THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING
IN SOLID AND STRUCTURAL MECHANICS

Modelling and optimization of gear shifting mechanism

Application to heavy vehicles transmission systems

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Cover:
Figure illustrates sub-phases of the generic synchronization.

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ABSTRACT

Without an efficient transmission system the idea of an efficient vehicle is incomplete. During running of the vehicle especially in case of heavy vehicles the transmission system has to face different situations during which the system possibly can lose its efficiency. Definitely variations in efficiency of the transmission system demand from engine to vary its power to maintain the system efficiency to regulate speed of the vehicle. Gear shifting mechanism of the transmission system is one of the responsible agents for losing rotational speed of the system. The engine needs to increase its power to compensate such kind of losses by injecting the more fuel. Ultimately there will be more emissions on exhaust side. In this scenario the conducted research is concentrated on the gear shifting mechanism.
To start with the well-defined aim a generic cone synchronizer mechanism modelled by a mechanical system with five degrees of freedom and comprising three rigid bodies is studied to understand the gear shifting process. To resolve complexity of the complete gear shifting processes detailed kinematic description of the phases and sub-phases gives an opportunity to capture the nature of bodies' interaction and forces arise during their interaction. In the project a mathematical model to represent the whole gear shifting process is developed based on Constrained Lagrangian Formalism. The developed model went through validation test by using experimental data. Because the developed mathematical model is flexible to adopt other relevant models, the friction model is applied to the developed mathematical model and analyzed the differences in the results. The next step is to optimize the gear shifting process based on the input parameters. Using the developed model the analysis is performed in two steps; in first step the sensitivity analysis is considered to study the effect of variations of individual parameter on the system performance and in second step effects of a set of synchronizer mechanism's parameters which vary simultaneously are studied by using the optimization technique. Time duration of the gear shifting and speed difference at end of the main phase of synchronization process are chosen as objective functions of the system. Parameters estimated quickness and smoothness (comfort) of synchronization processes are cone angle, cone coefficient of friction, applied shift force, blocker angle, blocker coefficient of friction, cone radius, gear moment of inertia and ring moment of inertia. Eight cases of the synchronizer mechanism performance are studied under different scenarios of master/slave and different operating conditions. Further analysis on results obtained from the Pareto optimization clarifies the degree of influence of the input parameters. It was found that optimal performance of the system can be obtained by tuning few of the system parameters which have higher degree of influence instead of changing all the parameters together. For example in the case where the sleeve is considered as a master at nominal condition optimal performance of the synchronizer can be obtained by paying attention to applied shift force, cones angle, cones coefficient of friction, blocker angle and blocker coefficient of friction instead of taking all eight input parameters. At the end a graphical user interface is developed to obtain the synchronization performance diagram.

Keywords: Generic synchronizer, constrained Lagrangian formalism, synchronization, sensitivity analysis, Pareto optimization

PREFACE

The research work has been carried out since 3rd of March 2014 at Division of Dynamic, at Department of Applied Mechanics, Chalmers University of Technology. The work is a part of the project of transmission cluster with collaboration of AB Volvo, Scania CV AB, Royal Institute of Technology, Chalmers University of Technology and VINNOVA. The overall aim of the project is to develop new knowledge and optimize the gear shifting process of transmission system with respect to life, robustness, power consumption and shift quality to meet the future demands. The project is funded by AB Volvo, Scania CV AB and VINNOVA.

First of all I would like to thank my ALLAH ALMIGHTY WHO is the most GRACIOUS and the most MERCIFUL.

I would like to thank my supervisor Viktor Berbyuk who aggressively tried to lead the research with his experience and to give the direction to produce the impressive research. I am also thankful to my co-supervisor Håkan Johansson for his deep interest in the work and his efforts to increase accuracy of the work. Collectively I would like to express my sincere gratitude to my both supervisors for contribution of their ideas to increase the quality of the research project.

A special thanks to Magnus Andersson from AB Volvo for discussing the issues during gear shifting process and conveying the demands in the work. Thanks to the project team in Stockholm, Ulf L Sellgren at Royal Institute of Technology, Daniel Häggström and Kenth Hellström at Scania CV AB for their fruitful discussion.

I take this opportunity to express my deepest gratitude to my mother, my father, other family members and my teacher Al-Shah Mazhar Fareed Subhani for their support and motivation to continue my higher education.

Gothenburg, December 2016
Muhammad Irfan

THESIS

This thesis includes an extended summary and the following appended papers:

Paper A	M. Irfan, V. Berbyuk and H. Johansson, (2015), “Modelling of heavy vehicle transmission synchronizer using constrained Lagrangian formalism”, *In Proc. of the International Conference on Engineering Vibration,* Ljubljana, 7 - 10 September ; [editors Miha Boltežar, Janko Slavič, Marian Wiercigroch]. - EBook. - Ljubljana: Faculty for Mechanical Engineering, 2015 p. 28-37.
Paper B	Irfan, M., Berbyuk, V., Johansson, H., (2016), “Dynamics and Pareto Optimization of a Generic Synchronizer Mechanism”, in *Rotating Machinery, Hybrid Test Methods, Vibro-Acoustic & Laser Vibrometry,* Proceedings of the 34th IMAC, A Conference and Exposition on Structural Dynamics 2016*, Editors James De Clerck and David S. Epp, Volume 8, pp. 417-425, 2016, Springer,* ISBN: 978-3-319-30084-9, http://dx.doi.org/10.1007/978-3-319-30084-9_38.
Paper C	M. Irfan, V. Berbyuk and H. Johansson, "Performance control of the transmission synchronizer via sensitivity analysis and parametric optimization," *To be submitted for international publication.*

The appended papers were prepared in collaboration with the co-authors. The author of this thesis was responsible for the major progress of the work in preparing the papers, i.e. took part in planning the papers, developing the theory, performing all implementations and numerical calculations, analysis of the results and writing.

In addition to papers A, B and C the following report has also been part of the research in this PhD project which is not included in this thesis:

Irfan, M., Berbyuk, V., Johansson, H., "*Constrained Lagrangian Formulation for modelling and analysis of transmission synchronizers*," 2015:05 Department of Applied Mechanics Chalmers University of Technology, Gothenburg, 2015.
http://publications.lib.chalmers.se/publication/233233.

CONTENTS

Part I

Extended Summary

1 Introduction and motivation

The vehicle engine needs to operate efficiently under favorable conditions regarding torque and speed. The concept of an efficient engine is not possible without improving the transmission system performance. Synchronizer is a crucial part of the transmission system of a vehicle which works during the gear shifting process. The gear shifting needs to be as quick as possible while maintaining the shift quality at the same time. During gear shifting, the selector disengages one gearwheel and engage another gearwheel at the same shaft, rotating at a different speed. The synchronizer mechanism reduces the speed difference to allow smooth engagement of the new gearwheel. Movement and interaction of bodies of the synchronizer are complex and includes several challenges such as the full transition between full hydrodynamic lubrication to boundary lubrication as well as internal frictional contact conditions. A lot of research have been done on different aspects of the gear shifting process, see e.g. [1-24].

