Dynamic Simulation of an Injection Molding Machine.

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Abstract:
Nowadays, products are often characterized by a strong interaction of different technologies, e.g. mechanics, electronics, or hydraulics. In order to analyze such mechatronic systems, it is necessary to implement specialized simulation tools. The present paper focuses on the dynamic simulation of the clamp unit of an injection molding machine with standard commercial simulation programs. The mechanical structure is modeled in the multi-body simulation software ADAMS where the flexibility of parts can be included by importing finite-element models generated with ANSYS. DSHplus is used to simulate the hydraulic system and MATLAB/Simulink to describe the controls system. By combining these tools, the interactions between the different components can be investigated. This gives a better understanding of the machine and helps in working out improvements.

1. Introduction
The analysis of products integrating different technologies, e.g. mechanical, hydraulic and controls systems, becomes more and more feasible with the constant development of simulation software and more performing computer hardware. The combination of specialized software packages is possible and allows the simulation of so-called mechatronic systems. If in the past such tools were mainly used in the aeronautic and automobile industries, they now find their way into more common engineering applications. In this case, the dynamic characteristics of the clamp unit of an injection molding machine from HUSKY is investigated. For this purpose, the finite-element (FE) program ANSYS, the multi-body simulation (MBS) software ADAMS, the fluid power simulation software DSHplus and the controls design tool MATLAB/Simulink are used. Combining these different simulation tools and applying them to the clamp unit, we can analyze and understand the dynamic behavior of the machine and the interaction between the different sub-systems. This is necessary to improve the performances, like reducing wear, cycle time or noise, or avoiding premature failure of parts.
The clamp unit is the mechanism that closes and opens the mold and keeps it effectively closed during the injection and the holding-pressure stages. The HUSKY QUADLOC™ clamp is a two-platen hydraulic clamping system. The main components are the moving platen, the clamp base and the stationary platen with the four tie bars. The platen locking and the clamping force are realized with the clamp pistons, which are integrated in the moving platen. The clamp pistons can be rotated by 45° to engage the tie bar teeth in order to lock the moving platen in its position. The main functions of the clamp unit are actuated hydraulically: hydraulic pressure is applied to the clamp pistons to generate the necessary clamping force and two hydraulic cylinders are used for the displacement of the moving platen. [1]

Figure 1: HUSKY QUADLOC™ clamp unit

The analysis focuses on two aspects: first, the quantification of the forces acting on the pads between the clamp base and the foundation in order to foresee and prevent any creep of the machine during operation, and second, the optimization of the stroke command signal in order to reduce the overall cycle time. Therefore the simulation model is limited to the moving platen stroke.

2. Simulation Models

The mechanical, hydraulic and controls systems are modeled in ADAMS, DSHplus and MATLAB/Simulink, respectively. ADAMS gives the possibility to include non-standard phenomena by linking user-written FORTRAN or C subroutines in the model. DSHplus uses this feature to make a co-simulation between both programs possible. Besides, ADAMS disposes of a plug-in that allows the user to connect the MBS model with Simulink. Thus it is possible to link the three simulation tools and simulate the complete machine. There are two possibilities for the computation: first, each program integrates its own set of differential equations and exchanges the necessary parameters with the other ones, second, the three
models are completely integrated and only the Simulink solver integrates the differential equations set.

Figure 2: co-simulation of moving platen stroke

Additionally, the flexibility of mechanical parts can be included in ADAMS. We use ANSYS to generate the necessary FE models and to reduce these large models to a few degrees of freedom before being integrated into ADAMS. The reduction method is based on the component mode synthesis technique introduced by Craig and Bampton.

