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Modelling, Simulation and Control of an Automotive Clutch System

Aseesh chintala[1], R. Lalitha Narayana[2], Ch. Srinivas[3]
1PG Student, 2Associate professor & Head of the Department, 3Associate professor
1, 2, 3 Department of Mechanical Engineering, A.S.R. College of Engineering, Tanuku.

Abstract—*In this paper it is intended to optimize the performance of the automotive clutch system. The modeling and Simulation of an automotive clutch system is carried out and a control strategy is derived for optimizing its performance. A mathematical model of a simplified Clutch system is built for the analysis of its dynamic behavior. Also a decoupling controller is derived which independently controls both the engine and the slip speed to avoid engine stalling and other problems. A sensitivity analysis was carried out to evaluate the effective structure of the model.*

Index Terms— Automotive clutch system using decoupling controller, karnopps approach, simulink clutch model

I. INTRODUCTION

1.1. Background

An automotive clutch is used to connect and disconnect the engine from the transmission. A clutch is required only for the vehicles with manual transmissions. But the clutch is not required as in the case of automatic transmissions because the engine and the transmission are disconnected automatically with the help of torque converter or fluid coupling.

When designing a clutch the following parameters like temperature rise and wear, torque delivery, energy loss, activating force, etc have to be taken into account. The following factors like good thermal conductivity, resistance to thermal fatigue, good high temperature strength and coefficient of friction that remains stable with temperature

1.2 Principle

For an automotive clutch, a drive disc is connected to the engine crankshaft and the driven disc is connected to the input shaft of the transmission.

When both the drive and the driven discs are not in contact, the engine crankshaft is set to rotate but the input shaft of the transmission will be stationary. But both the drive and driven disc is made to spin together when the input shaft of the transmission is locked to the engine crankshaft.

II. LITREATURE SURVEY

A dynamic model is built and mathematical modelling on engine, clutch, gearbox and axles, wheels is performed. Using Simulink, the dynamic model thus built is simulated for engaging the clutch. A control strategy is derived and the designed control law is implemented. A decoupling controller controlled by PLC is used to independently control both the engine and slip speed. It is noted that it was possible to engage the clutch within 1.5 seconds [M.H.M. Dassen].

For the investigation of the behavior of the drivelines, a test rig has been developed for the experimental evaluation of the designed control mechanisms. An analytical model is also developed for observing the clutch behavior at the time of closing and hence the vehicle’s acceleration from the standstill with the help of flywheel. Apart from the acceleration from the standstill, the simulations are carried out with a few adjustments made to the model for other situations too. This analytical model is then transformed into a Matlab Simulink model for performing simulations and further analysis on the same. At last a control strategy is derived using a steering and PI controllers to control the torque transmitted by the clutch and hence the experimental estimation of both the model and the controller’s parameters is achieved. [*M.J.W.H. Edelaar*]

A mathematical modelling of an engine, the torque converter, multi-plate clutch, the band brake and other mechanical elements has been performed for the better understanding of the overall system. An investigation of the shift transient characteristics of the vehicle provided with Ravigneaux type automatic transmission is carried out. The dynamic models such as transient characteristics can be used for the analysis on the performance of the powertrain and with the help of this dynamic model; the derivation of equations of motion is achieved. A control on optimum pressure and reduction of engine torque are carried out for the improvement of shift transient characteristics. [*Woosung Han and Seung- Jong Yi*]

In all the mechanical systems, slip-stick friction to some degree is present. The surfaces stick together when it approaches zero velocity; the irregularities present in the surface plays a significant role. The sliding can be followed when the force greater than the sliding friction force is necessary to break the static friction force. When the sliding takes place, a force is prescribed by the friction element. When the surfaces stick, a velocity is prescribed by the friction element. Some problems are found in this simulation model because of the switching of causality as the causal switch is subject to produce different state representations. Here a fixed causality friction model which can be used for computation is built. The model thus developed is quite similar to the LuGre mode. [*Donald Morgolis*].