Abdel-Halim, Barton, Crolla and Selim demonstrated the performance of improved multi-cone synchronizer for a typical vehicle by deriving the equations of motions validated by experimental studies at different oil temperature [1]. Abel, Schreiber and Schindler presented the solutions for the comfort of automatic shifting and their applications to address the challenges of simulation [2]. Gong, Zhang, Chen and Wang introduced the transmission assembly of heavy duty transport vehicle and the synchronizer components, and analyze the synchronization process [3]. They analyzed the factors actually affecting the synchronizer performance by running the gear shifting process on basis of real application conditions through developing the test rig. Gustavsson studied control system design, performed modelling and simulated the synchronization process of dual clutch transmission [4]. He focused to obtain a smooth shift instead of a fast shift and identified the biggest problem to control the sleeve position. Because of the complexity and shorter time of the synchronization process Hoshino developed a simulation model using ADAMS to clarify the abnormality in the shift reaction force during the upshift [5]. Kent created a model of synchronizer, selector system, driveline and transmission to identify the gearshift quality based on input parameters [6]. A shift feeling simulator was developed consisting of external and internal linkage dynamic models together with synchronizer and drivetrain models by Kim, Sung, Seok and kim [7]. A dynamic simulation model of the synchronizer by using ADAMS is presented by Liu and Tseng to analyze the abnormal impact during shifting and to provide the comfort and longer lifespan of the synchronizer [8]. Lovas dealt with synchromesh behavior and divided the process into eight phases [9]. He validated the results with measured data and studied different phenomena of stick-slip and double bump. Paffoni, Progri, Gras and Blouët studied the influence of radial and circumferential grooves on contact pressure, oil film load, coefficient of friction and transmitted torque [10]. A mathematical relationship of synchronization torque and index torque with significant parameters was established and validated experimentally by Razzacki and Hottenstein [11]. A concept of double indexing in single cone synchronizer has been introduced by Sandooja to increase the gear shifting performance [12]. Neto, Florencio, Rodrigues and Fernandez described concept and operation of synchronization step by step together with the description of working of gearshift lever and transmission components [13]. Yuming introduced working principle, designing formula and range of designing parameters of the synchronizer [14]. Häggström and Nordlander developed user friendly Matlab program for synchronization [15]. Häggström, Sellgren and Björklund have presented a numerical method to assess the performance of pre-synchronization. It is concluded from results of the numerical method that grooves on cones surfaces of synchronizer are more important than the grooves design [16]. Häggström, Stenström and Björklund developed a simulation model to measure the transient thermomechanical load of the synchronizer [17]. Häggström, Sellgren, Stenström and Björklund are studied effect of different external load and variation of design parameters on the temperature transient in the friction lining by using a generalized FE-based thermomechanical simulation model [18]. Häggström, Nyman, Sellgren and Björklund are developed a friction model for a lubricated molybdenum-steel contact by integrating the results from physical rig test and FEM simulation [19].

1.1 Aim, objectives and research questions

However, the above studies are not sufficient to find answer of the several questions. One of the important issue is to maintain the gear shift quality under different conditions. To start with study of the gear shifting mechanism there is a need of a tool to represent the shifting process thoroughly and deeply which is not develop in the research work done so far. One of the main aim of the project is to develop a mathematical model suitable for deep study of the dynamics of a generic synchronizer mechanism under different settings of master/slave and system operational scenarios. The model should be flexible to implement various realistic assumptions about friction between system components.

In particular the following research questions are in focus:

Is there any mathematical model to study the whole gear shifting process in detail? Is it possible to understand the complex movement of the bodies in synchronizer mechanism through the available models? Is it possible to capture the forces arise during interaction of the bodies?

What are the relevant objective functions to optimize the synchronization process?

Which structural and input parameters describing the synchronizer mechanism are influencing the gear shifting process?

What outcome can be expected from the results obtained from optimization technique?

Is the model sufficiently flexible and general to give opportunity to extend the research either on component connected with the synchronizer or for deep study of the synchronizer component?

1.2 Thesis outline

In chapter 1 a brief summary of the work, references of the research papers on synchronizer, aim and objectives, questions about the conducted research are outlined. In chapter 2 explanation of the synchronization process is given and later on the applied methodology to model the synchronization process is explained. Results of newly developed model of synchronization are presented. In chapter 3 problem statement of the Pareto optimization is given along the results from simulations. In chapter 4 appended papers are summarized briefly. The last chapter discussed the conclusions and outlook of the future work. Afterwards the published papers are attached.

2 Synchronizer and Synchronization

Transmission system transmits torque from engine to wheels of a vehicle at different driving conditions. There is a gearbox with different gear numbers inside the transmission system which matches the torque produced by engine with vehicle desire speed. It is clear that shifting between the gears is required for smooth driving of the vehicle. The mechanism used for gear shifting is called synchronizer. Still there is demand of automotive industry to complete the gear shifting process as quick as possible with smoothness. Before proceeding to analyze the gear shifting process for quickness and smoothness, in paper A the process is explained in detail together with the mathematical model. A short description of the gear shifting mechanism and the gear shifting process is given below.

The synchronizer can have different number of bodies with different shapes, and a variety of different concepts are used in vehicles today. To avoid specific existing commercially available synchronizers, this thesis will concern a generic synchronizer as shown in Figure 1. The synchronizer consists of three rigid bodies; sleeve, ring and gear. Basic purpose of the synchronizer is to engage the teeth of sleeve and gear which are rotating at different speed during shifting of the gear. The gear has its own speed and the sleeve has almost speed of the previously engaged gear because the driver first disengages the sleeve from the current gear, brings at neutral position and then moves the sleeve to next gear. Definitely if the teeth come in contact with different rotational speed, there will be clashing that will wear down the engaging teeth rapidly. So to reduce the speed difference between the engaging teeth there is a ring between the sleeve and the gear. In this synchronizer sliding friction is used to reduce the speed difference by making the frictional cones between two rotating bodies. The considered generic synchronizer has frictional cones between the sleeve and the ring. The engaging teeth during the time when frictional cones are sliding over each other and reducing the speed difference should not come in contact in order to produce clashing. For this purpose the ring and the gear have blocking chamfers which stops the sleeve to move further axially for engagement of teeth. When the speed difference approaches to zero, the blocking chamfer contact will release and the engaging teeth will come in full contact at end of the gear shift.

The gear shifting process, synchronization, is a complex process to figure out. To understand easily the synchronization process is explained in four phases which are further divided into several sub-phases (paper A). In the three body mechanism of synchronizer the sleeve is considered as a master which will retain its rotational speed throughout the synchronization, the gear behaves as a slave and the ring is free to rotate. In phase 1 which is called presynchronization the synchronizer prepares itself just to bring the friction cones together. When the shift force applies on the sleeve in phase 1a, it starts to move axially and cover the clearance between the sleeve and the ring before fluid contact of the cones. During phase 1b the ring gets its angular indexing position while in phase 1c the clearance finishes between chamfers of the ring and the gear. When the chamfers come in contact, the fluid starts to squeeze from the cones during the phase 1d. The phase 1e just shift the synchronizer from mixed friction phase to dry friction phase. It is considered that during phase 1f the shift force will overcome the spring axial resistance without considering the decrease in speed difference because of cones sliding. In short during phase 1 the ring gets its angular position and the fluid squeezes out from the cones. In phase 2 chamfers are in contact that’s why the sleeve cannot move further axially. Rotational speed difference decreases because of sliding of the cones over each other. At chamfers contact there are two main forces which are opposing each other in axial direction. One force is the gradually increasing axial force which is pushing the ring and the sleeve together to move further axially and second force is the gradually decreasing chamfers frictional force which is opposing the ring and the sleeve to not move axially further. When the first axial force cross over the second opposing force, the sleeve starts to move axially again with a surety of at least acceptable rotational speed difference which is the end of the phase 2. During the phase 3a, 3b and 3c the ring gets its axial indexing position, the engaging teeth chamfers come in contact and after the teeth leave chamfer contact respectively. In phase 4 the engaging teeth complete the gear shift after meshing completely. The all sub-phases are described in Figure 2. More detailed description of kinematics of synchronization process is presented in Paper A and in [21].

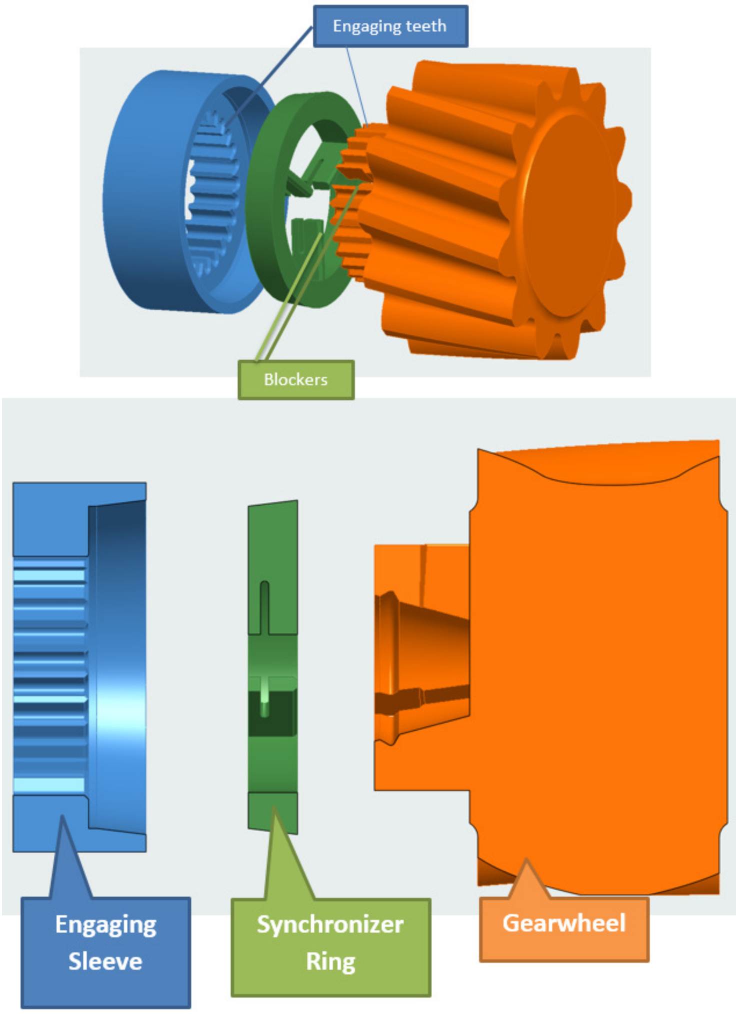

Figure 1: 3D and 2D front view of the generic synchronizer.

