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The Influence of Electronic Stability Control, Active Suspension, Driveline and Front Steering Integrated System on the Vehicle Ride and Handling

By Ahmed Elmarakbi, Chandrasekaran Rengaraj, Alan Wheatley
& Mustafa Elkady

University of Sunderland

Abstract - The main aim of this paper is to investigate the influence of integration of vehicle dynamics control systems by proposing new control architecture to integrate the braking, steering, suspension and driveline. A 6 DoF nonlinear vehicle handling model is developed for in Matlab/Simulink. The modelling work contains linear suspensions with nonlinear tyres suitable for combined slip conditions. The results are validated against commercially available vehicle dynamics software. All the four active chassis control systems are evaluated over the entire range of vehicle handling region in standalone mode through simulation. Based on the analysis of these four standalone controllers, a novel rule based integration strategy is proposed to improve the vehicle handling. A comparison of the proposed integrated control strategy and the standalone control strategy is carried out and the results of the simulation are found to prove that the integrated control strategy improves vehicle stability across the entire vehicle operating region.

Keywords : vehicle dynamics, integrated control systems, vehicle ride and handling, vehicle modeling, and numerical simulations.

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The Influence of Electronic Stability Control, Active Suspension, Driveline and Front Steering Integrated System on the Vehicle Ride and Handling

Ahmed Elmarakbi ^α, Chandrasekaran Rengaraj ^σ, Alan Wheatley ^ρ & Mustafa Elkady ^ω

Abstract - The main aim of this paper is to investigate the influence of integration of vehicle dynamics control systems by proposing new control architecture to integrate the braking, steering, suspension and driveline. A 6 DoF nonlinear vehicle handling model is developed for in Matlab/Simulink. The modelling work contains linear suspensions with nonlinear tyres suitable for combined slip conditions. The results are validated against commercially available vehicle dynamics software. All the four active chassis control systems are evaluated over the entire range of vehicle handling region in standalone mode through simulation. Based on the analysis of these four standalone controllers, a novel rule based integration strategy is proposed to improve the vehicle handling. A comparison of the proposed integrated control strategy and the standalone control strategy is carried out and the results of the simulation are found to prove that the integrated control strategy improves vehicle stability across the entire vehicle operating region.

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1. INTRODUCTION

Mechatronics and active control systems are playing an ever increasing role in automobiles. Modern vehicles typically include more than 40 actively controlled systems that play a major role in vehicle directional stability, ride comfort and safety. At present these systems generally work independently but it is widely accepted that integration of these standalone systems will lead to improved vehicle dynamic performance. Additional benefits include cost and weight reductions and reduced sensor requirements.

Both the automotive industry and the end users will benefit directly from this research. However, successful integration of such control systems is still largely in the research phase. Previous studies have identified that these systems were originally developed independently to perform specific tasks and some

Systems do co-exist, Junjie et al. (2006), Karbalei et al. (2007), and Kazuya et al. (2000). Researchers have succeeded in the successful integration of several systems, March and Shim (2007); however, potential conflicts are still a problem. Complete integration of many sub-systems is still a real technical challenge.

The overall aim of this research is to develop new control strategies/algorithms to enable successful integration of a subset of vehicle control systems. However, this paper focuses primarily on the methods of improving vehicle stability and emergency handling through the integration of four specific vehicle control systems: Active Front Steering (AFS), Active Suspension (AS), brake-based Electronic Stability Control (ESC) and driveline based Variable Torque distribution (VTD) system.

There are many ways to compare the performance improvement obtained by an integrated chassis controller against its standalone counterpart. Few of the techniques include comparing the reduction in energy consumption, reduced cost, less/modular parts, improvement in performance variable etc. In this paper, the improved performance objectives established from using the integrated chassis control approach are defined as a reduction in yaw rate and vehicle side-slip angle that lead to better handling capabilities.

The main aim of this paper is to investigate the influence of integration of vehicle dynamics control systems by proposing new control architecture to integrate the braking, steering, suspension and driveline. The active control systems investigated include brake-based electronic stability control (ESC), active suspension (AS) and active front steering (AFS) and variable torque distribution (VTD). The paper is organized as follows. In the vehicle modeling section, a detailed passive vehicle dynamics model with nonlinear tires suitable for combined slip and transient conditions is developed in Matlab/Simulink environment along with the dynamics of steering, braking, suspension and driveline systems.

In the standalone control systems section, the development of standalone control system models of active front steering, active suspension, a brake based

Author α σ ρ ω : Department of Computing, Engineering and Technology, Faculty of Applied Sciences, University of Sunderland, Sunderland SR60DD, UK.
E-mail : ahmed.elmarakbi@sunderland.ac.uk

electronic stability control and a variable torque distribution system are discussed. Various possible integrated control strategies amongst those systems in consideration are analysed and investigated in the integrated control system section. This section also explains a new integrated control strategy (ICC) developed from the results of the analysis and implemented then in MATLAB/Simulink. Finally the conclusions based on the new ICC strategy are presented.

II. VEHICLE MODELLING

A detailed study on various vehicle models available in literature was conducted for their suitability in this research. Considering the vehicle operations in a wide lateral acceleration range and interactions amongst various degrees of freedom (DoF) of vehicle, a non-linear vehicle handling model with 6 DoF for the chassis has been developed for this research. The ISO vehicle axis system is assumed throughout the modelling process. It is assumed that the steering angles of both front wheels are considered identically, the effect of un-sprung mass is only considered in the vertical direction and ignored in the vehicle's lateral and longitudinal directions and the tires and suspension remain normal to the ground during vehicle maneuvers. The chassis equations of motion based on Newton's laws can be derived as follows:

$$m_v(\dot{V}_x - V_y\dot{\psi} + V_z\dot{\theta}) = F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr} - (F_{yfr} + F_{yfl})\delta \quad (1)$$

$$m_v(\dot{V}_y - V_z\dot{\phi} + V_x\dot{\psi}) = F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr} + (F_{xfl} + F_{xfr})\delta \quad (2)$$

$$m_s(\dot{V}_z - V_x\dot{\theta} + V_y\dot{\phi}) = F_{sfl} + F_{sfr} + F_{srl} + F_{srr} - m_s g \quad (3)$$

$$J_r\ddot{\theta} = \frac{t}{2}(F_{sfl} + F_{srl}) - a(F_{sfl} - F_{sfr}) - h(F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}) \quad (4)$$

$$J_p\ddot{\phi} = b(F_{srl} + F_{srr} - F_{sfl} - F_{sfr}) + h(F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr}) \quad (5)$$

$$J_y\ddot{\psi} = a(F_{yfl} + F_{yfr}) - b(F_{yrl} + F_{yrr}) + \frac{t}{2}(F_{xfr} + F_{xrr} - F_{xfl} - F_{xrl}) \quad (6)$$

Figure 1 shows the schematic of the vehicle model used in this study. The vehicle model is divided into sub-models that describe the wheel, brake, suspension and steering dynamics.

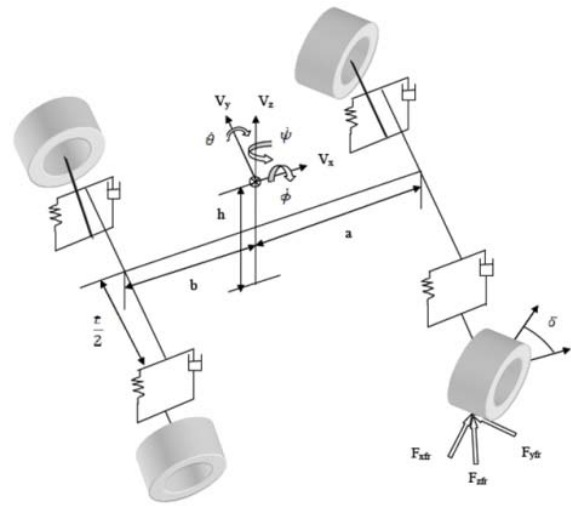


Fig. 1 : Vehicle Model

The dynamics of the tire-road interaction are dependent on the lateral and longitudinal tire slips. The lateral tire slip angles for each wheel can be calculated as follows:

$$\alpha_i = \tan^{-1}\left(\frac{V_y + a\dot{\psi}}{V_x \mp \frac{t}{2}\dot{\psi}}\right) - \delta; \quad \alpha_j = \tan^{-1}\left(\frac{V_y - b\dot{\psi}}{V_x \mp \frac{t}{2}\dot{\psi}}\right) \quad (7)$$

The component of the vehicle velocity of the wheel centre that is parallel to the wheel vertical plane is given as

$$V_i = \cos(\alpha_i) \sqrt{(V_x \mp \frac{t}{2}\dot{\psi})^2 + (V_y + a\dot{\psi})^2} \quad (8)$$

$$V_j = \cos(\alpha_j) \sqrt{(V_x \mp \frac{t}{2}\dot{\psi})^2 + (V_y - b\dot{\psi})^2} \quad (9)$$

The longitudinal wheel slip is defined as

$$\lambda_{ij} = \begin{cases} \frac{R_w \omega_i - V_{xi}}{R_w \omega_i}, & (\text{driving}) \\ \frac{V_{xi} - R_w \omega_i}{V_{xi}}, & (\text{braking}) \end{cases} \quad (10)$$

Capturing the tire behavior is probably the most difficult and important problem to tackle while building a vehicle model as realistically as possible. In the past a lot of different models have been created to solve this problem. The most realistic models are the most complicated but probably they are not useful in every kind of research. On account of our objectives, a too simple model is not applicable because it can provide correct results only if the slip angles are very small, but it cannot represent for example the forces the tires transfer during an emergency handling manoeuvre. For this reason a semi-empirical model usually called

"Magic formula", suggested by Pacejka and Besselink, (1997) is chosen which proved to be a good model but not too complicated. For simplification the camber has been set to zero in the current vehicle model. The general equation of the tire model is

$$y(x) = D \sin[C \tan^{-1}(Bx - E(Bx - \tan^{-1}(Bx)))] \quad (11)$$

Where $y(x)$ is F_x and F_y , respectively, if x is λ or α . The tire forces generated using the above equations in longitudinal and lateral directions are a function of pure slips in their respective directions. But in reality, these tire forces are generated as a function of combined slip that exists during typical combined braking and cornering situations such as braking before entering a corner and accelerating before exiting it. Weighing functions G as described by Bakker (1987) are introduced which when multiplied with the original pure slip functions produce the interactive effects of longitudinal slip on F_y and lateral slip on F_x . The cosine version of the magic formula is used to represent the hill shaped weighing function, G :

$$G = D \cos[C \tan^{-1}(Bx)] \quad (12)$$

The combined side force is described by the following formulae:

$$F_y = G_{yk} \cdot F_{yo} + S_{vyk} \quad (13)$$

Where, S_{vyk} , the effect due to ply-steer is assumed in this paper to be zero to reduce the complexity. The function G_{yk} is used as described by Rajamani and Hedrick (1995). And, the combined side force is described by the following formulae:

$$F_x = G_{x\alpha} \cdot F_{xo} \quad (14)$$

where G_{yk} and $G_{x\alpha}$ are described as follows:

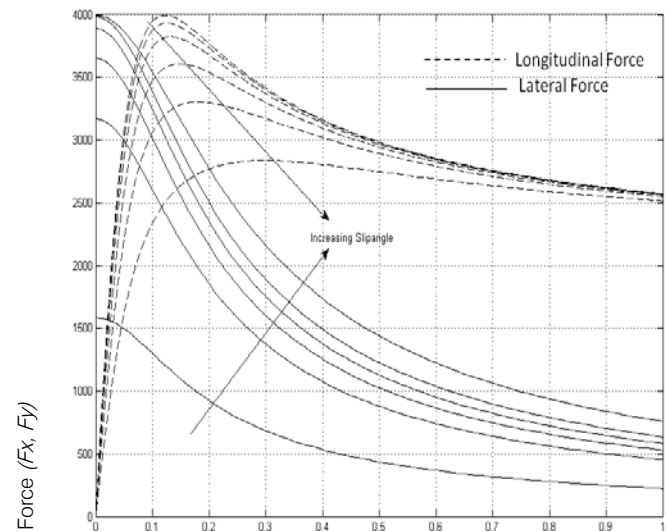
$$G_{yk} = \frac{\cos[C_{yk} \tan^{-1}(B_{yk} k_s)]}{\cos[C_{yk} \tan^{-1}(B_{yk} S_{Hyk})]} \quad (15)$$

$$G_{x\alpha} = \frac{\cos[C_{x\alpha} \tan^{-1}(B_{x\alpha} (\alpha + S_{Hx\alpha}))]}{\cos[C_{x\alpha} \tan^{-1}(B_{x\alpha} S_{Hx\alpha})]} \quad (16)$$

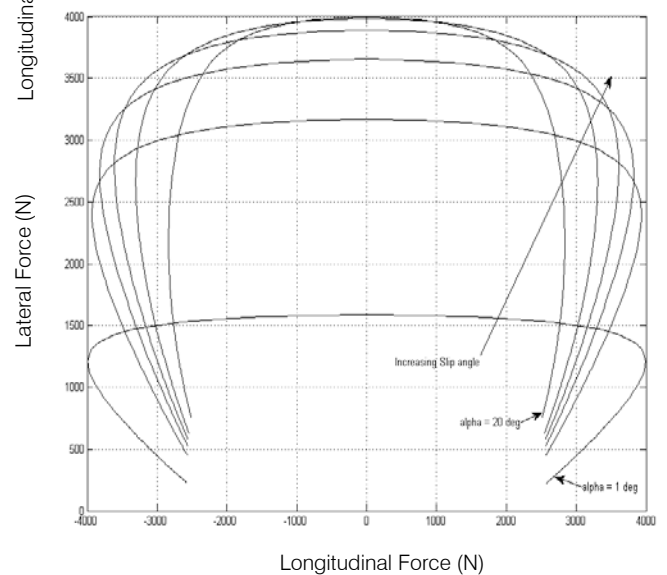
A detailed description of these weighing functions can be found in Bakker (1987). Figure 2 show the longitudinal and lateral tyre forces in combined braking and cornering conditions used in this paper obtained using the above mentioned equations. The

effect of tire force lag (Rajamani and Hedrick, 1995) is also taken into account according to the following equation.

$$\left. \begin{aligned} \frac{L_{Fx}}{V_x} \left(\frac{d}{dt}(F_x) \right) + F_x &= F_{xss} \\ \frac{L_{Fy}}{V_x} \left(\frac{d}{dt}(F_y) \right) + F_y &= F_{yss} \end{aligned} \right\} \quad (16)$$



Combined longitudinal and lateral tyre force vs. slip ratio



Tyre forces during combined braking and cornering

Fig. 2: Tyre forces in combined slip -friction ellipse

The equation of motion for each wheel in the wheel dynamics model is defined as:

$$I_w \dot{\omega}_{ij} = T_{ij} - F_{xij} R_w \quad (17)$$

The steering system modeled in this work has a hydraulic power steering mechanism. The input for the steering system is the angle of the steering wheel and steering column, while the output is the position of the rack, which determines the angle of the front wheels. There is a mechanical connection between the rack and the steering column with a pinion gear, which converts the rotational motion of the steering column to translational motion of the rack to turn the wheels.

The power assistance is provided by a hydraulic piston attached to the rack. A torsion valve determines which side of the piston receives pressurized hydraulic fluid. This torsion valve is attached to the steering column. The difference between the angular position of the steering wheel and the angular position of the pinion determines the fractional opening of the torsion valve. If the angular difference is positive, the pressure is applied to one side of the piston, and if the angular difference is negative, the pressure is applied to the other side of the piston. The power assistance continues until the difference between the steering wheel position and pinion position is approximately zero. The steering power assist curve is shown in figure 3.

The hydraulic brake system considered in this study is built upon a standard braking model. The standard passive brake system considered for this study consists of a mechanical brake pedal, a servo brake booster, a master cylinder, a hydraulic brake caliper and the friction pad. The brake mechanics considered here are explained as follows. The mechanical brake input is amplified by the servo booster. This is further amplified and converted to a hydraulic pressure called line pressure/supply pressure, which is fed through the brake lines. The line pressure is further amplified and converted to mechanical actuation at the brake calipers. This force moves the friction pads against the rotating wheel disc. The effect of pipe friction is taken into account in line with the real world brake dynamics.

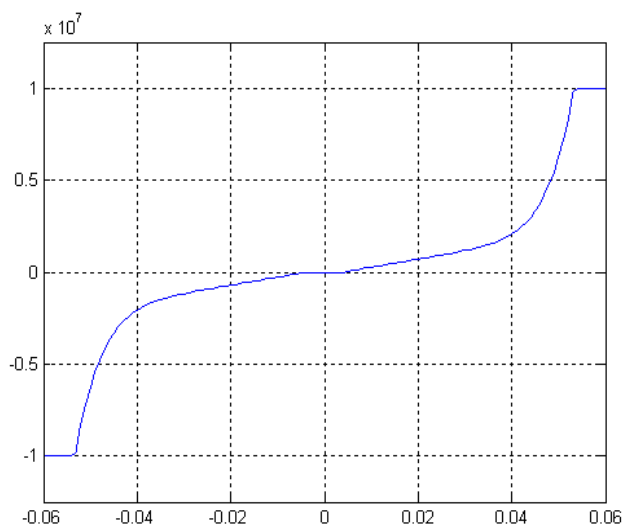


Fig. 3 : Hydraulic steering power assist curve

The resulting brake system model assumes non-laminar flow through restriction as described in the following equation by Fletcher et al. (2004).

$$Q = C_d A \sqrt{\frac{2}{\rho} (P_1 - P_2)} \quad (18)$$

III. DEVELOPMENT OF AUTOMOTIVE TOOLBOX IN MATLAB/SIMULINK

Simulation of dynamic systems such as vehicles is a complex and time consuming task. Most of the time the modeling tasks need to be repeated in order to perform system analysis such as "if-what" scenarios. Developing a toolbox will modularise the whole modeling process and reduce the model development and analysis time. Rodic (2003) developed a specialized piece of commercial software for modeling, control design and simulation of road vehicles. Poussot-Vassal et al. (2007) developed a unique toolbox during the course of his doctoral research to analyze active suspension and active brake systems. The automotive toolbox developed in this paper provides Simulink models and Matlab tools for vehicle dynamic simulation, analysis and development of vehicle dynamic control systems. It has modular Simulink models for quarter car, extended quarter car, half car for roll and pitch, vertical vehicle model, full vehicle model, linear, nonlinear tire models and other vehicle subsystem models.

This toolbox provides a flexible environment for vehicle dynamic research. It contains libraries with Simulink graphical blocks and Matlab functions, which can be connected to build vehicle models. Using this toolbox, it is also possible to subdivide the whole vehicle model into a number of smaller vehicle subsystems, which can be arranged in a neat way and validated separately. The use of block-diagrams greatly facilitates computer representation of vehicle dynamic systems.

IV. VEHICLE MODEL VALIDATION

This section describes the validation of the full vehicle model developed in the previous sections. The handling dynamics is evaluated and the simulation results were compared against industry standard software. Any software vehicle models developed needs to be validated either against experimental results or against other proven simulation software results.

The vehicle model developed in this paper is validated against the well-known commercial software CarSim. CarSim is vehicle dynamics simulation software developed by Mechanical Simulation Corporation in Ann Arbor, USA. It is parametric modeling software widely used both in academia and industry to simulate, predict and analyze vehicle dynamic behavior.

