Modeling and simulation of web-tension mechanism in a filling machine

Master's Dissertation by
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1 Abstract

When carton packages are formed in a filling machine, there have to be a certain tension in the material package to keep it stretched, however not so much that the material is damaged. In the Tetra Pak A1 filling machine this is achieved through a web tension-mechanism consisting of a number of rollers and a pneumatic cylinder.

The aim of the master thesis is to get a simulation model for this web-tension mechanism, with the end goal of being able to predict how the mechanism will work under different pneumatic pressures and with different package shapes.

To have the needed parameters for the simulation, the pneumatic cylinder was tested in a load testing equipment, and the parameters were extracted from test results.

The simulation model was split in two parts, with one being a Dymola logical model of the pneumatic subsystem (consisting of the cylinder, a regulator and a connecting tube). This model was built using the Modelon library package Pneumatics. The other part was a Finite Element model of the rollers and the package material, which was done in Abaqus Explicit.

The two models were cosimulated and the total force at the end of the moving web was logged. The simulated force variations were compared to variations that had been measured during actual production runs with the machine that was modelled.

The ultimate goal of the project is to have a small rheological Abaqus model containing dynamics of all parts, including a simplified version of the pneumatics. This could then be connected to the models of other parts of the machine that have earlier been modelled.
2 Introduction

2.1 Tetra Pak Carton Economy

_Tetra Pak Packaging Solutions_ produces carton packages for a wide variety of markets. In some of these markets, it is of high priority that the prices of packages are small and therefore, the package production costs also have to be low. _Carton Economy_ is concerned primarily with these markets and therefore keeping down production costs is an important objective for the department.

2.2 Web Tension-mechanism

Before the carton packages are formed, the package material is kept in the form of a long, flat stretch of package material, called a _web_. The web is then run through a _filling machine_ in which the material is prepared for forming, shaped into packages, filled with its content (e.g. milk) and then closed.

During the forming, the web has to have some tension, keeping it stretched in order to avoid forming defects. However, too much tension can result in damage to the packaging material giving rise to other defects. A system is therefore used to keep the force and the web tension almost constant. This system consists of a number of rollers and a pneumatic cylinder. The web force is not actually constant, and therefore _Carton Economy_ have been attempting to make a good model for the web tension-mechanism. For the machine 0800, the system sits in a chamber as seen in the figure 1. For this system, earlier tests have been made and therefore a model of this system would be possible to compare with measurements from the real mechanism. The objective of this thesis is to model this system.
2.3 Objectives

The final goal of the project is to make a model of the web tension-mechanism in Abaqus that can be incorporated in a bigger model for simulations of the package forming processes. However, for the thesis only a separate model of the web tension-mechanism was simulated. Because the attributes of the pneumatic cylinder were not known, tests were to be performed with a load testing equipment.

The pneumatic system was modelled in Dymola since Abaqus is not suited for modelling pneumatic components. The aim is to be able to include components in Abaqus that would have dynamics being much like those of the pneumatic cylinder.

A co-simulation between the two models was run in order to see whether the model was sufficiently accurate.

2.4 Dymola

Dymola is a program designed for modelling different types of physical systems. The program is based on the object oriented language Modelica. In this language different objects are written using a few governing equations. These objects are then connected together into a full model using a graphical interface. Before the model is run, Dymola rearranges the code depending on what objects are used, so that it will be runnable. This allows the user to simply write the governing equations without worrying about which variables are known and which are unknown. It also allows the same piece of code to be used in different contexts. When the code has been rearranged it is then turned into C code for faster running.
3 System

The web tension-system is seen in figure 2. The packaging material enters the system by the Driven Roller (the rightmost roller in the figures). At the other end the package forming system is located. It is the package forming system that forms the material into packages while at the same time pulling the web through the system. The forming process gives rise to a varying web velocity, and this also changes the tension in the material. In order to minimize the force variations, the middle roller is not set in place. It is instead allowed to move like a pendulum, hence the name pendulum roller. However, it cannot move freely because the web should always be stretched. To achieve this, a pneumatic cylinder is connected to the system, applying a force to the holder. This helps keeping the pendulum roller down.

![Figure 2: CAD models of the system with the chamber removed](image)

The driven roller, the bending roller and the bar are all kept in place but allowed to rotate.

The rollers are made out of steel and they are also hollow, which keeps them light. This is however not the case with the bar.

The movement of the material over the rollers is sketched out in figure 3 found on the next page.
3.1 Pneumatic Subsystem

The pneumatic subsystem is schematically illustrated in figure 4.

The parts have the following functions:

1. The air supply is the source of the pressurized air used in the pneumatic system. The air in this system has a near constant pressure of 5.5 bar gauge pressure.

2. The air supply is connected to the regulator. On the other side of this regulator the air should have a lower pressure than at the air supply. Whenever the pressure falls below the desired value, the regulator will let air flow from the air supply and into the pneumatic cylinder. Should the pressure rise above the desired value, the regulator will let air flow from the cylinder and out into the surrounding atmosphere.

3. The tube works like any tube: its only function is to connect the regulator to the pneumatic cylinder so that air can be transported between the two.
4. The cylinder is the centrepiece of the pneumatic system since it is the component that is connected to and exerts force on the mechanical system. The right end of the piston rod is the one connected to the mechanical parts but only the left chamber is pressurized. Therefore, the cylinder will always exert a force working to the right, helping keep the pendulum roller (see figure 2) in place.

Everywhere air travels between the different parts, there are small orifices through which the air flows. The size of these determine how much air can flow between the parts.

The working (desired) pressure in the cylinder is 2 bar gauge pressure (3 bar absolute pressure) and in this thesis that will be the used pressure except when explicitly stated otherwise.

4 Theory

For all quantities presented herein, explanations can be found both where they are introduced and in Appendix 11.1 where the units can be found. A list over subscripts can be found in the same place.

4.1 Pneumatic Cylinder

The pneumatic cylinder consists of two chambers, each with their own set of equations. For each chamber, the Dymola model is based on three equations. The first one is the well-known ideal gas law:

\[ P = \rho R_{\text{spec}} T \]  

\( R_{\text{spec}} \) being the specific gas constant for the gas used (in this case being air).

The second equation is the mass conservation law:

\[ \dot{m} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \]  

where \( \dot{m}_{\text{in}} \) and \( \dot{m}_{\text{out}} \) are the mass flow in and the mass flow out.

The third equation however requires a bit more work to reach.

The mathematical model of the pneumatic cylinder is based on the laws of conservation for energy and mass. For closed systems those equations are often formulated as:

\[ \frac{d}{dt}(B) = C \]
where $B$ is the total mass $m$ or the total energy $U$. When $B$ is mass then $C = 0$ while if $B$ is energy then $C = \dot{Q} - \dot{W}$. This only holds for closed systems though so for non-closed volumes such as a chamber of the pneumatic cylinder (see figure 5), modifications have to be made.

