TECHNIQUES FOR STUDYING A MOBILE HYDRAULIC CRANE IN VIRTUAL REALITY

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Abstract

Mobile hydraulic applications are exposed to changing environmental conditions and working processes. Furthermore, the fact that those systems consist of mechanical, fluid power and electronic control parts, make the design phase of the product to become complex. In a product development process, system configurations, components selections and parameter optimization must be accomplished in an evaluation-iteration method until fulfilling the performance specification. Replacing real physical prototypes by mathematical models and virtual prototyping in the design process is a major benefit in terms of reducing costs and time in the design phase. This paper introduces a modular method that generates dynamic models for a mobile hydraulic crane and a 3D graphical interface for visualizing of the simulation results in real-time. From the visual feedback provided by the interface, the user interacts with the course of the simulation by driving the crane model with joystick controllers. Such a tool is ideal to be utilised in virtual prototyping, since user can virtually drive and test the prototype and evaluate the system behaviour in real-time. The simulator also allows the user to instantly modify parameters and components of the model. A two degree of freedom hydraulically-driven crane is studied as an example.

Keywords: mobile crane, hydraulics-mechanics coupling, virtual reality, interface, real-time simulation, numerical solvers

1 Introduction

In mobile hydraulics applications, space and weight limitations have major influence on the fluid power design. The working process itself and the process environment are subject to changes, which makes the design phase complex and difficult. Moreover, heavy machinery production series are relatively small and unit prices are high, and therefore physical prototypes are rarely available for machine designers and researchers. On the other hand, the driving of mobile machinery involves a slow learning curve. Simulators are also of special interest as a tool for teaching driving skills to the pilots of mobile machines. It is obvious to conclude that the nature of mobile machinery makes the simulation approach especially justified.

Computer simulation has been a powerful and generally accepted practice for carrying out academic research and R&D action in the area of fluid power. Nowadays, with the available computational power, simulation models cover complete machine systems, and computer aided virtual prototyping is replacing physical prototype phases in product development.

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Most advantage of the simulation is available at the beginning of the product lifespan (Palmberg et al, 1995). In the early conceptual phase of the design process, basic design selections and comparison of different solutions can be accomplished with relatively small costs when using virtual prototyping rather than empirical testing.

Rapidly increasing computational power has created conditions for extensive numerical calculation. However, the modelling of hydraulic cranes is not widely considered on the published literature. Ellman et al (1996) propose rigid links in the modelling of a hydraulic crane. Mikkola and Handroos (1996) and Esqué et al (1999) use the finite element method (FEM) to model the structural flexibility of the links. Linjama and Virvalo (1999) use the assume modes method (AMM) to model flexibility, obtaining a low-order and highly efficient dynamic model.

An important feature of the virtual prototyping is the use of ‘man-in-the-loop’ simulation, especially in mobile applications, where the operator controls the machine. In such applications, mechanics, electronic control and hydraulics are in close interaction with each other. The design of this multi-domain system requires the dynamic properties of the system to be well-known.

The model of the hydraulic crane described in this paper is intended for studying the dynamic behaviour of a lifting mobile hydraulic crane when it is driven under different operating conditions.

The use of 3D graphics to represent simulation results has seen an increase in recent years. Traditionally, CAD programs have been used to represent geometry in a three dimensional space; however, they have been closed entities enabling interaction with the user only within the own program. Nowadays, commercial products such as IGRIP® provide an interface to their graphics engine allowing users to manipulate objects from external programs. For example, Raneda et al (2002) use the latter software package to display real time simulation results of a virtual prototype of a two degree of freedom (DOF) water hydraulic device under operator control. They made use of Matlab/Simulink® to run the simulation of the mechanical and fluid power models. The software showed good simulation times for relatively simple models but might present too heavy computational effort to run in real-time simulations for bigger systems. The simulator used in our application uses a distributed formulation and therefore it is more suitable for more complex systems.

Other applications to display 3D graphics can be custom made. Applications based on open source graphic libraries, such as OpenGL (which is a cross-platform standard for 3D rendering and 3D hardware acceleration) are easy to create due to the availability of resources to support, optimize and troubleshoot its implementation.

The need to display 3D graphics through the internet resulted in the development of VRML (Virtual Reality Modelling Language), an open standard for virtual reality on the Internet. VRML files define worlds, which can represent 3D computer-generated graphics, 3D sound, and hypermedia links. An attempt to generalize the use of VRML for scientific visualization is carried out by Mathworks®, by developing the Virtual Reality Toolbox for Matlab/Simulink®, which enables the animation of realistic three-dimensional virtual reality scenes that represent Simulink® models. For instance, some applications of the VRML have been used to display simulation results of military operations (Blais et al, 2002) and for interactivity in construction industry (Lipman and Reed, 2000)

A Virtual reality technique is used in the described application. This will allow a graphical visualization of the instantaneous simulation results and it will enable the user to interact 'man-in-the-loop' with the course of the simulation itself in real-time. The visualization will consist of a 3D graphical model of the crane and its working environment (terrain, truck, loads, obstacle, etc.).

