Modeling, Simulation and Analysis of Mechatronic Systems by Bond Graphs And Monte Carlo Simulations♠

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Abstract— Currently in a robust design of mechatronic systems one must be aware of shortcomings due to uncertainties. Taking into account uncertainties in model parameters values is a crucial point for studying robustness in modeling and in control. These may be caused by external hazardous perturbations, insufficient erroneous parameter identification, tolerances in the manufacturing process of the system, temperature, and aging, among other factors. In this work we'll determine the effects of dimensional parameters variances on a mechatronic system performance. A simple mechatronic system, modeled by bond graph approach, involving Monte Carlo simulation is used as a case study. Simulations and analysis are conducted on the 20-sim software.

Keywords: Mechatronic, Tolerancing, Bond Graph, Simulation, Monte Carlo simulation, 20-Sim.

1. INTRODUCTION

In the electro-mechanical systems which developed since about 1980, the integration of products or processes and electronics can be observed. These systems changed from systems with discrete electrical and mechanical parts to integrated electronic-mechanical systems with sensors, actuators, and digital microelectronics, these integrated systems are called mechatronic systems. A preliminary definition is given: "Mechatronic is the synergetic integration of mechanical engineering with electronics and intelligent computer control in the design and manufacturing of industrial products and processes", Bolton (2009). Modeling and simulation multi-domain systems, including interactions of physical effects from various energy domains, pose a formidable new challenges and demand new strategies and techniques for reliable solutions and performances that could not be obtained by solutions in one domain, Rzine et al. (2010). For this reason, engineers need modeling and simulation tools which allow a system analysis with respect to capabilities, capacities and behavior without really constructing the system.

The bond graph methodology is a convenient and useful complimentary tool for obtaining both the behavioral and the analysis of models. It uses a multi-energetic approach that allows the modeling of interdisciplinary models, the generation of the differential equations and the derivation of transfer functions. The use of the bond graph approach may be an alternative and a suitable tool for designing and understanding the physical phenomena in interdisciplinary systems. Mechatronic systems can be advantageously modeled and simulated using this approach.

Once the base design has been established, further optimization can be easily performed by studying the effects of component variances on the overall system performance. A robust design is one that is resistant to the effects of

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component variance due to tolerance, temperature, and aging, among other factors. Achieving a robust design involves careful analysis of the controller and plant operating together.

In this work we discuss how 20-sim software can be used to ensure the robustness of a mechatronic system design. The advantages of integrating both bond graphs and diagram block methods within 20-sim as the environment for Model-Based Design for mechatronic systems are also presented.

This paper is organized as follow: In section 2 a study of mechanical variations sources in mechatronic system is developed. Section 3 defines bond graph methodology and details the proposed method for modeling mechatronic system. Then this method is applied to a simple mechatronic system involving Monte Carlo simulations. Simulations and results are presented in section 4. Some conclusions are outlined in section 5.

2. SOURCES OF MECHANICAL VARIATIONS IN MECHATRONIC SYSTEMS

There are three main sources of variation, which must be accounted for in mechanical assemblies: Dimensional variations (lengths and angles), geometric form and feature variations (position, roundness, angularity, etc.) and Kinematic variations (small adjustments between mating parts).

Dimensional and form variations are the result of variations in the manufacturing processes or raw materials used in production. Kinematic variations occur at assembly time, whenever small adjustments between mating parts are required to accommodate dimensional or form variations.

The two-component assembly shown in Fig. 1. and Fig. 2. demonstrate the relationship between dimensional and form variations in an assembly and the small kinematic adjustments that occur at assembly time Chase et al. (1999). The parts are assembled by inserting the cylinder into the groove until it makes contact on the two sides of the groove. For each set of parts, the distance ***U*** will adjust to accommodate the current value of dimensions ***A***, ***R***, and ***θ***. The assembly resultant ***U*** represents the nominal position of the cylinder, while ***U*** + ***ΔU*** represents the position of the cylinder when the variations ***ΔA***, ***ΔR***, and ***Δθ*** are present. This adjustability of the assembly describes a kinematic constraint, or a closure constraint on the assembly.

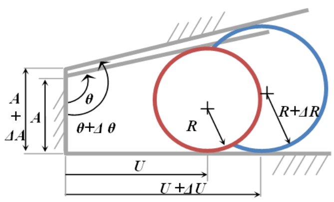

Fig. 1. Kinematic adjustments due to component dimension variations.

It is important to distinguish between component and assembly dimensions in Fig. 1. Whereas ***A***, ***R***, and ***θ*** are component dimensions, subject to random process variations, distance ***U*** is not a component dimension. It is a resultant assembly dimension. ***U*** is not a manufacturing process variable, it is a kinematic assembly variable. Variations in ***U*** can only be measured after the parts are assembled. ***A***, ***R***, and ***θ*** are the independent random sources of variation in this assembly. They are the inputs. ***U*** is a dependent assembly variable, it is the output.

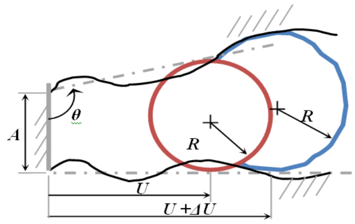

Fig. 2. Kinematic Adjustments due to geometric shape variation.

Fig. 2 illustrates the same assembly with exaggerated geometric feature variations. For production parts, the contact surfaces are not really flat and the cylinder is not perfectly round. The pattern of surface waviness will differ from one part to the next. In this assembly, the cylinder makes contact on a peak of the lower contact surface, while the next assembly may make contact in a valley. Similarly, the lower surface is in contact with a lobe of the cylinder, while the next assembly may make contact between lobes. Local surface variations such as these can propagate through an assembly and accumulate just as size variations do. Thus, in a complete assembly model all three sources of variation must be accounted for to assure realistic and accurate results.

