A MINIATURE NEEDLE VALVE

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ABSTRACT

It has been suggested that in order to obtain "the perfect valve" one should use large number of parallel connected, just one size of on/off-valves, to control the flow rate. With commercially available hydraulic on/off-valves, this is not practically reasonable because the number of valves needed is from tens to few hundred and the use of commercial on/off-valves leads to too big valve packages. In this paper a simple unidirectional spring return needle valve is miniaturized to size small enough to be used in digital flow control units consisting several dozens of valves. The most important design parameters are small size, operation on industrial hydraulic pressure level, fast response time and reasonable flow rate. A major problem in commercial valves is the size of the solenoid actuator thus the design includes both a hydraulic part and a solenoid actuator for it. Simulations, calculations, measurements and construction of the valve are presented. Capabilities of designed valve are discussed.

KEYWORDS: Digital hydraulics, miniaturization, on/off-valve

1. INTRODUCTION

In search for a way to "a perfect valve" solution can searched using the general scaling laws of physics [1]. These derivations, in case of fluid power, are also presented in [2]. Requirements for a good on/off-valve are small size, fast response, suitable flow rate and low energy consumption. High operating frequency is also important in some cases. It is clear, that smaller valves are faster and require less energy per actuation than bigger, because their actuation requires shorter transition distance and moving parts are lighter. As a negative thing, if one big flow channel is replaced with several very small, the total area of flow channels must increase if same flow rate is to be remained. While the shorter actuation distance and lighter parts should lead to smaller actuation energies and faster response time, increase of total opening area of a seat valves result bigger actuation forces needed to open the valve. We currently assume that practically good diameter of orifice of the one valve is in range of from 0.3 to 1.0 mm. A prototype of a small hydraulic on/off-valve was done to give some prove of this.

In this paper a miniature a needle valve, small enough to be used parallel connected as discrete control notch consisting only one size of on/off-valves, is designed and its performance is measured. Design includes CFD-calculations, thermal modeling and both static and dynamic electromagnetic modeling. Design methods are described in following chapters. Prototype is constructed and measured and the validity of the pre-prototype simulations and calculation results are checked. The new prototype is compared to our previous ones in the discussion part.

2. DESING OF THE VALVE

2.1. General design parameters

Our goal is to create a basic element for a digital replacement of a proportional valve and keep the size on the scale. Size of a typical sandwich-block of pilot operated mobile proportional valve with pressure compensators (e.g. BoschRexroth M4) is coarsely about 150x150x50 mm. In order to create a “proportional” valve from equally sized on/off-valves, one needs about 32 valves per control notch to reach somewhat similar controllability (based on practical experience of senior authors). In a four way valve there are four control notches so total number of on/off-valves in this kind of valve is 128. Square root of 128 is 11.3 and this number of valves should be installed in one row if they were in square formation. If we divide 150 mm for 11.3 valves we get area of 13.3x13.3 mm square for each valve. If we had cylindrical valve of Ø 10 mm and at least one millimeter free space around the valve in every direction we get diameter of 12 mm which fits on square of 13.3mm long edges. Therefore valve diameter of 10 mm was used as a design value. Length of the valve was limited to be less than 50 mm.

Small valve has to be simple in order to be economically feasible in mass production. Thus, the design was decided to be unidirectional spring return needle valve. Bistable actuation and bi-directional functionality would be beneficial but it increases complexity and were not used in the design. Previous prototypes include a force balanced poppet [3], symmetric and bistable poppet [4], bistable actuator with the balanced poppet [5] and hammer actuator for the balanced poppet [6] but those features are difficult to realize in very small scales. Beneficial they are of course, but they tend to cause requirement of tight tolerances and more complicated and expensive parts. If the valve is small enough and bi-directional flow is desired, two unidirectional valves can be put together to work as one bi-directional. Also if the energy needed to maintain the forced position is small enough, bistability doesn’t give much extra benefit.

The return spring of the valve was calculated to be strong enough to close the valve in two milliseconds which was also the designed opening time of the valve. Diameter of the valve orifice was decided to be as big as could be opened fast enough with the designed electromagnetic actuator. Since our highest design pressure was 20 MPa, the orifice must be on sub-millimeter scale. The space for the solenoid had to be maximized in order to maximize the force, so no separate pressure chamber was designed but the coil itself was made to be load bearing structure by molding it in and out to epoxy glue. Epoxy-copper-wire-composite should be strong enough to hold the pressure forces but in order to get better magnetic circuit the whole valve were installed inside a steel block. This however decreases heat dissipation from coil but this could be helped by choosing the most heat conducting epoxy on the market and using silicone oil as heat transfer agent in the small gap between coil and block. The coil was dry to avoid problems in wire throughputs from pressurized chamber but it led to cut of magnetic circuit which decreases the forces generated by solenoid.

Shortly, the design was max. Ø 10 mm outer diameter, spring return, needle valve, with actuation time of two milliseconds and flow rate of about Ø 0.5 mm orifice. Construction is presented later after electromagnetics influencing to design is presented.

2.2. CFD Calculations

In order to maximize the flow rate while minimizing the desired force needed to open the valve, some computational fluid dynamics were done. Problem to be solved by CFD was; of which needle angle causes the flow to saturate at least of lift and requires least force to move and stay in that position. Calculations were carried out by ANSYS/Fluent. Turbulence was modeled with *k-e* model and enhanced wall treatment was used. Turbulence parameters at the boundaries were approximated and no sensitivity analysis was made in this case. Boundaries were inlet and outlet pressures and pressure difference was 35 MPa, which was our very initial design pressure but it was relaxed to 20 MPa after some reconsiderations. New calculations were not made because actual numbers are not so significant but just shape of the curves. Problem was solved in single phase flow without any cavitation models. The mesh was created by automatic mesher but it was remeshed to finer near the orifice in order to ensure working wall function there. With purely automatic mesh it didn’t work properly. Problem was solved stationary in cases of several different needle angles and openings. Unexpected result was that pressure forces over the needle tip are not at maximum when the valve is completely closed but in the position the valve is just slightly open. This can be explained from surface plots of pressure in figure 1 where the zone of pressure drop is wider than the diameter of the valve orifice especially in case of the bluntest needle. In that case the pressure starts to drop far before the chamfer at the begin of the orifice. Also locally increased pressure at the tip of the needle in that case causes flow reduction. The results are shown in figure 2 and from them it can be seen that in case of sharp needles the flow saturates slowly as well slow is the disappearance of forces. On the other hand blunt needles cause very high forces at small openings and maximum flow is smaller than in case of medium sharp needles. From these result it seems that optimal needle angle is 100°. Around this angle the flow gets its maximum value while the force reduces also early without causing serious peaks at small openings. Also it can be seen that with this needle, the flow saturates at the lift of 0.3 mm so it was decided to be transition distance of the needle in the prototype.