2 Mobile Hydraulic Crane Model

The mobile hydraulic crane is the HIAB 031 model, which is designed for mid-weight tractors, with a lifting capacity of 2000 kg. The crane consists of a chain of three rigid booms which are articulated by rotational joints. Two hydraulic cylinders actuate the booms in the lifting and swinging movements (in this model the base boom is not actuated). The cylinders are commanded by electrically controlled proportional directional valves.

The main purpose of model presented in this paper is to simulate in real-time the dynamics of the crane when the crane is driven manually by an operator. The dynamic behaviour of hydraulically-driven boom is complex due to non-linear kinematics of the mechanism and non-linear properties of the fluid power components. The hydraulic crane is a multi-domain system, where the mechanical subsystem and the hydraulic subsystem operate in close interaction with each other.

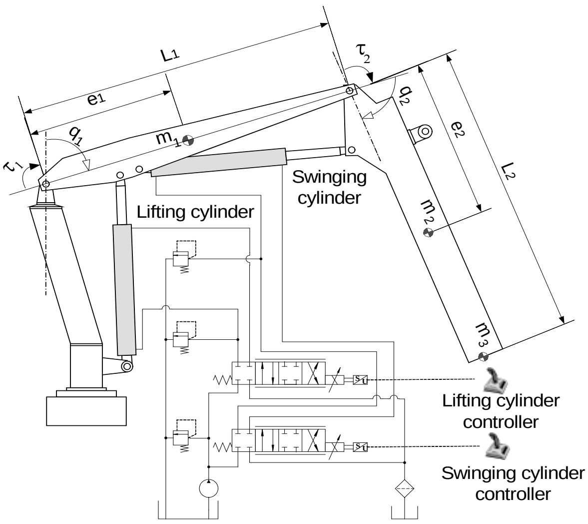

Fig. 1: *Schematic of the HIAB hydraulically-driven mobile crane*

Next subsections briefly introduce the dynamic equations from which mechanical and hydraulic models have been derived. The dynamic equations will show the coupling between mechanical and hydraulic variables, and a software implementation layout will show how this interaction is solved.

2.1 Model of the Mechanism

The model of the hydraulically-driven mechanism used in this study considers two degrees of freedom: the rotation q_1 of the lifting arm around the base arm and the rotation q_2 of the end arm versus the swinging arm. A further accurate model can be obtained when considering structural flexibility of the booms. However, the model described in this paper does not take into account additional DOF due to flexibility of the boom structure. The main reason is that the total contribution of the mechanical flexibility of the booms to the global dynamic response of the crane in negligible when compared to the contribution of the flexibility of the hydraulic system (Esqué et al, 1999). A second reason for not considering the booms' elasticity is to keep a computationally inexpensive model. However a limited flexibility (assuming only few modes) using the AMM might be considered since it would increase only slightly the mathematical formulation.

Mobile booms are usually rather simple structures consisting of a few DOF: rotational joints are used in common mobile applications and a combination of rotational and translational joints can be found in some telescopic boom applications. The dynamics of such simple articulated rigid multi-body systems can be studied easily using the Lagrangian equations of motion, which is commonly presented according to the following expression (Craig, 1989):

$$H\left(q_{\mathrm{j}}\right)\ddot{q}+C\left(\dot{q}_{\mathrm{j}},q_{\mathrm{j}}\right)+G\left(q_{\mathrm{j}}\right)=\tau\ ;\ \ j=1,...,n \tag{1}$$

Where q_{j} are the Lagrangian coordinates of the system, n is the number of DOF of the system, H represents the mass matrix of the multi-body system, C term contains the Coriolis and centrifugal forces; G is the force vector of gravitational forces and τ are the torques on the joints. According to the two degree of freedom mechanism shown in the schematic of Fig. 1, the Lagrangian coordinates q_{j} are chosen to be the joint angles q_1 and q_2.

Under the previous assignments, the angular acceleration $\ddot{q}$ of the joints can be isolated from the equation of motion (Eq. 1). In the two DOF boom mechanism of Fig. 1, the analytical expressions for H and G are given in the following set of Eq. 2 and 3. The term C that takes into account the Coriolis and centrifugal forces is neglected due to the relatively low operating angular speeds of the crane.

$$H\left(q_1,q_2\right)=\begin{pmatrix} H_{11} & H_{12} \\ H_{21} & H_{22} \end{pmatrix}\ ;\ \text{where}$$

$$\begin{aligned} H_{11} &= m_1e_1^2+m_2e_2^2+m_2L_1^2+m_3L_1^2+m_3L_2^2+ \\ &\qquad 2m_2L_1e_2\cos\left(q_2\right)+2m_3L_1L_2\cos\left(q_2\right) \end{aligned} \tag{2}$$

$$\begin{aligned} H_{12} &= m_2e_2^2+m_3L_2^2+m_2L_1e_2\cos\left(q_2\right)+m_3L_1L_2\cos\left(q_2\right) \\ H_{21} &= H_{12} \\ H_{22} &= m_2e_2^2+m_3L_2^2 \end{aligned}$$

$$G\left(q_1,q_2\right)=\begin{pmatrix} G_1 \\ G_2 \end{pmatrix}\ ;\ \text{where}$$

$$\begin{aligned} G_1 &= \left(m_2e_2g+m_3L_2g\right)\sin\left(q_1+q_2\right)+ \\ &\qquad \left(m_1e_1g+\left(m_2+m_3\right)L_2g\right)\sin\left(q_1\right) \\ G_2 &= \left(m_2e_2g+m_3L_2g\right)\sin\left(q_1+q_2\right) \end{aligned} \tag{3}$$