The validation methodology consists of three phases:

- Describing the validation test condition and procedures
- Simulation of full vehicle model
- Comparison of simulation prediction with the CarSim vehicle model simulation data

In order to be used in this research the model developed must be capable of evaluating vehicle dynamics both in normal and limit driving situations. Two standard test maneuvers are used to evaluate the vehicle model. First, a step steer input at a constant speed was provided so that it generates a lateral acceleration (latac) of 0.3g, 0.6g and 0.8g respectively. This evaluates the model across all the lateral acceleration range from low to the limit handling. Figures 4-7 present the results for the step steer input, showing the comparison of yaw rate at 0.3g latac, of vehicle side slip angle at 0.3g latac, of yaw rate at 0.6g and 0.8 latac, respectively. It is clearly shown that there are a great match between both CarSim and full vehicle model.

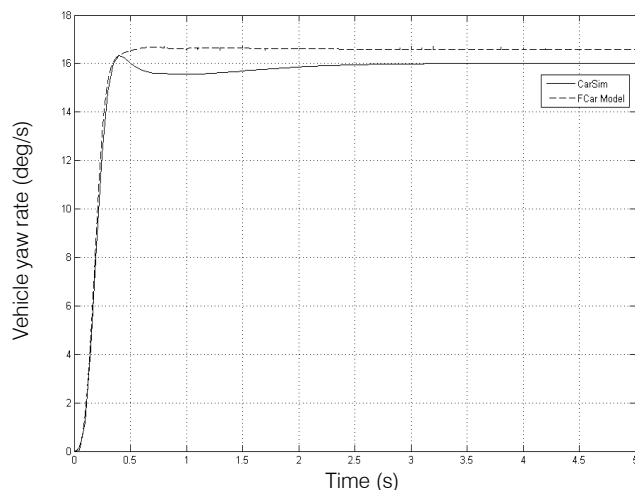


Fig. 4 : Comparison of yaw rate at 0.3g latac

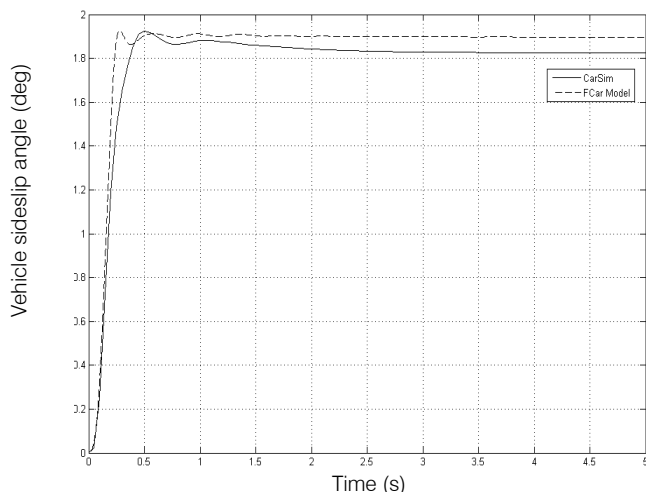


Fig. 5 : Comparison of vehicle sideslip angle at 0.3g latac

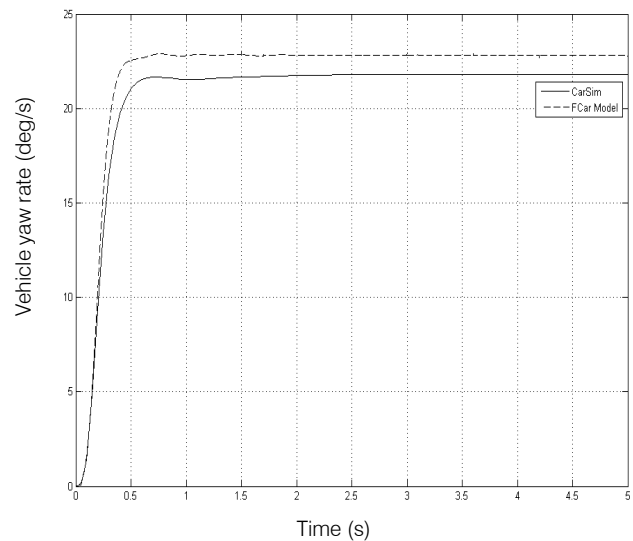


Fig. 6 : Comparison of yaw rate at 0.6g latac

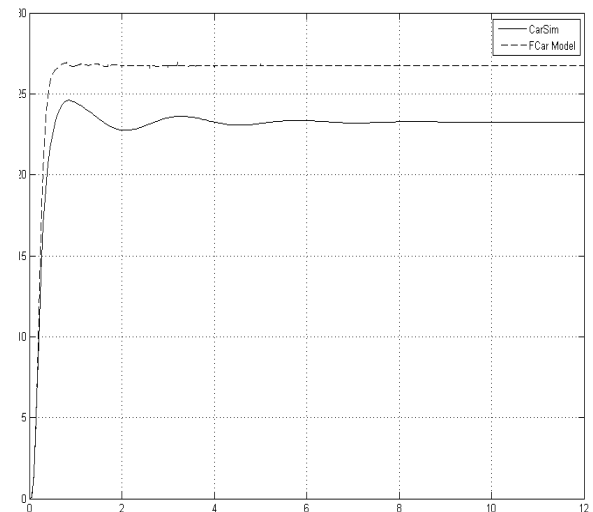


Fig. 7 : Comparison of yaw rate at 0.8g latac

Then a double lane change manoeuvre was also performed to validate the vehicle model. The test was performed at a speed of 80km/h on a flat dry surface whose coefficient of friction was 1 ($\mu = 1$). First the test was carried out using CarSim software. The vehicle parameters for a D Class Sedan were used. The results were imported to Matlab/Simulink workspace. Then the Full vehicle model was characterised with the CarSim vehicle parameters. The same steering data used to simulate the CarSim model was used as steering input to the full vehicle model. The simulation results of the full vehicle model were plotted along with the CarSim results for comparison. Figures 8-10 show comparisons of the vehicle yaw rate, of the vehicle sideslip angle, and of the vehicle path between CarSim and full vehicle model during an 80km/h double lane change manoeuvre. It is also clearly noticed that there are a great match between both results.

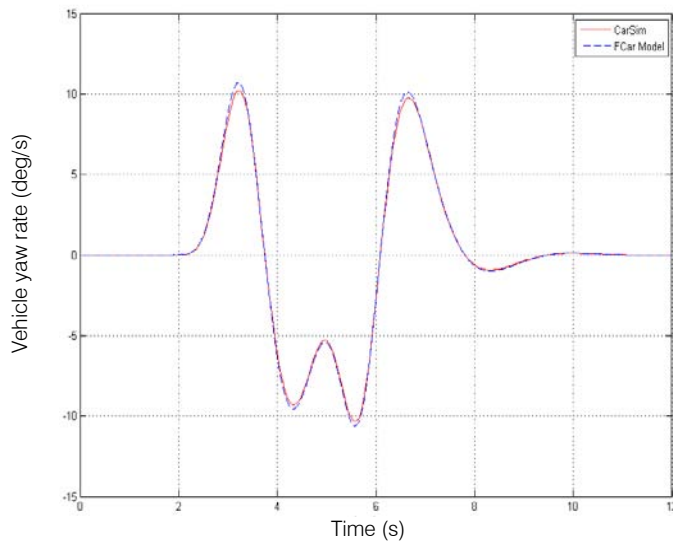


Fig. 8 : Comparison of vehicle yaw rate between CarSim and Full vehicle model (80km/h double lane change maneuver)

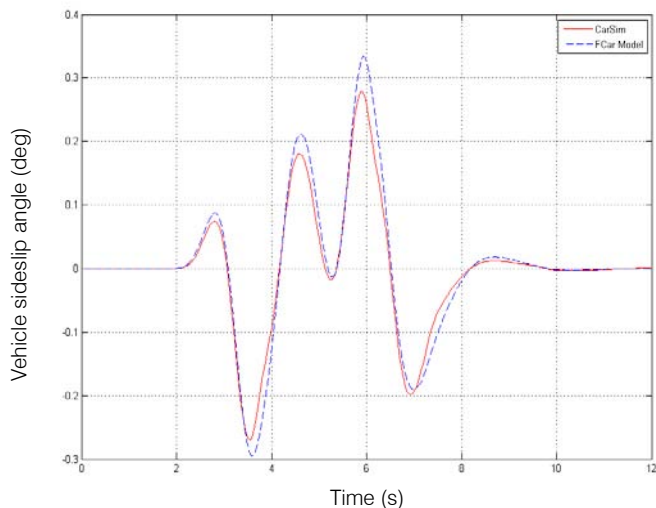


Fig. 9 : Comparison of vehicle sideslip angle between CarSim and full vehicle model during (80km/h double lane change maneuver)

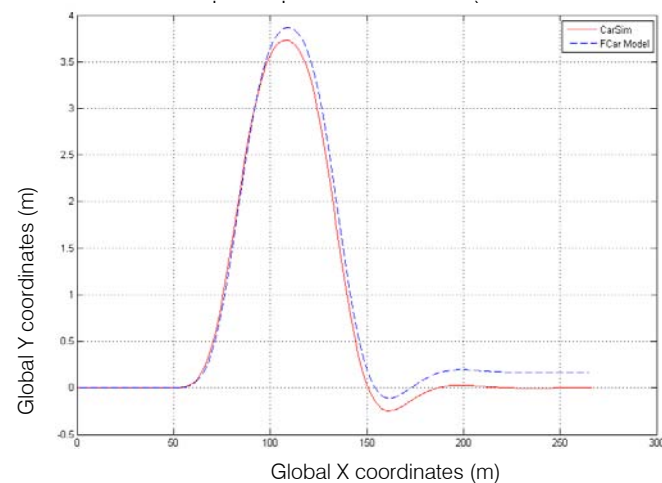


Fig. 10 : Comparison of vehicle path between CarSim and full vehicle model during (80km/h double lane change maneuver)

From these figures, it can be concluded that the responses of full vehicle model developed follows closely the responses of the CarSim vehicle model across various lateral acceleration range. The little deviations observed in the medium and high lateral acceleration range are largely due to the differences in the suspension kinematics between the models, the nonlinearity in the suspension elements in CarSim and limitations in transferring all the CarSim vehicle parameters in to the Full vehicle model. Moreover, as the CarSim model is validated against real-time experiments, a conclusion can be derived that the full vehicle model is also validated indirectly against experimental results. So it can be concluded that the full vehicle model developed is in par with the widely used commercial software vehicle model and is suitable to use in the vehicle dynamics studies such as integrated chassis control systems.

V. STANDALONE CONTROL SYSTEMS

a) Electronic Stability Control

Electronic stability control is used to stabilize a vehicle by generating an external yaw moment. The three strategies explained in literature to perform this are differential braking, active steering and differential drive torque distribution. In this section of the research, the differential braking, a brake (ABS) based ESC strategy is used. First a fuzzy logic ABS controller was developed and simulated for its performance. Then the ABS controller was extended to develop an ESC controller by additional sensor inputs, like steering angle, yaw rate and sideslip angle and supplemented with an ESC controller algorithm that is capable of enhancing the vehicle stability.

The control architecture as shown in figure 11, is designed to be a hierarchical, two layer control (Rajamani, 2006). The upper controller has the desired objective of ensuring yaw stability control and assumes that it can command any desired value of yaw torque. The lower controller ensures that the desired value of yaw torque commanded by the upper controller is indeed obtained from the differential braking system based on ABS.

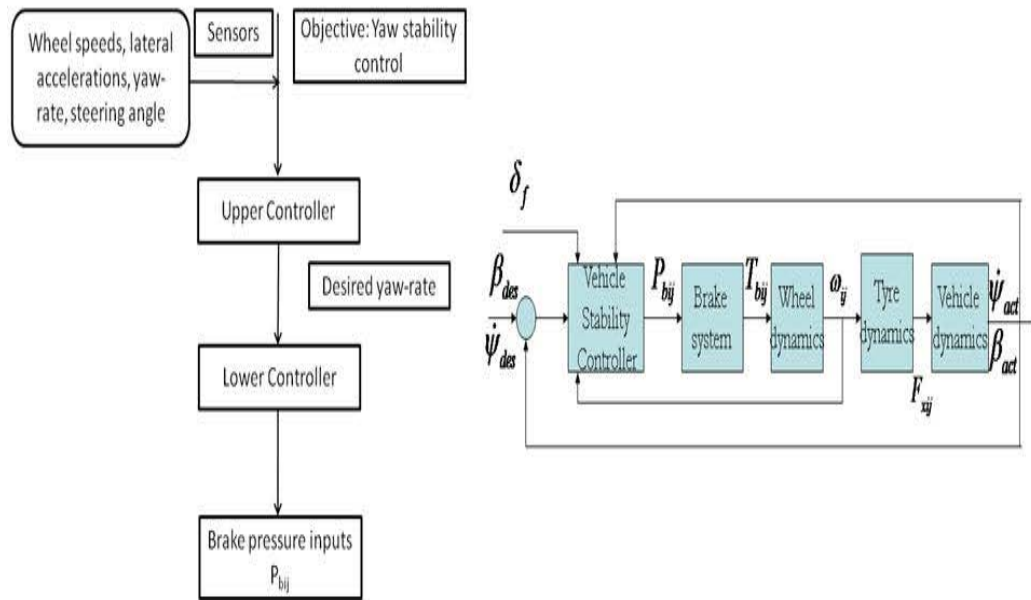


Fig. 11 : Schematic of electronic stability control (ESC)

The lower controller utilizes the wheel rotational dynamics and controls the braking pressure at each of the four wheels to provide the desired yaw torque for the vehicle. Figure 12 describes the relationship between the yaw rate error and its derivative to the controller output. Results of the earlier research show that the brake based ESC are more effective in a wide lateral acceleration range is validated through simulations.

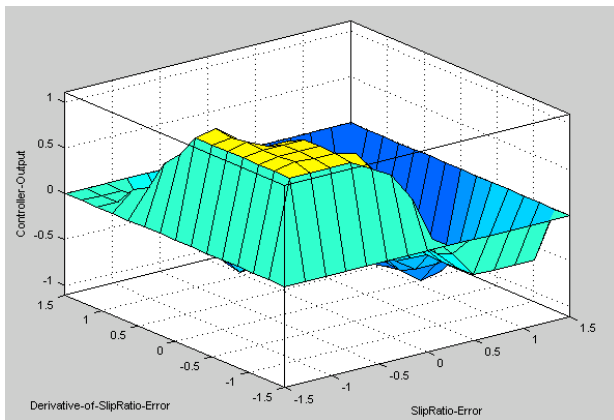


Fig. 12 : Control surface of ESC fuzzy controller

b) Active Suspension

Active suspension in this paper is another active vehicle control system that minimizes the longitudinal and lateral load transfer between the wheels. The AS model used in this research has hydraulic actuators at each wheels as shown in figure 13 that either add or subtract an extra force on each wheels and designed to optimize the normal forces on wheels based on signals from the active suspension controller as a function of various vehicle dynamic states.

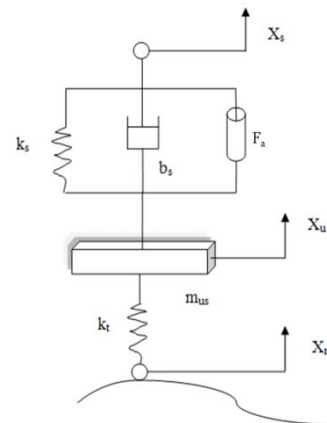


Fig. 13 : Quarter car suspension model

It ensures the tracking of the desired suspension force using PID and Fuzzy logic control strategies. Considerable literature can be found on the dynamics and control of hydraulic actuators for active automotive suspensions.

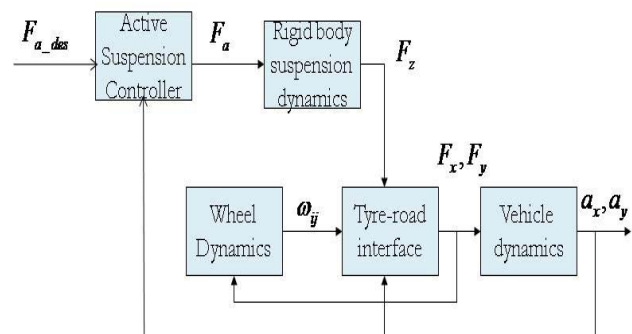


Fig. 14 : Schematic of active suspension system (AS)

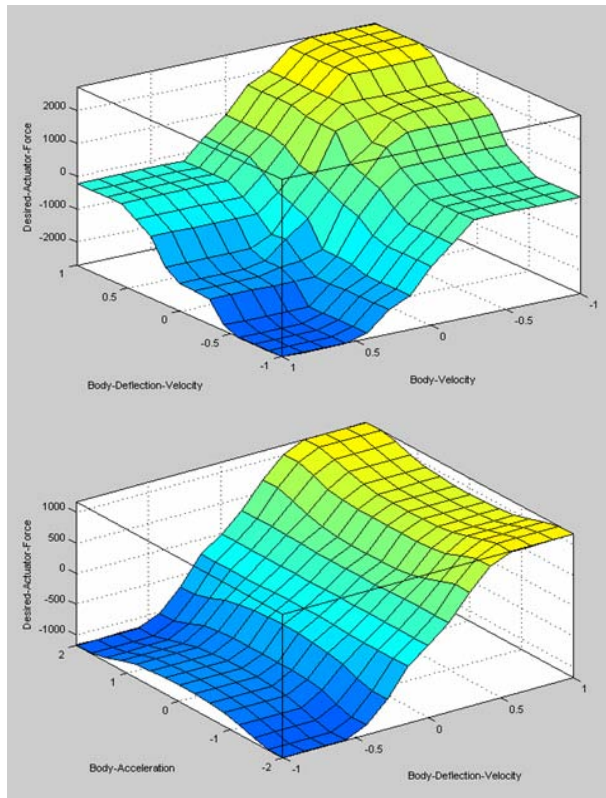


Fig. 15 : Control surface of AS fuzzy controller

The hydraulic actuator dynamics used in this paper includes the dynamics of a spool valve controlled hydraulic actuator model explained by Rajamani and Hedrick (1995). The schematic of the active suspension control strategy followed is described in figure 14 and few of the control surfaces used to develop the fuzzy logic suspension controller are shown in figure 15.

c) Variable Drive Torque Distribution

Another important way to stabilize a vehicle is active drive torque control. One of the recent and widely applied active driveline control techniques is variable drive torque distribution (Pinnel et al., 2004). The objective of this control strategy is to increase vehicle stability and handling capability by suitably distributing the drive torque between wheels. Different drive torque on left and right wheels yield a yaw moment about the vehicle's vertical axis and can be used to stabilize the yaw motion. A two layer control architecture similar to the one shown in figure 11 is used. A PI controller strategy is followed in this case and developed in Matlab/Simulink. The PI controller takes the yaw rate and side-slip angle errors as inputs and returns a control value between 0 and 1 giving the ratio of the drive torque transmitted to the left and right wheels. The control architecture of VTD used in this paper is as shown in figure 16.

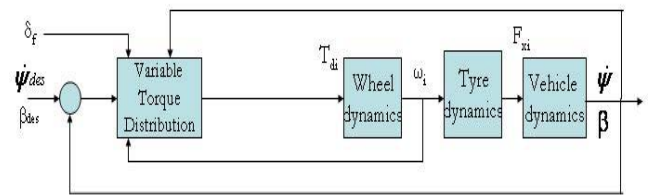


Fig. 16 : Schematic of variable torque distribution control (VTD)

d) Active Front Steering

The active front steering improves the vehicle dynamics in the lateral direction by extending the linear handling region experienced by the driver in a passive vehicle. In a typical vehicle active steering system, the steering angle at the tyre is set in part by the driver through the vehicle classical steering mechanism while an additional steering angle can be set by the AFS controller using hydraulic or DC motor actuators combined with a differential mechanical device. The schematic of the AFS control strategy followed in this paper is described in figure 17.

