

Model-Based Optimization for an AMT Clutch Control during the Vehicle Starting

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ABSTRACT

With the continuous growth in the emission requirements and higher riding comfort demand, the shift quality becomes more and more an important evaluation index of the automated transmission control algorithms. Traditionally, the shift quality is assessed subjectively by the driver's feeling and then adjusted based on the calibration engineer experience. This classical calibration has disadvantages, such as low reproducibility of the shifting event and a high dependence on the driver's driving habit, so here a model-based multi-objective optimization method is proposed, and the optimization of the clutch control parameters during the vehicle starting is used as an example.

Firstly, a Modelica® based vehicle model is introduced. A second-order sliding mode control is applied to track the clutch position trajectory. The clutch engagement point and the clutch engagement speed in the slipping stage are taken as the optimization objects. The shift quality, clutch engagement duration and mean longitudinal jerk during the vehicle starting, are taken as the evaluation criteria. An evolutionary multi-objective optimization method (non-dominated neighbor immune algorithm, NNIA) is chosen to find out the optimal control parameters. Finally the optimal clutch control parameters are determined based on a proposed mathematical function in different accelerator positions, and the testing results show that the optimized clutch control meets the driver's requirements in different conditions.

CITATION: Huang, H., Di, D., Chu, Y., and Guehmann, C., "Model-Based Optimization for an AMT Clutch Control during the Vehicle Starting," *SAE Int. J. Passeng. Cars – Electron. Electr. Syst.* 8(1):2015, doi:10.4271/2015-01-0161.

INTRODUCTION

Shift quality is one of the most important evaluation indexes for the automated transmission control algorithms and functions. Traditionally, the shift quality is assessed subjectively by the driver's feeling according to the ATZ grade from 1 (very dissatisfied) to 10 (outstandingly satisfied) [1]. Then the calibration engineer adjusts the control parameters in the transmission control unit (TCU) till the shift quality meets the requirements. However, this kind of subjective calibration faces the following challenges:

- influence of various environmental conditions (road surface, weather, traffic)
- low reproducibility of the shifting event
- impossibility of working at any time (night, weekend)
- complexity of control parameters to adjust
- high dependences on the driver's driving habit

In order to overcome these drawbacks, some researches on transmission shift quality have been carried out. In [2, 3] a score model, which is obtained through a regression analysis of subjective evaluations and objective parameters, is developed to objectively estimate the shift quality grade. Then through a testing planning (Design of Experiments) the control parameters are optimized. However, this kind of score model is based on large measurements

and highly depends on the driver's own experience. It restricts the widely portable application in other research object. In [4] a multi-objective genetic algorithm (MGA) is used to optimize the control strategy of clutch switch timing for a dual clutch transmission (DCT). A linear weighted method is defined to transform multi-objective based shift quality assessment criteria into a single level and then the optimization parameters are found through finding the minimum value. But the weights of assessment criteria are assigned by the expert advice, which has a personal experience limitation. In [5] a multi-objective shift quality optimization is described, the enhanced model based multi-objective optimization algorithm (EMMA) is chosen as the genetic algorithm. The sensitivity analysis is applied to choose the corresponding control parameters for different shift quality requirements. Since finding the correlation between the control parameters and the shift quality needs professional knowledge, and the TCUs from different suppliers have different control algorithms. These bring difficulties in the direct control parameters optimization. Moreover, the choice of the multi-objective optimization is also crucial since the optimization results highly depend on the algorithm convergence.

Based on these research results and contributions, a novel model-based multi-objective optimization method is proposed as a guided process for the automated transmission shift quality calibration. The general process overview is shown in Figure 1.

