Rexroth
Bosch Group

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# Simulation of a hydraulic variable axial piston double pump of bent axis design with subsystems

## Abstract

Hydraulic pumps of bent axis design are often used in mobile hydraulic applications like excavator and forest machines.

As a result of the construction these pump types have a very good efficiency coefficient at low rotational speed. The disadvantages is the more complex construction, because a double bearing of the connecting rods is necessary.

The drive of the piston drum by the connecting rod, and the cyclic change of it during one rotation or by a variation the rotational speed, result in torsion oscillations. The consequences could be a damage of the connecting rods and the gear wheels.

Regarding the complex kinematics of the double pump and the hydraulic compression and decompression phase inside the pump, a model based on mathematical equations is very complex and difficult to handle.

The simulation of the whole system in ADAMS in consideration of the elasticity of the drive (clutches, gear) and the variation of the speed of the diesel engine, delivers the forces of the piston and the torque of the drive shaft in critical load conditions.

The resulting dynamic torque is a more critical load for the pistons and gear than the static torque, which is often used for sizing systems.
The results were used in finite element calculation to find out the maximum stress under dynamic load and are the basis for optimising the pumps.

This presentation shows the modelling of the whole system and the calculation of the exact dynamic loads in consideration of the complex pump kinematics, elasticities and hydraulic subsystems.

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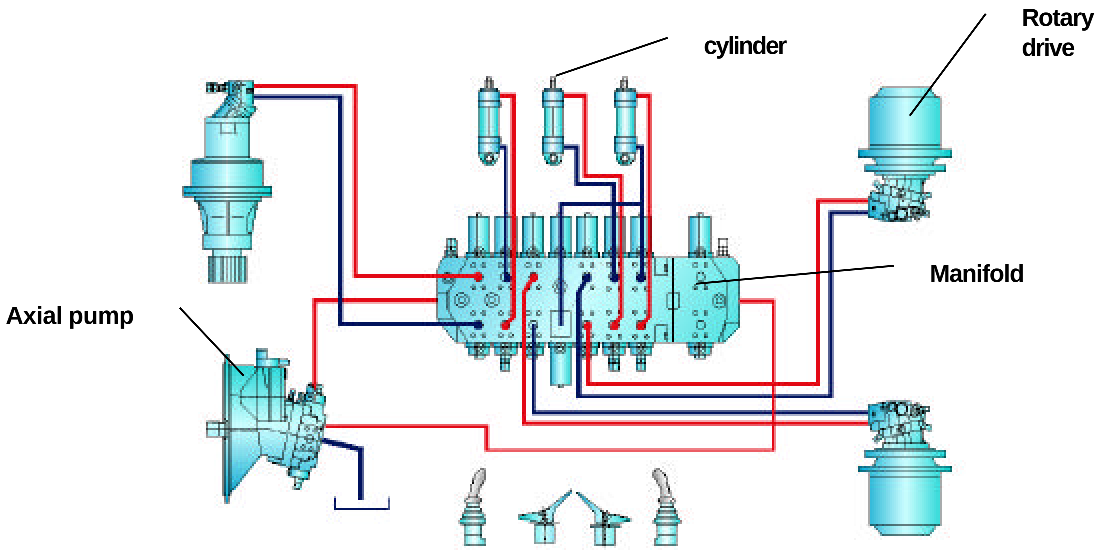
## Introduction

Hydraulic pumps of bent axis design are often used in mobile hydraulic applications like excavator and forest machines. The following pictures show a typical application for those pumps.

**Pic. 1: Real world (r) and virtual model (l)**

The pump is the central part of the system and supplies all other parts with hydraulic power and when it fails the whole machine is blocked. So it is necessary that this component is very reliable during different load conditions.

The pump converts the mechanical energy from the diesel engine in hydraulic power, which is controlled by a manifold and an electronic control system.