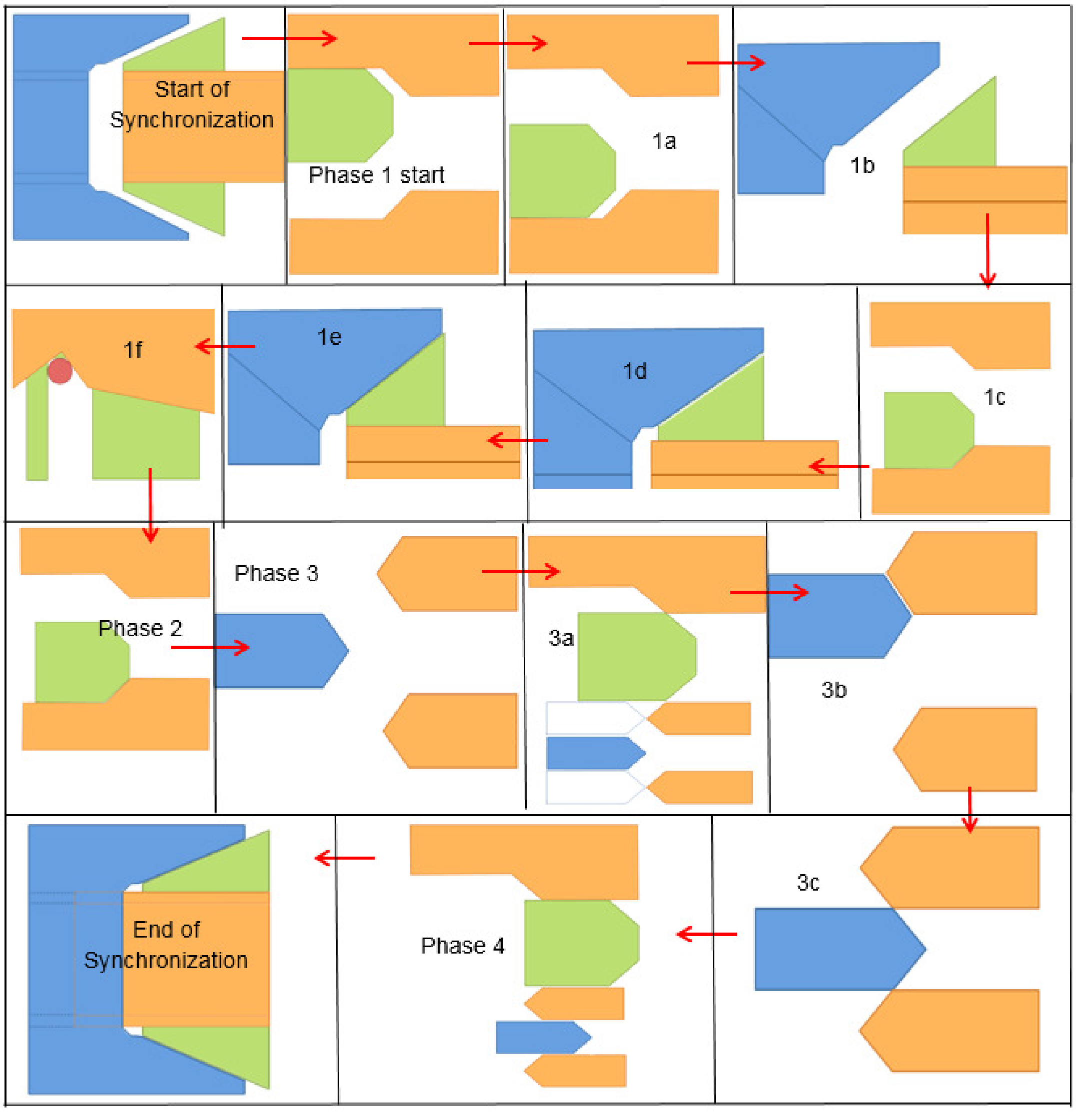

Figure 2: Sub-phases of the generic synchronization.

3 Methodology

The methodology used to develop the mathematical model for synchronization process of the generic synchronizer is given in paper A and [21]. Here we present elements of constrained Lagrangian formalism used to derive the equations of motion of a generic synchronizer mechanism.

Let's suppose multibody system comprises n bodies as shown in Figure 3. Some of the bodies are connected through kinematic constraints. Motion of some bodies get influence by other bodies.

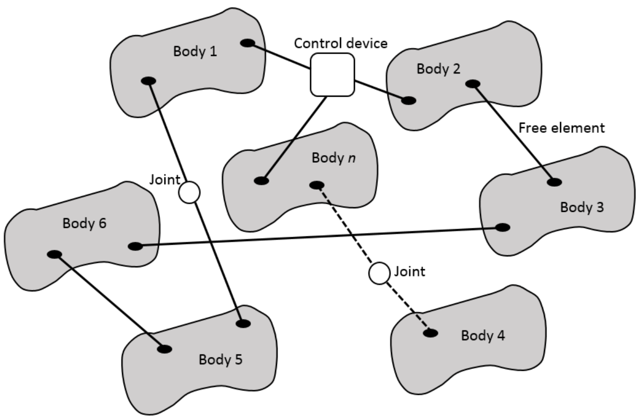

Figure 3: Multibody system.

Let the generalized coordinates of the multibody system is represented by $\boldsymbol{q} = [q_1\ q_2\ q_3\ .\ .\ .\ q_n]^T$ where n is the number of coordinates. Let also assume that the set of independent constraints is imposed on the system that can be represented by the following equations

$$\boldsymbol{C} = \left[C_1(\boldsymbol{q},t)\ \ C_2(\boldsymbol{q},t)\ \ .\ .\ .\ C_{n_c}(\boldsymbol{q},t)\right]^T = \boldsymbol{0} \tag{1}$$

If the equations of the constraints can be written in the above vector form, i.e. $\boldsymbol{C}\ (\boldsymbol{q},t) = \boldsymbol{0}$, the constraints as well as the respective system are called holonomic. In the holonomic system if t appears explicitly, the system is said to be rheonomic whereas if t does not appear explicitly, the system is said to be scleronomic. The constraints which can't be expressed in the form (1) and can be written as $\boldsymbol{C}\ (\boldsymbol{q},\dot{\boldsymbol{q}},t) = \boldsymbol{0}$ are called nonholonomic constraints.

According to the orthogonality theorem and Lagrange multiplier theorem [20] the constraint force can be written as

$$\boldsymbol{F}^c = -\,\boldsymbol{C}_{\boldsymbol{q}}^T\ \boldsymbol{\lambda} \tag{2}$$

Here $\boldsymbol{\lambda}$ is vector of the Lagrangian multipliers and $\boldsymbol{C}_{\boldsymbol{q}}^T$ is the constraints Jacobian of the system.

By introducing the Lagrangian L as

$$L = T - V$$

where V is the potential energy including strain energy and potential of any conservative external forces, the equations of motion of the multibody system can be written as follows

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\boldsymbol{q}}}\right)^T - \left(\frac{\partial L}{\partial \boldsymbol{q}}\right)^T + \boldsymbol{C}_{\boldsymbol{q}}^T\,\lambda = \boldsymbol{Q} \tag{3}$$

The equation (3) together with constraints (1) describe the motion of constrained multibody system and constitutes constrained Lagrangian formalism used to model the synchronization processes.

