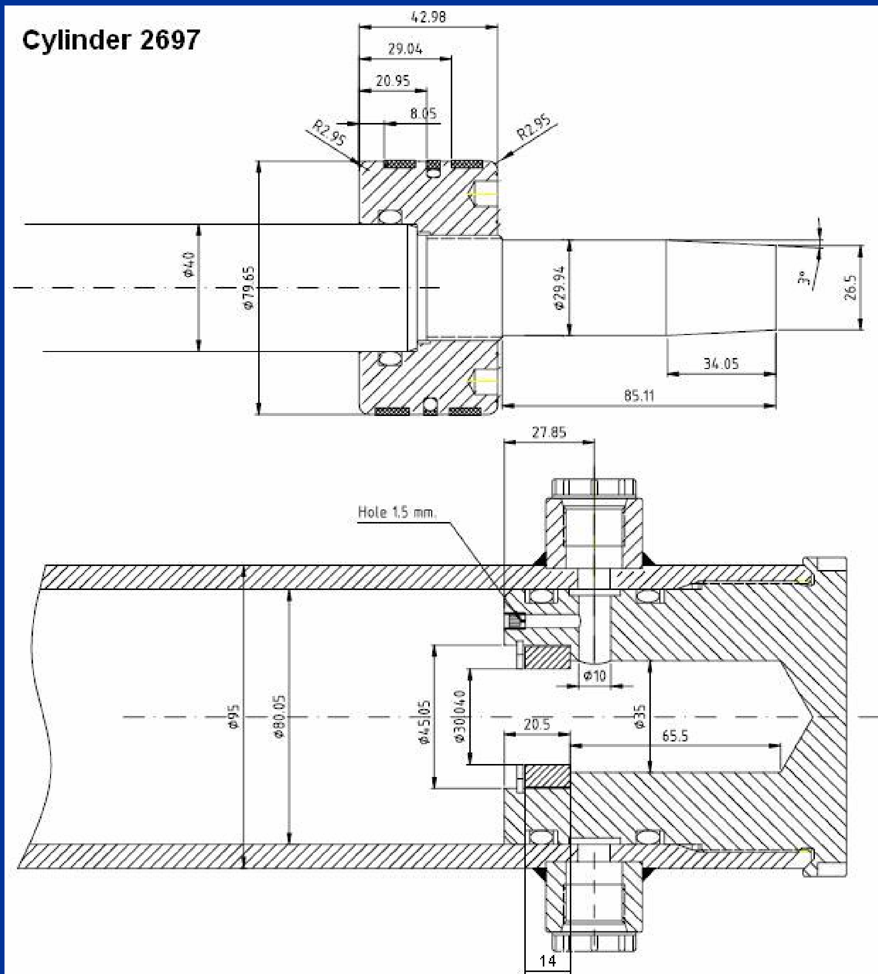
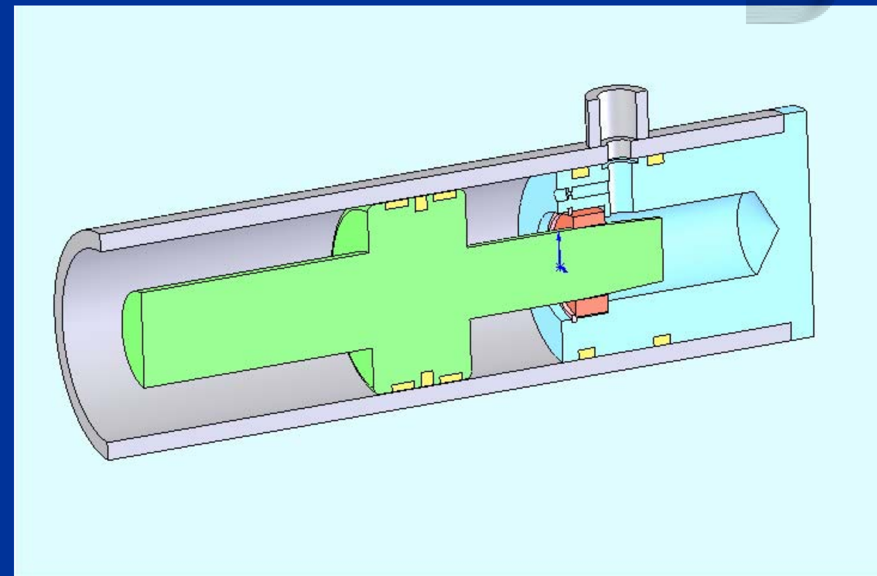
IMAMOTER – C.N.R.