The vector q of Lagrangian coordinates is then obtained from time integration of $\ddot{q}$ in the next equation:

$$\ddot{q}=H^{-1}\times\left(\tau-G\right) \tag{4}$$

The usage of the Lagrangian equations of motion for describing the dynamics of the two DOF mechanism requires a minimal computational effort, since inversion of matrix H in equation of motion (Eq. 1) can be realized analytically. Larger multi-body systems would increase the dimension of the Lagrangian system of equations. Systems consisting of more than two DOF might require the utilization of matrix-decomposition methods and therefore longer computational times would be required to solve the equation of motion.

2.2 Model of the Hydraulics

The hydraulic system governing the articulated crane consists of actuators, proportional valves, pressure relief valves and a pump. Hydraulic actuators act as an interface between the mechanism and the hydraulic systems, since the pressure in the chambers of the cylinders is converted to force applied to the mechanism. Proportional valves between the actuators and the pump control the flow rate to the actuators. Pressure relief valves are security devices which limit the maximum pressure of the fluid power circuit.

The hydraulic system model is based on two main equations: the pressure generation equation and the flow resistance formula. The pressure generation equation is used to describe the pressure dynamics in fluid volumes. Fluid power components such as accumulators, actuators, pipelines and tanks consist of fluid volumes, which are modelled according to the pressure generation equation applied to a control volume.

Different pressure generation models might be obtained depending on how density of the fluid is defined when computing the effective bulk modulus. In this study a constant effective bulk modulus is used in order to keep the mathematical model computationally inexpensive. The assumption of using a constant effective bulk modulus is valid for pressures above 70 bar (Nykänen et al, 2000). For lower pressures, a more advanced model needs to be used. Nykänen et al (2000) have derived a two-phase fluid model, which defines the density of the fluid as a liquid-gas mixture. This density model takes into account the fact that at low pressure levels, volume of entrapped air can grow substantially and therefore effective bulk modulus becomes dependent on pressure values.

The second general equation used in the modelling of hydraulic systems is the flow resistance formula, which determines the flow through an orifice which connects two fluid volumes with a pressure difference. Ellman and Piché (1996) derived a two-regime flow resistance formula with smooth transition between laminar and turbulent regime flows. This refined model replaces the infinite derivative of the conventional formula and allows numerical integrators to run smoother and more accurate. Fluid power components such as servovalves, pressure and flow control valves, proportional valves, etc. can be synthesized using the flow resistance formula.

The assembly of a fluid power circuit model consists of a combination of pressure build-up volume models and flow resistance models. Flow resistance models connect volume models together and transmit a two-way interaction between them. The methodology obeys the object-oriented structuring, since modularity of the system is achieved, and separate computational modules can be implemented. Esqué and Ellman (2002) have derived pressure generation equations as systems of ordinary differential equations (ODEs) for a fixed volume, a hydraulic cylinder and a short pipeline according to the topology described. The latter models have been used in the current simulation together with a constant pressure pump and models of pressure relief valves and proportional valves, which take into account first order dynamics of their spool. These fluid power models have been validated in experimental measurements (Ellman et al, 1996).

However, more realistic models, in accordance to the components equipped in this kind of machinery, might be used when the aim of the simulation is to obtain an accurate prediction of the dynamic response of the system. Hence, a load-sensing pump model (Käppi, 2001) consisting of a variable displacement pump with a load sensing regulator would replace the current constant pressure pump model. A mobile proportional valve model (Käppi and Ellman, 1999) integrating pressure compensation, anti-saturation, pressure relief and anticavitation features might also be needed. The addition of new components to the existing model will cause an increase of the computational effort during the simulation. New fluid power components, based on the flow resistance formula, need of new pressure build-up volumes in order to obey the connection topology when assembling the system.

2.3 Coupling of the Mechanical-hydraulic System

According to the schematic shown in Fig. 2, the dynamics of both mechanism and hydraulics are coupled in the following way: the hydraulic cylinder force F that the actuators generate to the booms causes acceleration in the mechanism. Such acceleration causes a change in the angular velocity $\dot{\varphi}$ of the mechanism, which affects at the same time at the velocity $\dot{x}$ of the cylinders. The displacement of the cylinder hence causes a variation in the pressure of the cylinder chambers resulting in variation of the cylinder force.