When the engine starts running, the cylinder ignites very high combustion to the crankshaft. These forces results in the effect of inconsistent rotational speed and the crankshaft bending. As a result of these excitations, it produces vibrations in both rotational and translational mode with the following range of frequencies. [*Patrick Kelly*]

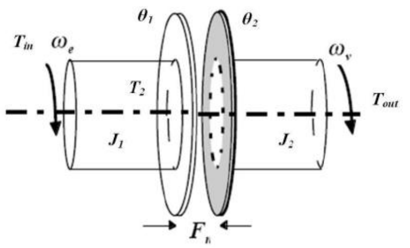
1. Gear whine in 1000 Hz.
2. Clutch judder complaints around 8 Hz.
3. Whoop around 300 Hz.
4. Gears rattle around 500 Hz.
5. Customer complaints less than 1000 Hz.

During the engagement of the Clutch at around 10 Hz, „Judder" is produced. It also occurs when vehicles with independent suspension systems for the driving wheel seem to be more sensitive when compared to the problem that arises with rigid axles, also can be associated with the movement of the engine on its mountings. It is usually not favorable when it produces Clutch Judder quite easily and hence the steps have to be taken to minimize it. The severity of this Judder can be understood clearly by the manner in which the vehicle is driven *[R.P.Jarvis and R.M.Oldershaw*]

III. Basic Model of a Clutch System

A basic driveline model of a clutch is built and using the equation of motions derived from the driveline model, a Simulink model can be developed which helps to calculate the speed and the torque produced by the system. The input parameters of the system can be varied and the simulations are performed to estimate the nature of the behaviour of the system.

The basic model of the clutch is shown in below figure



It consists of driver and driven plates with an input torque (T_{in}), driver torque T2 and an output torque T_{out} as shown.

General equations of motion for the transfer of torque in this clutch system are as follows:

For driver plate,

$$J_1\ddot{\Theta}_1 = Ti_n - K_1 (\Theta_2 - \Theta_1) - C_1 \Theta_1 - T_2$$

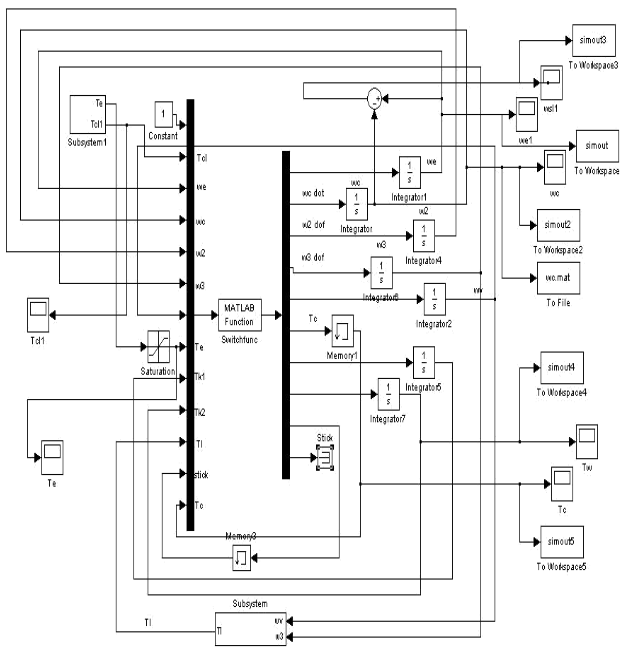
For driven plate,

$$J_2\ddot{\Theta}_2 = T_2 - K_2 \Theta_2 - C_2 - T_{2ut}$$

Where,

J_1, J_2 - moment of inertia for the driver and the driven plates respectively, in Kg.m^2

K_1,K_2 - Spring Stiffness of the clutch for the driver and



driven plates respectively, in N/m

T_{in}, T_2, T_{out} - Input Torque, Driver torque, Output torque respectively, in Nm

C_1, C_2 -Damping for the driver and driven plates, kg/s

θ - Angular displacement, in radians

IV. Clutch System MODELING

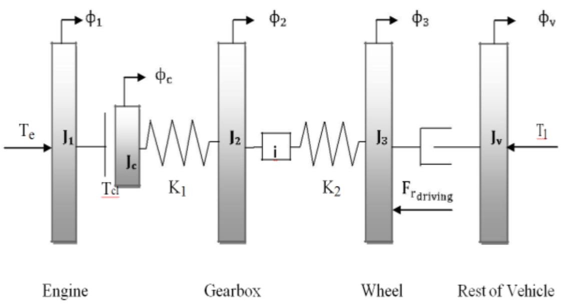
Simulink Model:

The equations of motions represented in the previous formulation can be integrated to determine the unknown variables. Simulink software was used to perform the integration of the above equation as shown

A driveline model is built in this chapter and the equations describing the various dynamic behavior of vehicle driveline system are discussed. Two methods of analysis are presented; the state space and the Karnopp approach.

