Temperature Estimation for Wear Prediction of Dry Clutches

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*Abstract –***For a reliable prognosis of clutch wear, the knowledge of friction lining temperature is crucial. Due to the lack of temperature sensors within the clutch, a reliable model is required to estimate the friction lining temperature. In this contribution, a thermal model for dry clutch temperature estimation is presented. The thermal system is described by a lumped-element model, which is capable of estimating the friction lining temperature based on the friction energy. Furthermore, external temperature influences ofcombustion engine and ambient air, as well as nonlinear effects,are considered. Results of modelidentification and validation for an automated manual transmission (AMT) on a test bench are shown in this article.**

Keywords –temperatureestimation, thermal modelling, dry clutch, wear prediction, predictive maintenance

I. INTRODUCTION

In the development of modern automobiles, the demand for durability and comfortissteadily increasing. Particularly wear and temperature dependent behaviour of the clutch influences these powertrain requirements.Hence, it is preferred to estimate wear and clutch temperature based on measured and estimated operating variables in vehicle's transmission control unit to handle the demand for better durability and comfort. In general, there is no economically satisfying solution for measuring temperatures inside the clutch. Therefore, a reliable thermal model is required for the friction lining temperature estimation.

This contribution shows an approach toa thermal clutch model with lumped-elements, whereby the clutch temperature can be used for correct estimation of clutch torque and prediction of future wear. The prediction of clutch wear can be used for clutch control on the one hand in order to sustain comfort over the remaining lifetime. On the other hand, the wear model can be used for failure diagnosis and predictive maintenance.

The presented model allows simulationof real temperature behaviour by means of a physical-based model structure. This structure was selected to ensure model computation in real-time environmentslike transmission control units. Model parameter identification is done by measurements on a transmission test bench. Furthermore, individual components of this model will be extended by nonlinearities. Nonlinear influences on clutch temperature are also investigated by test bench measurements.

II. RELATED RESULTS IN LITERATURE

In the field of thermalmodelling of dry clutches, several approaches are introduced in literature. One of the most detailed physical-based models is described in[1]. Here a lumped-element model based on several experimentswas presented. In[1] the influence of engine speedon heat transfer by convection was investigated, but the influence of clutch differentialspeed as well as clutch statewerenot considered. Simpler thermal models with total linear structures are shown in [2], [3] or [4]. In[3] and [4]the thermal models are used to describe the temperature dependency of the clutch torque, which is one of the main issues for optimal clutch control.Further contributionsare focused onthis topic, e.g.[5], [6] or[7], whereby the temperature influence on clutch torque was investigated in different experiments.

Furthermore, many researchersuseFinite element method (FEM) to simulate the clutch's thermal behaviour, e.g.[8], [9] or [10].In the area of FEM, especially clutch deformation and local thermal hot-spots are in focus. Due to the fact, that FEM is often too complex for online-estimation and that detailed FEM informationis not necessary for clutch wear prediction, here a physical-based approach is preferred.

The temperatureinfluence on clutch wear can bedescribed by different model approaches. Within the scope of this article, thewear model of [11] is used. A detailed literature survey on wear models can also be found in [11].

The current state of the art of thermal clutch modelling offers two possibilities for

improvement:Firstly,consideration of nonlinear influences, e.g. of differentialspeed and clutch state.Secondly,in previous works the estimated temperature is used to describe the influence on clutch torque, but not for wear prediction. Hence these issues will be considered in the following sections.

III. DESCRIPTION OF THE METHOD

A. WearEstimation:

The effect of wear $w$becomes visible by variation ofclutch torqueT_c.The clutch is considered as worn if$T_c$is smaller than the maximum engine torque$T_{e,max}$[11]. $T_c$can be calculated from the normal forceF_N, the number of friction plates z and the mean friction radius r.

$$T_c = z \cdot r \cdot \mu(F_N, \Delta\omega, \vartheta_{fl}) \cdot F_N(s_c, s_w, \vartheta_c) \quad (1)$$

The number of friction plates z and the friction radius $r$are assumed as constants. Hence, $T_c$depends on the friction coefficient μ and the normal forceF_N. The friction coefficient μ is influenced by theF_N, clutch differential speed $\Delta\omega$ and frictionliningtemperatureϑ_{fl}. In dry clutch systems$F_N$depends on clutch actuator positions_c, the frictionlining height $s_w$and the clutch temperatureϑ_c. Because of the significance for the calculation of the normal forceF_N, the membrane spring temperature is assumed asthe clutch temperatureϑ_c.

