Hybrid Modeling and Identification of Dynamic Yaw Simulator

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Abstract: The objective of this work is the study and execution of a dynamic model reproducing the dynamics of a road vehicle to implement it in a vehicle simulator. To do this we will use a generic approach using Matlab to model different phenomenons related to the vehicle. In this research, we have taken into account all relevant parameters in vehicle dynamics, specifying the longitudinal and lateral forces of wheel/ground contact.

For the calculation on an operational dynamic model of the vehicle, we chose the essential compnents to reduce system complexity while ensuring a degree of realism and effectiveness of modeling, integrating a status observer by sliding mode to reconstruct or estimate in real-time, the system status.

1 INTRODUCTION

Driving simulators have been becoming little by little a suitable tool oriented to improve the knowledge about the domain of driving research. The investigations that can be conducted with this type of tool concern both the driver's behaviour, the design/control of vehicles, testing assistance systems for driving and the roadway infrastructures impact. The benefits of simulation studies are many: lack of any real risk to users, reproducible situations, time savings and reduced testing costs. In addition, their flexibility allows to test situations that do not exist in reality or at least they rarely and randomly exist.

2 OVERVIEW OF THE DRIVING SIMULATOR

Motion cueing platforms are widely employed in driving simulator which became very accessible through technological progress due to the fact that computers have become more powerful and less expensive. Thus, several simulators of various architectures were built with an aim of either human factors study, or to test new car prototypes and functionalities, or for driver training and education. Most will agree that inappropriate motion cueing is likely to induce simulator sickness through multisensory conflict. For this reason, the INRETS-MSIS (which became INRETS/LCPC LEPSIS) decided to initiate the design of a mobile platform aimed primarily at studying the importance of the modalities of yaw rendering on virtual vehicle control and on simulator sickness. The aim of the present work is contributing to improve the performance and the results of this existing simulator with the creation and insertion of a new vehicle dynamic model. Dynamic driving simulator systems allow a driver to interact safely with a synthetic urban or highway environment via a motion cueing platform that feeds back the essential inertial components (acceleration and rotation) of the vehicles movements, in order to immerse the driver partially or completely. The complexity of dynamic driving simulators lies in the fact that the system is composed of interconnected subsystems of different nature (mechatronics, control laws, computer, etc.) of which a human subject is an integral part. Dynamic driving simulators should thus be studied in their entirety, including the human driver.

2.1 Motivation of the Platforms Architecture Choice

The choices of simulator structure and motion bases are motivated by the necessity to have a sufficient perception while driving as well as by financial design constraints. Thus, the objective of the simulator project is not to reproduce all of a real vehicles motions, but only the longitudinal movements (surge),

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and yaw. This inertial feedback is to be perceived by the human user in the intended applications, which include the study of the effects of yaw cueing on simulator sickness. Indeed, one of the more nauseating manoeuvres in a driving simulator is the negotiation of (sharp) curves, especially intersections in complex environments like agglomerations (S et al., 2009)(Hichem et al., 2010).

Figure 1: Vehicle simulator.

2.2 Platform Description

Here it is a driving simulator with an acceptable compromise between rendering quality, compactness and cost limitation. The mechatronics components of the proposed solution are described below (S et al., 2009)(Hichem et al., 2010):

- The cabin consists of an instrumented mobile part moving along a guide way mounted on the platform. It is the interface between the driver and the simulated environment. The cabin is equipped with acceleration and braking pedals, steering wheel, gearbox lever and other classical car control organs. (Figure 2)

Figure 2: Cabin of the vehicle simulator.

- Audio/video generators: the visual output is provided by a system of five projectors PROJECTIONDESIGN F20 sx + with 1280x1024 resolution. (Figure 3)

Figure 3: Projector system.

- The acquisition system is composed of an industrial micro controller. This allows the control of actuators in the desired position, speed or torque (used for the steering wheel force feedback). A bidirectional information exchange protocol is defined between this card and the PCs dedicated to vehicle and traffic models. The communication is performed through CAN port between the electronic card and the PC named XPC target: this one is connected with other PCs by means of Ethernet cables. 5 computers are involved in the data elaboration and acquisition (Figure 4):

Figure 4: Aquisition system.