Pic. 2: Mobile application of double pump

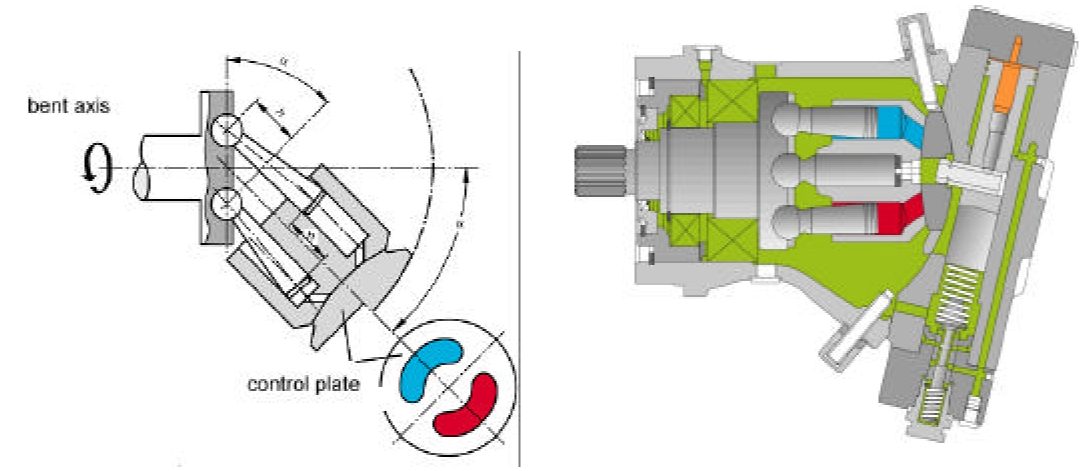
**Fundamentals of an axial piston pump of bent axis design**

Depending from the speed of the engine and the swivel angle the pump delivers a flow and the torque of the drive shaft is the result of the pressure difference between tank and load.

$$Q_P \approx z \cdot A_k \cdot D_k \cdot n_P \cdot \tan\alpha \qquad M_{Antr} = \frac{V_P \cdot (p_P - p_T)}{2 \cdot \pi}$$

The axial piston pumps are divided in two different groups. The axial piston pump with bent axis design and swash plate design.

The pistons of a pump of bent axis design are connected with spherical joints to the drive shaft and they are axial forced in the rotating piston drum.
The toggle of each piston must be allowed, because otherwise the system becomes




**Pic. 3: Principle of an axial piston pump of bent axis design**

kinematic blocked. The reason is that the angular of the drum builds an ellipse on the flange of the drive shaft.

The stroke of the pistons is depending from swivel angle between drive shaft and drum. The kidney-shaped openings of the control plate, which is not rotating, connects the piston with high pressure to power supply (red) and low pressure or tank (blue).

The control plate with its spherical bearing is moved by a bolt, which has also an spherical bearing so that the stroke form the piston is directly depending from the swivel angle.

The bearing of the pistons minimises the friction between pistons and drum so that the forces between them are very low.

As a result of the construction these pump types have a very good efficiency coefficient at low rotational speed. The disadvantages is the more complex construction, because a double bearing of the connecting rods are necessary.

The drive of the piston drum by the connecting rod, and the cyclic change of it during one rotation or by a variation the rotational speed, result in torsion oscillations. The consequences could be a damage of the connecting rods and the gear wheels and in worst case broken pistons.

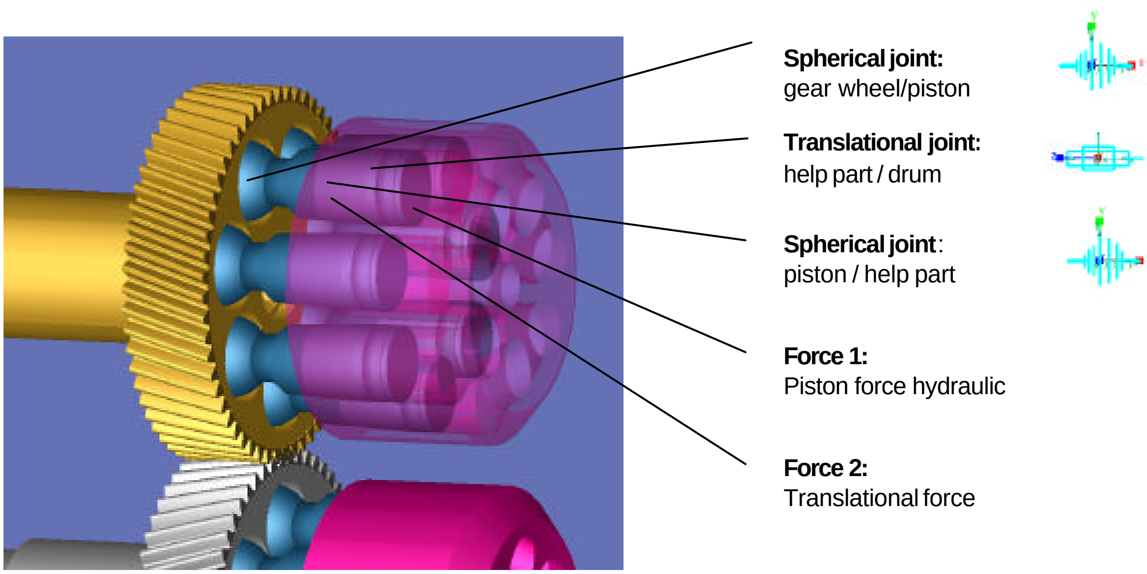
**ADAMS Model of double pump bent axis design**