The temperaturesϑ_c and ϑ_{fl} depend onthe friction energy E, which is generated during slipping timet_{slip}.

$$E = \int_0^{t_{slip}} T_c \, \Delta\omega_c dt \quad (2)$$

As shown in[11] wear w influences clutch torqueby change ofthe friction coefficient μ and friction lining height s_w:

$$w = f_w(\mu, s_w) \quad (3)$$

For predicting wearat first the current health state willbe calculated based on the estimation of μ and s_w, whereby trends of both values areestimated up to the current time. If $F_N(s_c, s_w, \vartheta_c)$ is known, μ can be estimated.This requires the measurement of the clutch spring characteristics in respect to the influence quantities $s_c, s_w$and ϑ_c. Assuming that these influencing quantities are measurable or computable.

Subsequently,future wear can be predicted by assuming a load scenario from previous data. By using thresholds for $\mu$and$s_w$remaining lifetime of the clutch can be calculated. A more accurate approach is the consideration ofprobabilities instead ofremaining lifetime thresholds to predict system failures.

Due to the temperature dependency of μ ands_w[11], the estimation of clutch temperature is essential for accurate wear prediction. Hence, a thermal model is required, which will be illustrated in the next subsection.

B. Thermal Model:

For thermal systemmodelling, it is necessary to understand the heat flow through the dry clutch system, which is depicted inFig.1. In the present case of a dry clutch, the heat input Q_{in} is generated by frictional power P occurring during slipping phase.

$$\dot{Q}_{in} = P = T_c \cdot \Delta\omega \quad (4)$$

The frictional power results from differential speed $\Delta\omega$ and transmitted clutch torqueT_c.
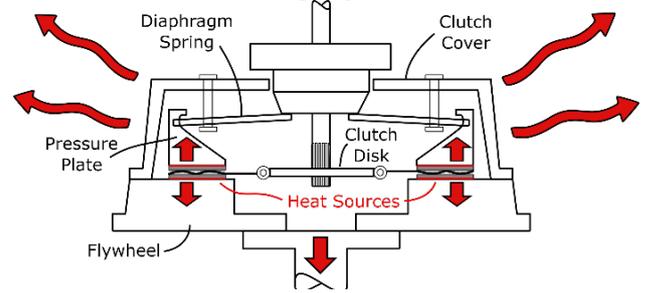

Fig. 1. Heat transfer of the dry clutch

As a result, heatspreads through the clutch, whereby at first adjacent metal parts of the flywheel and the pressure plate heat up. Afterwards, the diaphragm spring and the clutch housingwill heat up. The heat sinks during the slipping phase are the air in the clutch chamber and the combustion engine housing. For the clutch chamber, the heated air is exchanged by convection and for the combustion engine the heat transfers through the crankshaft connected to the flywheel. Finally, the heat of the air in the clutch chamber will distribute through the transmission housing to the ambient air or other components in the powertrain compartment.

Based on thedescribed heat flow, the thermal system isdivided into corresponding components as shown in Fig.2.The arrows in Fig.2illustrate the heat flow during the slipping phase.
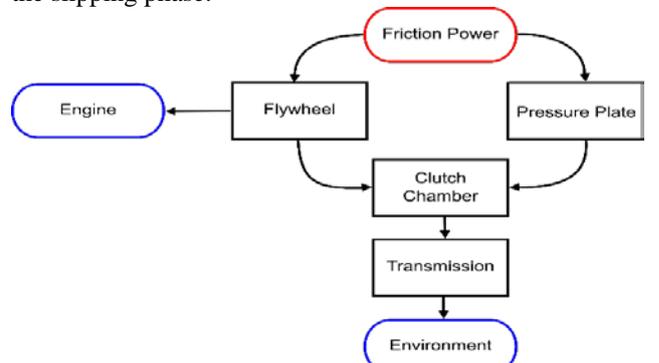

Fig. 2. Thermal components of the dry clutch

Of course, the powertrain compartment and the combustion engine also transfer heat into the thermal

system, if these are warmed up. Usually, the clutch will heat up to nearly the combustion engine temperaturewhenthe engine is started and the clutch is opened. Due to thehigh thermal resistance of the friction lining, the heat transfer between clutch disk and transmission shaft is neglected.