3 RELATED WORK

Various studies and researches have been done to edit a simplified vehicle model by adopting several assumptions with the aim to reduce its complexity in terms of Degree of Freedom (Dof) and to neglect redundant equations.

3.1 Type of Vehicle Dynamic Model

3.1.1 One Wheeled Model (Quarter of Vehicle)

For the study of suspension, the model quarter of vehicle is the simplest one. We find it in previous stydies about suspension and pneumatic model with only two degrees of freedom (NADJI, 2007).

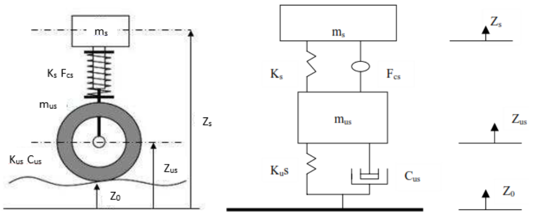

Figure 5: A quarter of vehicle.

3.1.2 Half-vehicle Model

Depending on modelings purpose, in literature it is possible to find half-vehicle models which, although far from a complete one, properly accomplish their tasks (JABALLAH, 2011)(Samuel, 2006).

Lateral Half-vehicle Model. This model represent a lateral view of the vehicle. It's used to study roll mouvements, mathematically il's a model with four degrees of freedom.

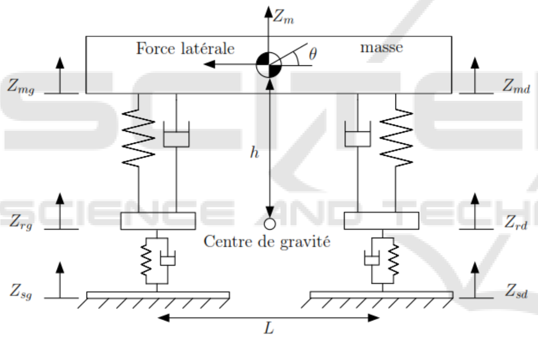

Figure 6: Lateral Half-Vehicle Model.

Longitudinal Half-vehicle Model (Bicycle Model). According to simply the model, the 4 wheels system, supposed a perfect symmetry between right and left parts, has been reduced to a bicycle model, focusing the study only to a half car. The forces applied are simply multiplied by two in order to take into account the four tires.

3.1.3 Complete Vehicle Modeling

The four wheels model is used for a complet and realistic model of the vehicle dynamics (SLEIMAN, 2010).

4 VEHICLE DYNAMIC MODEL

The proposed dynamic vehicle model is nonlinear.

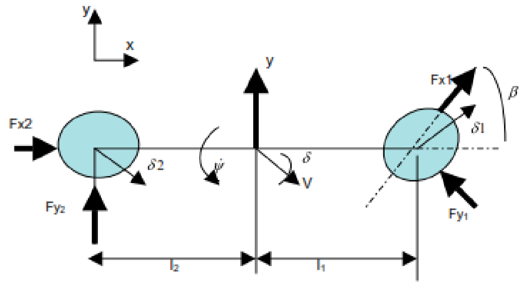

Figure 7: Longitudinal half-vehicle model.

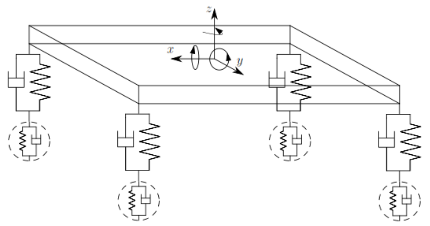

Figure 8: Complete vehicle modeling.

Moreover, the kinematic elements can greatly influence the vehicle dynamic behaviour. This is due to the existing interconnection between different parts of the vehicle. However, for the sake of simplicity, the complexity of the model may be reduced depending on the type of application and the purpose of modelling. Due to the complexity of a complete vehicle model, the vehicle model is limited to four interconnected subsystems:(VENTURE, 2003)

- The chassis.
- Suspension.
- Wheels and their interaction with the ground.
- The driver controls.

The chassis is treated as an unconstrained body in the space so it contains six DoFs; since it is consistently integrated with vehicle body, it counts three DoFs for translation along the longitudinal (Surge), lateral (Sway) and vertical (Heave) axis and three for rotations (Roll, Pitch and Yaw); they are all referred to vehicles Centre Of Gravity (COG) (Figure 9).