After having an idea about the different types of mechanical variations sources, we'll show in the next section how we can taking into account component dimension variations with the bond graph methodology, This will allow us to analyze their influence on system performance.

3. SYSTEM ORIENTED MODELING IN A BOND GRAPH SENSE

The aim of the system modeler is to obtain in mathematical form a description of the dynamical behavior of a system in terms of some physically significant variables. As the nature of the system changes, the system variables change. For example, the variables commonly used in electrical systems are voltage and current, in mechanical systems force and velocity, and in fluid systems pressure and volumetric flow rate. Despite the differences in the physical variables used to characterize systems in various disciplines, certain fundamental similarities exist, and it is in the analyst's interest to seek out and exploit these similarities in such a manner that the task of modeling is eased and our overall insight into the dynamic performance of physical systems is increased.

A suitable unifying concept which can be used for this purpose is energy. Bond graphs are a graphical method which emphasizes the energetic interactions in a system by coupling components with energy bonds. The mathematical model of the system is then derived from the graph using techniques analogous to those of network analysis, Brandes et al. (2005).

The bond graph theory has been presented extensively in the literature, Paynter (1961), Karnopp et al. (1990), Vergé et al. (2003), Borutzky (2010). the essential goals of this paper is to deal with all aspects of the bond graphs in the sense of component model, and the use of the bond graph modeling for mechatronic systems. Modeling and simulation multi-domain systems, like mechatronic systems involve many different transformations and field crossings. That is a reason why bond graph is particularly convenient to represent these systems. All physical domains can be drawn on the same design, with the same schematic elements, which greatly facilitates the system's analysis.

For systems with subsystems in different energy domains, it is obvious to construct a bond graph model for each subsystem and to connect the sub-models by models of the energy transducers. For illustration, consider the slider crank mechanism driven by an electrical ***DC*** motor with constant excitation as shown in Fig. 3. The armature winding has a self-inductance L_M and a resistance R_M. The rod connecting the bar of inertia J to a load of mass m is assumed to be mass-less. The load is moving against an external disturbance force. The ***DC*** motor is controlled by means of a PID-controller.

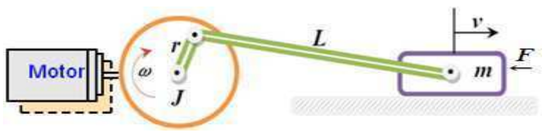

Fig. 3. Slider crank mechanism

Each sub-system constituting the slider crank mechanism can be separately modeled.

3.1 Bond graph model of electrical motor

The bond graph of the electrical subsystem is modeled by 20-sim software. The transmission including a gear element, PID controller and voltage source have been added to the DC motor as shown in Fig. 4.

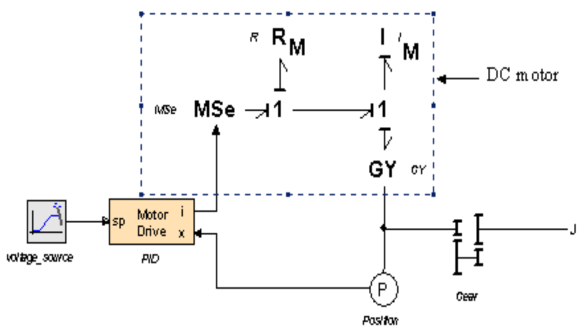

Fig. 4. Bond graph of the electric motor.

3.2 Bond graph model of crank slider mechanism

The rod establishes a geometric constraint between the angular position, φ(t) of the flywheel and the position of the piston.

$$\varphi(t) = \int_0^t \omega(\tau)\, d\tau \qquad (1)$$

Differentiation with respect to time yields a constraint between the angular velocity, ω, and the translational velocity, v, of the piston.

$$v = T(\varphi) \times \omega \qquad (2)$$

with

$$T(\varphi) = \frac{r\sin(\varphi)\left(r\cos(\varphi) + \sqrt{L^2 - r^2\sin^2(\varphi)}\right)}{\sqrt{L^2 - r^2\sin^2(\varphi)}} \qquad (3)$$

Assuming that rotational power is transformed into translational power without any losses, yields for the moment $\tilde{M}$ transformed into the force $\tilde{F}$ acting on the piston

$$\tilde{M} = T(\varphi)\, \tilde{F} \qquad (4)$$

This transformation can be represented by a displacement modulated transformer of modulus $T(\varphi)$, it was modeled by diagram blocks in 20-sim as seen in Fig. 5.

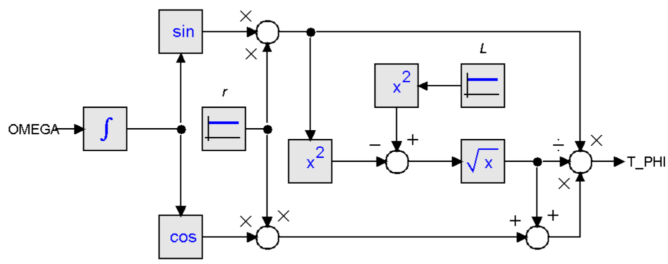

Fig. 5. $T(\varphi)$ Diagram block model.

The DC motor and $T(\varphi)$ model are introduced in bond graph of crank slider mechanism as diagram blocks presented in Fig 6.

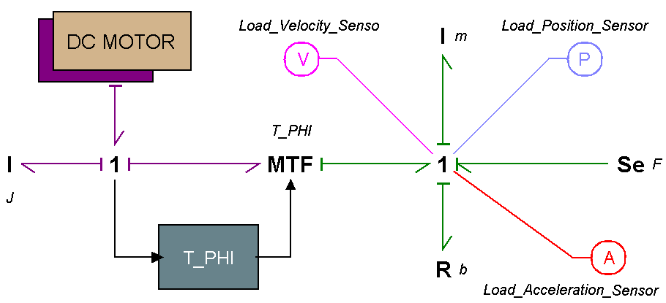

Fig. 6. Bond graph model of nominal crank slider mechanism.