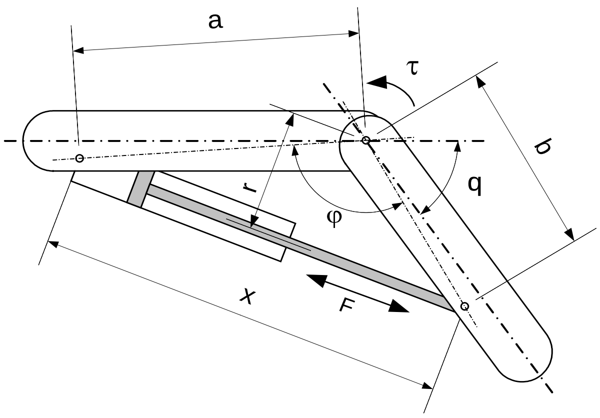

Fig. 2: *Transforming the translational cylinder movement into rotational joint movement*

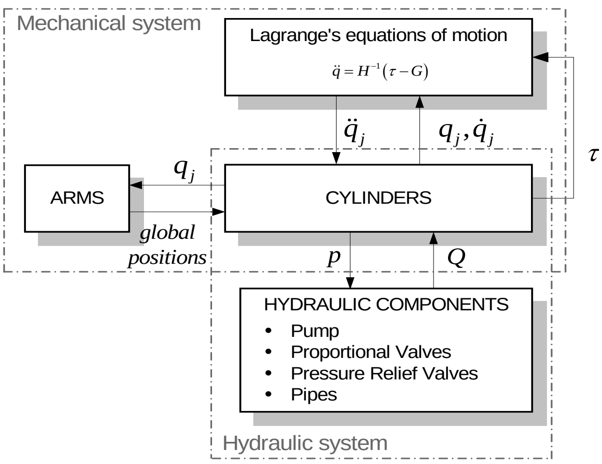

Fig. 3: *Implementation of the mechanical and fluid power circuit models in WinSimu*

The mathematical model of the hydraulically-driven crane is implemented using WinSimu (WinSimu, 2003), a fluid power simulator package developed at the Institute of Hydraulics and Automation at Tampere University of Technology. The simulator has an extensive library of fluid power components, which obeys the object-oriented methodology described in the previous subsection.

The mechanical model is divided into three functional submodels, which are showed in Fig. 3 as block diagrams:

- The cylinder model (Ellman et al, 1996) integrates the rotational acceleration $\ddot{q}$ provided by the Lagrange's equation of motion, Eq. 1, to obtain the Lagrangian coordinate q, from which the joint angle φ is directly obtained. Therefore, the cylinder stroke, velocity and accelerations are derived using Eq. 5 and its time derivatives.

$$x = \sqrt{a^2 + b^2 - 2ab\cos\varphi} \tag{5}$$

  The hydraulic cylinder model computes the flow rates (due to volume change) and chamber volumes from the cylinder stroke x and velocity $\dot{x}$ variables. Next, pressures in the chambers of the cylinders are obtained from integrations using pressure generation equation, where the input flows to the cylinder chambers are obtained from the hydraulic components block. The hydraulic force F of the cylinder is computed by balancing the forces generated by the pressures acting on both sides of the piston. However, the theoretical pressure force of the cylinder is reduced remarkably by friction forces due to the sealing. Sealing friction forces behaves highly nonlinear, and no general model methods are available for simulating purposes. A simple but still realistic dynamic model of friction was presented by Olsson (1996). In this study a yet simpler friction model is adopted in order to keep a low computational complexity. The sealing friction force is therefore modelled as a viscous friction force with constant viscous coefficient. Finally, the torque τ of the joint is obtained when multiplying the torque arm r of the cylinder by the hydraulic force F of the cylinder.
- At each integration step, new Lagrangian coordinates q_j are obtained, which cause rotations of the links of the mechanism. Therefore new global coordinates for cylinder and link joint positions have to be calculated. The submodel named arms calculates the new position of the joints in global coordinates as a function of q_j by means of a global kinematics transformation matrix.
- Lagrange's equation of motion module outputs the angular acceleration of the joints, which is computed from the mass matrix, gravity force and the joint torque according to Eq. 4.

Figure 3 shows the coupling of both mechanical and hydraulic sub-domains. Cylinders carry out the function of interface between both subsystems by means of the interaction of mechanical variables q, τ with hydraulic variables p, Q.

3 Virtual Reality Interface and Hardware Implementation

Two main tasks can be identified in the implementation of the VR interface: The simulation process of the mathematical model driven by the operator's commands and secondly the 3D visualization of the simulated results (trajectories of the booms, pressure level, etc.). In order to gain flexibility and computational power both processes are assigned to two different computers. The so-called *Simulation Computer* (SC) is dedicated exclusively to run the simulation of the dynamic model of the mobile crane described in previous section and the so-called *Visual & Communications Computer* (VCC) contains the IGRIP® software package which displays the VR representation of the graphical model. Furthermore, the VCC handles the communication between both computers and receives the operator commands generated by joysticks. Figure 4 shows a block diagram of the different tasks. First, the operator gives the input signal to the valves by means of the joystick. This signal is read by the communication interface module; the signal is then requested and read by the mobile crane model. The model is integrated one time step forward and the simulation results are outputted to the FP simulator graphical interface, which plots the received results and stores its numerical values in vector form. Parts of the simulated results sr_1 are also sent back to the communication interface, which allocates the values to a shared memory so that IGRIP® virtual environment can recall them in order to update graphically the position of the booms and the pressure levels in the cylinders. This cycle is repeated continuously until the operator stops manually the simulation. The following subsections offer a more detailed description of each process.