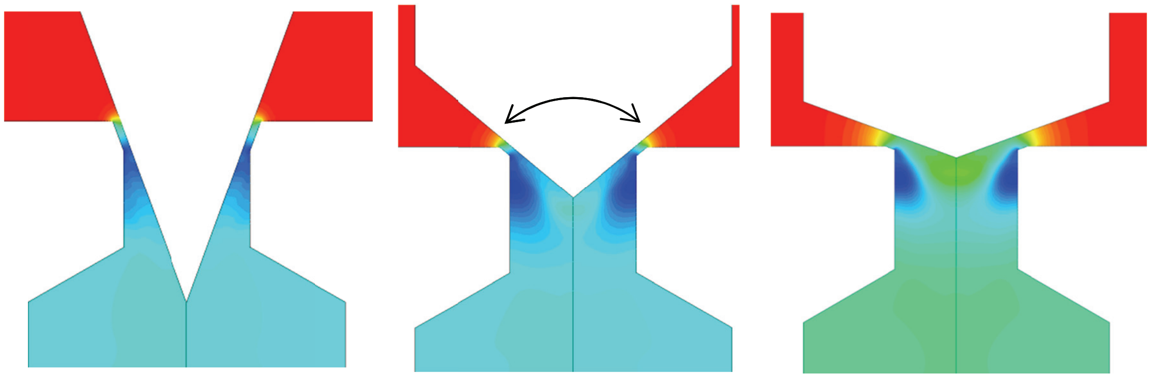

Figure 1. Three different needle angles. Sharp, medium and blunt. α-letter shows from where the needle angle was measured. Inlet pressure is 36 MPa and outlet pressure is 1MPa

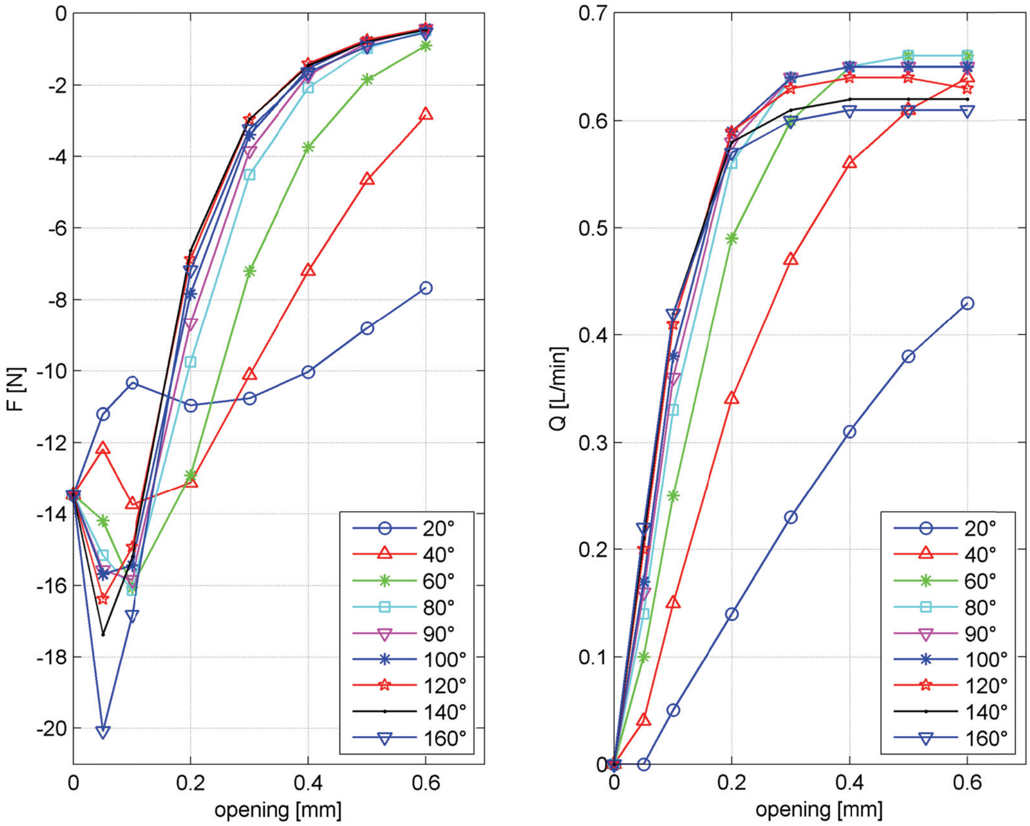

Figure 2. Required opening force and flow as function of opening at different needle angles in case of Ø 0.6 orifice and 35 MPa pressure differential. Angles from 80 to 120 are good. Very sharp and very blunt are not.

2.3. Modelling

The main idea was to make a simple needle valve that fulfils goals, with given limits, which were given at the beginning. Hydraulic modelling presents the hydraulic character of the valve. Electromagnetic modelling focused on the electromagnetic actuator and part of the manifold. Thermal modelling examined amount of the heat dissipation inside the valve. Working idea of this actuator is quite simple. The anchor moves when the coil is energized with a voltage. This movement will open hydraulic flow through the valve. Not until the voltage is removed the spring returns the anchor in its original place. The motion of the anchor is modelled to analyse response times and thermal losses. The heat dissipation of the coil is modelled with heat resistance network to predict highest applicable frequency size.