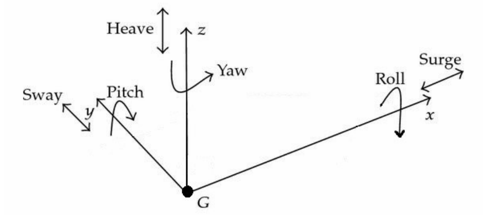

Figure 9: Vehicle degrees of freedom.

Its permissible to assume that each tire belonging to rear axis has two DoFs: one for the rotation and the other for suspension mechanism translation. Whereas the front axis is composed by two driving wheels able to steer, therefore they have one DoF more each (three DoFs). Finally the entire vehicle system owns 16 DoFs.

4.1 Chassis Dynamic Behaviour

High rigidity of the vehicle chassis can limit its flexibility study and its influence on the suspension system and the wheels system. In the present case, the chassis is considered as rigid. This rigidity helps in supporting axes with articulations of the elastic type. Therefore it can be considered as a suspended mass. The inertial parameters of the body are generally represented by:

- Its mass m (unsprung mass) and M (total mass of vehicle) ;
- Position of the centre of gravity G;
- Matrix of inertia J;

The equations of motion of the chassis are obtained by applying the fundamental principles of classical physics. This leads to three ordinary differential equations for the translational motion of the centre of gravity and three ordinary differential equations for the rotation.

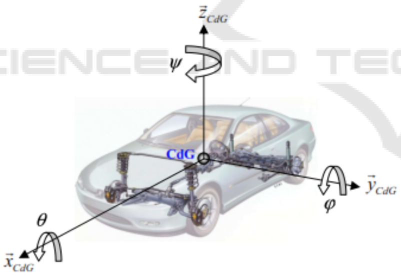

Figure 10: Translational and rotation motion of the centre of gravity.

4.1.1 Translation Motion

The sum of external forces applied to a solid body in motion, according to the principle of dynamics, is equal to its mass m multiplied by its acceleration:

$$m\dot{v}_{COG} = \sum F_{external forces} \tag{1}$$

The equilibrium of these forces along the three axes leads to the following relation (JABALLAH, 2011):

$$m\begin{bmatrix}\dot{V}_x\\ \dot{V}_y\\ \dot{V}_z\end{bmatrix} = T_r^c\begin{bmatrix}F_{xf1}+F_{xf2}+F_{xr1}+F_{xr2}+F_{wx}+F_{gx}\\ F_{yf1}+F_{yf2}+F_{yr1}+F_{yr2}+F_{wy}+F_{gy}\\ F_{zf1}+F_{zf2}+F_{zr1}+F_{zr2}+F_{wz}+F_{gz}\end{bmatrix} \tag{2}$$

Where $v=(v_x,v_y,v_z)^T$ indicates respectively longitudinal, lateral and vertical velocities.

Forces which contribute to body motion and which affect it are the following ones:

- *Contact Forces:* three for each tyre, developed in the pavement interface (for convention the subscripts f and r indicate respectively front and rear, whereas 1 and 2 indicate left and right).
- *Aerodynamic Forces:* $F_{wx} = -\frac{1}{2}C_{ax}\rho S V_x^2$
- *Gravity Forces:* $\begin{bmatrix}F_{gx}\\ F_{gy}\\ F_{gz}\end{bmatrix} = T_v^s\begin{bmatrix}0\\ 0\\ -mg\end{bmatrix}$

The vehicle system is illustrated in the Figure , where p_f and p_r represent respectively the front and rear half gauge (the sum of them provides the track width), whereas r_1 and r_2 represent the distance between the COG and front and rear axis respectively (the sum of them provides the wheelbase). m_1, m_2, m_3, m_4 depict the wheels weight, whereas m is the sprung mass (of the chassis).

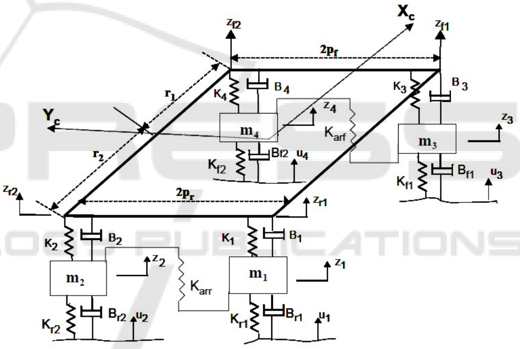

Figure 11: Vehicle With Suspension Model.

