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Planned Test of the Prototype 2B Linear Machine with Gas-Springs for Energy Harvesting from Waves

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***Abstract*— This paper describes a proposed test plan for harvesting energy from waves with a linear machine that uses gas-springs. The linear machine "Prototype 2B" has been put into production and will be finished. The machine has been constructed to withstand 300 bars of gas pressure and the maximum speed of the rotor is set to 10 m/s due to the sealing.**

The model made in [5] uses the fundamental equations and describes the system behavior of a linear machine with gas springs. The model has been slightly improved and the input specifications for the "Prototype 2B" have been used. The simulations shows that one needs to feed electric energy in some parts of the cycle, but the whole cycle will yield a positive net contribution of electrical energy. The model is not optimized, and hence should be improved due to the time consuming simulations.

Steel profiles have been proposed to use for the frame of the test-bench. The first draft of the drawing has been sent to NTNU and the workers have started to build the frame.

To ensure the vibration energy to spread itself into the surroundings, isolators need to be selected and adapted. The natural frequency of the test bench with four chosen isolators was calculated to 18,17Hz. The chosen isolators are therefore most likely good enough since the lowest operating frequency of the machine was simulated to 55 Hz.

Transferring the energy from the high-force-low-speed source (the wave) to the resonator has been discussed and some suggestions have been proposed. The energy transfer is done by raising pressure in a freestanding actuator with a hose connected to the gas spring. Hence, increasing the actuator pressure will increase the gas spring pressure and the force acting on the rotor will be significantly higher. A procedure to select the correct components are discussed and a Matlab script for analyzing the choice has been made.

The high-force-low-speed source has been discussed. Two solutions where a hydraulic system is used have been further investigated, and some calculations and analysis have been performed. Simulations show that one need to apply a maximum force of 24115N at one actuator with a 32mm effective piston area. The regulation of the high-force-low-speed source could be crucial to obtain a good test result since the regulation of the wave must adapt itself to the resonating force of the rotor. The regulation must therefore be investigated further as it can have tremendous impact on the possibility of extracting energy from the machine.

The electrical source and regulation control have been discussed. Timing of the electrical energy supplied is crucial and three operational simulations have been performed to investigate when the positive net contribution is at it`s highest. In order to extract the maximum possible net energy from the machine, the right electrical control is very important to implement. Two of the control systems have been proposed, where perhaps the simple proportional controller should be chosen to prove the concept. Later on a more sophisticated electrical controller should be applied since the efficiency of the wave energy converter could increase significantly.

The non-linear compression event in the gas-springs is not proportional to the electrical extracted energy. Simulations show that when the pressure increases in the springs one could extract more energy, but the non-linear event could not be recognized.

Simulations show that the efficiency of the system varies significantly with the controlling of the machine. If a simple proportional controller is adapted, the theoretical efficiency becomes 2,87% and with a more sophisticated controller one could get 10,02%. This means that there must be done more research on the electrical operating control system if energy harvesting could be profitable.

***Index Terms*— Linear permanent synchronous machine, permanent magnets, tubular machine, wave energy harvesting, resonance.**

I. Nomenclature

x_r	*Position of Rotor*
x_{ac}	*Absolute length of gas camber*
v	*Velocity*
q	*Point charge*
a_r	*Acceleration of Rotor*
m_r	*Mass of Rotor*
m_c	Mass of casing and stator
p	*Pressure*
p_0	*Initial Pressure in the gas spring*
p_{end}	*End Pressure*
$p_{0,x}$	*Initial Pressure in Volume x*
V	*Volume*
V_0	*Initial Volume*
V_{end}	*End Volume*
$V_{0,x}$	*Initial Volume in Volume x*
α	*Adiabatic gas constant 1,4 for Helium*
A_{spring}	*Area of gas Spring Piston*
D	*Mean Diameter of the Coil*
D_s	*Diameter of gas spring*
f	*Frequency of the rotor*
F_{el}	*Electrical Force*

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Resonator AS is a small R&D company that is developing a high resonating machine for different applications. They own the intellectual rights to the patent.

Email: resonator@resonator.no

F_{gSL}	*Force from left Gas Spring*
F_{gSR}	*Force from right Gas Spring*
F_{mSL}	*Force from left Mechanical Spring*
F_{mSR}	*Force from right Mechanical Spring*
F_{frictL}	*Friction force*
F_{frictR}	*Friction force*
μ_k	*Friction constant*
N	*Turns*
B	*Flux density*
ϕ	*Flux*
L	*Inductance*
R	*Resistance*
i_s	*Current in the coil*
U	*Voltage source*
U_{emf}	*Induced Voltage in the coil*
k_{iso}	*Spring constant of isolator/absorber*
F_{casing}	*Force of the casing (Whole machine)*
n	Amount of substance of gas
R_{gas}	*Universal gas constant*
T	*Temperature*
τ	*Response time*

II. Introduction

THE world oceans cover approximately three-fourths of the earth's surface. The many oceans, the land and the air above contain enormous energy resources. These resources have been utilized all over the world for decades. Hydropower and wind were used in the old days e.g. to power sawmills, grind flour and pump water. Today they are more or less only used to produce electricity, which is easy to transfer to the individual users. The idea of harvesting energy from waves was thought of more than several thousand years ago. The first patent for utilizing wave energy was in 1799 by Girad and his son [1] –[3].

Today several different patents exist on how to harvest energy from waves. Some of the most well known concepts are tidal energy plant, floating tube "Pelamis", bow technology "**CETO**" and linear motion generator concepts [4].

In this paper we will focus on the linear motion from etc. a float lying upon the ocean surface. The annual average power for a one metre wave front is 30-40 kW/m outside Lofoten and Stadt in Norway [8]. The power mentioned here equals a wave that is 3m high and have a period of 8 seconds. This emphasizes that the speed of a float is moving with the velocity of 1.2 m/s. Hence, the speed of the float is slow, and this results in large equipment size since the size of a regular linear electrical generator will increase proportionally with the force and speed of the linear motion [6]. The common denominator for all the different concepts mentioned here, is that all of them are associated with high costs because the structures are relatively massive. This results in a lack of desire to build wave power plants, as the investors face the risk and may not get their money back.

The concept discussed in this paper will be a linear tabular generator with gas springs. Several papers at NTNU have introduced the concept of linear tabular machines. This paper will focus on the concept of harvest energy from high-force-low-speed from waves. Pre work and simulations from [5] and [6] shows that by converting high-force-low-speed (wave energy) movements to high speed low energy in a mechanical frequency converter, a surplus of energy could be extracted in form of electricity. Papers [5] and [6] are beneficial to read if you want to get a deeper understanding of the concept. However, some of the basic principles will be reproduced in this paper.

The mechanical frequency converter, hence tabular linear machine with gas springs has been designed in cooperation with Resonator AS. The machine was planned to be built in the spring 2011. This machine is named the "prototype 2B"; as several planned and mediocre prototypes were build in 2007 and 2008. The machine is greatly improved and scaled up to one phase 230V and approximately 3kW.

A "test-bench" for testing the theoretical proven concept of extracting a surplus of energy from the frequency converter is planned. The bench has been designed to simulate waves that contain a great amount of force acting on the machine. This force has been made of standard commercial hydraulic components that can be rented or bought off the shelf and putted together. Parts of the bench have already been built at NTNU in Trondheim, the rest was planned to be built at the University of Ås.

III. Principal Description of The Machine

The tubular linear machine consists of different components, and a principal geometry of the model is shown in figure 1.

The rotor is placed in the center of the machine and made of magnets and relatively thick iron washers to make a specific mass and flux density in the machine.

The gas springs are located on each sides of the rotor and will be compressed and decompressed due to an extraneous source (the wave) and the movement of the rotor itself. The two gas chambers will be completely air tight to prevent any mass flow between the two chambers that could result in nonsymmetrical behaviors. The gas inside the cambers will act as springs on the rotor. When the springs are compressed enough, the gas-spring-force characteristic will change rapidly due to the non-linearity in the compression/decompression of the gas. This non-linear event is desirable to exploit, as more energy can be stored in the springs and hence force the rotor to bounce between the two springs with a higher frequency and speed.

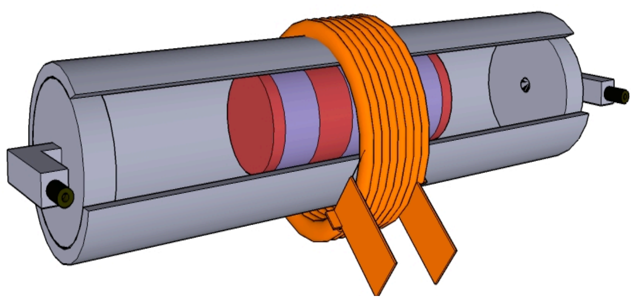

Fig.1. Principal drawing of the Resonator

To be able to move and extract energy from the rotor, a coil or several coils needs to surround the rotor. To be able to force the rotor to move back and forth, an alternating current needs to be applied to the coils and hence this will set up a magnetic field. The magnetic field set up by the current and coils will interact with the magnets in the rotor and hence lead to a mechanical movement of the rotor. The opposite happens if the rotor is moving in the nearby area of the coils. Both events follow Lenz Law, that says "An induced current is always in such a direction as to oppose the motion or change causing it" [9].

A. How To Extract Energy From The Machine

Figure 2 shows one principle cycle of the energy production.

1. *At the beginning of the cycle, an alternating current with the resonator`s dynamical natural frequency is applied to the stator coils and will force the rotor to bounce between the gas-springs in resonance. The high-force-low-speed (the wave) is now starting to compress the gas-springs inside the resonator. In this stage the machine will consume electrical energy due to friction and to keep the rotor in resonance, hence act as a motor.*
2. *As the gas-springs are being compressed, the rotor picks up higher frequency and speed due to the change in the spring constant. The movement of the rotor eventually reaches high enough velocity, and the back emf from the coils has now a slightly higher voltage than the source*, ref equation 12. *The current will now change direction, and the coils are now starting to deliver electrical energy back to the source.*
3. *When the high-force-low-speed have reached it`s maximum amplitude, the frequency and speed are at it`s highest.*
4. *The gas-springs are now on it`s retract, and the pressure inside the springs are decreasing. The spring constant will now change in the opposite direction, and the velocity and frequency is decreasing. When the rotor`s velocity has fallen enough, the current once again change direction. The coils are now consuming electrical energy because they are forced by the feedback control to dynamically maintain the natural frequency of the system.*
5. *The gas-springs are now fully decompressed, and the cycle repeats itself from 1. The velocity and frequency are now at its lowest value.*

We have now yielded one cycle of net electrical energy contribution from the resonator.

IV. Modeling The Machine

The machine concept was analyzed in autumn 2010. This led to a model made in Matlab-Simulink where fundamental equations and principles were integrated [5]. In this paper the mechanical springs are not included. This is because the influence from the mechanical spring is very low compared to the gas-springs, and hence do not affect significantly the system`s characteristics. The new prototype 2B does not include mechanical springs in its design, and hence the

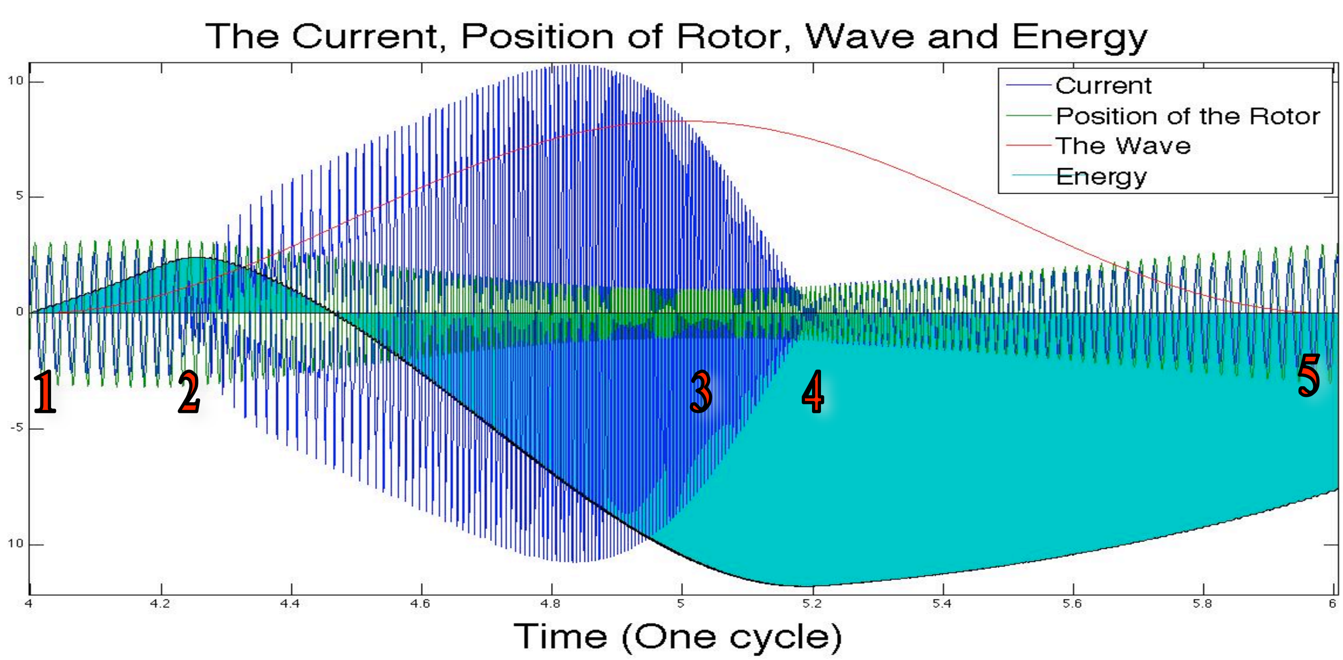

Fig.2. Principal operation of the energy production

mechanical springs should be removed from the model.

A. The Equations of the system

1) Modeling of the Rotor

Newton`s first law is used by considering the sum of all forces acting on the rotor

$$\sum F = F_{el} + F_{gSR} + F_{fricR} - F_{gSL} - F_{fricL} \qquad (1)$$

and Newton`s second law

$$\sum F = m_r a_r = m_r \frac{d^2 x_r}{dt^2} \qquad (2)$$

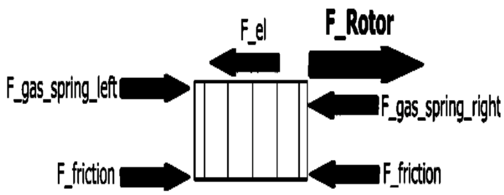

Fig.3. Forces acting on the rotor.

2) Modeling of the friction

When two surfaces are moving with respect to one another, the frictional resistance is almost constant over a wide range of speed.

$$F_{friction} = \mu \frac{dx}{dt} \qquad (3)$$

3) Modeling of the gas spring

The relationship between force, pressure and area;

$$F_{spring} = p_0 A_{spring} \qquad (4)$$

The volume inside the gas-spring will change and thus emphasizing a change in force acting on the rotor. This gives us;

$$F_{spring} = p_0 A_{spring} \frac{V_0}{V_{end}} \qquad (5)$$

and if one assume an adiabatic process (no heat transfer through the cylinder walls) where α is the adiabatic gas constant

$$F = p_0 A \left(\frac{V_0}{V_{end}}\right)^{\alpha} = p_0 A \left(\frac{x_{ac}}{x_{ac} - X}\right)^{\alpha} \qquad (6)$$

4) Modeling of the electromagnetic force from the coil

Lorentz force equation describes the interaction between a moving point charge and a magnetic field

$$F_{el} = qv \times \mathrm{B} \qquad (7)$$

If a conductor is transporting charged elements, the equation becomes

$$F_{el} = Ni\, L \times \mathrm{B} \qquad (8)$$

If N turns, the length of the conductor is πD and the flux density is orthogonally oriented to the coil one get

$$F_{el} = \pi \mathrm{D} N i \mathrm{B} \qquad (9)$$

5) Modeling of the current in the coil

Faradays law describes the induced voltage (electromotive force) in the coil.

$$e = U_{emf} = \frac{d\phi}{dt} \qquad (10)$$

The flux flowing through N loops of the coil is

$$\phi = N\pi \mathrm{D} B x \qquad (11)$$

and the induced voltage in the coil becomes

$$U_{emf} = N\pi \mathrm{DB} \frac{dx}{dt} \qquad (12)$$

To find an expression for the current in the coil, one need to use Ohm`s law.

$$U(t) = R i_s(t) + L \frac{di_s(t)}{dt} + U_{emf}(t) \qquad (13)$$

and with some simple algebraic calculation one get

$$\frac{di_s(t)}{dt} = \frac{\pi DBN}{L}\left(\frac{dx_r}{dt}\right) - \frac{R i_s(t)}{L} + U(t) \qquad (14)$$

B. The Implementation And Scheme

The following equations were implemented in Matlab-Simulink by using block diagrams and small calculation scripts. This method gives us an easy and simple way of connecting the principle equations of the system together. The block diagram made in this report is quite similar to the one in [5], but has been slightly improved to meet the challenges of the prototype 2B, see Appendix [1] for scheme, scripts and functions.

V. Design of the Machine

Resonator AS owns intellectual property rights to the resonator patent and has organized several MSc and PhD students to work on different areas of the patent. There is a strong desire to investigate these areas, and therefore several tests are planned this year. The tests could help the company to choose the most likely successful area to invest in.

Resonator AS and the MSc Student A.Bostad with the University at Ås have mainly developed the design of the prototype 2B machine. I have participated and contributed with ideas and suggestions at many of the discussions.

A. Requirements

Some of the main requirements to make this test successful were discussed:

- ✓ Efficiency of the machine
  - Low winding resistance
  - Low reluctance
  - Low eddy current loss
- ✓ Mechanical
  - Withstand very high pressure
  - Low friction
- ✓ Be able to measure
  - Position of the rotor
  - Pressure
  - Temperature
  - Voltage over the coil
  - Current in the coil

The efficiency of the machine is important due to the test. This is because simulations from [5] shows that a relatively small amount of average energy can be extracted from the machine when it is running. It is important to remember that the simulations are ideal and hence it is important to get the electrical efficiency as high as possible.

The gas springs must withstand high pressure. Simulations in Appendix [2] show that the more one compress and decompress the gas springs, the more energy can be extracted from the machine.

It is very important to have the ability to measure the different events in the system. The position of the rotor is necessary to measure due to control strategy of the machine. Pressure, temperature, voltage and current measurements are extremely important due to the documentation and further work. The measurements will help to decide if it is worth investigating this concept further.

B. Overview Of The Design

Resonator AS and A.Bostad have mainly been focusing on the electromagnetic design of the motor. The design priority has been to maximize the force density while having an acceptable motor efficiency [13]. This is mainly because the businesses area of the company is hard-rock hammer drilling.

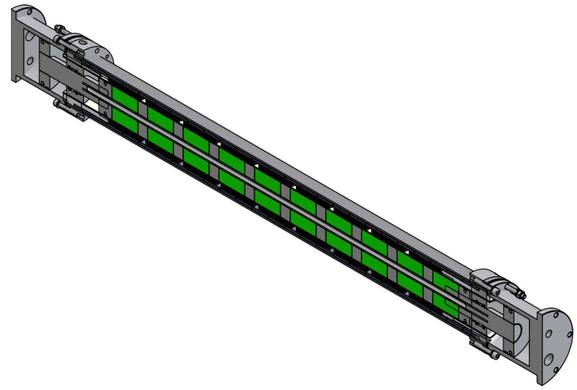

Fig.4. Planned design of Prototype 2B, more pictures in Appendix [3]

The design requirements Resonator AS had for the machine is to match and finally out perform the Wassera 50- or 80 water driven hammer in ROP. Hence, design priority one is to produce high impact force at a given frequency. The second priority is the machine efficiency [13].

Analytical and FEM design details of the machine are described in "New design of a reciprocating TLPMSM" by A.Dahl-Jacobsen at Resonator AS and the avid reader is urged to read it. The main design requirements and results are reviewed here.