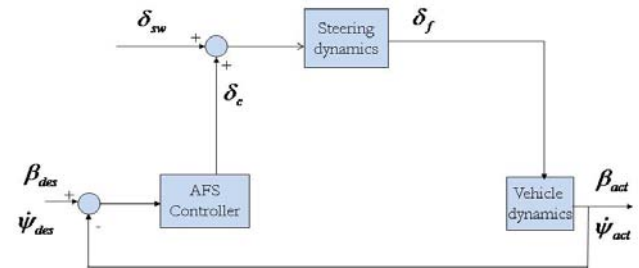


Fig. 17 : Schematic of active front steering (AFS)

Two commonly used control strategies, PID and Fuzzy logic, were used in the development of standalone steering controller in this paper. The vehicle yaw rate and sideslip angle errors (which are the functions of their nominal and actual values respectively) and their time derivatives are fed to the AFS controller to determine the controlled steer addition. Figure 18 shows one of the control surfaces used to develop the fuzzy logic steering controller. Results of earlier research literature in this field are validated here and confirm that performance of active front steering is limited within the linear vehicle handling region, i.e., low to medium lateral acceleration range.

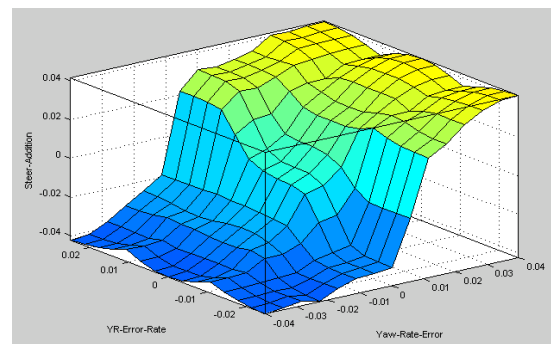


Fig. 18 : Control surface of AFS fuzzy controller

VI. INTEGRATED CONTROL SYSTEM

The goal of the integrated controller is to cohabit the active steering, active suspension, electronic stability control and variable torque distribution control strategies in order to attain a level of performance that would not otherwise be achievable by them in a stand-alone manner. All the integration strategies explained in this paper are obtained through simulation investigation of the vehicle handling dynamics by performing the ISO3888 double lane change maneuver.

The integrated controller was based primarily on yaw rate and vehicle side-slip angle errors, derived from the actual and desired values. A supervisory fuzzy logic controller (ICC) is used to coordinate among individual control systems. The transition of control authority between individual control systems takes place using a smooth blending function (Junjie et al., 2006) so that better stability and handling performance is maintained throughout the vehicle performance envelop.

To begin the investigation in developing the integrated controller, first, various possible combinations of control strategies between these four systems were analyzed. From the standalone controller systems development in the previous section, we observe that the control objective of ESC is to improve the vehicle lateral stability by minimizing the yaw rate and side-slip angle errors. And the control goal of the active suspension is to optimize the normal wheel forces as a function of various vehicle dynamic states irrespective of ESC's control objective. So, one possible concept of integrating ESC and AS is to optimize the normal wheel forces taking into account the vehicle's yaw rate and side-slip angle errors as well. The schematic of this strategy is described in figure 19 and leads to the co-existence of ESC and AS without affecting the present vehicle handling behavior.

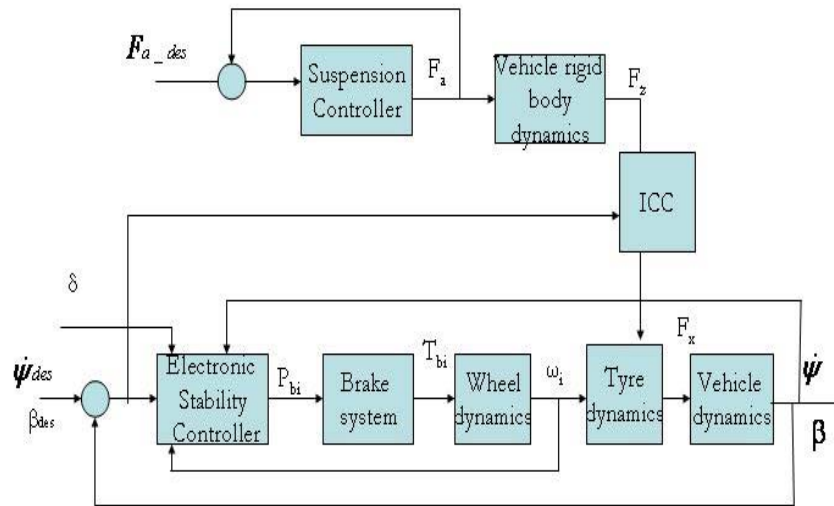


Fig. 19 : Schematic of ICC strategy between ESC & AS

The control objectives of both ESC and AFS are to improve vehicle lateral stability. From the analysis of these two standalone controllers and from the research literature, it is observed that the AFS is more effective in fulfilling its control goal in the linear vehicle handling region whereas the ESC is effective in achieving its control objective in a wide range of lateral accelerations.

Further results of the analysis and literatures show that the ESC is affecting the longitudinal dynamics of the vehicle while improving the lateral stability. The effect of ESC in longitudinal dynamics is perceived as an intrusion in drivers' point of view. Based on the above analysis, an integrated control strategy between ESC and AFS was developed and analyzed through simulation.

This integrated control strategy gives the control authority to AFS during low to medium lateral acceleration range thus providing more comfortable

lateral stability compare to ESC. Once the AFS's ability to stabilize the vehicle diminishes, the integrated controller switches the control authority to ESC, in medium to high lateral acceleration range to improve the vehicle stability in critical driving situations. The schematic of above mentioned strategy is shown in figure 20.

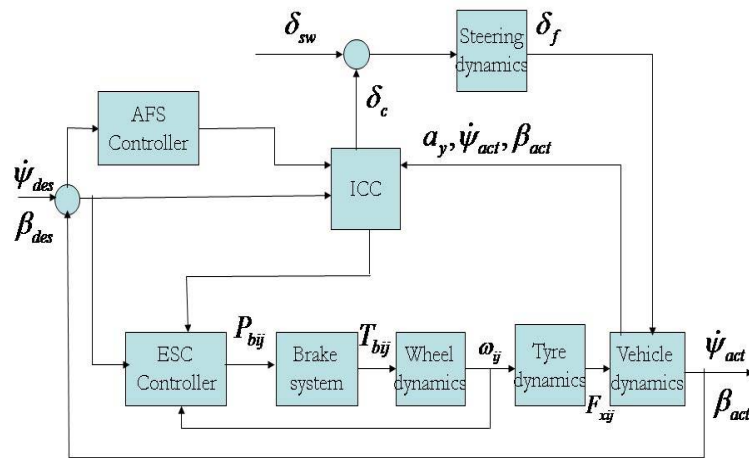


Fig. 20 : Schematic of ICC strategy between ESC & AFS

Similarly an integrated control strategy to improve the vehicle lateral dynamics stability by integrating AFS and AS is also developed and investigated. Here the active suspension optimizes the normal wheel forces taking into account the vehicle yaw rate and side-slip angle errors. Figure 21 describes this control strategy.

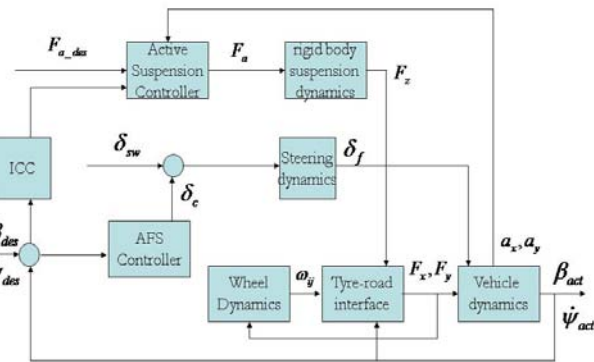


Fig. 21 : Schematic of ICC strategy between AFS & AS

Finally the control objectives of both ESC and VTD are to improve vehicle lateral stability by reducing the yaw-rate and side-slip angle errors. From the development of these two standalone controllers, it is observed that both the ESC and VTD are effective in stabilizing the vehicle in both the linear and non-linear handling regions, but the ESC is intruding and affecting the longitudinal dynamics of the vehicle whereas the intrusion of the VTD in longitudinal dynamics is more acceptable. The schematic of this control strategy is described in figure 22.

Based on these simulated investigations a new integrated control strategy (ICC) is proposed amongst AFS, ESC, VTD and AS to improve the present vehicle handling performance as follows. Here, the stability of the vehicle is determined using a stability criterion.

This stability criterion is based on a phase-plane characteristics (Vientinghoff, 2008) between the vehicle side-slip angle and its rate of change. The vehicle's stability is classified based on the stability criterion. Then a supervisory fuzzy logic controller is designed that

decides the weighting of control authority among the four active control systems to generate the necessary control steer angle, brake pressure, drive torque and suspension force to be added.

The proposed strategy is explained in figure 23 in a schematic form. Based on the new integrated control strategy the AFS, ESC, VTD and AS controllers were evaluated in both standalone and integrated manner and found that the integration of these controllers improves the vehicle lateral stability by reducing yaw rate and side-slip angle during an ISO 3888 DLC maneuver on both high and low frictional surfaces as shown in figures 24-27, respectively.

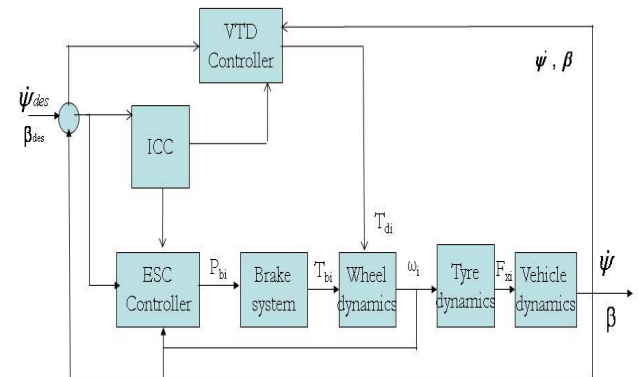


Fig. 22 : Schematic of ICC strategy between ESC & VTD

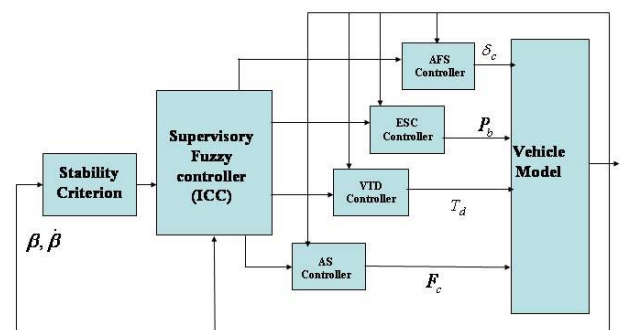


Fig. 23 : Schematic of proposed ICC strategy amongst ESC, AFS, VTD & AS

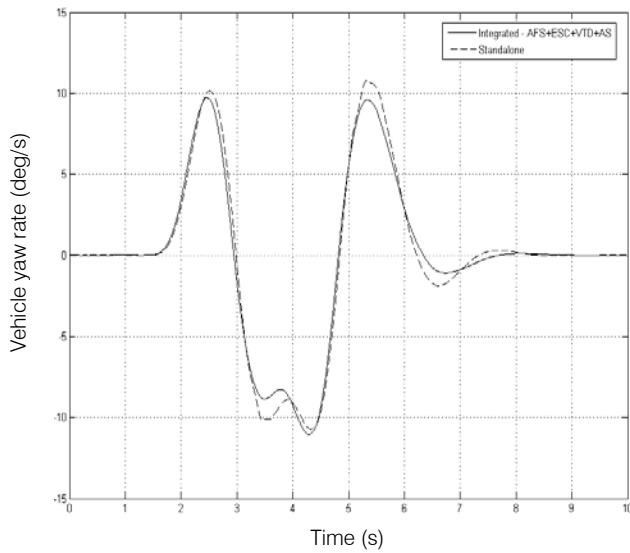


Fig. 24 : Yaw rate – integrated and standalone controllers – high friction surface

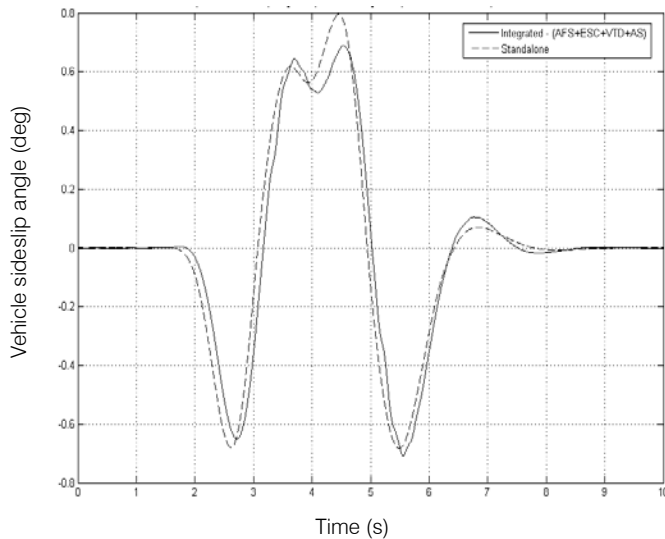


Fig. 25 : Sideslip angle – integrated and standalone controllers – high friction surface

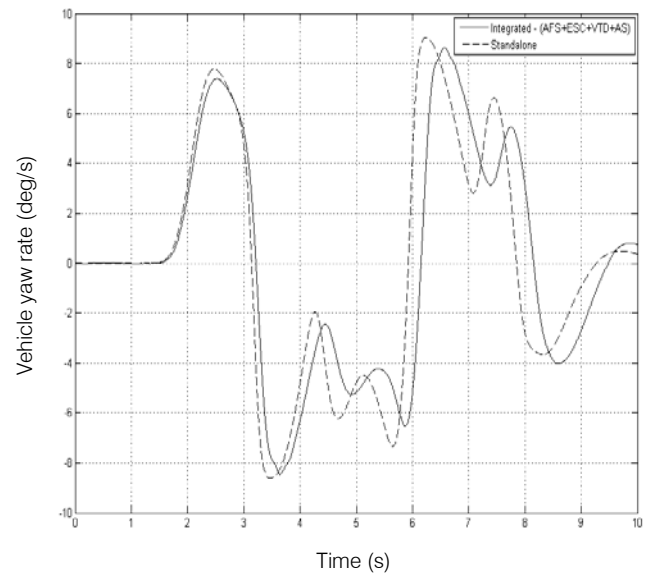


Fig. 26 : Yaw rate– integrated and standalone controllers low friction surface

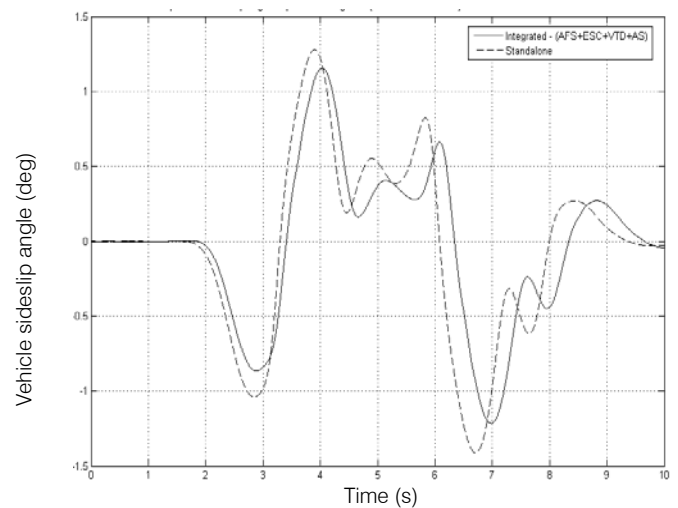


Fig. 27 : Sideslip angle – integrated and standalone controllers –low friction surface

VII. CONCLUSIONS

A detailed non-linear vehicle dynamics model is developed and four standalone vehicle control systems AFS, ESC, VTD and AS are modeled and simulated in MATLAB/Simulink. Various possible combinations of integrated strategies between these four systems were investigated through simulations. Based on these investigations a new integrated control strategy is proposed to make these four systems cohabit to improve the present vehicle handling performance. Simulations with the integrated controller demonstrated a significant improvement in the performance objectives. The vehicle motion during an emergency manoeuvre is much improved, showing better handling and characteristics during sudden, high speed maneuvers

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NOMENCLATURE

a	Longitudinal distance of CG from front axle
b	Longitudinal distance of CG from rear axle
B, C, D, E	Pacejka tire parameters
F_{sij}	Suspension force in N
F_{xij}	Longitudinal tire force
F_{xss}	Longitudinal steady state tire force
F_{yij}	Lateral tire force
F_{yss}	Lateral steady state tire force
g	Gravitational constant
h	Height of center of gravity from ground
i	Stands for front/rear $\{f, r\}$
I_w	Wheel rotational inertia
j	Stands for left/right $\{l, r\}$
J_r	Roll inertia of the vehicle in km^2/m^2
J_p	Pitch inertia of the vehicle in km^2/m^2
J_y	Yaw inertia of the vehicle in km^2/m^2
L_{Fx}	Longitudinal tire force lag
L_{Fy}	Lateral tire force lag
m_s	Sprung-mass of vehicle
m_v	Mass of total vehicle
R_w	Dynamic tire radius
t	Track width of vehicle
V_x	Sprung mass longitudinal velocity at CG
$\dot{V}_x$	Derivative of longitudinal velocity
V_y	Sprung mass lateral velocity at CG
$\dot{V}_y$	Derivative of lateral velocity
V_z	Sprung mass vertical velocity at CG
$\dot{V}_z$	Derivative of vertical velocity
α	Tire lateral slip angle
δ	Steering angle at the front wheels
θ	Vehicle pitch angle at CG

$\dot{\theta}$	Vehicle pitch rate at CG
$\ddot{\theta}$	Derivative of vehicle pitch velocity
λ	Tire longitudinal slip ratio
μ	Coefficient of friction
ϕ	Vehicle roll angle at CG
$\dot{\phi}$	Vehicle roll rate at CG
$\ddot{\phi}$	Derivative of vehicle roll velocity
ψ	Vehicle yaw angle at CG
$\dot{\psi}$	Vehicle yaw rate at CG
$\ddot{\psi}$	Derivative of vehicle yaw velocity
ω	Angular wheel velocity





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By S. S. Verma

S.L.I.E.T., Longowal

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Keywords : *compressed air vehicle, technological trends, energy input, energy released, emission control, storage & fueling, temperature.*

GJRE-B Classification : *FOR Code: 090299*



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I. INTRODUCTION

Compared to batteries, compressed air is favorable because of a high energy density, low toxicity, fast filling at low cost and long service life. These issues make it technically challenging to design air engines for all kind of compressed air driven vehicles. To meet the growing demand of public transportation, sustainable with environmental consciousness, people are in the search for the ultimate clean car with zero-emissions. Many concept vehicles were proposed that run on everything from solar power to algae, but most of them are expensive and require hard-to-find fuels. Compressed air vehicle project in the form of light utility vehicle (LUV) (i.e., air car in particular) has been a topic of great interest for the last decade and many theoretical and experimental investigations [1-19] have appeared on the subject in the literature. Many largest car manufacturers [5-8, 11-13, 19] all over the world have taken up the lead in this direction based on the initial technological concept of the pioneer-the French company Motor Development International (MDI) in the field. In 2008, India's largest car manufacturer also announced that it would begin production of world's first commercial vehicle to run on nothing but compressed air.

Author : Department of Physics, S.L.I.E.T., Longowal; Distt.-Sangrur (Punjab)-148 106, India.