**Mechanical Model**

The rigid-body model of the clamp unit is very simple and consists only of two parts: the moving platen and the stationary platen with clamp base and tie bars (refer to figure 4). The clamp base is fixed with linear spring-damper elements to the ground and the sliding of the moving platen is modeled with contact statements. Basically, the contact statement is a nonlinear spring-damper element where the force is proportional to the penetration depth $x$ and the penetration velocity $\dot{x}$:

$$
F = \begin{cases} 
  k \cdot x^e + d \cdot \dot{x} & x < 0 \\
  0 & x \geq 0
\end{cases}
$$

(1)

$k$ and $d$ are the contact stiffness and damping coefficients, respectively and $e$ is an exponent that for numerical reasons should be chosen greater than 1. If there is no penetration, then no force is applied, otherwise the location of contact, the normals at the points of contact and the force acting between both parts are computed. The statement also includes a Coulomb friction model. The transition from the static to the dynamic friction coefficient is based on the relative velocity of the two colliding geometries. Finally, the two stroke cylinder forces and a contact statement are defined between both platens.
In order to have a more accurate mechanical model, also in view of a more realistic
distribution of the forces on the clamp-base pads, the different parts are included as flexible
bodies. These flexible bodies are derived from FE models that are reduced before being
imported. The reduction method implemented in ADAMS is based on the component mode
synthesis (CMS) technique, i.e. the deformation is written as a linear combination of mode
shapes. In ADAMS, constraint modes and fixed-boundary normal modes are used to
generate the component-mode matrix $\Phi$. This approach is known as the Craig-Bampton
method. In the following, the basic principles and steps of integrating flexible bodies in
ADAMS are shown; a more detailed theoretical presentation can be found in [2].

First of all, a FE model is created that is detailed enough to correctly represent the mode
shapes of interest. The user has then to choose the nodes that serve as interface to the MBS
model. The kinematic constraints or forces are applied to these boundary nodes; the
remaining nodes are referred to as interior nodes.

By fixing the degrees of freedom (DOF) $u_b$ of the boundary nodes and solving an
eigenproblem, we get the fixed-boundary normal modes $\Phi_N$. This normal mode set is usually
trunnected. The constraint modes $\Phi_C$ are defined as the static deformation of the structure
when a unit displacement is applied to one DOF of a boundary node while the remaining
DOF of the boundary nodes are restrained (Guyan reduction). Finally the component mode
reduction matrix $\Phi$ is defined by the normal modes set $\Phi_N$ and the constraint modes set $\Phi_C$.

The relationship between the physical displacement coordinates $u$ and the component
generalized coordinates $q$ is

$$ u = [u_b, u_i] = \begin{bmatrix} 1 & 0 \\ \Phi_C & \Phi_N \end{bmatrix} [q_c, q_n] = \Phi q $$

With equation (2) the generally large number of physical DOF $u$ is drastically reduced to few
mixed physical and modal DOF $q$.

However, the Craig-Bampton modal basis $q$ has certain disadvantages that make it
sometimes difficult to directly use it in a multi-body simulation. The set of constraint modes
contains the 6 rigid-body DOF that must be replaced by the large displacement DOF of the
local body reference frame in ADAMS. They have to be removed and therefore the
component-mode matrix is transformed by solving the eigenproblem

$$ Kq = \lambda Mq $$

where $K$ and $M$ are the reduced stiffness and mass matrix, respectively. The manipulation
results in a modal basis where $q = N\hat{q}$; $N$ containing the eigenvectors from equation (3).

$$ u = \Phi q = \Phi N\hat{q} = \hat{\Phi} \hat{q} $$
The last step is a purely mathematical approach and does not further reduce the number of DOF. The new modal basis \( \dot{\mathbf{q}} \) has no direct physical meaning anymore but addresses the problems mentioned above.

