

Establishment of Fuel Economy Estimation Method Focused on Transmission Efficiency of Rubber Belt Type CVT

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ABSTRACT

A simulation tool has been developed that can be used to estimate a fuel economy while driving in a mode test of a motorcycle equipped with a continuously variable transmission (CVT) at an early stage of development. For a precise estimation of a mode fuel economy, it is necessary to accurately estimate the CVT ratio, the engine speed, and the crankshaft torque during driving in a mode. To achieve this, this study has generalized the transmission efficiency of a CVT system. This study has also derived developed balance equations that can take into account the transmission efficiency of CVT and the slippage that occurs when the centrifugal clutch is about to be engaged. In the proposed method, the pulley ratio of CVT, the engine speed, and the torque at the crankshaft were obtained first by solving the developed balance equations at discrete times during driving in a mode. Then, the results were checked against a predefined fuel consumption map, which was based on actual measurements, to obtain the fuel consumption at discrete times. Lastly, the mode fuel economy was calculated through integration over time of the fuel consumption data. Comparison of the calculated result with the actual measurement showed that the difference between the two was within 3.5%. This method has been found useful as it enables accurate estimation of the fuel economy of CVT-equipped motorcycles at an early stage of development.

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INTRODUCTION

In recent years there has been a strong call for a reduction in carbon dioxide emissions as one of the measures to prevent global warming, and there is also an ongoing requirement for motorcycles to meet the increasing need for better fuel economy. The need for lower fuel consumption is particularly high in developing countries where the sales volume of motorcycles equipped with continuously variable transmission (CVT) is increasing. To realize an efficient development of a motorcycle with a good fuel economy, it is necessary to be able to accurately estimate the fuel economy at an early stage of vehicle development before a prototype model is built.

One of the indicators used to evaluate the fuel economy performance of vehicles is the mode fuel economy. In estimating the mode fuel economy, it is important to accurately estimate the pulley ratio of CVT, the engine speed, and the crankshaft torque (hereinafter referred to as "ratio," " N_e ," and " T_r ," respectively) during driving in a mode. In the case of manual transmission (MT) motorcycles, the ratio during driving in a mode is determined by the gear shift pattern of the mode. On the other hand, in the case of motorcycles with belt type CVTs, in the process of delivering driving force from the drive pulley to the driven pulley, the pulley ratio is continuously changed depending on the equilibrium between a thrust force applied to the drive pulley and that to the driven pulley, which is determined by a running resistance. While the way in which these parameters change

differs depending on the specifications for the vehicle and CVT system, there is a typical method to calculate the ratio, N_e and T_r , which involves solving the following three relationships:

- Thrust forces on drive and driven pulleys follow the equation proposed by Ogasawara [1]
- Converted driving force on the rear wheel calculated by T_r = Running resistance defined in a mode
- Converted vehicle velocity from N_e = Vehicle velocity defined in a mode

Using this method, the ratio, N_e and T_r can be obtained at discrete times during driving in a mode. Also, the fuel consumption (hereinafter referred to as "FC") is measured beforehand for various N_e and T_r combinations to create a fuel consumption map (hereinafter referred to as "FC map"). An FC value at each discrete point of time that corresponds to the values of N_e and T_r is picked up from FC map. The picked up values for individual time are integrated to yield the total amount of fuel in the condition of traveling under the mode.

However, when this method was used to estimate the mode fuel economy, it often resulted in a significant difference of as much as 15% between the calculated and measured values. After investigating the cause of this difference, it was found that the transmission efficiency of CVT and the effect of the clutch slippage that occurs when the centrifugal clutch is about to be engaged had a large effect.

considered. As such, it is now clear that the issue that needs to be solved in order to accurately estimate the mode fuel economy is how to include these two factors in the calculation.

