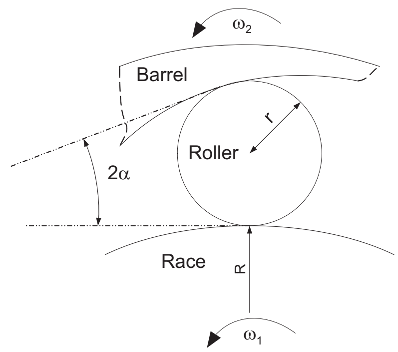

Fig. 5. Free Body Diagram of the Roller One-Way Clutch

$$
T_{maxss} = f(N, L, R, r)\sin(\alpha) \qquad (2)
$$

where $f(N,L,R,r)$ is an algebraic function of design variables such as the number of rollers (N), the length (L) and radius (r) of rollers and the radius (R) of the race.

As illustrated in Figure 2, during torque transmission stage, the two systems are not completely locked with each other. There always exists a velocity difference. For convenience, a new variable $\triangle\omega = \omega_1 - \omega_2$ (the absolute velocity slip) is introduced. The torque transmitted by the roller OWC is given by the following set of equations:

$$
T = \begin{cases} 0 & \text{if } \triangle\omega < 0 \\ T_1 S_1(\triangle\omega) & \text{if } \triangle\omega \geq \triangle\omega_1 \\ T_1 + T_{maxss} S_2(\triangle\omega) & \text{if } \triangle\omega \geq \triangle\omega_2 \end{cases}
$$

where $S_1(\cdot)$ and $S_2(\cdot)$ are two monotonically increasing saturation functions; and their range is between zero and one. $\triangle\omega_{s1}$ and $\triangle\omega_{s2}$ are design parameters and are always positive. Also, $\triangle\omega_1$ is always less than $\triangle\omega_2$. T_1 is the input torque to the race of the OWC. The behavior of the torque transmitted by the OWC is depicted in Figure 6. When $\triangle\omega_1$ and $\triangle\omega_2$ are very small compared to fluctuations in the engine RPM, the variation in the selection of their values will not affect the simulation results.

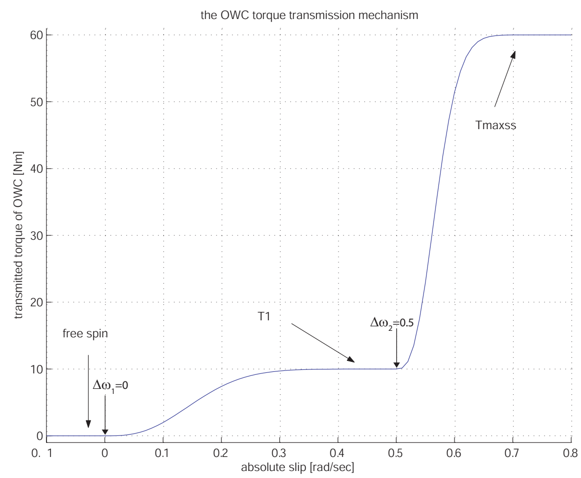

Fig. 6. Torque transmission mechanism of the OWC when $T_1 = 10Nm$

Of course, there are many choices when selecting the saturation function. For simplicity, the Maxwell probability function (a continuous function) is selected. The reasons behind this kind of a formulation are:

- The continuous saturation function captures the fact that the torque transmitted by the OWC is directly related to the deformation of the barrel cage which in turn is affected by $\triangle\omega$.
- The stepwise hard saturation function wholly approximates the torque transmission mechanism, but creates numerical problems in computer simulation.

C. Stability Analysis

The torque transmitted by the OWC is chosen to vary based on the relative velocity between the input and output shafts as follows:

- When $\triangle\omega \gg \triangle\omega_2$, the torque transmitted by the OWC is T_{maxss}.
- When $\triangle\omega_1 < \triangle\omega < \triangle\omega_2$, the torque transmitted by the OWC equals the input torque i.e. $T_1 = G_{gr1}T_s$.

The power transmission ratio of the OWC is:

$$
P_R = \frac{P_2}{P_1} = \frac{T\omega_2}{T\omega_1} = \frac{\omega_2}{\omega_1} < 1
$$

This formulation for the torque transmission mechanism guarantees that the energy is always dissipative. Energy loss may be through heat dissipation due to metal shear. For simplicity, we neglect the potential energy stored as inertia in the OWC.