1. Driveline Model

The input torque, T_e is applied directly on the engine inertia J_1 with its angle ϕ_1 as shown in the figure. This engine inertia is connected to the pressure plate through a rigid axle. The torque transmitted by the clutch, T_{cl} acts on the friction plate with inertia J_c and corresponding angle ϕ as shown. The clutch can be attached to the gearbox with inertia J_2 and angle



ϕ_2 with the help of axle spring stiffness, K_1. The rolling resistance $F_{r\,driving}$ acts on the inertia of the driving wheel J3 with corresponding angle ϕ_3 .The torque transferred at the wheel T_w can be expressed as

$$T_w = K2\ (i\phi 2 - \phi 3)$$

2. Karnopp's Approach:

According to Karnopp, both the sticking and slipping systems can be described using one expression. The outcome of using this method of formulation is that the same set of equations can be used for both sticking and slipping systems and therefore the switching is not necessary within this system. Since the equations of motions derived from the driveline model are valid for both slipping and sticking systems, the following expression can be used.

When the Clutch locks,

→ F_n is no longer influencing the transmitted torque, $T_{cl.}$

Torque T_{cl}, transmitted at the interface of J_1 and J_2 are calculated by,

$$T_{cl} = J_1K_1\ (\phi_e - \phi_2) + J_eT_e / (J_1 + J_e)$$

When the clutch sticks, there is a variation only with the torque transmitted by the clutch, T_{cl} and hence the switching is not needed.

V. CLUTCH CONTROL.

The engagement of dry clutches is considered as a very important process ensuring small facing wear and better performance of the driveline. Small friction losses, minimal required time for the engagement of the clutch, the preservation of driven comfort are some of the important factors to be considered when controlling the engagement of the clutch (Powell, 1998).

There are two conditions to be satisfied for the effective engagement of the clutch. These are:

1. No-kill condition is the major constraint of the clutch engagement process, i.e. stalling the engine must be avoided.

When the normal force F, applied to the clutch disk and the torque transmitted to the clutch T_{cl} are very huge, the engine would stall due to the loss of the engine speed ϕ_1 . This condition can be expressed as:

$$\phi_1 \geq \phi_1^{min}$$

2. No-lurch condition is the other constraint to be satisfied during the clutch engagement. When the engagement of the clutch is smooth, it results in the decent level of comfort for the driver. The oscillations present in the drive train are the major reason for the driver discomfort. These oscillations are dependent upon the time derivative of the slip speed, i.e. $\omega_{sl} = (\phi_3 - \phi_v)$.

Using the assumption of constant torque T_e, the formulation for no-lurch condition can be given by:

$$\omega_c(t^+) - \omega_c(t^-) = J_1{*}\omega_{sl}(t^-)/(J_1+J_e)$$

$$J_1{*}\omega_{sl}(t^-)/(J_1+J_e) \leq C$$

Where C is a positive constant.

Decoupling Controller:

The derivative of the slip speed, $\omega_{sl} = (\phi_1 - \phi_c)$ should be continuous to eliminate the vibrations in the system. The derivation of the controller which independently controls the engine speed and the slip speed can be achieved which helps to avoid the discontinuities that exist in the derivative of the difference between $\phi_1 - \phi_c$.

When assuming perfect rigid axes with no stiffness terms, the following equations were found for the derivation of the controller.

Engine and Slip Speed Controller:

The PID controllers for both v1 and v2 can be substituted in the equation 5.5 to independently control both the engine and the slip speed. The details of the PID controllers are listed below.