The system includes four road inputs u_1, u_2, u_3 and u provided by roadway longitudinal profile. The vertical displacements of the corners Z_{r1}, Z_{r2}, Z_{r3} and Z_{r4} depend on the angles ϕ and θ and the vertical displacement z of the sprung mass as shown in the following equations (IMINE, 2003):

$$\begin{cases} z_{r1} = z - p_{r1}\sin\theta - r_2\sin\phi \\ z_{r2} = z - p_{r2}\sin\theta + r_2\sin\phi \\ z_{f1} = z - p_{f1}\sin\theta + r_1\sin\phi \\ z_{f2} = z + p_{f2}\sin\theta + r_1\sin\phi \end{cases} \tag{3}$$

4.1.2 Rotational Motion

In literature it is common to find as the total equilibrium of rotational motion around (X_c, Y_c, Z_c) axis the following expression:

$$I\begin{bmatrix}\ddot{\theta}\\ \ddot{\phi}\\ \ddot{\psi}\end{bmatrix}=\begin{bmatrix}(F_{zf1}-F_{zf2})P_f+(F_{zr1}-F_{zr2})P_r+(k_{arr}+k_{arf})\theta\\ -(F_{zf1}+F_{zf2})r_1+(F_{zr1}+F_{zr2})r_1\\ (F_{yf1}-F_{yf2})r_1-(F_{yr1}+F_{yr2})r_2+(F_{xf2}-F_{xf1})P_f+(F_{xr2}-F_{xr1})P_r\end{bmatrix} \tag{4}$$

where $[\ddot{\theta},\ddot{\phi},\ddot{\psi}]$represents respectively the accelerations of the roll, the pitch, and the yaw.
The inertia matrix referred to R_c frame having cross elements neglected:

$$I=\begin{bmatrix}I_{xx} & 0 & 0\\ 0 & I_{yy} & 0\\ 0 & 0 & I_{zz}\end{bmatrix} \tag{5}$$

4.2 Global Model

The equations developped before by Newton-euler laws are synthesized in the state representation below:

$$\dot{\begin{bmatrix}x\\ \dot{x}\\ y\\ \dot{y}\\ z\\ \dot{z}\\ \theta\\ \dot{\theta}\\ \phi\\ \dot{\phi}\\ \psi\\ \dot{\psi}\\ z_1\\ \dot{z}_1\\ z_2\\ \dot{z}_2\\ z_3\\ \dot{z}_3\\ z_4\\ \dot{z}_4\end{bmatrix}}=\dot{\begin{bmatrix}x_1\\ x_2\\ x_3\\ x_4\\ x_5\\ x_6\\ x_7\\ x_8\\ x_9\\ x_{10}\\ x_{11}\\ x_{12}\\ x_{13}\\ x_{14}\\ x_{15}\\ x_{16}\\ x_{17}\\ x_{18}\\ x_{19}\\ x_{20}\end{bmatrix}}=\begin{bmatrix}x_2\\ \ddot{x}\\ x_4\\ \ddot{y}\\ x_6\\ \ddot{z}\\ x_8\\ \ddot{\theta}\\ x_{10}\\ \ddot{\phi}\\ x_{12}\\ \ddot{\psi}\\ x_{14}\\ \ddot{z}_1\\ x_{16}\\ \ddot{z}_2\\ x_{18}\\ \ddot{z}_3\\ x_{20}\\ \ddot{z}_4\end{bmatrix} \tag{6}$$

5 SIMULATION AND TEST

For the tests we had many simulations scenarios, in the figure below we tested a straight line followed by two turns:

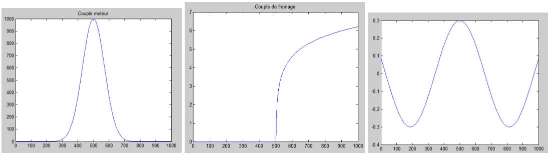

Figure 12: The simulation conditions: left engine torque, the brake in the middle and on the right the floor signal.