Fig. 1 : Model of air car by Tata Motors

The car was said to achieve speeds of up to 68 mph with a range of 125 miles between fill-ups, all for less than \$13,000. Unfortunately, the dates for the air car's much-publicized release in both Indian and American markets have come and gone with no word about when the vehicle might actually hit the streets. Tata says that the first "proof of the technical concept" phase of the program is now complete with "the compressed air engine concept having been demonstrated in two Tata Motors vehicles." Phase two, which involves, "completing detailed development of the compressed air engine into specific vehicle and stationary applications," is now underway and, "the two companies are working together to complete detailed development of the technology and required technical processes to industrialize a market ready product application over the coming years."

II. TECHNOLOGICAL TRENDS

- In Europe inventors have made a simple air engine, thus opening a new field for compressed air car technology. These engines allow cars to run on compressed air instead of fuel. The air, super compressed and powerful, pumps the pistons in the car instead of small gas explosions. Pumping air instead of exploding gasoline means these cars have zero emission motors—no pollution, no oil. In addition, current average family cost of fuel is 60 dollars a week and half that for a hybrid car. The new air engines will give a whole week of driving for a few dollars. The company, MDI plans to sell this clean fuel vehicle and a compressed air hybrid in Europe for less than 15,000 dollars in near future.

- To refit scooters with an air-compression motor that is about three-quarters of a foot in diameter. In the engine's schematics, a tank of compressed air fires into the chambers of a turbine whose axis is set off-center from its housing. The vanes of the turbine extend as they rotate, allowing the chambers to accommodate the volume of air as it expands and contributes to the drive. Unlike the Air Car, the retrofitted scooters would run off the pressure it takes to fill a tire at the gas station. Although they are hoping to eventually solve engineering problems related to torque and range for the scooters (which currently only hold enough air to travel 18 miles), it might be a while until they can really solve the emissions problem.
- An inventor in California (USA) has made a car running on compressed air, stored in scuba diving tanks. He modified an engine used on the Honda RC51 998 cc Superbike. He blocked off one of the cylinders of the engine and used the spark plug hole on the other cylinder to feed the compressed air. The compressed air drives the piston down as the power stroke. At the end of the power stroke, the compressed air is released through the exhaust valves and the exhaust is only air. The pistons were connected to the wheels through the Honda bike's six-speed transmission. This modified engine was mounted on a tubular frame and a body that looked like a curious crossbreed of a motorbike with a racing car. A bank of three scuba tanks was used to store compressed air at 3500 psi and throttled it to 250 psi at the engine inlet with a self-designed throttle valve, linked to the accelerator pedal. The three tanks were sufficient for the test run over the 2 mile, where an average speed of 46.723 mph was achieved with a top speed of 54.058 mph. Further, a speed to the level of 300 mph is expected with compressed air. However, several issues like large size and heavy model due to the use of a number of tanks still remain before this concept can translate into a usable idea.
- Another inventor in Uzbekistan has modified a car to run it with compressed air, stored in a tank and claims that as the car runs, it will take in air and store it in the tank at pressure, resulting in a perpetual motion machine. Perpetual motion machines are, of course, not possible under the known laws of physics but the idea is not being dismissed as crank. The concept of compressed air being regenerated while the car is in motion is intriguing. It is akin to batteries of electric vehicles using regenerative charging and even a part regeneration of compressed air would be an advance from the present understanding of the technology.

III. DEVELOPERS AND MANUFACTURERS

Various companies are investing in the research, development and deployment of compressed air cars. Overoptimistic reports of impending production date back to at least May 1999. Most of the cars under development also rely on using similar technology to low-energy vehicles in order to increase the range and performance of their cars. Here is a brief list of few leading manufacturers who are putting their concerted efforts towards making compressed air vehicle a reality for public use.

- APUQ (Association de Promotion des Usages de la Quasiturbine) has made the APUQ Air Car, a car powered by a quasiturbine.
- MDI (Motor Development International), France has proposed a range of vehicles made up of AirPod, OneFlowAir, CityFlowAir, MiniFlowAir and MultiFlowAir. One of the main innovations of this company is its implementation of its "active chamber", which is a compartment which heats the air (through the use of a fuel) in order to double the energy output.
- Tata Motors, India, as of January 2009 had planned to launch a car with an MDI compressed air engine in 2011. In December 2009 Tata's vice president of engineering systems confirmed that the limited range and low engine temperatures were causing problems. Tata Motors announced in May 2012 that they have assessed the design passing phase 1, the "proof of the technical concept" towards full production for the Indian market. Tata has moved onto phase 2, "completing detailed development of the compressed air engine into specific vehicle and stationary applications".
- Air Car Factories South Africa is proposing to develop and build a compressed air engine.
- The Energine Corporation, a South Korean company claimed to deliver fully assembled cars running on a hybrid compressed air and electric engine. These cars are more precisely named pneumatic-hybrid electric vehicles. Engineers from this company made, starting from a Daewoo Matiz, a prototype of a hybrid electric/compressed-air engine. The compressed-air engine is used to activate an alternator, which extends the autonomous operating capacity of the car.
- A similar concept using a pneumatic accumulator in a largely hydraulic system has been developed by U.S. government research laboratories and industry. It uses compressed air only for recovery of braking energy, and in 2007 was introduced for certain heavy vehicle applications such as refuse trucks.
- K'Airmobiles vehicles were intended to be commercialized from a project developed in France in 2006-2007 by a small group of researchers.

However, the project has not been able to gather the necessary funds. In the meantime, the team has recognized the physical impossibility to use on-board stored compressed air due to its poor energy capacity and the thermal losses resulting from the expansion of the gas.

- Engine-air is an Australian company which manufactures small industrial vehicles using an air engine of its own design.
- In 2010, Honda presented the Honda Air concept car at the LA Auto Show.

IV. PROBLEMS WITH COMPRESSED AIR

A compressed air vehicle is a type of alternative fuel vehicle which uses a motor powered by compressed air. Compressed air cars are not yet available in most places but the technology behind them is being perfected so that they can be introduced into the market. These cars can be powered solely by air or by a combination of air and fuel, such as diesel, ethanol, or gasoline which is how a hybrid electric vehicle runs. Compressed air car engines are fueled by a tank of compressed air, instead of an engine that runs with pistons and an ignited fuel air mixture. Basically, compressed air cars are powered by the expansion of compressed air. Vehicles that run on compressed air sound like a fantastic idea on paper, but bringing this technology to the masses have proven, well, a difficult road to travel because of some inherited technical problems with compressed air [2-4, 14, 18]. The present article gives a brief report to highlight such problems so that some methods can be designed to counter to improve the efficiency of compressed air vehicle.

a) Engine Technology

The air-powered car runs on a pneumatic motor that is powered by compressed air stored onboard the vehicle. Once compressed air is transferred into the onboard storage tank, it is slowly released to power the car's pistons. The motor then converts the air power into mechanical power. That power is then transferred to the wheels and becomes the source of power for the car. The engine that is installed in a compressed air car uses compressed air which is stored in the car's tank at a pressure as high as 4500 psi. The technology used by air car engines is totally different from the technology that is used in conventional fuel cars. They use the pressure generated by the expansion of compressed air to run their pistons.

b) Air Storage & Refuelling

The cars are designed to be filled up at a high-pressure pump and thus the tanks must be designed to safety standards appropriate for a pressure vessel. The storage tank may be made of metal or composite

materials. The fiber materials are considerably lighter than metals but generally more expensive. Metal tanks can withstand a large number of pressure cycles, but must be checked for corrosion periodically. It may be possible to store compressed air at lower pressure using an absorption material within the tank. Absorption materials such as activated carbon, or a metal organic framework is used to store compressed natural gas at 500 psi instead of 4500 psi, which amounts to a large energy saving. One company stores air in tanks at 4,500 pounds per square inch (about 30 MPa) and hold nearly 3,200 cubic feet (around 90 cubic metres) of air. The tanks may be refilled at a service station equipped with heat exchangers, or in a few hours at home or in parking lots, plugging the car into the electrical grid via an on-board compressor. The Tata/MDI air car version had 4,350 psi in its tanks, which would require stations to install new high-tech air pumps, a difficult investment for station owners. As thought by engineers and designers, the storage tank would be made up of carbon fiber to reduce the car's weight and prevent an explosion, in case of a direct collision. Carbon-fiber tanks are capable of containing air pressure up to 4500 psi, something the steel tanks are not capable of. For fueling the car tank with air, the compressor needs to be plugged into the car, which would use the air that is around to fill the compressed air tank. This could be a slow process of fueling; at least until air cars are commonly used by people, after which high-end compressors would be available at gas stations that would fuel the car in no time at all.

c) Input Energy

Compressed air to very large pressures is extremely expensive to produce and it is very inefficient to use. Of the energy required to produce compressed air, less than 20% of input energy is left for use. That means 80% of what is paid for is used up before compressed air is put in the distribution system. In order to use air storage in vehicles or aircraft for practical land or air transportation, the energy storage system must be compact and lightweight. Energy density is the engineering term that defines these desired qualities. Compressed air has a low energy density. In 300 bar containers, about 0.1 MJ/L and 0.1 MJ/kg is achievable, comparable to the values of electrochemical lead-acid batteries. While batteries can somewhat maintain their voltage throughout their discharge and chemical fuel tanks provide the same power densities from the first to the last litre, the pressure of compressed air tanks falls as air is drawn off. A consumer-automobile of conventional size and shape typically consumes 0.3-0.5 kWh (1.1-1.8 MJ) at the drive shaft per mile of use, though unconventional sizes may perform with significantly less. It is important to take into consideration that the energy needed to

compress the air into the tanks comes from the electrical grid. Such electric grids are generally supported with use of fossil fuels. However, efforts are going on to make use of non-conventional sources of energy like solar and wind to compress air. Compressed air has relatively low energy density. Air at

30 MPa (4,500 psi) contains about 50 Wh of energy per liter (and normally weighs 372g per liter). From figure 2, it is very clear that power input requirement depends directly on the mass flow rate of the air to be compressed at a desired compression pressure [14, 16-17].

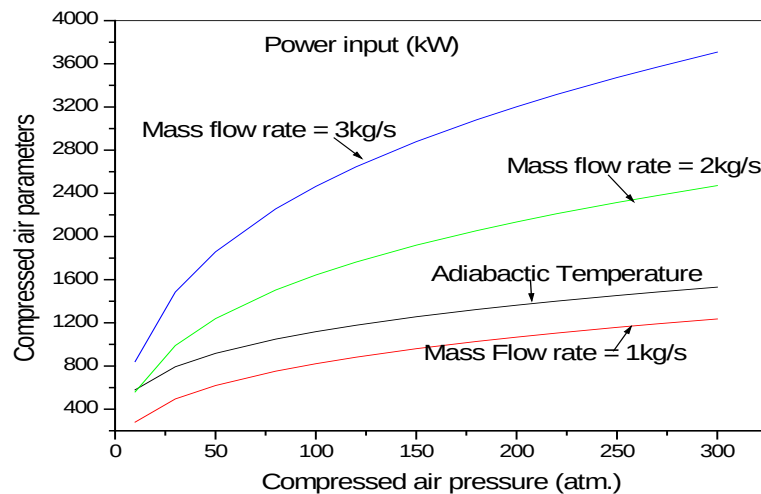


Fig. 2: Power input requirement for compressed air as a function of compressed air pressure

d) Temperature Control

In adiabatic process of compression of air [15], the heat of compression is retained, that means, there is no heat exchange resulting in zero entropy change, so the compressed air becomes very hot. Compressed air would experience a temperature rise due to the energy that has been added to the gas by the compressor (Figure 2) which can be controlled with isothermal or adiabatic compression processes. In an isothermal compression process, the gas in the system is kept at a constant temperature throughout. This necessarily requires removal of heat from the gas. This heat removal can be achieved by heat exchangers (intercooling) between subsequent stages in the compressor. To avoid wasted energy, the intercoolers must be optimized for high heat transfer and low pressure drop. Naturally this is only an approximation to an isothermal compression, since the heating and compression occurs in discrete phases. Some smaller compressors can approximate isothermal compression even without intercooling, due to the relatively ratio of surface area to volume and the resulting improvement in heat dissipation from the compressor body itself. An adiabatic process is one where there is no heat transfer between the fluid and the surroundings: the system is insulated against heat transfer. If the process is furthermore internally reversible (smooth, slow and frictionless, to the ideal limit) then it will additionally be isentropic. An adiabatic storage system does away with the intercooling during the compression process,

and simply allows the gas to heat up during compression, and likewise to cool down during expansion. This is attractive, since the energy losses associated with the heat transfer are avoided, but the downside is that the storage vessel must be insulated against heat loss.

e) Multistage Compression

Power requirement can be controlled through multistage compression of air [16-17]. The compression generates heat, the compressed gas is to be cooled between stages making the compression less adiabatic and more isothermal. The inter-stage coolers typically result in some partial condensation that is removed in vapor-liquid separators. By cooling the compressed air between each stage, the compression curve moves to near isothermal. The work done by the compressor is less if it is multi-staged. A multi-stage compressor is one in which there are several cylinders of different diameters. The intake of air in the first stage gets compressed and then it is passed over a cooler to achieve a temperature very close to ambient air. This cooled air is passed to the intermediate stage where it is again getting compressed and heated. This air is again passed over a cooler to achieve a temperature as close to ambient as possible. Then this compressed air is passed to the final or the third stage of the air compressor where it is compressed to the required pressure and delivered to the air receiver after cooling sufficiently in an after-cooler. Figure 3 indicates the

power required is highest for single stage compression and lowers with multistage compression. Table 1 gives the Lorentz fit equation (Fig. 3) constants for a ready

hand calculation of power requirement with multistage air compression.

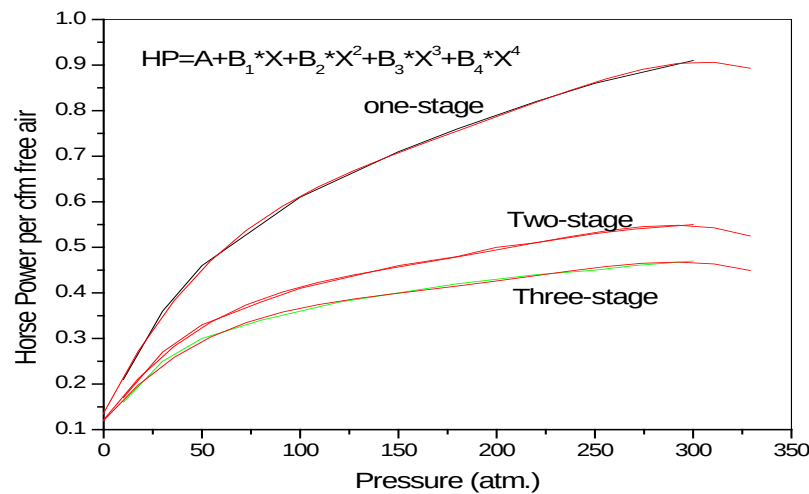


Fig. 3 : Multi-stage power requirement for air compression

Table 1 : Power input Lorentz fit equation constants

Constants	Value of constants		
	Single-stage	Two-stage	Three-stage
A	0.13677	0.12183	0.12042
B ₁	0.00862	0.00582	0.00501
B ₂	-5.5601x10 ⁻⁵	-4.2453x10 ⁻⁵	-3.7148x10 ⁻⁵
B ₃	1.95142 x10 ⁻⁷	1.54986 x10 ⁻⁷	1.33957x10 ⁻⁷
B ₄	-2.57x10 ⁻¹⁰	-2.0780 x10 ⁻¹⁰	-1.7669x10 ⁻¹⁰

f) Properties of compressed air

Compressed air energy storage is a way to store energy generated at one time for use at another time. Air driven motors use the energy of a compressed gas to do useful work. The power output is simply the inlet enthalpy minus the discharge enthalpy times your mass flow rate. The variation of some of its thermodynamical and physical properties like enthalpy, internal energy and density with compression pressure is shown in figure 4. The inlet enthalpy is a known state which we can calculate knowing the inlet pressure and temperature [15, 16-17]. Internal energy gives the energy contained in the compressed air and the idea about how much work can thus be extracted from it. Similarly, density of the compressed air can indicate the volume level to which the air can be compressed corresponding to a desired value of pressure.

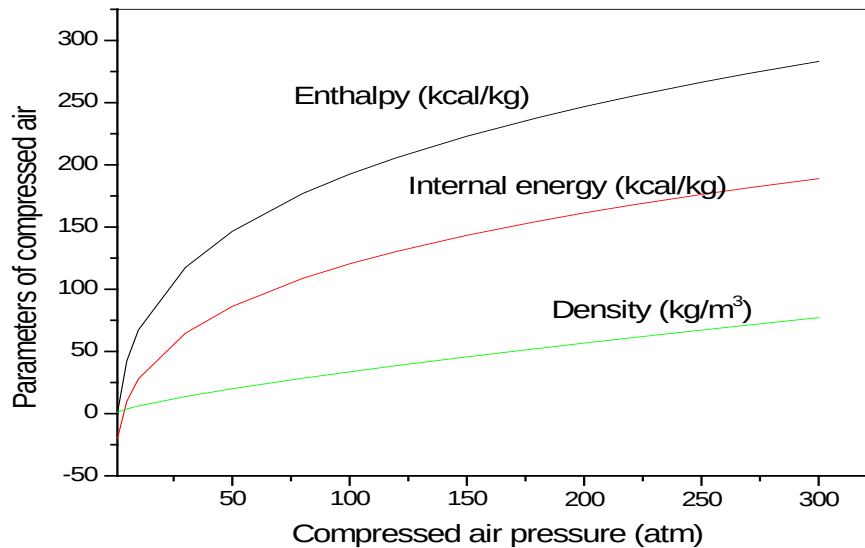


Fig. 4 : Variation of enthalpy, internal energy and density of compressed air

g) Energy Released

Compressed air is more expensive than many other utilities when evaluated on a per unit energy delivered basis. Therefore, energy released by the compressed air is very important and thus are the factor like volume of compressed air, compressed air pressure and expansion flow rate under isothermal or adiabatic conditions of expansion on which it depends [14, 16-17]. Figure 5 shows the typical control characteristics of energy release by the compressed air on its isothermal or adiabatic expansion as a function of compressed air pressure, volume of compressed air and flow rate.

Applying best energy release management and control practices, the desired energy can be achieved. Figure 5a,b, c indicate that energy released for isothermal expansion is always higher than adiabatic expansion when volume of compressed air and flow rate are controlled at a particular pressure of compression. However, when compression pressure is fixed with respect to flow rate and volume of air, then adiabatic expansion gives more energy release as compared to isothermal expansion.