The reason for this simulation was to simulate the dynamic load for the gear in a critical load condition.

The idea was to use the 3D-contact model available in ADAMS 12.0 for simulating gear and piston contact model. First simulation show that this works in principle, but the calculation time increased.
That means that the 3D-contact model will work at a very low speed but not at 2000 rpm, which is the normal speed of the engine. So another simplified model had to be created to simulate the pump system.

**Bearings and forces of each piston**

The bearing of the ADAMS-Model are close to the real bearing in the model except a help part of the piston.




**Pic. 4: Bearing and force model axial piston pump of bent axis design**

The help part added to the piston is necessary, because the piston has a spherical bearing but is also connected with the rotating drum.

**Contact forces models**

The simulation of the contact forces is based on 3D spring model with clearance. The distance between help part and the centre marker of the piston is a dimension of the deflection of the piston. The clearance of the piston is realised by a half sine function so that the translational force is faded in depending from the distance.

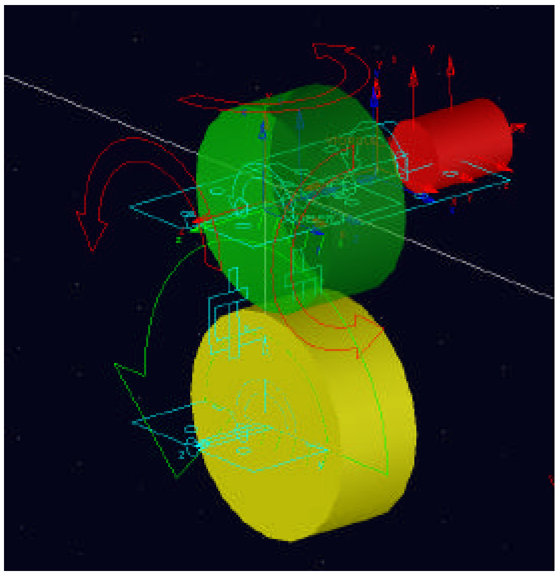
$$F_{Trans} = c_{Mat} \cdot \left(dm(M_1, M_2) - x_{clear}\right) \cdot hav\sin\left(x_{clear}, 0.0, x_{clear\,max.}, 1.0\right)$$

$$dm = \sqrt{dx^2 + dy^2 + dz^2}$$

The result is that the piston can move without translational force when **dm** is between 0.0 and $x_{clear}$.

### Stiffness and clearance of gear and diesel clutch

The same principle was used to simulate the elasticity and clearance of the gear. The drive shaft gear is connected by a coupler to the rotational joint of a help part.



**Pic. 5: Sample model of stiffness and clutch**

This is connected by a torque vector to the output shaft, which has a joint connection to the ground part. The torque vector consists only of one component in rotation direction of the gear and has basically the same structure like the translation force but an additional part for the damping is used.