2.3.1. Static modelling

Static modelling including reluctance method [7] was used to estimate the construction of the actuator. It needs to produce a magnetic force within the physical limits given which is greater than resisting forces to be sure that the valve works. The structure of the coil is also modelled to be able to estimate the diameter of the copper wire.

The actuator consists of a coil, an iron yoke, an anchor and returning springs. In the figure 3 only the electromagnetically important parts are shown in two dimensions using axial symmetry. The magnetic circuit is formed by the iron yoke and the anchor. There is a gap between the anchor and the iron yoke above (later called: upper yoke), which makes the action of the valve possible. The valve opens when the anchor moves toward the upper yoke and the gap closes. Springs between the upper yoke and the anchor resist this movement.

The static modelling was done using Matlab and Opera-2D steady-state analysis. Figure 4 shows the results of the analysis, when the valve is driven with 12V assuming 80°C temperature. The force, exerted on the anchor depends on the magnitude of the current density and the gap. The closer the anchor and the upper yoke get the greater is the magnetic force. The sum of the resisting forces at the beginning when the gap has its maximum value is F_{res} = 6.3 N. The magnetic force needs to exceed this resisting force before the anchor moves.

Resisting forces are taken into account when the motion is modelled. The major forces are the pressure force, which is assumed simply to be constant along with the spring force. Spring force depends on the position of the anchor. There are also resisting forces following from the movement of the anchor: viscous friction and shear forces. The gravitational force is also taking into account. Both static and dynamic modelling is based on these resisting forces.

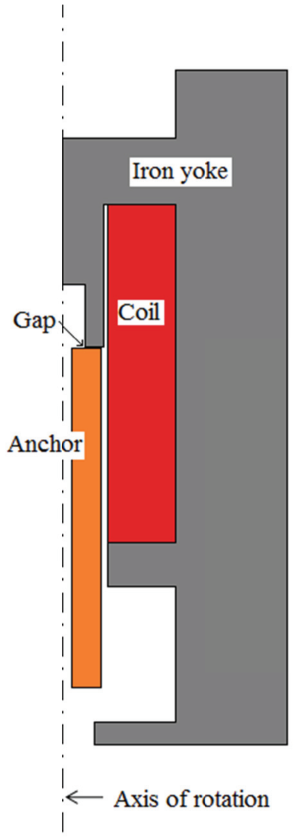

Figure 3. The geometry for electromagnetic modelling of the actuator.

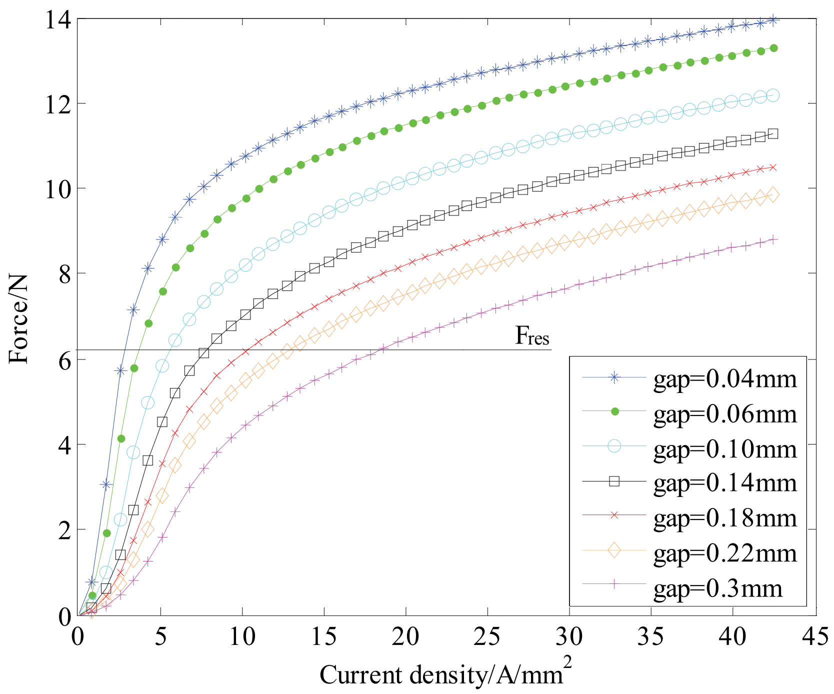

Figure 4. Steady state analysis of the magnetic force in the anchor as a function of the current density and the gap. F_{res} is the sum of the resisting forces.

2.3.2. *Thermal modelling*

One of the restricting things in designing the actuator is thermal losses. Thermal losses may cause the overheating of the coil and break down the valve. Thermal losses are caused by the copper losses inside the coil and the iron losses caused by electromagnetic induction and hysteresis. The analytic model of the heat flux from the coil to the surrounding material is essential. This model doesn't include radiation because the temperature differences are less than 100 °C degrees.

Using simple network of thermal resistance the heat flux from the coil to the air and the oil is model as described in figure 5. P_c is average thermal loss produced by the coil. The total thermal resistance is R_{air} from the coil to the air above the yoke. R_{oil} is the total thermal resistance from the coil to the oil under the yoke. T_{coil} is the temperature of the coil and T_{env} is the temperature of the air and oil. The temperature difference between the environment $T_{env} = 50°\text{C}$ and the coil $T_{coil} = 80°\text{C}$ is $\Delta T = 30K$.

The air and the oil around the valve are assumed to be in natural state without any forced flow. In the table 1 the values of the thermal resistance and received thermal losses are given. The values of convective heat transfer are taken from tables for free convection of gases and liquids [8]. Iron losses are not noticed in this model because of the size of the iron yoke and the used current frequency [9]. This thermal model is not exact but it will give a rough estimate of the operation point. It also gives an idea of the working frequency of the actuator.

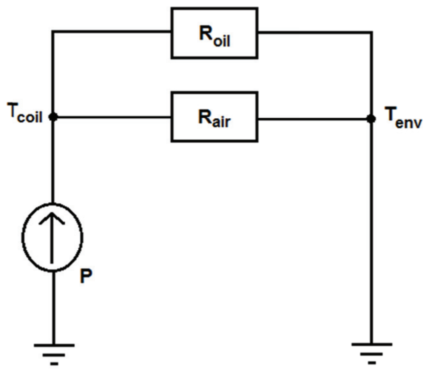

Figure 5. The thermal resistance network used in this work. $\boldsymbol{P_c}$ is thermal losses in the coil. $\boldsymbol{R_{oil}}$ is the thermal resistance from the coil to the hydraulic oil and $\boldsymbol{R_{air}}$ is to the surrounding air.