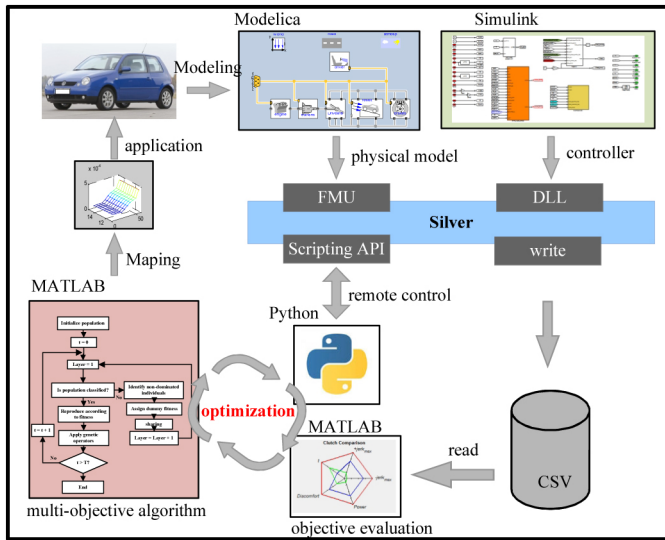


Figure 1. Model-based multi-objective optimization process

Firstly, based on the shifting measurements from a real vehicle, a non-linear Modelica[®] based vehicle model, especially the transmission components, is built in detail. The corresponding TCU and ECU control algorithms and functions are developed with Simulink[®]. In order to co-simulate these two parts seamlessly in the Model-in-the-Loop (MiL) simulation, Silver[®] is used, which consists of a virtual platform for the model-based calibration. Then a MATLAB[®] based multi-objective evolutionary algorithm is applied for the automatic optimization. During the optimization loop, the virtual vehicle platform is remotely controlled by a Python[™] based script. Through the Microsoft[®] Excel based measurements of the gear shifting results, and the feedback of the MATLAB[®] based shift quality objective evaluation system, the genetic algorithm finds the Pareto-optimal control parameters set and defines the initial input parameters for the next generation loop till the termination criterion is reached. Finally, the optimal of the control parameters are mapped, applied and tested in the real vehicle.

In this paper, the clutch control parameters optimization for an automated manual transmission (AMT) during the vehicle starting on a flat road is used as an example to present in detail the model-based multi-objective optimization process. The structure of this paper is: Firstly, the Modelica[®] based modeling process, especially the clutch components are described in [section 2](#). Then, in order to find out the correlation between the clutch position variation and the vehicle starting quality, the second-order sliding mode based position tracking controller is developed in [section 3](#). Thirdly, in [section 4](#) an evolutionary multi-objective optimization method (non-dominated neighbor immune algorithm, NNIA) is applied to optimize the clutch control parameters based on the objective shift quality criteria. Since the accelerator position depths represent the driver's driving intention, a mathematical function is defined to find out the corresponding optimal parameters based on the accelerator position value (from 0 % to 100 %). Finally, a summary is concluded.

MODELICA[®] BASED MODELING

Modelica[®] is chosen as the language for the modeling, it is a unified object-oriented, general equation based language for modeling multi-domain physical systems, it helps model developers to look at modeling from a physical perspective instead from a mathematical one [6, 7].

In order to build a suitable model for the shift quality calibration, the detailed modeling of the synchronizer and the flexible connection between different components are necessary. Since this paper main focus is put on clutch system, only the clutch modeling is introduced here, others can be found in [8, 9].

Clutch Model

Figure 2 depicts the push-type single-plate dry clutch used in the investigated AMT system. The clutch working condition depends on the torque transferred from the engine. Generally, the clutch is closed, the gearbox input shaft torque equals to the engine output torque. If the engine output torque is greater than the clutch capacity (the torque the clutch can transfer), both sides of the clutch have a speed difference, the clutch slips and the gearbox input shaft torque behind the clutch equals to the clutch capacity.

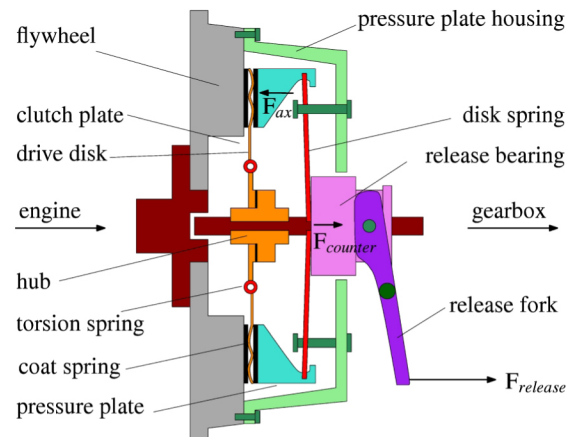


Figure 2. Schematic diagram of clutch

The clutch capacity is expressed in [equation 1](#) [1],

$$T_c \approx F_{ax} \mu_c z \frac{2(r_o^3 - r_i^3)}{3(r_o^2 - r_i^2)} \quad (1)$$

with axial force F_{ax} on the friction lining, speed depended friction coefficient μ_c , friction surfaces plates number z , outer and inner friction surface radius r_o and r_i .

The release bearing counter force $F_{counter}$ is a resultant force of disk and coat springs. It can be obtained under the assumption that the disk spring is a stiff lever. But actually the disk spring has a variable stiffness which depends on the compressed clutch displacement.