A thermal capacity is assigned to each component and a thermal resistance is used for describing the heat transfer between two components. The thermal resistance between two components x and y and thethermal capacity of one component xare defined as follows[12]:

$$\dot{Q}_R = \frac{\vartheta_x - \vartheta_y}{R_{xy}} \quad \text{and} \quad \dot{Q}_C = C_x \frac{d\vartheta_x}{dt} \qquad (5)$$

By the first law of thermodynamics the thermal system can be described according to Fig.2by using(5) as follows:

$$\frac{P}{2} = \frac{\vartheta_P - \vartheta_A}{R_{P,A}} + C_P \frac{d\vartheta_P}{dt} \qquad (6)$$

$$\frac{P}{2} = \frac{\vartheta_F - \vartheta_A}{R_{F,A}} + \frac{\vartheta_F - \vartheta_{eng}}{R_{F,eng}} + C_F \frac{d\vartheta_F}{dt} \qquad (7)$$

$$\frac{\vartheta_F - \vartheta_A}{R_{F,A}} + \frac{\vartheta_P - \vartheta_A}{R_{P,A}} = \frac{\vartheta_A - \vartheta_T}{R_{A,T}} + C_A \frac{d\vartheta_A}{dt} \qquad (8)$$

$$\frac{\vartheta_A - \vartheta_T}{R_{A,T}} = \frac{\vartheta_T - \vartheta_{amb}}{R_{T,amb}} + C_T \frac{d\vartheta_T}{dt} \qquad (9)$$

whereby the system is described non-dimensional. The indexesof (6),(7), (8) and (9) correspond to the following temperatures:$\vartheta_P$pressure plate, $\vartheta_F$flywheel, $\vartheta_A$air in clutch chamber, ϑ_T transmission housing, ϑ_{eng}engineand ϑ_{amb}ambient air.

The clutch temperature $\vartheta_c$from equation (1) is assumed as ϑ_A, because of the low thermal resistance between pressure plate and membrane spring. Furthermore the friction lining temperature ϑ_{fl} is assumed as ϑ_F, which has nearly the same surface temperature because of the contact during slipping. During slipping phase theactual temperatures of ϑ_{fl} and ϑ_c have slightlydifferent dynamicsas the approximated values of ϑ_F and ϑ_A, but averagetemperatures for the slipping phase are similar.For wear prediction average temperatures during slipping phase are more significant.Hence, both assumptions are valid, if the thermal model is used for wear estimation.

For clutch temperature estimation, the identification of the linear parameters$R_{A,T}$, $R_{F,eng}$, $R_{T,amb}$, C_P, C_F, C_A and C_T is necessary.$R_{P,A}$ and $R_{F,A}$ are considered as a nonlinear function f_R. The resistances $R_{F,A}$ and $R_{P,A}$, which define heat transfer from flywheeland pressure plate to surrounding air, describe the convection. Convectiondependson motor speedω_e, differential speed $\Delta\omega$ and clutch state S.

$$R_{F,A} \approx R_{P,A} = f_R(\omega_e, \Delta\omega, S) \qquad (10)$$

The parameter S describes the clutch state froma thermal point of view. Hence, for S = 1 (closed) both sides of clutch are thermally connected and for S = 0 (open) both are thermally separated. Detailed influences of the quantitiesin(10)are investigated by experiments.

IV.   RESULTS AND DISCUSSIONS

The required measurement data for model parameter identificationis generated through transmission test bench experiments [13].TwoPT1000 sensorsare placed on the inner side of the flywheel surface, to acquiring ϑ_F.These temperature sensorsare connected to anelectronic device, called ClutchSensorNode[14]. Thisnode,integrated into the flywheel, transmits data wirelessly to an outside sensor node, called PC node.In contrast to [14] here the energy supply of the Clutch Sensor Node is realized by a resonant inductive coupling instead of a battery, which allows to measure higher clutch temperatures and to run long time experiments.The unmeasured temperature of the pressure plate is assumed as $\vartheta_P \approx \vartheta_F$.

Additionally, the air temperature in the clutch chamber$\vartheta_c$is measured by a wired PT1000 sensor, which is placed in an opening of the transmission housing. Also,ϑ_T and $\vartheta_{amb}$are measured by PT1000 sensors.

The test bench does not contain a combustion engine. Therefore, the influence of the engine temperature is neglected by assuming $\vartheta_{amb} \approx \vartheta_{eng}$.