In the case of which this thesis is concerned, the mass airflow in to and out from the chamber will be given by other components and so it is trivial to see that the change in mass is given by equation (2).

where $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass flow in and the mass flow out of the cylinder.

For the energy conversation Reynolds’s Transport Theorem can be used. This states that the conservation equation can be expressed:

$$\frac{dB}{dt} = \frac{d}{dt} \int_{CV} \rho \beta dV + \int_{CS} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS \quad (4)$$

where:

- $\beta$ is $e + \frac{v_f^2}{2}$,
- $e$ is the internal energy per mass unit,
- $\bar{v}_f$ is the fluid velocity and $v_f$ the fluid speed,
- $CV$ is the control volume and $CS$ is the surface of $CV$,
- $\bar{v}_s$ is the movement of $CS$ and $v_s$ is the speed of the movement,
- $\hat{n}$ is the unity vector normal to the surface, pointing outwards (see figure 5).

![Figure 5: Control Volume of a pneumatic chamber. Fluid can enter and leave the volume at the inflow and outflow orifices.](image-url)
Looking at figure 5 it can be seen that the only parts where \( \bar{v}_f \) and \( \bar{v}_s \) can have components normal to the surface are at the piston and at the orifices. Therefore, the second term on the right hand side can be split into three parts:

\[
\int_{CS} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS = \int_{S_p} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS + \int_{S_{in}} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS + \int_{S_{out}} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS
\] (5)

where \( S_p, S_{in}, \) and \( S_{out} \) are the surfaces at the piston and the orifices of the chamber.

Over the piston, the fluid velocity equals the velocity of the control surface, so the first part becomes zero. Thus:

\[
\int_{S_p} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS = 0
\] (6)

For the orifices \( \bar{v}_s = 0 \). Denoting the mean values of \( \beta \) over the inflow and outflow orifices as \( \beta_{in} \) and \( \beta_{out} \) the other parts become:

\[
\int_{S_{in}} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS = \beta_{in} \dot{m}_{in}
\] (7)

\[
\int_{S_{out}} \rho \beta (\bar{v}_f - \bar{v}_s) \hat{n} dS = -\beta_{out} \dot{m}_{out}
\] (8)

where \( \dot{m}_{in} \) and \( \dot{m}_{out} \) are incoming and outgoing mass flows.

If mass density \( \rho \) and energy concentration per mass \( \beta \) are considered to be uniform throughout the chamber, the total amount of B becomes

\[
B = \beta \cdot m
\] (9)

and the rate of change of B becomes

\[
\frac{dB}{dt} = \dot{\beta} m + \beta \dot{m}
\] (10)

The full conservation equations are thereby given on the form:

\[
\dot{\beta} m + \beta \dot{m} = \frac{d}{dt} \int_{CV} \rho \beta dV + \beta_{in} \dot{m}_{in} - \beta_{out} \dot{m}_{out}
\] (11)

Inserting that \( \beta = e + \frac{v_f^2}{2} \) for energy, equation (11) becomes:

\[
\frac{d}{dt} \left( e + \frac{v_f^2}{2} \right) m + \left( e + \frac{v_f^2}{2} \right) \dot{m} = \frac{d}{dt} \int_{CV} \rho \left( e + \frac{v_f^2}{2} \right) dV + \left( e + \frac{v_f^2}{2} \right)_{in} \dot{m}_{in} - \left( e + \frac{v_f^2}{2} \right)_{out} \dot{m}_{out}
\] (12)

The first term of the right hand side is divided into two parts: the effect from in- and outgoing gas and the heat transfer to the surroundings.
\[
\frac{d}{dt} \int_{CV} \rho \left( e + \frac{v_f^2}{2} \right) dV = \int_{CS} -P \bar{v} \hat{n} dS - \Phi
\]  

(13)

where \( \Phi \) is the heat transfer to the environment and \( P \) is the pressure in the chamber. The term \( \frac{d}{dt} \int_{CS} -P \bar{v} \hat{n} dS \) arises from the power of pressurized air doing a work at the edges of the control volume. The loss of internal energy this results in for an incremental surface segment \( dS \) is equal to \( P \bar{v} \hat{n} dS \) and so the energy gain is \( -P \bar{v} \hat{n} dS \).

\( P \) is assumed to be homogeneous in the chamber and \( \Phi \) is assumed to be a convection between the air and the cylinder walls. Further assuming that the cylinder walls have the same temperature as the surrounding atmosphere gives:

\[
\Phi = h A (T - T_a)
\]  

(14)

where:
- \( A \) is the area of the control surface,
- \( h \) is the convection coefficient,
- \( T \) is the temperature of the air in the chamber,
- \( T_a \) is the temperature of the surrounding atmosphere.

Since the fluid movement have a non-zero component normal to the control surface only on the piston surface and at the orifices, equation (13) becomes:

\[
\frac{d}{dt} \int_{CV} \rho \left( e + \frac{v_f^2}{2} \right) dV = -P v_p a_p + P_{in} v_{in} a_{in} - P_{out} v_{out} a_{out} - h A (T - T_a)
\]  

(15)

where:
- \( v_p \) is the piston speed and \( a_p \) is the piston area,
- \( v_{in} \) is the speed of incoming air and \( a_{in} \) is the area of the orifice for incoming air,
- \( v_{out} \) is the speed of outgoing air and \( a_{out} \) is the area of the orifice for outgoing air.

Inserting the new terms into (12) yields:

\[
\frac{d}{dt} \left( e + \frac{v_f^2}{2} \right) m + \left( e + \frac{v_f^2}{2} \right) \dot{m} = -P v_p a_p + P_{in} v_{in} a_{in} - P_{out} v_{out} a_{out} + \left( e + \frac{v_f^2}{2} \right)_{in} \dot{m}_{in} - \left( e + \frac{v_f^2}{2} \right)_{out} \dot{m}_{out} - h A (T - T_a)
\]  

(16)

Assuming that the kinetic energy is negligible, the only interesting energy will be the internal energy, \( e \). The assumption is made that the specific heat capacity is the same as it would be if the volume was constant. Then introducing \( c_v \) as the specific heat capacity at constant volume, \( e = c_v T \), equation (16) becomes:

\[
c_v \dot{T} m + c_v T \dot{m} = -P v_p a_p + P_{in} v_{in} a_{in} - P_{in} v_{out} a_{out} + c_v T_{in} \dot{m}_{in} - c_v T \dot{m}_{out} - h A (T - T_a)
\]  

(17)
with \( \dot{m} \) being the mass flow rate.