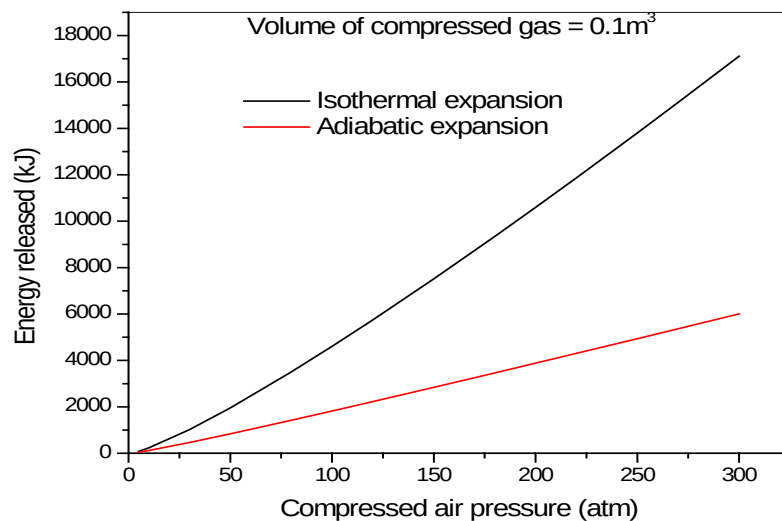


Fig. 5a : Energy released as a function of compression pressure at constant volume of compressed air

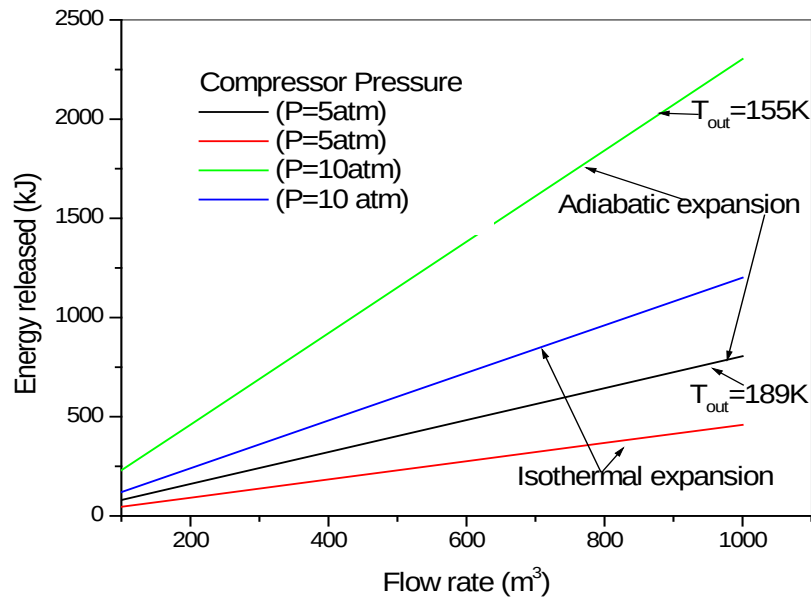


Fig. 5b : Energy released as a function of air flow rate at a particular compression pressure for isothermal and adiabatic expansion

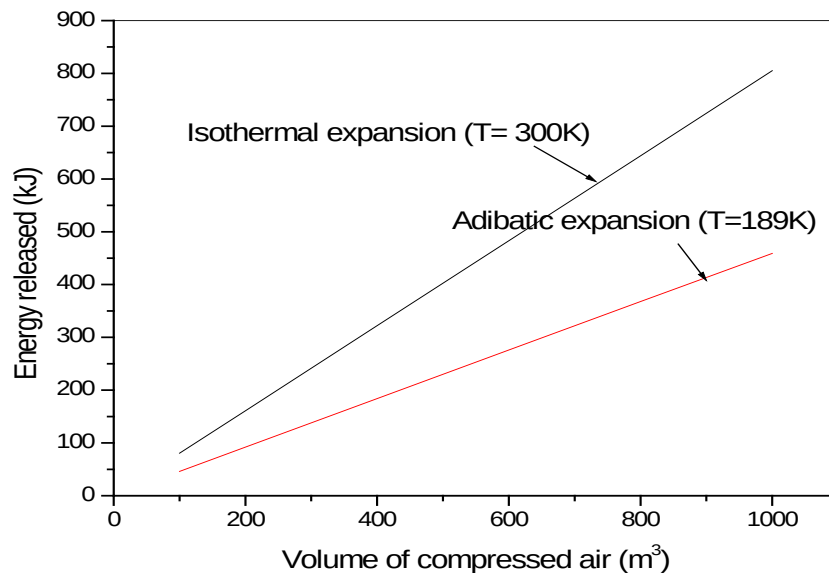


Fig. 5c : Energy released as a function of volume of compressed air at a particular compression pressure and flow rate for isothermal and adiabatic expansion

h) Adiabatic Expansion

Compressed-air vehicles operate to a thermodynamic process as air cools down when expanding and heats up when being compressed [14-17]. It is said that the compressed air vehicle project is facing difficulties in terms of vehicle range and cooling as excessively low engine temperature caused by the adiabatic expansion of the compressed air is a major

problem. Typical air engines use one or more expander pistons or rotary expander. Figure 6 clearly shows a sharp decrease in temperature of the air at the outlet under adiabatic expansion conditions. The cooling of air is very large in case of higher compression pressure. It is necessary to heat the air or the engine during expansion. As it is not possible in practice to use a theoretically ideal process, losses occur and

improvements may involve reducing these, e.g., by using large heat exchangers in order to use heat from the ambient air and at the same time provide air cooling in the passenger compartment. At the other end, the

heat produced during compression can be stored in water systems, physical or chemical systems and reused later.

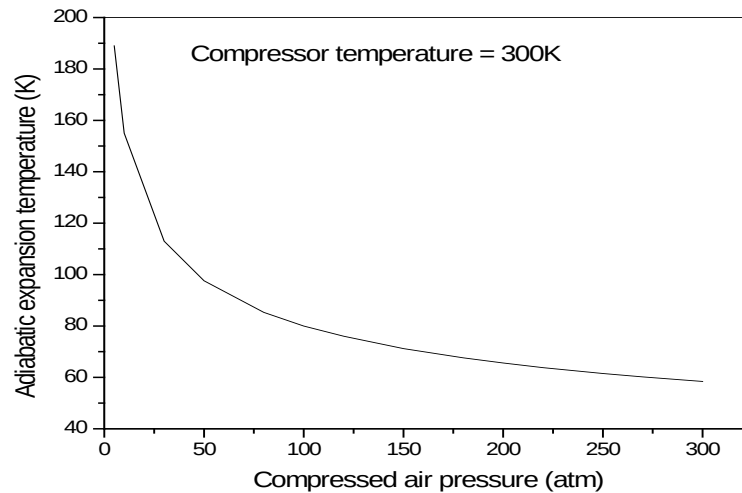


Fig. 6 : Compressed air temperature after adiabatic expansion

i) Pollutant Emissions

Like other non-combustion energy storage technologies, an air vehicle displaces the emission source from the vehicle's tail pipe to the central electrical generating plant. Where low emissions sources are available, net production of pollutants can be reduced. Emission control measures at a central generating plant may be more effective and less costly than treating the emissions of widely dispersed vehicles. Since the compressed air is filtered to protect the compressor machinery, the air discharged has less suspended dust in it, though there may be carry-over of lubricants used in the engine. The air-powered car would normally emit air, as it's what it would solely use. But it would totally depend on the purity of air that is put into the air tank as well as the temperature maintained inside the storage tanks during the process of compression. The rise in temperature of the compressed air as a function of compressed pressure [14-17] is shown in figure 7. As can be seen from the figure 7, temperature rises significantly if the air is compressed single stage without providing any facility in the tank to absorb the heat produced. This increase in temperature can lead to the dissociation of air nitrogen part and its chemical reaction with oxygen to produce unwanted gaseous compounds like NO, N₂O and NO₂ in significant concentrations. This type of emission can increase if impure air is filled in the tank.

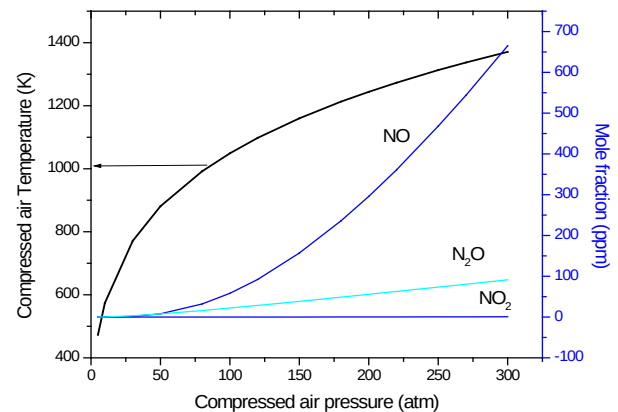


Fig. 7 : Concentration variation of NO, N₂O and NO₂ with compression air temperature rise as a function of compressed air pressure

V. CONCLUSION

It's important to remember that while vehicles running on only compressed air might seem like a distant dream, but they still have public interest due to their environmental friendly nature. Efforts should be to make them light, safe, cost effective and economical for deriving. The storage of compressed air (initially as well as during journey) with all benefits like no heating, high energy density and provisions to make use of cooling produced during adiabatic expansion during the energy release have to be taken care off in a much more controlled manner. Electric-powered cars and bikes already available on the market put a strong competition to compressed air car not only in terms of cost but also

their environment friendly role. The technology still looks distant but that has not deterred inventors from working on it.

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Epicyclic Power Split Transmissions for Hybrid Electric Vehicles: Fuel Consumption, Vehicle Performance and Driving Aggressiveness

By Ahmed Elmarakbi, Qinglian Ren & Rob Trimble

University of Sunderland

Abstract - This paper focuses on the improvement of fuel consumption, vehicle performance, and driving aggressiveness for Hybrid Electric Vehicles (HEV) using epicyclic power split transmission (PST). In this paper, three and four branch transmission system models are analyzed and implemented into two HEVs. The modeling of the hybrid electric vehicle performance is carried using the QSS Toolkit. The models for the two vehicles are built based on the same baseline vehicle with the same vehicle parameters but with different transmissions. The vehicles are simulated over typical driving cycles; New European Driving Cycle (NEDC) and USA Federal Test Procedure FTP 75. The HEV results are also compared against a conventional IC engine plus manual gearbox vehicle. Computer programs for the analysis of epicyclic transmission based on a matrix method are developed and used. The simulations show that the 4 branch epicyclic offers substantial improvements of reduction in energy consumption, though the benefits are sensitive to the driving cycle used.

Keywords : *hybrid electric vehicle, transmission, modeling, numerical simulations, efficiency, energy consumption, performance, aggressiveness.*

GJRE-B Classification : *FOR Code: 090205*



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Ahmed Elmarakbi ^α, Qinglian Ren ^σ & Rob Trimble ^ρ

Abstract - This paper focuses on the improvement of fuel consumption, vehicle performance, and driving aggressiveness for Hybrid Electric Vehicles (HEV) using epicyclic power split transmission (PST). In this paper, three and four branch transmission system models are analyzed and implemented into two HEVs. The modeling of the hybrid electric vehicle performance is carried using the QSS Toolkit. The models for the two vehicles are built based on the same baseline vehicle with the same vehicle parameters but with different transmissions. The vehicles are simulated over typical driving cycles; New European Driving Cycle (NEDC) and USA Federal Test Procedure FTP 75. The HEV results are also compared against a conventional IC engine plus manual gearbox vehicle. Computer programs for the analysis of epicyclic transmission based on a matrix method are developed and used. The simulations show that the 4 branch epicyclic offers substantial improvements of reduction in energy consumption, though the benefits are sensitive to the driving cycle used.

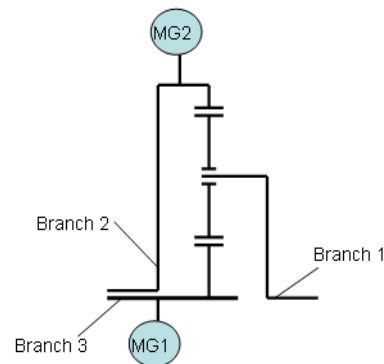
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1. INTRODUCTION

Continuously Variable Transmissions (CVTs) have been around for many years and the cost-benefit issues relating to CVTs are well understood. The potential advantages are improved performance, economy and emissions or more importantly an improved compromise between them. Their disadvantages have been cost, complexity, noise and driving refinement. Only over the past five years or so has the development of CVTs reached a stage at which they are beginning to be genuinely competitive with the alternatives, e.g. conventional, torque converter automatics and automated manual gearboxes, such as the twin clutch VW DSG system.

An important type of CVT is the E-CVT (Electronically-Controlled Continuously Variable Transmission), a good example of which is Toyota Hybrid System (THS). It combines the characteristics of an electric drive and a continuously variable

Transmission, using motor generator units in addition to toothed gears (Toyota Prius Transmission, 2006; Miller and Miller, 2005). In a THS system (see figure 1), one of the motor generators (MG2) is mounted on the driveshaft, and thus couples torque into or out of the driveshaft. The second motor generator (MG1) is connected with the sun gear and used to change the sun gear speed. Because MG2 is connected with the driveshaft, it cannot change speed and torque freely. Hence there are three power input/output branches in the system: the engine, MG1, the output, MG2. Because the speed of the output shaft is decided by the speed of the vehicle, there is some limitation on the control strategy to achieve optimum performance.

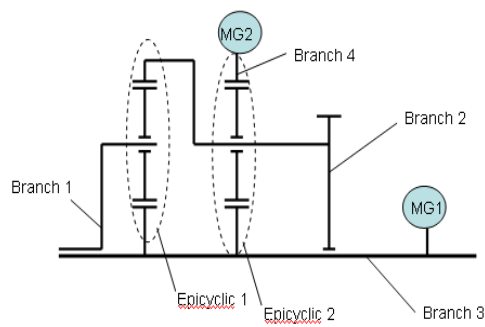


Branch 1- engine input shaft; Branch 2 – output shaft, also connected with MG2; Branch 3 – shaft connected with MG1

Fig. 1 : Three branch system

In a four branch transmission system, which is presented in this paper, neither of the motor generator units is mounted on the driveshaft or on the engine input shaft, which gives more freedom and benefits to the system (see figure 2).

^{Author ^α ^σ ^ρ} : Department of Computing, Engineering and Technology, Faculty of Applied Sciences, University of Sunderland, Sunderland SR60DD, UK. E-mail : ahmed.elmarakbi@sunderland.ac.uk



Branch 1- engine input shaft; Branch 2 – output shaft; Branch 3 - connected with MG1; Branch 4 – connected with MG2

Fig. 2 : Four branch system

One motor/generator is connected to the sun gear, and the other motor/generator is connected with a ring gear. So there are four branches of power input/output: the engine, the output shaft, and two motor generator units, MG1 and MG2.

This type of four branch transmission system has been described recently by Moeller (2006) who proposed that it offers advantages in many automotive applications. However, its usage in a hybrid electric vehicle driveline will be studied here.

A matrix method is used here to analyse the planetary transmission system, as introduced by Tian and Lu (1997). The key point of this method is to generate matrices to represent all the planetary train elements and other auxiliary components of the transmission.

II. MATHEMATICAL MODELING

The modeling of the hybrid electric vehicle performance is carried out using the QSS Toolkit (Guzzella and Amstutz, 2005). This is a quasistatic simulation package based on a collection of Simulink blocks and the appropriate parameter files that can be run in any Matlab/Simulink environment. The traditional ICE vehicle model itself is straightforward and is shown in figure 3.

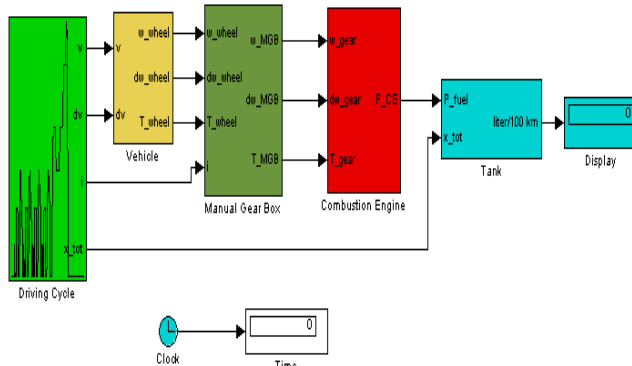


Fig. 3 : Overview of the conventional ICE vehicle model

There are 5 sub-systems: the driving cycle subsystem, vehicle subsystem, the gearbox subsystem, the combustion engine subsystem, and the fuel tank subsystem. The data for the engine and gearbox are taken from generic data in the QSS package. The function of the engine subsystem is to compute the fuel consumption from a consumption map, according to the torque and the rotational speed demand. The gearbox has 5 gears ranging from 3.84 to 0.63. The differential gear ratio is 3.95. The other vehicle data are shown in table 1. It is not intended to represent any specific vehicle – but rather to act as a generic vehicle platform to focus attention on the differences obtainable from the three vehicle models.

Table 1 : Vehicle Parameters (Miller, 2004)

Vehicle curb weight	1257 kg
Drag coefficient, C_d	0.29
Frontal Area	2.23 m ²
Tyre radius	0.292 m
Final drive	3.95:1

The models for the two hybrid electrical vehicles are built based on the same baseline vehicle, with the same vehicle parameters, but different transmissions, as shown in figure 4. The input to the model is one of the standard driving cycles – the NEDC and USA FTP-75 cycles are used extensively in this work – and the solution procedure is based on stepping through the driving cycle at typically one second steps, calculating the equilibrium condition and then collecting all the data for plotting at the end of the cycle. Thus, the focus of attention is on the overall efficiency of the engine and motor generator units and the major issue of whether it is possible to improve overall energy usage by operating the whole system at or near to the best efficiency points.

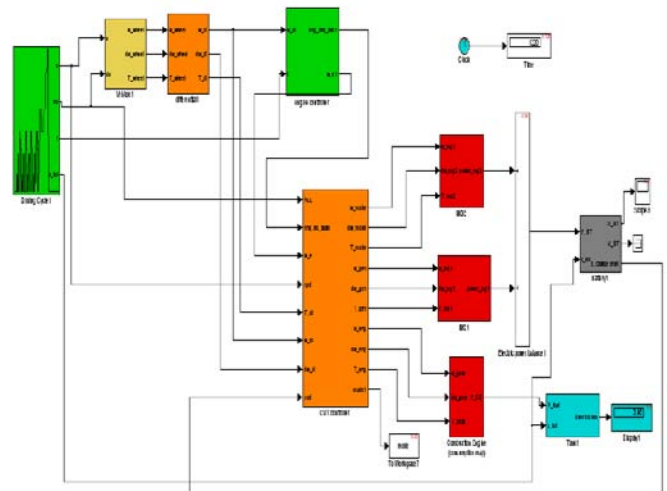


Fig. 4 : Overview of the hybrid vehicle model

III. CONTROL OF HYBRID VEHICLE DRIVELINES

The overall control of a hybrid drive train incorporating a power splitting transmission has proved to be an enormous challenge (Ren, and D. Crolla, 2007). This has already been shown to be the case for the first generation, single mode systems (Inoue et al., 2000), but looking at the complexity of the emerging dual mode systems (Cho et al., 2006), it is set to become even more challenging.

The overall control strategy can be summarized in broad terms as:

- Maximizing fuel economy
- Minimizing emissions
- Maintaining battery state of charge (SOC)
- Providing good performance and drivability

Central to these control objectives is the careful management of the power flows through the electrical and mechanical parts. Optimizing overall efficiency depends on a difficult compromise between operating the engine and transmission simultaneously around their most efficient conditions. The strategy for controlling the engine has received an enormous amount of attention, since this is an important issue irrespective of what transmission is used. Much emphasis has, not surprisingly, been placed on minimizing the brake specific fuel consumption (BSFC).

An overview of the high level, supervisory controller is shown in figure 5. Usually, the system components, e.g. engine, motor, generator etc. will also have their own local feedback loops and these are not considered here. Whatever the details of the actual controller, the approach used here is to combine it with a MATLAB model (e.g. figure 6) of the vehicle to compare the performance of alternative control strategies.