As seen in Fig. 6. an R_element is introduced to model mechanical friction of the load. Tree sensors for measuring load displacement, velocity and acceleration are also introduced.

This example illustrates how modern software can help to come up with a model that has the complexity that is needed for a particular problem. Physical models, in the form of an iconic diagram or as a bond graph may help in this modeling process. Both are based on connecting elements by means of power ports. The user can select the preferred view, whether this is a bond graph, an iconic diagram with ideal physical element or a view using higher lever sub-models. They contain the same information and it is up to the user, which one of the two representations is preferred.

4. SIMULATIONS AND RESULTS

4.1 Nominal model simulation

In order to verify the validity of the proposed crank slider bond graphs model, and determine the effect of uncertainties on system performance. First, the crank slider mechanism kinematic with nominal parameters is simulated. The overall performance metric might be obtained by the measuring of the load position, velocity and acceleration as shown in Fig. 7. Nominal parameters values are dressed in table 1.

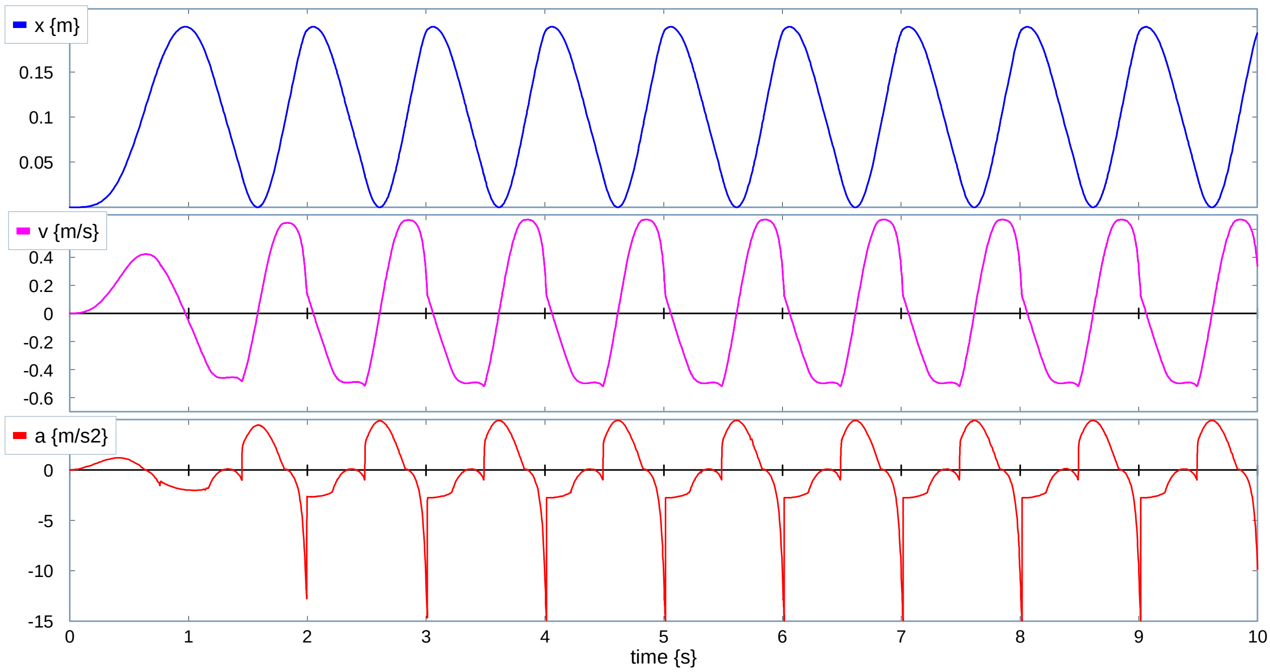

Fig. 7. Load displacement, velocity and acceleration with nominal parameters

1. Nominal parameters values

Parameters	Nominal dimensions
r (m)	0.100
L (m)	0.500
b (N.s/m)	0.1
m (Kg)	2
J(Kg.m^2)	30.10^{-6}
F(N)	10
ω(rad/s)	62.8

One of the challenges of any physical model is validating that the simulation results are accurate and represent reality. The model parameters are the degrees of freedom for adjusting the model performance. In some cases, these parameters can be populated directly from the datasheet of a component manufacturer. Many times, however, good data is not available and the model parameters must be manually adjusted. This is typically a tedious trial-and-error (adjust, simulate, repeat) process until a reasonable result is obtained.

4.2 model with variable parameters

The nominal simulation results with optimized parameters are useful for testing the controller and verifying the overall system performance. However, an optimized design does not necessarily ensure that the design is robust. A robust design, Calafiore et al. (2006), is one that is immune to component variances due to tolerances, temperature, aging, and other factors. Once the nominal performance has been validated, it is important to consider these variances and account for their effect on system performance when modeling a physical system. The 20-sim software can be used to automatically measure various aspects of the simulation result.

A common technique for analyzing the affect of parametric variances is Monte Carlo simulation, Gentle (2003).The Monte Carlo method is just one of many methods for analyzing uncertainty propagation, where the goal is to determine how random variations, lack of knowledge, or error affects the sensitivity, performance, or reliability of the system that is being modeled. Monte Carlo simulation is categorized as a sampling method because the inputs are randomly generated from probability distributions to simulate the process of sampling from an actual population. So, we try to choose a distribution for the inputs that most closely matches data we already have, or best represents our current state of knowledge. The data generated from the simulation can be represented as probability distributions (or histograms) or converted to error bars, reliability predictions, tolerance zones, and confidence intervals. Fig. 8.