TABLE I
DESIGN REQUIREMENTS [13]

PARAMETER	VALUE	UNIT
Phases	1	1
Voltage	230	V
Outer Diameter	80	mm
Length	<1	M
Thrust force	High as possible	N
Frequency	100	Hz

TABLE II
SUMMERY OF THE DESIGNED MACHINE

COMPONENT		VALUE	UNIT
Number of turns	$N =$	560	1
Max Pressure	$P_{max} =$	300	bar
Length of gas chamber	$x_{ac} =$	0,24	m
Total length of machine	$x_{machine} =$	0,952	m
Diameter of gas spring	$D_s =$	0,063	m
Gas constant for helium	$\alpha =$	1,4	1
Mass of rotor	$m_r =$	8,541	kg
Mass of casing and stator	$m_c =$	17,577	kg
Flux density (Estimated)	$B =$	0,3	T
Mean Diameter of Coil	$D =$	0.056	m
Resistance in the coil (Teoretical)	$R =$	0,4	Ω
Friction factor between casing and Rotor/Piston	$\mu =$	0.001	1
Inductance	$L =$	0.0018	H
Position sensor LVDT [Appendix 4] not yet decided	2		pieces

VI. Simulations Of The Machine

The input values used in the simulations can be found in table 2. These simulations will give a hint of what one can expect from the real test with a easy proportional controller that helps the resonator to maintain it`s natural frequency at a given pressure. Perhaps the proportional controller is not the optimal choice, but to prove the concept, it should do the work. More comprehensive control regimes will be looked at later in this paper.

When the machine is running as a motor, the voltage source and hence the amplitude is limited to 130. This could be changed to another value, but several simulations show that the voltage should be in the area of +- 130V to be able to extract a reasonable amount of electrical energy. The voltage is of course dependent on the gain of proportional controller and at the same time the pressure in the gas springs. The gain in the proportional controller was set to 28,9 and the pressure to maximum 300 bar.

The high-force-low-speed external source is set to 0.5Hz (hence 2s period). Here it is assumed that the external source is big enough to withstand the compressed gas-springs force caused by the movement of the rotor and displacement of the gas spring chamber. The external force is ideal and will therefore not be affected by the resonator system.

The initial pressure is set to 10bar. Higher initial pressure will cause the gas springs to store more energy and this will also lead to a higher frequency and speed in the system. Higher frequency and speed will produce more heat and hence losses.

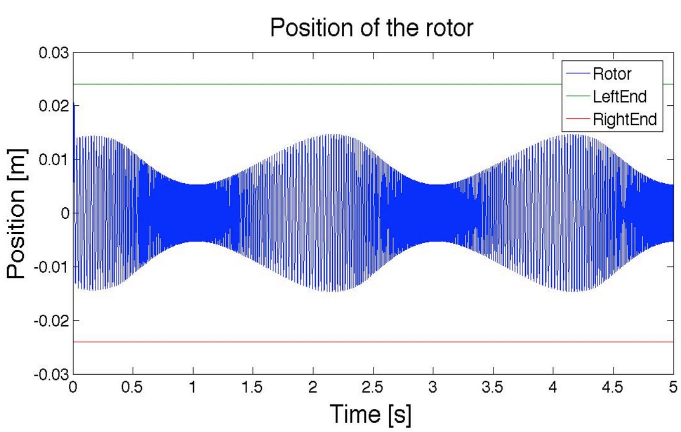

Fig.5. The position of the rotor to reference

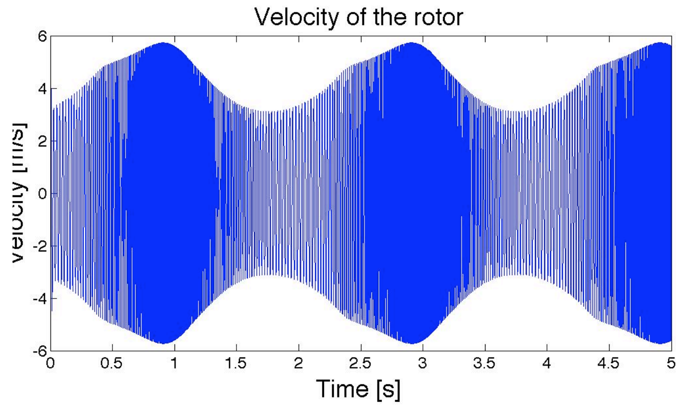

Fig.6. The Velocity of the rotor

Figure 5 and 6 shows the rotor`s position and velocity as a function of time. Notice that the frequency and stroke length changes as the gas-springs are compressed; hence the frequency and speed increases and the stroke length decrease. The frequency is measured to approximately 55 and 200 Hz.

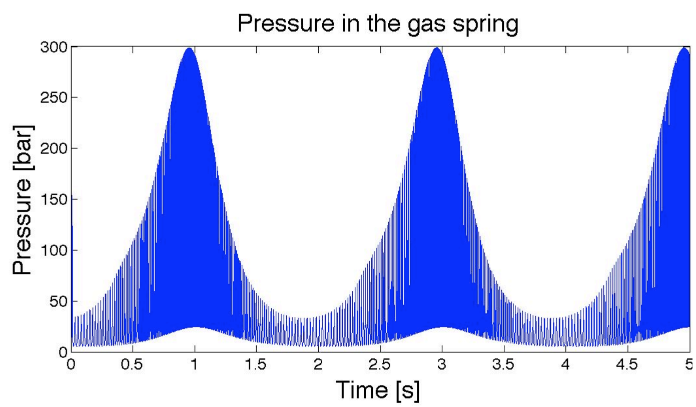

Fig.7. The pressure from the gas-springs

Figure 7 shows the pressure in the springs as a function of time. The maximum potential force in one particular gas spring will be $F = pA$, hence 93,5kN.

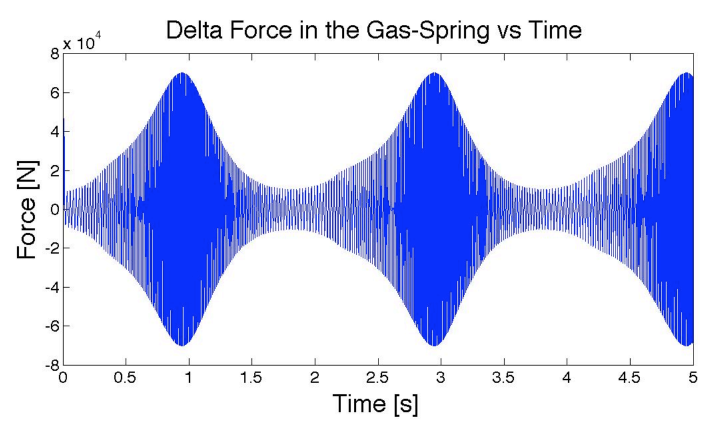

Fig.8. The difference in force between the two gas-springs

It is worth noticing the extremely high forces in the gas-springs that will be produced inside the machine. Maximum delta force will be approximately 7 tons. When the test is going to be performed, the achieved force and pressure in the machine are very high, and hence a safe room should be adopted.

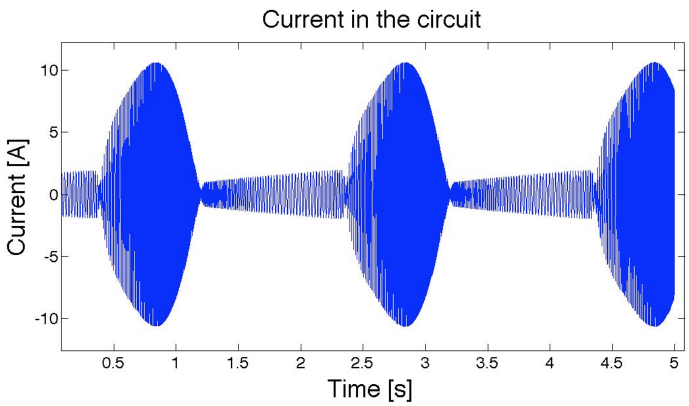

Fig.9. The current in the coils

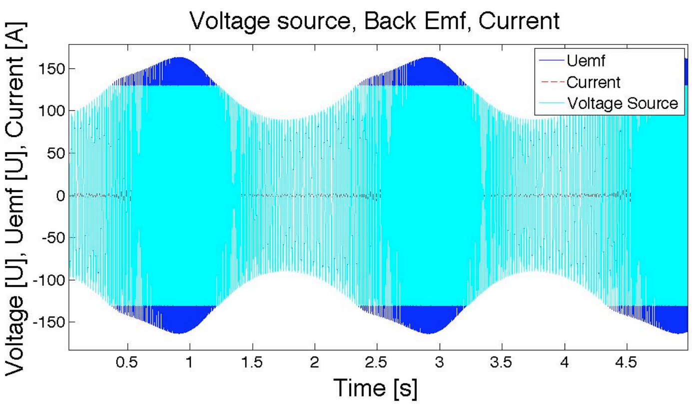

Fig.10. The voltage source and back emf

Figure 9 and 10 shows the current and voltages in the circuit. It can easily be seen that the Uemf is proportional to the velocity of the rotor ref equation 12. The current and voltage is only 10 Ampere and 160 Voltage and are suitable for most commercial frequency converters made today.

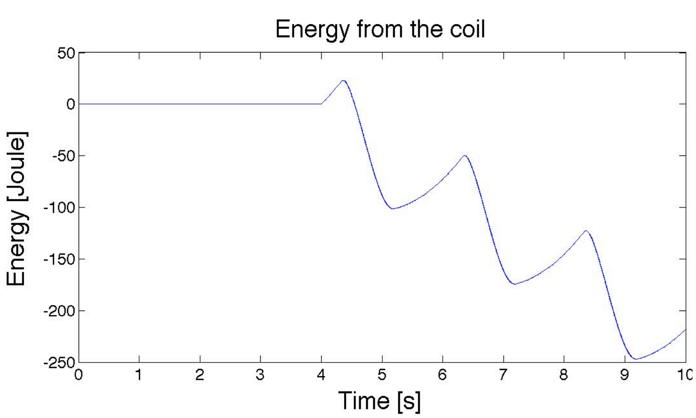

Fig.11. The energy from the coil

Figure 11 shows the possible theoretical accumulated energy that could be extracted with the given details in table 2. Notice that the machine needs some time to get into steady state and therefore the measurements are plotted after 4 seconds. At 10 seconds and a measurement period of 6 seconds, the energy is measured to 220 Joule. The effect is $P = Energy/Time$, hence 33W.

VII. Design of the Test Bench

A. Basic Concept

Figure 12 illustrate a basic concept of how one can utilize the energy from the waves. The float is moved up and down by the wave, forcing it to decompress and compress the gas-volume inside the gas cylinders. When the cylinders are compressed, the pressure inside increases and hence a small amount of mass-flow appears from the cylinders trough the hoses and into the resonator`s gas springs. This result is an instant increase in pressure of the resonator`s gas springs. The potential energy in the gas spring inside the resonator increases, and hence some of this energy is transferred to the rotor forcing it to bounce between the gas springs with a higher frequency and speed.

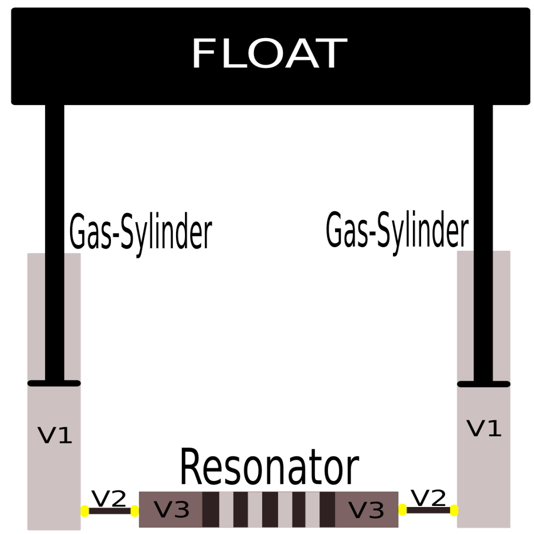

Fig.12. Principle drawing of the wave concept

B. Test Bench Frame

An illustrating picture of how the test-bench could be made is shown in figure 13. The frame of the test-bench is mainly built of steel. Different pipes (50x50x5mm), angular (50x50x5mm) and flat steel are welded together to make a solid frame for the machine. Holes are drilled in the lower steel elements to make it possible to clamp the frame to the floor. The open space is meant for supplies to produce the high-force-low-speed motion and auxiliary equipment. More pictures can be seen in appendix [5]

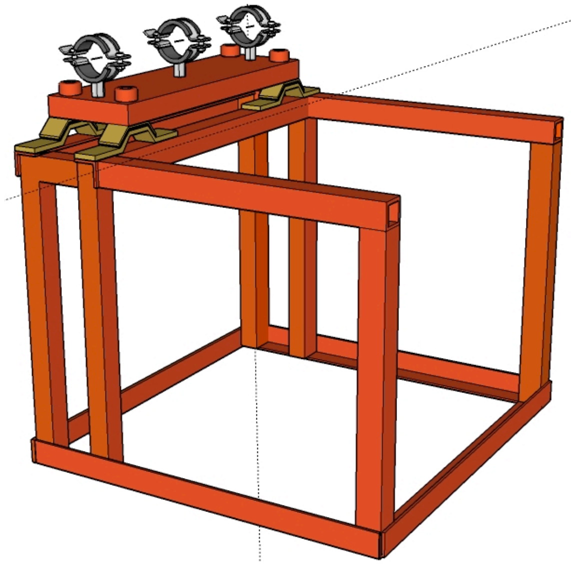

Fig.13. Proposed test bench frame

C. The Camps And Absorbers/Isolator

The resonator is mounted to a steel-bed with steel clamps that includes rubber cushion. Several vendors sell the rubber bands and clamps to hold the resonator. Two examples of clamps are shown in Appendix [6].

The steel-bed is lying upon four rubber isolators that will absorb and mainly isolate the shear (horizontal force) vibrations produced by the machine. The use of isolation is primarily for reducing the effect of the dynamic forces generated by moving parts in a machine into the surrounding structure [14]. If not anti-vibration rubber was used; some of the kinetic energy produced by the machine would be transferred to the surroundings and hence lost.

There are many companies who sell isolators. To find out which can be used, three principle factors control the selection of an isolator. The first is the weight to be supported, the second is the disturbing frequency of the machine and the third is the rigidity of the structure supporting the machine [14].

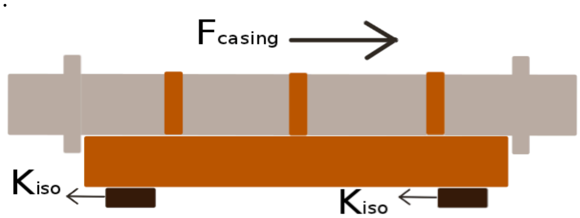

Fig.14. Principal scheme of the steel-bed and resonator

The weight of the components lying upon the four isolators needs to be summed to find out how much mass that is moving. Hence, the steel bed, clamps, bolts and the machine casing needs to be included. The rotor needs to be separated from the rest since this affects the movement of the machine. One assume that the vertical forces will be minimized compared to the horizontal, and hence do not take the vertical forces into account. Figure 14 show a scheme of the set up and its horizontal forces.

From the simulations, one knows that the operating frequency for the resonator will be 55-200Hz. To cope with the vibrations produced by the machine, the isolators need to have the right stiffness. Companies who sell isolators provide the information about the stiffness/spring constant.

To calculate the natural frequency of the machine casing including the steel-bed, clamps and bolts the following equation can be used;

$$f_{natural} = \frac{1}{2\pi}\sqrt{\frac{4*k_{iso}}{m_c+m_{steel-bed}+m_{clamps}+m_{bolts}}} \quad (15)$$

TABLE I I I

Mass of components

COMPONENT	VALUE	UNIT
Steel-bed (estimated)	10	kg
Clamps (estimated)	2	kg
Bolts (estimated)	0.5	kg
Stator and Casing with Springs	17,577	kg

Appendix [7] shows one supplier of isolators that may be used. If one choose the elBe 130 with the spring-constant 98N/mm and calculate equation 15 one get a natural frequency of 18,17Hz. This frequency is relatively far away from 55Hz and the isolators might be good enough to hinder unwanted interference and resonance. If there is a lot of vibrations and uncontrolled behavior under operation, it would probably be better to choose isolators with a lower spring-constant or change them from four to only two. Two isolators give a natural frequency at 12,85Hz that is most likely better than 18,17Hz.

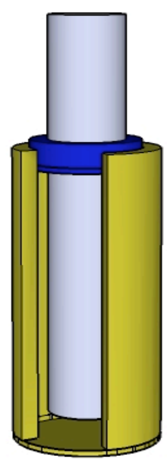

Fig.15. Example on gas-spring

D. The Gas/Pneumatic Cylinder

To be able to compress gas into the gas-spring chamber of the resonator, a gas-cylinder needs to be bought or made. However, it has been hard to discover companies who can deliver gas-cylinders that are able to compress gas from 10 bar and up to 300 bar.

The main issue here is that most companies deliver gas-springs or gas-cylinders with a progression factor "k" lower that 1,5. The progression factor is the relationship between the initial pressure and the end pressure.

$$k = \frac{P_{end}}{P_{init}} \quad (16)$$

The factor is mainly related to the rod that displaces the volume of the cylinder housing, see fig 15. The rod is slightly smaller than the volume, and hence the compression will not increase to the desire pressure.

The best solution would be to use a kind of shock absorber that has a rod that displaces all of the volume inside the

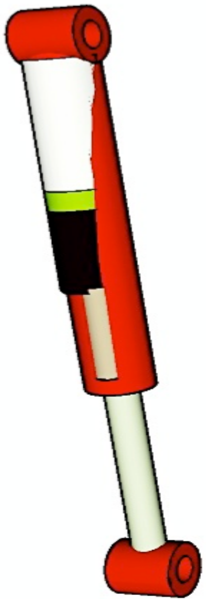

Fig.16. Cylinder with oil-cap.

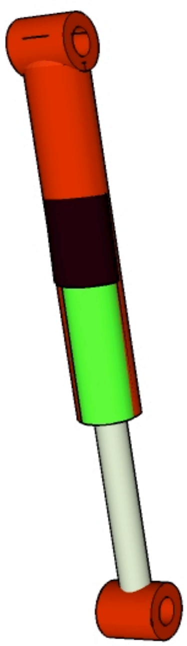

Fig.17. Cylinder with oil.

cylinder housing. Standard pneumatic actuators do this, but the problem here is that the operating pressure often is 5,5 – 6,9 bar [12] and hence not suitable for the test.

Since a gas cylinder or spring could not be found to match the system requirements, the author was conceived of a plan. The concept was discussed with the managing engineer for hydraulic and hard chromium plating Mr. Soelberg at Tromsø Mekaniske AS.

Figure 16 show a standard hydraulic cylinder that could be bought off the shelf. If the hydraulic cylinder is filled with gas, the gas-fluid would leak from one side to the other side of the cylinder piston. This will then leak further out through the gland (sealing-box) in the bottom of the figure.

The solution discussed was to mount a standard cylinder in a vertical position, as the one in figure 16. Then by adding a small amount of hydraulic oil (green colour in figure 16) into the inlet at the top, the oil could then seal the piston. The gas would then be above the oil, and the oil will be forced against the piston by gravity and hence not leak out of the cylinder. A mechanical force could then be applied to the rod to compress and decompress the gas.

The last solution thought of was to use a standard 300bars hydraulic cylinder being compressed with oil (green) from the minus side see figure 17. The solution here is perhaps the best one for this test, since there will not be any leakage between the piston and the cylinder walls and out through the sealing. At the same time the complexity could be less since one would only need to mount two cylinders to the test bench with no mechanical force acting on them. However, requirements for regulation are important in this context because it can be difficult to get an ideal wave motion. Regulation of the two concepts will be discussed later on in chapter 9.

E. The wave

Buford's wind scale [7] in Table 4 shows different wave heights and periods in relation to wind speed. One can notice that the wave period and the wave height, is relatively large when the wind speed increases. This results in a low frequency due to the period, but a slight increase in the float speed. Be also aware that the waves need time, regular wind and open sea to build up.

TABLE IV
OCEAN WAVE PROPERTIES IN DIFFERENT CONDITIONS

Description	Wind speed (m/s)	Wave height (Fully developed) (m)	Wave period (s)
Light air	0.3-1.5	0.08	0.7
Light breeze	1.6-3.3	0.2	1.8
Gentle breeze	3.4-5.4	0.6	3.3
Moderate breeze	5.5-7.9	1.1	6.3
Fresh breeze	8.0-10.7	1.9	7.3
Strong breeze	10.8-13.8	3.4	8.5
Near gale	13.9-17.1	5.3	9.7
Gale	17.2-20.7	8.0	10.6
Strong gale	20.8-24.4	11.3	12.1
Storm	24.5-28.4	15.5	13.9
Violent storm	28.5-32.6	18.0	16.2

Appendix [8] shows us the wind properties at Kråkenes Fyr in Norway. This is known as one of the places where the weather is rough and the wind speed is high. The annual average wind speed from Appendix [6] is 7,8 m/s. This emphasizes from Buford's wind table an average moderate breeze with wave height at 1,1m and a wave period at 6,3s.