Engine speed controller (v1):

The parameters for the engine speed controller, v1 are:

P = 0.875

I = 0.012

D = 0.017

Slip speed Controller (v2):

The parameters for the slip speed controller, v2 are:

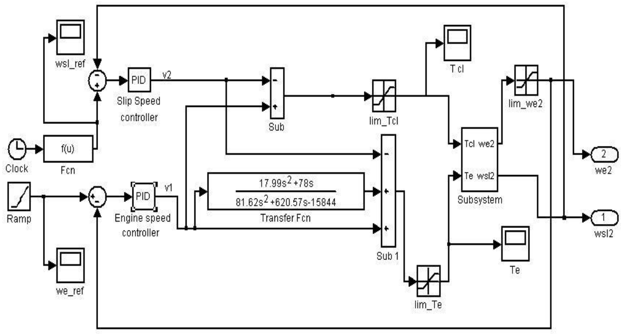
P = 1.52

I = 0.0085

D = 0.018

Simulink model:

A schematic form of this decoupling controller is shown.



Schematic representation of a Decoupling Controller

The value of G(s) can be included in the decoupling controller using the transfer function block. The engine torque Te and the torque transferred by the clutch Tcl are the input for the clutch system which independently controls the engine and the slip speed we and wsl respectively.

VI. RESULTS.

The simulation results for the basic model of the clutch, Karnopp model, and the decoupling controller model are given in this chapter. A sensitivity analysis was carried out for optimizing clutch system performance.

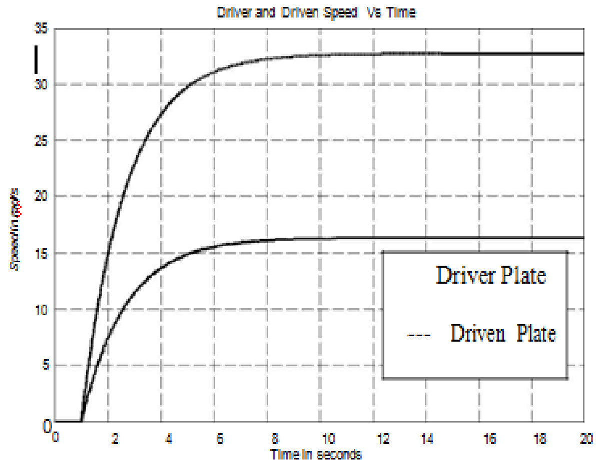
1. Simulation Results for the Basic Clutch model:

The main output of these models is speed and torque response and will be presented in this chapter after each simulation set-up as graphical output. The values of the input parameters used for the first simulation are:

Viscous friction = 0.6 N, Coulomb friction = 0.4 N

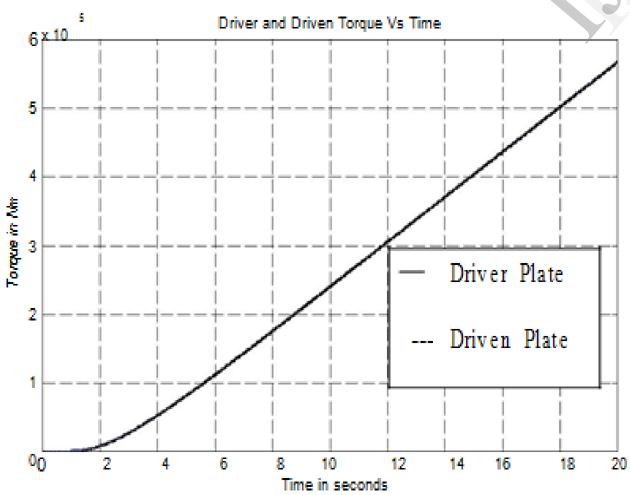
Inertia, $J_1 = J_2 = 1$ Kg.m^2

Stiffness, $K_1 = K_2 = 2000$ N/m



Driver and the driven speed Vs Time

It can be noted that the speed of the driver plate at the end of the simulation is about 32.5 rad/s whereas the speed of the driven plate is about 17 rad/s. Both the speed of the driver and the driven plates increases very quickly and attains a maximum steady state after about 10 seconds.