Objectives

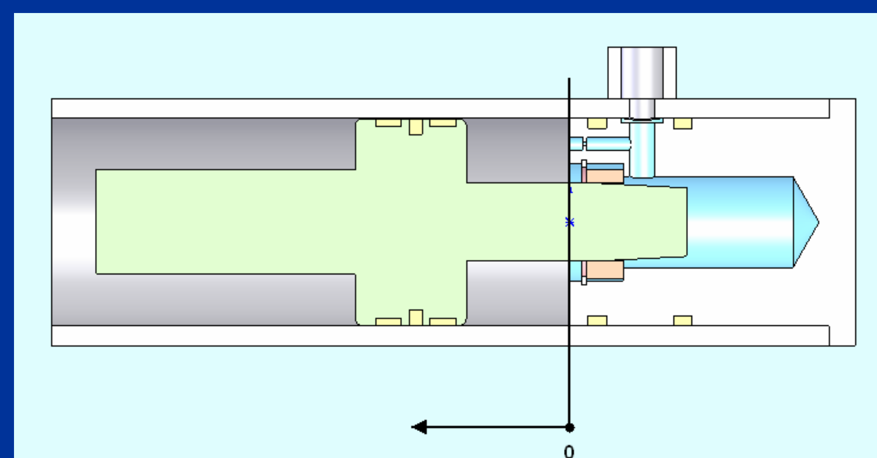
- Investigation of the third cushioning geometry
- Comparison with experimental data for third reporting period
- Deeper investigation of radial forces effect on geometry I (new)
- Guidelines for next cushioning design