Table 1: First 13 eigenfrequencies of the moving platen

<table>
<thead>
<tr>
<th>full FE model</th>
<th>modal basis ( \dot{\mathbf{q}} )</th>
<th>comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>60543 DOF</td>
<td>29 DOF</td>
<td></td>
</tr>
<tr>
<td>0.0 Hz</td>
<td>0.0 Hz</td>
<td>6 rigid-body modes</td>
</tr>
<tr>
<td>113.8 Hz</td>
<td>114.6 Hz</td>
<td>0.7 %</td>
</tr>
<tr>
<td>114.9 Hz</td>
<td>116.2 Hz</td>
<td>1.1 %</td>
</tr>
<tr>
<td>157.6 Hz</td>
<td>161.9 Hz</td>
<td>2.7 %</td>
</tr>
<tr>
<td>161.8 Hz</td>
<td>165.4 Hz</td>
<td>2.2 %</td>
</tr>
<tr>
<td>200.4 Hz</td>
<td>202.6 Hz</td>
<td>1.1 %</td>
</tr>
<tr>
<td>211.0 Hz</td>
<td>211.1 Hz</td>
<td>0.1 %</td>
</tr>
<tr>
<td>284.4 Hz</td>
<td>288.8 Hz</td>
<td>1.5 %</td>
</tr>
<tr>
<td>\vdots</td>
<td>\vdots</td>
<td>\vdots</td>
</tr>
</tbody>
</table>

The motion of a flexible body is derived from the same equation as for a rigid-body, i.e. Lagrange’s equations. In order to calculate the kinetic and potential energy, the position and velocity of an arbitrary point on the flexible body is expressed with the generalized coordinates.

\[
\xi = \begin{bmatrix} x & y & z & \psi & \theta & \phi & \dot{\mathbf{q}}^\text{T} \end{bmatrix}^\text{T}
\]

Where \( x, y, z, \psi, \theta, \phi \), are the coordinates of the local reference frame attached to the flexible body and describe the six rigid-body modes. The final form of the equation of motion is

\[
\mathbf{M} \ddot{\xi} + \mathbf{M} \dot{\xi} + \frac{1}{2} \left[ \begin{bmatrix} \frac{\partial \mathbf{M}}{\partial \xi} \end{bmatrix} \right] \dot{\xi} + \mathbf{K} \xi + \frac{\partial \mathbf{V}_g}{\partial \xi} + \mathbf{D} \dot{\xi} + \mathbf{\Psi}_\lambda = \mathbf{Q}
\]

where \( \xi \) are the generalized coordinates,
- \( \mathbf{M} \) the generalized mass matrix depending on \( \xi \),
- \( \mathbf{K} \) the generalized stiffness matrix only depending on \( \dot{\mathbf{q}} \),
- \( \mathbf{V}_g \) the gravitational energy,
- \( \mathbf{D} \) the damping matrix defined using modal damping ratios \( \zeta_i \), thus \( \mathbf{D} \) is diagonal,
- \( \mathbf{\Psi} \) the kinematic constraint equations applied to the flexible body,
- \( \lambda \) the Lagrange multipliers and
- \( \mathbf{Q} \) the generalized applied forces.
Adding flexible bodies to an ADAMS model is quite straightforward. Nevertheless, there are some limitations regarding forces and joints that can be defined to them. Especially the problem of a moving force on a flexible body, i.e. moving platen sliding on clamp base, is an open issue in multi-body dynamics. However, there are "standard" workarounds which work well and which have proven their usefulness.

The technique implemented in our flexible-body model is based on the contact statement mentioned above. Basically it works as follows: for each of the selected nodes along the sliding path, a force is computed according to equation (1) that depends on the relative vertical position $y$ and velocity $\dot{y}$ of the node to the moving platen. However, the force is only activated when the node and the moving platen effectively overlap. In fact, it is weighted by a function that depends on the horizontal distance $x$ between the node and the moving platen. The force is ramped up from zero or ramped down to zero in order to guarantee a smooth application and to minimize any discontinuities. No contact points and contact normals are computed. The distance and velocity of a node relative to the moving platen are taken in the global coordinate system and the contact and friction forces are always collinear with the coordinate system unit vectors. Nevertheless, this approach gives acceptable results, as the deformation of the clamp base is very small. A Coulomb friction force is applied in the same way.