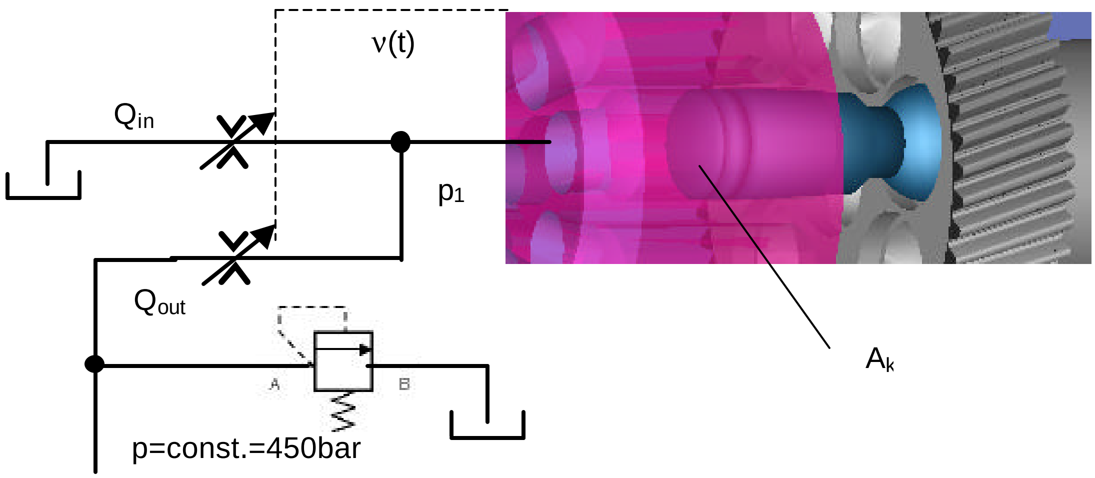
$$M_T = [c_T(az(M_1, M_2) - z_{clear}) + d_T \cdot wz(M_1, M_2)] \cdot havsin(z_{clear}, 0.0, z_{clearmax.}, 1.0)$$

$C_T$ – Stiffness gear
$d_T$ – Damping
$M_{1,2}$ – Marker 1,2

The diesel engine is connected by a clutch with the drive shaft of the gear, which has a nonlinear stiffness. The model contains a standard torsion spring which allows to modify in the stiffness as a spline function depending from the deformation.

## Piston force model and hydraulic model

During the rotation of the pump with its seven piston they are cyclic connected to high pressure and low pressure side depending from the openings of the control plate.




**Pic. 6: Hydraulic model of single piston**

$$\dot{p}_{1..7} = \frac{E_{Oil}}{V(x)} \cdot \left[Q_{i\_in} - Q_{i\_out}\right] - A_k \cdot \dot{x}_k - k_L \cdot \Delta p \qquad F_1 = p_1 \cdot A_k$$

$$Q_{in} = \alpha_D \cdot A(\varphi) \cdot \sqrt{\frac{2}{\rho}\left|p_V - p_1\right|} \cdot \mathrm{sign}(p_V - p_1)$$

$$Q_{out} = \alpha_D \cdot A(\varphi) \cdot \sqrt{\frac{2}{\rho}\left|p_1 - p_T\right|} \cdot \mathrm{sign}(p_1 - p_T)$$

$$Q_{ges} = \sum_{i=1}^{7} Q_i$$

The equations were modelled with the differential equation from ADAMS and state variables. The pump pressure is constant at 450bar. (pressure relief valve, which is modelled in ADAMS, here constant pressure). This model is realised for all 7 pistons and the result is the total flow.