4 Synchronization

The developed mathematical model of a generic synchronizer mechanism went through validation test and used for modelling and analysis of dynamics of synchronization processes for different scenarios. In the thesis work the gear shifting process is simulated by the mathematical model where the sleeve is considered as a master and gear is considered as a slave at nominal condition. Values of the parameters to simulate the gear shifting process are considered as used in [21]. In Figure 4 when the shift force applies on the synchronizer, the sleeve starts to move axially but retains its rotational speed because of considering as a master. The gear loses its rotational speed during the phase 1a and 1b but its speed increases from phase 1c till end of the main phase 2 by increasing the shift force gradually. The ring gets its angular indexing position during the phase 1a and afterwards it rotates with the gear. During the phase 1f and 2 the sleeve does not move axially. The phase 2 ends with a speed difference. In the phase 3 the sleeve moves again axially and at end of the phase 4 it gets the full engagement position. More results of simulation of dynamics of a generic synchronizer mechanism for different scenarios and conditions (transmission vibrations, road grade, others) are presented in details in Papers A-C and [21].

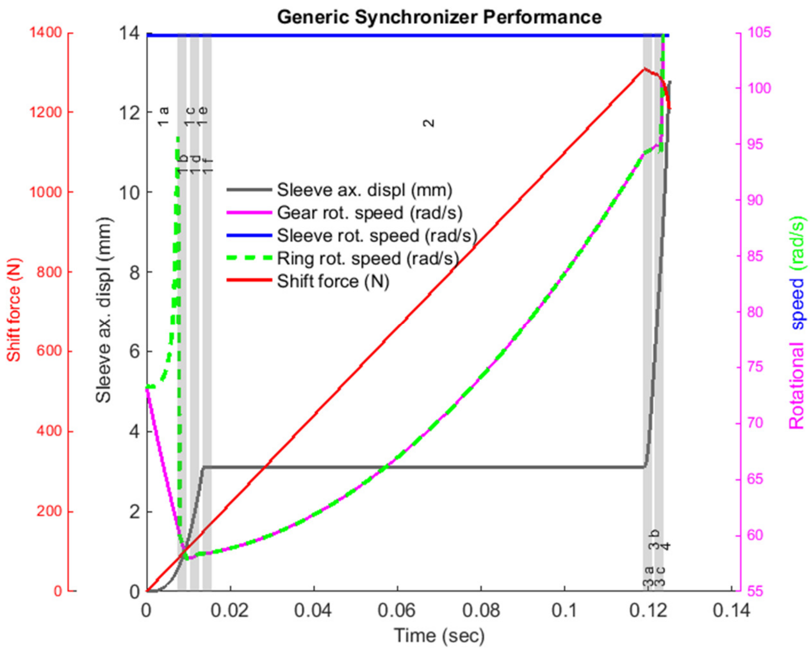

Figure 4: The generic synchronizer performance diagram.

Next question after studying performance of the synchronizer by the developed mathematical model is that how the performance will vary by changing parameters of the synchronizer. By the sensitivity analysis effects of the parameters are studied in chapter 6.

5 Implementation of the friction model

One of the advantages of the mathematical model based on CLF is the adoptability that means other relevant model to the synchronization can be applied in the developed mathematical model. In this task the friction model proposed in [19] is applied. The synchronization time is compared obtained from the developed CLF based mathematical model and from mathematical model of friction used in [19].

Fourier series is used to make an expression for the coefficient of friction in [19]. The Fourier series expression of coefficient of friction is applied in the developed model and values of the coefficients are given below.

$a_o = -1.6096e^{08}$, $a_1 = 2.5715e^{08}$, $a_2 = -1.2478e^{08}$, $a_3 = 2.7386e^{07}$, $a_4 = 5.0845e^{06}$, $a_5 = -5.0180e^{06}$, $a_6 = 1.2499e^{06}$, $a_7 = -1.1006e^{05}$, $b_1 = 1.1733e^{08}$, $b_2 = -1.4381e^{08}$, $b_3 = 9.2911e^{07}$, $b_4 = -3.5729e^{07}$, $b_5 = 7.8406e^{06}$, $b_6 = -8.0752e^{05}$, $b_7 = 1.6157e^{04}$, $w = 3.0897$

where *w* is the fundamental frequency.

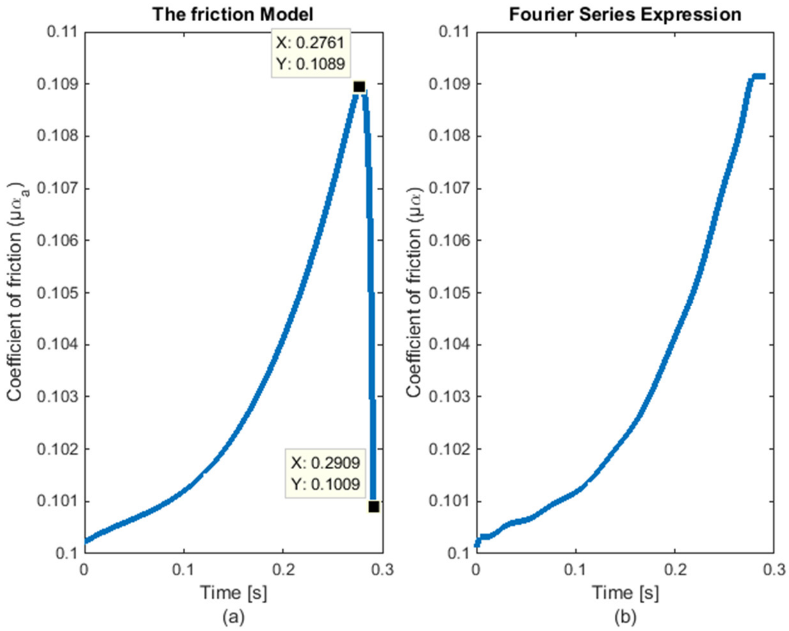

Figure 5: Coefficient of friction in [19] and by Fourier series.

μ_{α_a} in Figure 5 (a) is the coefficient of friction between the cones proposed in [19] and μ_α in Figure (b) is the coefficient of friction obtained by the Fourier series. μ_α has constant value after the maximum value as shown in Figure 5 (b) because it is assumed in Figure 5 (a) the synchronization process has completed till the maximum value.

Five common variables in the developed mathematical model and friction model proposed in [19] are taken as input variables. Synchronization time is obtained at two particular values as shown in Table 1. Synchronization time at nominal values of the input variables by [19] is 0.2957 sec and by the developed model is 0.3334 sec.

	Common input variables	Nominal values of input variables	Particular values of input variables	Synchronization time (sec) in [19]	Shift time (sec) by the developed model
1	Inertia	0.42	0.3-0.7 (kgm^2)	0.2499-0.3829	0.2894-0.3977
2	Initial Speed difference	60	10-85 (rad/s)	0.1016-0.3570	0.1515-0.3903
3	Rate of shift force	7000	7000-20000 (N/s)	0.2957-0.1841	0.3334-0.1970
4	Cone radius	0.085	0.05-0.12 (m)	0.3777-0.2510	0.3977-0.2827
5	Cone angle	6.5	5-15 (degree)	0.2598-0.4554	0.2942-0.3980

Table 1: Implementation of the friction model into the developed model on synchronization time.

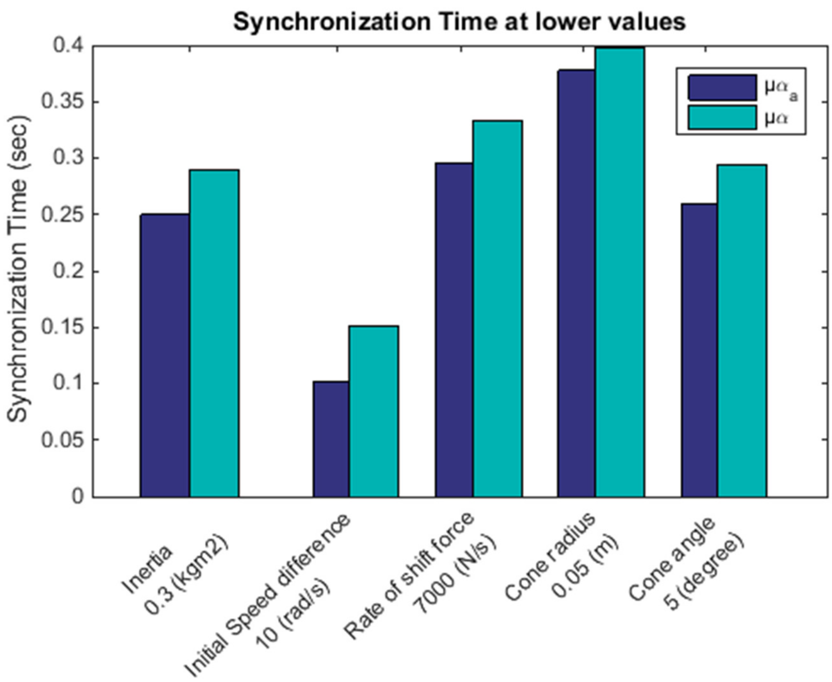

Figure 6: Synchronization time at lower values of common variables in [19] and the developed model.