With the input values from table 2 and a initial pressure at 10 bar, the simulations show that if one choose a wave period at 6,3 seconds, the time to get a positive net contribution from the machine increases.

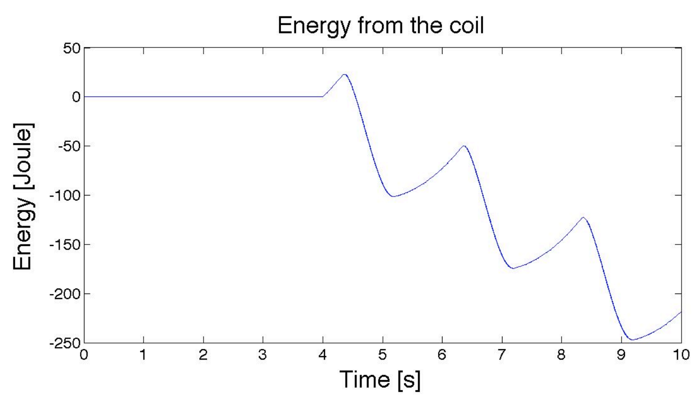

Fig.18. Energy from the coil with a period of 2 seconds (0.5Hz) and 300 bars

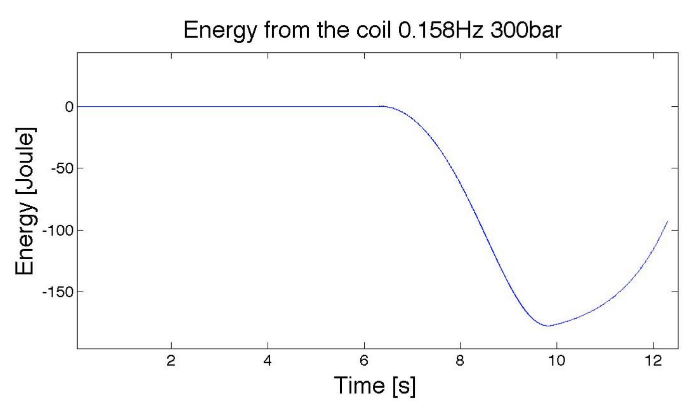

Fig.19. Energy from the coil with a period of 6,3 second

Figure 18 and 19 shows the accumulated energy in the coil. Notice that the figures show different timespans. The real measurement in figure 18 starts at 4 seconds and the real measurement in figure 19 starts at 6,3 seconds. This is because the machine needs to get into steady state before the energy could be measured and hence the effective measurement period for both is only 6 seconds.

One can see from the figures that the energy produced with a long period vs. short is significantly lower. This give us a hint of where there can be most beneficial to place a finished product.

The energy produced from the coil with 2 seconds period is more than 215 Joule after 6 seconds and the one with 6,3 seconds period is only 93 Joule after 6 seconds. Hence there is a desire to increase the frequency of the wave so we can limit the test period due to uncertainties that can occur in the newly designed machine.

VIII. The Choice and Calculations Of The Gas Cylinder

As mention before, it has been difficult to locate a supplier that can deliver gas-cylinders with high progression factor and withstand 300bar of gas pressure. Choosing one of the two solutions in figure 16 or 17 hopefully accommodates this. To find out which gas-cylinders to use, one needs to analyze the problem.

The resonator is rated to 300bar pressure and hence the chosen gas-cylinder must also withstand this pressure.

a) Cylinder and Gas Spring Theory

From the ideal gas law we have [17];

$$pV = nR_{gas}T \qquad (17)$$

for an ideal gas undergoing a reversible (i.e., no entropy generation) adiabatic process is [17];

$$pV^{\alpha} = constant \qquad (18)$$

If we look at figure 11, you can notice three volumes. Hence, the gas-cylinder *"V1"*, the hose *"V2"* and the resonator gas-spring *"V3"*.

If we assume adiabatic process, the relationship between these volumes and the pressure becomes;

$$p_{0,1}V_{0,1}^{\alpha} + p_{0,2}V_{0,2}^{\alpha} + p_{0,3}V_{0,3}^{\alpha} = p_1V_1^{\alpha} + p_2V_2^{\alpha} + p_3V_3^{\alpha} \quad (19)$$

and if we assume the same pressure in the system (no pressure drop between the gas-cylinder and the gas-spring) we get;

$$p_0\left(V_{0,1}^{\alpha} + V_{0,2}^{\alpha} + V_{0,3}^{\alpha}\right) = p\left(V_1^{\alpha} + V_2^{\alpha} + V_3^{\alpha}\right) \qquad (20)$$

The relationship between force, pressure and area;

$$F = P_0A \qquad (21)$$

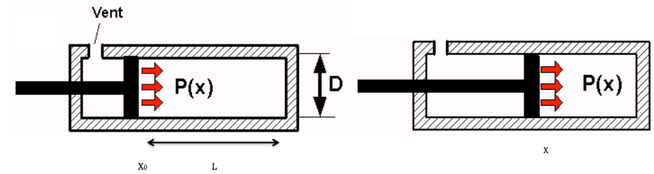

Fig.20. and Fig.21.
Decompressed and compressed gas-spring [16]

Let us assume that a mechanical force is applied to the piston, moving it from x_0 to x (Fig.20. & Fig.21). The volume inside the casing will decrease and thus emphasizes that a higher force needs to keep the piston in the current position. This gives us [5];

$$F = P_0A\frac{V_1}{V_2} \qquad (22)$$

and if we assume an adiabatic process (no heat transfer through the cylinder walls)

$$F = P_0A\left(\frac{V_1}{V_2}\right)^{\alpha} = P_0A\left(\frac{x_{ac}}{x_{ac}-X}\right)^{\alpha} \qquad (23)$$

b) Choice of gas-cylinder

From equation 22 and 23 one can easily see that the force acting on the rod will change in proportion to the piston area, hence depending on the gas constant. For the test there is a desire to keep the force acting on the rod as low as possible. This is because a lot of force will require a relatively big and more expensive hydraulic system. In real life one would not have this issue because of the tremendous force from the wave.

It is important to notice that smaller area of the piston emphasizes that the total volume inside the cylinder will decrease. The only way to cope with a smaller diameter or area is to make the cylinder longer and the consequence would be a longer rod stroke length. It is important to find a cylinder that has enough volume that can be displaced. If the volume of the cylinder is too low, the progression factor of the whole system is too low and hence the pressure in the system will not increase to the desired one.

Appendix [9] shows different standardized hydraulic cylinders that could be bought of the self. These cylinders are not suitable for the test since they are only rated to 200 bars but give us a hint of the dimensions. A 300 bar hydraulic cylinder is often regarded as specialist equipment, and the delivery time can vary from 3 – 6 weeks [23].

To find out which cylinder that should be chosen one need to follow this procedure (be aware that one are only looking at one gas-cylinder, hose and gas-spring at the time);

1. First run the improved script made in Simulink and Matlab. The script can be seen as two pistons that are displacing the volume in the gas-springs and hence compressing or decompressing the gas chamber with a relatively slow speed (figure 22). By choosing the right displacement amplitude in the script one is able to find out when the machine will reach 300bars, see figure 7.

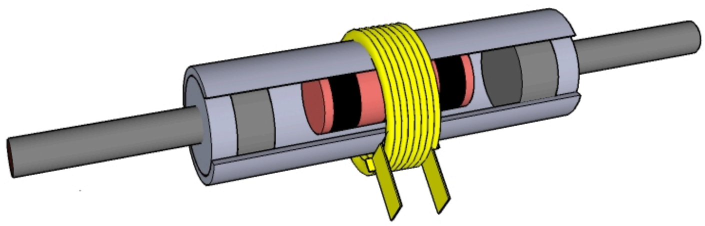

Fig.22. Illustrating picture of the resonator piston-displacement.

2. Before making a new simulation one disconnects the electrical system that is connected to the resonator. The rotor will not move since no current is applied to the coil and therefore it will orient itself into the middle of the machine. When the script runs once again without the electrical system, one can measure the pressure that will be obtained by the displacement inside the gas-springs, see figure 23.

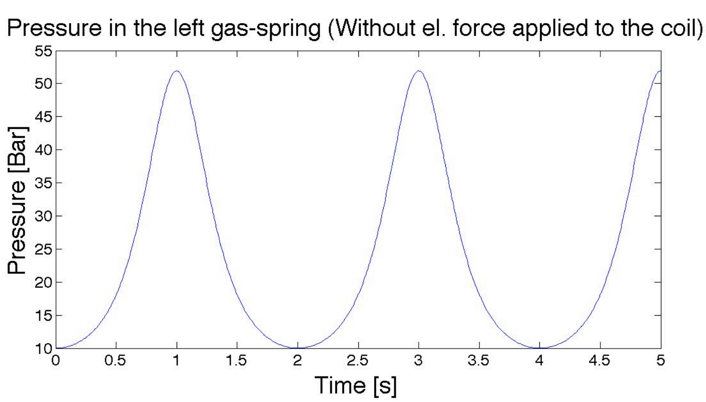

Fig.23. The pressure inside the left gas-spring without electrical force applied to the coil.

Since no current is applied to the coil, the rotor will not bounce back and forth. The maximum pressure inside the gas springs will therefore only become 52 bars when they are fully compressed. This result tells us approximately how much that is needed to compress the total volume (hose, gas-cylinder and resonator gas-spring) with a given initial pressure.

3. A Matlab script has been developed for analyzing how to choose an appropriate cylinder for the test. The script is based on the equations from the "Cylinder and Gas Spring Theory" part. The script does not include mass flow losses that occur in the system. The script can be viewed in appendix [10] and it calculates the pressure, force and ratio in relation to the stroke length of a given cylinder. The initial pressure is set to 10 bar.

From Appendix [9] the cylinder "HC-507-32-20-320" is chosen, and its details will be the input data for the script. Hence;

TABLE V
Part "HC 507-32-20-320" Cylinder details

SPECIFICATION	**VALUE**	**UNIT**
Stroke length	320	mm
Piston effective diameter	32	mm
Rod diameter	20	mm
Pressure	200	bar

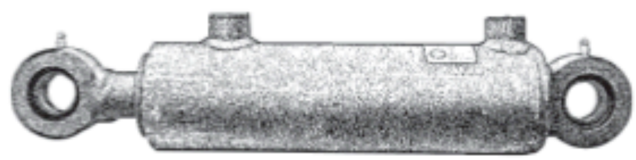

Fig.24. Hydraulic cylinder [18]

The input values give the following results;

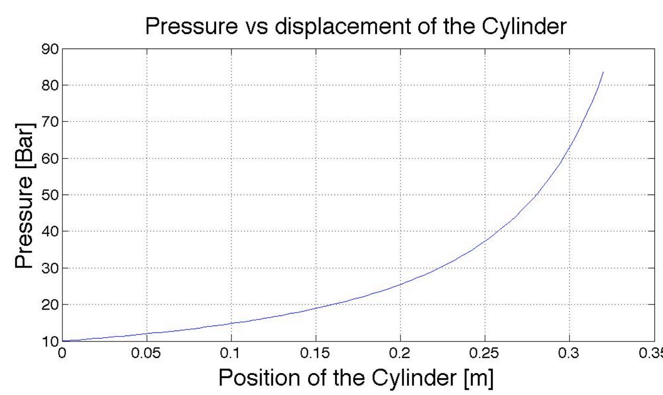

Fig.25. Pressure vs. displacement of the cylinder

One can notice from figure 25 that 52 bars gives a stroke length of the cylinder`s rod to 0,284m. This means that the cylinder is not fully compressed, but sufficiently to meet the requirements for the test of the resonator wave-harvesting concept. One now know the approximately the required stroke length that is needed for the given cylinder.

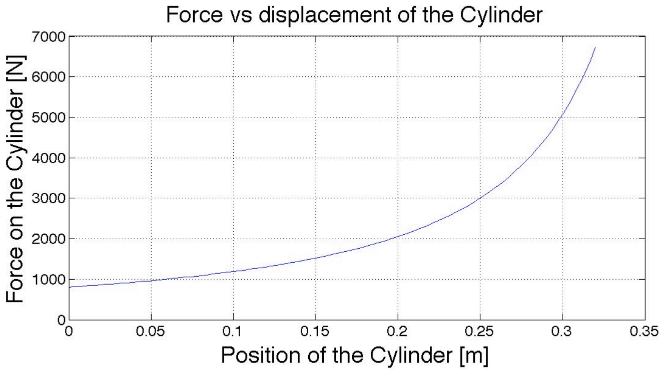

Fig.26. Force vs. displacement of the cylinder

Figure 26 shows the force produced by the effective area of the piston and the pressure acting on it. The force that is needed at 0,284m is 4200 Newton. However, it is important to remember that this is the force needed when there is no electrical current applied to the coil. When the resonator is operating, the peak pressure inside the gas-cylinder will become 300 bar and hence the force acting on the rod needs to be $F = pA = 24115$ Newton. This is a relatively high force but absolutely durable for hydraulic systems.

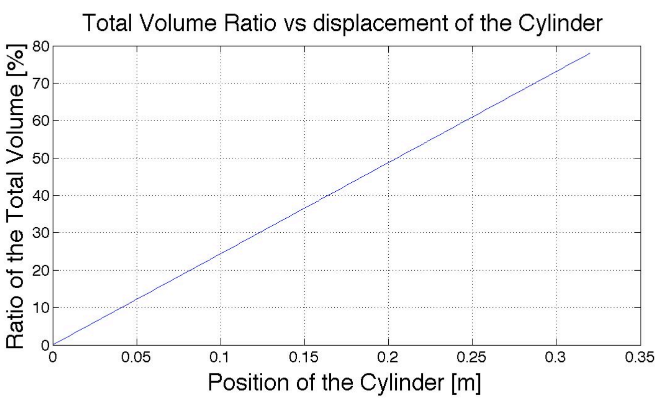

Fig.27. Pressure vs. displacement of the cylinder

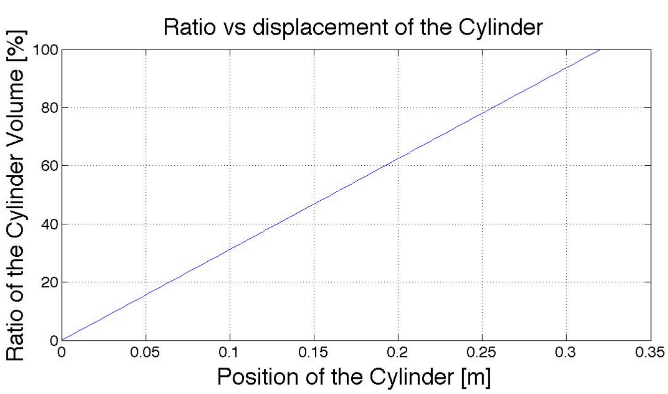

Fig.28. Pressure vs. displacement of the cylinder

Figure 27 and 28 shows the ratio between the displacement of the total volume and the cylinder volume. One can notice that the displacing of the total system is approximately 70%. More simulations of different cylinders can be found in appendix [9].

IX. The High-Force-Low-Speed System

To produce the high force low speed motion (hence the wave), different solutions have been discussed. In the test one wants to have the wave motion as ideal and sinusoidal as possible. However, to be able to produce a smooth high-force-low-speed sinusoidal motion, relative expensive equipment is required.

The easiest and smoothest solution might be to use a powerful electric actuator to compress and decompress the two gas cylinders in figure 12. The electric actuator would then be controlled with a smooth signal from a standard signal-generator or PLS. The benefit of using this solution is the ideal motion, easy controlling, smaller and no pollution associated with oil spills. The disadvantage is that it is quite expensive.

Some investigations have been done, where one was found with a maximum force of 45000N. However, it cannot accelerate fast enough with this force and hence the wave-motion period would be slow. If two electrical actuators were bought, and worked on each gas-cylinder, it might be fast enough to move the rod of the gas-cylinder back and forth with a period of 2 seconds and a stroke length at 320mm. Appendix [10] shows one supplier of electrical actuators that could be used.

The other solution and perhaps the best one might be to use a hydraulic system. Hydraulic systems are well developed, they are often very reliable, standardized couplings/hoses and could be cheaper. The author conducted several investigations, where several companies were contacted and where different hydraulic solutions, both cheap and expensive were discussed.

It is important to remember that a cheap hydraulic system does not produce a smooth sinusoidal motion, but a more triangular motion. This is often because the hydraulic system is pumping the oil to the hydraulic cylinders with a constant rate of flow. When the cylinder reaches the ending point, an electric driven valve changes the direction of the oil-flow forcing the cylinder in the opposite direction. To cope with this a rather expensive proportional or servo valve could be connected between the cylinder pipelines/hoses and the hydraulic pump.

If one implements a triangle wave motion source in the Matlab-Simulink model, this gives us a quite similar curve as the one in figure 11 but with slightly less energy produced. From figure 11 (The sinusoidal wave motion), one can see that we get approximately 70 Joule per cycle and with the triangle only 50 Joule. This makes sense since a sinusoidal motion-curve has a bigger area and hence contains more energy than a triangular.

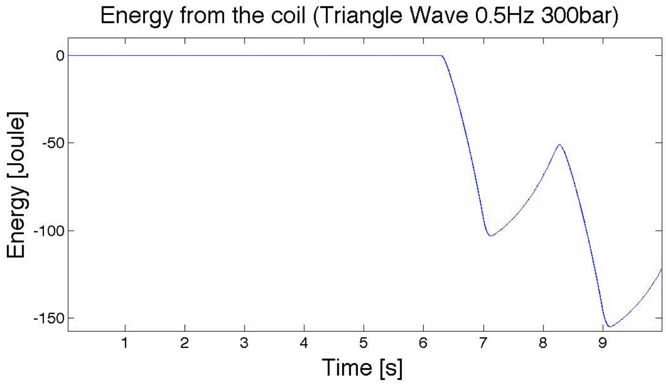

Fig.29. Energy from the coil with a period of 2 seconds

Two examples are proposed here to generate the wave-movement for the prototype 2B machine. Both include a relative commercial hydraulic system.

1) Example 1

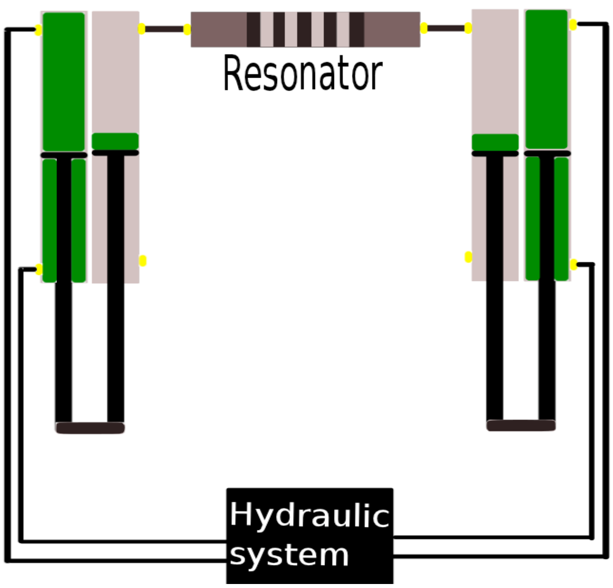

Fig.30. Two cylinders with oil cap being compressed by two hydraulic cylinders and it's corresponding control system.

Figure 30 shows us the principal scheme of four hydraulic cylinders and one hydraulic control system. The outer cylinders are mechanically connected to the inner "pneumatic" gas cylinders. The outer cylinders are driven by a controlled hydraulic system so one obtains the ideal sinus/triangle wave-motion.

The outer hydraulic cylinders will have a higher cylinder diameter than the inner. Hydraulic components have a tendency to be quite expensive when the rated pressure is more than 250 bar [23].