First, the nonlinear influences on $R_{P,A}$ and $R_{F,A}$are experimentally investigated. Therefore the system is stimulated by constant load until the clutch temperature attains steady state at 70 °C. Afterwards the system is cooled down to 60 °C with different sets of ω_e, $\Delta\omega$ and S.The experimental results for the influence of ω_e and $S$are shown in Fig.3.
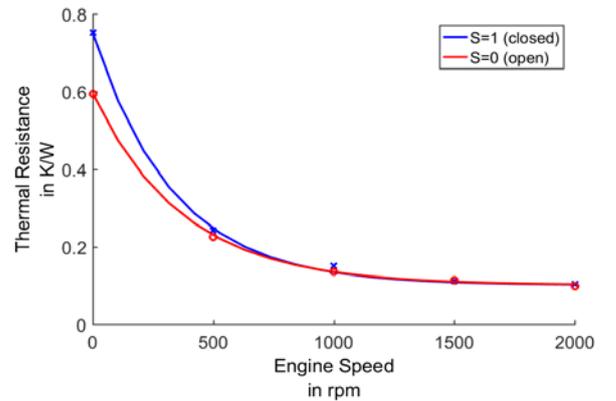
Fig. 3. Nonlinear influence of engine speed and clutch state on thermal resistance for convection inside the clutch chamber

For ω_ea significant nonlinear dependenceis identifiableduring cooling down. In comparison to $\omega_e$the influence of S is onlynotable for a stopped vehicle

($\omega_e < 600$ rpm). The influence of $\Delta\omega(= \omega_e - \omega_c)$ increases slightly for $\Delta\omega < 0$, which is assumed as an unlikely operating scenario. For $\Delta\omega \geq 0$ the influence of $\Delta\omega$ on the thermal resistance is in the range from 0.1 to 0.15 K/W. Therefore, the influence of $\Delta\omega$ in comparison to ω_e and S is negligible. Hence, the nonlinearity can be assumed as:

$$f_R(\omega_e, \Delta\omega, S) = \begin{cases} a_1 e^{b_1 \omega_e} + R_0 \text{ for } S=1 \\ a_2 e^{b_2 \omega_e} + R_0 \text{ for } S=0 \end{cases} \quad (11)$$

If (11) is inserted in (6), (7) and (8), the nonlinear parameters a_1, a_2, b_1, b_2 and R_0 need to be identified in addition to the linear model parameters.

In order to identify the thermal model, the clutch system is stimulated by random variations of friction energy and cool downtime according to Table 1. Whereby the friction energy is varied by different settings of engine speed, clutch torque and load torque. These parameters are varied for each load cycle. For each load cycle a vehicle launch and stop is simulated on the test bench. Engine speed, clutch torque and load torque are kept constant during clutch engagement by different controllers. This ensures a better comparability and reproducibility of the experiments.

Table 1. Parameters for thermal model identification

parameter	value range
engine speed	[1000, 1500, 2000] rpm
clutch torque	[30, 45, 65] Nm
load torque	[100, 300, 500] Nm
load inertia	70 kgm²
cool downtime	[30, 50, 60,80,90,120] s

Parameter identification of the presented thermal model is done in Matlab by Grey-Box model estimation with a nonlinear least squares solver. The execution time for the simulation is chosen as 10ms by taking into account of the fastest dynamics for the heating process during slipping, which is assumed as 2 s.

The resulting measurements in comparison to the estimated system behaviour for ϑ_F and ϑ_A are illustrated in Fig.4. The thermal model fits well with a root-mean-square error (RMSE) of 2.5 % for the identification data and less than 10 % for 5 validation data sets.
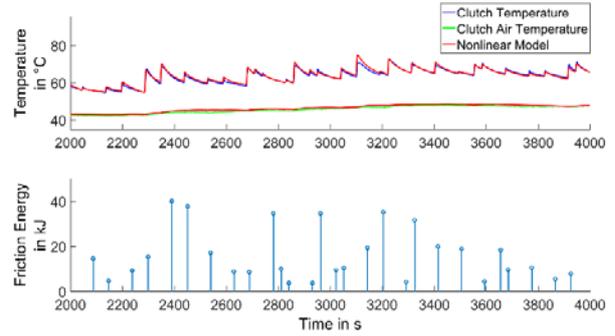

Fig. 4. Measured and estimated dry clutch temperature

Fig.4 shows deviations up to 5% for some peaks caused by the first order model structure, which only fits first order dynamics perfectly. Despite this limitation the model can estimate the mean temperature over long time intervals, which is important for an accurate wear prediction.