The ideal gas law \( PV = nRT \) can be rewritten to:

\[
P = \rho R_{\text{spec}} T
\]  

\( R_{\text{spec}} \) being the specific gas constant, which is also \( c_p - c_v \) where \( c_p \) is the specific heat capacity at constant pressure.

Using this form of the ideal gas law results in:

\[
c_v \dot{T}m + c_v T \dot{m} = -P v_p a_p + \rho R_{\text{spec}} T_{\text{in}} v_{\text{in}} a_{\text{in}} - \rho R_{\text{spec}} T_{\text{out}} a_{\text{out}} \\
+ c_v T_{\text{in}} \dot{m}_{\text{in}} - c_v T \dot{m}_{\text{out}} - hA(T - T_a)
\]

\[
= -P v_p a_p + R_{\text{spec}} T_{\text{in}} \dot{m}_{\text{in}} - R_{\text{spec}} T \dot{m}_{\text{out}} + c_v T_{\text{in}} \dot{m}_{\text{in}} - c_v T \dot{m}_{\text{out}} - hA(T - T_a)
\]  

(19)

This can be simplified into:

\[
c_v \dot{T}m + c_v T \dot{m} = -P v_p a_p + c_p T_{\text{in}} \dot{m}_{\text{in}} - c_p T \dot{m}_{\text{out}} - hA(T - T_a)
\]  

(20)

With some rearrangements of the equation, the energy conservation law takes the form:

\[
\dot{T}m + T \dot{m} = \frac{-P v_p a_p}{c_v} + \gamma T_{\text{in}} \dot{m}_{\text{in}} - \gamma T \dot{m}_{\text{out}} - \frac{hA(T - T_a)}{c_v}
\]  

(21)

Finally, using that \( \gamma \) turns up in two terms, equation (21) is simplified to:

\[
\dot{T}m + T \dot{m} = \frac{-P v_p a_p}{c_v} + \gamma (T_{\text{in}} \dot{m}_{\text{in}} - T \dot{m}_{\text{out}}) - \frac{hA(T - T_a)}{c_v}
\]  

(22)

where: \( T \) is the temperature in the chamber and \( \dot{T} \) is the rate of temperature change.

\( P \) is the chamber pressure,

\( v_p \) is the speed of the piston, counted as negative when it compresses the chamber,

\( a_p \) is the piston area,

\( c_v \) is the specific heat capacity of the air at constant volume,

\( \gamma = \frac{c_p}{c_v} \) is the ratio of specific heats,

\( T_{\text{in}} \) is the temperature of the incoming air,

\( h \) is the heat convection ratio,

\( A \) is the convection surface area,

\( T_a \) is the temperature of the surrounding atmosphere. (Modelon, 2010, p. 84),(Maré et al., 2000)

This model calculates the derivative of the temperature, updates the temperature and then calculates the pressure as a function of the temperature. Another way of modelling
the cylinder chambers would be to make a model for the pressure derivative and calculate
the pressure from there (Li et al. 2013).

4.2 Tubes

The air tubes involved in the *Pneumatics* package are modelled such that they consist
of a number of segments (see figure 6). This model has two governing equations, one for
pressure variation and one for mass flow rate variation. The equation for pressure variations
in one segment of the tube is given from equation (1):

\[ P_1 = \rho R_{\text{spec}} T = \frac{R_{\text{spec}} T m}{A_t L} \]  

(23)

where:
- \( m \) is the mass of the air in the tube,
- \( A_t \) is the cross-sectional area of the tube,
- \( L \) is the length of the tube segment,
- \( \dot{m}_1 \) and \( \dot{m}_2 \) are the mass flows at the first and second end respectively,
- \( P_1 \) and \( P_2 \) are the pressures at the first and second end respectively.

![Figure 6: Overview of a tube segment](image)

Under the assumption that the temperature is constant and uniform in the tube segment
and that the pressure in the segment is uniform with the value \( P_1 \) (see figure 6), the rate
of pressure change is given as:

\[ \frac{dP_1}{dt} = \frac{R_{\text{spec}} T}{A_t L} \dot{m} = \frac{R_{\text{spec}} T}{A_t L} \left( \dot{m}_1 - \dot{m}_2 \right) \]  

(24)
For the mass flow rate, the forces acting on the air in the segment are assumed to be the pressure forces from the two ends of the segment, as well as one force from the air friction in the segment. Newton’s second equation for the segment then becomes:

\[
\rho \cdot A_t \cdot L \cdot \frac{dv_f}{dt} = A_t \cdot P_0 - A_t \cdot P_1 - A_t \cdot \Delta P_{\text{friction}} \Rightarrow \tag{25}
\]

\[
\Rightarrow \rho \cdot L \cdot \frac{dv_f}{dt} = P_0 - P_1 - \Delta P_{\text{friction}} \tag{26}
\]

Assuming that the term \( \frac{\partial P}{\partial x} \) is the same through the whole segment, (26) turns into:

\[
\rho \cdot \frac{dv_f}{dt} = P_0 - \left( P_0 + \frac{\partial P}{\partial x} L \right) - \Delta P_{\text{friction}} \Rightarrow \tag{27}
\]

\[
\Rightarrow \frac{dv_f}{dt} + \frac{1}{\rho} \frac{\partial P}{\partial x} = - \frac{\Delta P_{\text{friction}}}{\rho L} \tag{28}
\]

where \( \Delta P_{\text{friction}} \) is the pressure drop due to frictional flow.

Inserting \( \dot{m}_1 = \rho A_t v_f \) into (28) gives:

\[
\frac{d}{dt} \frac{\dot{m}_1}{\rho} + \frac{1}{\rho} \frac{\partial P}{\partial x} = - \frac{\Delta P_{\text{friction}}}{\rho L} \Rightarrow \tag{29}
\]

\[
\Rightarrow \frac{1}{\rho A_t} \frac{d\dot{m}_1}{dt} + \frac{\dot{m}_1}{A_t} \frac{\partial P}{\rho \partial x} = - \frac{\Delta P_{\text{friction}}}{\rho L} \Rightarrow \tag{30}
\]

Solved for \( \frac{d\dot{m}_1}{dt} \), (30) becomes:

\[
\frac{d\dot{m}_1}{dt} = - A_t \frac{\partial P}{\partial x} - \rho \dot{m}_1 \frac{d^2}{dt^2} - A_t \frac{\Delta P_{\text{friction}}}{L} \tag{31}
\]

Neglecting the term \( \rho \dot{m} \frac{d^2}{dt^2} \) (see (Modelon, 2010, p. 55)), and assuming the segment is short enough that \( \frac{\partial P}{\partial x} \) can be considered uniform throughout it gives:

\[
\frac{d\dot{m}_1}{dt} = A_t \left( P_1 - P_2 \right) - A_t \frac{\Delta P_{\text{friction}}}{L} \tag{32}
\]

The pressure drop due to frictional flow \( \Delta P_{\text{friction}} \) is given by the relation:

\[
\Delta P_{\text{friction}} = \frac{\lambda \rho v_f^2 L}{2D} \tag{33}
\]

where: \( \lambda \) is the friction factor,

\( D \) is the diameter of the tube,
$v_f$ is the fluid velocity. (Modelon, 2010, p. 33)

$\lambda$ is itself given through different relations depending on the *Reynold’s number*:

$$Re = \frac{v_f D \rho}{\mu} = \frac{4 A_i}{D \pi} \cdot \frac{v_f \rho}{\mu} = \frac{4 \dot{m}}{\mu D \pi}$$

(34)

where $\mu$ is the dynamic viscosity of the fluid.