2) Example 2

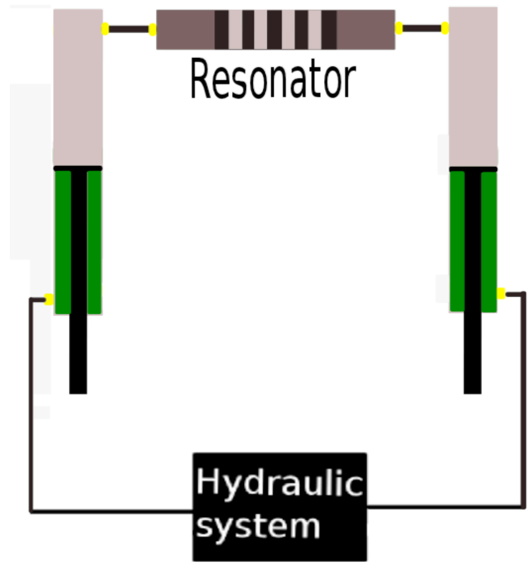

Fig.31. Two cylinders being compressed by a controlled hydraulic system.

Figure 31 shows two hydraulic cylinders operated by a hydraulic system. This is a fairly simple system, but needs a more comprehensive control system. As the pressure will increase to 300 bar on the upper part of the cylinder, a relatively high force will occur and tries to push the piston and rod downwards. This will be important to control since the downward motion of the piston and rod from the top can have a significantly high velocity at the beginning, but a rather slow velocity from half way to the end.

Since the maximum pressure will become 300 bar inside the upper part of the cylinder, the hydraulic system needs at least to be rated to 300 bar. This emphasizes higher costs for the hydraulic system.

3) The hydraulic control system design

There have been several discussions with different suppliers about how to design a proper hydraulic control system for the resonator. One supplier of hydraulic components for agricultural machinery was contacted and asked to design the hydraulic system. At first the work went smoothly, but after a few weeks with work we realized the complexity of the system and had to drop the cooperation.

PhD Pingju Li at Resonator has several of years with experience from designing hydraulic systems. In cooperation with him some basic calculations and investigations about how to control the hydraulic system have been completed.

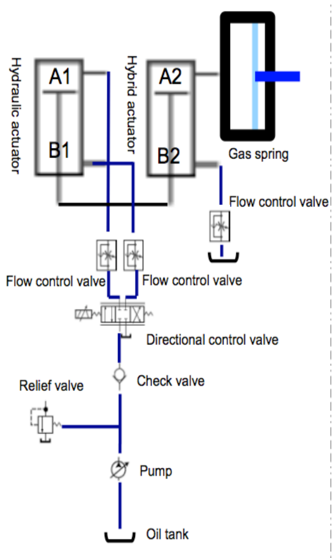

Fig.32. Configuration 1

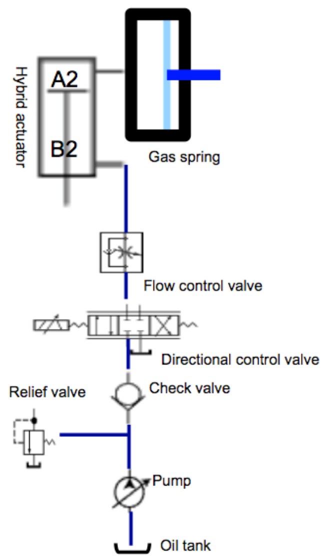

Fig.33.Configuration 2

Figure 32 and 33 show us the how the system could be designed.

Configuration 1 uses a directional control valve to control the flow of the oil into the cylinder. Three flow control valves have been placed onto the cylinders to adjust the speed of the hydraulic pistons. The hydraulic actuator has a piston diameter of 50mm. When the system runs (compressing the hybrid actuator), the maximum pressure will be slightly different on each side of the piston due to the piston rod. The piston rod takes up a part of the effective area at B1, the resulting area ratio A1/B1 shows that the area B1 is significantly lower than A1. Calculation shows that the max pressure will be 125 bar on A1 and 195 bar at B1. Hence the price of the hydraulic components could be relatively cheap.

The main difference between configuration 1 and 2 is that configuration 2 uses only one control valve. At the same time the directional valve, pump, flow control and check valves needs to be rated to a higher pressure than in configuration 1. Calculations shows that if one choose a cylinder with piston diameter of 32mm and a stroke length of 250mm, the pressure in B2 needs to be almost 500 bar. This configuration is possible to perform, but the price for the components will be expensive.

Both of these configurations should have a relief valve on the gas-cylinder (hybrid actuator), so the gas pressure does not increase above 300 bar. This is mainly to protect the resonator as it is only rated to 300.

More specific descriptions, calculations, pros and cons can be found in appendix [18].

Ph.D. Li at Resonator has looked deeper into how to control configuration 2 and the motion of the actuators behave. When the machine is running, the rotor will bounce back and forth inside the resonator, hence changing the pressure rapidly in each gas springs. To be able to have a smooth motion of the hybrid actuator, the oil that flows into the actuator needs to be smoothly controlled with the same frequency as the rotor. Ph.D. Li says that this can be solved with a rather expensive proportional or servo valve that has a very high accuracy and response time. The rated pressure of the valve must be at least 500 bar and the response time should be or better;

$$\tau = 1/\,(\max freqency\ of\ rotor) = \frac{1}{200Hz} = 0{,}005s \qquad (24)$$

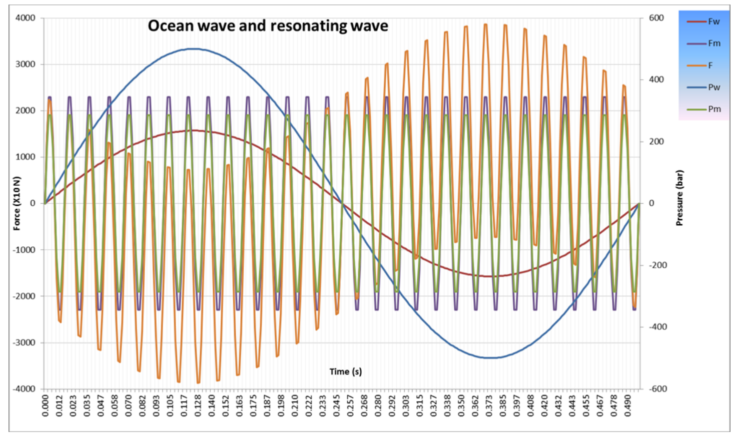

Fig.34.The forces and pressure produced by the wave and rotor movement under operation for configuration 2 (at lowest frequency, hence 50Hz). See bigger picture in Appendix [18].

Figure 34 shows the simulation of the configuration 2. The wave period is 2 seconds and a rotor frequency is 50 Hz. The general force from the hydraulic system ("F", orange color), will be needed to changes rapidly due to the resonating rotor. The best way to handle this problem would be to use a valve that has a high enough response time, or one can perhaps oversize the hydraulic system so the impact from the rotor will have minor influence.

X. THE ELECTRICAL CONTROLLING

If the machine is going to run smoothly and in resonance, then the electrical controlling is crucial. Requirements to make the test successful are listed in table 6 and are mainly based on the simulations done in this paper.

TABLE VI
REQUIREMENTS FOR ELECTRICAL CONTROL

PARAMETER	VALUE	UNIT
Voltage	100-150	V
Current	15	A
Frequency	55-200	Hz
FEATURES		
Regenerative	Needed	
Feedback Analog sensor-coil	Needed	

1) *How the machine should be controlled and operated*

Different solutions have been discussed with Resonator AS about how to operate and control the machine. These have mainly been;

a) *Apply a constant sinusoidal voltage that is limited to +- 130 V with the natural frequency of the system, but only when the gas-springs are being compressed. The overshooting voltage is hence rectified and fed back to the source/control system. When the decompression is happening, the control system switches the source off and does not feed electrical energy to the coil. The electric duty cycle will hence be 50% of the wave.*

b) *Apply a constant sinusoidal voltage that is limited to +- 130 V with the natural frequency of the system, but only when the gas-springs are being decompressed. When the compression is happening, the control system switches the source off and does not help the rotor to resonance. The movement from the rotor will hence feed electric energy back to the source. The electric duty cycle will hence be 50% of the wave.*

c) *Apply a constant sinusoidal voltage that is limited to +- 130 V, with the natural frequency of the system. The overshooting voltage is hence rectified and fed back to the source/control system.*

d) *Apply a voltage pulse that is limited to +- 130 V, with the natural frequency of the system. The overshooting voltage is hence rectified and fed back to the source/control system.*

Suggestion a and b will be discussed here, while c and d are very similar to that which was simulated earlier in [5] and the simulation from section 6 in this paper. Solutions c and d will only be briefly commented upon. For your support; an 0.5Hz (2 sec period) wave motion example can be looked at in appendix [13]. All of the simulations are done with the given period.

Solution a)

The maximum pressure is set to 300bar and the gain for the proportional controller had to be adjusted to 42x to get a reasonable result from the machine.

The machine runs into steady state, and at 5 seconds one disconnect the source and let the current flow directly back through the rectifier and further on to a braking resistance for measuring. At 6 seconds one connects the source to accelerate the rotor into resonance and steady state. The cycle repeats itself with a two second period. See appendix [13] for wave motion.

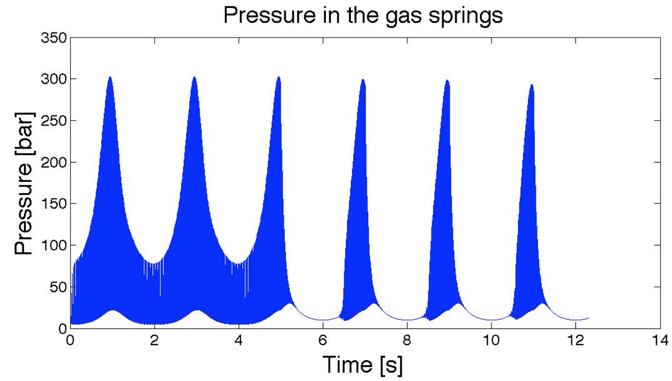

Fig.35. Pressure in the gas spring when switching of supply

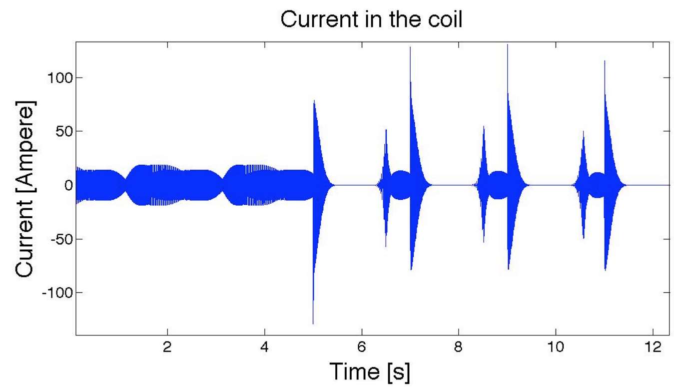

Fig.36. Current in the coil

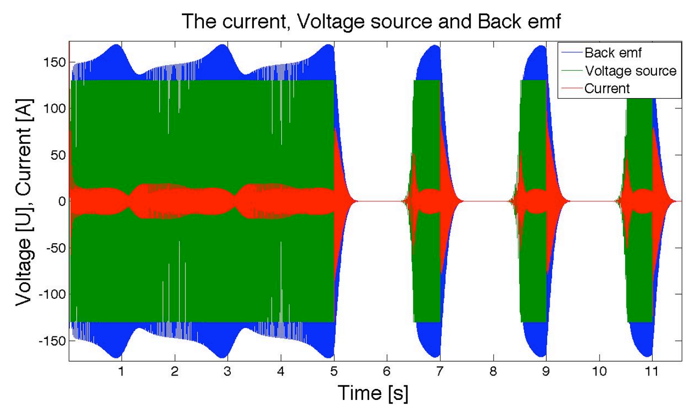

Fig.37. The current, voltage source and back emf

Figure 36 and 37 shows us the current, voltage source and the back emf from the coil. From the figures one can see that the peak currents in the circuit are quite high. The high peak current is a result of the voltage source that is being turned off by the controller. At this moment the coil contains a lot of energy and will now try to discharge itself through the resistance in the system. The coils in the resonator and the electrical regulating equipment must therefore be designed to withstand this current.

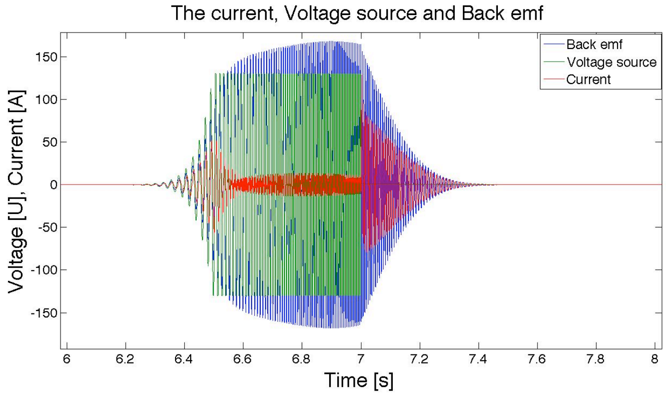

Fig.38. The current, voltage source and back emf

Figure 38 shows one cycle (one wave). The wave starts to compress the gas-springs at 6 seconds, and decompress at 7 seconds. See appendix [14] for the position and speed of the rotor. One can notice that it takes 0,3 seconds before the rotor starts to move. The reason for this is because the source has been disconnected for one second before the cycle starts. One can easily see from appendix [14] that the velocity of the rotor drops to zero around 7,4 seconds, and since the proportional controller is following the speed, the supply will also drop to zero and hence the rotor will be at rest. It could perhaps be better to superimpose a high voltage pulse at the start of every cycle to help the rotor to resonate.

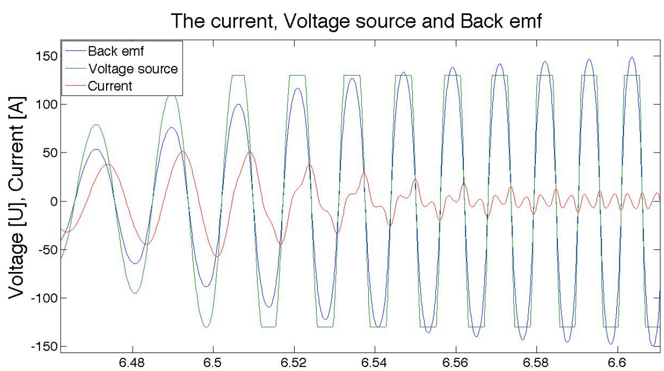

Fig.39. The current, voltage source and back emf (start)

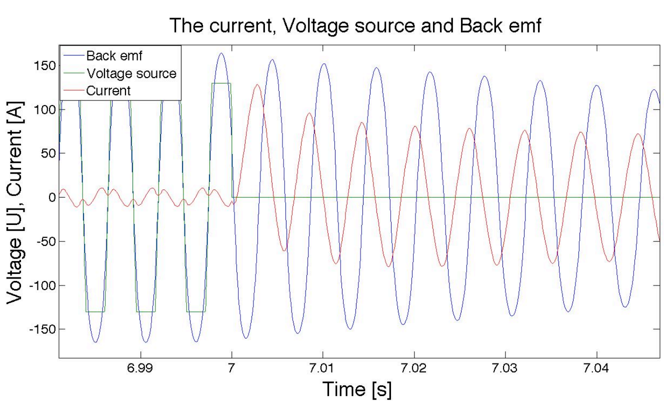

Fig.40. The current, voltage source and back emf (end)

Figure 39 and 40 show us a part of the beginning and end of the electrical cycle. As seen from the figures, the voltage source is saturated to ± 130V. This is because the voltage will increase towards infinity with the current proportional controller if no saturation point is set. The disadvantage of having the presented simple controller with saturation function is that it produces harmonics in the system and hence losses. This could perhaps be coped with, if the control system makes a smooth sinusoidal voltage so the peaks are not chopped. These harmonic should be investigated further if this solution is chosen.

Figure 39 shows the back emf starts to catch up the voltage source. When the back emf reaches a higher voltage than the source, the current starts to go in the opposite direction and hence the machine operates as a generator and produce power. From the figures it is worth noticing that the voltage and current are phase-shifted. This implies that the machine, now acting as a generator, is producing a relatively high amount of reactive power.

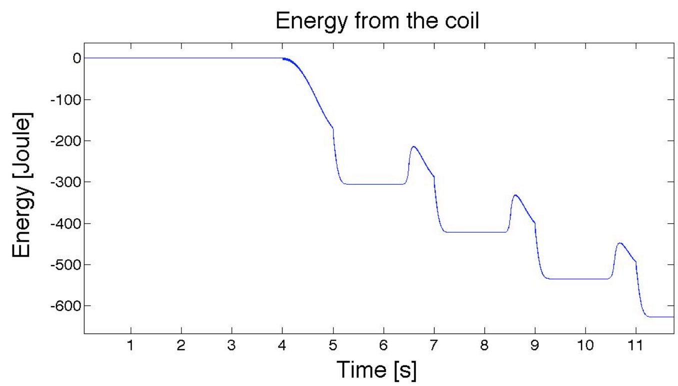

Fig.41. Energy from the coil

Figure 41 shows us the accumulated energy from the coils with the given control regime. One can notice that the machine is producing more than two times as much energy (500Joule, at 10 sec) compared to the simulations done in the first part of this paper (figure 11). The maximum frequency is also measured to 180Hz that is slightly less than 200Hz.

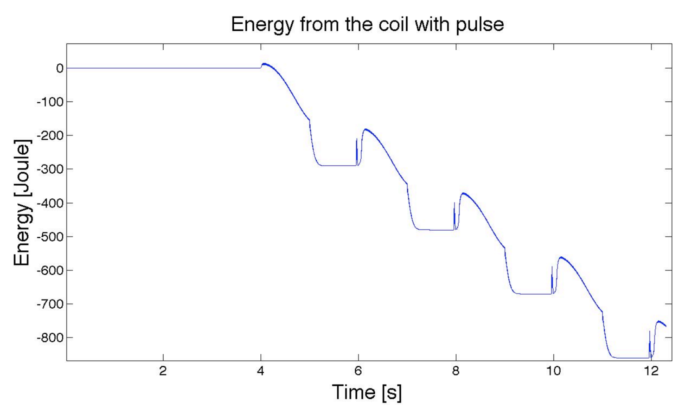

Fig.42. Energy from the coil with a start pulse

Figure 42 shows the accumulated energy from the coil with a superimposed pulse right before the compression of the springs. One can see from the figure that if the pulse of 100V is applied right before the compression happens, one can get approximately 200 Joule more energy within the same timespan. This emphasizes a power of 108W, which is considerably higher than the simulations done in chapter 6.

From appendix [15] one can notice that the peak current rises to 350 Ampere due to the pulse, and this needs to be accounted for when designing the control system and coils.

Solution b)

The pressure is set to maximum 300bar, and the gain has been chosen to 47x after several simulations.
The machine runs into steady state, and at four seconds when the gas-springs start to be compressed by the high-force-low-speed motion one disconnect the source. See appendix [13] for wave motion. When the wave is on it`s retract after five seconds, the source is connected again and forcing the rotor to resonate again.

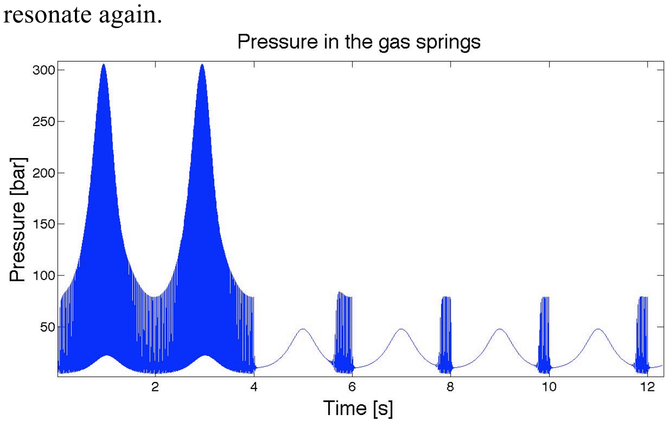

Fig.43. The Pressure in the gas-springs.

From figure 43 one can see the pressure in the gas springs as a function of time. It is apparent to notice that the pressure does not rise to 300 bars although the source is turned on at five seconds. The reason for this is the high potential energy, and hence forces from the gas-springs. If one increases the gain in the proportional controller, the pressure increases but the drawback is the start-current in the coil that also increases.