Elastic deformation is considered here to accurately describe the displacement relationship between the big end and the small end of the disk spring, in equation 2,

$$\begin{aligned}
 s_2 &= s_2' + s_2'' \\
 s_2' &= ks_{disk} \\
 s_2'' &= \frac{6F_d r_f^2}{\pi E_{disk} t_{disk}^3} \left(\frac{A_1}{\beta_1} + \frac{A_2}{\beta_2} \right) \\
 k &= \frac{l - r_f}{L - l} \\
 A_1 &= 0.5(d^2 - 1) - 2(d - 1) + \ln(d) \\
 A_2 &= 0.5(c^2 - d^2) - 2(c - d) + \ln\left(\frac{c}{d}\right) \\
 \beta_1 &= 1 - \frac{\delta_1 n}{\pi(r_e + r_f)} \\
 \beta_2 &= 1 - \frac{\delta_2 n}{\pi(r_e + r)} \\
 c &= \frac{r}{r_f} \\
 d &= \frac{r_e}{r_f}
 \end{aligned} \quad (2)$$

where s_2 is the displacement on the disk spring small end, it includes rigid displacement s_2' obtained according to lever ratio k and elastic deformation s_2'' , F_d is the force from the release bearing, r_f is the radius of the loading point at the small end, r_e is the inner radius on the small end, δ_1 and δ_2 are the groove widths on the small end and in the middle position, the corresponding structure diagram is shown in Figure 3, and Figure 4 is the comparison result of the simulated clutch counter force with measurements, it shows that the simulated model describes the similar force characteristic as the measured one.

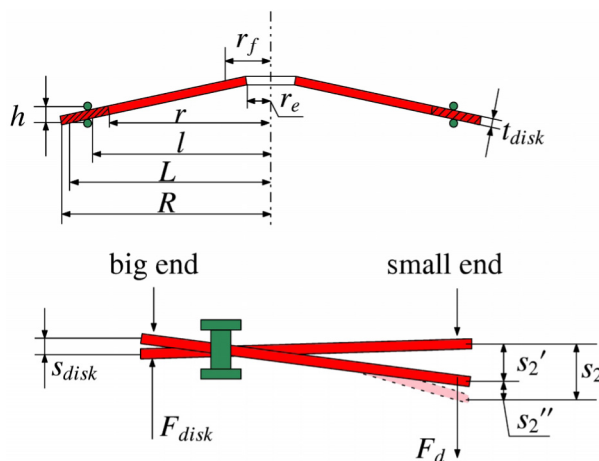


Figure 3. Schematic diagram of disk spring

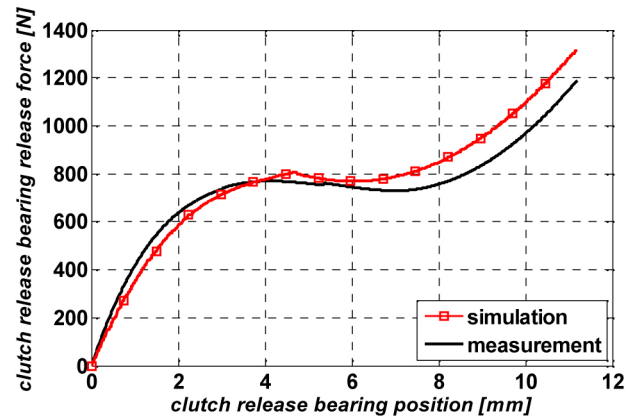


Figure 4. Comparison of the simulated clutch release bearing counter force with measurements

The disk spring force F_{disk} is estimated based on the Almen and László formula [10], in equation 3, E_{disk} is the elasticity modulus, t_{disk} is the disk spring section thickness, s_{disk} is the displacement, μ is the Poisson's ratio, R and r are the outer and inner radius of the disk spring, L and l are the pressure plate and supporting ring loading point radius, h is the height of the truncated cone.

$$F_{disk} = \frac{\pi E_{disk} t_{disk} s_{disk} \ln\left(\frac{R}{r}\right)}{6(1 - \mu^2)(L - l)^2} \left[\left(h - \frac{R - r}{L - l} s_{disk} \right) \left(h - 0.5 \frac{R - r}{L - l} s_{disk} \right) + t_{disk}^2 \right] \quad (3)$$

The Modelica® based clutch module is shown in Figure 5. The block *Springs* is used to calculate the force based on equations 2 and 3. T_c is used to estimate the clutch capacity and transfers this torque to the clutch block. In order to increase the clutch system dynamic behavior, the torsion damping system in the clutch plate is also built.

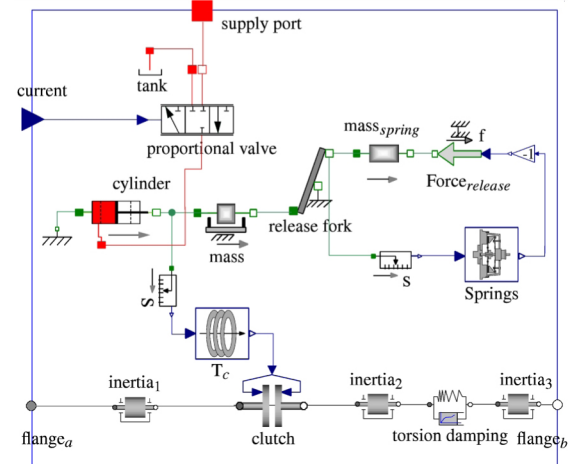


Figure 5. Dynamic module of clutch system