Table 1 The values of the network of thermal resistance.

	$\boldsymbol{\Delta T = 30K}$
$\boldsymbol{R_{air}}(W/K)$	874-75
$\boldsymbol{R_{oil}}\ (W/K)$	77-16
$\boldsymbol{P_c}\ (W)$	0-3.4

2.3.3. Dynamic modelling

Dynamic modelling gives more accurate results on the actuator taking into account electromagnetic induction which causes some changes in the coil and the actuator. The skin depth in the wire of the coil gives information of the current density. Its value is depended on the material properties of the wire but also on the frequency of the current. In this model the skin depth is larger than the used copper wire. The skin depth of the anchor describes how deep the magnetic field density is penetrating into the anchor. In the figure 6 is given the magnetic field amplitude (T) and the vector potential (Wb/m) when the anchor is in case (a) closed at the time 0.2ms, in case (b) still not moved yet at the time 1ms and in the case (c) completely opened at the time 2.2ms. The skin depth grows by the time. The motion of the anchor is model using Opera-2D linear motion analysis. Dynamic model takes into account movement of the anchor. When the anchor moves, changes the inductance of the coil effecting on the time constant of the coil.

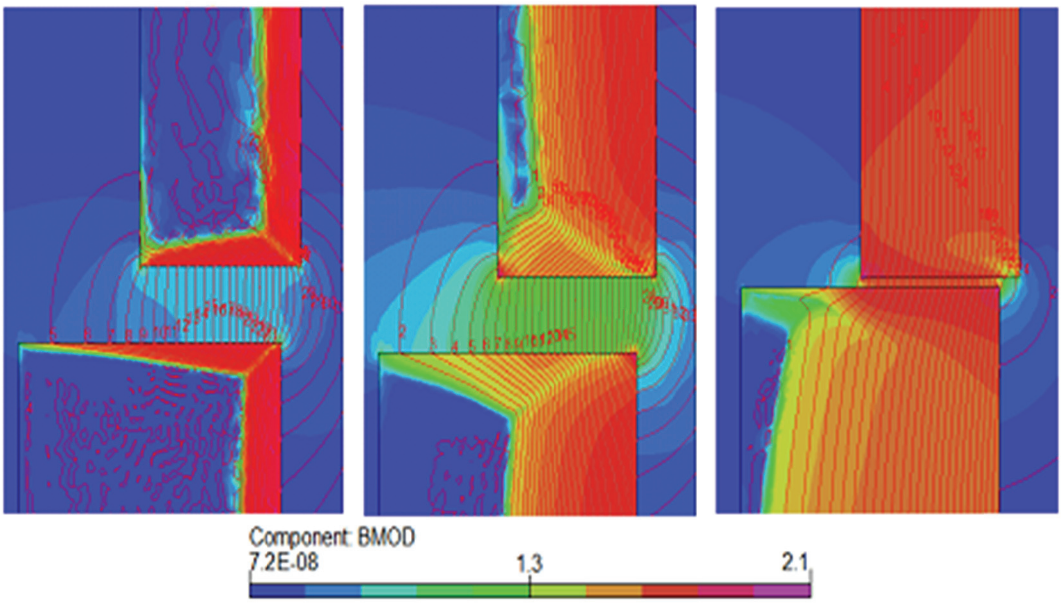

Figure 6. Magnetic field amplitude and magnetic potential before any movement in subplots (a) & (b) and in final state in subplot(c).

2.4. Mechanical design

The goal of mechanical design was to make the valve of only few parts and minimize the number of tight geometrical tolerances. The coil was decided to be "dry" because sealing of wires coming from pressurized volume is tricky. Epoxy-copperwire-composite-coil is a load bearing structure by itself so no separate frame structure was needed. Installation of valve was designed to be simple so that they could be just dropped in the block and tightened with a tightening screw. The needle was made of only one part and it was designed to be partially hardened but it was not done for this prototype because of some practical problems in time table. The whole needle should not be hardened because impurities which cause iron to harden would decrease magnetic conductance. The valve assembly is presented in figure 7 and parts are listed in the caption. The material of the needle, upper-iron-core and iron-ring was Armco Iron which has very good ferromagnetic properties but it is very soft metal. The valve body and return springs were made of stainless steel. The block and the installation screw-ring were ferromagnetic manifold steel. Photo of the prototype is shown in the figure 8.