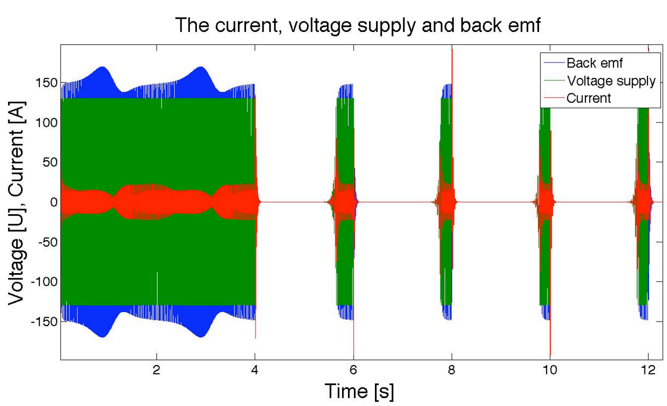

Fig.44. The current, voltage supply and back emf.

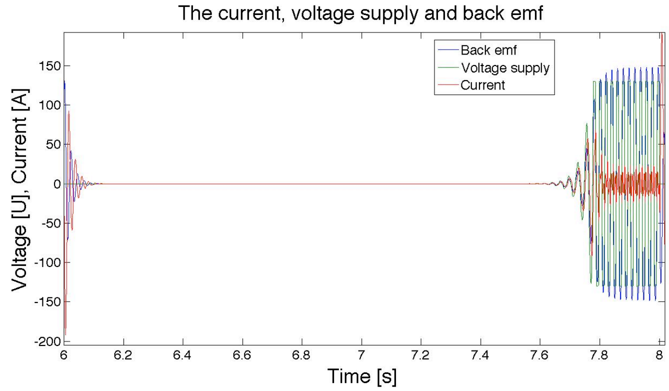

Fig.45. The current, voltage supply and back emf (zoomed).

Figure 44 and 45 shows the current, voltage supply and back emf as a function of time. The peak current from the figures is also a consequence of the controller turning off the voltage source. The current is a bit higher that in figure 37 and 38 and the main reason for this is the instant velocity reduction of the rotor due to the start of the compression coincident with no electrical supply from the source. See appendix [16].

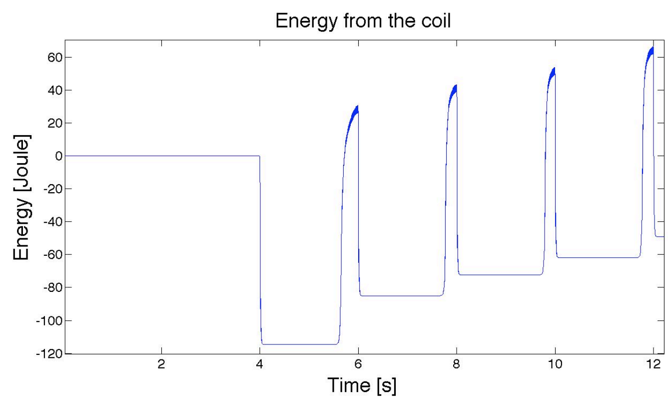

Fig.46. The energy from the coil

Figure 46 shows the accumulated energy from the coil. One can instantly see when the compression of the springs takes place, since the energy spikes downward on the graph. At this time one get a lot of energy fed back to the source. After the energy is fed back, the rotor is at rest and does not consume or emit energy. At 5,6 seconds the coil starts to consume energy. One can see from the figure that it consumes more energy than it feeds back to the source. This emphasizes that if the current control solution is chosen, the machine will in the long term consume net energy and hence not act as desired.

Solution c) and d)

Simulations completed in chapter 6 show approximately how the machine will behave to the following suggestions. The main difference from these two will be that solution d) will produce more harmonics than solution c) in the system. The energy that can be extracted from the coil will differ, but not remarkably.

2) The choice of controller

From the simulations done in this paper, one can notice that we extracts most energy from solution a) with a superimposed pulse. This is a relatively comprehensive control regime, but absolutely solvable for the experienced person.

To prove that it is possible to extract a surplus of energy when a high-force-low-speed motion impacts the resonator, an easy proportional controller with feedback could be used. One can then expect results that are similar to the simulations done in chapter 6.

<u>*Vacon frequency converter*</u>

There has been located a controller that has the ability to perform the desired requirements listed in table 6. The controller is a sophisticated frequency converter with PID-regulation that has been developed by the company Vacon. See figure 47. The controller or hence the inverter that it is often called, have several I/O gates that makes it able to control and run the resonator. Some of the specifications are listed in table 7, the rest can be viewed in appendix [17] and ref [19].

Fig.47. Vacon NXP FR6 [19]

TABLE VII

VACON NXP "0048 2 A 2 H 1 SSS"

PARAMETER	VALUE	UNIT
Input Voltage	208-240	V
Input Power 10% Overload at 230V	11	kW
Output Voltage	0-240	V
Output Current Continuous/+40 °C	48	A
Output Current Maximum	62	A
Output Nominal Frequency	8-320	Hz
Max Breaking Current	21,7	A
FEATURES		
Regenerative	Yes, Break chopper/resistor	
Feedback	Yes, several Input/output - A/D	
Programmable	Yes	

a) Startup of the resonator

The script made in this paper starts the rotor by applying a relatively high voltage pulse at the start. This pulse is applied to the coil and forcing the rotor to bounce into the desired direction. The proportional controller then takes over and feed the coil with a signal proportional to the velocity of the rotor.

The chosen inverter has several programmable startup procedures. Hence, ramp, s-curve ramp, pulse and flying start.

b) The feedback control

It has not been decided which position sensors are going to be used in the machine. However, the inverter has two analogue input gates, and here the selected sensors be connected. The position is very important to measure since this will tell the controller when to apply the right signal to help it resonate with the desired amplitude.

c) The Brake chopper (Regenerative)

When the frequency converter is decelerating the motor, the inertia of the motor and the load are fed into an external brake resistor. This enables the frequency converter to decelerate the load with a torque equal to that of acceleration (provided that the correct brake resistor has been selected) [19]. The current and voltage through and over the resistor can then be measured, and hence the energy could be calculated. It is worth noticing that the maximum breaking current is 21,7 Ampere for the inverter. Simulation from figure 9 shows that the current should not be higher than 11,5 Ampere, and hence this inverter is suitable for this test.

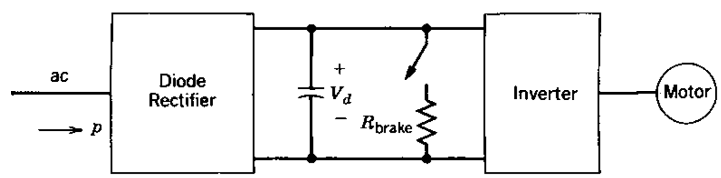

Fig.48. Principle of a brake chopper (braking resistance) [22]

<u>*Frequency converter with DSP controlling*</u>

The PhD Candidate Shugjun Zang with the Department of Electric Power Engineering at NTNU has made a Dynamic model, estimations, regulation and Control design of the linear electric machine. His main focus has been on designing a more complicated control strategy for the linear machine. He has therefore been working on DSP controlling to achieve better performance and more advanced control with auxiliary functions. This work has resulted in a homemade control system that has been tested on the prototype 2. The test was successful.

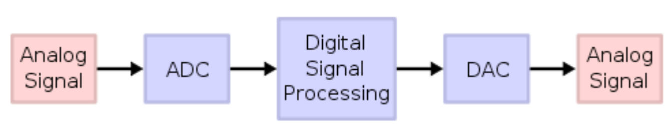

Fig.49. Basic scheme of the DSP-controller [20]

Since the DSP control system gives the resonator more opportunities for an optimal and smooth control than the Vacon controller, this could be beneficial if another control solution is chosen as opposed to the simple proportional controller in chapter 6. Mr. Zang has confirmed that his control system could work with the resonator if he changed some hardware components and had written program for the

DSP. "If a Simulink control scheme program existed, this would be very easy to convert into a program for the DSP controller"[21].

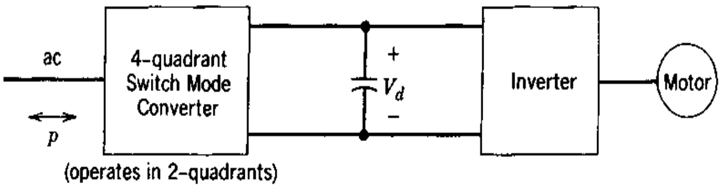

Fig.50. Regenerative Braking [22]

The system Mr. Zang made for the prototype 2 does not have a regenerative part. Figure 48 and 50 shows the basic scheme of two different solutions for this task. The first one is using a breaking resistor that is similar to the Vacon controller, and the second is a 4-quadrant switch that can be used if one wants to send the energy back to the grid. These solutions are well known and a lot of suppliers sell these components.

XI. Energy Considerations

The specific dimensions on the gas cylinder are from table 5 and are used in the following simulations and calculations.

A. *The energy needed from the wave*

Simulations performed in chapter 8, shows that we need to move the piston rod 0,284m to obtain 300bar.

Fundamental equations;

$$Work = Force * deltaX \quad (24)$$

$$Work = pressure * Area * deltaX \quad (25)$$

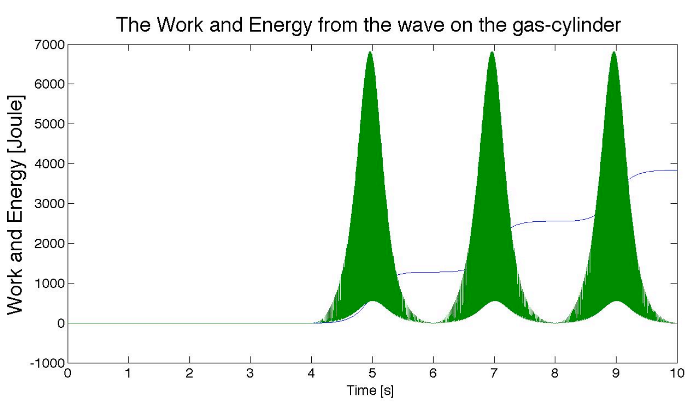

Fig.51. The work and accumulated energy needed from the wave.

Figure 51 shows the work and accumulated energy that is needed to compress one gas spring in the resonator. The simulation set up uses the easy proportional controller. We can see from the figure that the maximum work needed is 6820 Joule, and the accumulated energy is 3836 Joule. The electrical net energy with the proportional controller is 220 Joule (see chapter 6) and hence the theoretical efficiency of the system becomes;

$$\eta = \frac{E_{el}}{2{*}E_{wave_{needed}}} = \frac{220Joule}{2{*}3836Joule} * 100\% = 2{,}87\%$$

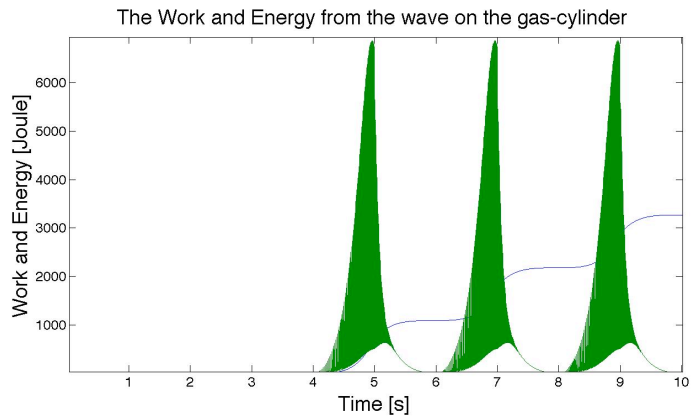

Fig.52. The work and accumulated energy needed from the wave.

Figure 52 shows the work and energy from solution a) with a superimposed pulse, hence chapter 10. We can see from the figure that the maximum work needed is 6820 Joule, and the accumulated energy is 3836 Joule. The theoretical efficiency of the system becomes therefore;

$$\eta = \frac{E_{el}}{2{*}E_{wave_{needed}}} = \frac{655Joule}{2{*}3270Joule} * 100\% = 10.02\%$$

XII. Conclusion And Further Work

The model made in [5] has been improved and adapted to the new linear machine "Prototype 2B". The "Prototype 2B", hence the resonator is expected to be completed in July 2011. Simulations show that it is possible to extract a surplus of energy from the system when a high-force-low-speed motion impacts on the resonator. The model has been slightly improved, but has still weaknesses that need be handled. The simulations are very time consuming since different adjustments and set points like gain, pressure, voltage source and wave-amplitude needs to be adjusted and simulated each time.

The flux density is set to a mean value of the "prototype 2B" machine. This approximated value originates from simulations performed by A.Dahl-Jacobsen at Resonator AS. However, one knows that the flux density distribution is not homogenious over the whole pole pitch, and therefore should FEM simulations be performed and the results should be implemented in the model.

The "Prototype 2B" has mainly been developed and designed to satisfy the specifications of the Wassera -50 & 80 driven hammer in ROP. The relationship between mass, pressure, number of turns and voltage must be investigated to locate the optimal solution for this concept.

The electrical control of the machine seems to be crucial, since small changes to the control regime will have huge impacts on the potential energy that could be extracted. By using the simple proportional controller, the simulations show that we can only extract 33W. By doing a simple modification to the control, hence applying a 50% electric duty cycle and a startup pulse we could extract 108W. There exists several opportunities in how to regulate the machine. One could for example combine the 50% duty cycle and also superimpose a small constant disturbance the whole time to help the machine resonate.

In addition the harmonics and reactive power should be investigated. Simulations show that one get harmonics if we saturate the proportional controller so that the sinusoidal voltage is chopped. The impacts of these harmonics have not been determined and should be investigated. The current and voltage is phase-shifted and this results in reactive power consumed and extracted from the machine. This event must also be investigated to make the machine as efficient as possible.

The Vacon drive should be chosen to prove the concept of harvesting energy from waves. This inverter is quite sophisticated and will be able to run as a proportional controller and hence do the work. The crew at Resonator AS and the supplier of Vacon can assist the set up and connection of the inverter to the machine.

The test bench has been constructed using different steel profiles to make it solid. The resonator is clamped to a steel-bed, and isolated from the surrounding environment. The energy would then be conserved in the system and not released to the surroundings. The steel-bed is suggested to rest and mounted upon rubber isolators that would satisfy the requirements. This has to be tested in real life to verify that they will withstand the vibration forces and frequencies.
To transfer the high-force-low-speed energy to the resonator two different solutions have been suggested. Both of the solutions could work, but to ensure that no gas leaks out of the cylinder, then the solution in figure 17 is probably the best choice. To find out which cylinder should be used, the self-composed Matlab script in this paper could be used.

The high-force-low-speed source has been a challenge. At first one though this was an easy and cheap configuration, but after several weeks with planning and investigation, the opposite was confirmed. One of the main reasons for the difficulty was due to ignorance and economic considerations. A lot of time has been used to find a cheap but durable solution for the high-force-low-speed source and several suppliers have been contacted.

The best choice is perhaps to use a strong electric actuator that can easily be controlled to emulate the wave. However, this was not investigated further due to the cost of the system.

Two hydraulic system proposals have been suggested in this paper. Here configuration 2 should be chosen due to it`s simplicity. After some simulations and investigations in cooperation with Ph.D. Pingju Li, the main problem became the regulation of the hydraulic system. Simulations show that the force produced by the resonating rotor needs to be accounted for when the wave is being emulated. This can most likely be solved, by choosing a relatively expensive directional valve. The solution has to be investigated further since an emulated wave source that is not controlled properly, can be crucial for the net electrical energy extraction.

It has been mentioned among the professors and PhD that the non-linear compression event in the springs could be proportional to the electrical energy that can be extracted from the machine. A few simulations were performed and the event could not be confirmed. Simulations showed [Appendix 2] that one could extract more energy when the pressure increased in the springs, but the non-linear event could not be recognized. More and deeper investigations should be done on this area.

The efficiency of the system has been calculated, and if the proportional controller is used, the maximum theoretical efficiency with the specifications of the cylinder in table 5 is only 2,87%. By doing simple changes to the control regime, the efficiency seems to increase more than 10%. The relatively low theoretical efficiencies that are obtained by the simulations, tell us that the method of harvesting energy from a linear machine with gas springs is probably not profitable. Further work should be to investigate how much theoretical net electrical energy that is possible to extract from the machine, and hence the control strategy needs to be optimized. If the efficiency is not remarkable higher after the investigation, then the concept should be shutdown.

The test should be performed in a secure room due to high gas pressure in the system. Resonator AS has bought a reinforced container that is meant for test purposes and the container is located at the University of Ås.

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[23] PMC servi Cylinderservice, RISSA.

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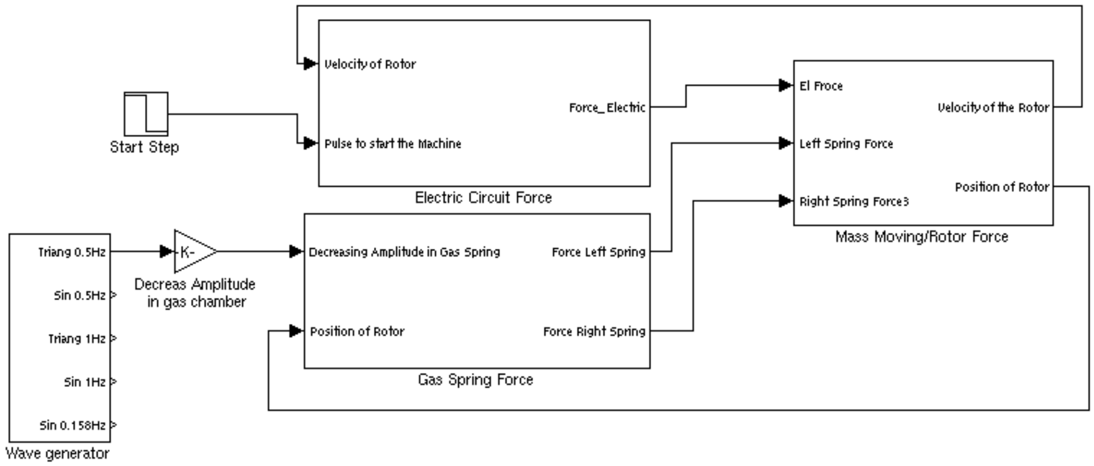
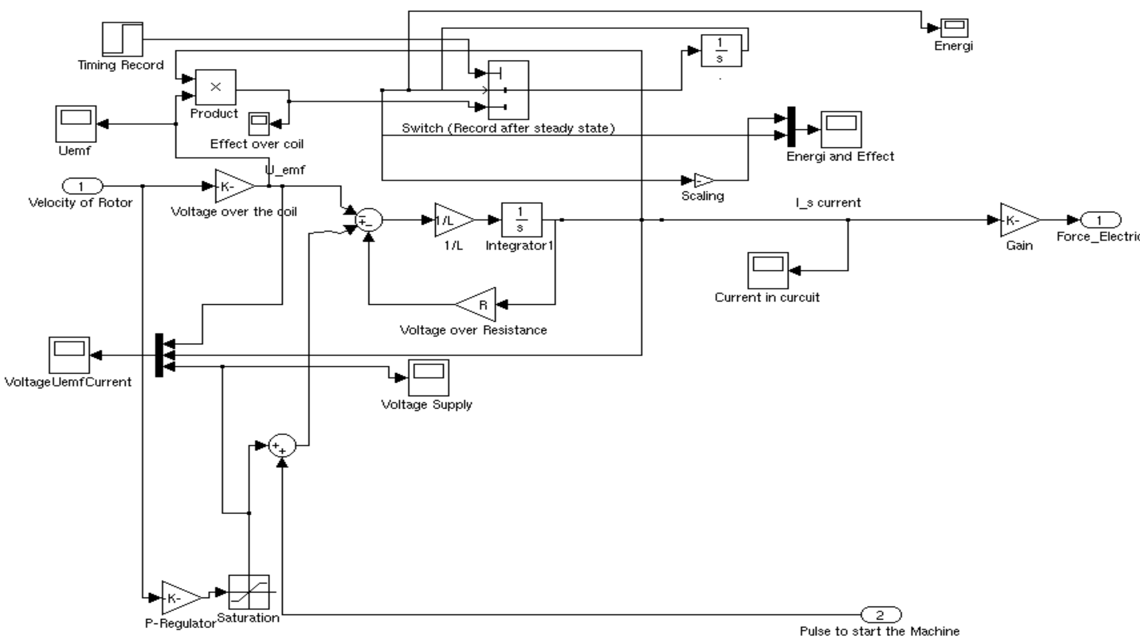
He is currently heading Professor in Energy Conversion Group at Department of Electrical Power Engineering, NTNU. His research interests include Electromagnetic filed calculations and design of electrical machines. The third paragraph begins with the author's title and last name (e.g., Dr. Smith, Prof. Jones, Mr. Kajor, Ms. Hunter). List any memberships in professional societies other than the IEEE. Finally, list any awards and work for IEEE committees and publications. If a photograph is provided, the biography will be indented around it. The photograph is placed at the top left of the biography. Personal hobbies will be deleted from the biography.