Regarding the first issue, or transmission efficiency, there is a case study where the mapping of the actual measured variable speed characteristics has been incorporated in the calculation. As the mapping of variable speed characteristics that is based on actual measurements includes the effect of transmission efficiency, the accuracy of the calculation should be improved. However, the issue with this approach is that the variable speed characteristics can be measured and used in the estimation of mode fuel economy only in the later phase of vehicle development, after the specifications for the vehicle have been finalized and a prototype vehicle has become available for testing. Furthermore, the variable speed characteristics must be measured each time the CVT specifications are revised. To solve these issues, it is necessary to develop a method that can generalize the transmission efficiency for incorporation into the conventional calculation method. Therefore, after identifying the factors that affect the transmission efficiency, we formed a generalized expression of the transmission efficiency.

Regarding the second issue of how to incorporate the slippage effect of the centrifugal clutch, there is a case study where the effect of engaging and disengaging the clutch has been considered [4]. However, clutch slippage was not taken into account. Accordingly, the accuracy of the calculation is deteriorated when there is a slipping at the clutch that frequently occurs at a starting of a vehicle. The torque that is transmitted in a centrifugal clutch at each point of time, which was already formulated in generalized formula, was utilized to solve this issue.

By incorporating these results into the previously mentioned three balance equations, a developed calculation method was established. As a result, the mode fuel economy was successfully estimated in an early stage of development with a level of accuracy that is suitable for practical use. This paper reports on the proposed calculation method and the adequacy of the results.

CONFIGURATION OF BELT TYPE CVT SYSTEM FOR MOTORCYCLES

Figure 1 shows the target CVT system of this study, which is a typical rubber belt type CVT system for motorcycles. The power of the crankshaft is transferred from the drive pulley on the crankshaft to the driven pulley via the belt, and it is further transferred to the rear wheel after the speed is reduced in the reduction gear. Here, the CVT system is realized by automatically changing the wound radius of the belt according to the running resistance of the vehicle as the thrust force on the drive pulley that presses against the sides of the belt balances with the similar thrust force on the driven pulley to determine the balancing point. This type of mechanical CVT system is widely used in motorcycles, all-terrain vehicles, farm machines, and so on, due to its simple mechanism, ease of manufacture, and low cost.

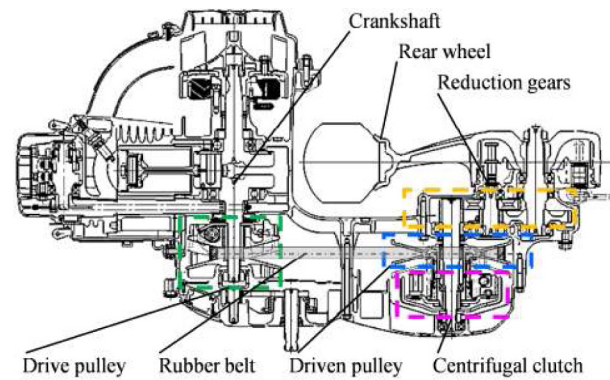


Figure 1. Configuration of typical CVT with rubber V-belt for motorcycles

Figure 2 is a schematic depiction of the CVT system that shows the relationship between the drive and driven pulleys and the belt. The CVT pulley ratio is given in Equation (1) based on geometry using the circumference of the belt, the distance between the shafts, and the wound angle of the belt.

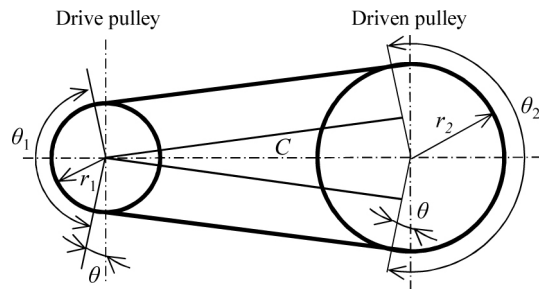


Figure 2. Schematic drawing of CVT system

$$i = 1 + \frac{2}{\pi \left\{ \frac{L_p/2C - \cos\theta}{\sin\theta} - \theta \right\} - 1} \quad (1)$$