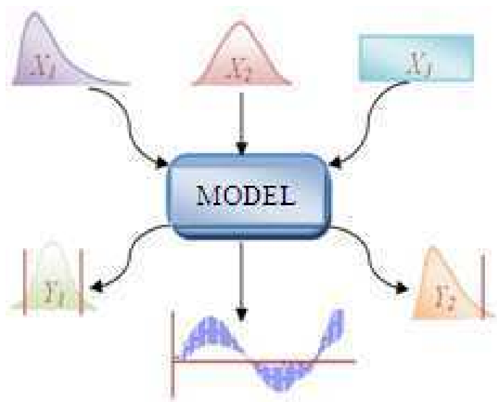

Fig. 8. Principal of stochastic uncertainty propagation.

With today's computing power and tools to distribute the simulation runs across multiple computers, a large number of runs can be simulated and the results automatically post-processed with little or no human intervention.

To demonstrate the influence of dimensional parameters variations on the operation of this mechatronic system, we consider only uncertainties on ***L*** and ***r.*** Other parameters are considered ideals.

r and ***L*** are no longer constant, and should be considered as statistically distributed variables to account for the variations of kinematic configuration. These variations can be represented by the mean $\boldsymbol{\mu}$ and tolerance ***T*** as:

$$L = \mu_L \pm T \text{ and } r = \mu_r \pm T \tag{5}$$

Since it is known that the tolerances of mechanical systems can be generally represented by a normal distribution, Choi et al. (1997), Park et al. (1996), design variables can be assumed to have a normal distribution.

Let's assume that it is desired that variables whose mean and variance are $\boldsymbol{\mu}$ and $\boldsymbol{\sigma}^2$, respectively, are within $\boldsymbol{\mu} \pm 3\,\boldsymbol{\sigma}$. As shown in Fig. 9. The probability that the variables are within this range is 99.7% in case of normal distribution.

The relation between tolerance $\boldsymbol{T}_i$ and variance $\boldsymbol{\sigma_i}$ of the i-th variable can be given as :

$$\sigma_i^2 = \frac{T_i^2}{9} \tag{6}$$

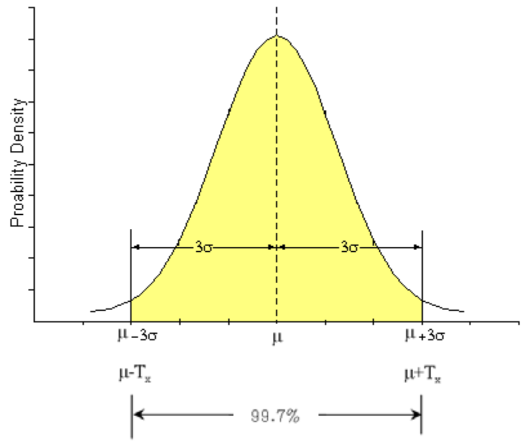

Fig. 9. Probability associated with normal distribution.

Using the variances, the tolerances of the ***r***, ***L*** components are computed, and the results are derived through Monte-Carlo method.

Tests used to assign tolerances to dimensional parameters, perform 4000 simulation runs, and collect the performance measurement data. For simplicity a normal (Gaussian) distribution with 10% tolerance was assigned to dimensional parameters as seen in Fig. 10.

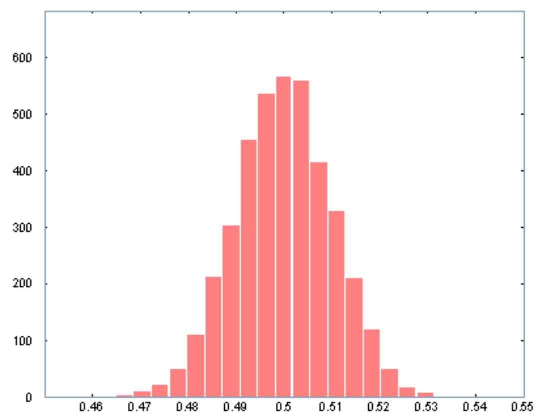

Fig. 10. Normal distribution (10% tolerance) assigned to ***L***

During the Monte Carlo simulations, the load position, velocity and acceleration measurements were applied after each run and the results were collected in a histogram plot.

The simulation results in Fig. 11 show the effect of the ***L*** variances on the load kinematics.