APPENDIX 1

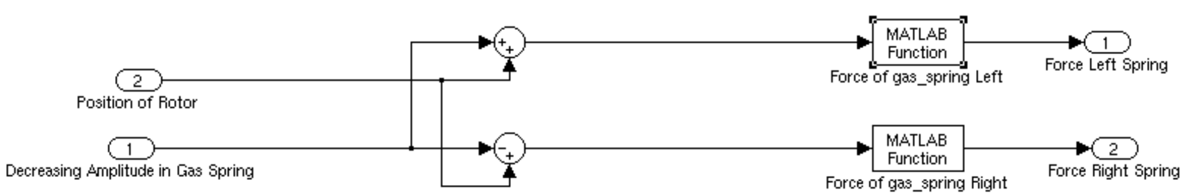
The system is build up of three main subsystems, hence electrical circuit, force from gas springs and the dynamic force from the moving mass.



The electrical circuit



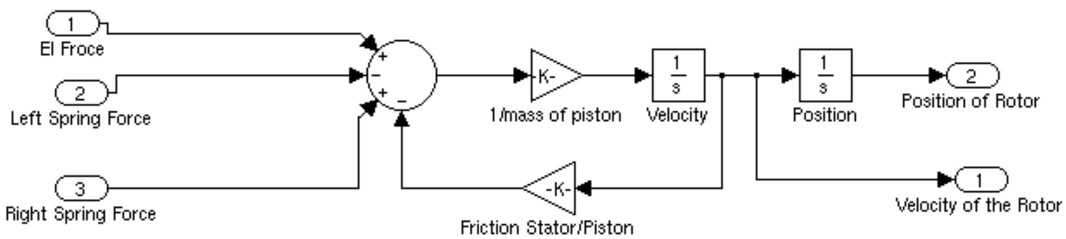
Force from gas springs



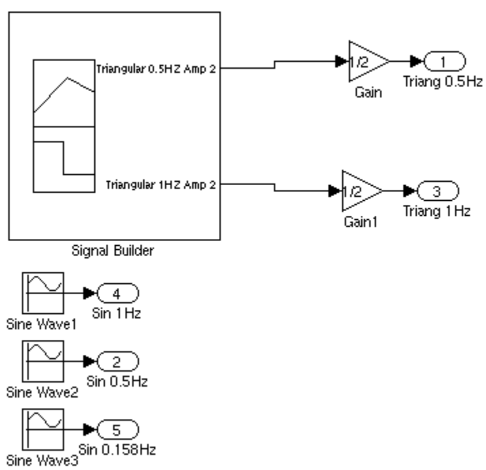
Matlab Functions of the gas springs

```
%GAS_SPRING (Left Side)
function [ force ] = gas_spring( u, Area , Length_camber, gas_const,P0 )
force = P0*100000*Area*(Length_camber/(Length_camber - u))^gas_const;
end

%GAS_SPRING (Right Side)
function [ force ] = gas_spring( u, Area , Length_camber, gas_const,P0 )
force = P0*100000*Area*(Length_camber/(Length_camber + u))^gas_const;
end
```

Dynamic force from the moving mass



Wave Generator

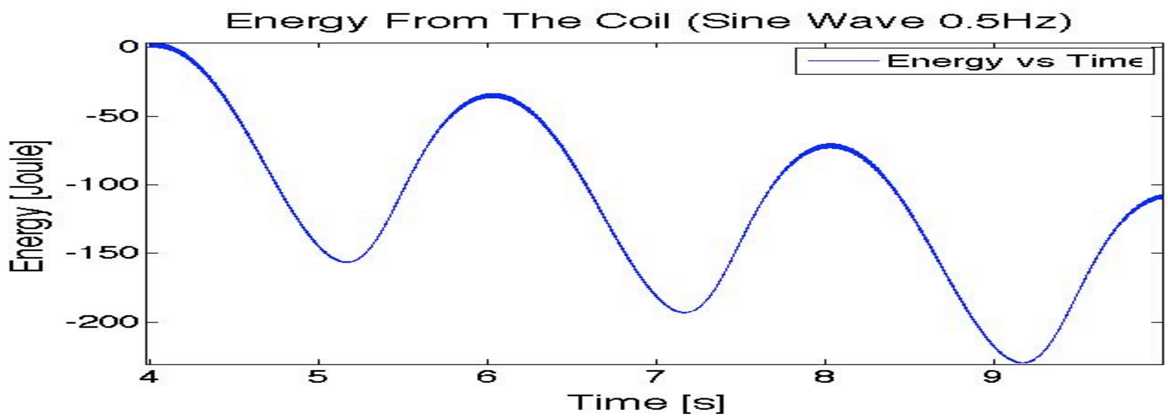


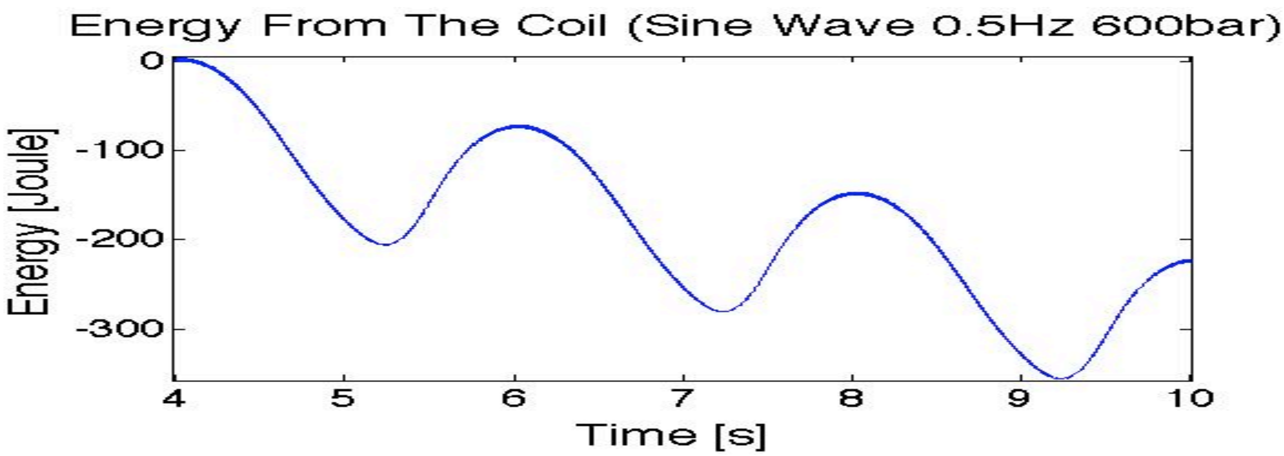
Input Values in the Model

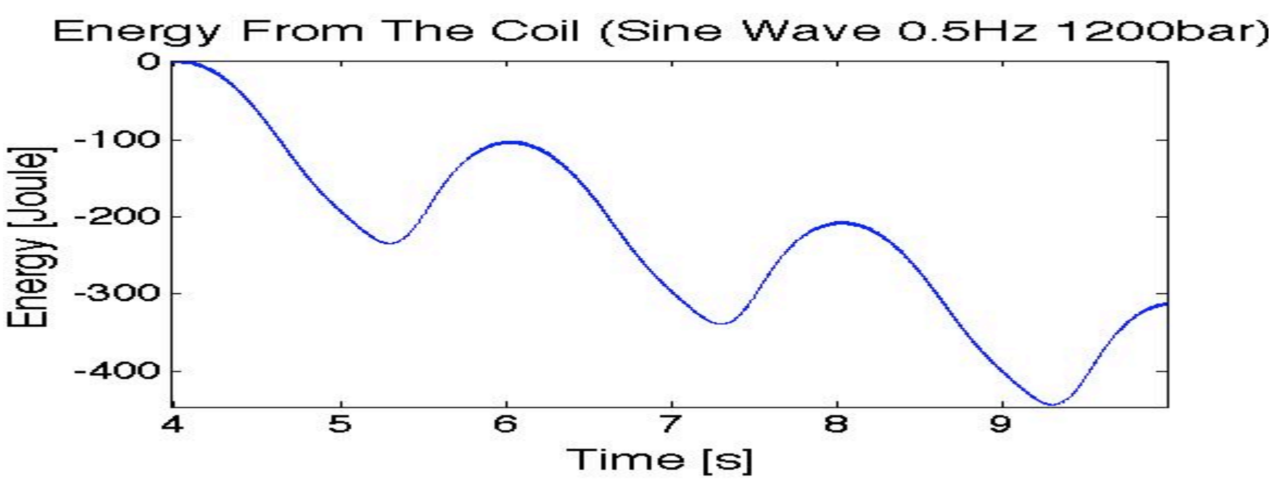
```
clear all

kinternal=0 ;              % [N/m]     Equivalent mechanical spring
BMagnet=0.3;               % [Tesla]   Flux density, rotor
dm=0.056;                  % Mean coil diameter
%dm=0.0476;                % Diameter of rotor chamber
N=540;                     % Turns of stator coil
mp=8.541;                  % mass of piston/rotor
fricp=0.001;               % friction constant between piston and stator
R=0.4;                     % Resistance of coil
L = 1.8e-3;                % Inductance of coil

%% The Gas Spring
Ds = 0.063;               %Diameter of gas spring
P0 =10;                   %Pressure [bar]
Area = pi*(Ds/2)^2;       %Area of the pistn [m^2]
gas_const= 1.4;           %Gas constant (Nitrogen/Air = 1.4)
Length_camber = 0.024;    %[m]
```

APPENDIX 2

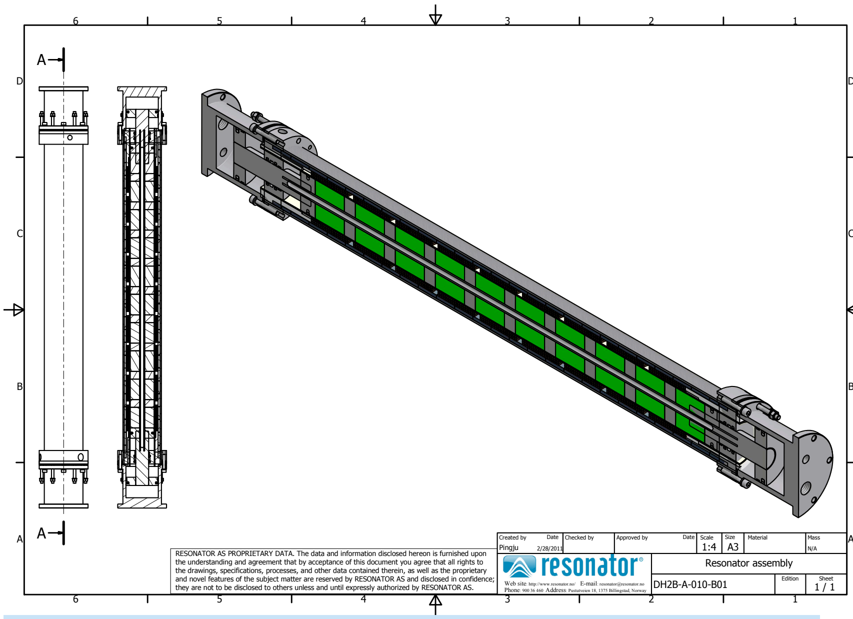
Input values from table 2 are used in the simulations and the gas springs are compressed simultaneously by an external source to the desired pressure. The pressure in the first figure is maximum 300bar.







APPENDIX 3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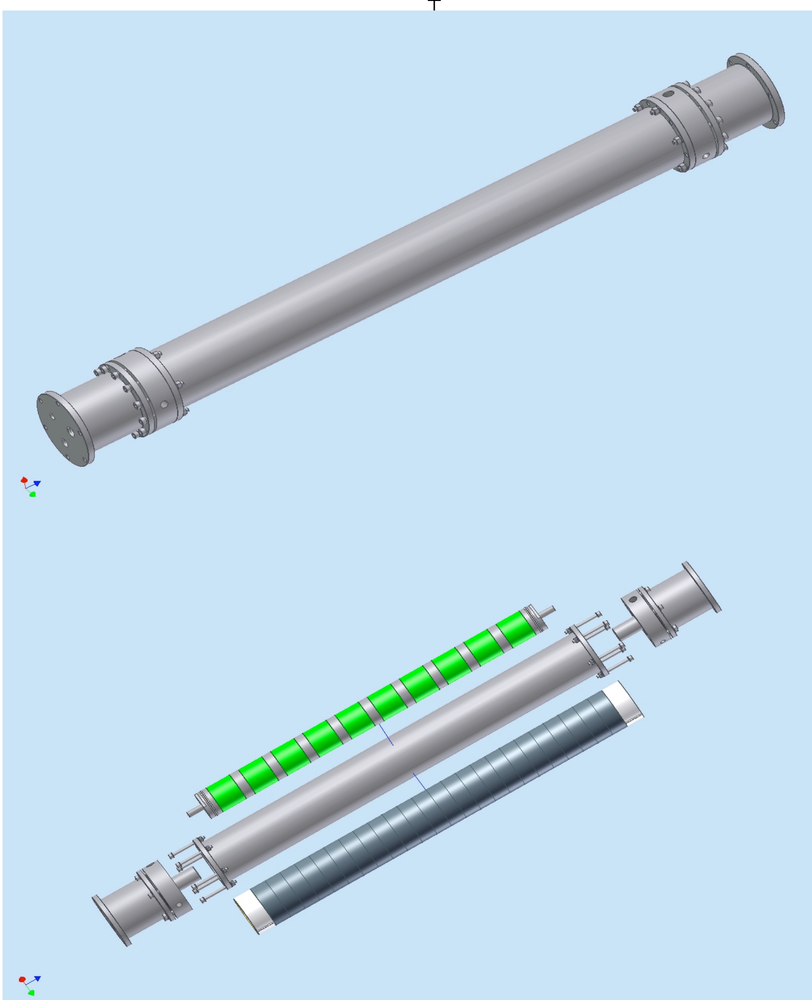


APPENDIX 4

How Does An LVDT Work?

Figure 2 illustrates what happens when the LVDT's core is in different axial positions. The LVDT's primary winding, P, is energized by a constant amplitude AC source. The magnetic flux thus developed is coupled by the core to the adjacent secondary windings, S_1 and S_2. If the core is located midway between S_1 and S_2, equal flux is coupled to each secondary so the voltages, E_1 and E_2, induced in windings S_1 and S_2 respectively, are equal. At this reference midway core position, known as the null point, the differential voltage output, (E_1 - E_2), is essentially zero.

As shown in Figure 2, if the core is moved closer to S_1 than to S_2, more flux is coupled to S_1 and less to S_2, so the induced voltage E_1 is increased while E_2 is decreased, resulting in the differential voltage (E_1 - E_2). Conversely, if the core is moved closer to S_2, more flux is coupled to S_2 and less to S_1, so E_2 is increased as E_1 is decreased, resulting in the differential voltage (E_2 - E_1).

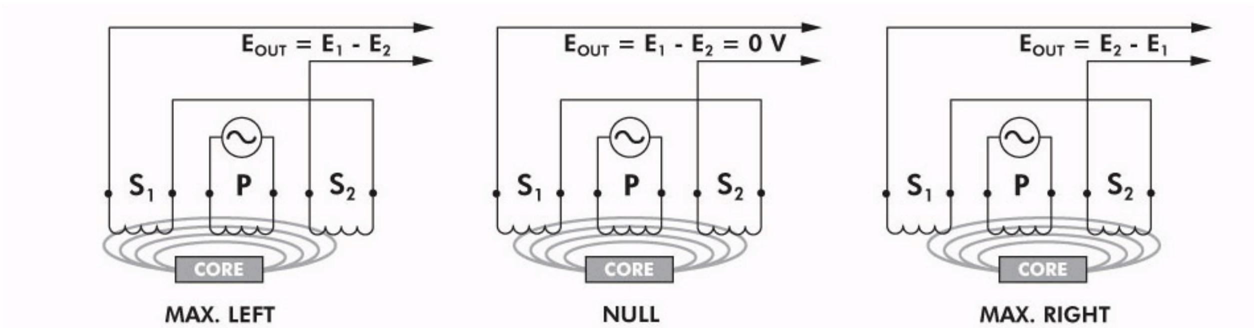


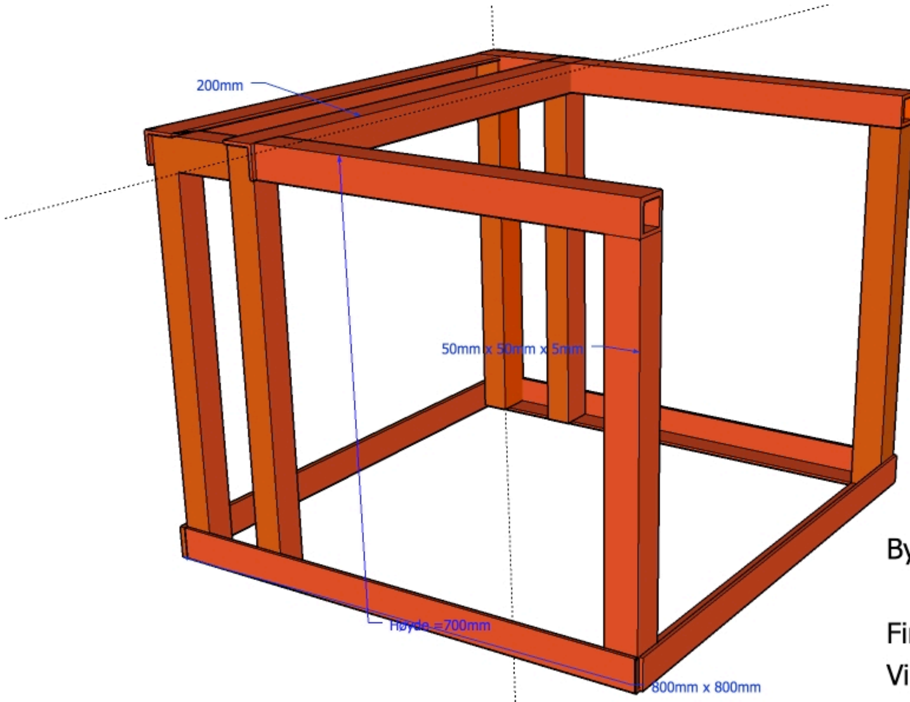
Figure 2

Figure 3A shows how the magnitude of the differential output voltage, E_{OUT}, varies with core position. The value of E_{OUT} at maximum core displacement from null depends upon the amplitude of the primary excitation voltage and the sensitivity factor of the particular LVDT, but is typically several volts RMS. The phase angle of this AC output voltage, E_{OUT}, referenced to the primary excitation voltage, stays constant until the center of the core passes the null point, where the phase angle changes abruptly by 180 degrees, as shown graphically in Figure 3B.

This 180 degree phase shift can be used to determine the direction of the core from the null point by means of appropriate circuitry. This is shown in Figure 3C, where the polarity of the output signal represents the core's positional relationship to the null point. The figure shows also that the output of an LVDT is very linear over its specified range of core motion, but that the sensor can be used over an extended range with some reduction in output linearity.