Where,

i = Pulley ratio of CVT

C = Distance between drive and driven shafts

L_p = Belt pitch perimeter

r_1 = Belt wound radius on drive pulley

r_2 = Belt wound radius on driven pulley

θ_1 = Belt wound angle on drive pulley

θ_2 = Belt wound angle on driven pulley

$\theta = |(180\text{deg}-\theta_1)/2| = |(180\text{deg}-\theta_2)/2|$

Figure 3 shows a simplified depiction of the centrifugal governor mechanism on the drive side. The movable drive pulley on the drive side can slide in the axial direction. Similarly, the ramp plate rotates together with the crankshaft and transfers the rotation to the movable drive pulley. Accordingly, the movable drive pulley also rotates

together with the crankshaft at the same rotation speed. The weight roller positioned between the movable drive pulley and the ramp plate also rotates at the same speed as the crankshaft to generate the centrifugal force F_{wt} , as given in Equation (2). This centrifugal force is transferred via the movable drive pulley to the side of the belt as thrust force Q_1 , as given in Equation (3).

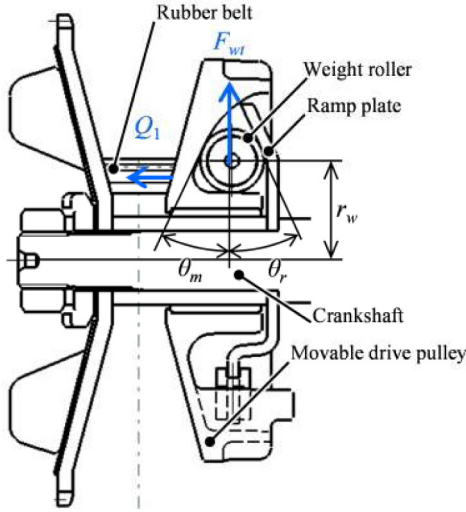


Figure 3. Mechanism of governor using centrifugal force installed on drive side pulley

$$F_{wt} = W_t \cdot r_w \cdot \left(\frac{2\pi}{60} \cdot N_e \right)^2 = \frac{W_t \cdot r_w \cdot \pi^2}{900} N_e^2 \quad (2)$$

$$Q_1 = \frac{F_{wt}}{(\tan\theta_m + \tan\theta_r)} = \frac{W_t \cdot r_w \cdot \pi^2}{900(\tan\theta_m + \tan\theta_r)} N_e^2 \quad (3)$$

Where,

N_e = Engine speed [r/min]

F_{wt} = Total of centrifugal forces on weight rollers

Q_1 = Thrust force of the centrifugal governor on the drive side

W_t = Total weight of weight rollers

θ_m = Angle of contact between weight roller and movable drive pulley

θ_r = Angle of contact between weight roller and ramp plate

r_w = Turning radius of weight roller

On the driven shaft, there is a torque cam mechanism that generates a thrust in proportion to the drive torque and a compression spring that generates a thrust in proportion to the stroke of the movable driven pulley. The components around the driven shaft are shown in Figure 4, and a partial cross-section of Figure 4 is illustrated in Figure 5. The torque cam thrust is determined by the drive torque, the torque cam angle of the torque cam slot in which the torque cam pin moves, and the torque cam radius. The torque cam thrust Q_c is given by Equation (4).