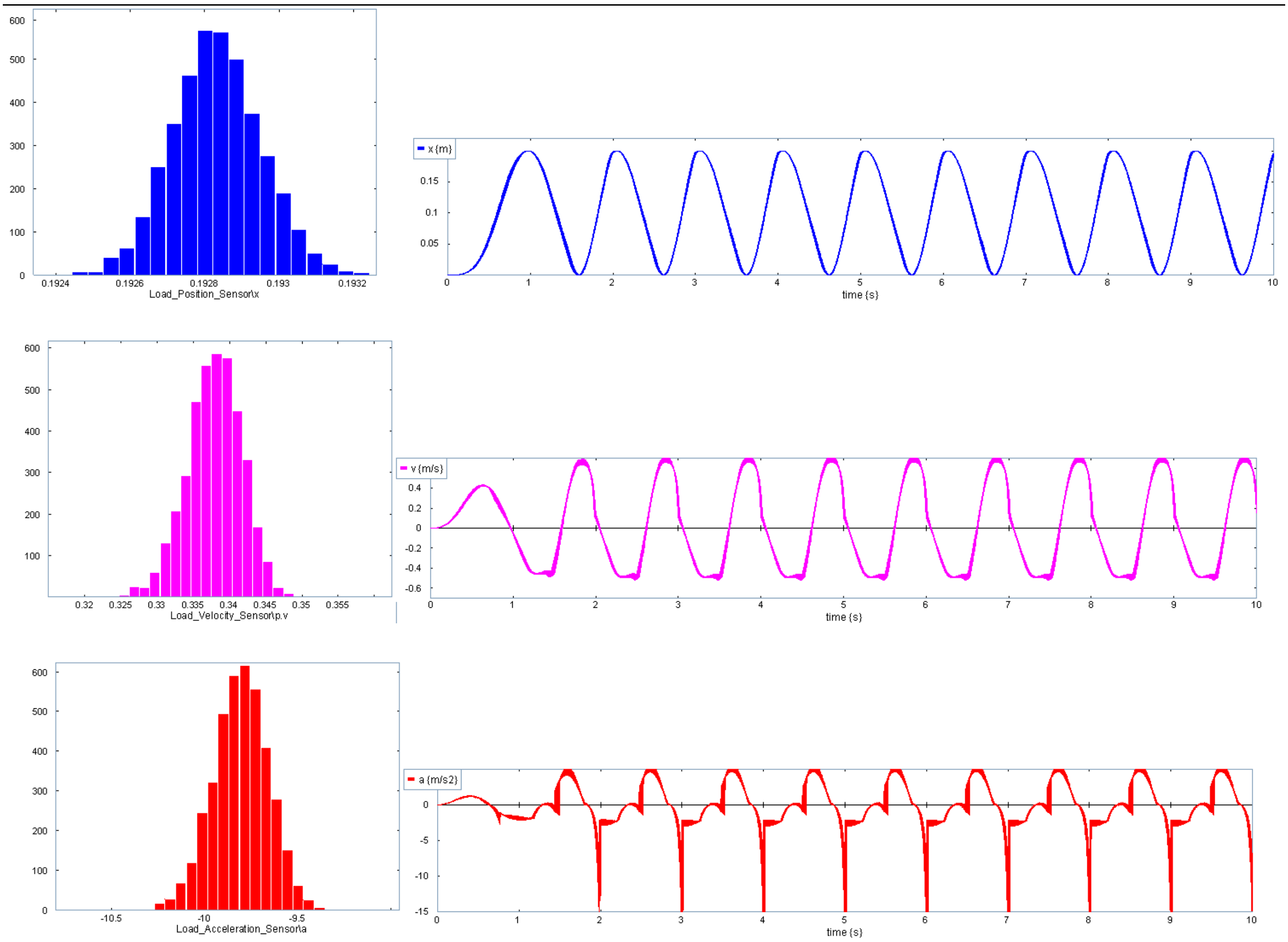

Fig. 11. Effect of L variances on the load position, velocity and acceleration during the Monte Carlo simulation

The curves in Fig 11 show that ***L*** don't have a great influence on the kinematics of the mechanism. The histograms show that the load position, velocity and acceleration variations follow a Gaussian distribution.

In order to show the effect of the ***r*** variances on the load position, velocity and acceleration, a Gaussian distribution with 10% tolerance was also assigned to ***r*** as shown in Fig. 12.

The simulation results in Fig. 13 show the effect of the ***r*** variances on the load position, velocity and acceleration. During the Monte Carlo simulations, the load position, velocity and acceleration measurements were applied after each run and the results were collected in a histogram plot.

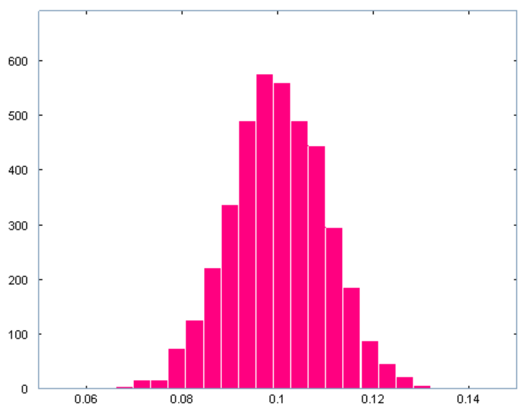

Fig. 12. Normal distribution (10% tolerance) assigned to r.