The synchronization time in Figure 6-7 and in Figure with μ_α is higher than the synchronization time with μ_{α_a} because the developed CLF based mathematical model describes more detail kinematics of the generic synchronization by several sub-phases than the model presented in [19].

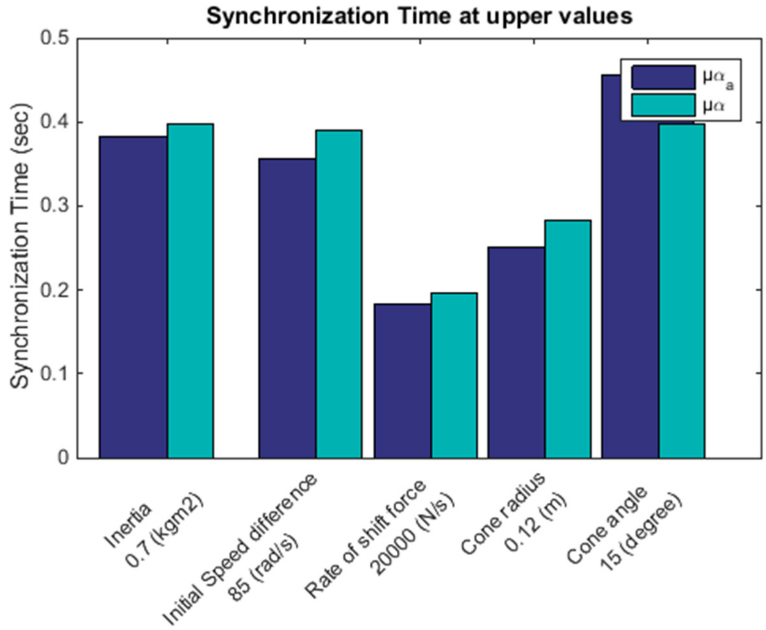

Figure 7: Synchronization time at upper values of common variables in [19] and the developed model.

6 Sensitivity analysis

One of the main objectives under all circumstances is to shift the gear as quick as possible. But it is also demanded to make the gear shift as smooth as possible. Synchronization time and speed difference are taken as objectives which should be minimized. Synchronization time is considered as quickness and speed difference is considered as smoothness of the synchronization process.

To this end two objectives of the generic synchronizer are defined as follows

1. Synchronization time; time from neutral to full engagement of the teeth $\left(t_{synch}\right)$ as a measure of quickness.
2. Speed difference; the difference between rpm of the master and the slave at end of the phase 2 from where the sleeve starts to move axially again $\left(\Delta\omega_{s-g}\right)$ as a measure of smoothness.

The developed mathematical model has been used for sensitivity analysis of dynamics of synchronization processes with respect to structural system parameters as well as shift force. Sensitivity analysis with eight parameters is given in paper C. In Figure 8 and Figure 9 degree of influence of five parameters: cone angle, cone coefficient of friction, rate of shift force, blocker angle and blocker coefficient of friction, is analyzed with respect to synchronization time and speed difference at end of the phase 2.

In Figure 8 synchronization time increases with increasing cone angle and blocker angle but the time decreases with increasing cone coefficient of friction, rate of shift force and blocker coefficient of friction. Change in the synchronization time is approximately 0.05 sec within prescribed range of blocker angle and rate of shift force. But within prescribed range of other parameters, change in the synchronization time is approximately 0.09 sec.

The objective functions t_{synch} and $\Delta\omega_{s-g}$ have contradicting behavior with the parameters. For instance t_{synch} increases with increasing cone angle but decreases with increasing rate of shift force. $\Delta\omega_{s-g}$ has opposite trends than t_{synch} with cone angle and rate of shift force. With such a kind of conflicting behaviour of objective functions with parameters it is not easy to find optimal values of the parameters of the synchronizer. The Pareto optimization is taken into account to find the optimal parameters as explained in chapter 7.

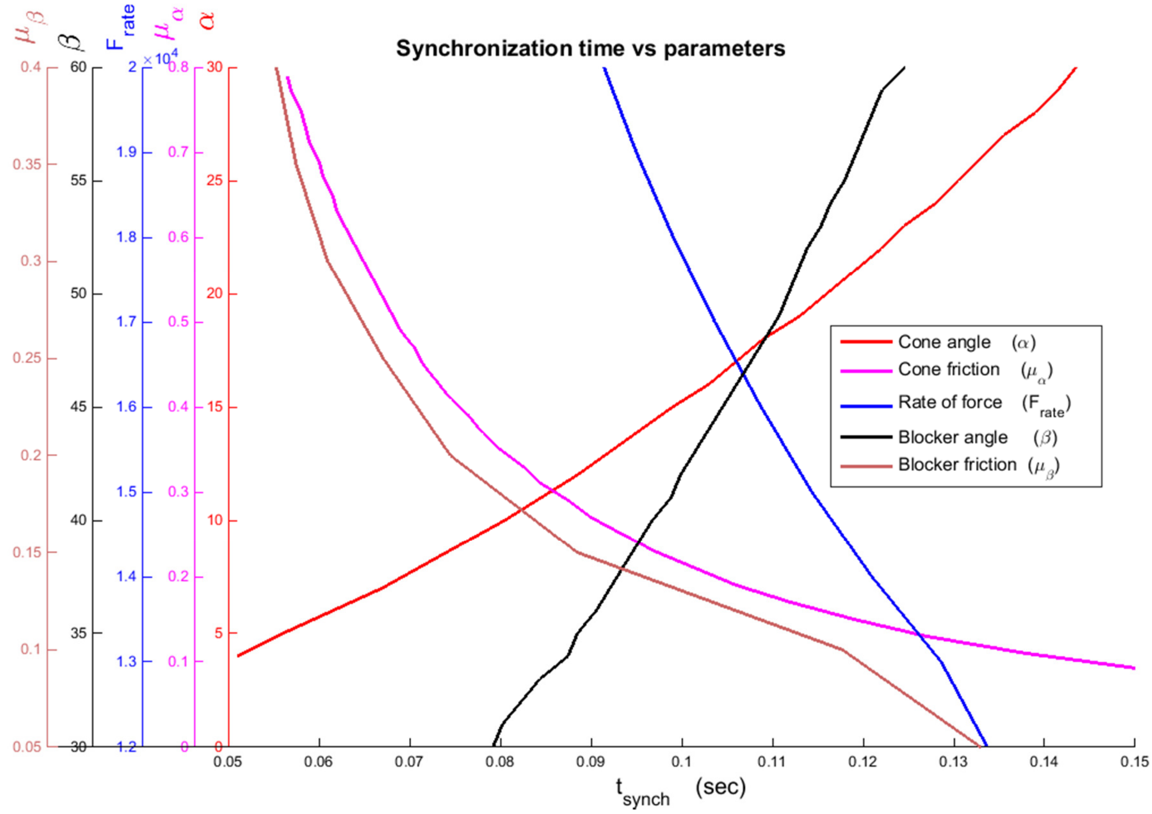

Figure 8: Effect of five parameters on synchronization time.