3.1 Simulation Computer

The SC is a regular workstation which hosts the fluid power simulator software together with mathematical model of the mobile crane. As Fig. 4 shows, the inputs of the SC are the signals to the proportional valves and its outputs are the simulated results generated by the simulator. The input signal (*signal to the valves*) is transmitted continuously from the VCC to the SC at a frequency rate of 75 Hz, which is the same rate at which the SC outputs the instantaneous simulated results sr_1 (obtained from a real-time simulation) to the VCC. The transmission of the simulated results sr_1 consists of a packet of nine values: strokes of the two cylinders, pressures in the four cylinder chambers, relative opening of the two valves, and the simulation time. The *mobile crane model* block contains the mathematical model of the hydraulic crane system. The solver uses a constant integration step size of 0.1 milliseconds (10 000 Hz), which is small enough to assure numerical stability and an acceptable accuracy of the solution. The *mobile crane model* module also sends the simulated results to the *fluid power simulator graphical interface* (WinSimu, 2003), which plots the time-history evolution of the simulated variables such as pressures, cylinder loads, strokes, end-tip position, velocity, etc. A picture of the simulator graphical interface is shown in the upper-right part of Fig. 4.

As well as plotting the instantaneous simulated results, the *FP simulator graphical interface* can also handle the parameterisation of all the submodels comprising the mathematical model of the mobile crane. Therefore, by means of a user-friendly interface, the user can set all the hydraulic and mechanical parame ters involved in the modelling of the crane such as size, lengths and friction coefficients of the cylinders,

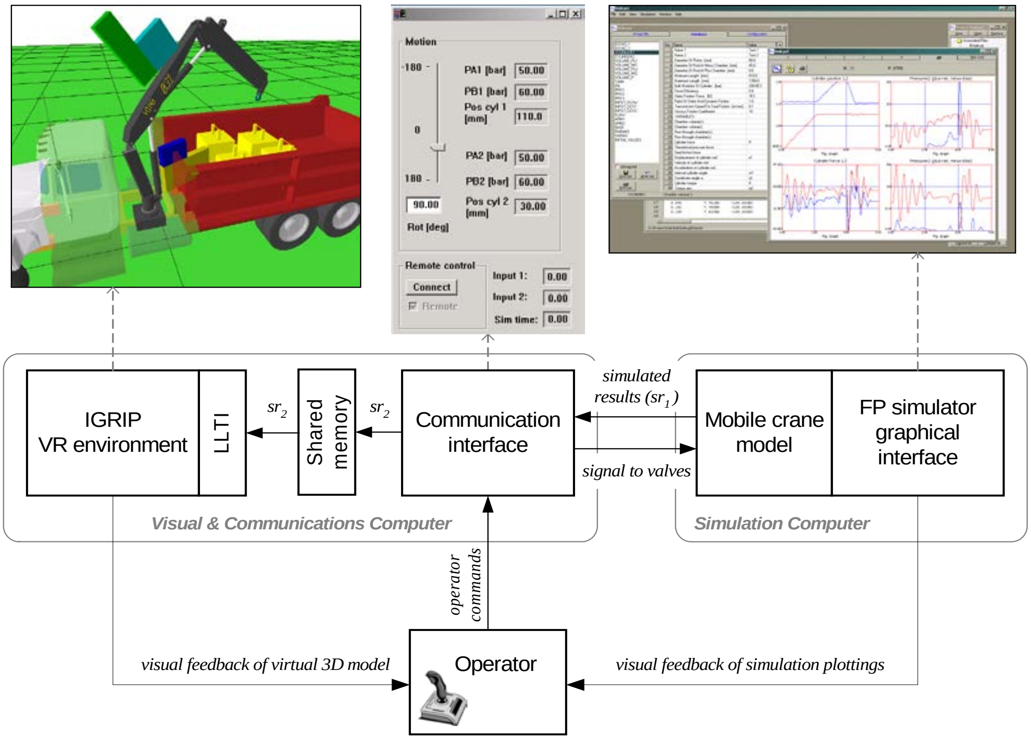

Fig. 4: *Hardware and software implementation*

characteristic curves of the pressure valves, length and size of pipes, characteristics of the pump, length and mass of the booms, position of the joints, loads, etc.

3.2 Visual & Communications Computer

The VCC consists of a visual workstation whose main function is to display a virtual 3D environment where the evolution of the simulated trajectories, pressure levels, etc. of the crane is visualized in real-time. The commercial software used to accomplish this is IGRIP®, distributed by DELMIA. IGRIP® is a modular virtual prototyping and simulation-based design software with a physics-based, scalable robotic simulation solution for modelling and off-line programming complex, multi-device robotic work cells. It incorporates motion attributes, collision detection, kinematics and I/O logic. The graphical models of the crane and the truck have been created by using common commercial CAD software. Graphic models are imported to IGRIP®, where constraints between different parts of the crane and cylinders are defined in order to build the multi-body system and to set the requested number of DOF of the system.