With Reynold’s number established $\lambda$ is, in the *Pneumatics* package, modelled as:

If $Re < 2000$ (laminar flow):

$$\lambda = \frac{64 \mu}{\rho v_f D}$$

(35)

If $2000 < Re < 4000$ (transition between laminar and turbulent flow):

$$\lambda = 0.00276 \cdot Re^{0.322}$$

(36)

If $4000 < Re < 80000$ (turbulent flow):

$$\lambda = \frac{0.3164}{Re^{0.25}}$$

(37)

If $Re > 80000$ (very turbulent flow):

$$\lambda = 0.0032 + 0.221 Re^{-0.237}$$

(38)

(Modelon, 2010, pp. 33-34)

An alternative approach is to calculate the change in mass flow at the cylinder using the equation:

$$\dot{m}(L_t, t) = \begin{cases} 
0 & \text{if } t < L_t/c \\
\exp\left(-R_t R_{spec} T \frac{L_t}{2P} \frac{c}{h}\left(t - \frac{L_t}{c}\right)\right) & \text{if } t > L_t/c 
\end{cases}$$

where: $L_t$ is the length of the tube,
$c$ is the speed of sound,
$R_t$ is the airflow resistance of the tube,
$P$ is the pressure in the cylinder chamber. (Richer et al, 2000, pp. 418-419)

This model would allow for different shapes of tubes, however it requires knowledge of the tube resistance and was therefore ultimately discarded.
4.3 Orifice Equation

Every time air flows from one pneumatic part to another, it has to travel through an orifice (see figure 7 below) which affects the air flow between the parts.

The orifices in the Pneumatics package are modelled according to the ISO 6358, which is based on the equation:

\[
\dot{m} = \begin{cases} 
  P_u C \rho_0 \sqrt{\frac{T_0}{T_u}} \cdot \sqrt{1 - \left( \frac{P_d/P_u - b}{1 - b} \right)^2} & \text{if } \frac{P_d}{P_u} > b \\
  P_u C \rho_0 \sqrt{\frac{T_0}{T_u}} & \text{if } \frac{P_d}{P_u} \leq b
\end{cases}
\]

where: 
- \( P_u \) is the upstream pressure,
- \( P_d \) is the downstream pressure,
- \( C \) is the sonic conductance of the orifice,
- \( \rho_0 \) is the air density at reference conditions,
- \( T_0 \) is the temperature at reference condition,
- \( T_u \) is the temperature of the upstream fluid,
- \( b \) is the critical pressure ratio. Its value for air is found in section 4.7.

When \( \frac{P_d}{P_u} \leq b \) the flow is **choked** and further lowering \( P_d \) will not affect the air flow.

ISO 6358 also gives the reference condition:

- \( T_0 = 293.15 \) K,
- \( \rho_0 = 1.185 \) kg/m\(^3\),
- Relative air humidity 65%. (Modelon, 2010, pp. 37-38)
For the conductance, if the ratio between length and diameter of the orifice have a value between 0.33 and 10, equation (39) is used:

\[ C = 8d^2 \frac{l}{\text{min} \cdot \text{bar}} \]  

where \( d \) is the diameter given in mm (Modelon, 2010, p. 43).

### 4.4 Regulator

If the inner workings of a regulator is known, a model can be used which is based on the movement equations of a small mass affected by pressure, friction and forces applied from internal parts, as is done in (Wang et al., 2007). Since no good description of the dynamics of the regulator in the system could be found, a simple steady state-model is instead used. In this model it is assumed that the inflow is given by:

\[
\dot{m}_{in} = \begin{cases} 
(P_u - P_d)C_{in} & \text{if } P_d < (P_{nom} - P_{work}) \\
(P_u - P_d)\frac{(P_{nom} - P_d)}{P_{work}}C_{in} & \text{if } (P_{nom} - P_{work}) < P_d < P_{nom} \\
0 & \text{if } P_d > P_{nom}
\end{cases}
\]

and the outflow by:

\[
\dot{m}_{out} = \begin{cases} 
(P_d - P_u)C_{out} & \text{if } P_d > (P_{nom} + P_{work}) \\
(P_d - P_u)\frac{(P_l - P_{nom})}{P_{work}}C_{out} & \text{if } P_{nom} < P_d < (P_{nom} + P_{work}) \\
0 & \text{if } P_d < P_{nom}
\end{cases}
\]

(Modelon, 2010, p. 126-127)

A schematic overview of the in- and outflow can be seen in figure 8. The thick line is the inflow and the thinner line is the outflow.
4.5 Piston Movement

The main forces working on the piston are thought to be the pressure forces from the cylinder chambers and environment, the load force and the friction force. Newton’s second law applied to the piston then becomes:

\[ M \cdot a = A_1 P_1 - A_2 P_2 - A_r P_a + F_{ext} - F_{fric} \]  

where: 
- \( M \) is the piston mass,
- \( A_1 \) and \( A_2 \) are the areas of the piston on side 1 and 2 respectively,
- \( A_r \) is the cross-sectional area of the piston rod,
- \( P_1 \) and \( P_2 \) are the pressures of cylinder chambers 1 and 2,
- \( P_a \) is the pressure of the surrounding atmosphere,
- \( F_{ext} \) is the external applied force,
- \( F_{fric} \) is the internal friction force, (see figure 9)
4.6 Friction Force

There are many different models for friction (Olsson et al, 1997). The one used in the *Pneumatics* package is given by the model:

\[
F_{fric} = F_C + K_{prop}v + F_{stribeck}e^{-f_{exp}v}
\]  \( (41) \)

where: 
- \( F_C \) is the coulomb friction force,
- \( K_{prop} \) is the viscous friction constant,
- \( F_{stribeck} \) is the Striebeck effect constant,
- \( f_{exp} \) is the coefficient of exponential decay for the Striebeck effect.