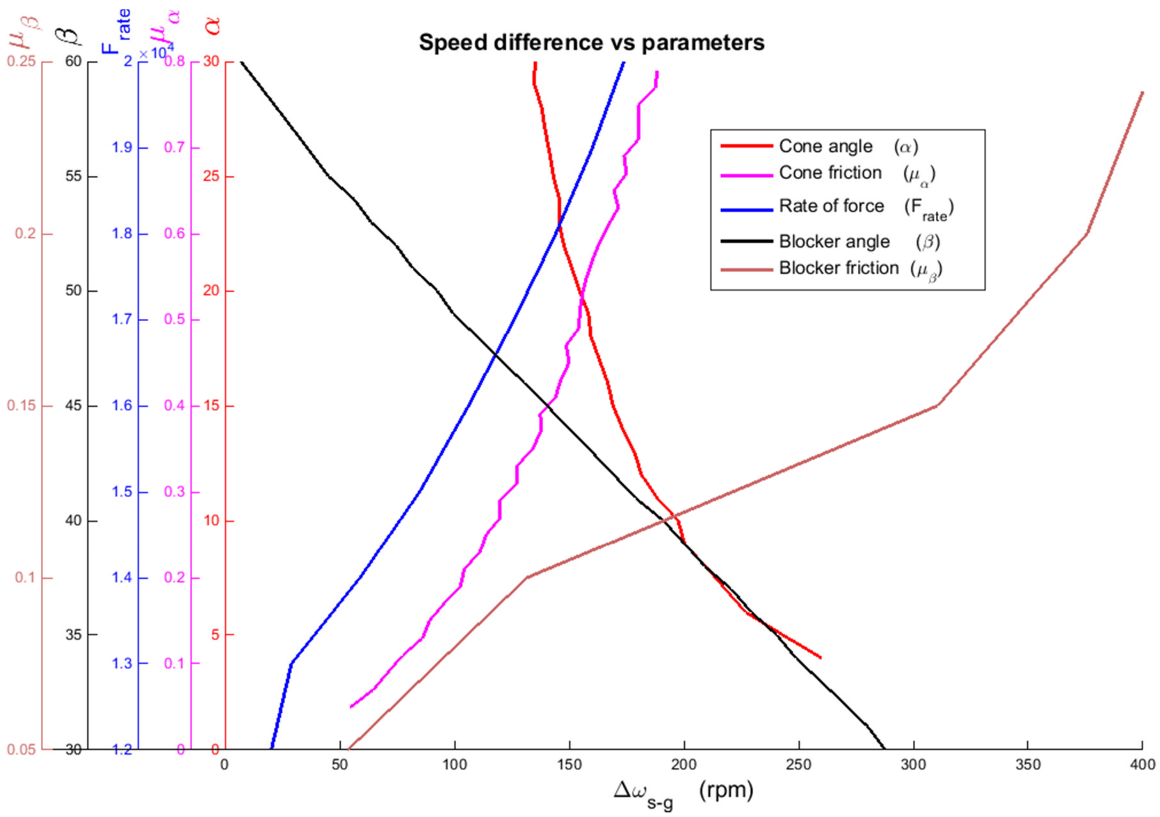

Figure 9: Effect of five parameters on speed difference.

7 Pareto optimization

The results of optimizations of the synchronization process at eight different cases are presented in paper C. Here a brief description is given about performing optimization of the synchronization process.

Two objective functions (quality factors) are taken into account to measure performance of the synchronizer; first is synchronization time, t_{synch}, and second is speed difference, $\Delta\omega_{s-g}$, at end of the phase 2. Both objective functions need to be minimized at the same time for robustness of the synchronization. Eight parameters are considered as input parameters which are cone angle (α), cone coefficient of friction (μ_α), shift force (F_{rate}), blocker angle (β), blocker coefficient of friction (μ_β), cone radius (r_α), gear moment of inertia (I_g) and ring moment of inertia (I_r). Before applying the Matlab routine of multi-objective optimization with genetic algorithm two constraints are defined. The constraint of wedging condition ($\tan(\alpha) > \mu_\alpha$) means tangent of cone angle must be greater than cone coefficient of friction. The second constraint correspond to the condition that cone torque (T_c) must be greater than the index torque (T_I) otherwise the synchronizer will produce clashing all the time.

Mathematical statement of the problem is given below

$$\begin{cases} \min\limits_{\boldsymbol{q}}\left(t_{synch}(\boldsymbol{q}), \Delta\omega_{s-g}(\boldsymbol{q})\right) \\ \quad \begin{cases} \tan(q_1) > q_2 \\ T_c > T_I \end{cases} \\ \quad \boldsymbol{q}_l < \boldsymbol{q} < \boldsymbol{q}_u \end{cases}$$

where $\boldsymbol{q} = [\, q_1, q_2, q_3, q_4, q_5, q_6, q_7, q_8]^T = [\, \alpha, \mu_\alpha, \dot{F_s}, \beta, \mu_\beta, r_\alpha, I_g, I_r]^T$,

$\boldsymbol{q_l} = [\, 6{,}0.1{,}6000{,}40{,}0.05{,}0.1{,}0.1{,}0.003]^T$, $\boldsymbol{q_u} = [\, 15{,}0.5{,}20000{,}60{,}0.5{,}0.5{,}0.5{,}0.01]^T$

$$T_c = \frac{F_s\, \mu_\alpha\, r_\alpha}{\sin(\alpha)}\,, \qquad T_I = r_\beta\, F_s \left(\frac{1 - \mu_\beta \tan(\beta)}{\mu_\beta + \tan(\beta)}\right)$$

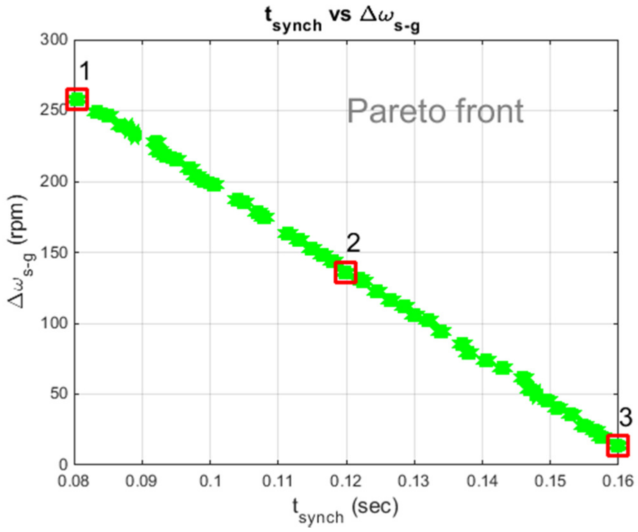

Figure 10: Pareto front between synchronization time and speed difference.

Pareto front is shown in Figure 10 between the synchronization time t_{synch} and the speed difference $\Delta\omega_{s-g}$. Both objective functions t_{synch} and $\Delta\omega_{s-g}$ have contradicting behavior with each other. The synchronization selected at point 1 has minimum t_{synch} but highest $\Delta\omega_{s-g}$ and the synchronization selected at point 3 has highest $\Delta\omega_{s-g}$ but minimum t_{synch}. The point 2 is tradeoff between $\Delta\omega_{s-g}$ and t_{synch}.

Figure 11 shows variations of the Pareto sets along the objective functions t_{synch} and $\Delta\omega_{s-g}$. Results of the Pareto optimization in Figure 11 show optimal values of the parameters at which the synchronizer can give optimal performance. Percentage change and scattering behavior in the results indicate degree of influence of the parameter on objective functions. F_{rate} has smooth trend and highest percentage change in values (see Figure 11, h) but other pareto sets have non-smooth trends and less percentage changes in values along the objective functions. F_{rate} has highest degree of influence on the objective functions than rest of the parameters ($\alpha, \mu_\alpha, \beta$ and μ_β, I_r, I_g and r_α). In Figure 11 (f) most of the points are along the single value of cone angle which is almost 8 degree and in Figure 11 (g) along single value of cone coefficient of friction which is almost 0.14. This kind of trend of the particular parameters shows that different values of the objective functions can be obtained at single value of the particular parameter while changing the values of rest of the parameters or in other words the particular parameters can be ignored from the optimization process if the values of these parameters are selected at a point explained above.

More results on sensitivity analysis of synchronizer dynamics and Pareto optimization are presented in Paper C.