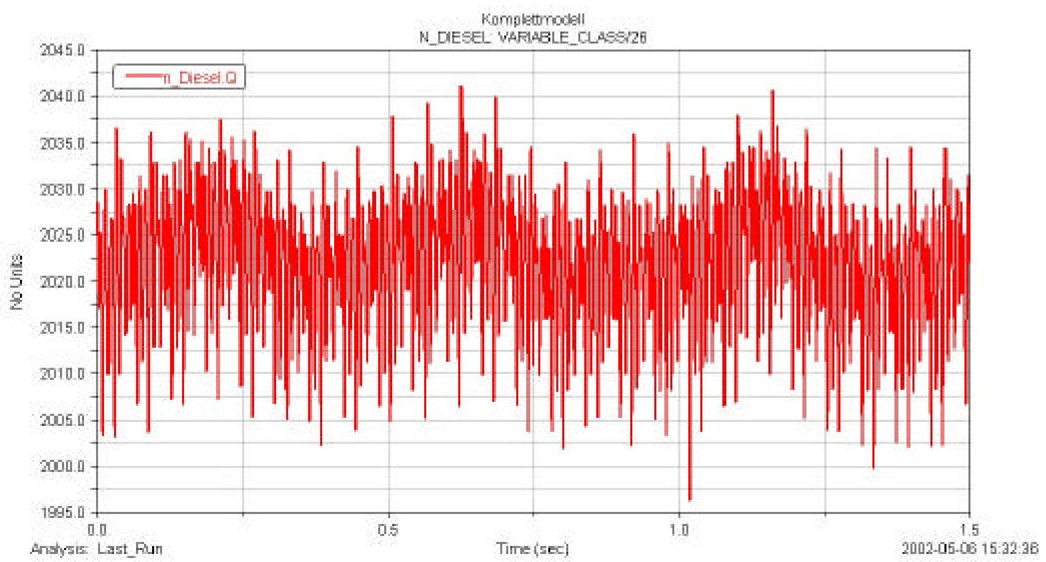
For simulating more complex load conditions or a more complex hydraulic circuit to simulate the whole excavator it is better to simulate the hydraulic circuit with an external simulation program based on co-simulation.

The reason for that is that it is very complex to handle high order differential equation with the formula editor of ADAMS and a schematic orientated simulation program for hydraulic systems makes it easier, because only the interface has to be defined.

## Motions of system

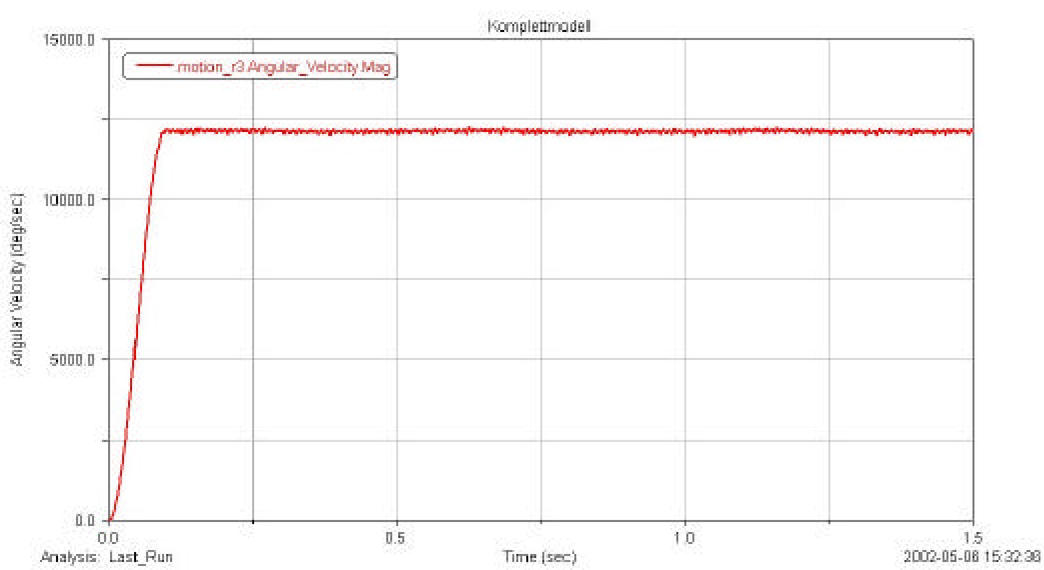
The pump is driven by a diesel engine at 2000 rpm. It is connected by a clutch to the drive shaft of the pump. The connection between both system is realised with torsion spring damper system with a non linear spring characteristic, which was measured in tests.

This was necessary because the diesel system contains its own oscillations which may influence the behaviour of the pump too. The feed back from the pump to the diesel engine is not considered here.