The coulomb friction force is given by:

\[
F_C = \begin{cases} 
F_{tot} \text{sign}(v) & \text{if } |F_{tot}| < F_{Cmax}, \quad v_p = 0 \\
F_{Cmax} & \text{else}
\end{cases}
\]

\( F_{Cmax} \) being the force at which the piston starts to move and \( F_{tot} \) being the sum of all non-frictional forces. (Modelon, 2010, p. 87)

The above model works well for simulations where the applied force is predetermined and the displacements are to be computed. However, when the displacements are known and the forces unknown, it gives rise to numerical problems since the force is not uniquely determined and have a noticeable discontinuity at \( v_p = 0 \). Having a model that works for known displacements is desirable since the known movements in the system are based on predetermined velocities rather than forces. Therefore a slight modification was made, introducing a small velocity span, denoted \( v_{span} \), and turning the friction model into:

\[
\text{if } |v| > v_{span} \\
F_{fric} = F_C + K_{prop}v + F_{stribeck}e^{-f_{exp}v}
\]

\( (42) \)

\[
\text{if } |v| \leq v_{span} \\
F_{fric} = \left( F_C + K_{prop}v_{span} + F_{stribeck}e^{-f_{exp}|v_{span}|} \right) \frac{v_{rel}}{v_{span}}
\]

\( (43) \)

The two models are compared in figure 10.
Figure 10: An overview of the two friction models. The blue line shows the original model while the red line shows the modified model.

For known applied forces and a small value for $v_{span}$, the two friction models give similar results (see figure 29 in Appendix 11.2), but the second one also work for controlled displacements. Therefore it was the one used in the simulation.

It can be noted that in some models the Coulomb Friction is given as a dependant of the pressure setting (Kleidon, 1984). This was not woven into the model in this thesis because there seemed to be no such correlation in our tests (see section 6).

The damping could also be modelled as non-viscous (Lin et al. 1992). However this would make the model more complicated and nothing was found indicating that it would improve accuracy.

4.7 Air Properties

The following units are for air at room temperature ($20^\circ C$).

- $c_p$: 1010 \( \frac{J}{kgK} \)
- $c_v$: 718 \( \frac{J}{kgK} \)
- $R_{spec}$: 288 \( \frac{J}{kgK} \)
- $\mu$: 1.98$ \cdot 10^{-5}$ \( Pa \cdot s \)
- $b$: 0.528 \[1\] (TheEngineeringToolbox, 2014)
where: 

- $c_p$ is the specific heat capacity at constant pressure
- $c_v$ is the specific heat capacity at constant volume
- $R_{\text{spec}}$ is the specific gas constant, $R_{\text{spec}} = c_p - c_v$
- $\mu$ is the dynamic viscosity.
4.8 Combining the equations

When a model has been made, *Dymola* will take all equations and combine them into one set of ordinary differential equations. Since all this is done by the program itself, hidden from human eyes, the exact look of the set will not be known, however it will be on the general form:

\[
F(y', y, t) = 0 \\
y(t_0) = y_0
\]

where the set will be made up by a combination of the equations found in this chapter.

There are multiple methods for solving ordinary differential equations, but as standard solver, *Dymola* uses a solver from the *Dassl* family.

4.8.1 Dassl

Dassl solvers are based on the numerical discretizations known as Backwards Differentiation Formulas (BDFs), which are of the type:

\[
F(t_{n+1}, y_{n+1}, \frac{1}{h} \sum_{i=0}^{k} \alpha_i y_{n+1-i}) = 0 (44)
\]

In the dassl family, the order \( k \) is limited to: \( 1 \leq k \leq 5 \). The values of the different \( \alpha_i \) are dependent on the order of the method used. In *Dymola*, the order of the BDF is hidden from the user.

Using a BDF method results in a set of equations which have to be solved using some iterating method. In *Dassl*, the method is a modified Newton iteration. (Petzold, 1995)

In Newton’s Method, the Jacobian, \( F'(y) \), is computed at every iteration and the iteration follows the scheme:

\[
F'(y_n) \Delta y_n = -F(y_n) \\
y_{n+1} = y_n + \Delta y_n
\]

(Führer et al, 2008).

In Dassl, \( F'(y_n) = F'(y_0) \) is used throughout the whole iteration. It is only in case the iteration doesn’t converge that \( F'(y_n) \) is updated.
5 Execution

A number of tests were performed on the pneumatic cylinder to determine its force characteristics for different pressures, among them the normal working pressure of the cylinder. The characteristics for this pressure were then inserted into a Dymola model of the pneumatic subsystem. This model was in turn connected to an Abaqus model of the mechanical subsystem and a co-simulation between the two programs was run.

5.1 Experimental Tests

The tests were performed on the Division of Solid Mechanics at the Lund’s University Faculty of Engineering (Lunds Tekniska Högskola, LTH). The machine used a load cell able to apply force up to 25 kN.

First, the cylinder was inserted into the load testing machine. The actual cylinder was fitted to an unmoving part of the machine so that it would be kept still throughout the tests. The piston rod was fitted to a part that was movable along one axis (see figure 11). The movement of this part could be programmed either by applying predetermined forces, or through giving it a predetermined displacement curve. When the cylinder was in place, it was pressurized to 3 bar absolute pressure. Initially, the piston was kept in place in the middle of the cylinder. Two sine formed displacement curves were then applied to the piston and the forces working on the unmoving end was measured. The tests were run for several cycles each in order to get a sufficient amount of data to establish a cylinder model.

Figure 11: The pneumatic cylinder. Note that the tube in the figure is connected to the end with the rod protruding from the cylinder. In the actual tests, it was the other end that was pressurized.
The following values were used for the sine curves:

1. 5 Hz frequency, 2 mm amplitude
2. 2 Hz frequency, 4 mm amplitude

Since the friction model was velocity dependent the forces measured by the load cell were plotted versus the velocities. This was done for the standard working pressure of 3 bar, as well as for the pressures 2.5 bar and 3.5 bar.

5.2 Implementation

5.2.1 Dymola Model

The Dymola model was built up according to figure 12.

![Figure 12: The structure of the Dymola model of the pneumatic subsystem](image)

The different parts work in the following way:

1. The part DISP takes input from Abaqus giving the displacement of the connection point in the Abaqus model.

2. The input from DISP is only given as a real number. This part however tells that the values for DISP are positions (as opposed to e.g. a force). However, if the value given is zero, the pneumatic model will place the piston at one of the cylinders endpoints. Since the piston should be placed in the middle of the cylinder when the displacement is zero, a value corresponding to half the stroke have to be added to the values from DISP before they are sent as input to the position part.
3. The force sensor can tell the forces acting in the cylinder. This is what makes it possible for Dymola to send Abaqus information.

4. The cylinder is the central part of the pneumatic model just as it is of the system. The positions given to the cylinder tells it how to move the piston. The movement of the piston then gives rise to friction forces and pressure variations. The total forces are the ones that will be sent to Abaqus through the force sensor, since these are the forces that act on the mechanical system modelled in Abaqus.