III. Simulation Results

The simulation is tailored so as to match with the test data obtained from a real engine. In order to avoid using the friction model which is currently under development, the friction torque is derived from real test data.

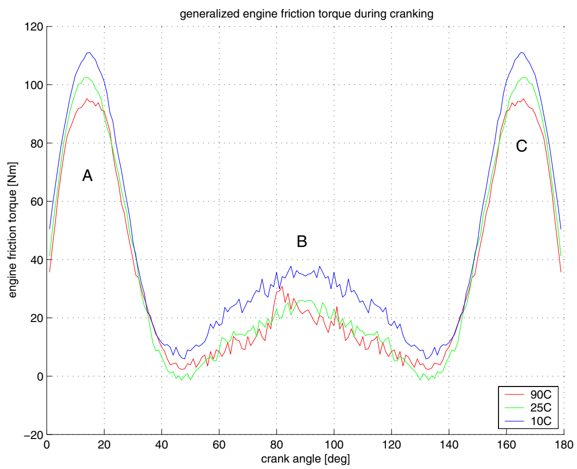

Fig. 7. Engine friction torque during cranking at different temperatures

A. Engine Friction Setup

The friction used in the computer simulation is derived from the work done by [1] [3] [2]. Although the estimated $T_f - T_s$ conveys only partial information about the true engine friction during cranking phase, the missing information can be recovered from basic engine friction theory. In general, within two consecutive compression strokes of two adjacent (firing order) cylinders of a typical four cylinder engine, the engine friction torque has a shape as shown in Figure 7 [1].

The positions "A" and "C" correspond to expansion and compression strokes of two adjacent cylinders [10]. The location and magnitude of the peak friction torque reflects the contribution by the crank pin-bearing friction [6]. At position "B", the friction torque is dominated by piston-cylinder wall friction because this directly depends on the engine RPM. The "W" shaped friction torque may not necessarily be symmetric about the middle peak, especially for the friction at position "B". During engine firing, the friction torque at position "A" (as seen from real engine data) can be tripled or even greater. But the "W" shape is preserved. The detailed development of an engine friction model will appear in an upcoming publication, but is beyond the scope of this paper. For fast simulation techniques such as HIL, it is sufficient just to use the generalized engine friction as shown in Figure 7.

B. Results

The simulation results against experimental data at 90°C 10°C are shown in Figure 8 and 9. At all the temperatures, the RPM trace seems to follow the measured RPM quite well. The measured RPM starts from a nonzero value because the data acquisition system is initialized only after a small angle of rotation of the crank shaft. The mismatch in the RPM traces is due to the fact that the simulation model does not incorporate the reciprocating inertia torque because it does not contribute too much to the scope of our work, but only increases the simulation overhead. For HIL simulation at cranking speed, its effect can be neglected [12]. It can be inferred from the RPM plot that the concept of discontinuous slip of the starter system and crank shaft, may indeed be true.

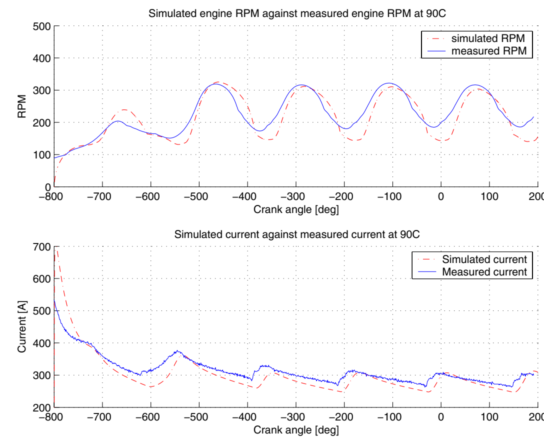

Fig. 8. Comparison of engine RPM, Input and output torques of the OWC and Starter current at 90°C