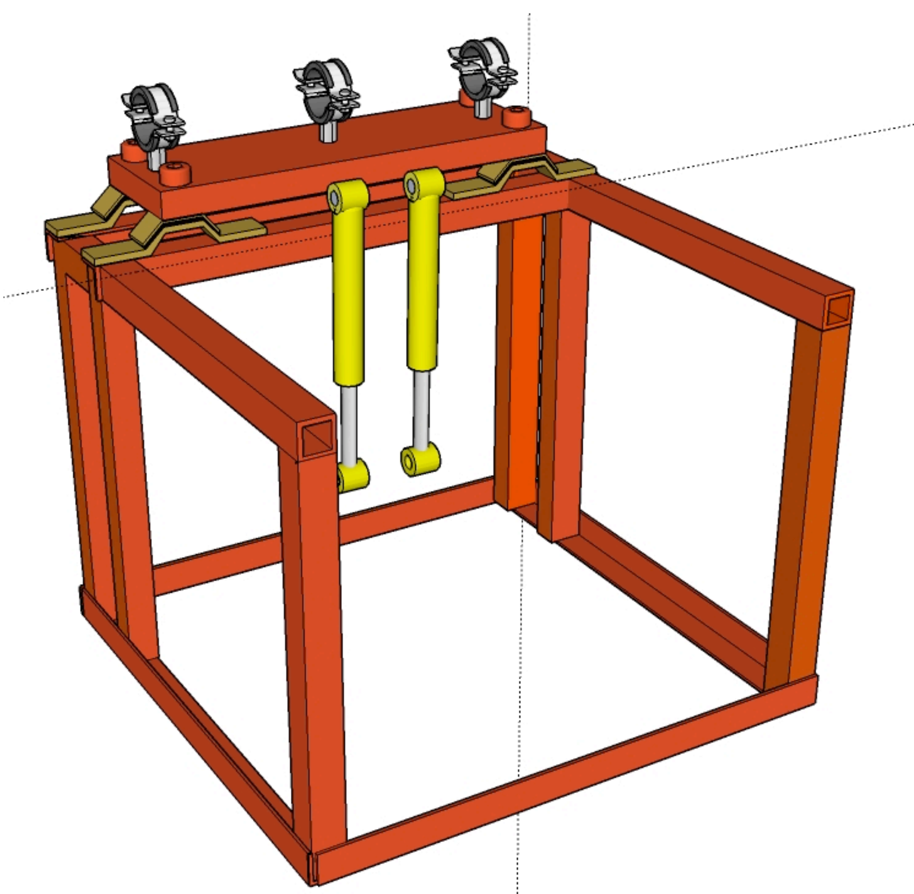
http://www.macrosensors.com/lvdt_tutorial.html

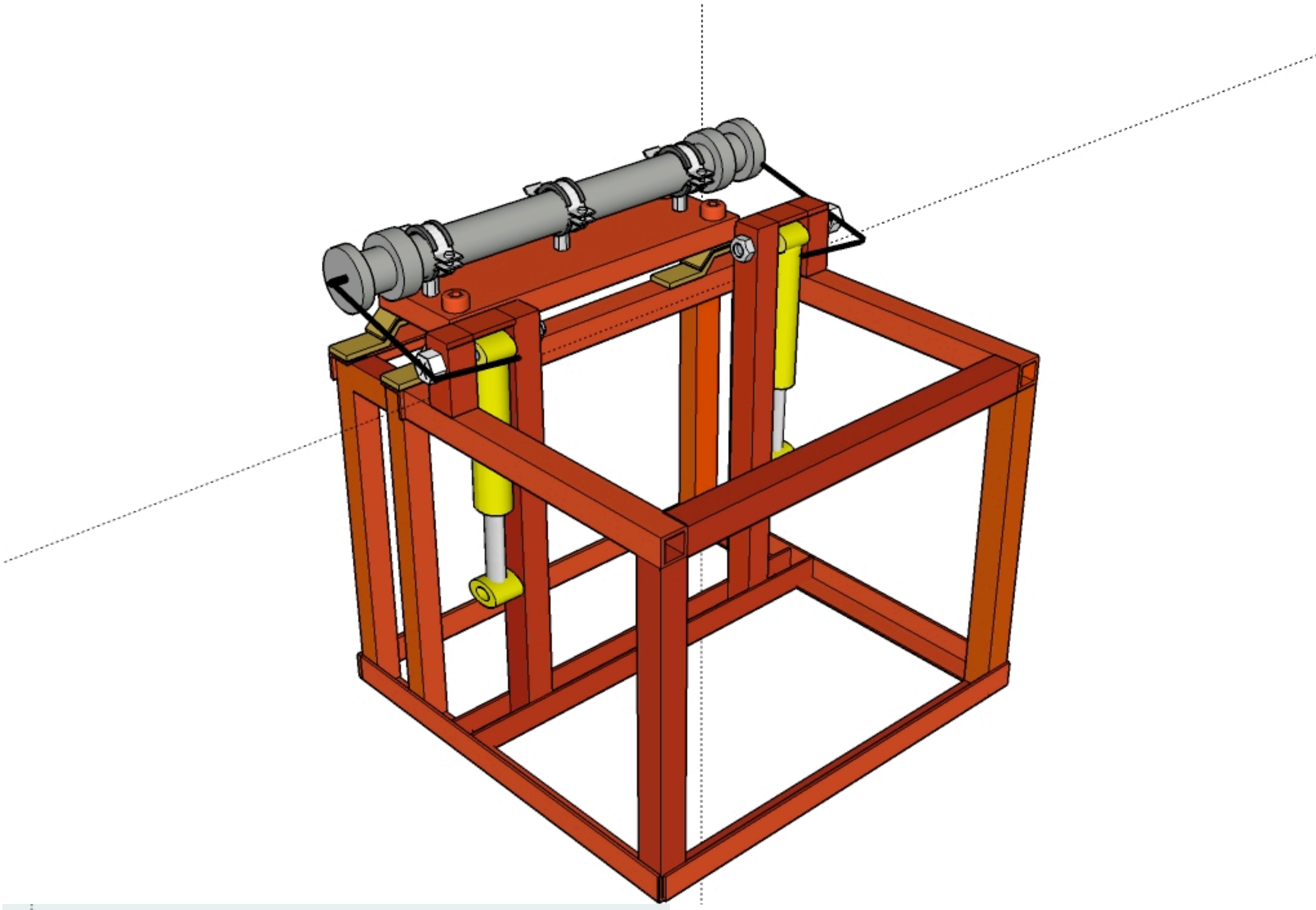
APPENDIX 5

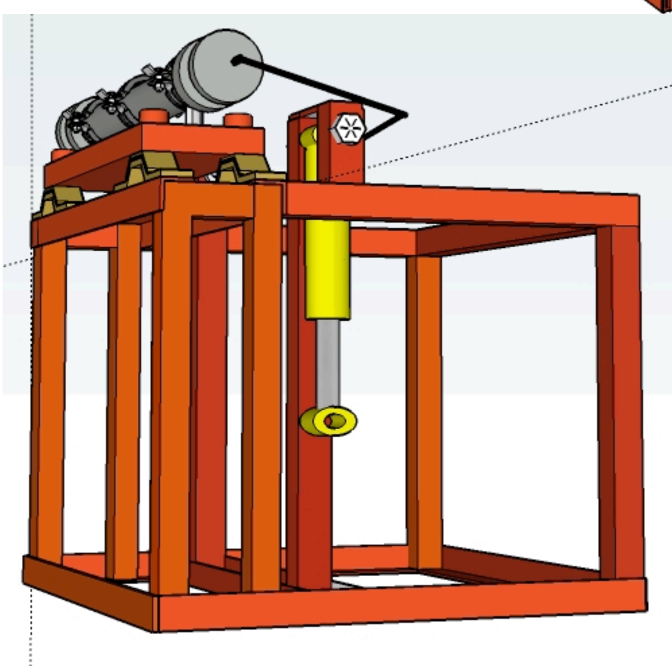


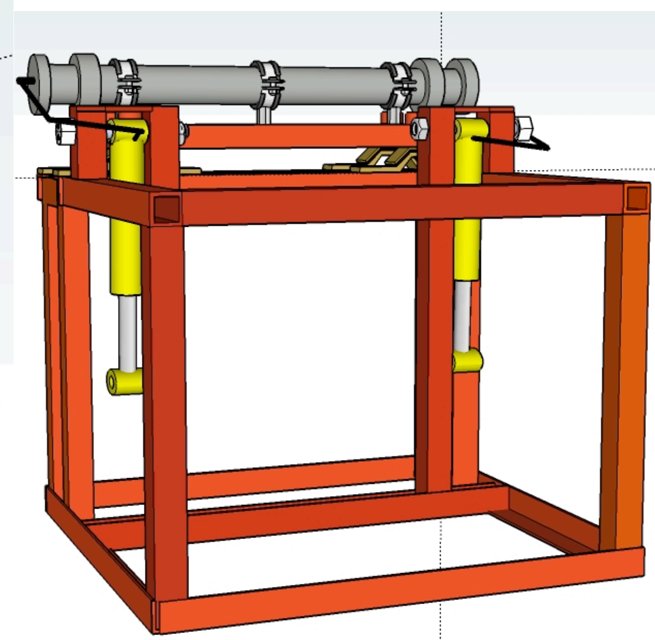
Bygges av:

Firkantstål 50x50x5mm
Vinkelstål 50x5mm









APPENDIX 6

Røroppheng

Skruklammer med dempgummi, kraftig

- Dempgummi EPDM temp.bestandig -50 °C til +110 °C
- Kraftig elforsinket klammer mod. HDPCI
- NU - max belastning sikkerhetsfaktor 1:1 - 200 kg **(14- 65 mm)**

350 kg **(67- 116 mm)**
500 kg **(117- 171 mm)**
850 kg **(177- 225 mm)**
1000 kg **(226- 316 mm)**

NRF nr.	Dim. mm	Tommer	Anslutning	Forpakning	Pris u.mva	Enh	RG	LS
353 23 56	14 - 18	3/8	M 8/ M 10	50	**30,30**	STK	YC	*
353 23 57	19 - 23	1/2	M 8/ M 10	50	**31,80**	STK	YC	*
353 23 58	24 - 28	3/4	M 8/ M 10	50	**33,70**	STK	YC	*
353 23 59	29 - 33		M 8/ M 10	50	**35,80**	STK	YC	*
353 23 61	33 - 37	1	M 8/ M 10	50	**37,70**	STK	YC	*
353 23 62	40 - 45	1 1/4	M 8/ M 10	50	**40,20**	STK	YC	*
353 23 63	47 - 52	1 1/2	M 8/ M 10	50	**42,70**	STK	YC	*
353 23 64	53 - 58		M 8/ M 10	50	**44,70**	STK	YC	*
353 23 65	60 - 65	2	M 8/ M 10	50	**48,70**	STK	YC	*
353 23 66	67 - 72		M 10/ M 12	25	**56,60**	STK	YC	
353 23 67	73 - 78	2 1/2	M 10/ M 12	25	**67,10**	STK	YC	*
353 23 68	79 - 85		M 10/ M 12	25	**70,00**	STK	YC	
353 23 69	88 - 93	3	M 10/ M 12	25	**72,50**	STK	YC	*
353 23 71	94 - 99		M 10/ M 12	25	**75,50**	STK	YC	
353 23 72	100 - 106		M 10/ M 12	25	**79,50**	STK	YC	
353 23 73	108 - 116	4	M 10/ M 12	25	**89,40**	STK	YC	*
353 23 74	117 - 123		M 10/ M 12	25	**107,00**	STK	YC	
353 23 75	124 - 129		M 10/ M 12	10	**119,00**	STK	YC	*
353 23 76	131 - 137		M 12/ M 16	10	**152,00**	STK	YC	

Klammer, to-delte SSG

- Primet rød
- Leveres uten skrue og mutter

NRF nr.	Dim. mm	Skrue dim.	Forpakning	Flatt stål	Pris u.mva	Enh	RG	LS
353 51 49	20 - 22	6 x 20	10	40 x 4	**37,70**	PAR	YC	*
353 51 51	26 - 28	6 x 20	10	40 x 4	**47,20**	PAR	YC	*
353 51 52	33 - 35	10 x 40	10	40 x 4	**50,60**	PAR	YC	*
353 51 53	40 - 43	10 x 40	10	40 x 4	**54,40**	PAR	YC	*
353 51 54	48 - 49	10 x 40	10	40 x 4	**56,60**	PAR	YC	*
353 51 55	57 - 61	10 x 40	10	40 x 4	**55,00**	PAR	YC	*
353 51 56	75 - 77	10 x 40	10	40 x 4	**84,90**	PAR	YC	*
353 51 57	88 - 90	10 x 40	10	40 x 4	**92,00**	PAR	YC	*
353 51 58	110 - 116	16 x 60	10	50 x 6	**127,00**	PAR	YC	*
353 54 14	139 - 141	16 x 60	10	50 x 6	**138,00**	PAR	YC	*
353 51 59	165 - 170	16 x 60	10	50 x 6	**160,00**	PAR	YC	*
353 51 61	216-222	20 x 80	5	60 x 8	**312,00**	PAR	YC	*
353 20 51	267 - 276	20 x 80	1	60 x 8	**408,00**	PAR	YC	*
353 20 52	321 - 327	20 x 80	1	60 x 8	**464,00**	PAR	YC	*
353 20 53	403 - 410	24 x 80	1	70 x 10	**773,00**	PAR	YC	*

www.bd.no

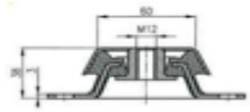
I would recommend the red steel clamps since they can be welded to the steel-bed. This is because there will be a tremendous vibrating force produced by the Resonator and the red ones will be stronger.

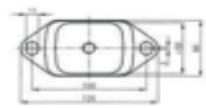
APPENDIX 7

Motorlabber elBe 130

elBe 130

Hardhet (Shore)	Stivhet (N/mm)	Laster Kg Min	Max	Gjenger
45	98	17	40	M12
50	157	25	60	M12
60	216	40	90	M12
70	294	60	120	M12





Vis større bilde...

Art. nr.: elBe 130

Motorlabber

elBe

Elastisk vibrasjonsisolator for vekter fra 11 kg til 890 kg.
Disse elementene gir en høy grad av isolering ved valg av rett type og hardhet på gummi, egenfrekvens
fra ca 7,5 Hz.
Elbe har myk vertikal stivhet, men er avstivet i horisontalretning, egner seg derfor godt for
bruk pÅ båter. Sikret mot at gummi og ståldeler kan skilles fra hverandre.
Utførelse i forsinket stål/naturgummi. elBe 400 leveres også i rustfritt stål/ naturgummi.

Leveres med egnet bolt for høydejustering av motorer etc.

http://www.marine-industrisalg.no/index.php?valg=produkt&gruppe_nr=134&produkt_nr=elBe%20130

Motorlabber elBe 130

Art. nr.: elBe 130

Pris: 435,-

Detaljer Velg antall: 1 Legg i handlekurven

Motorlabber elBe 400

Art. nr.: elBe 400

Pris: 550,-

Detaljer Velg antall: 1 Legg i handlekurven

Other suppliers:
http://www.avproductsinc.com/
http://www.vibrationisolationproducts.co.uk
http://www.marineshop.no/CBC.aspx?q=c:100500

Appendix 8

Wind Properties From Kråkenes Fyr

Måneder	Vind	
	Gjennom-snitt	Sterkest vind
apr 2010	8,4 m/s	25,7 m/s 7. apr
mai 2010	5,7 m/s	19,9 m/s 11. mai
jun 2010	7,7 m/s	24,4 m/s 19. jun
jul 2010	8,6 m/s	26,3 m/s 5. jul
aug 2010	6,2 m/s	26,0 m/s 22. aug
sep 2010	7,0 m/s	22,4 m/s 23. sep
okt 2010	9,9 m/s	32,3 m/s 7. okt
nov 2010	6,1 m/s	25,8 m/s 3. nov
des 2010	7,5 m/s	24,7 m/s 31. des
jan 2011	9,8 m/s	27,7 m/s 31. jan
feb 2011	9,4 m/s	29,0 m/s 4. feb
mar 2011	7,3 m/s	33,0 m/s 2. mar

www.yr.no

Appendix 9

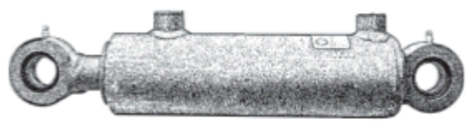
HYDRAULIKK

Hydrauliske sylindere

MED ALTERNATIVE STANG-

OG INNFESTNINGSMÅL

TEKNISK SPESIFIKASJON
Stempelstang.......Stål UNI C 45 SAE 1045 25 ±5 micron crom
Sylinder.......St. 52.3 DIN 2393 ISO H9, rullepolert
Trykk-klasse.......200 bar



På forespørsel skaffer vi også sylindere med mål som er oppgitt nedenfor.

TYPE 507 MED SFÆRISKE LAGER

BEST.NR.	SYL. DIAMETER	STANG-DIAMETER	SLAGLENGDE	MIN L	MAX L	OLJEPORTER	HULL FOR REDSKAP	BREDDE INNFESTING
	mm	mm	mm	cc hull	cc hull	BSP	mm	mm
HC 507-32-20-125	32	20	125	280	405	1/4"	15	12
HC 507-32-20-200	32	20	200	355	555	1/4"	15	12
HC 507-32-20-275	32	20	275	430	705	1/4"	15	12
HC 507-32-20-320	32	20	320	475	795	1/4"	15	12
HC 507-32-20-400	32	20	400	555	955	1/4"	15	12
HC 507-32-20-630	32	20	630	805	1435	1/4"	15	12
HC 507-40-25-200	40	25	200	380	580	1/4"	20	16
HC 507-40-25-250	40	25	250	430	680	1/4"	20	16
HC 507-40-25-300	40	25	300	480	780	1/4"	20	16
HC 507-40-25-350	40	25	350	530	880	1/4"	20	16
HC 507-40-25-400	40	25	400	580	980	1/4"	20	16
HC 507-40-25-500	40	25	500	680	1180	1/4"	20	16
HC 507-40-25-630	40	25	630	810	1440	1/4"	20	16
HC 507-50-30-150	50	30	150	359	509	3/8"	25	20
HC 507-50-30-200	50	30	200	409	609	3/8"	25	20
HC 507-50-30-250	50	30	250	459	709	3/8"	25	20
HC 507-50-30-320	50	30	320	529	849	3/8"	25	20
HC 507-50-30-400	50	30	400	609	1009	3/8"	25	20
HC 507-50-30-500	50	30	500	709	1209	3/8"	25	20
HC 507-50-30-630	50	30	630	839	1469	3/8"	25	20
HC 507-50-30-800	50	30	800	1009	1809	3/8"	25	20
HC 507-50-30-1200	50	30	1200	1409	2609	3/8"	25	20
HC 507-63-40-160	63	40	160	386	546	1/2"	30	22
HC 507-63-40-200	63	40	200	426	626	1/2"	30	22
HC 507-63-40-250	63	40	250	476	726	1/2"	30	22
HC 507-63-40-320	63	40	320	546	866	1/2"	30	22
HC 507-63-40-400	63	40	400	626	1026	1/2"	30	22
HC 507-63-40-460	63	40	460	686	1146	1/2"	30	22
HC 507-63-40-550	63	40	550	776	1326	1/2"	30	22
HC 507-63-40-675	63	40	675	901	1576	1/2"	30	22
HC 507-63-40-800	63	40	800	1026	1826	1/2"	30	22
HC 507-63-40-1075	63	40	1075	1301	2376	1/2"	30	22