5. The air supply is modelled as a large reservoir with a constant pressure since the physical air supply should have a constant pressure.

6. The regulator adjusts air flow to and from the cylinder in order to maintain the pressure at a certain level. When the pressure on the downside (right side in the picture) falls below the desired value, the regulator lets air flow from the supply to the tube. If, however, the pressure rises above the desired value, air flows from the tube out into the atmosphere.

7. The tube lets air flow between the regulator and the cylinder depending on the pressures at the ends of the tube. The length and diameter of the tube affects how quickly air flows to and from the cylinder as well as how much of the air flowing through the regulator actually reaches the cylinder. Thereby it determines how quickly the pressure can be returned to the desired value.

Wherever two classes connects to each other in the Dymola model, an orifice working according to section 4.3 is included in the model. This can however not be seen in the figure.

For the most parts, the components of the model were the same as in the actual physical machine. However, a few changes were made:

- In the model there was a large tank with constant atmospheric pressure connected to the non-regulated cylinder chamber. In the real system this chamber is just open straight into the open air.

- The real cylinder has only one rod, the one in the model had two. However, by setting the diameter and length of one rod to zero, the model was in practice made identical to a model with only one rod. The reason that the two-rod cylinder was used was because it was the only one where the friction model could be modified according to Eq. (42)-(43).
No force sensor is present in the physical system. The one in the model is there to pick out the force in the cylinder from all the other cylinder variables and forward it to the Abaqus model.

The Dymola model was run for the same displacements that were used in the experimental tests. The friction parameters, as well as the conductances of the cylinder orifices, were then adjusted to make the simulated forces correspond to the measured forces as much as possible. The reason that the conductances were set this way was that choking valves were attached to the cylinder and it was difficult to determine the conductance of those valves.

5.2.2 Fitting Of Cylinder Parameters

The forces developed in the model are affected by the following unknown parameters:

- The conductance of the in- and outflow orifice of the pressurized chamber.
- The conductance of the in- and outflow orifice of the non-pressurized chamber. (see section 4.3).
- The Coulomb friction working on the piston.
- The damping coefficient of the cylinder.
- The Stribbeck friction effect on the piston.
- The exponential decay rate of the Stribbeck effect. (See equations (42)-(43)).

Therefore, these parameters were modified to make the force curves of the model fit those of the experimental tests for the working pressure of 3 bar. To see how the parameters held up for different pressures, the pressure was changed in the model while the rest of the parameters were kept unchanged. For the absolute pressures 2.5 bar and 3.5 bar, the simulated results were then compared with the experimental results.
5.2.3 Abaqus Model

The Abaqus model was built up according to figure 13-14.

Figure 13: The structure of the Abaqus model representing the mechanical subsystem

Figure 14: Nomenclature of parts in the Abaqus model
All parts except the paper web were modelled as rigid bodies rather than deformable parts, in order to speed up the simulations. It should be noted that one of the parts was modelled as a combination of a beam element and a hinge element rather than a physical model using the approximate geometry of the real part. This was done because including a model of the real part would require a contact interaction between two circular surfaces, something that can give inaccurate results. All elements in the Finite Element model were linear elements.

The material model for the package material was a linear, orthotropic model. Plasticity was not included in the model although the material has been tested, with the plasticity levels being determined to somewhere above 10 MPa for tension and 5 MPa for compression.

Throughout the whole simulation, a gravitational force was defined, acting only on the pendulum roller and the web. This was done since the gravity would have no effect on the driven roller and the bending roller, while its effect on the other parts was thought to be negligible.

The Abaqus model was simulated in three stages, which in Abaqus are called *steps*.

In the first step, a predefined velocity downwards was applied at the right end of the web, while the left end was restricted from moving vertically. This had the effect of stretching the web (see figure 15). However, there was quite a bit of waves moving through the web.

![Figure 15: The model after the web has been stretched out](image)
Since there was a long stretch of material hanging from the driven roller (only the uppermost part is present in figure 13), these waves were allowed to grow rather large. To stop this, two metal plates in the form of analytical surfaces were added to the model according to figure 16a and 16b. These were then pushed against the web, making the distance the web could swing much smaller. This made it impossible for the waves to grow large. Lessening the waves was desired since they would affect the measured forces.

The whole system with the added plates

Close-up of the plates

Figure 16: Figures of the wave reducing system

The second step was a stabilization step. In this step, a constant force was applied at the right end of the web, while the left end was kept unmoving. This allowed for the waves in the web to die out while still keeping the web stretched.

In both the first and the second steps, all the rollers were kept in the same positions but allowed to rotate.

The cosimulation is run during the third step. This was done because Abaqus only allows cosimulation during one step. The third step represented the actual forming process
and the objective of the model was to properly simulate the forming process.

5.2.4 Cosimulation

The cosimulation was run from Abaqus, using a Dymola plugin. For the Dymola model, the only thing that had to be done was to have the program reorganize the code and turn it into C-code.

For the Abaqus model however, a few modifications had to be made. Firstly, the pendulum roller was now allowed to move as in the physical machine. Then the left end of the web in figure 14 was given a prescribed velocity downwards, with the amplitude given from previous runs of the real process. Also, the driven roller was given a constant prescribed angular velocity that made the surface of the roller move with the same average speed as the left end of the web in figure 13. Lastly, the interfaces to the Dymola model were introduced as acting on the reference point of the cylinder holder. This could not be done from the start since, as was mentioned in section 5.2.3, Abaqus only allows one step to be included in a cosimulation.

Also the output variables from Abaqus were modified so that the total reaction force in the left edge of the web was being saved throughout the simulation. This is done by calculating the total of all reaction forces acting on the edge of the web, and setting that total as the reaction force in the point called $RP-1$. Then the reaction forces in this point are recorded and saved throughout the simulation of the forming process.
6 Calibrations

The forces for the different displacement curves, given as functions of velocity, are found in figures 17-18.