**Pic. 7: Speed diesel engine**

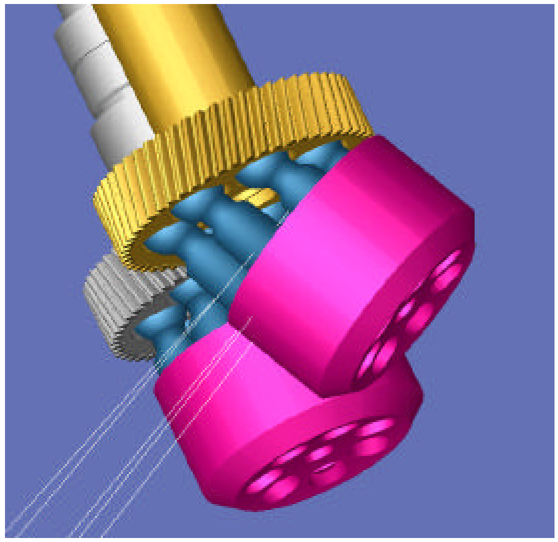
The start up of the system was realised with a fade in function in 0.25s to get soft start and no numerical problems.




**Pic. 8: Fade in speed diesel engine**

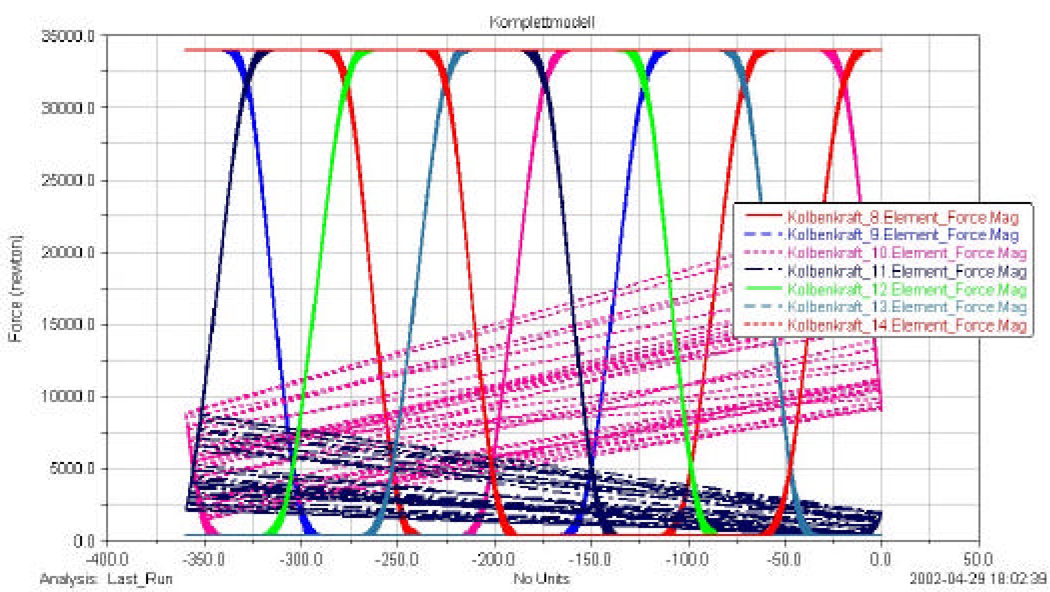
## Results of dynamic simulation

Experiments show that the critical load condition is, when one pump delivers maximum flow at maximum pressure and the other one is in zero position Q=0.0 L/min.



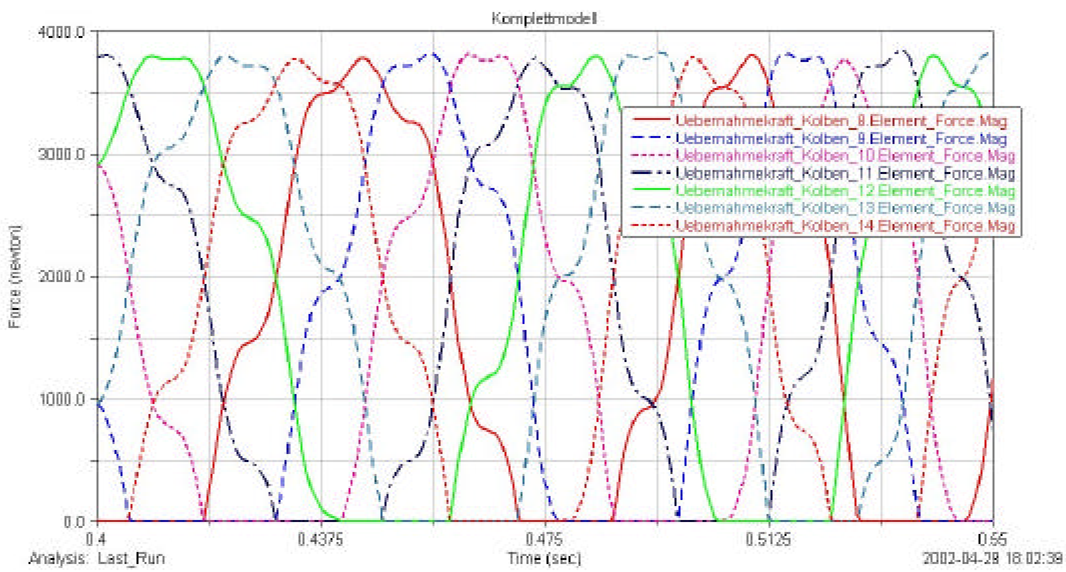
**Pic. 9: ADAMS model of axial piston pump**