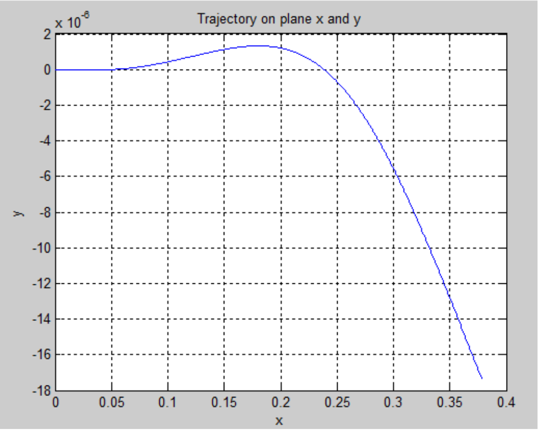

Figure 13: Result of the trajectories simulated.

We see that the model respect the instructions of the scenario, but it doesn't mean that all the vehicle dynamics are represented, because we neglected some parameters. We note also, the rotations motions don't impacts the trajectory because the undulation of the road are weak.

5.1 States Estimations

to improve the accuracy of the model, we chose to estimate the rotation angles that are the roll, pitch and yaw. Due to measurements made on an instrumented car Peugeot 406 provided by IFSTTAR, we compared these results with what we've got in our simulation model. Using a sliding mode observer we tried to estimate the equations that govern the movements of rotations.

The Steps of the Estimate

- the calculation error between the simulated and measured value

$$\begin{aligned}e_\theta&=\theta_{mesur}-\theta_{calcul}\\ e_\phi&=\phi_{mesur}-\phi_{calcul}\\ e_\psi&=\psi_{mesur}-\psi_{calcul}\end{aligned} \tag{7}$$

- Calculation of the sliding surfaces

$$S=(\frac{\partial}{\partial x}+\gamma)e \tag{8}$$

$$\begin{aligned}S_\theta&=\dot{e}_\theta+\gamma_\theta e_\theta\Rightarrow\dot{S}_\theta=\ddot{e}_\theta+\gamma_\theta\dot{e}_\theta\Rightarrow-k_\theta sign(S_\theta)\\ S_\phi&=\dot{e}_\phi+\gamma_\phi e_\phi\Rightarrow\dot{S}_\phi=\ddot{e}_\phi+\gamma_\phi\dot{e}_\phi\Rightarrow-k_\phi sign(S_\phi)\\ S_\psi&=\dot{e}_\psi+\gamma_\psi e_\psi\Rightarrow\dot{S}_\psi=\ddot{e}_\psi+\gamma_\psi\dot{e}_\psi\Rightarrow-k_\psi sign(S_\psi)\end{aligned} \tag{9}$$

- Study of stability through the theorems lyapunov
  We put:

$$\tilde{w}=\begin{pmatrix}\tilde{\theta}\\ \tilde{\phi}\\ \tilde{\psi}\end{pmatrix}=\begin{pmatrix}\theta-\hat{\theta}\\ \phi-\hat{\phi}\\ \psi-\hat{\psi}\end{pmatrix} \tag{10}$$

We choose the following Lyapunov function:

$$\begin{aligned} V(x) &= \tfrac{1}{2} S_T S + \tfrac{1}{2} T_r(\tilde{w}^T \tilde{w}) \\ \dot{V}(x) &= \dot{S}_T \dot{S} + T_r(\tilde{w}^T \dot{\tilde{w}}) \\ \dot{V}(x) &= S_T \dot{S} + T_r(\tilde{w}^T \dot{\tilde{w}}) \end{aligned} \tag{11}$$

$$\dot{V}(x) = (S_\theta \; S_\phi \; S_\psi) \begin{pmatrix} \dot{S}_\theta \\ \dot{S}_\phi \\ \dot{S}_\psi \end{pmatrix} + T_r((\tilde{\theta} \; \tilde{\phi} \; \tilde{\psi}) \begin{pmatrix} \dot{\tilde{\theta}} \\ \dot{\tilde{\phi}} \\ \dot{\tilde{\psi}} \end{pmatrix}) \tag{12}$$