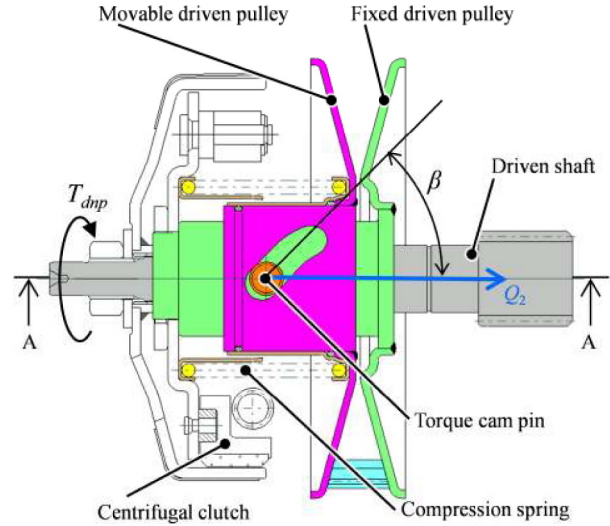


Figure 4. Components configured around driven shaft

$$Q_c = \frac{1}{2r_c} T_{dnp} \cdot \tan\beta \quad (4)$$

Where,

Q_c = Torque cam thrust

T_{dnp} = Torque on driven pulley

β = Torque cam angle

r_c = Torque cam radius

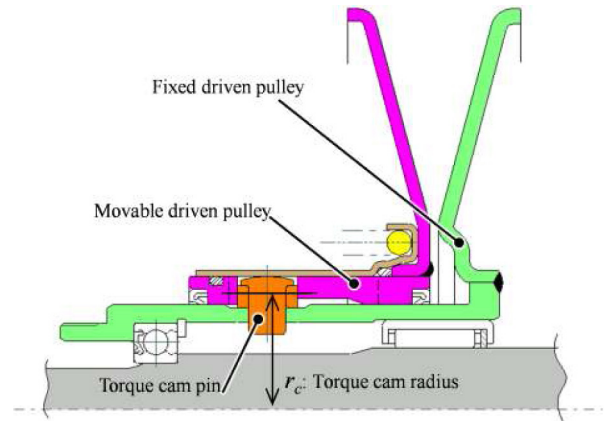


Figure 5. Cross section of driven pulley (section AA on Figure 4)

The thrust force of the compression spring is given by Equation (5).

$$Q_s = k \times \delta = k(l_f - l_i) \quad (5)$$

Where,

Q_s = Thrust force of the spring

k = Spring constant

δ = Spring stroke

l_f = Free length of the spring

l_i = Spring length in operation

Q_2 = Thrust force on the driven pulley

Using the above Equations (4) and (5), the thrust on the driven side Q_2 can be expressed by Equation (6).

$$Q_2 = Q_s + Q_c = k(l_f - l_i) + \frac{1}{2r_c} T_{dnp} \cdot \tan \beta \quad (6)$$

CALCULATION OF RATIO, N_e , T_r , AND RESULTING FUEL CONSUMPTION USING CONVENTIONAL METHOD

To obtain the ratio, N_e , and T_r values during a driving in a mode that constantly varies according to the equilibrium between the running resistance and the thrust force, the following three relationships expressions (i.e., the conventional formula) were solved:

- Thrust forces on drive and driven pulleys follow the equation proposed by Ogasawara [1]
- Converted driving force on the rear wheel calculated by T_r = Running resistance defined in a mode
- Converted vehicle velocity from N_e = Vehicle velocity defined in a mode

Relationships between Thrust Forces on Drive and Driven Sides of CVT

According to [1], the drive thrust Q_1 and driven thrust Q_2 can be expressed by Equations (7) and (8), respectively, based on Hooke's law for shearing stress, on small segments of the V belt and the force balance in the radial direction of the pulleys. These shows different representations of the drive thrust Q_1 in Equation (3) and driven thrust Q_2 in Equation (6). In Figure 6 and 7, the forces applied to the V-belt, which are involved in the derivations of Equation (7) and (8), are shown. The cross sectional diagram of the V-belt is shown in Figure 8.

$$Q_1 = \frac{\theta_1 \cdot \cot(\alpha + \rho)}{2} \left\{ (T_1 - \rho AV^2) - \frac{1}{2} T_b \frac{\tanh(\lambda \cdot R_1 \cdot \theta_1)}{\lambda \cdot R_1 \cdot \theta_1} \right\} \quad (7)$$

$$Q_2 = \frac{\theta_2 \cdot \cot(\alpha + \rho)}{2} \left\{ (T_2 - \rho AV^2) - \frac{1}{2} T_b \frac{\tanh(\lambda \cdot R_2 \cdot \theta_2)}{\lambda \cdot R_2 \cdot \theta_2} \right\} \quad (8)$$