![Figure 3: moving platen sliding model](image)

The main disadvantage is that a huge number of interface nodes are needed to have any sound representation of the moving contact forces. Unfortunately, this gives a huge number of flexible-body DOF and therefore unacceptable computation times. Now, instead of defining these nodes as interface nodes, they remain interior nodes. From a purely theoretical point of view, the accuracy of the results is not guaranteed anymore when using interior nodes as interface nodes. However, choosing more normal modes can reduce the error. The comparison of a model using interior nodes for the moving platen sliding with one using
interface nodes shows a very good compliance of the results while having a much faster computation time. Therefore we used this model for the following simulations. Other methods were not tested but some of them are presented in [3] and [4].

The flexible bodies are created in ANSYS. Simplified CAD geometries of both platens were imported in the FE program and meshed automatically with tetrahedral SOLID187 elements. The clamp base was generated "manually" with SHELL63 elements and the tie bars are modeled with BEAM4 elements. Stationary platen, clamp base and tie bars were put together to one assembly and the different components connected via spring-damper elements (COMBIN14). A macro for the computation of the modified Craig-Brampton basis is available in ANSYS. It allows the user to specify the interface nodes and the number of normal modes. The resulting modal basis is written to a file that has to be imported into ADAMS.

The ANSYS macro automatically selects the six DOF of each interface node as $u_b$. However, it is not imperative to select all the six DOF. This allows us to furthermore reduce the number of static modes and thus the number of flexible-body DOF. The macro has been changed accordingly to select only the effectively required DOF $u_b$. Finally, the moving platen has 29 flexible-body DOF and the stationary platen, clamp base and tie bars assembly has 95 flexible-body DOF.

![Figure 4: ADAMS flexible-body model](image-url)
The structure of the flexible-body model is similar to the rigid-body model. It consists of two flexible bodies: the moving platen and the stationary platen, clamp base and tie bars assembly. The clamp base is fixed with linear spring-damper elements to the ground and the moving platen sliding is modeled with “customized” contact statements as explained above. The stroke cylinder forces and a contact statement are defined between both platens. Additionally, a contact statement between each tie bar and the moving platen is added to the model.

**Hydraulic Model**

Considering that the discharge $Q$ and the pressure $\Delta p$ are equivalent to a current and a voltage, respectively, then it is possible to define an analogy between a fluid power system and an electrical circuit. The concept of resistance, capacitance and inductance can be defined [5]. DSHplus uses this analogy for the dynamic simulation of fluid-power systems. The basic modeling components are resistances of flow and volumes of fluid, they are combined to create simple or more elaborated models. The formulas behind the program are based on the one-dimensional flow theory supplemented by empirical considerations. DSHplus cannot model for example the unsteady non-uniform flow through a valve; for such problems Computational Fluid Dynamics (CFD) simulations tools are necessary. Despite the simplifications, tools like DSHplus give acceptable results for the dynamic calculation of complete fluid-technical systems in common applications.

**Resistance**

\[ \Delta p = R_H \cdot Q^2 \quad \text{with} \quad R_H = \frac{\rho}{2 \cdot A^2} \]

all components responsible for a pressure loss: orifice, throttle, valve, pipe, etc

**Capacitance**

\[ \Delta p = \frac{1}{C_H} \cdot \int Q \cdot dt \quad \text{with} \quad C_H = \frac{V_0}{E_{\text{el}}} \]

compressibility of a fluid volume

**Inductance**

\[ \Delta p = L_H \cdot \frac{dQ}{dt} \quad \text{with} \quad L_H = \frac{\rho \cdot V}{A^2} \]

inertia of a fluid volume

Figure 5: electrical - hydraulic analogy

The hydraulic system for the moving platen stroke is quite complex and includes several valves, orifices and piping. In principle, a proportional valve controls the two stroke cylinders and thus the velocity and the positioning of the moving platen.