Third Geometry

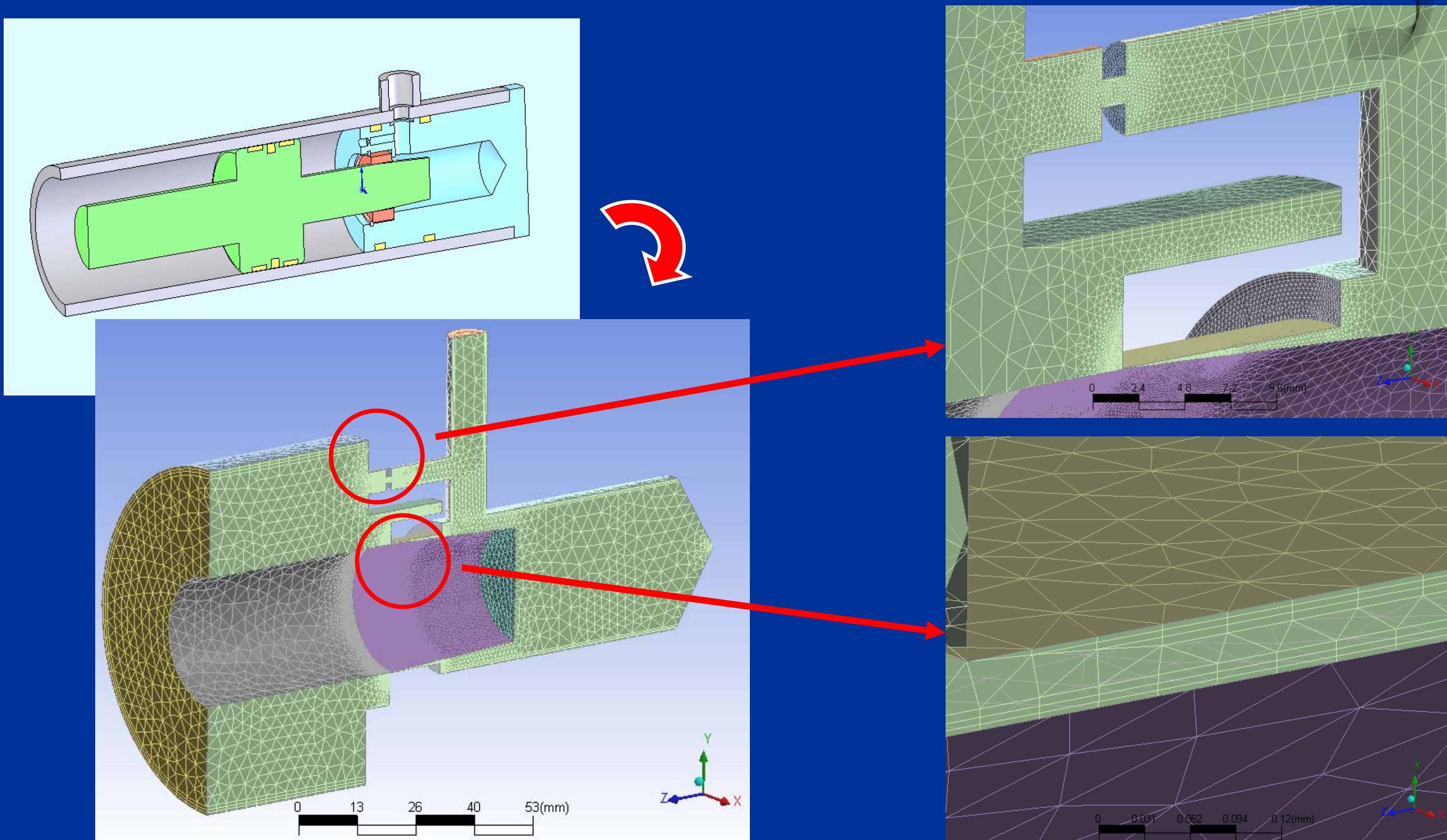




Reference system



CFD Mesh

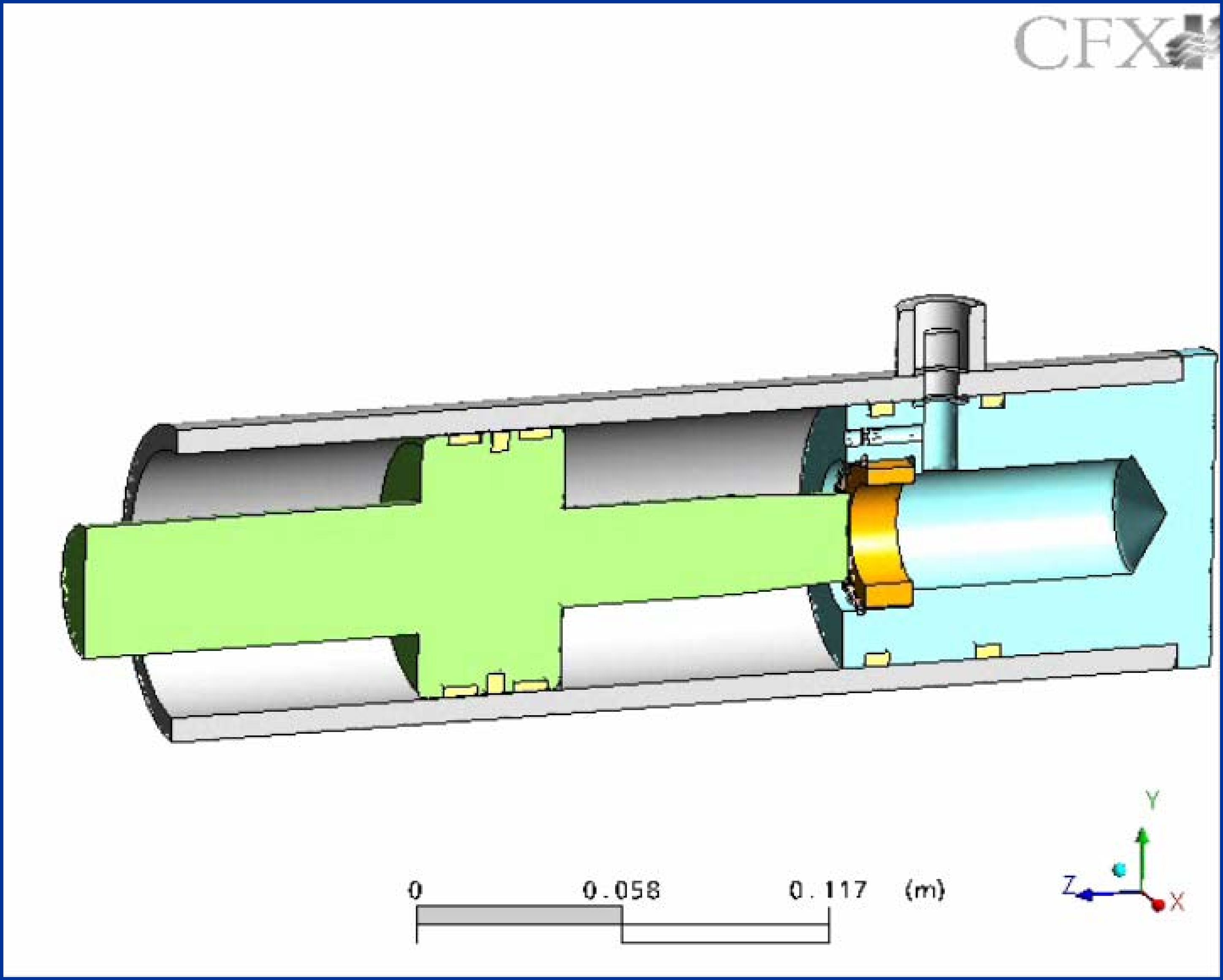


Fine resolution of the throttler, 8 elements in the gap

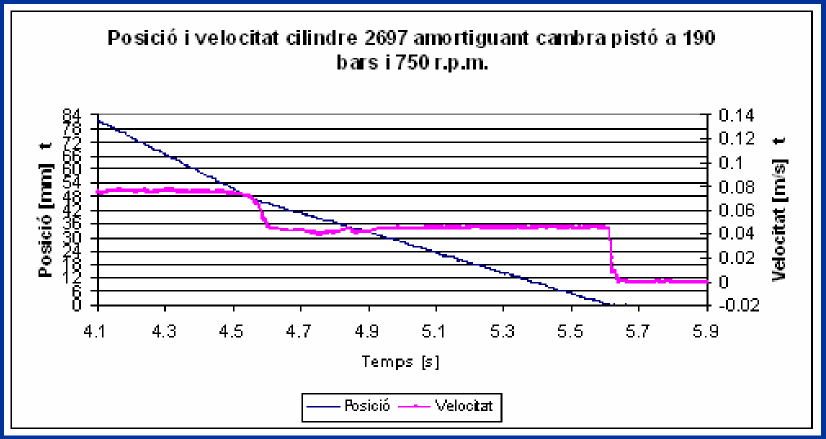
CFD analysis settings

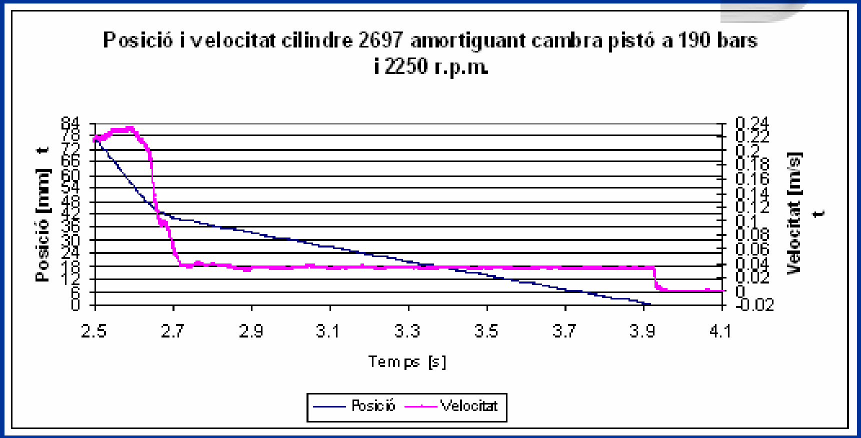
- 2 piston speed: 0.075 m/s and 0.22 m/s
- Many positions of analysis: 15 in the first case, 9 in the second (dependent by supply circuit relief valve setting (190 bar)
- Outlet at bench test discharge port pressure: 3.5 bar in the first case, 30 in the second
- Oil: Mobil DTE 32 and 46 Excel Series
- Turbulence models: k-epsilon, k-omega, BSL Reynolds stress
- Target average normal residual: 10^{-4}