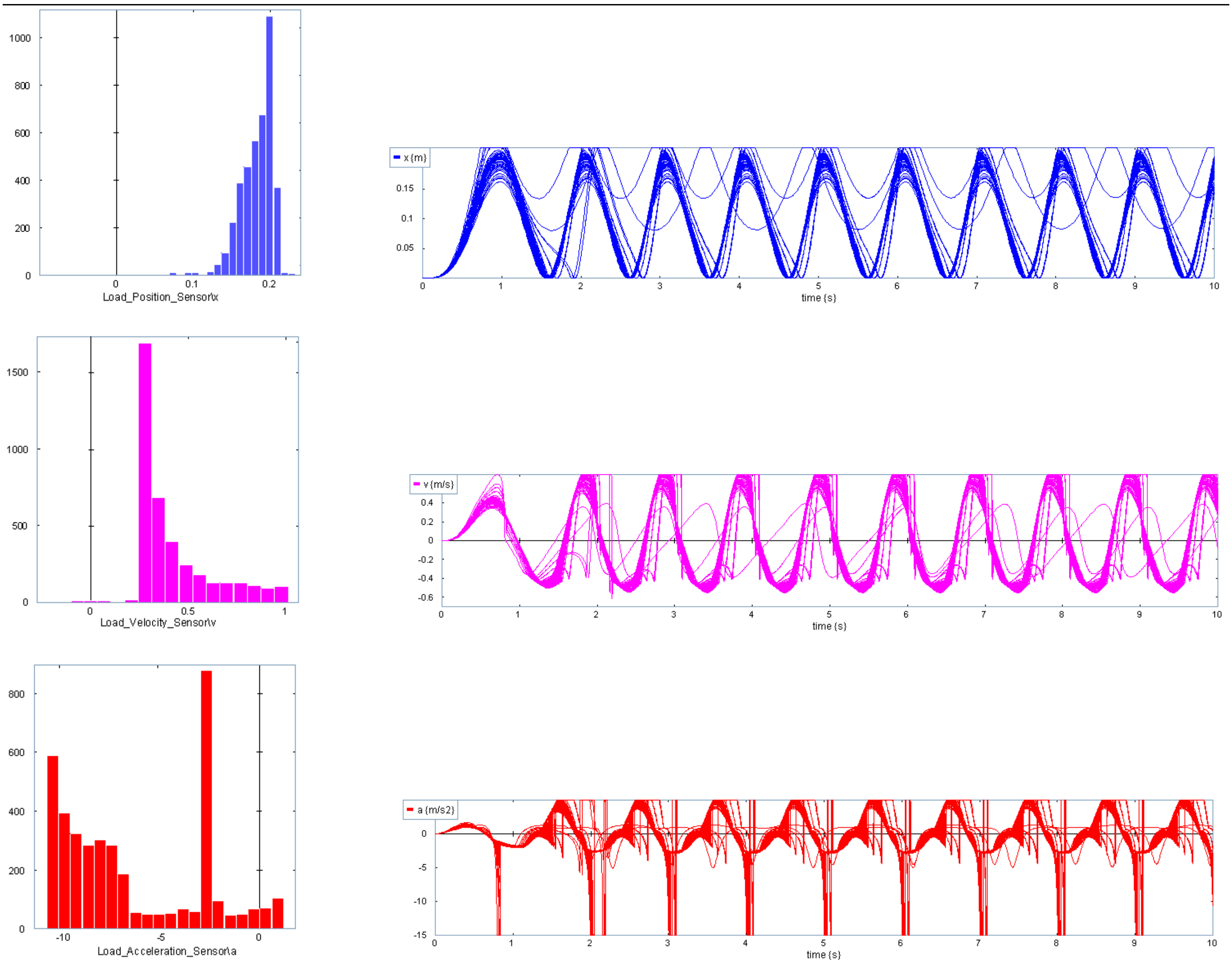

Fig. 13. Effect of *r* variances on the load position, velocity and acceleration during the Monte Carlo simulations

In order to show the effect of both ***r*** and ***L*** variances on the load position, velocity and acceleration, a normal distribution with 10% tolerance, was assigned to both ***L*** and ***r*.**

As shown in Fig. 14 and Fig. 15, new histograms representing ***L*** and ***r*** variances are generated using Monte Carlo simulations.

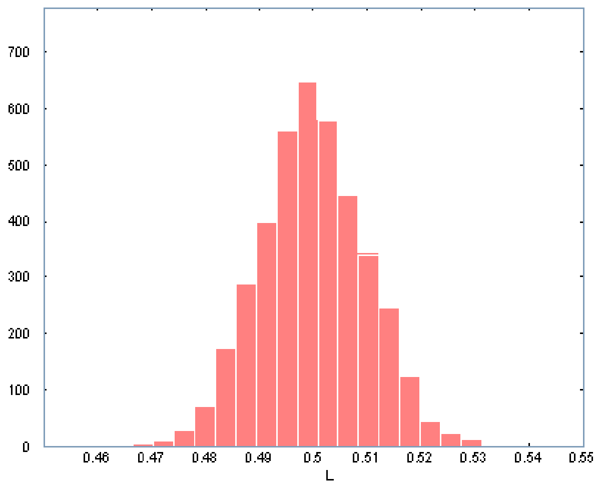

Fig. 14. Normal distribution (10% tolerance) assigned ***L*.**

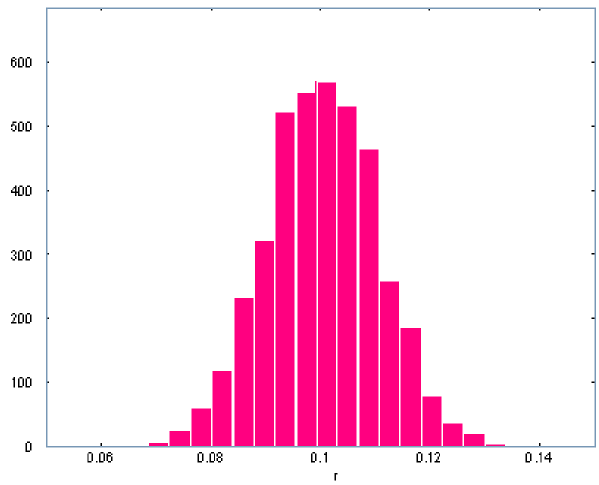

Fig. 15. Normal distribution (10% tolerance) assigned ***r***.