The following diagram shows axial forces versus the rotation angle (ν) of the drum. The curves are depending from the opening and closing point of the control plate.



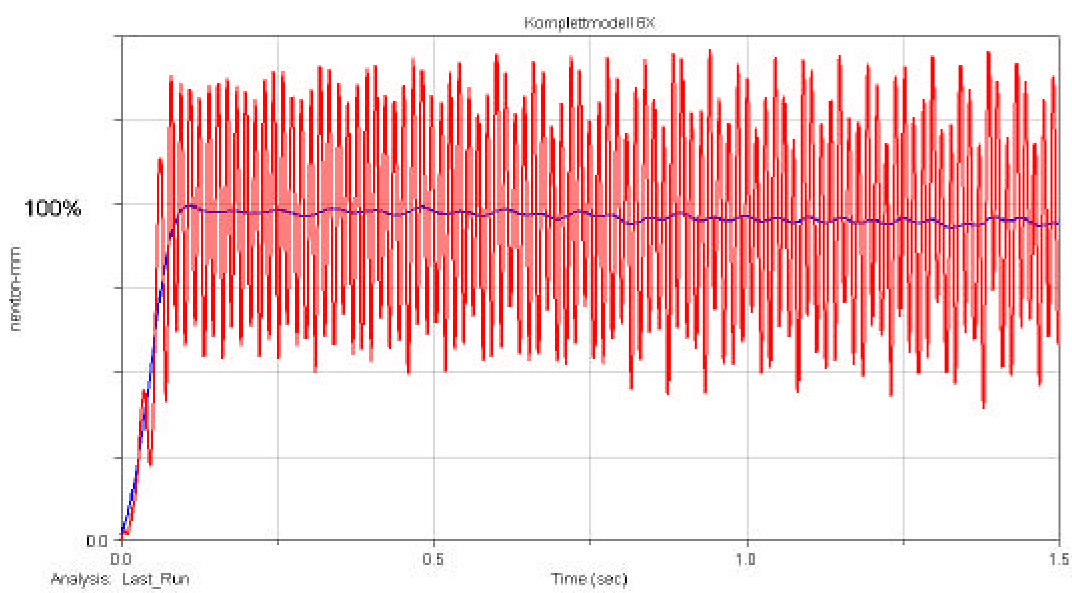

**Pic. 10: Axial forces of pistons**

The next diagram shows the “take over forces” of the piston, which move the drum. Four or five pistons are bearing the radial main force to transport the drum. The oscillation show the interaction between the pistons and the drum and this is the reason for possible torsional oscillation of drum and pistons.