Where, α = Belt pinching half angle

ρ = Belt friction angle

λ = Characteristic value of belt

l_f = Free length of the spring

T_1 = Belt tension on tight side

T_2 = Belt tension on slack side

T_b = Effective tension of V-belt

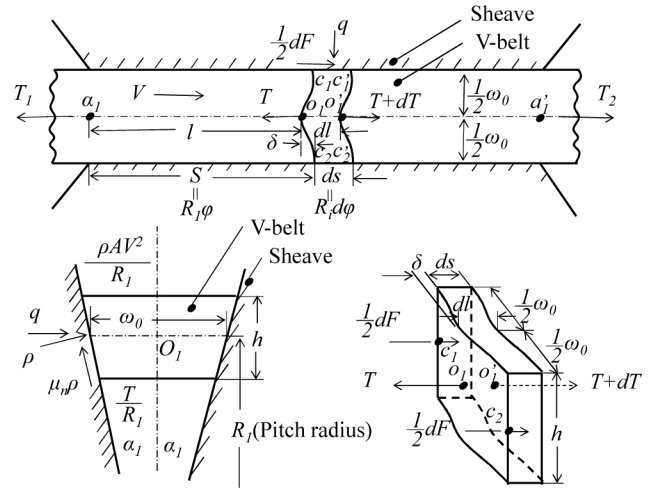


Figure 6. Forces applied to V-belt

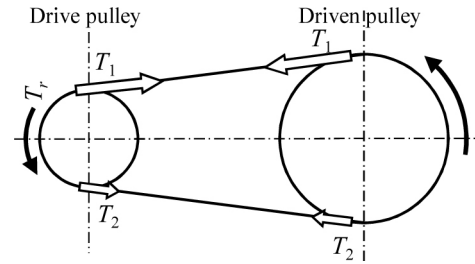


Figure 7. Biaxial model of belt driving system

The characteristic value λ of the belt is a coefficient determined by the stiffness and shape of the belt, and is given by Equation (9). The angle of friction ρ of the belt is a coefficient for the coefficient of friction μ , and is given by Equation (10).

$$\lambda = \sqrt{\frac{h_v G}{\omega A_b E}} \quad (9)$$

Where, h_v = Height of V-belt

G = Modulus of transverse elasticity

ω = Width at the center of cross section

l_f = Free length of the spring

A_b = Cross section area of belt

E = Longitudinal elastic modulus

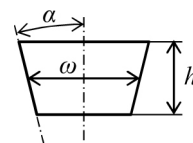


Figure 8. Cross-sectional diagram of V-belt

$$\rho = \tan^{-1} \mu \quad (10)$$

Where, μ = Coefficient of static friction between belt and pulley

Using the relationship in Equation (11) between the tight side tension T_1 , slack side tension T_2 , and effective tension T_b , Equations (7) and (8) can be combined into Equation (12).

$$T_b = T_1 - T_2 \quad (11)$$

$$Q_1 = \frac{Tr \cdot \theta_1}{2r_1 \cdot \tan(\alpha + \rho)} \left[1 - \frac{1}{2} \left(\frac{\tanh(\lambda \cdot r_1 \cdot \theta_1)}{\lambda \cdot r_1 \cdot \theta_1} + \frac{\tanh(\lambda \cdot r_2 \cdot \theta_2)}{\lambda \cdot r_2 \cdot \theta_2} \right) \right] + Q_2 \cdot \frac{\theta_1}{\theta_2} \quad (12)$$

Equation (12) shows the thrust balance between the drive and driven pulleys via the belt. The torque applied on the driven pulley T_{dnp} during driving in a mode can be given by Equation (13) using the running resistance as follows:

$$T_{dnp} = \frac{R \cdot r_T}{i_f \cdot \eta_f} \quad (13)$$

Where, R = Running resistance

i_f = Final reduction ratio

η_f = Efficiency of final reduction gear

r_T = Radius of rear tire

By substituting this in Equation (6), the thrust Q_2 on the driven side is given by Equation (14).

$$Q_2 = Q_s + Q_c = k(l_f - l_i) + \frac{1}{2r_c} \cdot \frac{R \cdot r_T}{i_f \cdot \eta_f} \cdot \tan \beta \quad (14)$$