![Figure 17](image1.png)

Figure 17: Measured forces for a displacement curve with 2Hz frequency and 4 mm amplitude

![Figure 18](image2.png)

Figure 18: Measured forces for a displacement curve with 5Hz frequency and 2 mm amplitude

An analysis of the curves showed that the average force was around 176 N, which corresponded to an absolute pressure of 3.2 bar rather than 3.0 bar. Therefore, the pressure of the model was changed to 3.2 bar when trying to fit the parameters of the cylinders to the measured forces. Fitting simulated curves to the experimental results gave the parameter values seen in table 1.
Table 1: Cylinder parameters

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet conductance pressurized chamber</td>
<td>$C_a$</td>
<td>1.7</td>
<td>l min·bar</td>
</tr>
<tr>
<td>Inlet conductance non-pressurized chamber</td>
<td>$C_b$</td>
<td>2</td>
<td>l min·bar</td>
</tr>
<tr>
<td>Coulomb Friction</td>
<td>$F_C$</td>
<td>13</td>
<td>N</td>
</tr>
<tr>
<td>Damping Coefficient</td>
<td>$K_{prop}$</td>
<td>360</td>
<td>N·s m</td>
</tr>
<tr>
<td>Strubeck Friction</td>
<td>$F_{strbeck}$</td>
<td>0</td>
<td>N</td>
</tr>
<tr>
<td>Exponential decay</td>
<td>$f_{exp}$</td>
<td>0</td>
<td>[1]</td>
</tr>
</tbody>
</table>

It is noticeable that this configuration does not include any Strubeck effect, thus simplifying the friction model.

The results given by simulations with the model in table 1 can be seen beside the experimental results in figures 19-20. The red lines are the experimentally measured forces and the smoother blue lines are the simulated forces.

Figure 19: Measured and simulated forces for a displacement curve with 2Hz frequency and 4 mm amplitude
For the 2.5 bar test, the actual pressure turned out to be 2.76 bar and for the test were the pressure was supposed to be 3.5 bar, it turned out to be 3.73 bar. The results for the 2.5 bar pressure can be seen in figures 21-22.
Figure 22: Measured and simulated forces for a displacement curve with 5Hz frequency and 2 mm amplitude at 2.5 bar pressure

The results for the 3.5 bar pressure can be seen in figures 23-24.

Figure 23: Measured and simulated forces for a displacement curve with 2Hz frequency and 4 mm amplitude at 3.5 bar pressure
Figure 24: Measured and simulated forces for a displacement curve with 5Hz frequency and 2 mm amplitude at 3.5 bar pressure
7 Results Web Force

The simulated results of the web tension force, with the standard pressure of 3 bar absolute pressure, are seen in figure 25.

![Web Force Graph](image)

Figure 25: Variations of the simulated reaction force in the end of the web

In figure 26, which is found on next page, the results from the simulation are plotted together with the measured results from earlier tests performed on the system. The simulated results have been filtered so that variations with a frequency over 100 Hz are removed in order to eliminate noise arising from numerical inaccuracies. For reasons that will be discussed in chapter 8, the first time of the simulation is excluded from the result comparison.
The mean force in the simulations is lower than the measured forces while the variations are larger. This is quantified in table 2.

<table>
<thead>
<tr>
<th>Force:</th>
<th>Mean:</th>
<th>Amplitude:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured</td>
<td>48</td>
<td>7N</td>
</tr>
<tr>
<td>Simulated</td>
<td>34</td>
<td>22N</td>
</tr>
</tbody>
</table>

The forming cycle is half a second long. Thus a periodicity of 2 Hz should be present in the simulated results and this is also the case.
7.1 Material Stresses

The stresses along the paper web in the simulation was below the plasticity level of the material for most parts of the web. However, in the parts which were curved before the start of the simulation, the stresses reached higher levels, and over the rollers small sections, only one element in size, occasionally reached below $-5 \text{ MPa}$ (5 MPa compression). See figure 27.

![Figure 27: Snapshot of the web colorcoded after stress levels. The black parts are where the stresses reached below the plasticity limit.](image_url)
7.2 Filtering Of Results

In order to make sure that the force variation amplitudes weren’t considerably affected by the filtering out of high frequency components, the filtered simulation results were plotted together with the unfiltered ones. The result is seen in figure 28.

Figure 28: Comparison between filtered and unfiltered results

(a) Larger comparison

(b) Zoomed in view. The sharp edges are from the unfiltered results
8 Discussion

8.1 Experimental Tests

For the experimental tests it would probably be better to use a load cell dimensioned for smaller forces. Mostly that should be useful since such a cell would probably have a higher force resolution allowing it to record the forces in the cylinder more accurately. However, it is quite possible that this would only result in a less broad band of measured forces but following the same curves. Also, sometimes the forces applied were not the ones that the machine was programmed to apply. Therefore, a smaller load cell might allow a wider range of tests with smaller force variations.

8.2 Cylinder Parameters

Most noticeable is probably the fact that there is no steep slope around $v = 0$ but rather one lesser slope on each side (see figures 17-20). This makes it impossible for the model to efficiently follow the curve around $v = 0$. Therefore focus was put on making the forces from the model match the experimental forces at the somewhat higher speeds. In these ranges, the model seems to match the experiments rather well (see figures 19 and 20). The correspondence with the experiments could be better for each of the curves but the model used was the best compromise that could be found.

Figures 21-24 indicates that the model still works well for different pressures in the cylinder chamber. The correspondence seems especially good for the lower pressure. This may be valuable since the mean of the measured forces suggested the pressure setting was a bit too high and a highly pressure-dependent model would thus be rendered insufficient.

8.3 Web Force Results

The very large and rapid variations in force that can be seen in the beginning of figure 25 are the results of a sudden movement. When the web starts moving, the velocity of the end goes from zero to the full speed of the process in one time step. This produces a very unstable behaviour which should be ignored. Therefore, only the results from 5 seconds into the simulations were considered.

The results for the Web Force were not very consistent with the actual measured forces. A number of factors may have affected this and what amount of difference each of these makes is hard to estimate. The three most prominent factors are thought to be that:

- The rollers in the model are rolling freely, without any friction affecting the movement.
  This is of course not the case with the rollers in the physical system, although it is
not known how much friction is present. Considering that the movement of the web have a prescribed velocity, including friction in the model should increase the forces developed in the web. Whether it will affect the force variations is however unclear.

- The Driven Roller in the actual system contains a regulating unit so that the web should move with the same speed over that roller as over the bending roller. In the thesis it was assumed that the angular velocity of the roller was kept constant with a speed so that on average the web moved as much over the driven roller as over the bending roller. However, the regulating unit may keep the movements more synchronized, thus keeping down the amplitude of the force variations.

- The web on the right side of the driven roller in figure 13 is swinging when the process starts which makes it move differently than it does in the real world. Adding a pinching device that stops the web from swinging just before it reaches the driven roller might be a possible solution to this problem.

The positive part is that the cyclical behaviour that was expected is indeed present. The machine forms two packages every second so the forces in the web should show a more or less cyclical behaviour with a frequency of $2 \text{ Hz}$. This is also what is seen, so it would seem the basic dynamic behaviour is correct. Therefore, the model shows a certain promise for the future.