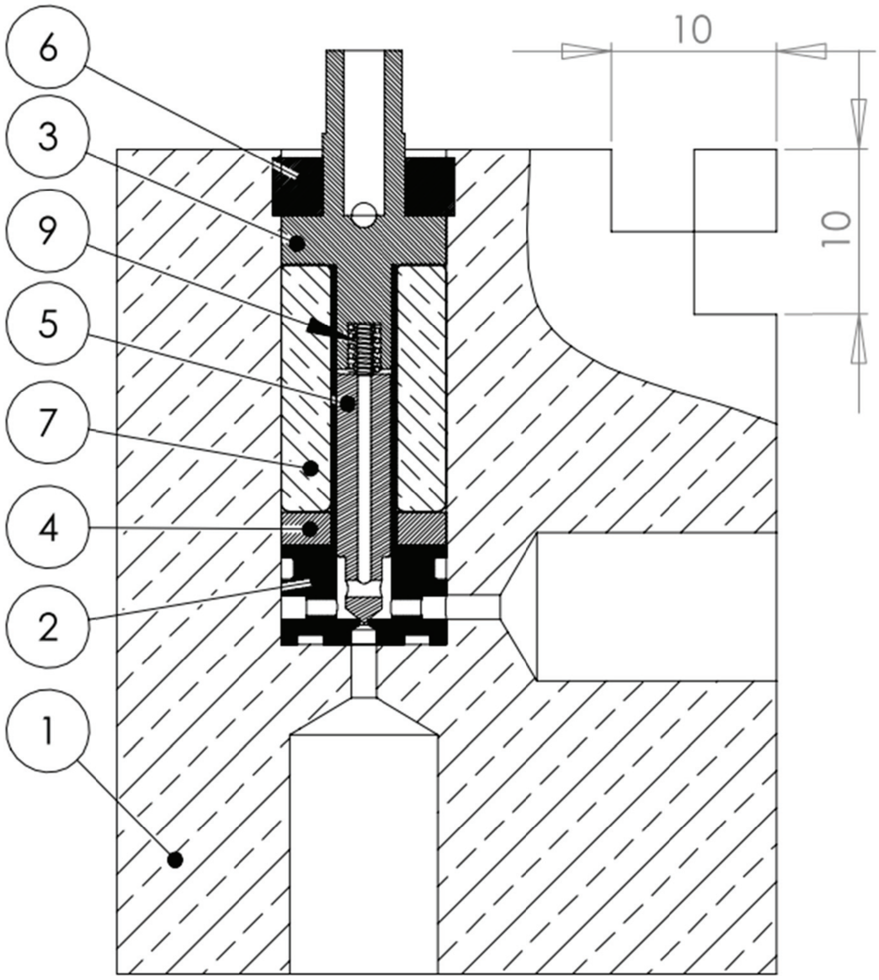

Figure 7. CAD-image with scale of assembled prototype. Parts are 1-Manifold, 2-Valve Body, 3-Upper Iron Core, 4-Iron ring, 5-The Needle, 6-Installation Screw Ring, 7-The Coil, 9-Return Springs(two pcs.)

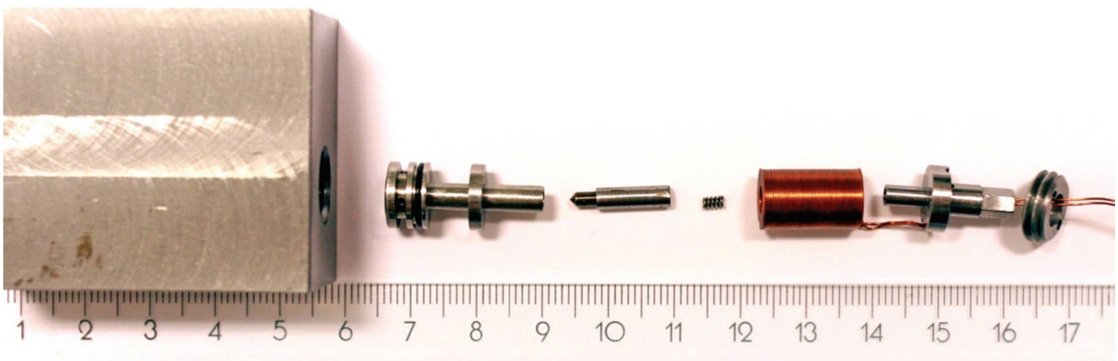

Figure 8. Exploding view of real components.

2.5. Simulation and calculation results

To approximate the performance of the designed valve a simulation model for the prototype was made. The simulated motion of the anchor is presented in the figure 9. The driving voltage is first 12V and then after 2ms dropped to the holding voltage 2V. The holding current 0.5A is small enough to keep the valve open and doesn't generate too much heat. The response time is 2.2ms and he switching energy E_k is 0.10 J. The frequency area where the valve can work is simulated using average thermal losses. The highest frequency of the valve is 55Hz.

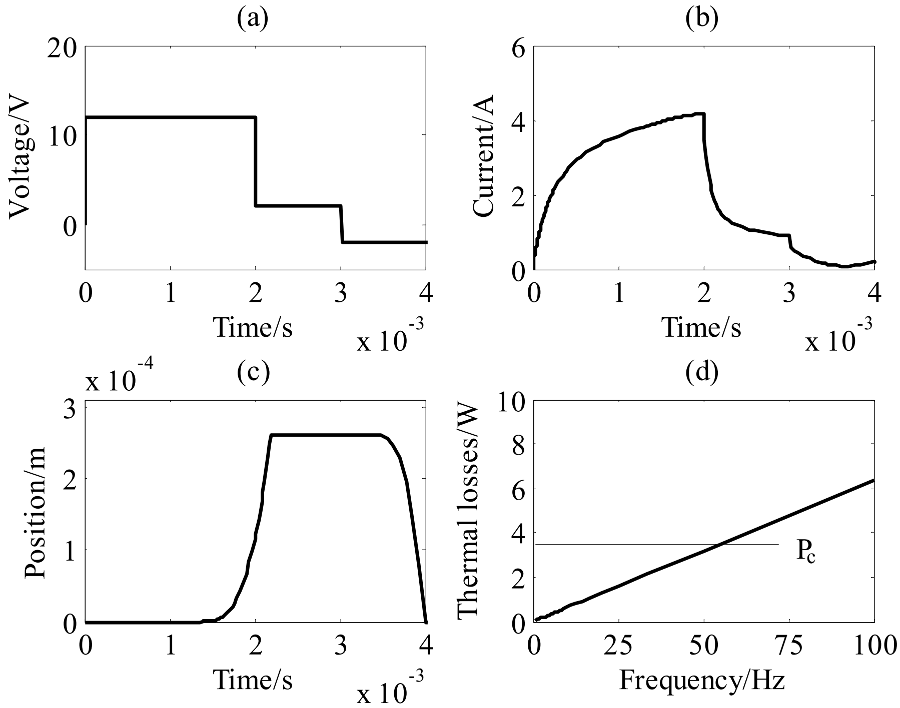

Figure 9. Driving voltage is 12V and holding voltage 2V in figure (a). Highest value of the current is 4A (b). The response time of the valve is 2.2ms (c). The highest frequency where the modelled valve works is 55Hz (d).

3. PERFORMANCE OF THE VALVE

3.1. Measurement system

Performance of the prototype was measured in valve measurement bench. Measurement system is presented in figure 10 and list of instrumentation is given in table 2. Oil in the system is Shell Tellus ISO VG 46. Oil temperature was tried to keep at 40 °C but because the flow was quite small it was not controlled at good accuracy.