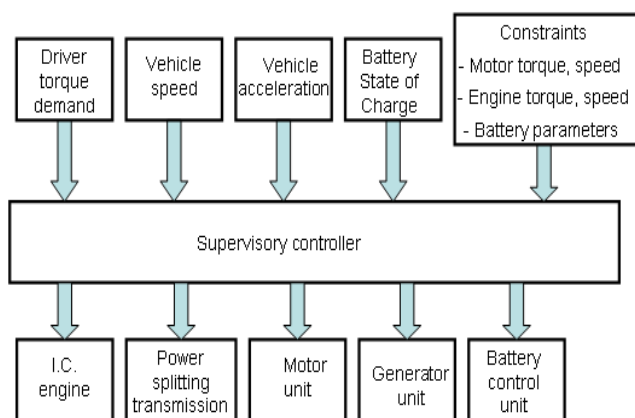


Fig. 5 : Controller Model

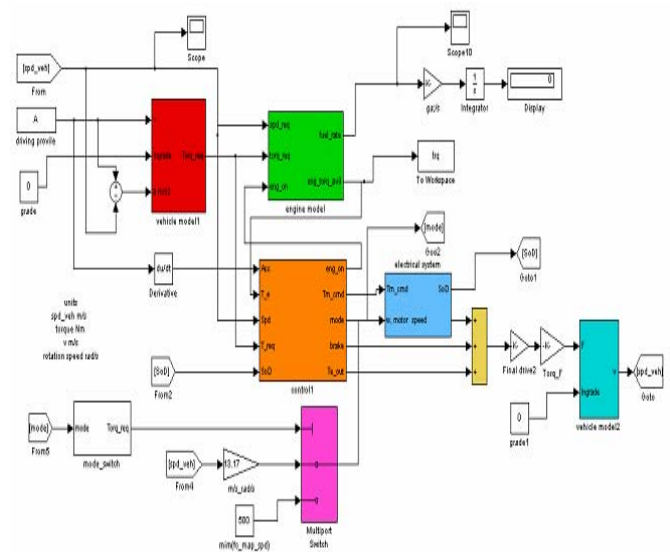


Fig. 6 : Vehicle Model

The control problem is often referred to as an optimal control problem, although this is often used in a general sense as opposed to a strict mathematical definition. The problem may be characterized in many ways depending on the performance objective, the hybrid vehicle model considered, the constraints imposed and the available control actions. However, as a general dynamics and control problem it is clearly not straightforward, and is likely to involve combinations of linear and non-linear elements, discrete and continuous systems, algebraic and dynamical systems

IV. FUEL CONSUMPTION

For HEVs, the difference between the initial and final battery SOC can significantly affect the measurement of fuel economy. To eliminate this effect, the concept of 'overall fuel consumption (OFC)' was introduced. The total additional energy stored or drawn from the battery (kWh) is calculated and then converted into how much fuel (liter) would be used for the engine to produce this amount of energy.

Engine fuel consumption (EFC, litre/100km): actual fuel burned by the engine divided by the driving distance Overall fuel consumption (OFC, litre/100km): the fuel consumption after taking the battery energy changed (BEC) into consideration

$$OFC = EFC + \frac{100 \times BEC \times \eta_{eng}}{\rho} / D \quad (1)$$

in which ρ is the fuel density (g/ml), η_{eng} is the engine efficiency (g/kWh) and D is the driving distance (m). The values for ρ and η are 0.76 g/ml and 240 g/kWh, respectively.

In the simulation, BEC is positive if energy is drawn from the battery and negative if the energy is stored into the battery. So at the end of each driving cycle, if final SOC is smaller than the initial SOC, namely the energy is drawn from the battery, overall fuel consumption is greater than the engine fuel consumption, and vice versa.

It is very important to take account of the battery SOC in the calculations, because if it is different at the end of the driving cycle from its value at the start then some net energy has effectively been lost or gained in the vehicle calculations. In several examples of results in the literature, it is not clear whether this effect has been accounted for. Also, some researches actually use the control system to ensure that the battery start and finish conditions are exactly the same. However, this can cause difficulties because the control system is not necessarily representative of what it would be doing during normal practical driving.

The first set of results was used to compare the two PST arrangements with a baseline, conventional vehicle equipped with a five speed gearbox (3.84, 2.11, 1.36, 0.86 and 0.63 with the same final drive ratio). The control strategies for the two PST arrangements were based on a rule-based approach to compromise between overall energy efficiency and maintaining the battery state of charge (SOC) under control. The vehicle models were run over NEDC and USA FTP-75 driving cycles, and the overall fuel consumption results are shown in table 2.

As expected, both hybrid vehicles show economy advantages over the conventional, manual gearbox vehicle. However, the improvements are not as great as published in some other studies, but this is understandable because the systems used here – and in particular their controllers – have not yet been optimized. The main aim of this work was rather to compare the 3 and 4 branch drivelines under exactly comparable conditions; this comparison is shown in table 3 and it shows that the 4 branch system offers around significant improvements over all the driving cycles. The improvements vary substantially with the different cycles, varying from 7.4% for the NEDC to 20% over the USA FTP-75 cycle.

Table 2 : Fuel consumption for the hybrid vehicle fitted with the 3 and 4 branch systems compared with a conventional, manual gearbox vehicle over different driving cycles

Driving cycle	Fuel consumption over driving cycle, l/100km				
	Traditional ICE	Single epicyclic system (3 branch system)		Dual epicyclic system (4 branch system)	
		Engine FC	Overall FC	Engine FC	Overall FC
Europe	3.8	4.0	2.7	3.9	2.5

NEDC					
USA FTP-75	3.6	3.8	3.0	3.7	2.4

Table 3 : Percentage improvement of the 4 branch over the 3 branch driveline over different driving cycles

Driving cycle	Percentage improvement of 4 branch over the 3 branch driveline (%)
Europe NEDC	7.4
USA FTP-75	20.0

The associated engine utilization maps are shown in figures 7 - 12 for the baseline gearbox, the single epicyclic gearbox and the twin epicyclic gearbox vehicles, respectively. Each point on the map of engine torque vs. speed is the solution at a single point during the NEDC cycle; the cycle defines input from $t = 0$ s to $t = 1220$ s. However, the NEDC cycle contains a percentage of constant speed running conditions, so that several points will sit on top of each other.

First, these results highlight in figures 7 and 10 the shortcoming associated with conventional IC engine cars – namely that they inevitably spend considerable time at part load conditions well away from the areas of maximum efficiency. In contrast, it can be seen in figures 8 and 11 for the single epicyclic gearbox, that it manages the IC engine rather well through the combination of effectively its continuously variable gear ratio plus its ability to manage the electrical power flows in and out of the battery. This results in the usage points being constrained around the area of maximum specific fuel consumption of the engine. Finally, in figures 9 and 12 it can be see that the dual epicyclic gearbox actually manages some further improvement and also reduces the use of the higher engine speeds.

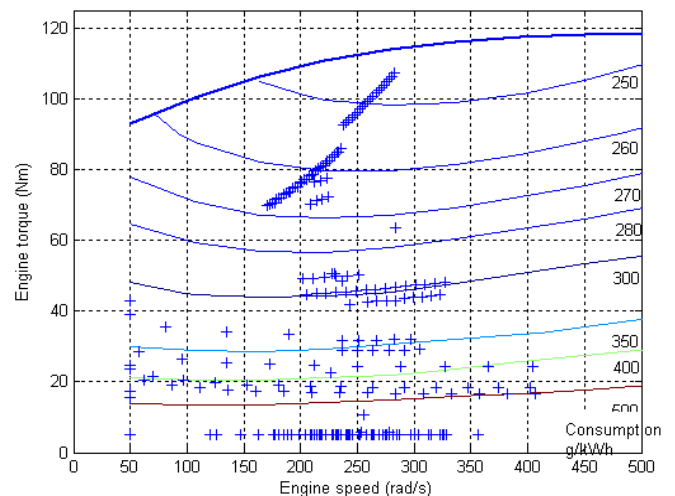


Fig. 7 : Engine operation points, NEDC cycle, traditional ICE vehicle

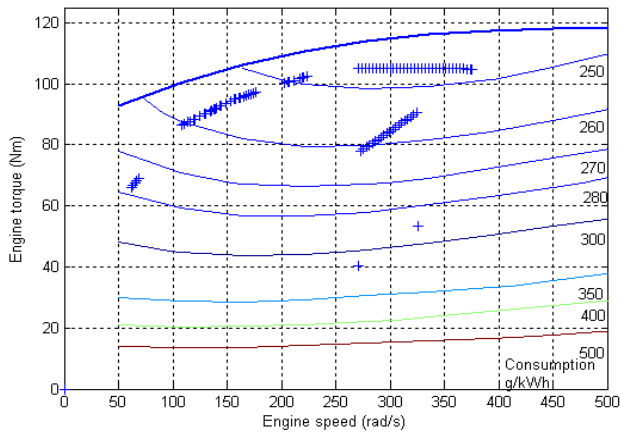


Fig. 8 : Engine operation points, NEDC cycle, single epicyclic syste

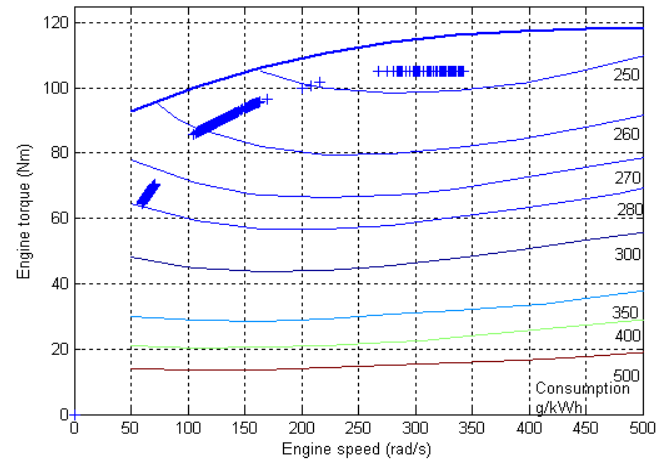


Fig. 11 : Engine operation points, FTP75 cycle, single epicyclic system

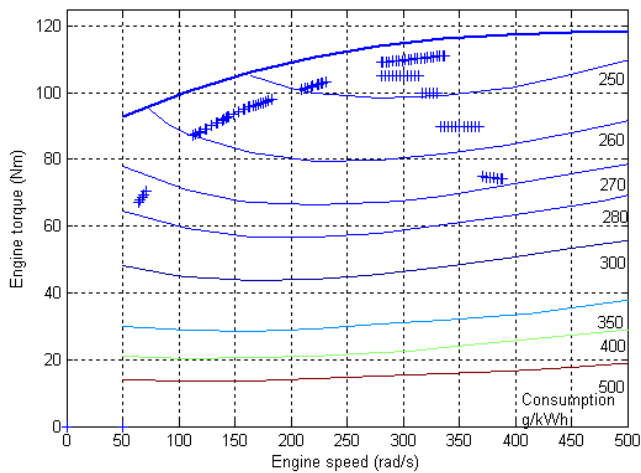


Fig. 9 : Engine operation points, NEDC cycle, dual epicyclic system

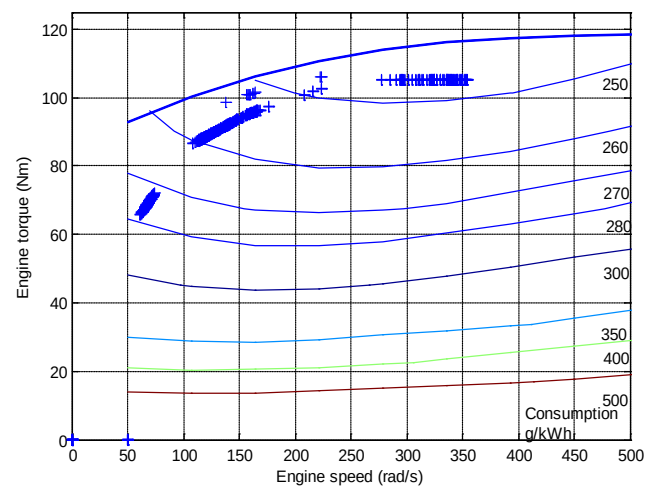


Fig. 12 : Engine operation points, FTP75 cycle, dual epicyclic system

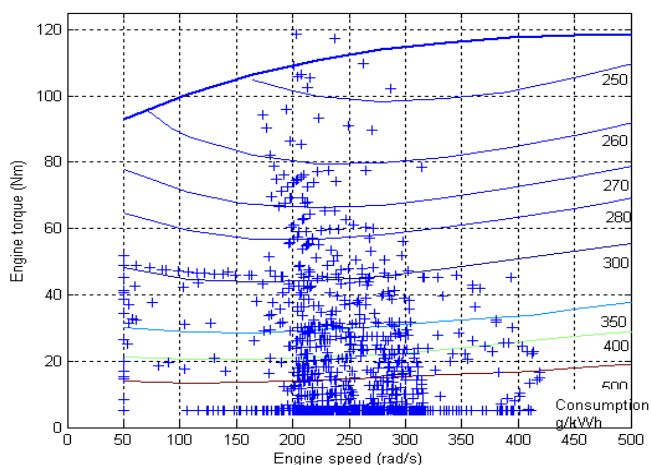


Fig. 10 : Engine operating points, FTP 75, traditional ICE vehicle

Further insight into the detailed behavior of the single and dual epicyclic gearboxes can be seen in the time history plots in figures. 13 and 14 for the NEDC cycle and figures. 15 and 16 for the USA FTP-75 cycle. The power utilization of the IC engine and two motor generator units, MG1 and MG2 are plotted along with the vehicle speed profile specified in each of these driving cycles.

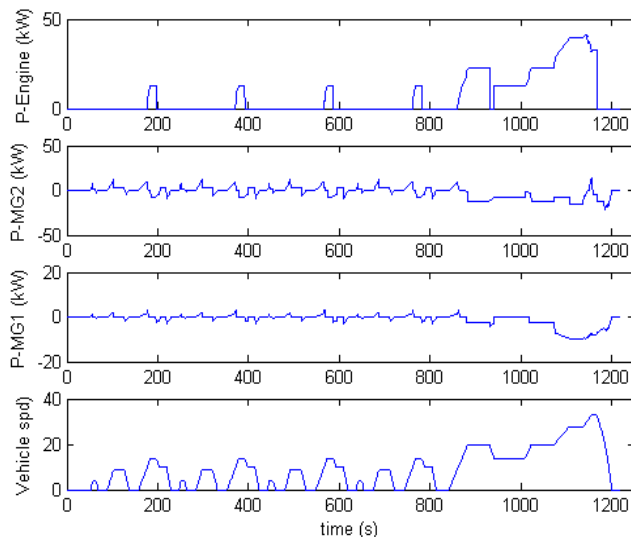


Fig. 13 : Power flows in the HEV with the single epicyclic gearbox over NEDC driving cycle

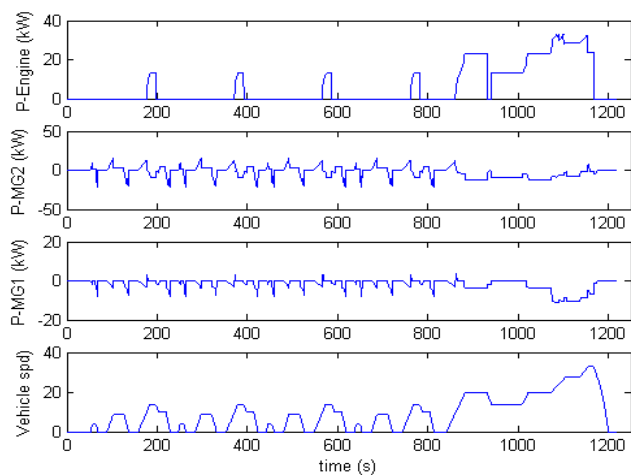


Fig. 14 : Power flows in the HEV with the dual epicyclic gearbox over NEDC driving cycle

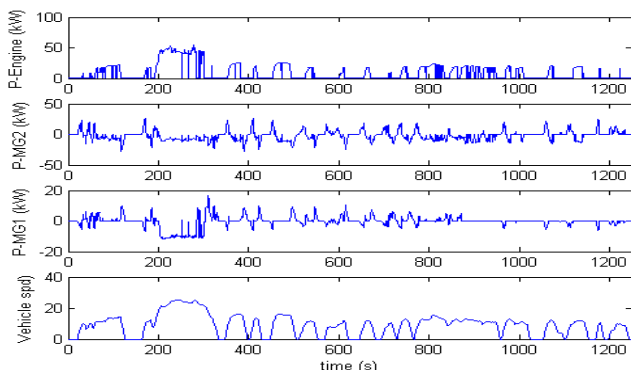


Fig. 15 : Power flows in the HEV with the single epicyclic gearbox over FTP-75 driving cycle

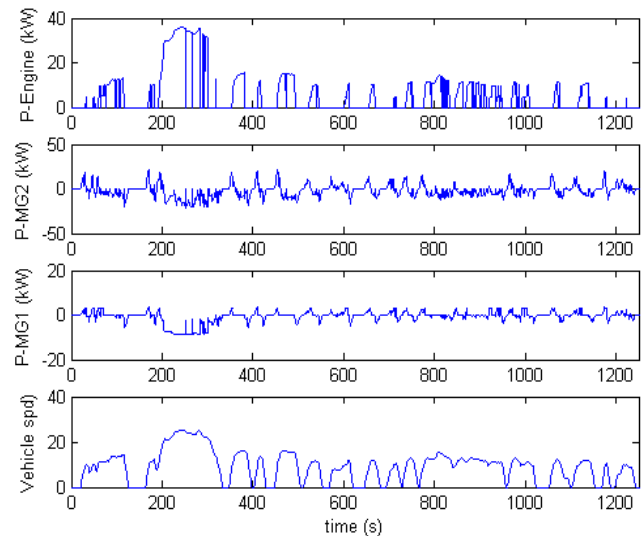


Fig. 16 : Power flows in the HEV with the dual epicyclic gearbox over the FTP-75 driving cycle

The results in figures 13 to 16 also highlight the substantial differences between the European and USA standard driving cycles; the European version contains a much greater number of stationary and constant speed running events, whereas the USA version is almost continually changing speed in its version of an urban cycle. This reflects a fundamental difficulty within the vehicle industry when it has to try and make fair comparisons of competing driveline technologies regarding their energy consumption – what actually constitutes a representative driving pattern over which to make comparisons? The answer is likely to vary across the three major global automotive markets – Europe, USA and the Far East.

The acceptance of a so-called standard cycle also raises another potential problem – that in the quest for the best headline figures, the driveline and particularly its controller is actually optimized around the specific cycle. This can lead to engineering developments based more around the standard than around a drivable, efficient vehicle across a wider range of operating conditions.

V. VEHICLE PERFORMANCE

a) Top Speed

For conventional IC engine vehicle, the top speed that can be reached on level road with a given transmission ratio can be found by intersecting the curve of the available power at the wheels with that of the required power on level road (Genta and Morello, 208). The available power for gears I, II, III, IV and V are calculated as: for the given gear ratio, assume the engine always works on the “engine speed-maximum engine torque” curve.

$$P_a = \eta_t P_e = \eta_t \omega_{eng} \times T_{eng_max} \quad (2)$$

$$P_n = (F_{rr} + F_{ad}) \times V \quad (3)$$

Where P_a is the available power, P_n is the required power, T_{eng_max} is the engine's maximum torque, F_{rr} is the rolling resistance force, F_{ad} is the aerodynamic drag, and η_t is the efficiency of the whole powertrain. The vehicle speed under each gear is

$$V = \frac{\omega_{eng}}{i_{gear} \times i_{differential}} \times r \quad (4)$$

where r is the radius of the tire, i_{gear} is the transmission ratio of each gear, and $i_{differential}$ is the transmission ratio of the differential. The power required and the available power for each gear ratio are shown in figure 17. For the IC engine vehicle the top speed is 54.5 m/s (196.2km/h). For full hybrid vehicles, if the engine is not downsized, the vehicle is almost retaining to have better overall performance compared with a conventional vehicle (Chan, 2007). As far as the top speed is concerned, because the power split device is actually a CVT, it can change the transmission ratio to achieve the highest top speed, as shown in figure 18. The top speed for the HEV with an E_CVT is 58.6 m/s (210.9 km/h).