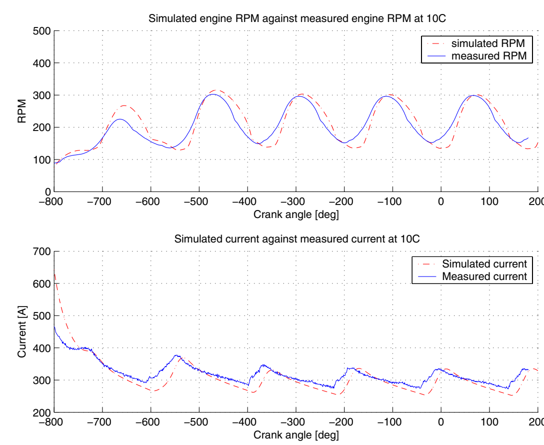

Fig. 9. Comparison of engine RPM, Input and output torques of the OWC and Starter current at 10°C

The torque transmission mechanism is depicted in figures 10and 11. As experienced in the computer simulation, a non-smooth torque transmission model of the OWC causes an oscillating effect on the torque output T. This observation may be influenced by the solver for the differential equations or by the true nature of the OWC [13]. The current traces too seem to match exceptionally well. As the starter current governs the torque output, the result again vindicates the concept of discontinuous engagement and torque boost.

IV. CONCLUSION AND FUTURE WORK

A simple model of the roller OWC together with an effective torque transmission mechanism thus completes the starter system model. It can be seen that the simulation results match real engine data sufficiently well. Although this

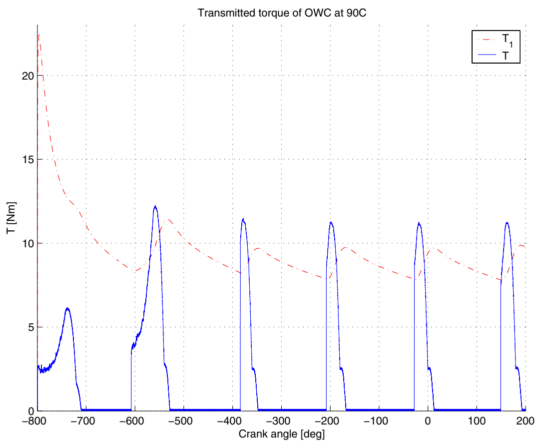

Fig. 10. Input and output torques of the OWC at 90°C

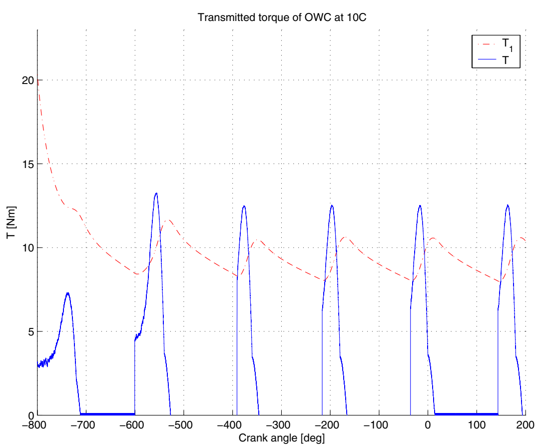

Fig. 11. Input and output torques of the OWC at 10°C

simulation and its result seem to follow the expected trend, the speed of the starter motor has not been measured and thus a thorough comparison was not carried out. Comparing the simulation results to true value of the starter speed will further strengthen the work done so far and will contribute towards future work in the field.

Accurate modeling of the engine friction for simulation is essential because of the popularity of model based calibration in the automotive industry. Ideally, the engine friction torque can be obtained if the motor acceleration is known. As a result, the torque transmitted by the OWC can be represented by the following equation.

$$T \approx G_{r1} k_t i_a - J_{owc} \frac{\triangle \omega_1}{\triangle \theta_1} \omega_1$$

The engine friction torque comprising the torque transmission logic explained above can be calculated as follows.

$$T_f = T_{cp} - T_{rot} - T_{rec} + G_{r2} T$$

In this paper, the starter motor current information is used to derive the starter motor speed. Due to instability issues associated with numerical differentiation, we cannot obtain a reliable angular acceleration of the starter motor. In order to confirm the concept of torque boosting of OWC during slip engagement, speed measurement of the starter motor shaft is needed. In addition, because of the dependence of the simulation results on accurate engine friction information, a detailed engine friction model is desired, which is not discussed in this paper. Future work will elaborate on the detailed friction model.

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