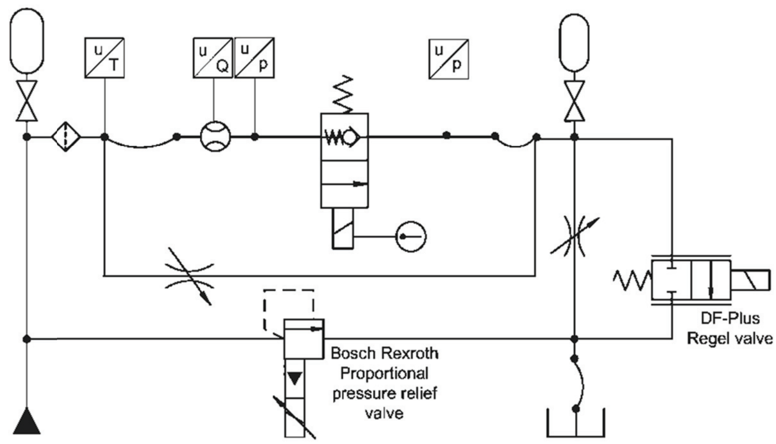

Figure 10. Valve measurement system. The measured valve is located at the centre of the figure.

The valve control electronics was designed and made in IHA and it is updated version of the circuit presented earlier in [10]. Basically the booster holds the desired boost voltage for desired time and later uses PWM to generate the hold current which is set by adjusting the duty ratio. When the valve is commanded to close, negative voltage pulse is used to drop the current to the zero quickly. Designed valve control voltages were 12 V boost for 2 ms and 1 V hold voltage for infinite time if necessary but because of some resistance in the circuit, the boost pulse was set to 16 V from which it dropped by itself to 12 V during the first 1 ms while the total boost time was 1.8 ms. During measurements, no internal or external leakage was recognized from the valve.

Table 2 Measurement instruments

Piece of Equipement	**Type**	**Accuracy**
Measuring computer	dSpace DS1103	-
Pressure transducer	Trafag NAT 250	0.3 % 25 MPa, delay 0.3 ms
Flowmeter	Volutronic VC 0.025	< 0.1 %
Multimeter	Fluke 77	-
Clamp current meter	Kyoritsu Model 8113	± 1.5 % rdg ± 5mA
Temperature sensor (Oil)	PT-100 & Nokeval	-
Temperature sensor (Coil)	Thermo couple & Fluke 52	-

3.2. Results

Valve switching time was measured by commanding the valve with 2 Hz square wave signal. Pressure differential was 12.5 MPa instead of design pressure of 20 MPa. Reason for that is discussed later.

Multiple switches are plotted over each other in figure 11 and the time scale is so that the commands are given at the time of zero. The plotted signals are the inlet pressure, the flow and the coil current. In this case, response time cannot be detected from the coil currents but the pressure and flow subplots are very clear. Valve is assumed to have switched state when changes in those signals start to happen, not when the signals settle down. Settling time depends of the measurement system but no transient can start if the valve is not changing or changed state. The biggest part of valve operation time is delay, not needle movement time which is, based on simulations, fractions of milliseconds. Based on these, it can be said that valve switching times in used measurement conditions are 1.9 ms open and 2.2 ms close. Used hold current during open state was about 0.8 A. The resistance of the coil was about 2.3 Ω(at 23 °C) so hold power was about 1.5 W. Based on calculations this could be smaller but the used booster electronics was not capable of giving less than 1.5 V hold voltage.

Because the measured pressure differential and control voltage didn't match those used in simulations in design phase, new simulations were calculated with updated parameters to get verification for the model. These results are presented in figure 12. The driving voltage (fig. 12a) is first 16V and then after 1ms dropped to 12V. The holding voltage is 2V after 1.8ms. After 3ms the driving voltage is dropped down to -10V for a short period and then kept in 0V. The current follows the driving voltage (fig. 12b). The maximum value of the current is 6.5A. The electromagnetic force depends on the current and the position of the anchor (fig. 12c). At the moment the force is greater than the resisting force 6.3N the anchor starts to move (fig. 12d). The response time is 1.9 ms open and 0.75 ms close in this model (fig 12d). The switching energy E_k is 0.14J. Compared to the measured results, the opening time matches but the closing time does not. Reason for the error is discussed later.

The average thermal losses the coil produced when the valve opens and closes are calculated by using the simulation results of the coil current in figure 12. Thermal losses are given in respect to different applies frequencies in figure 13. Estimate of the frequency that can be used continuously without overheating the coil depends on the driving voltage. Using the previous model of the thermal losses the valve can tolerate the highest frequency where the valve works is 22Hz.

Measured characteristic curves (pQ-charts) are presented in figure 14. Measured curves are inlet pressure ramp of 0 - 12.5 - 0 MPa while outlet pressure is zero and outlet pressure ramp of 12.5 - 0 - 12.5 MPa while inlet pressure is 12.5 MPa. The ramp time is 30 s. From the figure 14 it is clearly visible that cavitation choking occurs if outlet pressure is too low. This is the case all the time in inlet pressure ramp. Valve flow rate is about 0.5 L/min @ 2 MPa.