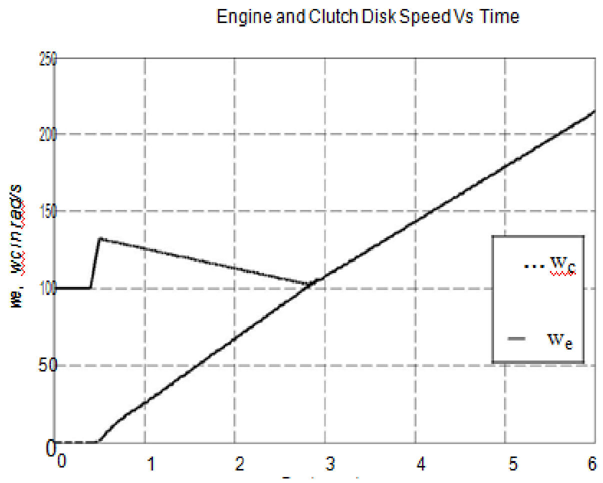
Driver and driven plate torque Vs Time

It is noted that a very high torque is produced. Both the driver and the driven plates produce the same torque throughout the simulation and increases almost linearly with respect to time.

2. Simulation Results for the Karnopp model:

The simulation was performed and the results are discussed as follows.

2.1. Engine and the Clutch disk Speed (we and wc)

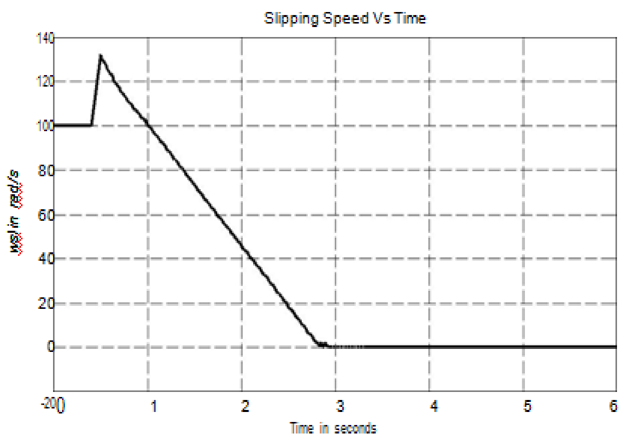


Engine and the clutch disk speed at the time of engagement (we & wc)

The engine and the clutch disk speeds are plotted together as shown. The specified Tcl value is responsible for the slow engagement of the clutch. When the Tcl is larger, the engine speed (we) drops heavily resulting in an engine stall.

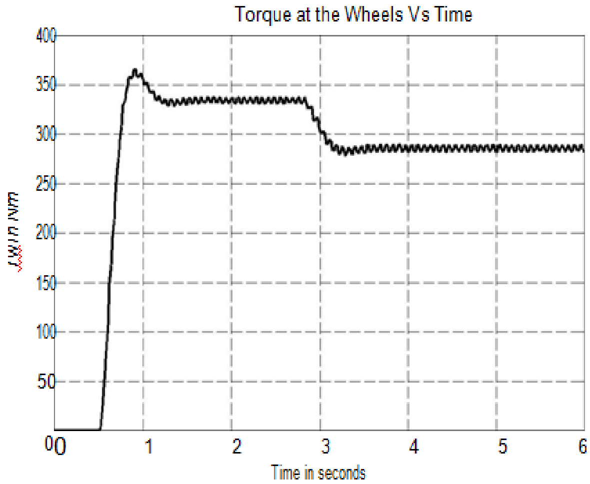
2.2 Slipping Speed (wsl)

It was observed from Figure 6.12 that the slipping speed (engine and the clutch speed difference) drops to zero. Initially it rises for a very short time and then it continues to drop and



Slipping Speed during engagement (wsl)

3. Torque at the wheels (Tw)



Torque transferred at the wheels

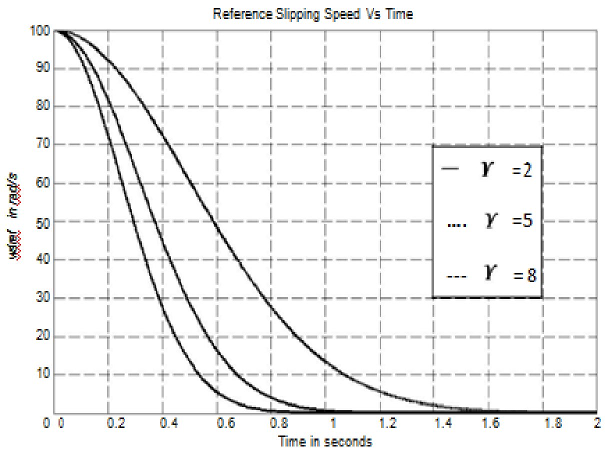
The peak value of torque was reached at the time of slipping phase. In the slipping phase, the torque transmitted by the clutch was heavy and when the clutch sticks, level of the torque drops down for a particular time interval and then it remains constant throughout the simulation. The springs present in the driveline produce some vibrations which are the reason for an irregular curve and it takes a little while to damp out