**Pic. 12: Take over forces pistons**

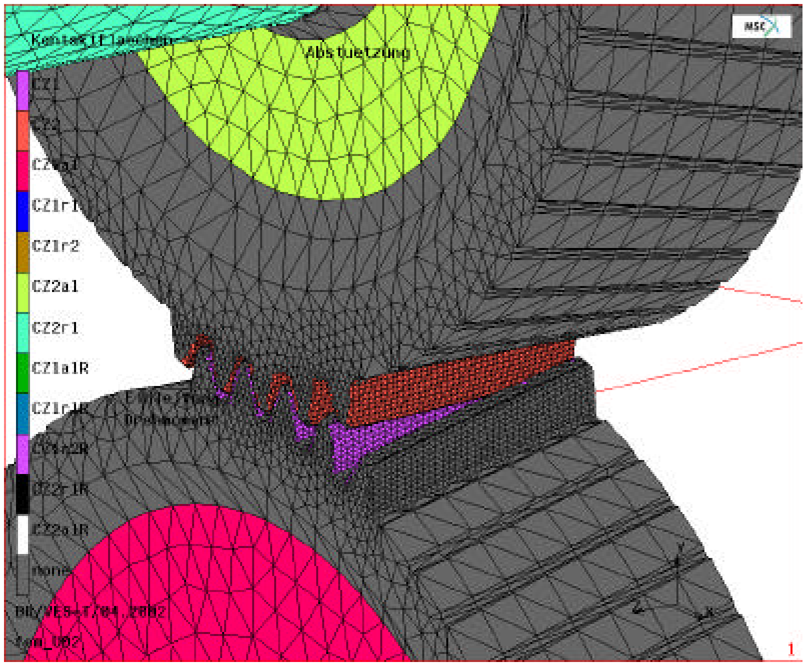
The next picture show the dynamic torque of the drive shaft.




**Pic. 13: Dynamic torque of the drive shaft**

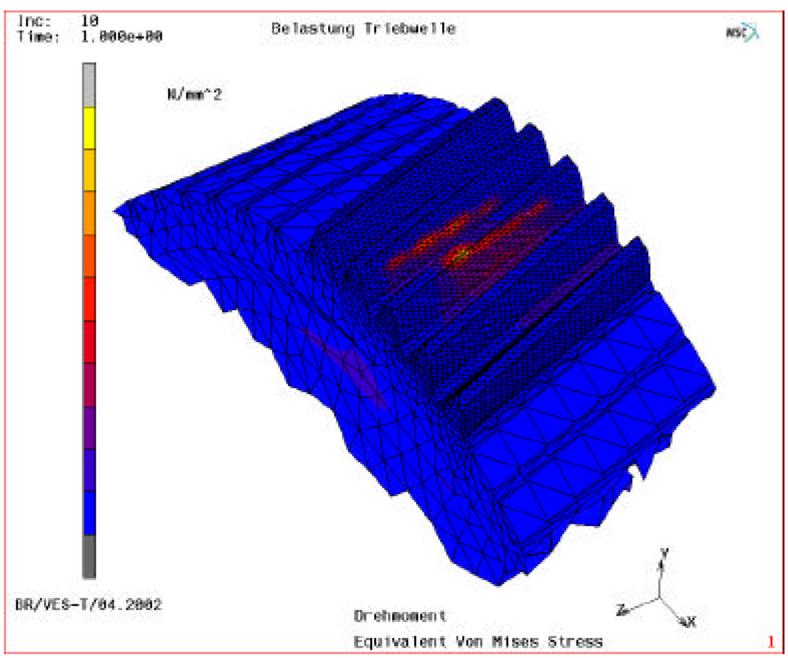
The normal static load of the drive shaft is about 100%. The rest is the result of the dynamic forces of the cinematic and hydraulic system of the pumps.

The result show that the dynamic load is higher that expected and these results are used to make a FEM calculation to verify the maximum stress inside the gear.




**Pic. 14: FEM contact model of axial gear pump**

The FEM calculation showed the influence of the higher dynamic load to the maximum stress.




**Pic. 15: FEM stress of gear**

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## Summary

The analyses of double pump of bent axis design show how ADAMS can be used the simulate the load of a very complex multi body system including subsystems. The result is a dynamic load of the system, which can be used in FEM analyses as shown to optimise the gear wheels, reducing stress inside to get a maximum of life time of the pump.

The next steps are to simulate a more complex hydraulic system to simulate different systems at the high pressure side e.g. axis motor, cylinders and other actuators as shown in picture 1.

These additional subsystems (cylinders, rotary drives, load sensing systems) will not be simulated with the equation builder but with an interface based on co-simulation between MOSIHS (**M**odular **S**imulation **H**ydrau**l**ic **S**ystems) and ADAMS 12.0. This allows to simulate hydraulics and multibody system with more flexibility, because only the interface of forces, positions, velocity has to be defined.

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