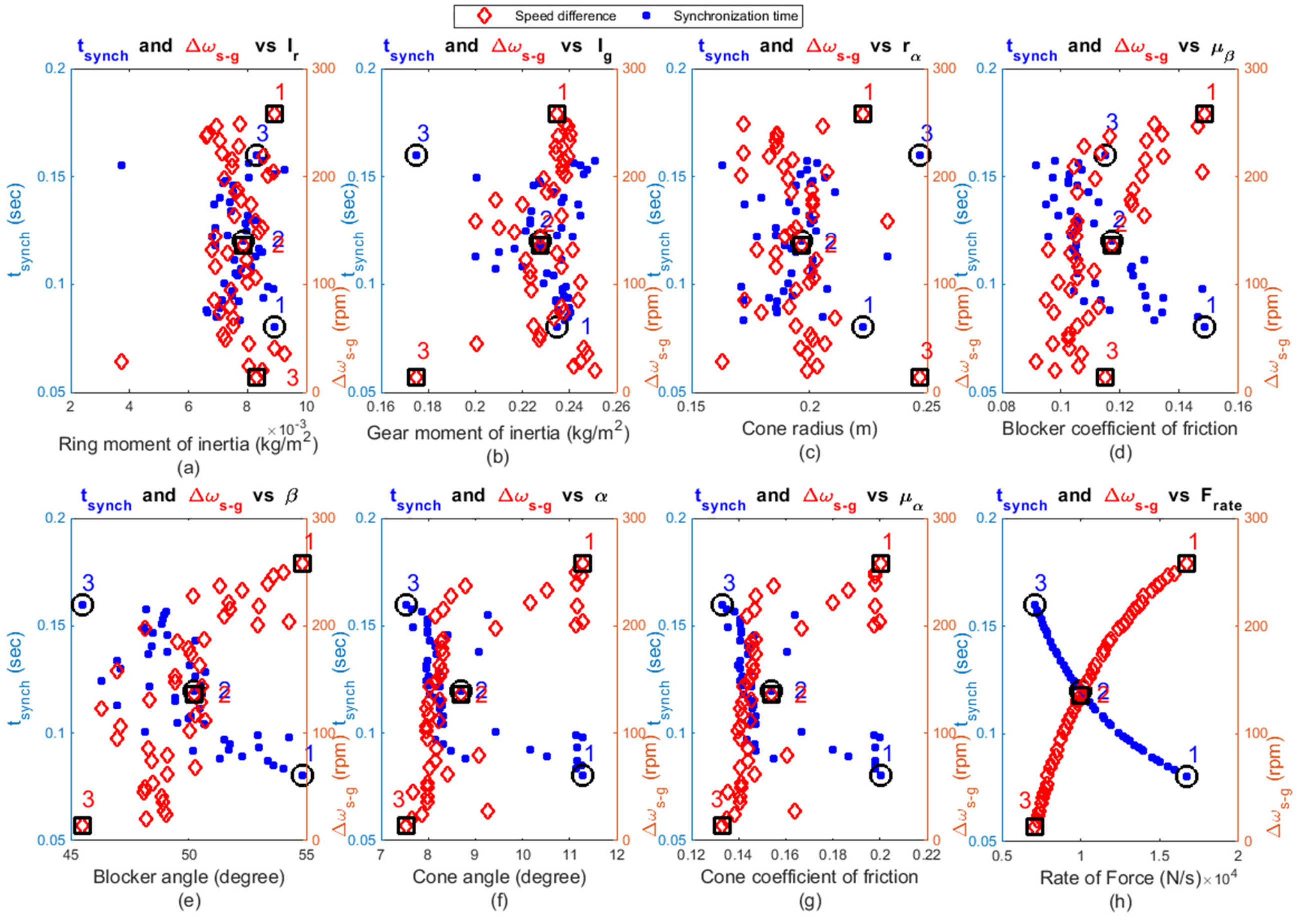

Figure 11: Pareto sets along synchronization time and speed difference.

8 Graphical user interface in Matlab

Sometimes it is quite tedious to operate with the developed mathematical model by handling the input variables. A graphical user interface is developed in Matlab where the user can easily select values of the five variables (cone angle, cones coefficient of friction, rate of force, blocker angle and cones radius) as shown in Figure 12 and can see the synchronization performance diagram. When user select values of the variables, graphical user interface call the developed mathematical model for synchronization process and display the results. Graphical user interfaces for sensitivity analysis of the input variables with respect to the speed difference and the synchronization time are developed as shown in Figure 13 and Figure 14.

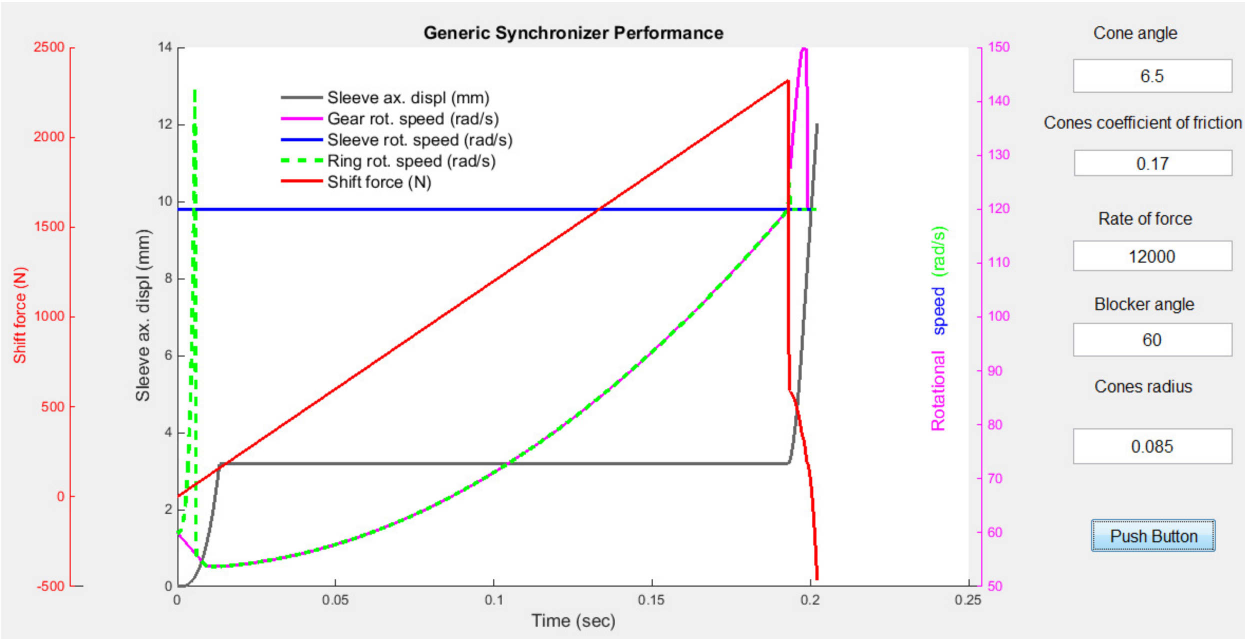

Figure 12: Graphical user interface of the generic synchronization in Matlab.

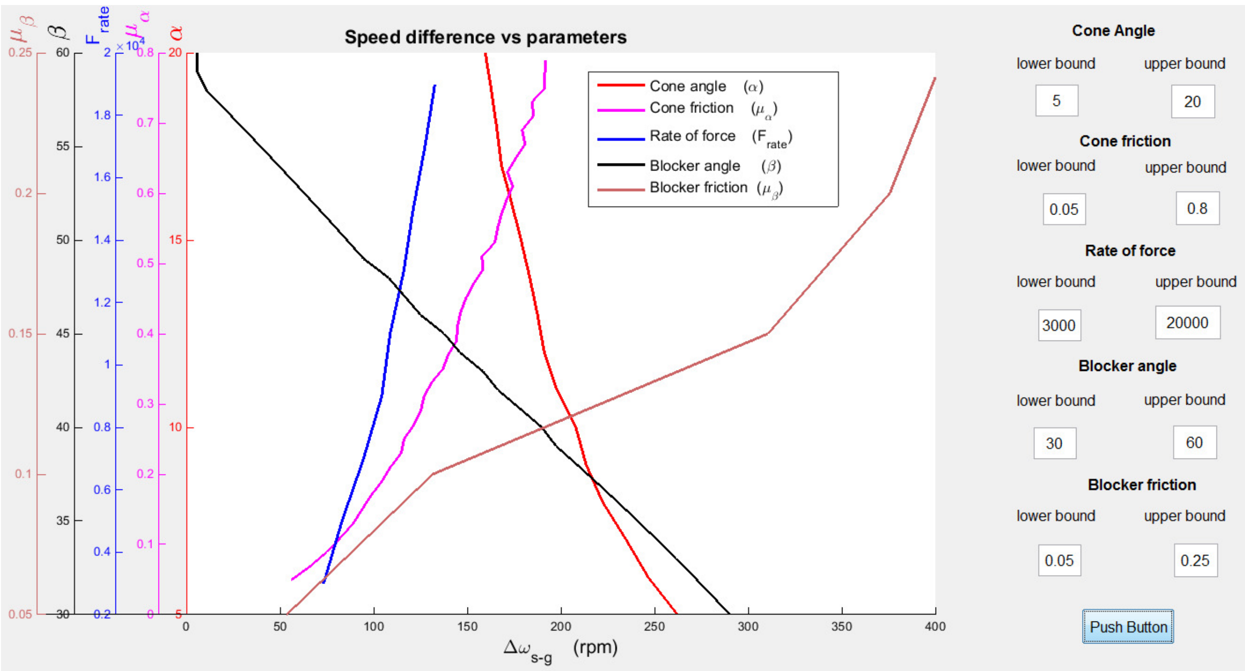

Figure 13: Graphical user interface of sensitivity analysis with respect to speed difference in Matlab.