4. Simulation Results for the Decoupling Controller

4.1. Reference Slipping Speed:

The simulations were performed for the various time and the constants γ is set at 2, 5, and 8, with tuned engine speed controller, v_1 and slip speed controller, v_2 for every constant. The reference for the slip speed can be plotted

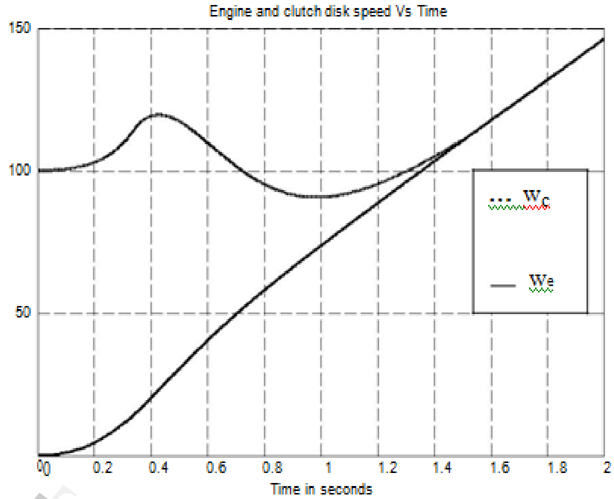
From the figure, it is noted that the shortest time for which the function drops to zero is 1 seconds. So the corresponding time which is responsible to drop the function to zero and remains to stay in zero throughout the simulation is found to be γ = 5. So the further simulations were performed with the time constant γ = 5.



Reference for the Slipping Speed

4.2 Engine and the Clutch Speed (wc and we):

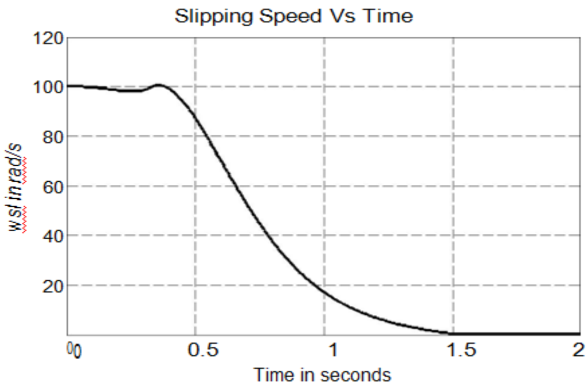
From the figure, it was observed that the clutch can be engaged at 1.5 seconds with the speed of the engine dropping not lower than 100 rad/s as how it started. The unlimited value of the engine speed controller v_1 and the limited power of the engine are the two main reasons for the drop of the engine speed. The lower limit of the engine speed was set to 90 rad/s in order to obtain a fast engagement of the clutch.



Engine and the Clutch disk Speed during engagement

4.3. Slipping Speed:

It was observed from the figure that the function reaches zero at 1.5 seconds which means the clutch sticking time is found to be 1.5 seconds.



Slipping Speed during engagement

VII.CONSLUSIONS

The main findings of this thesis can be summarised as

1. The mathematical modelling of a clutch, engine, gearbox and axles, wheels has been performed for the better understanding of the overall system.
2. The driveline model thus built was incorporated into Matlab/Simulink and further simulations were performed.
3. A decoupling controller was derived to prevent engine stalling by independently controlling both the engine and

the slip speeds.

4. The working of this controller was made possible with the help of PID controllers to control engine and the slip speeds.
5. With the help of this decoupling controller, the engagement of the clutch was attained within 1.5 seconds without many disturbances in the torque at the wheels.
6. Based on the simulations, a sensitivity analysis was carried out for optimising the performance of the clutch system.
7. The clutch sticking time increases when the gears were shifted from 1-5. Also when the damping coefficient value was increased, the clutch sticks very soon.
8. There are chances for the decoupling controller to fail when the model used seems inaccurate. Hence the reliability of these types of controllers is limited.
9. The performance of this system shall be affected due to un-modelled disturbances since PID controllers tuning seems very sensitive.

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