By applying the drive thrust Q_1 of Equation (3) and driven thrust Q_2 of Equation (14) in Equation (12), Equation (15) can be derived. This equation is hereinafter called "Equation F_1 ".

$$F_1(N_e, T_r, i) = \frac{W \cdot r_w \cdot \pi^2}{900(\tan \theta_m + \tan \theta_r)} N_e^2 - \frac{T_r \cdot \theta_1}{2R_1 \cdot \tan(\alpha + \rho)} \cdot \left[1 - \frac{1}{2} \left(\frac{\tanh(\lambda \cdot r_1 \cdot \theta_1)}{\lambda \cdot r_1 \cdot \theta_1} + \frac{\tanh(\lambda \cdot r_2 \cdot \theta_2)}{\lambda \cdot r_2 \cdot \theta_2} \right) \right] - \left\{ k(l_f - l_i) + \frac{1}{2r_c} \cdot \frac{R \cdot r_T}{i_f \cdot \eta_f} \cdot \tan \beta \right\} \cdot \frac{\theta_1}{\theta_2} = 0 \quad (15)$$

In Equation (15), the parameters θ_m , θ_r , r_1 , r_2 , θ_1 , θ_2 , l_p and β are uniquely determined for individual ratios. As such, F_1 is simply a function of N_e , T_r , and ratio. However, it should be noted that Equation F_1 is valid only in the area between the LOW and TOP ratios, as shown in Figure 9, and it is no longer valid if the ratio is fixed to LOW or TOP.

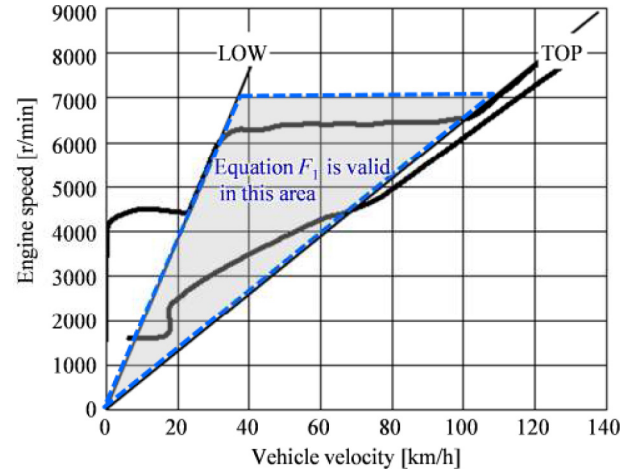


Figure 9. Characteristics of engine speed to vehicle speed in power train system with CVT

Equation of Balance between Converted Driving Force Calculated by Tr and Running Resistance Defined in Mode

In this balance relationship, the running resistance is calculated as the sum of the rolling resistance, the acceleration resistance, and the air resistance (aerodynamic drag). These resistances are given by Equations (16), (17), and (18), respectively. The acceleration to be used in determining the acceleration resistance as well as the vehicle speed to be used in determining the air resistance is specified for each driving mode. Therefore the acceleration resistance and air resistance at discrete times can be determined as constants. Accordingly, the running resistance at a discrete time is also a constant. The running resistance is given by Equation (19) as the sum of all types of resistance.

$$R_k = R_f + R_r \quad (16)$$

Where, R_k = Rolling resistance

R_f = Rolling resistance of front wheel

R_r = Rolling resistance of rear wheel

$$R_a = F_w \frac{a}{g} \quad (17)$$

Where, R_a = Acceleration resistance

F_w = Weight of motorcycle and rider

a = Acceleration

g = Gravitational acceleration

$$R_v = c \cdot V^2 \quad (18)$$

Where, R_v = Air resistance

c = Drag coefficient

V = Vehicle velocity

$$R = R_k + R_a + R_v = R_f + R_r + F_w \frac{a}{g} + c \cdot V^2 \quad (19)$$

Next, the driving force on the rear wheel F_r can be represented by Equation (20) using T_r , the final reduction ratio, and other parameters.