The advantage of the model nonlinearity (11) becomes visible for long cool down processes. During cool down the linear model is unable to estimate accurately, if engine speed or clutch state are varying. For illustration an example with varying engine speed is shown in Fig.5.
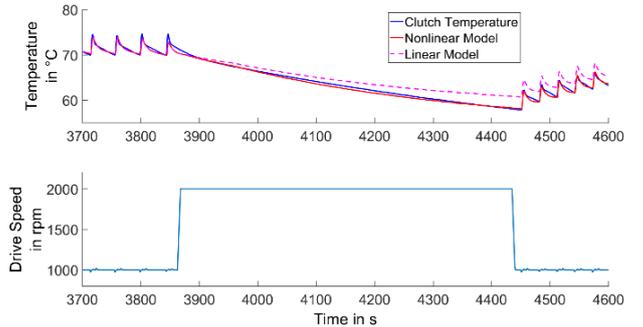

Fig. 5. Comparison of linear and nonlinear model during cool down with varying engine speed

The linear model in this case achieves an RMSE of 8.4% and the nonlinear model achieves an RMSE of 3.8%. Despite the fact, that the linear model achieved a high estimation accuracy, similar to the nonlinear model, by an RMSE of 3.5% for a cool down with 1000 rpm engine speed, it is not capable to estimate the cool down for 2000 rpm. For example, the linear model estimation in Fig. 5 shows a deviation of 9.4 % after cooling down with 2000 rpm for 550 s.

In the end, the illustrated model already provides a very good temperature estimation despite some limitations. One general limitation is the extrapolation for unknown load scenarios. An accurate estimation of the clutch temperature is only possible if the current load scenario is part of the identification data. Enormous effort will be necessary to achieve a perfect model fitting for all possible load scenarios. Otherwise, the model will have an estimation error for extrapolated load scenarios.

Further limitations are assumptions for the missing

temperature of the pressure plate and the missing heat source of the combustion engine. The influence of the combustion engine was tested by simulating this missing heat source with an additional heating device. As a result, the steady state temperature was increased, which agrees with the thermal model behaviour of the simulation. Due to the lack of additional temperature sensors, the pressure plate temperature could not be determined. In general, the pressure plate temperature canbe assumed slightly higher as the flywheel temperature due to the smaller thermal capacity and the negligible thermal resistance between both components[1]. Therefore, wear should differ onboth clutch sides.

V. CONCLUSIONS AND OUTLOOK

In summary, the presented thermal model was identified and validated on a transmission test bench. In comparison to existing thermal models for dry clutches, nonlinear effects and environmental conditions were taken into consideration resulting in reduced RMSE values. The investigated nonlinear behaviour can be integrated into the thermal model through thermal resistances with state variables dependence.Thus model accuracy can be improved for different operating scenarios, especially during long cooldown phases, which last more than 120s.The presented model is capable of online simulation in a transmission control unit due to its simplicity and low simulation execution time.Hence, the deployment of the presented approach in a real vehicle seems to be possible, if the identification is repeated for vehicle environment.

The estimated clutch temperature can be used for online wear prediction in passenger cars.This provides an opportunity to diagnosis of current health state and control of future health state. Due tocontrol of future abrasion, a minimum clutch lifetime can be guaranteed. Furthermore, damages caused by thermal overload can be avoided. As a result, predictive maintenance can be performed for dry clutches.

The mentioned model limitations should be in the focus of future research. To improve the model accuracy for extrapolation without increasing the measurement effort, the installation of an additional sensor at the transmission housing or inside the clutch chamber is recommended. By the acquired sensor data the model can be reinitialized during operation to minimize estimation errors. Furthermore, this sensor makes the thermal system observable and grants the opportunity to implement an observer for a more accurate temperature estimation.

Even existing sensor values like the transmission oil temperature can be used for the model reinitializing, but not as an observer input. Due to the fact that transmission oil circuits in general are temperature controlled, the system observability is not given.If additional sensors are not installable in automotive application due to economic reasons, the validation of the temperature estimation should be done by temperature characteristics of the clutch torque.

For a more accurate wear prognosis, the knowledge of the pressure plate temperature is desirable. Hence, the measurement of pressure plate temperature could be investigated in future research. If this temperature is unknown, it should be estimated by thermal capacity of pressure plate and the known flywheel temperature.

VI. ACKNOWLEDGMENT

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