In general, the problem in such hydraulic simulations is not the generation of the model itself, but the definition of the input parameters for the different components. Most information can be taken from manufacturer catalogues. However, the given values have generally been determined experimentally at specific conditions. First simulation results showed some
unacceptable deviations from measured data. Extensive measurements on the hydraulic system were necessary to adapt the model and the input parameters. They also showed that the actual design of the manifold block\(^1\) has a significant influence on the pressure-flow characteristic of the valves and therefore on the behavior of the complete hydraulic system.

**Controls Model**

Actually, the position and the velocity of the moving platen are controlled through an open-loop system. The controller computes a reference velocity profile for the moving platen according to the input parameters defined by the operator. Based on a look-up table, the command signal for the proportional valve is directly generated from the velocity profile.

Figure 6: command signal generation

The controller logic has been translated into a MATLAB/Simulink block diagram. Of course, as the controller is open-loop, the command signal could be directly applied to the ADAMS-DSHplus model. However, this was done to check the feasibility of connecting the three simulation tools in the view of elaborating a closed-loop solution to enhance the overall cycle time.

3. Results of simulation model

We carried out regularly measurements on the mechanical and hydraulic systems during the set-up of the models. They are needed to validate the simulations and are essential to determine and to tune unknown parameters. On the following pages, the simulation results are compared to measurements. The relative correlation of the computed results with the measured data is fairly good. Nevertheless, there are differences in the absolute values for

\(^1\) A manifold block is a machined steel block on which the different valves, cartridges and orifices are mounted. This allows a compact construction of the hydraulic system by eliminating the use of subplates and piping.
some parameters. In addition, we present the possibilities and limits of the different simulation models.

Hammer-impact modal analysis measurements on the individual parts were done to verify that the FE models correctly reflect their stiffness. As for the moving platen, the tolerance for the other parts is about 4 % up to 300 Hz, except for the clamp base where it is 10 %.

Table 2: Modal analysis of moving platen

<table>
<thead>
<tr>
<th>measurements</th>
<th>full FE model</th>
<th>relative deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>117.0 Hz</td>
<td>113.8 Hz</td>
<td>-2.7 %</td>
</tr>
<tr>
<td>118.0 Hz</td>
<td>114.9 Hz</td>
<td>-2.6 %</td>
</tr>
<tr>
<td>158.0 Hz</td>
<td>157.6 Hz</td>
<td>-0.3 %</td>
</tr>
<tr>
<td>163.0 Hz</td>
<td>161.8 Hz</td>
<td>-0.7 %</td>
</tr>
<tr>
<td>205.5 Hz</td>
<td>200.4 Hz</td>
<td>-2.5 %</td>
</tr>
<tr>
<td>211.0 Hz</td>
<td>211.0 Hz</td>
<td>-0.0 %</td>
</tr>
<tr>
<td>289.5 Hz</td>
<td>284.4 Hz</td>
<td>-1.8 %</td>
</tr>
</tbody>
</table>

In a first step, simple accelerometer measurements were carried out on a clamp unit under operation to determine the frequencies of interest. The accelerometers were fixed on the two platens in the stroke direction and on the clamp base in the vertical direction. Three frequencies clearly appear in the spectrum of the time signals: 8.5 Hz, 223 Hz and 447 Hz. The two latter frequencies are due to the operating pumps and are found in the three spectrums. The 8.5 Hz corresponds to a deflection of the stationary platen and the clamp base and is only observed when the moving platen starts to move.

Figure 7: a) FFT of measured accelerometer data, b) FFT of computed accelerations
Afterwards, more exhaustive measurements were carried out in order to generate operating deflection shapes (ODS). Several sets of measurements were necessary where the vertical and horizontal accelerations at several locations along the clamp base and the stationary platen were measured in reference to the vertical and horizontal accelerations at a specific location. The acquired data was evaluated with the algorithm implemented in ME'scopeVES. The ODS gives an information on the deformation under operation of the structure at a certain frequency [6]. With the ADAMS/Linear plug-in, the MBS model is linearized and the eigenfrequencies of the complete clamp unit, with the corresponding mode shapes, can be computed. Actually, figure 8 shows that the ODS at 8.5 Hz corresponds to a mode shape of the clamp unit. At the start-up of the moving platen acceleration, the force impulse from the stroke cylinders excites this mode.