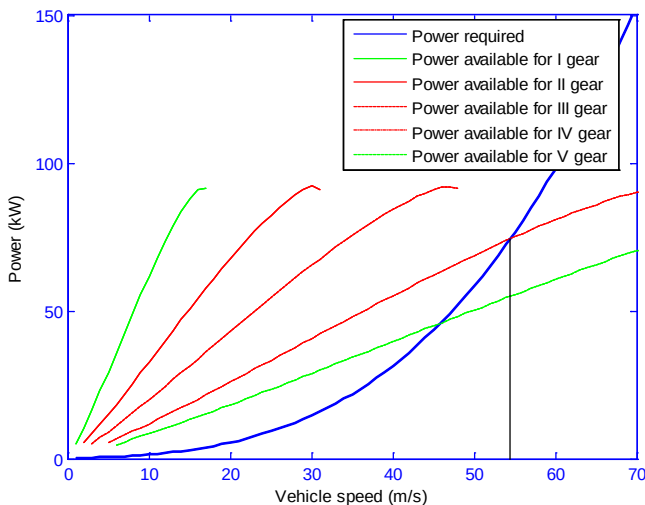


Fig. 17 : Top speed for the conventional vehicle (55.92m/s)

In practice, for a full HEV it is normal design practice to downsize the IC engine. The challenge for the controller design is then to maximize the time spent by the engine toward its optimum efficiency region by controlling the power flows to and from the battery.

However, this should not compromise overall performance and drivability compared to the equivalent IC engine vehicle. So the controller exploits the continuously variable transmission ratio to address this compromise.

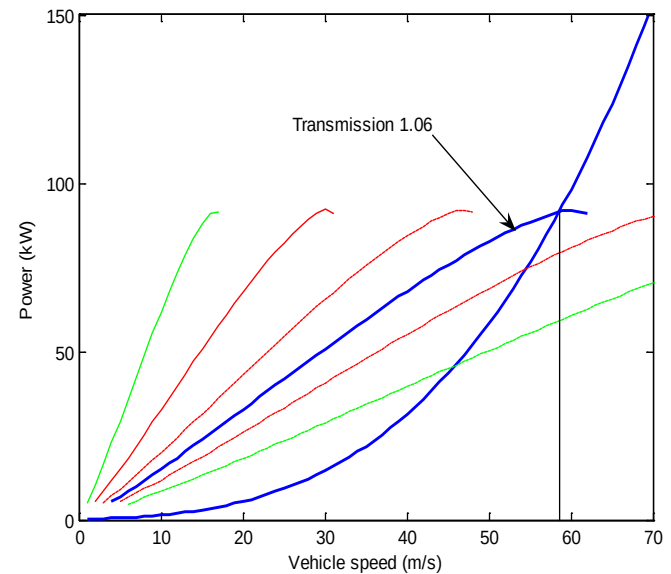


Fig. 18 : Top speed for a HEV with an E-CVT

b) Acceleration

The maximum acceleration a vehicle is capable of at various speeds is [9]:

$$\left(\frac{dV}{dt} \right)_{\max} = \frac{\eta_t P_e - P_n}{m_e V} = \frac{\eta_t P_e - (F_{rr} + F_{ad})V}{m_e (V + 1)} \quad (5)$$

where m_e is the equivalent mass of the vehicle,

P_e is the engine power, and P_n is the required power. The plot of maximum acceleration versus vehicle speed of the tradition IC engine vehicle with a five speed gear box is shown in figure 19. The minimum time need to accelerate from speed V_1 to V_2 can be calculated by integrating Eq. (5), but usually numerical integration is performed. A graphical interpretation of the integration is shown in figure 20, which plots $(1/\text{acceleration})$ vs. vehicle speed. The area under the curve $1/a$ is the minimum time required for the acceleration.

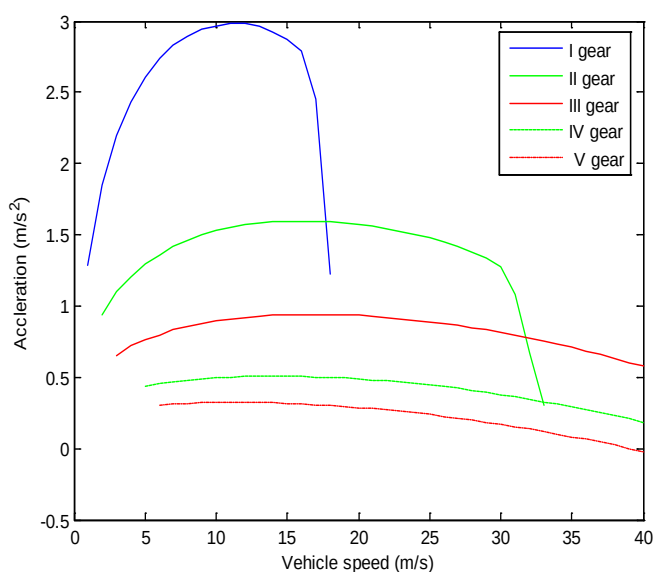


Fig. 19 : Maximum acceleration vs. vehicle speed

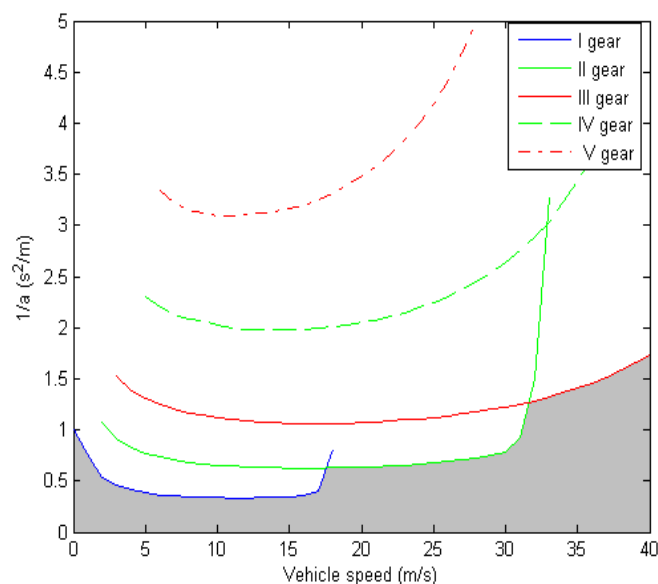


Fig. 20 : 1/a versus vehicle speed

The minimum time of acceleration for a HEV with a PST (power split transmission) is shown in figure 21. The dark area is the time to speed for a HEV with an E-CVT. The time-speed curve, as shown in figure 22 can be obtained by integrating the dark area in figures. 20 and 21. From figure 22, it shows that the time to accelerate from 0 to 100 km/h (27.78 m/s) for a vehicle with a geared transmission and a HEV with a PST is 12.3 s and 14.4 s, respectively.

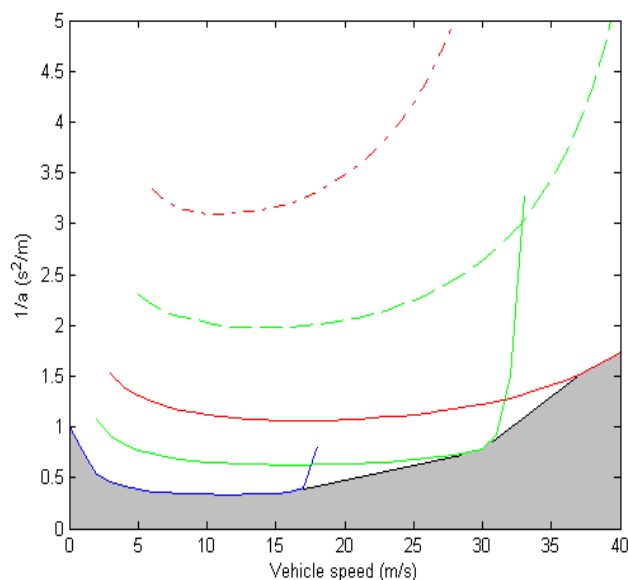


Fig. 21 : 1/a versus vehicle speed

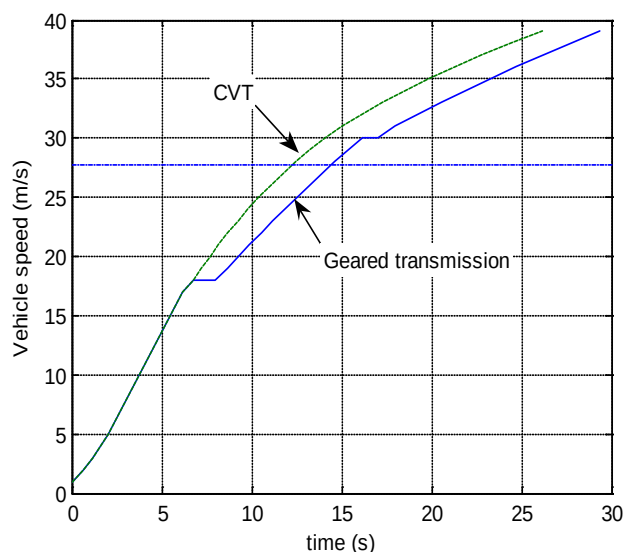


Fig. 22 : Speed vs. time curve

VI. DRIVING AGGRESSIVENESS

A study at the Argonne laboratory demonstrated that HEVs have higher fuel consumption sensitivity to aggressive driving (Sharer, 2007). They define more aggressive driving to mean more use of periods of higher accelerations that indicated in the typical driving cycles. In this study, the sensitivity of the two HEV models to driving aggressiveness was calculated and the results are shown below.

The FTP 75 driving cycle is used as a baseline driving input for the comparisons. Then, a simple multiplier factor of 0.8, 0.9, 1.1 and 1.2 was imposed to get cycles that represent different driving aggressiveness. The bigger the factor, the more aggressive the driving cycle becomes since the speeds and hence accelerations are simply increased. Factor

0.8 means the cycle is less aggressive than the baseline cycle and factor 1.2 means the cycle is more aggressive than the baseline cycle, as shown in figure 23. The simulation results are summarized in table. 4.

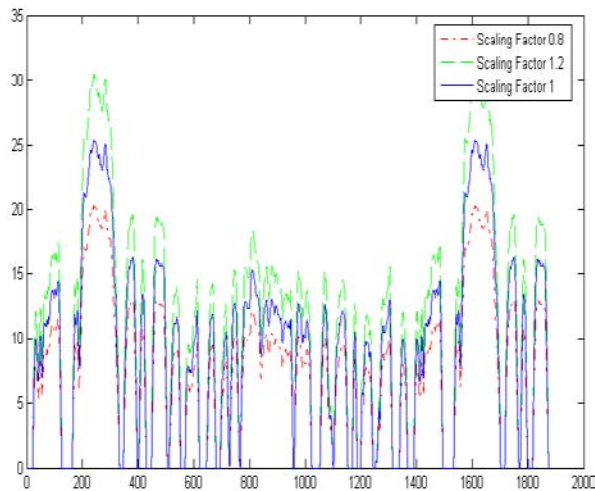


Fig. 23 : Driving cycles with different driving aggressiveness

Table 4 : Fuel consumptions with different driving aggressiveness

Scaling factor	Traditional ICE		Single epicyclic system		Dual epicyclic system	
	Overall FC	Normalized FC	Overall FC	Normalized FC	Overall FC	Normalized FC
0.8	3.39	0.93	3.07	0.84	2.64	0.73
0.9	3.50	0.96	3.03	0.83	2.37	0.65
1.0	3.64	1.0	3.02	0.83	2.36	0.65
1.1	3.79	1.04	3.57	0.98	2.64	0.72
1.2	3.97	1.09	4.42	1.21	3.22	0.88

Note: all the normalized fuel consumptions are normalized to 3.64, which is the overall fuel consumption of the traditional ICE with scaling factor 1.

The overall benefits of the dual epicyclic over the single epicyclic and the traditional ICE are shown to occur consistently over all limited range of driving aggressiveness tested here. Then, the same results are plotted in figure 24 showing the trend of the fuel consumption figures normalized to the traditional ICE at a scaling factor of 1. These curves provide information on how sensitive each system is to driving aggressiveness as indicated by the slope of the lines. The ICE case is shown to be rather insensitive; the twin epicyclic is somewhat less sensitive than the single epicyclic as the driving aggressiveness increases above 1. This again relates to the greater flexibility of managing the power flows between the mechanical and electrical paths.

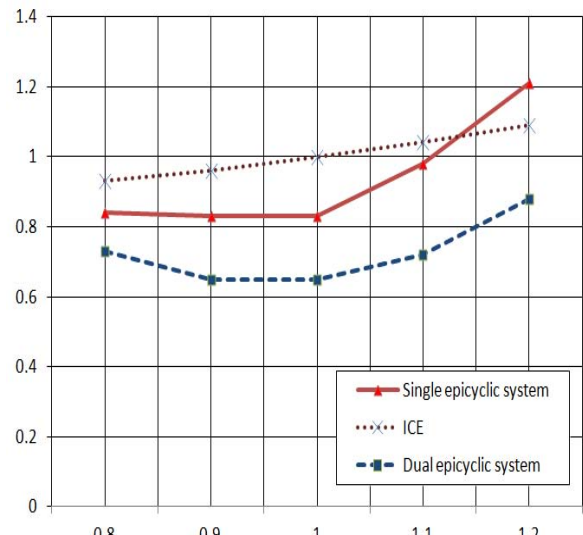


Fig. 24 : Sensitivity to driving aggressiveness – fuel consumption normalized to the baseline ICE condition

VII. CONCLUSIONS

The following conclusions are made:

- The four branch dual epicyclic gearbox arrangement offers a significant performance benefit over the three branch, single epicyclic arrangement; fuel economy improvements of 7 and 20% were shown over the two main European and USA driving cycles.
- The performance benefits arise from the greater flexibility of control over the torques, speeds and power flows through the two motor generator units available with the dual epicyclic scheme.
- For a HEV with a PST, which provides the benefits of a CVT gearbox, if the engine is not downsized, the HEV will have better drivability, namely higher top speed and shorter acceleration time.
- In practice, the normal design approach for a HEV is to downsize the engine, and improve overall fuel consumption, whilst exploiting the transmission and controller properties to obtain similar performance to the equivalent ICE vehicle.
- The four branch system is slightly less sensitive to increased driving aggressiveness than the 3 branch system, because it can control the power flows better according to different driving conditions.
- In practice, further benefits are probably available; first, the dual arrangement has two nodal positions at which zero electrical power circulates and these can be designed to occur at convenient speeds, e.g. in the UK, 30 mile/h in urban driving and 70 mile/h motorway cruising. Second, with the dual arrangement it is possible to downsize the motor

generator units to retain the same driveability but with reduced weight and cost.

- Although the results presented here have been based on relatively simple vehicle models, it is likely that the promising results for the twin epicyclic transmission could be further improved using more sophisticated optimisation strategies for the control system.

ACKNOWLEDGEMENT

The authors acknowledge with sadness, the contribution of Prof. Dave Crolla who has passed away during the period of this research.

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Sheet Metal Forming Simulations for Heavy Commercial Vehicle Parts by LS-DYNA

By Abdulla Mohammad Gous Shaikh & Tippa Bhimasankara Rao

Institute of Science

Abstract - The design of new forming tools gets more problemmatic as the geometries get more complicated and the materials less formable. The idea with this project is to evaluate if an implementation of simulation software in the designing process, to simulate the forming process before actually building the tools, could help to avoid expensive mistakes. To evaluate this, the commercial FEM simulation software LS-DYNA was used in a complicated project, where the design of the forming tools for forming a girder was considered. The main objective was to avoid cracking and severe wrinkling which may result in the forming process. With help of simulations a stable forming process which did not yield cracks or severe wrinkling, was eventually found. The girder was almost impossible to form without cracking, but the breakthrough came when we tried to simulate a performing step which solved the problem. Without simulation software this would never have been tested since it would be too risky and expensive to try an idea which could turn out to be of no use. The simulations also showed that the spring back - shape deformation occurring after pressing - was large and hard to predict without simulations. Therefore, the tools were also finally spring back compensated.

Keywords : forming, forming limit diagram (FLD), LS-DYNA, spring back, stamping, stress, strain.

GJRE-B Classification : FOR Code: 090299



SHEET METAL FORMING SIMULATIONS FOR HEAVY COMMERCIAL VEHICLE PARTS BY LS-DYNA

Strictly as per the compliance and regulations of :



Sheet Metal Forming Simulations for Heavy Commercial Vehicle Parts by LS-DYNA

Abdulla Mohammad Gous Shaikh ^α & TippaBhimasankara Rao ^σ

Abstract - The design of new forming tools gets more problematic as the geometries get more complicated and the materials less formable. The idea with this project is to evaluate if an implementation of simulation software in the designing process, to simulate the forming process before actually building the tools, could help to avoid expensive mistakes. To evaluate this, the commercial FEM simulation software LS-DYNA was used in a complicated project, where the design of the forming tools for forming a girder was considered. The main objective was to avoid cracking and severe wrinkling which may result in the forming process. With help of simulations a stable forming process which did not yield cracks or severe wrinkling, was eventually found. The girder was almost impossible to form without cracking, but the breakthrough came when we tried to simulate a performing step which solved the problem. Without simulation software this would never have been tested since it would be too risky and expensive to try an idea which could turn out to be of no use. The simulations also showed that the spring back - shape deformation occurring after pressing - was large and hard to predict without simulations. Therefore, the tools were also finally spring back compensated.

We concluded that simulations are very effective to quickly test new ideas which may be necessary when designing the tools for forming complicated parts. Simulation also provided detailed quantitative information about the

expected cracks, wrinkles, and weaknesses of the resulting pieces. Even though there is cost associated with simulations, it is obvious from this project that simulation software is actually very less as compared to actual tool building and making mistakes.

Keywords : forming, forming limit diagram (FLD), LS-DYNA, spring back, stamping, stress, strain.

I. INTRODUCTION

a) The forming process

Sheet metal forming, or stamping, is a process where a material, referred to as the blank, is formed by stretching it between a punch and a die, see figure (1). First, the punch and the binder or blank-holder is in an uplifted position. The binder then moves down and clamps the blank in such a way that the blank is allowed to be drawn inwards, but still creates tension to stretch the blank between the die and the punch. In somecases, draw-beads are also used to further increase the resistance of inwards drawing. After this, the punch moves down and the sheet is drawn through the opening of the die ring.

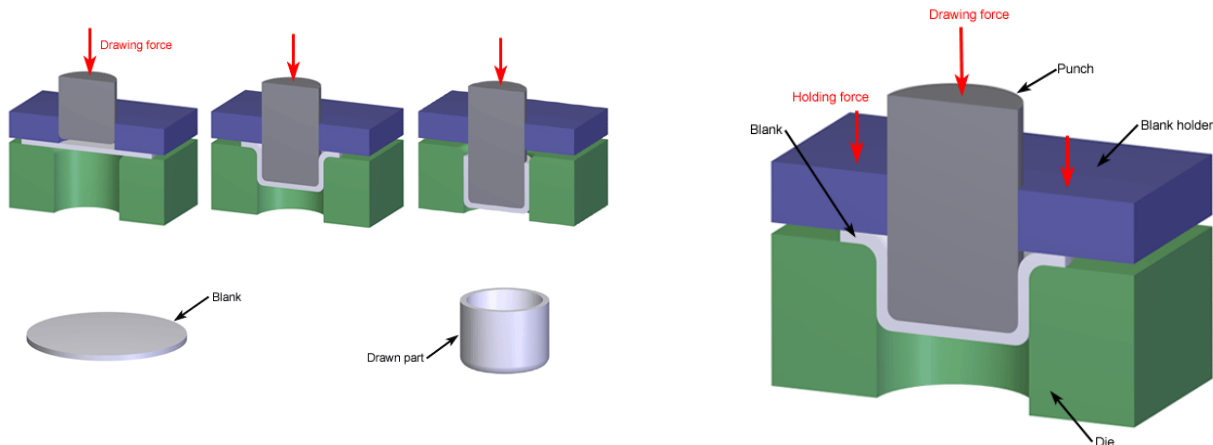


Figure 1

Note that the forming process is not a compression process, but rather a stretching process,

i.e. the blank is stretched over the tools. Since the blank, for the most part, only is in contact with the tooling on one side, no through-thickness compression is present. The flow stress in the sheet is generally much larger than the contact pressure, and thus, it is assumed that there is no through-thickness compression. This is called plane stress deformation. Stamping or forming

Author ^α : PG Student, Department of Mechanical Engineering, Nimra Institute of Science and Technology Vijayawada, AP, India.