The inputs to graphical software are the simulated results, which are stored in the *shared memory*. As outputs the 3D graphic software displays a rendered animation of the instantaneous position of the crane (upper-left in Fig. 4). Based on the inputted simulated cylinder strokes the software computes the angles of each joint and generates the rendering of the 3D graphical model. The pressure levels of the cylinders are also animated by means of bar plots located above the cylinders' chambers.

Inputs to IGRIP® are collected from the shared library by using its Low Level Telerobotic Interface (LLTI). The LLTI environment allows directly changing the state of the virtual mobile crane from external information. LLTI is a binary packet command interface in which a stream of binary command information may be connected to a particular device in the simulation environment. The connection utilise a set of user I/O routines in the shared library. LLTI commands are processed continuously; when commands are processed, then a graphics update occurs.

The *Communication interface* module is a user-written interface in C language whose main functions are to (see Fig. 4):

- Read the joystick signals, convert the signals into relative opening values and send them to the SC.
- Read the simulated results sr_1 sent by the SC and allocates them in the shared memory as sr_2

The data acquisition frequency in the described transmissions is set to 75 Hz. With this frequency an optimal rate of joystick position lectures is achieved. Data transfer between computers is carried out through Internet by utilising the protocol TCP/IP wirelessly (Wireless LAN IEEE 802.11b standard).

The two joysticks are connected to the USB ports of the VCC, although game port or parallel port can also be used. The *Communication Interface* block reads and sends the joysticks' positions by using the routines provided by the Win32 Application Program Interface (API).

4 System Performance

There are some aspects which might affect the achievement of real-time when performing simulations:

- Data transmission rate between computers. The data transmission between different modules through the

LAN should be established and processed fast enough in order to avoid any delay in data acquisition and data transmission from and to the simulator. The 75 Hz frequency rate used in this application did not show any remarkable delay on the man-in-the-loop simulation.

- Visualization refresh rate. The graphical rendering showing the instantaneous state of the virtual system should be updated at an acceptable frequency rate according to the human eye perception. According to our tests the maximum LLTI update rate obtained can be as high as 10.000 Hz. However, a refreshing rate of 30 Hz is enough for achieving a continuous and smooth perception. The test has consisted on moving the 2-DOF crane mounted on the truck following standard operating motions. The 3D graphic models contained about 14.000 polygons. The hardware used in the test was a PC computer equipped with a PIV 1600 MHz processor, 256 MB RAM memory, and 64 MB OpenGL graphic card. By utilising a refreshing rate of 30 Hz, the graphical system has not shown any difficulty.
- Numerical integration rate. The rate (or step size) of the integration together with the type of numerical method utilised to solve the mathematical model of the system will determine the following properties: stability of the numerical method, accuracy of the solution and efficiency of the overall integration (how fast the simulation is completed). A numerical method must show good stability properties otherwise the integration process might collapse. In most of the numerical methods a suitable stability is achieved by reducing the integration time step sufficiently. This might lead to excessive long integration times which do not fulfil real-time performance. In addition, the size of the model also affects the computational time required for solving each integration step. The numerical method used in this application uses a constant integration step size due to the requirements that demand man-in-the-loop simulations. In the next subsection, extended emphasis is made to that particular problem.

4.1 Numerical Solvers

The mathematical model of the mobile crane is formulated basically by means of differential algebraic equations and ODEs arising from both Lagrange's equations of motion (Eq. 4) of the mechanism subsystem and the pressure generation equations of the hydraulic subsystem. The mathematical model is formulated obeying a modular methodology in which each fluid power or mechanical component is considered as a separate block that exchange information with adjacent modules. Therefore, each module or block is formulated and solved independently.

The Lagrangian equation of motion is integrated using a simple trapezoidal method. However the ODEs arising from pressure generation equations are generally stiff. Stiffness in differential equations means that there is a difference of orders of magnitude between time constants in the system. This system behaviour cause stability problems in the numerical integration and special numerical methods suitable for stiff systems must be employed (Hairer and Wanner, 1996).

In this application Krus' method (Krus, 1986) is being used for integrating the pressure equations. The method is a single-step two-stage second-order algorithm belonging to the family of Runge-Kutta semi-implicit methods. The method has proven to be reliable in many mobile hydraulics simulations, even in the ones that presented strong discontinuities. The algorithm has *A*-stable stability properties. *A*-stable methods are numerically stable for any positive value of time increment advancing the integration, provided that the system of ODEs is also stable. Ideally, such methods are the preferred to solve stiff equations. However, the method shows that solutions associated to very stiff components damp out too slowly. This undesirable asymptotic behaviour results in oscillatory solutions for very stiff problems. The described numerical oscillations disappear when a sufficient small time step is used. By reducing the step size for improving stability, the overall time required to run the simulation is lengthened, which is not desirable in order to achieve real-time simulations.