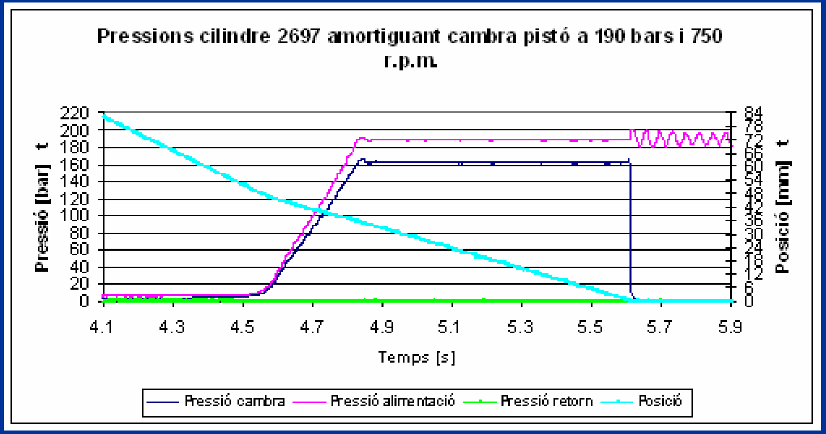
Example of results

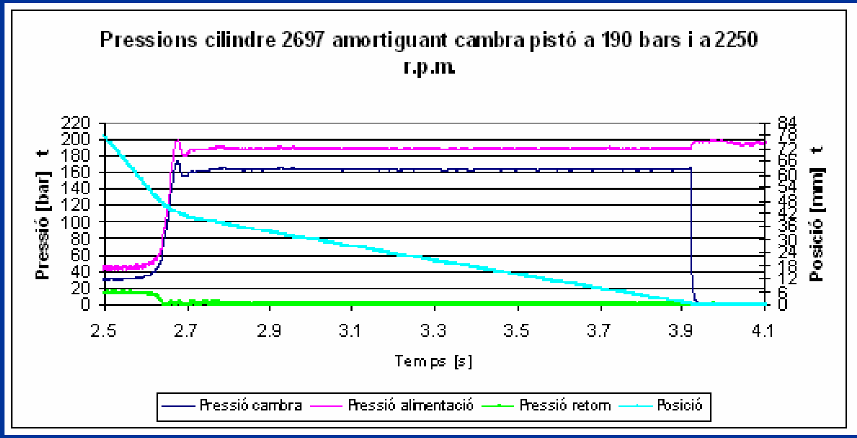


Bench results





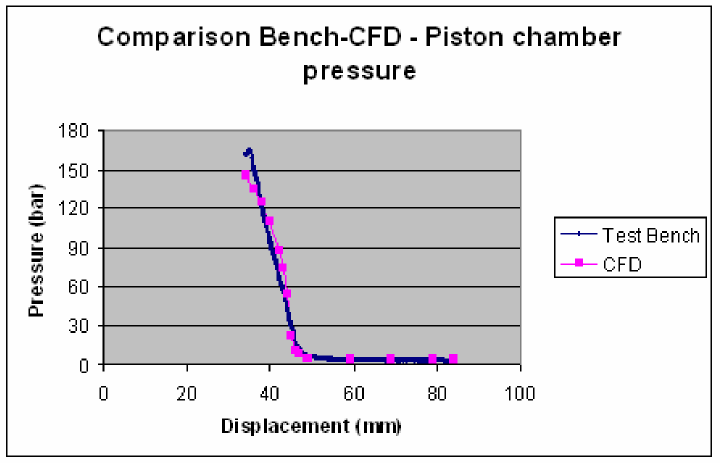


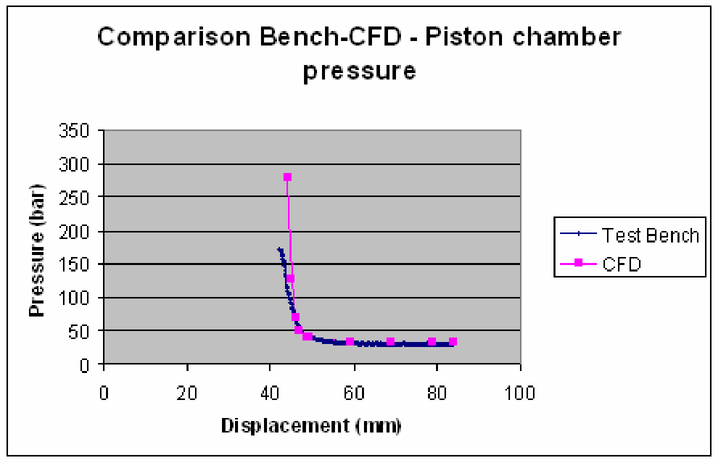


Starting piston speed 0,075 m/s

Starting piston speed 0,22 m/s

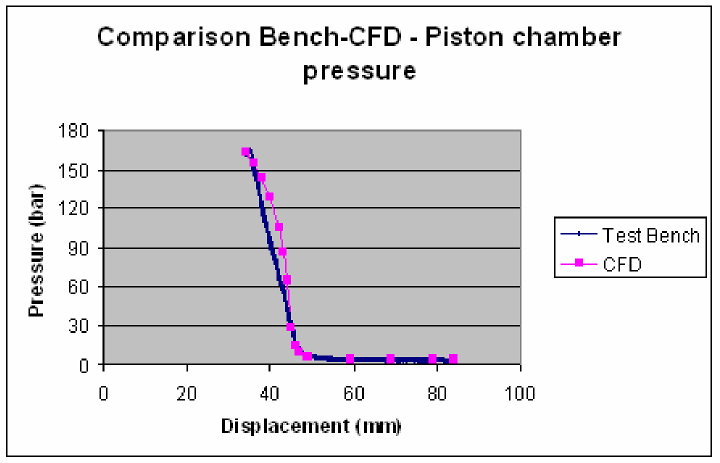
Comparison between bench and analysis

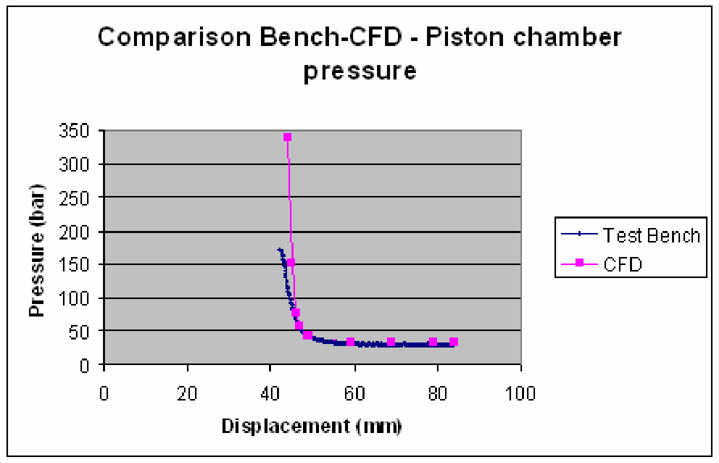




DTE 32

DTE 46





Piston speed: 0.075 m/s

Piston speed: 0.22 m/s

Comparison between bench and analysis

Reading figures from right to left according to the stroke:

- Starting pressure is the same as expected because CFD outlet pressure was set at the starting piston chamber value of tests in order to consider pressure losses between cylinder and tank in simulations.

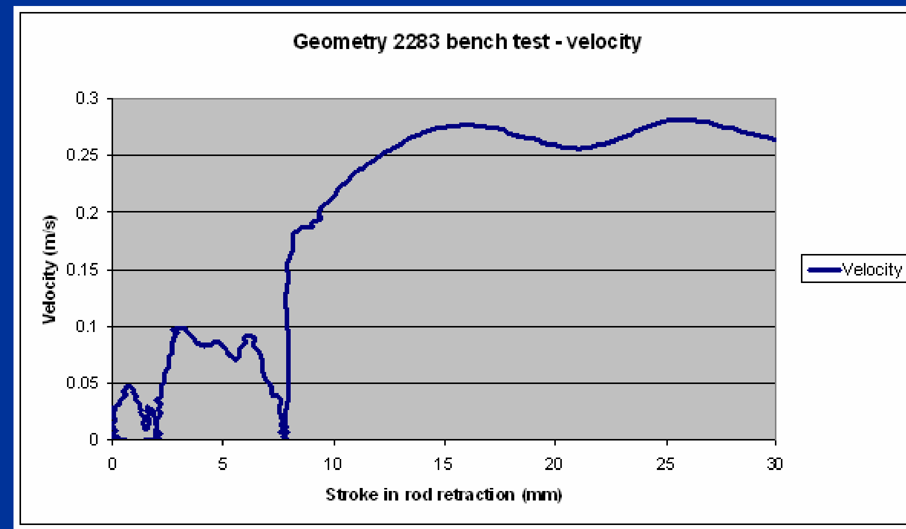
- At 46-47 mm the pressure increasing begins, trends have a very good agreement. Analysis at 45, 46, 47 and 49 mm follow the path of bench tests reproducing the right pressure gradient and the variation of the gradient. Then the CFD pressure increasing seems to be more rough than bench tests, probably the real case has a time lag due to the fluid compressibility and dynamic response of the cylinder

- At the lower speed (0.075 m/s), bench delay is briefly recovered and final values of trends are very close. After 34 mm relief valve opens also at the lower piston speed, so a direct comparison between CFD and bench test is not possible further on.