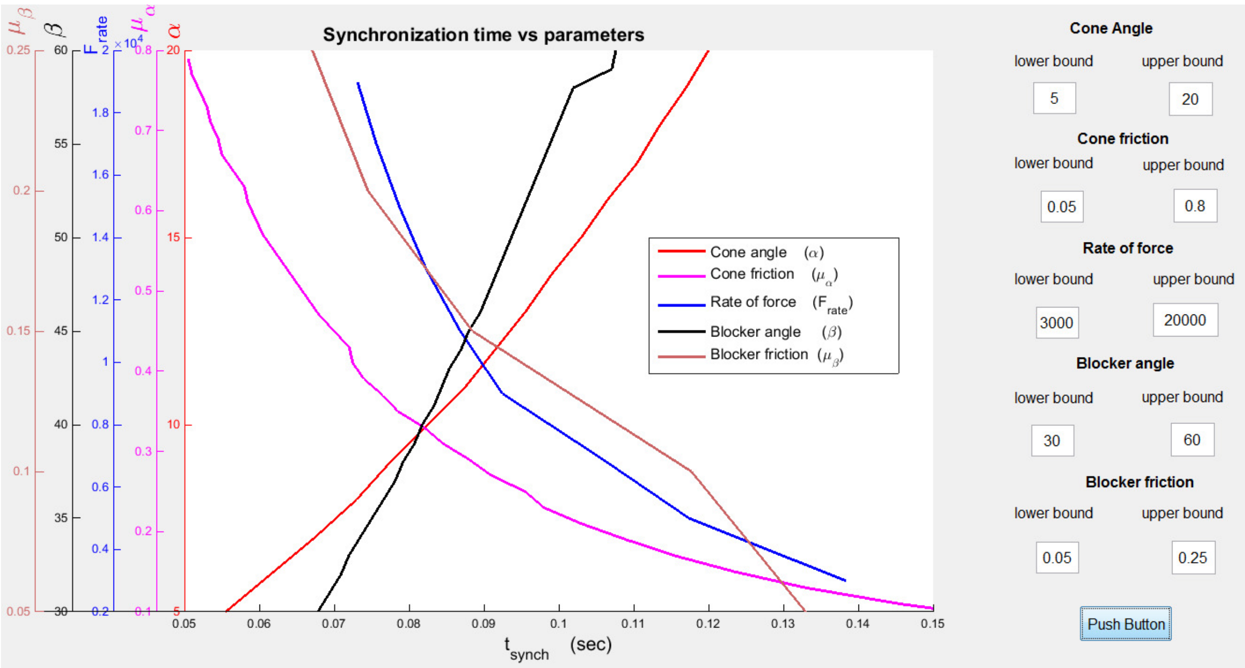

Figure 14: Graphical user interface of sensitivity analysis with respect to synchronization time in Matlab.

9 Summary of Appended Papers

Paper A: Modelling of Heavy Vehicle Transmission Synchronizer using Constrained Lagrangian Formalism.

The generic synchronization process is a complex phenomenon because of design of the bodies (sleeve, ring, gear) and their movement. Constrained Lagrangian formalism (CLF) explained the whole synchronization process in a unified manner with unilateral or/and bilateral constraints. Mathematical model of used CLF turns out into differential-algebraic equations to model the kinematics and kinetics of the generic synchronizer. Before solving the equations of motions by using the numerical algorithm, the sleeve is considered as a master and the gear is considered as a slave. Validation of modified form of the generic synchronizer with available experimental test rig predicted reasonable accuracy of the developed mathematical model of using CLF. Effect of sleeve vibrational motion, cone angle, cone coefficient of friction and shift force on synchronization time is also analyzed.

Paper B: Dynamics and Pareto Optimization of a Generic Synchronizer Mechanism.

In Paper B, the optimization problem to improve the shift quality by shifting the gear as quick as possible and as smooth as possible is addressed. Two objectives are selected as measures of shift quality; one is synchronization time and another is speed difference at end of the main phase. Speed difference is the main characteristic to measure smoothness of the generic synchronizer. To contribute to the goal effect of three parameters (cone angle, cone coefficient of friction and rate of applying shift force) are studied on two objectives. It is concluded that both objectives have contradicting behavior. Synchronization time decreases but speed difference increases with increasing friction coefficient and rate of force. Whereas synchronization time increases and speed difference decreases with increasing cone angle. Because of conflicting behavior of the objectives optimization is performed by taking the three parameters as inputs. Optimized values of the parameters in form of Pareto sets and trade-off values of the objectives in form of Pareto front are presented.

Paper C: Performance Control of the Transmission Synchronizer via Sensitivity Analysis and Parametric Optimization.

Time duration of the gear shifting for heavy vehicles sometimes exceeds than the normal time duration. The phenomenon of abnormal gear shifting is expected to occur more frequently during different operating conditions. Besides the nominal operating condition effect of the vibrational motion and road grade are also studied on the gear shifting mechanism (synchronizer). Delay in the gear shifting (synchronization) puts more burden on the engine to work smoothly that results in more emissions. To optimize the gear shifting process the research is conducted by optimizing parameters of the mechanism. Before starting the optimization the developed mathematical model of the gear shifting mechanism is validated against the available test rig. Synchronization time and speed difference are selected as performance measures of the synchronizer. Through sensitivity analysis it is found that the eight selected parameters are prominent parameters to measure the performance of the synchronizer. The synchronizer mechanism can work in three kind of settings by considering the sleeve as a master, the gear as a master and both as slaves. Optimization is performed with different operating conditions under three type of settings of master and slave. It is concluded that instead of taking eight parameters together only the highest influencing parameters are sufficient to measure the performance of the synchronizer.

10 Conclusions and outlook of future work

10.1 Conclusions

It is concluded that the developed mathematical model based on CLF can represent the synchronization process. The model is validated by the available experimental setup. The developed model can not only describe in detail the synchronization process by sub phases but also can provide the opportunity to implement other relevant mathematical models. For example the friction model used in [19] is applied in the mathematical model. Parameters: cone angle, cone coefficient of friction, applied shift force, blocker angle, blocker coefficient of friction, cones radius, gear moment of inertia and ring moment of inertia are considered as input parameters. The synchronization time (quickness) and the speed difference (smoothness) are considered as objective functions in the sensitivity analysis. It is predicted from the sensitivity analysis that both objective functions have same trends with increasing cone radius, gear moment of inertia and ring moment of inertia but both objectives have opposite trends with increasing rest of the parameters. Because of the conflicting behavior between quickness and smoothness Pareto optimization is performed to determine optimal values of the parameters through which the synchronizer can perform optimally. In addition to optimal values of the parameters it is also found from Pareto optimization analysis that the robust gear shifting process can be achieved by just taking into account the parameters which have higher degree of influence instead of taking all effecting parameters. The most influencing parameters are different in different cases with different settings of master/slave and with different conditions of vibrations, road grade and nominal. For instance applied shift force, cones angle, cones coefficient of friction, blocker angle and blocker coefficient of friction are the highest influencing parameters where the sleeve is considered as a master at nominal condition. At conditions of vibrations the ring moment of inertia replaces the blocker angle and at condition of road grade the cones radius replaces the cones angle among the highest influencing parameters. But some of the parameters are common in some cases which have highest degree of influence. For example the applied shift force is among the highest influencing parameters in the cases where the gear or the sleeve is considered as a master. In short the research work has contributed a mathematical model of the synchronization process and found the parameters with their optimal values which can give optimal performance of the gear shifting process. To deal easily with the mathematical model a graphical user interface is developed where the user can choose values of the input variables and can see the synchronization performance diagram.

10.2 Future work

It is suggested to verify the developed synchronizer model and relevant results with industrial simulation tool. The multi-objective optimization at different load cases will be studied in future. Idea of concentrating on the presynchronization phase in order to decrease time of the phase has also given for future work.

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