Another modular approach for modelling and solving such mechanical-hydraulic coupled system is to use the Lagrangian Differential Algebraic Equations (DAE) method. The Lagrangian DAE (Layton, 1998) is a systematic and unified method for mathematical modelling of lumped engineering systems which may involve coupled interactions of different physical domains (such as mechanical, electrical and hydraulic domains). Although the Lagrangian DAE method has not been used in a modular way, since modelling of the systems have always been formulated from scratch, Piché and Palmroth (2000) have developed a modular approach to the Lagrangian DAE method by coupling submodels using kinematic constraints. A drawback in using this approach is that hydraulics models are not easy to formulate according to the energy functions involved with the Lagrangian equations, and therefore implicit efforts and dynamic variables must be used. A second difficulty arises from the numerical methods for solving DAEs (Hairer and Wanner, 1996). Algorithms for ODEs are generally unsuitable for solving DAEs. The development of robust numerical algorithms for integrating DAEs is an area of active research.

Yet another approach is to keep the ODE formulation and use an *L*-stable numerical integration method. *L*-stable methods are *A*-stable methods whose components of the response associated with the highly stiff modes are extinguished immediately. Whereas widely used *A*-stable methods require small integration step sizes in order to avoid numerical oscillations in stiff systems, *L*-stable methods do not show those numerical oscillations and hence the size of the integration step does not need to be reduced to improve the stability. The size of the integration step is therefore adjusted exclusively to meet the required accuracy of the solution. Esqué et al (2002) have presented an *L*-stable method of the family of Rosenbrock methods (Rosenbrock, 1963; Wolfbrandt, 1977) for integrating distributed systems of non-linear ODEs. The algorithm is implemented with an estimator of the local truncation error and a predictor of the step size. As a consequence, sufficient small time steps are taken only when fast transitions occur during the simulation, while smooth solutions can be integrated with larger step sizes. Despite the new algorithm is still being developed. They have tested the new formula by simulating some *individual* fluid power components formulated by Esqué and Ellman (2002). As far as this, the method has been shown to have important advantages in terms of stability and efficiency when solving systems of ODEs of individual fluid power components When integrating a stiff system consisting of a loaded hydraulic cylinder Esqué et al (2002) show that while a constant step Runge-Kutta formula needs a time step size of 0.1 milliseconds in order to make the numerical oscillations vanish, the new Rosenbrock formula just needs an average step size of 76 milliseconds for performing the same simulation with tolerance of 1% in terms of accuracy of the solution. Since the computational complexity of the Runge-Kutta method and the Rosenbrock method are similar, we can confirm that for that particular simulation, the Rosenbrock method ran the simulation about the order of 1000 times faster. So far, Rosenbrock methods have not been implemented in fluid power simulator packages. Despite the promising preliminary results obtained with the Rosenbrock method, the algorithm is still in the phase of development and current research of the author is focused on expanding the usability of the *L*-stable Rosenbrock formula for integrating fluid power system models formulated according to a distributed approach.

4.2 Simulation Tests

The performance of the simulation of the mobile hydraulic crane model is analysed in the following section. Previously, the dynamic model of the mobile crane is validated by comparing simulated results and measurements, which are plotted in Fig. 5 (Ellman et al, 1996). Simulated and measured displacements and pressures of cylinders show an acceptable coincidence.

The main dimensions and masses of the crane and the load are shown in Table 1. The course of the simulation is driven man-in-the-loop by the input signals, commanded by the operator, to the proportional valves. The signals consist of ramp functions. The numerical solver used in these simulations is the previously described two-stage second-order Runge-Kutta algorithm with a constant step size h of 0.1 ms.

The effect of the size of the integration step h on the accuracy and the computational time of the simulation has been evaluated. Computational time has also been measured for different DOF models. Table 2 shows the DOF and h used for each simulation, as well as the computational time and accuracy error of the solution obtained.