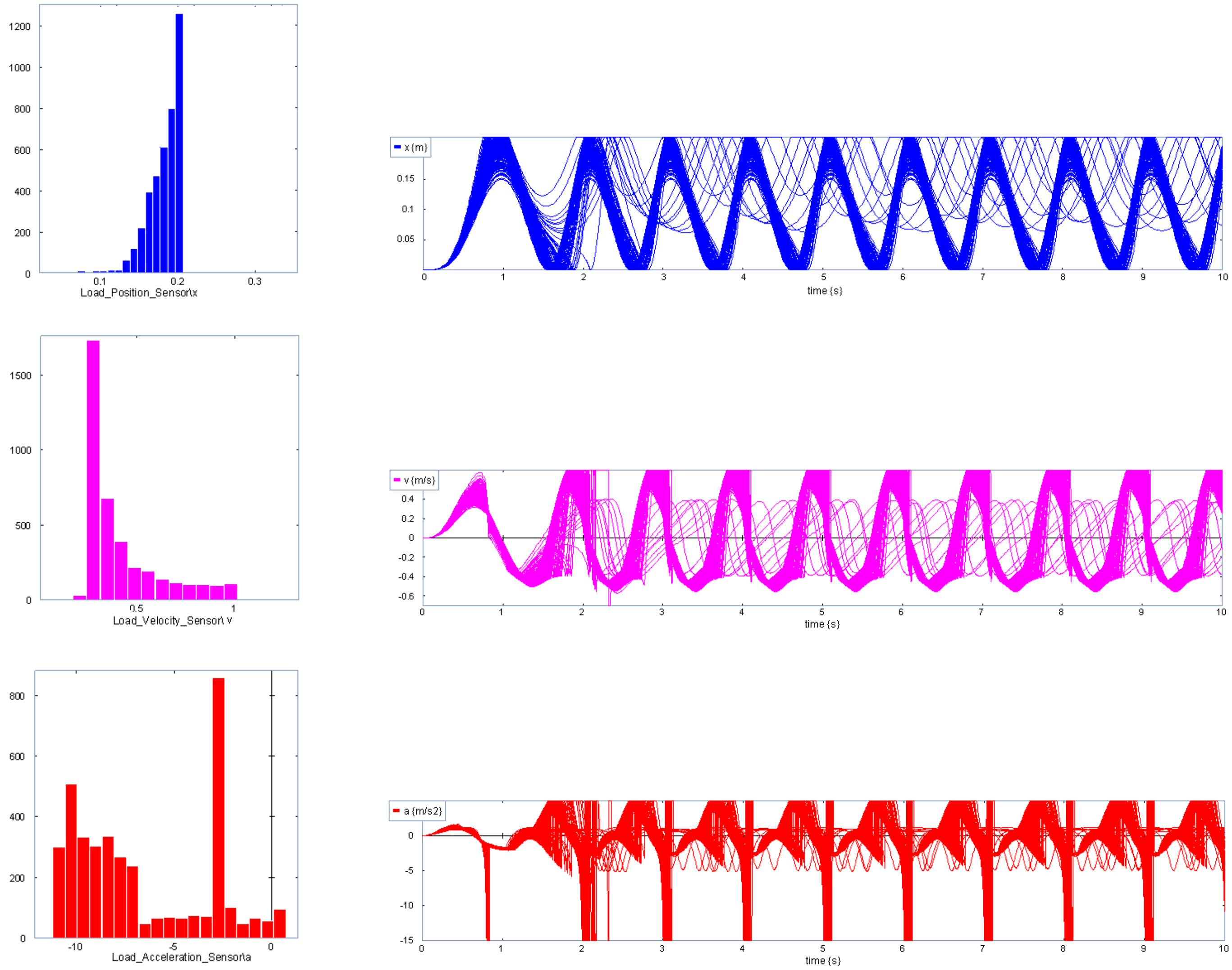

Fig. 16. Effect of ***L*** and ***r*** variances on the load position, velocity and acceleration during the Monte Carlo simulation

The curves in Fig. 13 have the same shape as those in Fig. 16. We can conclude that ***L*** don't have a great influence on the kinematics of the mechanism. ***r*** has a great influence on the kinematics of the mechanism. The load position, velocity and acceleration are very sensible to the small ***r*** variations.

The histograms provide with a statistical view of the system performance that takes into account parameter variances. From this, we arrive at a quantifiable assessment of the robustness of our design and can use the results to directly determine if the observed performance variation falls with acceptable limits based on requirements.

Requirements management tools such as 20-sim verification and validation can then use this information to automatically generate reports to communicate the results to other members of the design team.

If the performance variation is not acceptable, the next logical step is to determine the major contributors to this variation. Others plotting and visualization capability can be used to further analyze the data. The scatter plot like in excel, shown in Fig. 17, is one useful representation.

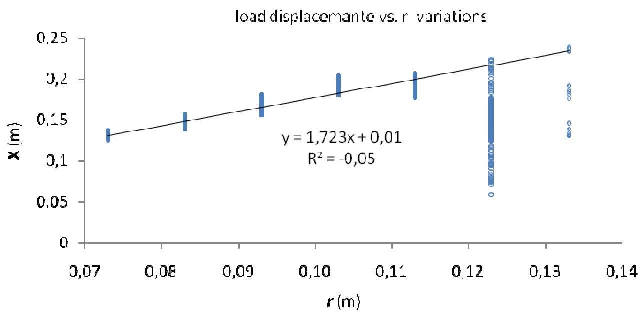

Fig. 17. Correlation scatter plot.

We have plotted the measured position, for each simulation run depending on the value of one of the parameters. The scatter plot data points reveals a visual trend of increasing position for increasing values of the parameter value. Additional measurements quantify this data into sensitivity (slope of best-fit line) and correlation coefficient (measure of deviation from the best-fit line). This information can be used to assess the allowable tolerances for a given parameter, but for a complex mechatronics design with many physical parameter variations and multiple performance measurements, a more efficient mechanism for visualizing the data is needed.

5. CONCLUSIONS

Taking advantage of the fact that a Bond graph approach is inherently suitable for tolerance analysis of mechatronic systems because a Bond graph system can consider part level tolerance variables, a procedure for performing tolerance analysis and corresponding sensitivity analysis for a mechatronic system is proposed.

First, a formulation for a Bond graph system tolerance analysis is developed for a detailed computational scheme to obtain total system tolerance for given part-level tolerances. In the process of computing system tolerance, the sensitivity of system tolerance with respect to part-level tolerances can be calculated with this formulation. Thus, this formulation enables the optimal design of system tolerance.