![Figure 8: a) ODS, b) ADAMS modal analysis](image)

As stated earlier, the information contained in the manufacturer catalogues proved to be incomplete and not adequate so that measurements on the hydraulic system were essential. We measured the pressure at different locations in the manifold block assembly to determine the pressure losses for several valves; the corresponding flow was derived from the moving platen velocity. These results served as input for several valves in our hydraulic model. Until now, measurements at only two operating points, i.e. for two different maximum velocities of the moving platen, are available. Additional measurements would be required to generate more accurate pressure-flow characteristics. Nevertheless, the simulation gives already reasonable results. In figure 9, the stroke command signal, the moving platen velocity and the pressures at the proportional valve during mold-close are plotted. During this process, the rod side of the stroke cylinders is connected to the pump and the bore side to the tank.
The comparison clearly shows that our simulation still lacks an accurate modelization of the pumps. Indeed, as the complete pump system is simply modeled by an ideal pressure source, the fluctuations of the system pressure due to the adjustment of the variable displacement pumps by the controller cannot be represented. The discrepancies during the deceleration phase are probably due to the approximated pressure-flow characteristics of some valves. The correlation of the model should be improved with additional measurements.
The model allows us to simulate various design parameters of the hydraulic system. The effect of different valve characteristics onto the system’s dynamic or how the system performs under different operating conditions can be studied. The same is true for the controls. We can run through several command signals for the stroke valve in order to get some benchmark data that will help for the design of an improved controller.

Figure 10: a) Modified deceleration profile, b) Modified acceleration profile

The modifications to the command signal presented above are rather trivial and should rather demonstrate the possibilities of the simulation model. In figure 10 b), the proportional valve is opened completely during acceleration in order to get the maximal possible flow. This allows reaching the set velocity earlier and shortening the overall cycle time. In figure 10 a), the acceleration profile is used for the deceleration of the moving platen. The moving platen is decelerated abruptly. Because of the fast closure of the proportional valve, the pressure rise at bore side is even so important that the moving platen moves backwards for a brief moment, which in turn is responsible that the pressure drops down to 0 bar. This leads to cavitation that is unacceptable as the hydraulic components can be damaged.

Even though the simulation model does not always give exact absolute values, it helps to make qualitative and quantitative statements to the influence of different parameters on the dynamic behavior of the complete machine.
4. Conclusion

Today the interest in an integrated dynamic analysis of mechatronic systems increases. For this purpose different approaches are possible depending on the tools available to the user. In this paper, commercial simulation software packages are used and combined. They include special interfaces that allow the set up of co-simulations.

In our case, ADAMS is the backbone of the analysis as the MBS software inherently models large non-linear motion. This is only conditionally possible with ANSYS and normally at the expense of long computation times. The use of flexible bodies generated by a FE program allows the user to account for the flexibility of components in ADAMS if necessary. The computation effort is limited as the component mode synthesis method considerably reduces the number of DOF. The FE-MBS model can be linked with DSHplus and MATLAB/Simulink to integrate hydraulics and controls and to simulate mechatronic systems.

Fig. 11: co-simulation of mechatronic system

This case study shows the potentials of an integrated analysis. It gives a better understanding of the dynamic characteristics of the overall system. The interactions between the different subsystems, generally difficult to assess, can be investigated. Furthermore, the model allows to simulate with various design parameters and to predict the effect of modifications to these parameters on the performance of the overall system. This is very helpful to work out improvements and generally faster than proceeding experimentally. Such a simulation is also a valuable tool in the early design stages. It serves as a fast method of evaluating different designs to meet the specified objectives and can reduce the need of costly prototypes.
5. References