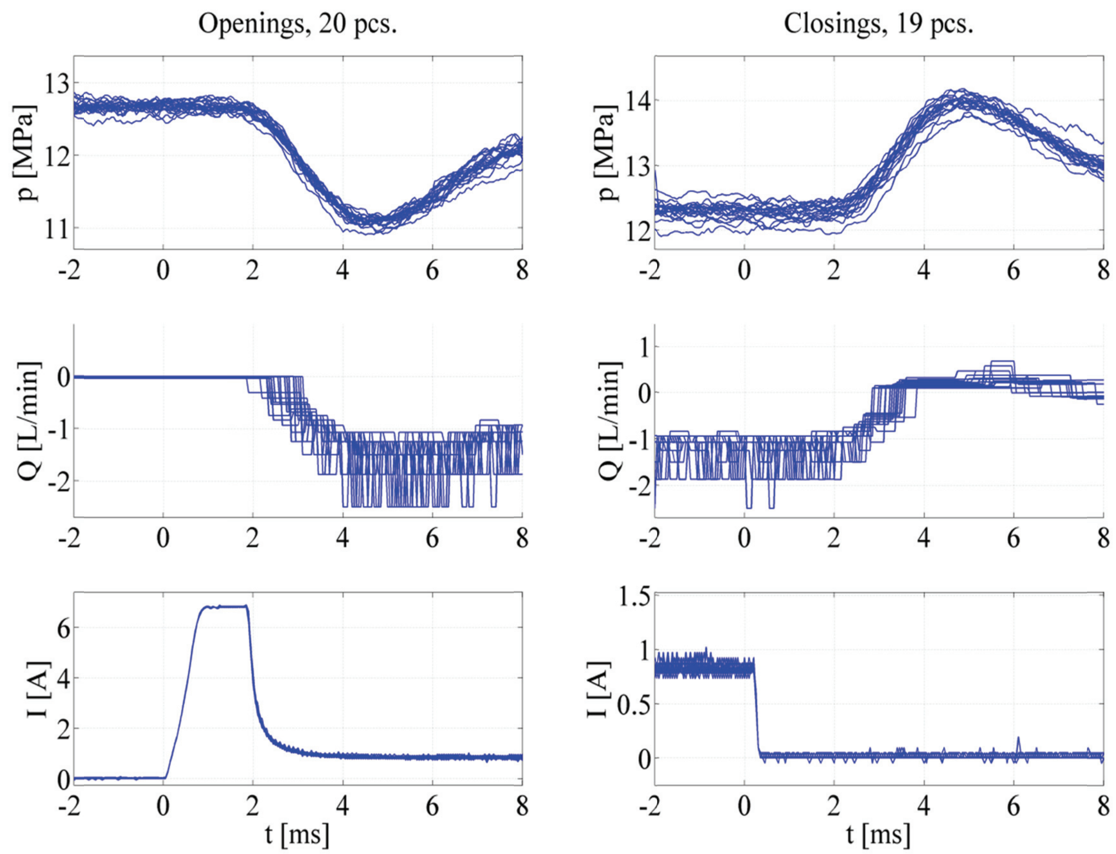

Figure 11. Plotted openings and closings. Response times can be seen from pressure curves and they are aprox. 1.9 open and 2.2 close.

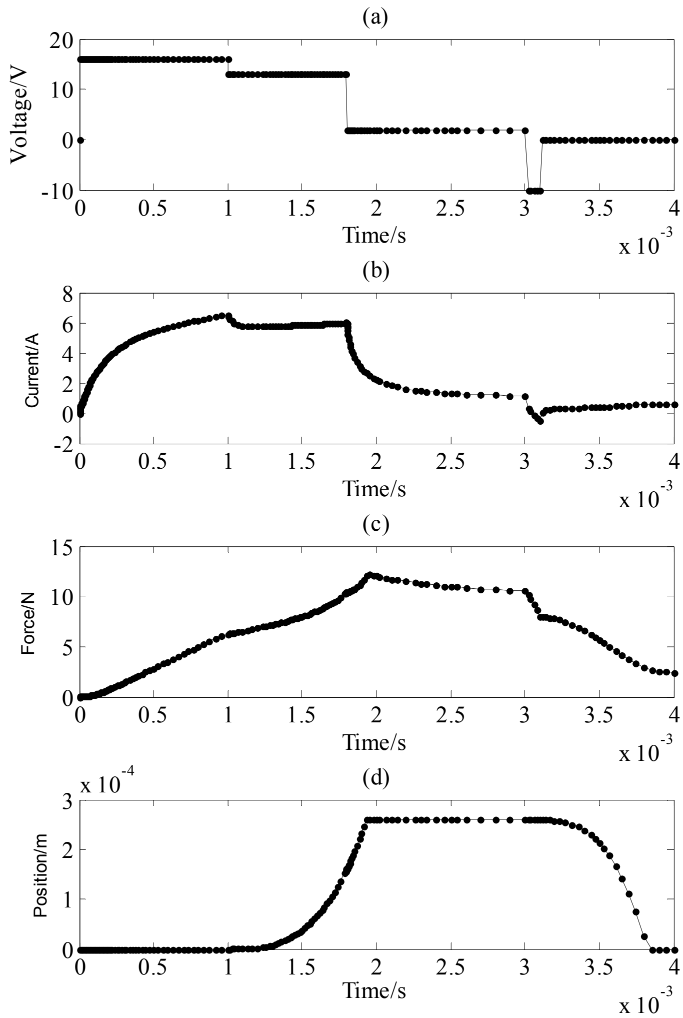

Figure 12. Simulated results corresponding to the measured case. The opening time seems to match with the measurements but the closing (commanded at the time of 3 ms) does not and is over one millisecond shorter than the measured value.

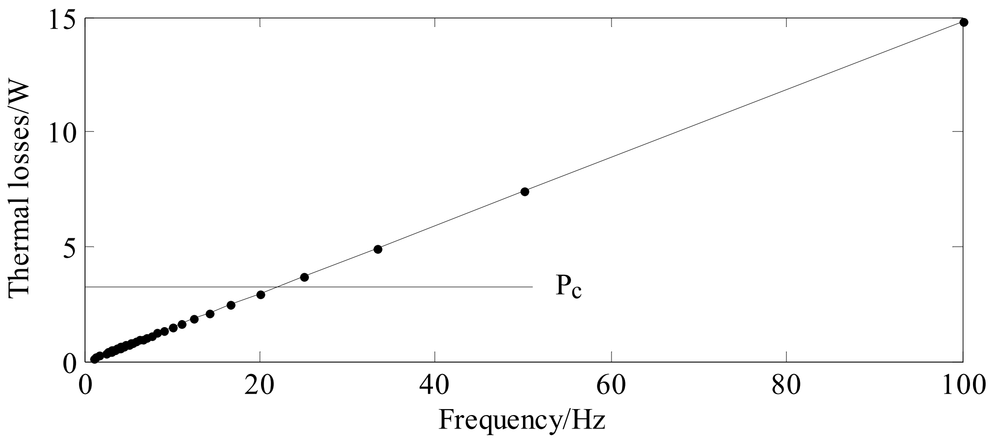

Figure 13. Thermal losses the coil produced in the previous (fig. 12) simulations using different frequencies. The highest frequency where the valve works infinite cycles without over heating is 22Hz

