

System Design and Parameter Matching for the New Generation of Hybrid Electric City Bus

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ABSTRACT

In this paper, based on the research of the first generation of a hybrid electric city bus, principle researches have been offered on the parameters matching of the power-train and main components for the new generation of hybrid electric city bus. The vehicle system was designed integrated, and then simulation platform was developed and established. The dynamic and economy performance can be predicted based on the above work, results show that the designed hybrid electric city bus can meet the performance demands and validate rationality of the parameters matching.

INTRODUCTION

The technology of HEVs will become the main development as well as the research hotspot of the vehicles in the next several decades because of its advantages[1,2,8,9] over the traditional vehicles on the energy economy and low emissions and also its advantages over the EVs whose development is limited by the bottle-neck on the battery technology. HEVs have been under research in and out abroad for many years and the technology on it has been improved continuously[3-5]. To deal with this challenge, various domestic institutions and automobile manufactures have been paying great attention to it gradually and thus raise the high tide of research in HEVs, for example, the Technical Center of FAW has developed its first generation city hybrid and they will focus on the integration and optimization design of the power assembly and system further. This paper focuses on the research in the matching of the parameters and system design for the whole vehicle of the next generation city hybrid bus, and thus provides proper evidence to the basic theory for the product size in the corporation.

ANALYSIS OF CONFIGURATION SCHEME

CONFIGURATION LAYOUT

While the 1st generation parallel hybrid bus was configured with the scheme that contains two axles and one motor shown in Figure.1, the 2nd generation parallel hybrid bus was changed to the configuration scheme with a single axle and two motors. The hybrid configuration is shown in Figure.2. One motor is linked to the input axle of the gearbox and the other is linked to the output axles of the gearbox separately

The other optional configuration layout is shown in Figure.3 where only one motor was used and was linked to the output axle of the gearbox (II axle). The advantage of this scheme over the others is that the power isn't

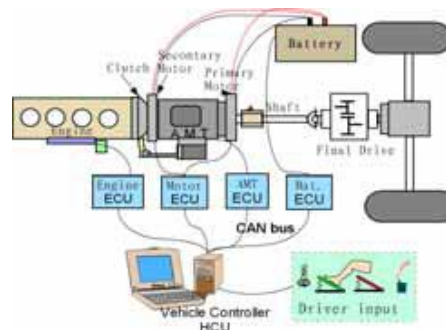


Figure 1 Configuration layout for the 2nd generation parallel hybrid bus

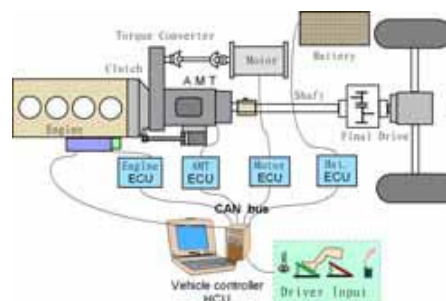


Figure 2 Configuration layout for the 1st generation parallel hybrid bus

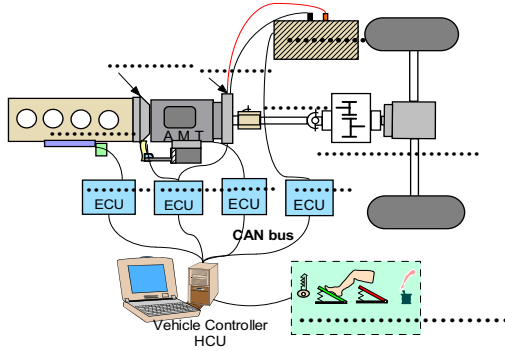


Figure 3 Other optional configuration layout

interrupted during AMT shifting, and the hybrid bus can be driven only by the batteries.

COMPARISON OF CONFIGURATION LAYOUTS

With comparison between the configuration layout for the 2nd generation parallel hybrid bus and that for the 1st generation parallel hybrid bus, the advantages of the former can be explained on several aspects:

- With the same power, it is comparatively easier to layout the motors on the bus because the size of the motors will decrease with two separate motors. And because the two motors will link to the gearbox separately, the torque converter can be omitted and thus the more space will be saved
- The configuration scheme with the single motor to the input axle increases the moment of inertia of the input axle of AMT, which can lead to the difficult AMT shifting. Even though the problem can be solved by reducing the shifting time with the isochronous shifting, the large current impact will be caused and the shifting of AMT will be further influenced. The choice of small motors in the new scheme not only causes little impact and influence on the shifting of AMT and increases the stability, but also ensures to prevent the power from being interrupted in the process of shifting and thus improves the dynamic performance of the bus.
- With motors coupled with the input axle and the output axle, there are two major technical advantages. On the one side, not only it can save the fuel by idling, but also it brings little adverse influence to the overmuch shifting of the AMT, because it reserves the start-stop function by the motor coupling with the input axle. On the other side, the motor coupling with the output axle ensures the power to go continuously without being interrupted when shifting in order to improve the dynamic performance. In addition, it possesses several advantages such as high recycling efficiency of the braking energy because the motor is fairly close to the driving wheels.

POWER MATCHING OF DYNAMICAL SOURCES

Power demand is determined according to the dynamic performance [6,7] (the max speed, the max acceleration and the max up grade) for the most vehicles, and the max power of the dynamic source is usually determined by the accelerating performance. In other words, as long as the accelerating performance demand is met, the other dynamic requirements will also be met. A new design method is introduced by which the overall power of dynamic sources can be determined from the accelerating target directly.

The dynamic performance target for a hybrid vehicle includes the max speed $v_{\max}$ (km/h), $v_0 - v_t$ (km/h) accelerating time and the max up grade $i_{\max}$ (%).

First of all, the max power $P_{\max 1}$ can be determined by the max speed $i_{\max}$ as follows:

$$P_{\max 1} = \frac{v_{\max}}{3600 \cdot \eta_t} (m \cdot g \cdot f + \frac{C_D \cdot A \cdot v_{\max}^2}{21.15})$$

Where η_t is the transmission efficiency of the power.

$$\eta_t = \eta_{cl} \cdot \eta_{tc} \cdot \eta_{gb} \cdot \eta_{fd};$$

Where η_{cl} is the efficiency of the clutch; η_{tc} the efficiency of the torque converter; η_{gb} the efficiency of gearbox; η_{fd} the efficiency of rear bridge; f the rolling resistance coefficient.

Secondly, the max power is determined according to the climbing performance:

$$P_{\max 2} = \frac{v_i}{3600 \cdot \eta_t} (m \cdot g \cdot f \cdot \cos \alpha_{\max} + m \cdot g \cdot \sin \alpha_{\max} + \frac{C_D \cdot A \cdot v_i^2}{21.15})$$

Where $\alpha_{\max} = \tan^{-1}(i_{\max})$.

The vehicle velocity during the starting and accelerating process can be expressed as follows:

$$v = v_m \left(\frac{t}{t_m} \right)^x$$

Where x is the coefficient which is 0.5 usually.

t_m, v_m is the acceleration time and the terminal velocity.

Given the assumption that the bus is accelerated on the smooth road, according to the dynamic function in