Not including plasticity in the material model also seems to have been correct. von Mises stresses reaching above the plasticity limit could be found in a few parts of the web. However, almost all these parts were the parts in which the web was curved before the simulation started. Since no parts of the material in the real system are curved, these parts can be ignored. After 4 seconds the curved parts are no longer in the system. The other parts that reached the plasticity levels were few and small enough that they seem to be the results of elements belonging to the web sticking to elements belonging to the roller. This was also supported by the fact that stretching the web should not result in large compression as is the case in these regions. Therefore these stresses were an occurrence not representing the real system why not including plasticity seemed not to be an error source.

The sticking together of elements itself might have caused some errors. However, since there was so little of it happening, its effects on the results were probably negligible.

The biggest problem with the results is that the average force is lower in simulations than in physical tests. To at least have an average value corresponding well with the real value would quite possibly make the model a better alternative to the current model with a constant Web Force.
8.3.1 Filtering Of Results

Filtering the results seems to have been a good idea since the results were smoothed out while the overall behaviours and amplitudes were preserved.

8.4 Programs

The cosimulation between Abaqus and Dymola carried with it a rather large amount of problems. The first problem that was encountered was the fact that an extra license for Dymola was required in order to make the code available for cosimulation. This license was rather expensive and took some time to get a hold of so that the cosimulations could get started.

The cosimulation interface was developed so that each version of Abaqus should be able to run cosimulations with the corresponding version of Dymola. However, this was not something their local support in Sweden was informed on. This caused some rather confusing problems since some cosimulations would work while others would not. This turned out to be because newer versions of Dymola use newer versions of Modelica but models using an older version could still be run. Therefore, if one is to run a cosimulation between the programs it is advisable to make sure that they are of corresponding versions.

The cosimulations were run through a plug-in to Abaqus which had Abaqus write a file with instructions on how to run the simulation. The plug-in then added a few lines of text to the file enabling and finally ran the cosimulation. For some models, including the final version, the lines were not added though. This forced the lines to be added manually and the cosimulation being run from a terminal. This way the simulations finally worked.

It should be noted that solving the problems mentioned above was rather time-consuming. Therefore, I am not sure that I would recommend using an Abaqus-Dymola cosimulation for future projects. However, this project have given some valuable testing of the cosimulation which might lead to improvements in the cosimulation interface. Therefore it is quite possible that it will be easier to use the cosimulation for other people needing to combine a FEM model with a model of a system that is not suited for Finite Element treatment.
9 Future Work

Going forward, there are a few steps to be taken:

- The model should be modified to include frictional effects from all rollers except the driven one in order to get better results. Implementing the regulating unit from the real system will probably not be doable since no information of its workings is available. However it is thought that this system has a considerably lower effect on the forces than the neglected frictions have.

- The model should also be included in the larger forming model used at Carton Ecenomy. Currently, the forming model assumes that the Web Force is constant. There is a definite possibility that the model included in this thesis will still improve the results since it allows variations and it is thought that force variations are responsible for some of the forming defects that are not reproduced in the current forming simulations.

- If the model shows signs of improving results, an attempt will be made to construct Abaqus elements with characteristics corresponding to those from the pneumatic system. This would lower the dependency on the cosimulation interface which seems somewhat unreliable and slows down the simulation a bit.
10 References


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11 Appendices

11.1 Quantities and units

11.1.1 Cylinder chamber

**Quantity:**

- $B$, total quantity in conservation equation [$kg, J$]
- $U$, total energy in system [$J$]
- $\beta$, conversation quantity per mass unit [$J/kg, 1$]
- $e$, internal energy per mass unit [$J$]
- $CV$, control volume [$m^3$]
- $CS$, surface of control volume [$m^2$]
- $\bar{v}_f, v_f$, fluid velocity and fluid speed [$m/s$]
- $\bar{v}_s, v_s$, velocity and speed of control surface [$m/s$]
- $\hat{n}$, unity vector normal to surface, pointing outwards [1]
- $S_p$, piston surface [$m^2$]
- $S_{in}, S_{out}$, surfaces for in- and outflow [$m^2$]
- $\Phi$, effect of convection [W]
- $h$, convection coefficient [W/m²K]
- $A$, area of the control surface [$m^2$]
- $T, T_a$, temperature in chamber and in surrounding atmosphere [K]
- $v_p$, speed of the piston movement [$m/s$]
- $a_p$, area of the piston [$m^2$]
- $v_{in}, v_{out}$, speed of incoming and outgoing fluid [$m/s$]
- $a_{in}, a_{out}$, area of inflow orifice and outflow orifice [$m^2$]
- $c_v$, specific heat capacity at constant volume [$J/(kgK)$]
- $c_p$, specific heat capacity at constant pressure [$J/(kgK)$]
- $R_{spec}$, specific gas constant for the gas, $c_p - c_v$ [$J/(kgK)$]
11.1.2 Air orifice

**Quantity:**

- $P_u$, pressure upstreams of an orifice $[Pa]$
- $P_d$, pressure downstreams of an orifice $[Pa]$
- $C_o$, sonic conductance of an orifice $[J/(kgK)]$
- $\rho_0$, air density under ISO 6358 reference conditions $[kg/m^3]$
- $T_0$, air temperature under ISO 6358 reference conditions $[K]$
- $T_u$, air temperature upstreams of an orifice $[K]$
- $b$, critical pressure ratio for choked flow through orifice, $c_p - c_v$ $[J/(kgK)]$

11.1.3 Forces

**Quantity:**

- $A_1$ and $A_2$, areas of the piston on sides 1 and 2 respectively $[m^2]$
- $A_r$, area of the piston rod $[m^2]$
- $P_1$ and $P_2$, pressure in chamber 1 and chamber 2 $[Pa]$
- $P_a$, pressure of the atmosphere $[Pa]$
- $F_{ext}$, external applied force $[N]$
- $F_{fric}$, piston friction force $[N]$
- $F_C$, coulomb friction on piston $[N]$
- $K_{Prop}$, viscous friction constant $[\frac{N}{m/s}]$
- $F_{stribek}$, Stribeck effect $[N]$
- $f_{exp}$, coefficient of exponential decay $[s/m]$

11.1.4 Tubes

- $A_t$, cross-sectional area of tube $[m^2]$
- $L$, length of tube segment $[m]$
\( \dot{m}_1, \dot{m}_2 \), mass air flow at the end points of a tube segment \([kg/s]\)

\( P_1, P_2 \), pressures at the end points of a tube segment \([Pa]\)

\( \Delta P_{friction} \), pressure drop in tube segment due to friction \([Pa]\)

\( D \), tube diameter \([m]\)

\( \lambda \), friction factor of air \([Pa \cdot s]\)

\( \mu \), dynamic viscosity of fluid \([Pa \cdot s]\)

11.2 Comparison of friction models with controlled forces

![Figure 29: An overview of the two friction models. The blue (upper) line is from Modelon’s original model and the red (lower) line is from the modified model.](image-url)