Author ^σ : HOD, Department of Mechanical Engineering, Nimra Institute of Science and Technology, Vijayawada, AP, India.

can be done in a single or double-acting press. A double-acting press is a press in which the clamping and punch actions are separate. Usually the forming has to be done in several stages, different presses, if the desired geometry is complex.

II. MECHANICS OF SHEET METAL FORMING

Formability is the ability of a given metal work piece to undergo plastic deformation without being damaged. The plastic deformation capacity of metallic materials, however, is limited to a certain extent, at which point, the material could experience tearing or fracture (breakage).

Sheet metal parts are produced in large quantities using special tooling and high-volume production techniques. Most processes are tensile in nature and deformation in one stage is limited by the

onset of tensile instability, necking and tearing. Because of the compressive stress in some part of thin sheet membrane, wrinkling and buckling may also occur. This makes very important to understand the mechanics of the forming process, knowledge of failure analysis and considerable experience in manufacturing.

The true stress-strain curve as shown in fig 2. Derived from a tensile test can be applied to other deformation processes that may occur in typical sheet-forming operations. A common feature of many sheet-forming processes is that compared with the stresses in the plane of the sheet (the membrane stresses), the stress perpendicular to the surface of the sheet is small. If this normal stress is assumed zero, a major simplification is possible. Such a process is called plane stress deformation.

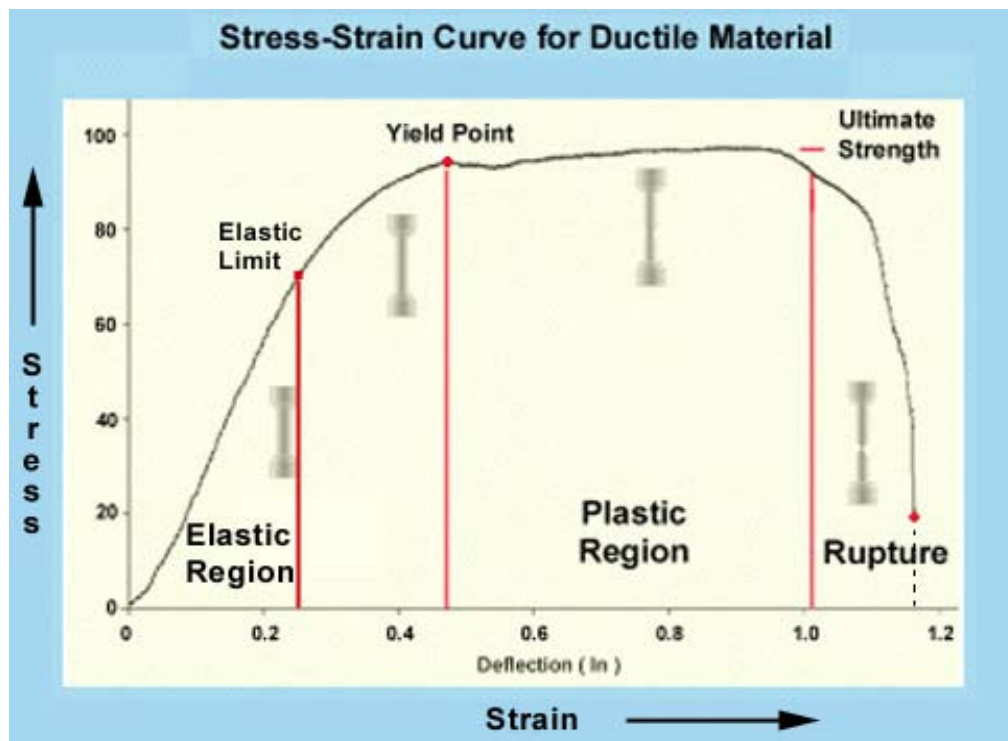


Figure 2

a) Forming-Limit Diagrams (FLD)

A forming limit diagram, also known as a forming limit curve, is used in sheet metal forming for predicting forming behaviour of sheet metal. The diagram attempts to provide a graphical description of material failure tests, such as a punched dome test.

In order to determine whether a given region has failed, a mechanical test is performed. The mechanical test is performed by placing a circular mark on the work piece prior to deformation and then measuring the post-deformation ellipse that is generated from the action on this circle. By repeating the mechanical test to generate a range of stress states,

the formability limit diagram can be generated as a line at which failure is onset.

Sheet metal is marked with small circles, stretched over a punch, and deformation is observed in failure areas. FLD shows boundary between safe and failure zones.

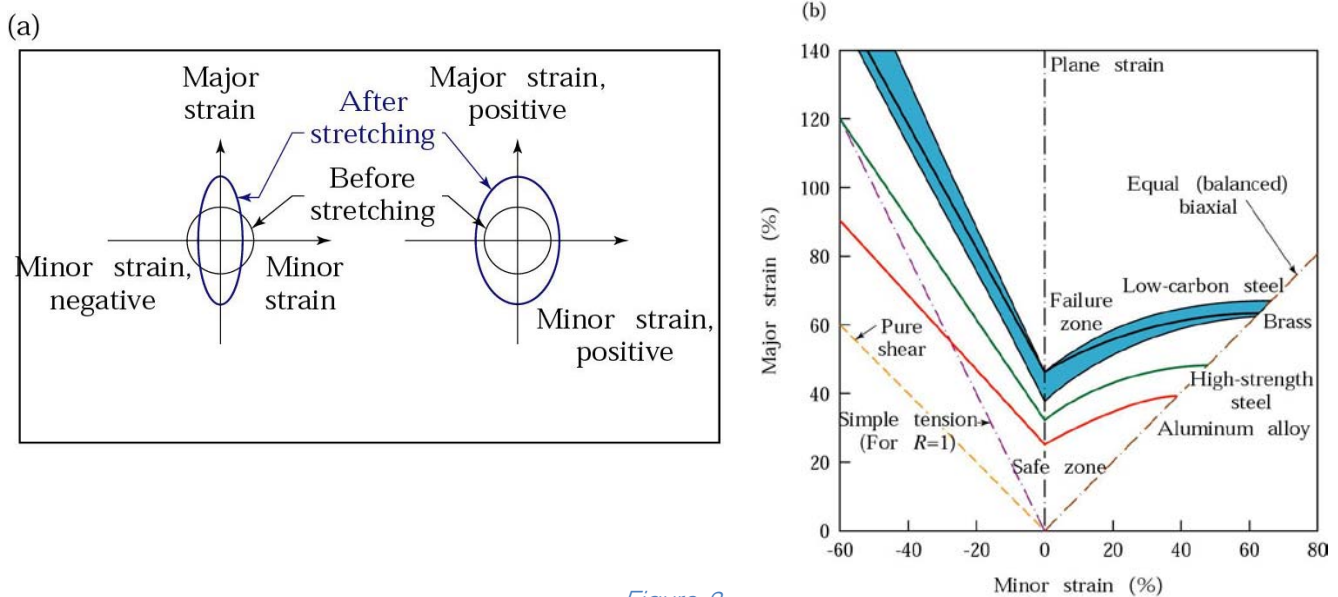


Figure 3

(a) Strains in deformed circular grid patterns. (b) Forming-limit diagrams (FLD) for various sheet metals. Although the major strain is always positive (stretching), the minor strain may be either positive or negative. In the lower left of the diagram, R is the normal anisotropy of the sheet.

III. FINITE ELEMENT METHOD

FEM is Method for numerical solution of field problems. In mathematics, finite element method (FEM) is a numerical technique for finding approximate solutions to boundary value problems. It uses variational methods (the Calculus of variations) to minimize an error function and produce a stable solution. FEM cuts a structure into several elements (pieces of the structure). Then reconnects elements at "nodes" as if nodes were pins or drops of glue that hold elements together. This process results in a set of simultaneous algebraic equations. Method for numerical solution of field problems.

a) LS-DYNA

LS-DYNA is an advanced general-purpose multiphysics simulation software package developed by the Livermore Software Technology Corporation (LSTC). LS-DYNA is capable of simulating complex real world problems. It is used by the automobile, aerospace, construction, military, manufacturing, and bioengineering industries. LS-DYNA is optimized for shared and distributed memory UNIX, Linux, and Windows based, platforms. The code's origins lie in highly nonlinear, transient dynamic finite element analysis using explicit time integration.

LS-DYNA's potential applications are numerous and can be tailored to many fields. In a given simulation, any of LS-DYNA's many features can be combined to model a wide range of physical events. An example of a simulation, which involves a unique combination of features, is the NASA JPL Mars Pathfinder landing simulation which simulated the space probe's use of airbags to aid in its landing. LS-DYNA is one of the most

flexible finite element analysis software packages available.

LS-DYNA consists of a single executable file and is entirely command line driven. Therefore all that is required to run LS-DYNA is a command shell, the executable, an input file, and enough free disk space to run the calculation. All input files are in simple ASCII format and thus can be prepared using any text editor. Input files can also be prepared with the aid of a graphical pre-processor.

There are many third party software products available for pre-processing LS-DYNA input files. LSTC also develops its own pre-processor, LS-Pre-Post, which is freely distributed and runs without a license. Licensees of LS-DYNA automatically have access to all of the program's capabilities, from simple linear static mechanical analysis up to advanced thermal and flow solving methods. Furthermore, they have full use of LS-OPT, a standalone design optimization and probabilistic analysis package with an interface to LS-DYNA.

One of LS-DYNA's most widely used applications is sheet metal forming. LS-DYNA accurately predicts the stresses and deformations experienced by the metal, and determine if the metal will fail. LS-DYNA supports adaptive remeshing and will refine the mesh during the analysis, as necessary, to increase accuracy and save time.

Metal forming applications include:

- Metal stamping
- Hydro forming
- Forging
- Deep drawing
- Multi-stage processes

IV. PROBLEM DEFINITION

The production of new stamping tools gets more troublesome, as the geometries get more complex and smaller tolerances on the dimensions are requested. This increases the risk for unexpected scenarios that could compromise the entire project. Thus, to deal with this, we want to try a more advanced simulation tool, than the one-step solver they already have, to investigate what it could mean for future development of the company.

To investigate this, we want to use a simulation program in a real project to see its benefits at work and then evaluate if investment in a simulation tool and in personal would be profitable.

In this project the design of stamping tools, for stamping a girder, placed in a truck, will be considered, see figure (4). Note that the geometry is very complex and stamping in a single forming step is not possible but several stages will be necessary to form the part.

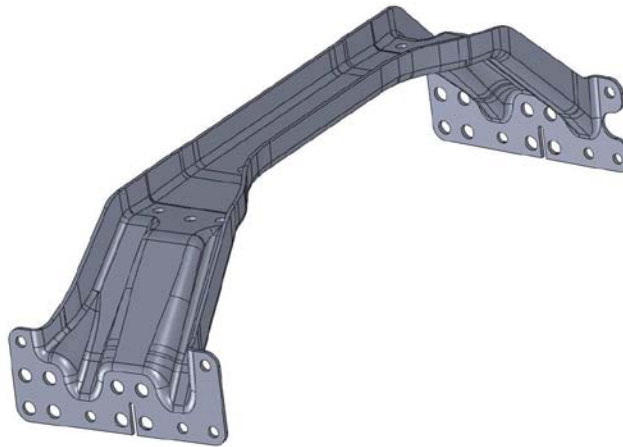


Figure 4 : Girder with sheet thickness 6 mm in Domex 500 MC

The girder has a very complex form and will, therefore, be formed in several stages.

V. SETTING UP SIMULATION

The steps for setting up a forming simulation in LS-DYNA are:

- Import the CAD-geometry into LS-DYNA and check surfaces.
- Create the blank or import it from previous forming step.
- Mesh the tool surfaces and the blank.
- Define material for the blank.
- Define the tool motions.

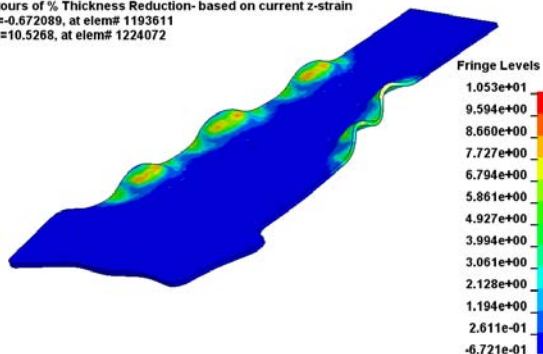
- Set up the controls for adaptive refinements, time-step etc.
- Run the simulations, for the trimming and forming steps, in LS-DYNA.
- Define constraints for spring back simulation in LS Pre-Post.
- Simulate the spring back.
- Evaluate the results in LS-Pre-Post.

This can further be generalised into the following three main steps: Pre-processing, Executing the simulation, Post processing.

VI. RESULTS

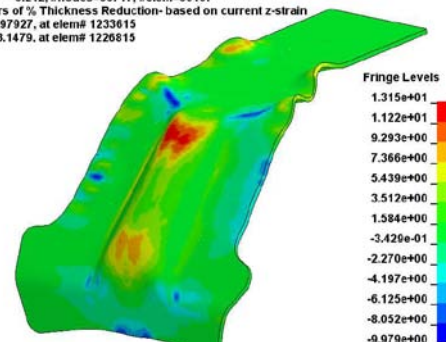
This section presents the final results for the forming simulations after spring back compensation.

forming2 / untitled
Time = 0.016981, #nodes=48236, #elem=46945
Contours of % Thickness Reduction- based on current z-strain
min=-0.672089, at elem# 1193611
max=10.5268, at elem# 1224072



(a)

forming / untitled
Time = 0.212, #nodes=68717, #elem=66137
Contours of % Thickness Reduction- based on current z-strain
min=-9.97927, at elem# 1233615
max=13.1479, at elem# 1226815



(b)

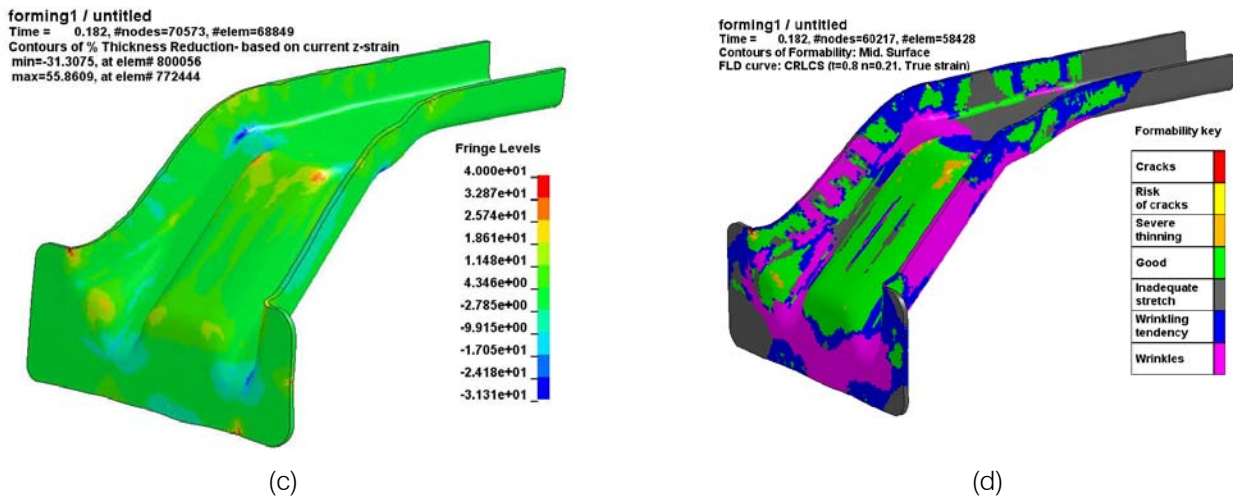


Figure 5 : (a), (b): Thickness reduction in the first (pre-forming) step and the second forming step respectively. (c): Thickness reduction after the third forming step. The only region with cracks is at the small radii on the top of the flanges at the far end of the girder, but, as mentioned before, this was not considered since a design change will be made here. (d): FLD after the third forming step

To remove the crack on the right flange, in figure (5), an additional bump was drawn in the pre-forming step. Figure (5) shows the results for the final forming simulations for all the forming steps. As can be seen in (c) and (d), the forming now looks acceptable and no cracks can be seen, except for the small radii on the top of the flanges at the far end of the girder. But, since a design change will be made here, this was left for now. What one could try to do is to draw additional bumps, in the pre-forming steps, here to avoid cracking.

After one has found a method where the forming is acceptable, i.e. does not yield cracks, too much wrinkling etc, one also has to look at the material distribution such that the formed part really has the shape requested and that the geometry is within the given tolerances. To do this, the model, after the simulations, was imported together with the CAD-model of the girder. The models were placed on top of each other to see where the formed part differed from CAD-model. The blank geometry was then slightly modified if required.

VII. CONCLUSIONS

There are some main conclusions that can be drawn from this project. Was the project successful? The main objectives were met and even though the hope was to be able to design the girder in two forming steps only, we finally reached a feasible solution. The main objective was to design the tools for forming the girder and then evaluate if investment in simulation software would be profitable. The answer to the question if this would be profitable, as can be understood from reading the discussion, depends greatly on how we choose to profligate. If we focus on forming tools and especially advanced forming tools, an investment in

simulation software would most certainly be profitable in terms of more orders, faster cheaper and safer production.

However, if we only produce a few forming tools per year, buying simulations externally, only when necessary, would maybe be a better option. But from a future perspective, one can expect the forming operations to be even more complicated and the need for simulation software will rise quickly as the demands on the products get higher.

Another important aspect that was learned during this project is that if one encounters problems during the forming with, for example, large cracks one usually has to make a drastic design change. Making small design changes usually have very small effect on the results. One can try to alter the friction coefficient or try to explain the forming problems on some other small parameter, but the difference will be negligible. If there is a severe forming problem, something drastic has to be made.

Below are some of the main conclusions from this study listed:

- LS-DYNA has shown to be good for setting up forming simulations and also to simulate spring back.
- Designing complicated forming tools without simulation software compels the design engineer to work within very narrow boundaries and only use ideas that have previously been tested.
- A simulation software is a must if any company wants to be a leading company specialized on forming tools.

VIII. ACKNOWLEDGMENT

All praises and humble thanks are to Almighty ALLAH who gave me the valour and strength for

successful completion of such a momentous work. No intellectual findings are the result of works of authors only; it includes also the precious contributions of others, which pave the path of success and immortality. Same is true for this research work.

My deep gratitude goes to my supervisor Professor Dr. TippaBhimasankara Rao for his initiation and supervision of this study and for his continuous support and engagement in the project. I believe that the effort embodies in this dissertation attained its present form only due to the vibrant and skillful leadership of him.

I would also like to express my gratitude to all those who have helped me with my work. Finally, and most importantly, I would like to express my warmest appreciation to my family and friends. Throughout this process, it has been a comforting feeling to have the support of loved ones that not only show genuine interest in my work, but who also help me to complete my work.

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