$$F_r = \frac{T_r \cdot i \cdot i_f \cdot \eta_f}{R_T} \quad (20)$$

Where, F_r = Driving force on the rear wheel

As the running resistance in Equation (19) is equal to the driving force on the rear wheel F_r in Equation (20) while the motorcycle is running, as shown in Equation (21), it results in Equation (22). This equation is hereinafter called "Equation F_2 ". Note that Equation F_2 is a function of T_r and pulley ratio.

$$R = F_r \quad (21)$$

$$F_2(T_r, i) = R - F_r = R_f + R_r + F_w \frac{a}{g} + c \cdot V^2 - \frac{T_r \cdot i \cdot i_f \cdot \eta_f}{r_T} = 0 \quad (22)$$

Equation of Balance between Converted Vehicle Velocity from N_e and Vehicle Velocity Defined in Mode

The third balance equation indicates that the vehicle speed V_c as converted from N_e is equal to the vehicle velocity defined in a mode V . Here, V_c is given by Equation (23) using the ratio and other parameters.

$$V_c = \frac{N_e \cdot 60 \cdot 2\pi \cdot r_T}{i \cdot i_f} \quad (23)$$

Where, V_c = Vehicle velocity converted from N_e

As $V = V_c$, Equation (24) can be obtained. This equation is hereinafter called "Equation F_3 ". Note that F_3 is a function of N_e and pulley ratio.

$$F_3(N_e, i) = V - V_c = V - \frac{N_e \cdot 60 \cdot 2\pi \cdot r_T}{i \cdot i_f} = 0 \quad (24)$$

Calculation of Ratio, N_e , and T_r

From the above discussion, the following three balance equations F_1 , F_2 , and F_3 were obtained with the variables N_e , T_r , and ratio.

$$\begin{cases} F_1(N_e, T_r, i) = 0 & (15) \end{cases}$$

$$\begin{cases} F_2(T_r, i) = 0 & (22) \end{cases}$$

$$\begin{cases} F_3(N_e, i) = 0 & (24) \end{cases}$$

By solving these equations for each discrete time during the driving in a mode, the N_e , T_r , and ratio values at each time point can be obtained. However, because these equations are quite complicated, the equations are unable to be solved directly to obtain N_e , T_r , and ratio. Therefore, these equations were solved using an iterative calculation method to obtain convergence. To be specific, by applying the simultaneous equation composed of F_1 and F_2 , T_r is eliminated and F_{12} of Equation (25) is yielded.

$$F_{12}(N_e, i) = 0 \quad (25)$$

In the next step, the conversion condition was set as $F_3(N_e, i) = 0$ and, the combinations of N_e and ratio that satisfy F_3 picked up from the combinations satisfying F_{12} were calculated at each discrete point of time. In the case that the equation F_1 cannot be applied with the fixed ratios for LOW or TOP, as shown in Figure 9, N_e and T_r can be calculated based on equations of F_2 and F_3 as the pulley ratio is known.

Calculation of FC and Mode Fuel Economy

By locating the point that corresponds to the calculated N_e and T_r values on the FC map, the fuel consumption at each discrete time point can be determined. Then, a provisional calculation of the mode fuel economy can be made by integrating the fuel consumption data over time. In this study, an FC map that is based on actual measurements was used to evaluate the effect of N_e and T_r calculated by the conventional method on the accuracy of fuel economy estimation. In calculating the fuel consumption, the P_{sf} correction for oil temperature rise, fuel cut throttle with closed, O₂ feedback, idling stop, and other conditions were also taken into account, as shown in Figure 10.

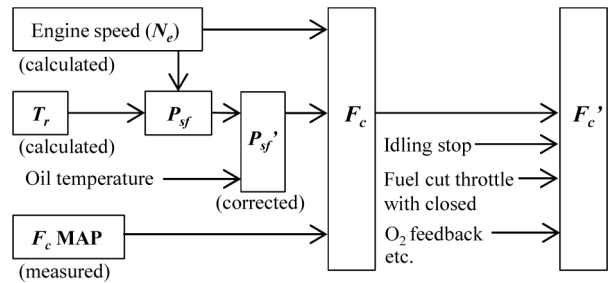


Figure 10. Calculation diagram of FC