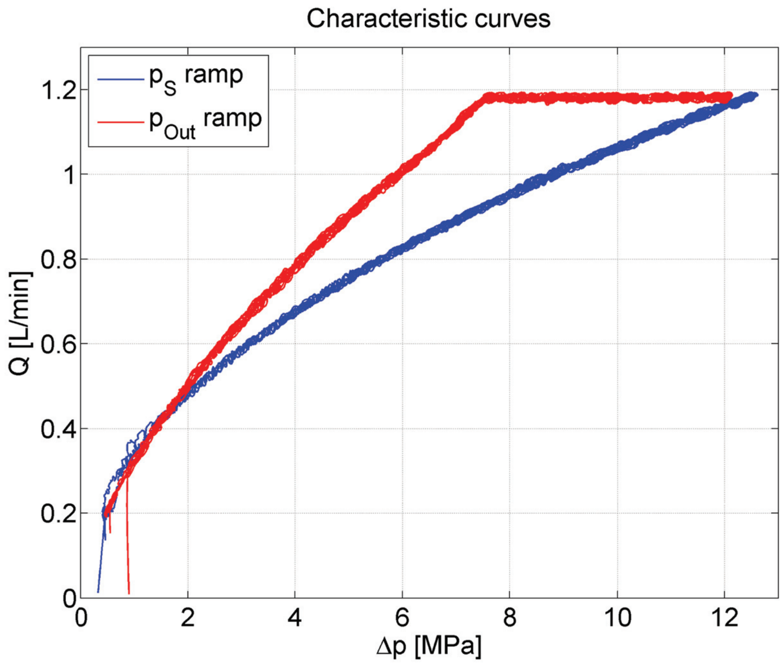

Figure 14. Characteristic curves with inlet and outlet pressure ramps. The valve flow rate is about 0.5 L/min @ 2 MPa

4. DISCUSSIONS

The valve response time, both open and close, were about 2 ms in measured conditions. The valve was not measured against highest design pressure differential of 20 MPa because clear deformation at the tip of the needle was recognized during visual inspection after presented measurements. Deformation was though expected because tip of the needle was not hardened and the material it was made is very soft iron. In the future we shall try if Boron hardening at the tip of the needle helps needle to keep its shape. Cavitation choking is a major problem on higher pressure. This might be helped by designing valve flow path better way. The flow rate of the valve was 0.5 L/min @ 2 MPa. A good target value for valve flow rate in PNM coded 32 bit DFCU would be about 1 L/min @ 1 MPa.

Compared to smallest commercial hydraulic on/off-valve found, 250S type valve from the Lee Company [11], the prototype is better. Maximum flow rate of 250S is 1.2L/min at maximum pressure of 21 MPa. From turbulent orifice equation one can calculate diameter for similar orifice to be about 0.4 mm. Diameter of the orifice in our prototype is 0.5 mm. Outer diameter of the solenoid of 250S is 15 mm while our solenoid was just 10 mm. Length of the Lee valve is 40 mm, our valve is 30 mm. Last but not least, The Lee gives to their valve response time of 15 - 40 ms which is extremely poor value. Our prototype has response time about 2 ms. Comparison to previous prototypes [5] & [6] and to the Lee 250S is presented in table 3.

Table 3 Comparison of properties of four different valves

Property	ProtoØ10	Hammer valve	Bistable valve	Lee 250S
t [ms]	2	2	3.5	15-40
f cont. [Hz]	22	50	30	N/A
f max [Hz]	N/A	200	100	N/A
Q_nom [L/min]	0.3	3.3	4.4	N/A
Q_max [L/min]	1.2	17	18	1.17
Dp max [MPa]	12.5 (21)*	21	21	21
V [cm^3]	2.4	7.0	58	4.4**
q nom [L/min/cm^3]	0.13	0.47	0.21	N/A
q max [L/min/cm^3]	0.5	2.4	0.94	0.23
P cont. [W]	1.5	0 (bistable)	0 (bistable)	4.8
E act [J]***	0.10	0.14	0.68	N/A
*	Design pressure was 21 MPa but measured only with 12.5			
**	Only solenoid and valve, flange excluded			
***	Based on electromagnetic simulations only.			

Measurements and simulations matched quite well. In simulations the delay in closing the valve was not the same as in the measurements. It could be explained to be caused by hydraulic stiction, some slight differences between the model and actual measurements and remaining electromagnetic field in the manifold and thus magnetic force exerted on the anchor. In this project, actuator and hydraulic part was designed separately. There could probably be better way to integrate parts and optimize them together in order to make a better valve.

5. CONCLUSIONS

The miniature needle valve presented, named ProtoØ10, is unidirectional, simple spring return needle valve which has response time about 2 ms. Compared to the previous prototypes, the most important feature is that ProtoØ10 is made of simpler parts and therefore can be even more miniaturized. In conclusion of [12] it is mentioned that hydraulic part used in [5] & [6] may be problematic to scale smaller and therefore cannot be used in digital micro hydraulics but new hydraulic parts are required. Results indicate that even in simple spring return on/off-valves on the market are not optimized for maximum possible performance.

No counter evidence which tackles our ideas of digital microhydraulic valve were not found but we still learned few things which must be handled better way in next prototypes. This valve was the very first prototype of digital microhydraulics research project.

6. ACKNOWLEDGEMENTS

This research is done in the project DiHy that is part of the Finnish Metals and Engineering Competence Cluster, FIMECC-program EFFIMA [13].

ANSYS/Fluent used in computational fluid dynamics was offered by CSC. “CSC - IT Center for Science Ltd is administered by the Ministry of Education, Science and Culture. CSC provides IT support and resources for academia, research institutes and companies” [14].

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