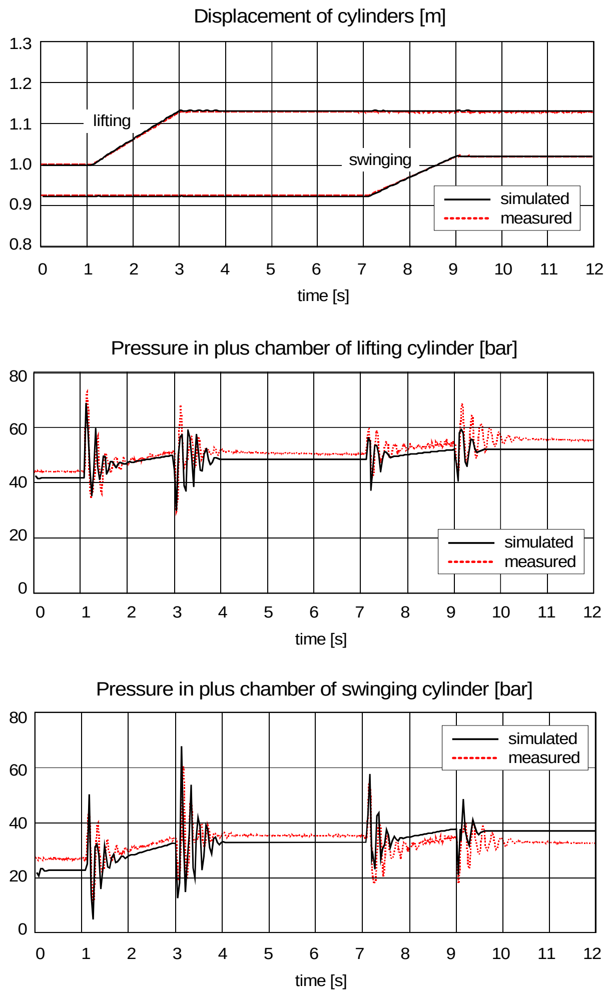

Fig. 5: *Validation of the model*

If the smallest integration step size (h = 0.1 ms.) is used, the simulation runs closely to real-time. For a second-order integration method, the estimated local truncation error of the simulated pressures when h = 0.1 ms is about the order of 0.2 Pa, which can be approximated as the exact solution. When changing the step size to h = 1 ms, the computational time is reduced drastically to 0.13 s. Changing the integration time step to h = 1 ms also involves greater truncation errors in the numerical integration and the theoretical local truncation error in the simulated results is of the order of 20 Pa, which is still acceptable. However, simulation experiments using h = 1 ms showed errors of 0.28 MPa (4.8% of relative error) in the parts of the simulation process where stiff conditions were present. As stated previously, those numerical artefacts are predictable when integrating stiff systems with a time step size not sufficiently small.

When the number of DOF of the system is changed from two to one, fluid power components associated to the swinging cylinder are not part of the model anymore. From table 2, comparison of computational times between 1-DOF and 2-DOF systems do not show significant differences. The 1-DOF model runs between 15-25% faster than the 2-DOF model.

Table 1: Basic dimensions and masses of the crane

L_1 [m]	L_2 [m]	m_1 [kg]	m_2 [kg]	m_3 [kg]	Cylinders [mm]
1.6	1.65	80	100	200	80/45-1350

Table 2: Performance results

DOF	Integ. time step [ms]	CPU time* [s] (% below real-time)		Max. relative error observed
2	0.1	1 .0	(0 %)	10^{-3} %
2	1.0	0.13	(87 %)	4.8 %
1	0.1	0.85	(15 %)	10^{-3} %
1	1.0	0.10	(90 %)	4.8 %

**CPU time measured in a PIII 700 MHz Mobile processor*

We can conclude that in order to run the simulation successfully (i.e. achieving real-time simulation, an acceptable accuracy, and good stability performance) a proper size of the integration step must be chosen. According to the simulation tests presented, the maximum size of the integration step is imposed by a) stability problems (or numerical artefacts) that increase locally the error of the solution and b) the computational time required to run the simulation. In the study presented, both conditions a) and b) require that the maximum value of the integration step should be of the order of h_{max} = 0.1 ms.

5 Conclusions

A virtual reality interface for a real-time simulation of a two degree of freedom mobile hydraulically driven crane has been presented. A simple model for the hydraulic system has been used, while the booms of the crane have been considered as rigid bodies. The solution to the coupling between hydraulic and mechanical system has been solved by using an object-oriented modelling methodology. The modular approach of modelling components as systems of ODEs simplify the system assembly, the reusability and hierarchically of the components and allows the solver to use any of the several well-known ODE numerical integrators available in the literature.

It has been shown that the actual bottleneck of the real-time simulation of stiff systems with a VR interface comes from a) the computational time required to solve the mathematical model of the fluid power circuit and b) stability of numerical integration methods. The continuous development of fastest microprocessors might help to speed computational times. However, it has been discussed that the computational time can be significantly reduced when implementing more efficient and robust numerical integration algorithms. The *L*-stable Rosenbrock formula can be categorized as a good candidate for being used in real-time simulation applications. The good stability properties and efficient formulation of the method would allow the achievement of real-time simulation of more complex systems compared to the one investigated in this study.

Nomenclature

a, b	distance between boom and cylinder joints	[m]
C	vector of Coriolis and centrifugal forces	[N]
e_1	position of centre mass of lifting arm	[m]
e_2	position of centre mass of end arm	[m]
F	cylinder acting force	[N]
G	vector of gravitational forces	[N]
g	gravitational acceleration	[m/s^2]
H	mass matrix	[kg]
h	integration step size	[s]
L_1	length of lifting arm	[m]
L_2	length of swinging arm	[m]
m_1	mass of lifting arm	[kg]
m_2	mass of swinging arm	[kg]
m_3	mass of the load	[kg]
p	pressure	[Pa]
Q	volumetric flow rate	[m^3/s]
q_j	Lagrangian coordinates	[rad]
x	cylinder stroke	[m]
T_j	joint torques	[Nm]
φ	relative angular position of the booms	[rad]

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