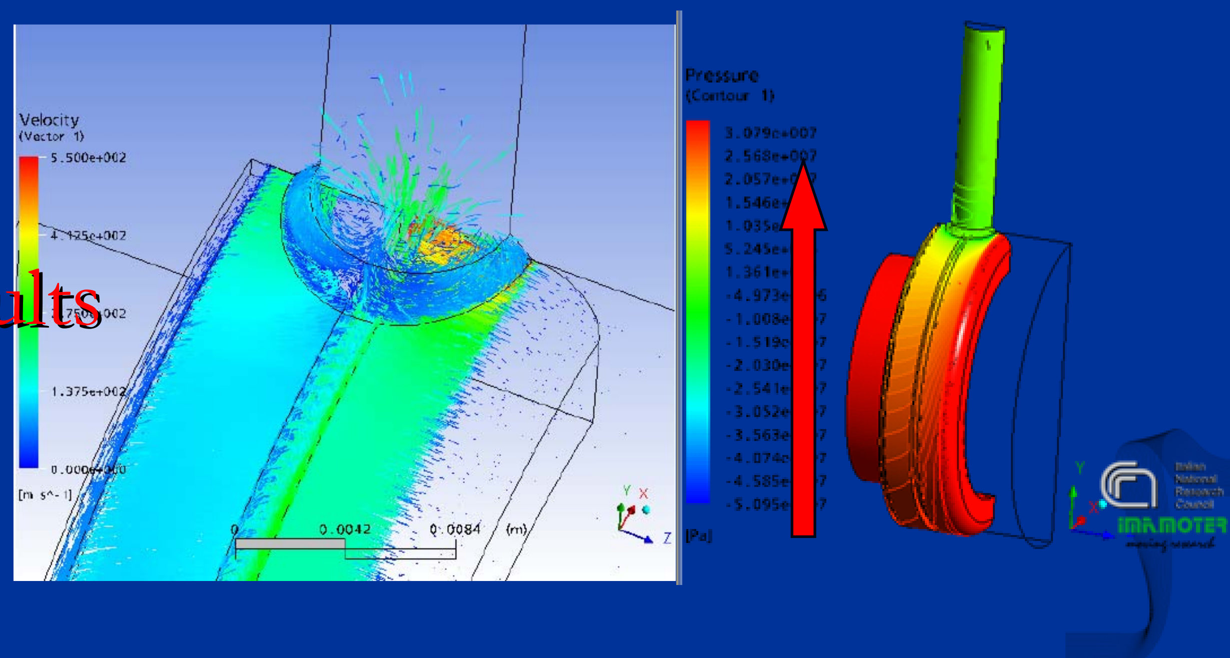
- At the higher speed (0.22 m/s) the relief valve opening is reached between 44 and 45 mm, so the description of phenomena after this point is not useful.

Radial-force-induced oscillations

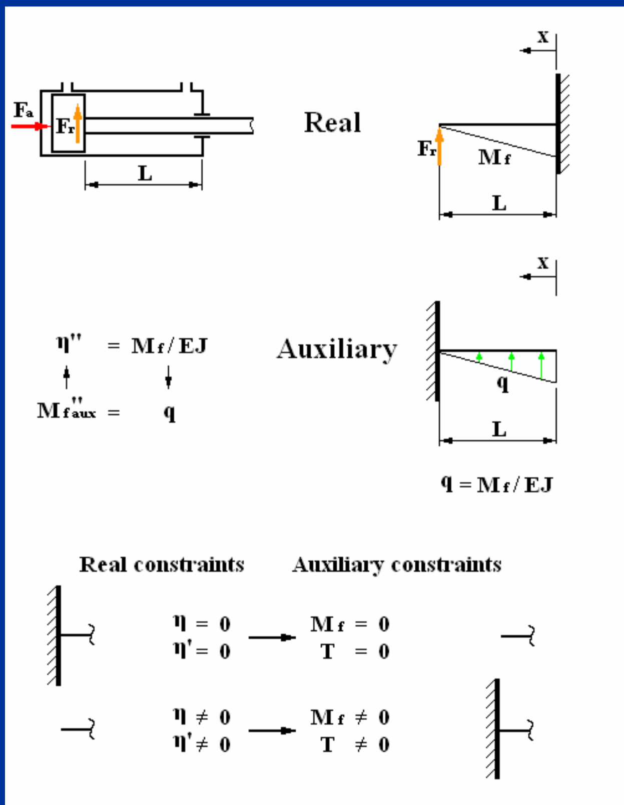
- Experimental Evidence



- Numerical results



Theoretical investigation



Flexural moment in real beam:

$$Mf = Fr \cdot (L - x)$$

Auxiliary distribuited load:

$$q = Mf / EJ = Fr \cdot (L - x) / (EJ)$$

Auxiliary flexing moment = displacement in real beam

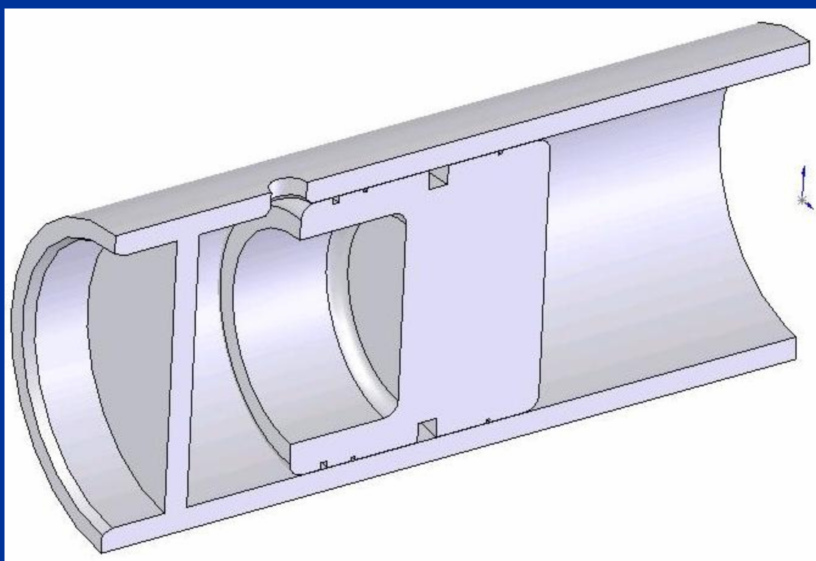
$$\mu = Mfaux = \int_0^x (q \cdot x \cdot dx) = \int_0^x \frac{Fr \cdot (L - x) \cdot x}{EJ} \cdot dx = \frac{Fr}{EJ} \cdot \left(\int_0^x L \cdot x \cdot dx - \int_0^x x^2 \cdot dx \right) = \frac{F}{EJ} \cdot \left(\frac{1}{2} L \cdot x^2 - \frac{1}{3} \cdot x^3 \right)$$

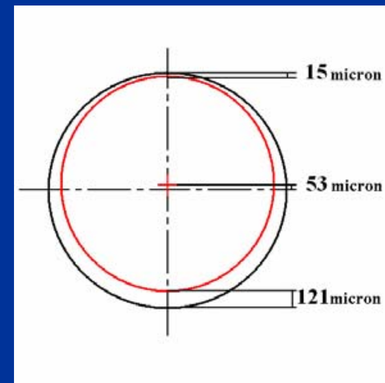
If x=L

$$\mu = \frac{1}{6} \cdot \frac{Fr}{EJ} \cdot L^3$$

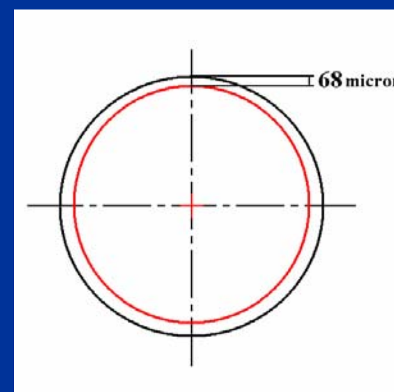
= 560 micron (gap 68 micron)

CFD analysis with eccentric piston head

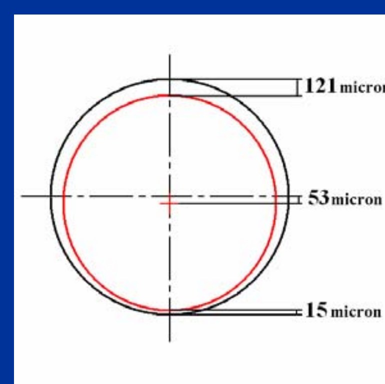




Up: 53 micron of eccentricity. Gap of 15 micron up, 121 down



Centre: gap 68 micron all around (max gap according to tollerance)



Down: 53 micron of eccentricity. Gap of 121 micron up, 15 down

Kind of meshes

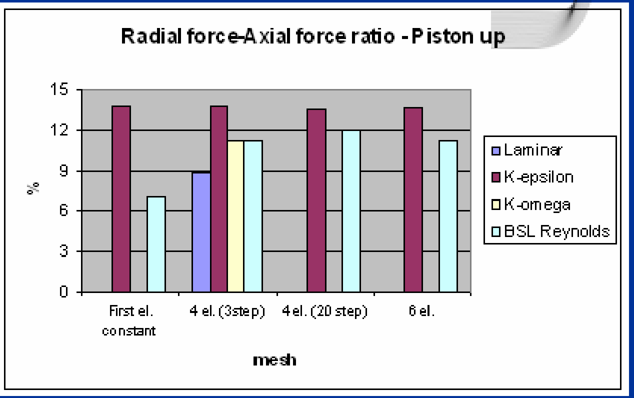
KIND of Mesh	Position	Elements
1) 4 elements into the gap	Centre	1334206
2) 8 elements into the gap	Centre	3787816
3) First element thickness constant all around	Up	6874326
4) First element thickness constant all around	Down	8041672
5) 4 el. into the gap, first el. thickn. variable (3 steps)	Up	6931733
6) 4 el. into the gap, first el. thickn. variable (3 steps)	Down	7916568
7) 4 el. into the gap, first el. thickn. variable (20 steps)	Up	7903211
8) 4 el. into the gap, first el. thickn. variable (20 steps)	Down	8010524
9) 6 el. into the gap, first el. thickn. variable (20 steps)	Up	10104916