- Equation resolution for obtaining the estimated variables θ, ϕ et ψ;
  Stability condition:

$$\dot{V}(x) \leq 0 \tag{13}$$

$$(S_\theta \; S_\phi \; S_\psi) \begin{pmatrix} \dot{S}_\theta \\ \dot{S}_\phi \\ \dot{S}_\psi \end{pmatrix} + T_r((\tilde{\theta} \; \tilde{\phi} \; \tilde{\psi}) \begin{pmatrix} -\dot{\hat{\theta}} \\ -\dot{\hat{\phi}} \\ -\dot{\hat{\psi}} \end{pmatrix}) \leq 0 \tag{14}$$

The problem posed is to appear $\begin{pmatrix} \hat{\theta} \\ \hat{\phi} \\ \hat{\psi} \end{pmatrix}$ in the equations that govern the accelerations, but unfortunately we managed to do it only for 2 variable from 3. That return us to think about another type of observation, or downright invites us to reconsider our equations.

6 VALIDATION

To determine the reliability of the model we have developed, we rely on *Prosper OKTAL* software used by the world's leading players in the transport sector as your: *AIRBUS, ENAC, RENAULT, PSA, DGA, VALEO, SNCF, KEOLIS, RATP, ALSTOM or BOMBARDIER*. About this tool as an expertise which refers more and more academic and scientific community.

7 CONCLUSIONS

This work deals with the topic of modeling and estimating the state of the vehicle subsystems. First, we looked at different knowledge models and dynamics of vehicle behavior in the literature. These models are then used throughout this work based on the problematic of the study.

The modeling of motor vehicles was developed to understand their dynamic behavior. Indeed, such a study has allowed us to understand the complexity of the various phenomena that interfere in this area.

As regards the aspects of vehicle dynamics; we realized that the most comprehensible way for the study of vehicle behavior is to split the various dynamics in parts. In this case, the chassis model, the aerodynamic forces, gravity, suspension and wheel.

REFERENCES

www.ifsttar.fr.

Boualem, B. (2009). *Caracterisation du comportement non lineaire en dynamique du vehicule*. PhD thesis, Universit de technologie de Belfort Montbeliard, Ecole Doctoral Sciences pour L'Ingenieur et Microtechniques.

Hamadi, N. (2009). *Modlisation d'un vehicule en prsence des forces de contact roues/sol*. PhD thesis, Universite EL-Hadj Lakhdar Batna Algerie.

Hanane, B. (2013). *Observateur non lineaire mode glissant*. PhD thesis, Universite FERHAT ABBAS Setif Algerie.

Hichem, A., Salim, H., Lamri, N., Rene J V, B., and Stephane, E. (2010). From design to experiments of a 2 dof vehicle driving simulator. *IEE*.

Imine, H. (2003). *Observation d'etats d'un vehicule pour l'estimation du profil dans les traces de roulement*. PhD thesis, Universite de Versailles Saint Quentin en Yvelines.

Jaballah, B. (2011). *Observateur robustes pour le diagnostic et la dynamique des vehicules*. PhD thesis, Universite Paul Cezanne Aix-Marseille III et l'Ecole Nationale d'Ingenieurs de Monastir.

Maakaroun, S. (2011). *Modelisation et simulation dynamique d'un vehicule urbain innovant en utilisant le formalisme de la robotique*. PhD thesis, Universite de nantes, Ecole Doctoral de l'ecole des mines de nantes.

Nadji, M. (2007). *Adequation De La Dynamique De Vehicule a La Geometrie Des Virages Routiers, Apport a La Securite Routiere*. PhD thesis, Institut National Des Sciences Appliquees Insa De Lyon.

Ouahi, M. (2011). *Observation de systemes e entrees inconnues, applications a la dynamique automobile*. PhD thesis, Universite de Limoges Ecole Doctorale ED521 Sciences et Ingenierie pour l'information.

S, E., R, B., P, S., P, P., H, A., S, B., and D, D. (2009). Simulateur de detection des alterations du comportement de conduite liees l'attention et la vigilance (vigisim).

Samuel, G.-B. (2006). *Etude d'un systeme de controle pour suspension automobile*. PhD thesis, Universite du Quebec.

Sleiman, H. (2010). *Systemes de suspension semi-active a base de fluide magnetorheologique pour l'automobile*. PhD thesis, Paris Tech, Ecole Nationale Superieure d'Arts et Metiers.

Venture, G. (2003). *Identification des parametres dynamiques d'une voiture*. PhD thesis, Universite de nantes, Ecole Doctoral de Nantes.