The kinematics of crank slider system is redefined in terms of generalized part-level tolerance variables. Variations in the geometry of a body are specified in terms of the dimensional tolerances on linkages.

To demonstrate the validity and effectiveness of the proposed tolerance analysis procedure, tolerance analysis of a crank slider mechanism is performed. For future works, the optimization of part tolerances to minimize the cost associated with maintaining system tolerance within an allowable range will be performed.

REFERENCES

Bolton, W. (2009). *Mechatronics: A Multidisciplinary Approach,* Fourth edition, Prentice Hall, England.

Borutzky, W. (2010). *Bond Graph Methodology: Development and Analysis of Multidisciplinary Dynamic System Models,* Springer-Verlag, London.

Brandes, U., & Erlebach, Th. (2005). *Network analysis: methological fondations,* Springer, Berlin.

Calafiore, G., Dabbene, F. (2006). *Probabilistic and Randomized Methods for Design under Uncertainty*. Springer Verlag London Ltd.

Chase, K.W., (1999). Multi-Dimensional Tolerance Analysis. *Dimensioning &Tolerancing Handbook,* ed. e, Jr., New York: McGraw Hill.

Choi, J. H., Lee, S. J., and Choi, D. H., (1997) Mechanical error analysis in planar linkages due to tolerances, Transactions of the KSME A 21 (4) 663-672.

Gentle, J. E., (2003). *Random number generation and Monte Carlo methods*", Springer-Verlag, 2nd edition.

Karnopp, D. C., Margolis, D., & Rosenberg, R. (1990). *System dynamics: A unified approach.* Wiley-Interscience (2nd ed.), New York.

Park, K. H., Han, H. S., and Park, T. W. (1996). A study on tolerance design of mechanisms using the Taguchi method, Journal of KSPE 13, 66-77.

Paynter, H. A., (1961). *Analysis and design of engineering systems*. MA: MIT Press Cambridge.

Rzine, B., Moujibi, N., Saka, A., Radouani, M., El fahime, B. (2010). Subsystem Interaction of Mechatronic System with Clearance. *International Journal of Systems Control,* volume (1-2010/Iss.3), pp. 122-130.

Vergé, M., and Jaume, D. (2003). *Modélisation structurée des systèmes avec les Bond Graphs,* Technip, France.

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A Simple DSP based Speed Sensorless Field Oriented Control of Induction Motor

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Abstract— In the past two decades power semiconductor device technology has seen significant advances, which have changed the world of industrial drives. With the availability of smaller and faster power semiconductor devices and also compact high speed computing processors like Digital Signal Processor (DSPs) has enabled us to implement highly computational intensive control techniques feasible inexpensively. This has led to the widespread use of AC drives using complex control techniques such as Field-Oriented Control (FOC) or direct torque control. The conventional FOC drive requires shaft-mounted speed sensors, which renders them less attractive. These sensors are usually expensive and bulky and also they increase the cost and size of the drive systems. This paper presents a simplified model of ac motor through which stator flux as well as rotor speed can be estimated to implement a sensorless drive. The proposal is based on principle of natural orientation of flux and induced emfs.

Keywords: Field Oriented control, stator flux estimation, rotor speed, slips speed, parameter sensitivity.

1. INTRODUCTION

Induction motor drives are now being used in variable speed drives, where earlier separately excited DC drives were preferred. DC drives provided fast dynamic response, independent torque and flux control. Robustness, small size, reduced cost and maintenance are some of the features in favour of induction motor drives. Variable speed drive with independent torque and flux control of induction motor is possible in Field Oriented Control (FOC) technique, which also provides excellent dynamic response (P. Vas 1998 and W. Leonhard 1992). The FOC drives require accurate shaft encoders or mechanical sensors (tachometers, position encoders) for accurate speed control operation. These sensors need additional electronics, extra wiring, and careful mounting which detracts from the inherent robustness and increase cost of a drive considerably. At lower rating drives in the range of 2 to 5 KW, the cost of a precise speed sensor is almost equal to the motor cost where as in case of drives with rating of the order of 50 KW the cost is 20 to 30% of the motor cost. There has been significant research in developing high performance AC drives that does not require a speed or position transducer for its operation. Estimators, observers and spectral analysis methods are commonly used techniques for speed estimation (J.Jiang et al., 1997). Estimators, such as those in model reference adaptive system (P.L.Jansen et al., 1995, L.C Zai et al., 1987 and Y.R.Kim et al,. 1994) depend on accurate machine model and parameter estimation. However, the induction motors are nonlinear and their parameters vary with time and operating conditions. Observers and spectral analysis method have a relatively long delay and data processing time that can limit the dynamic response of the drive, hence real-time implementation may be difficult. Rotor slot harmonics (RSH) can also be used to estimate rotor speed, as it is independent of motor parameter variation and operating conditions. Different techniques have been suggested for speed estimation (M.Arkan. 2008, R.M.Bharadwaj et al., 2004., J.M. Aller et al., 2002., K.D. Hurst et al., 1992, 1997 and A. Ferrah et al., 1998) through RSH. The main problem associated with this method is the low level of the RSH signal at low speeds which make it difficult to filter the fundamental harmonic from other harmonics arising both from the inverter and from the machine itself (K.D. Hurst et al., 1992). Speed estimation using neural network and artificial intelligence have also been developed (R.M.Bharadwaj et al., 2004. R.S.Toqeer et al., 2003 and B.K. Bose et al., 1997) and in some applications both neural networks and fuzzy logic has been used for speed estimation.

In this paper, a simplified model is developed to estimate the speed for field oriented induction motor drives. This model is dependent on the machine parameters and therefore a sensitivity analysis is provided for variation of the stator resistance and rotor resistances which usually vary during the