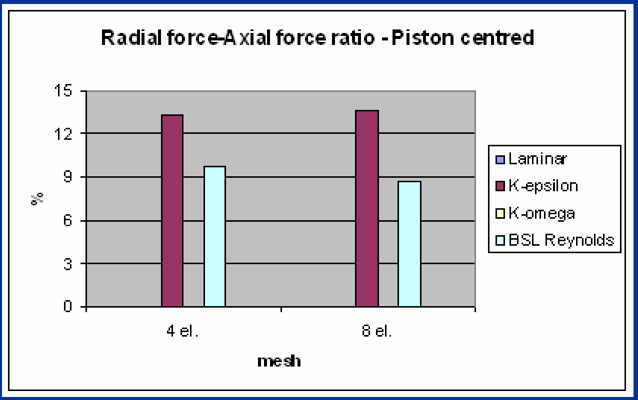
Turbulence models used

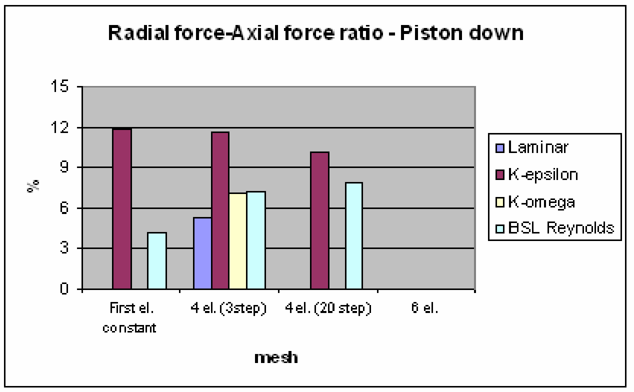
KIND of Mesh	Position	Turbulence models
1) 4 elements	Centre	K-epsilon, BSL Reynolds stress
2) 8 elements	Centre	K-epsilon, BSL Reynolds stress
3) First el. thickn. constant	Up	K-epsilon, BSL Reynolds stress
4) First el. thickn. constant	Down	K-epsilon, BSL Reynolds stress
5) 4 el. first el. thickn. variable (3 steps)	Up	Laminar, k-epsilon, k-omega, BSL Reynolds stress
6) 4 el. first el. thickn. variable (3 steps)	Down	Laminar, k-epsilon, k-omega, BSL Reynolds stress
7) 4 el. first el. thickn. variable (20 steps)	Up	K-epsilon, BSL Reynolds stress
8) 4 el. first el. thickn. variable (20 steps)	Down	K-epsilon, BSL Reynolds stress
9) 6 el. first el. thickn. variable (20 steps)	Up	K-epsilon, BSL Reynolds stress

Results: measured radial forces

Radial force / Axial force (%)					
Piston position	Turbulence model				
	Kind of mesh	laminar	k-epsilon	k-omega	BSL Restress
Up	First el. constant		14		7
	4 el. (3step)	9	14	11	11
	4 el. (20 step)		13		12
	6 el.		14		11
Centre	4 el.		13		10
	8 el.		14		9
Down	First el. constant		12		4
	4 el. (3step)	5	12	7	7
	4 el. (20 step)		10		8
	6 el.				







- Ratio of 0.13 between radial and axial force is confirmed in k-epsilon (much easier to handle)
- K-ω and BSL Reynolds model give only marginal improvement in radial force computation
- Ratio is not dependent on piston position

Conclusions first geometry - rod bending

•There is an important rod bending due by radial force, so resonance could start. It is shown through calculation but an important consideration is introduced: piston touches the inner wall of the cylinder only for the action of the discharge hole on its side at the piston chamber pressure (142.5 bar) able to open the supply circuit relief valve.

•changes in cylinder geometry due by rod bending create an important pressure variation (about a ratio of 2 comparing centre position to up and down), emphasizing radial forces and rod instability.

When piston movement closes the discharge hole, pressure increases and radial force displaces the rod amplyfying the increase. During this time piston stops as a consequence of the cushioning chamber pressure. Flow and pressure decrease, combined with the piston dead weight, can explain the dynamics of the oscillation, when the adjoint mass effect is accounted for.

Full project summary
IMAMOTER – C.N.R.

Italian National Research Council
IMAMOTER moving research

Sixth Framework Programme
2002 - 2006

PRO HIPP

Objectives

- Three geometries cushioning investigation (CFD mainly)
- IMAMOTER active Tasks
  - T2.2 (fluid dynamic design)
  - T8.1 T9.1 (Best practice rules and exhibitions)
  - WP12-14 (Training and dissemination)
- Active partnership with: Roquet SA, UPC-Labson, CIMNE (and information exchange with many others)

Methods

- Non linear dynamic analysis
- CFD investigation of cushioning (more than 1000 different computational runs)
- 3 MSc Thesis, exchange of researchers with UPC-Labson
- Benchmark on FEM analysis of cylinder oil port stress concentration

Main achievements

- Full review of patents and literature on cushioning
- Full simulation of fluid flow in cushioning for three geometries (reference configurations)
- Non-linear simulation of cushioning dynamics
- Predictive methods for cushioning performance
- Effect of radial forces on cushioning devices
- Guidelines for CFD and GDSS concurrent use in cylinder design
- Benchmark on cylinder oil port stress concentration

State of the art improvement

Topic	State of the art	PROHIPP achievement
Cushioning design	Empiric approach based on nomograms and consolidated practice	CFD and GDSS use with rating of different methods effectiveness
Disturbances evaluation in cushioning	N.A.	Radial forces effect, oscillatory behaviour comprehension
Cumulative damage estimation in cushioning	Empirical	Analytical estimate based on forces on sealings and structure
Cushioning device performance evaluation	Experimental, in-service	Virtual mockup, predictive