Ref.www.okonomi-deler.no

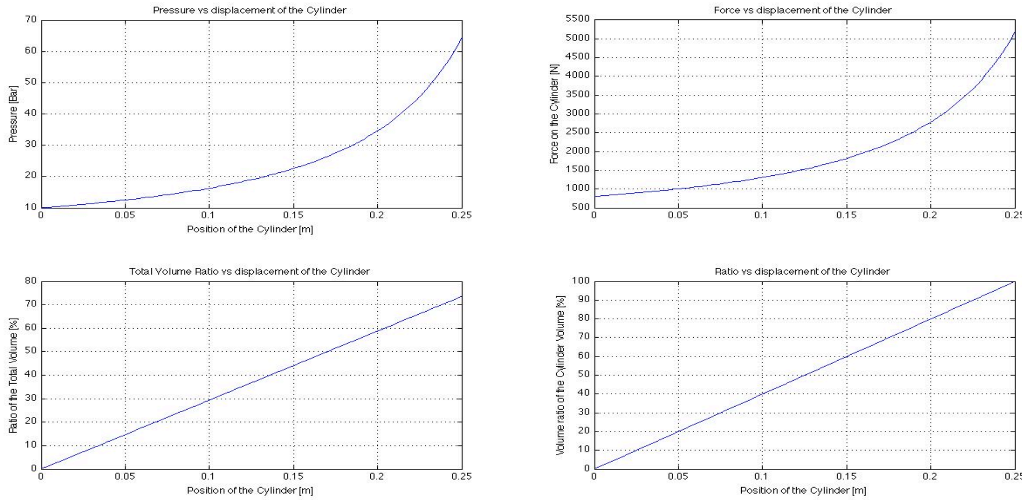
APPENDIX 10

```
clc
clear all
% Two Syliders connected together with a hose.
% Calculation of P_Init*V_tot_Init = P_end*V_end

%We have three volumes.
%1. The gasspring in the Resonator Machine
%2. The hose
%3. The Gas Spring (or pump) from Kaller

%%
%Initial pressure


P_0 = 10; %[bar]
Gas_Const = 1.4; %[1] Adiabatic Gas Constant for Air, Helium and Nitrogen

%%
%Specifications for the Resonator Machine gas spring

Length_Res_Gas      = 0.02;    %[m] From Drawings of Prototype 2B (24mm)
Diameter_Res_Gas    = 0.063;    %[m] From Drawings of Prototype 2B (63mm)
%Length_Res_Worst_Case = 0.004;  %[m] Strokelength the Resonator piston will move
Volume_Res_Gas      = pi*((Diameter_Res_Gas/2)^2)*Length_Res_Gas %[m^3]

%t = 0.01 :0.001 : 0.4;
%Volume_Res_Gas_Worst = pi*((Diameter_Res_Gas/2)^2)*(Length_Res_Gas-
Length_Res_Worst_Case*sin(100*pi*t));

%%
%Spesifications of the hose

Length_Hose     = 0.5;          %[m]
Diameter_Hose   = 0.005;        %[m]

Volume_Hose = pi*((Diameter_Hose/2)^2)*Length_Hose %[m^3]

%%
%Spesifications of the Gas Cylinder  (Or Kaller gas spring)
 %**************************************************
Length_Kal_Gas_mm       = 250;       %[mm] The stroke length of the cylinder
Diameter_Kal_Gas_mm     = 40;        %[mm] Diameter of the effective piston area of the cylinder
Volume_Cylinder_Liter  =
((pi*(((Diameter_Kal_Gas_mm/1000)/2)^2)*(Length_Kal_Gas_mm/1000))*1000)*1.001; %[L] Here is the
calculated Volume of the cylinder piston + 0.1% more volume in the cylinder than the piston
%Volume_Cylinder_Liter  = 0.2413;   %[Liter] Listed in the Kaller table PS! There is gas sourrouding
the piston, therefore "that much"
%**************************************************
%Converting and Calculations of the specifications
Length_Kal_Gas    = Length_Kal_Gas_mm/1000;        %[m]
Diameter_Kal_Gas = Diameter_Kal_Gas_mm/1000;      %[m]
Table_Volume_Kal_Gas    = Volume_Cylinder_Liter/1000 %[m^3] From the table
Volume_Kal_Gas = pi*((Diameter_Kal_Gas/2)^2)*Length_Kal_Gas % [m^3] Calculated

%%
%Total Initial Pressure and Volume of all componets
Volume_Tot_Init =  Table_Volume_Kal_Gas + Volume_Hose +Volume_Res_Gas; %[m^3]
P_Init = P_0; %[bar]

%%
% DeltaX_Kal_Spring. The Length is splitted up in 100...and from here
% used futher on in the calculation of the total pressure in the system.

deltaX_Kal_Gas(1) = 0;

for j =1:99

   deltaX_Kal_Gas(j+1) = (Length_Kal_Gas/99)+deltaX_Kal_Gas(j);

end



%Calculating the system pressure and the force acting from the kaller-spring
for i = 1:100

    P_end(i) = P_Init*((Volume_Tot_Init)^Gas_Const)/((Volume_Hose+Volume_Res_Gas+(Table_Volume_Kal_Gas
- pi*((Diameter_Kal_Gas/2)^2)*deltaX_Kal_Gas(i))))^Gas_Const;%[bar] Calculaton P_end =
P_Init*(Volume_Init)^Gas_Const/(Volume_End)^Gas_Const
```

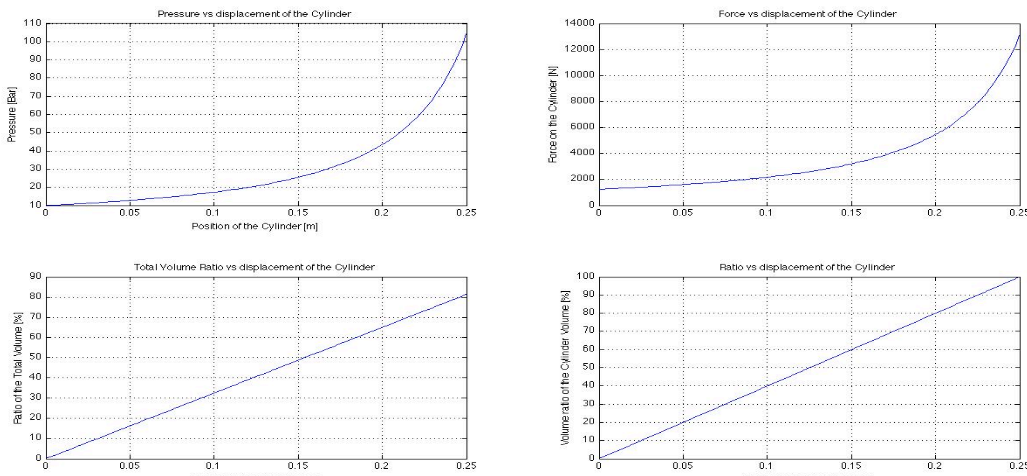
```
    %P_Init*((Volume_Tot_Init)/((Volume_Hose+Volume_Res_Gas+(Table_Volume_Kal_Gas -
pi*((Diameter_Kal_Gas/2)^2)*deltaX_Kal_Gas(i))))); %[bar] Calculaton P_end =
P_Init*Volume_Init/Volume_End

    Force_Kal_Gas(i) = (((P_end(i)*1e5)*pi*(Diameter_Kal_Gas/2)^2)); %[N]

    Volume_Ratio(i) = 100*(1-((Volume_Hose+Volume_Res_Gas+(Table_Volume_Kal_Gas -
pi*((Diameter_Kal_Gas/2)^2)*deltaX_Kal_Gas(i))))/Volume_Tot_Init); % The volume ratio compression

    Volume_Ratio_Kal(i) = (( pi*((Diameter_Kal_Gas/2)^2)*deltaX_Kal_Gas(i))/Table_Volume_Kal_Gas)*100;


end

%%
% Potting the Pressure and Force vs displacement of the Kaller Gas spring

subplot(2,2,1);
figure(1);
plot(deltaX_Kal_Gas , P_end);
grid
ylabel('Pressure [Bar]')
title('Pressure vs displacement of the Cylinder')


subplot(2,2,2);
plot(deltaX_Kal_Gas , Force_Kal_Gas);
grid
ylabel('Force on the Cylinder [N]')
title('Force vs displacement of the Cylinder')

subplot(2,2,3);
plot(deltaX_Kal_Gas , Volume_Ratio);
grid
xlabel('Position of the Cylinder [m]')
ylabel('Ratio of the Total Volume [%]')
title('Total Volume Ratio vs displacement of the Cylinder')

subplot(2,2,4);
plot(deltaX_Kal_Gas , Volume_Ratio_Kal);
grid
xlabel('Position of the Cylinder [m]')
ylabel('Volume ratio of the Cylinder Volume [%]')
title('Ratio vs displacement of the Cylinder')
```

Appendix 11

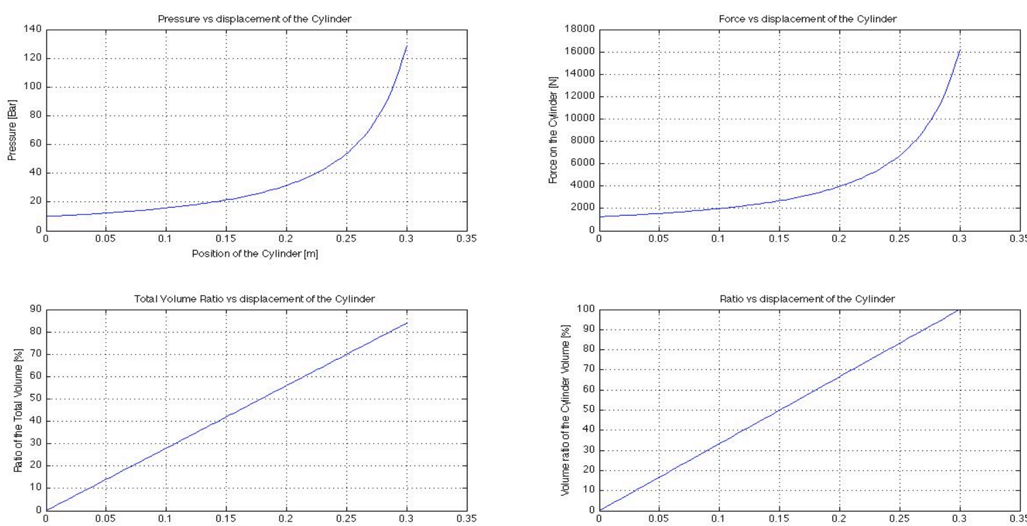
D32-L250



D40-L250



D40-L300



APPENDIX 12

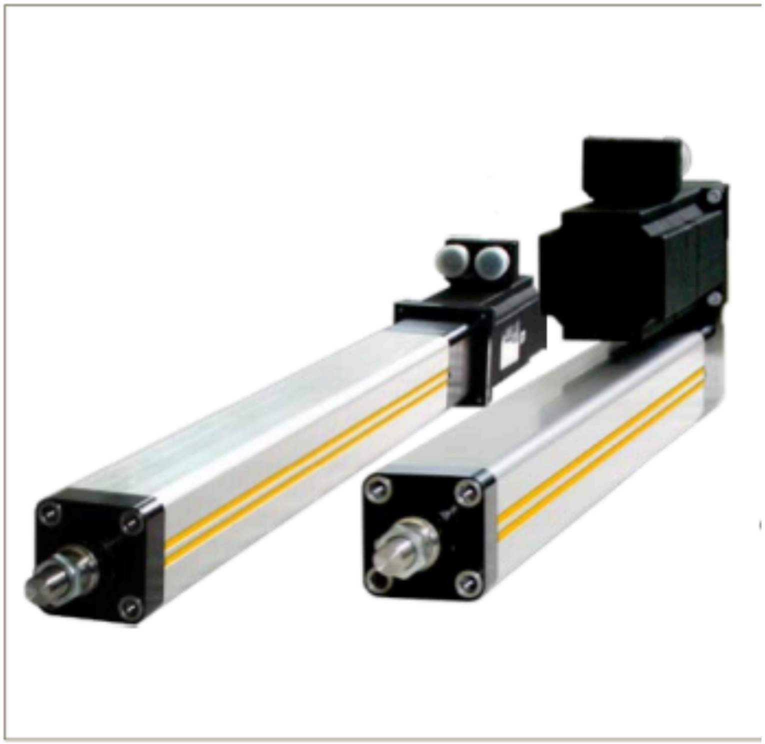
ETH - Electro Cylinder

Parker High Force Electro Thrust Cylinder

New

The well-known series of ET Electro Cylinders, which is widely used in thousands of different applications, was completely redesigned and enhanced. With the new ETH Electro Cylinder, we succeeded in setting new standards in power density and lifetime of electromechanic linear actuators:

- Unrivaled power density - high forces and small frame sizes
- Initiators / initiator cables can be concealed in the profile
- Optimized for safe handling and simple cleaning
- High service life
- Reduced maintenance costs thanks to relubricating hole in the profile
- Easy replacement due to pneumatic ISO flange norm conformity (DIN ISO 15552:2005-12)
- Anti-rotation device integrated
- Reduced noise emission
- All from one source
  We offer the complete drive train: Drive controllers, motors and gearboxes matching the Electro Cylinder



Contact Information

Parker Hannifin GmbH
Electromechanical Automation
Robert-Bosch-Straße 22
D-77656 Offenburg, Germany
+49 (0)781 / 509-0
+49 (0)781 / 509-98176
sales.automation@parker.com

www.parker-eme.com/eth

Product Features

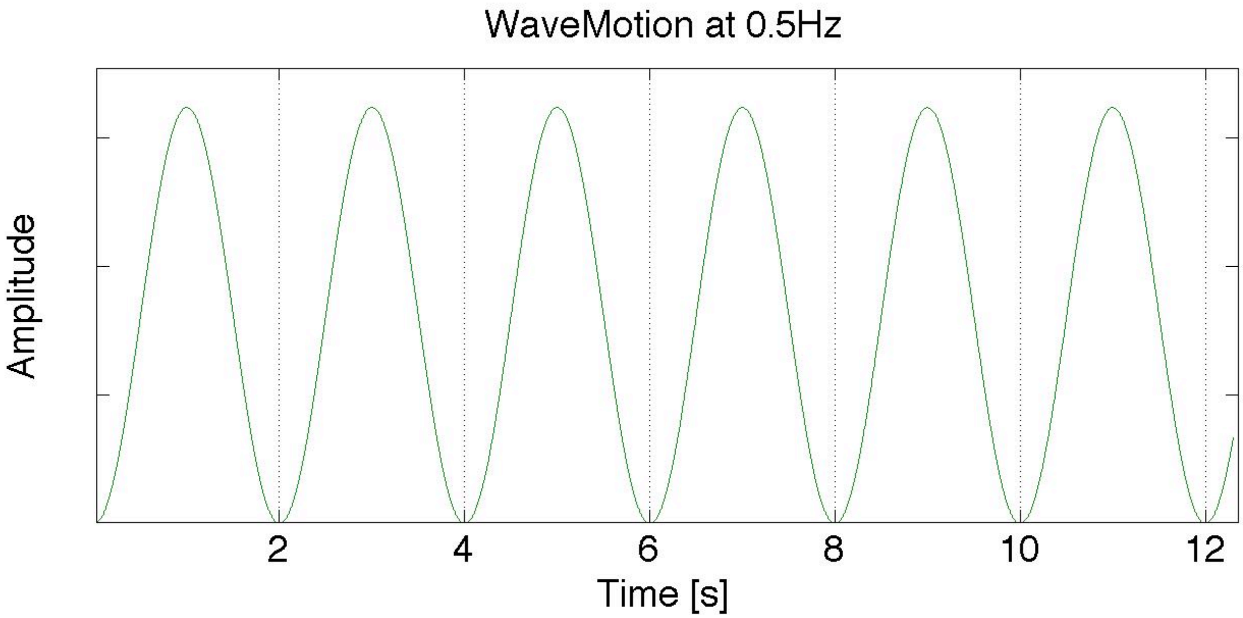
	Unit	ETH032	ETH050	ETH080
Screw lead	mm	5 / 10 / 16	5 / 10 / 20	5 / 10 / 32
Max. stroke	mm	up to 1000	up to 1200	up to 1600
Max. speed	mm/s	up to 1067	up to 1333	up to 1707
Max. acceleration	m/s²	up to 12	up to 15	up to 15
Max. axial traction / thrust force	N	up to 3700	up to 9300	up to 25 100
Equivalent dynamic axial force at a lifetime of 2500 km	N	up to 1700	up to 3230	up to 7500
Repeatability (ISO230-2)	mm	±0.03		

Parker

ENGINEERING YOUR SUCCESS.

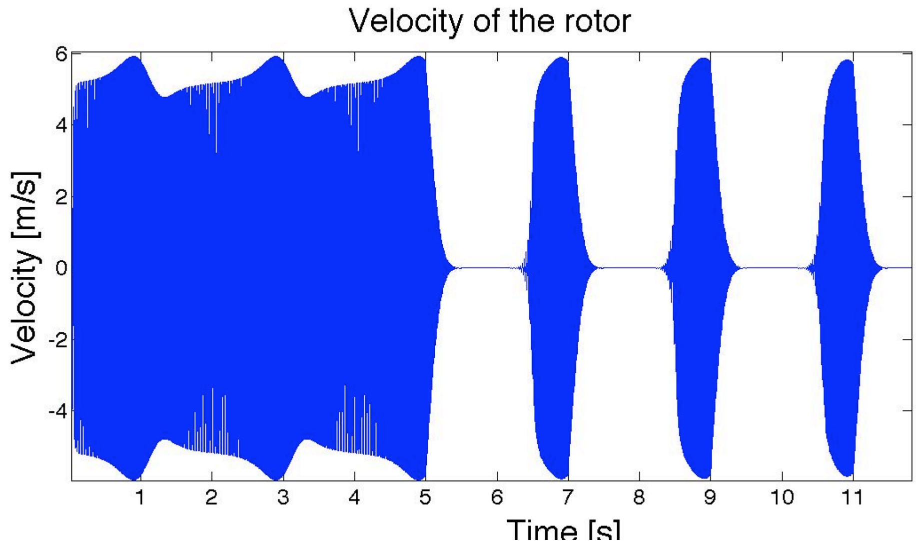
www.parker.com

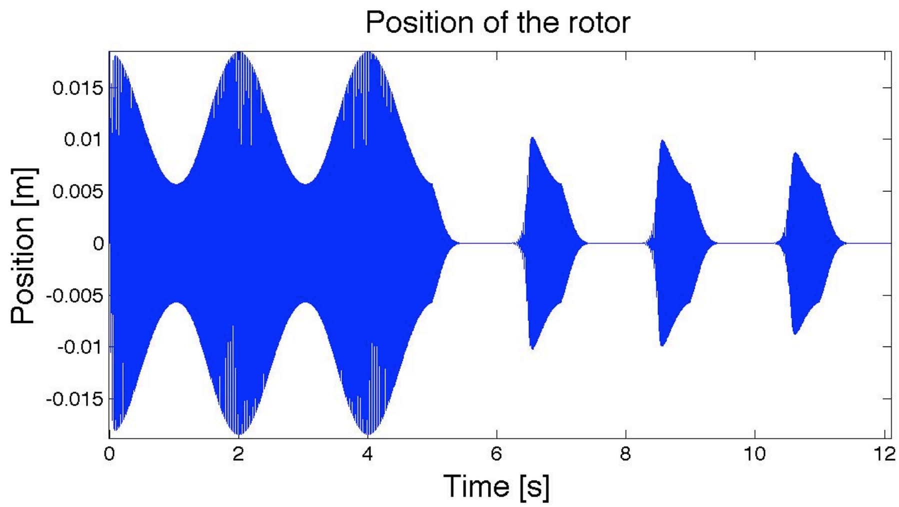
The one with 45000N is not listed here, but the product name is ET125 and can be found on www.parker.com

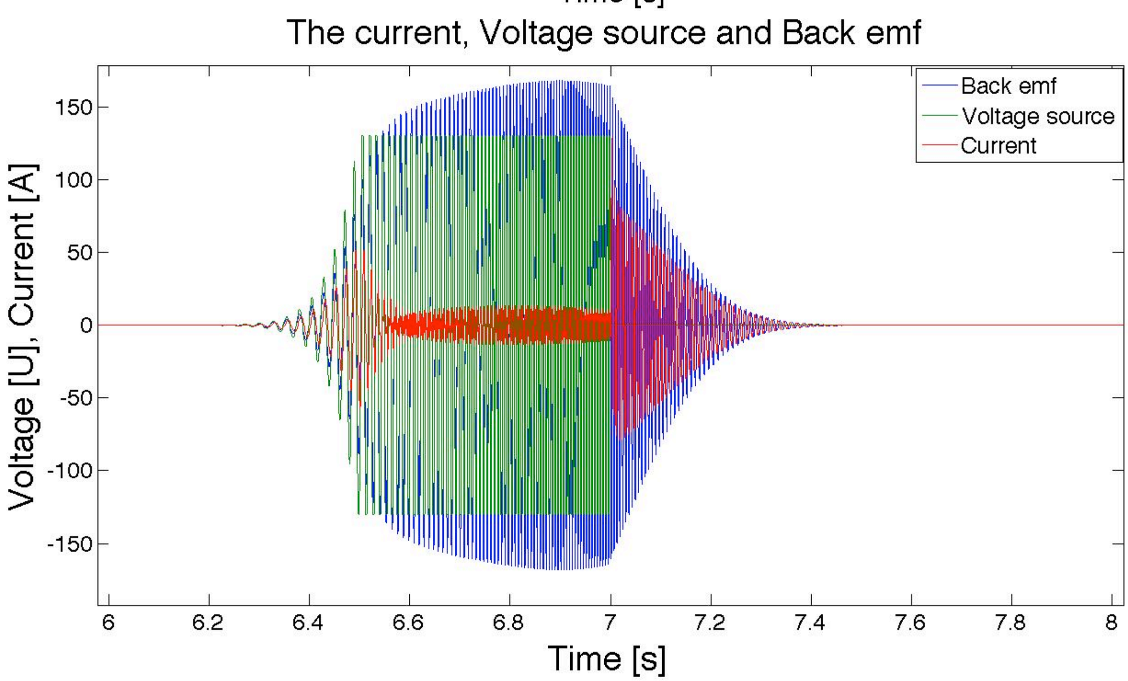

WaveMotion at 0.5Hz
Amplitude
2
4
6
8
10
12
Time [s]

APPENDIX 14

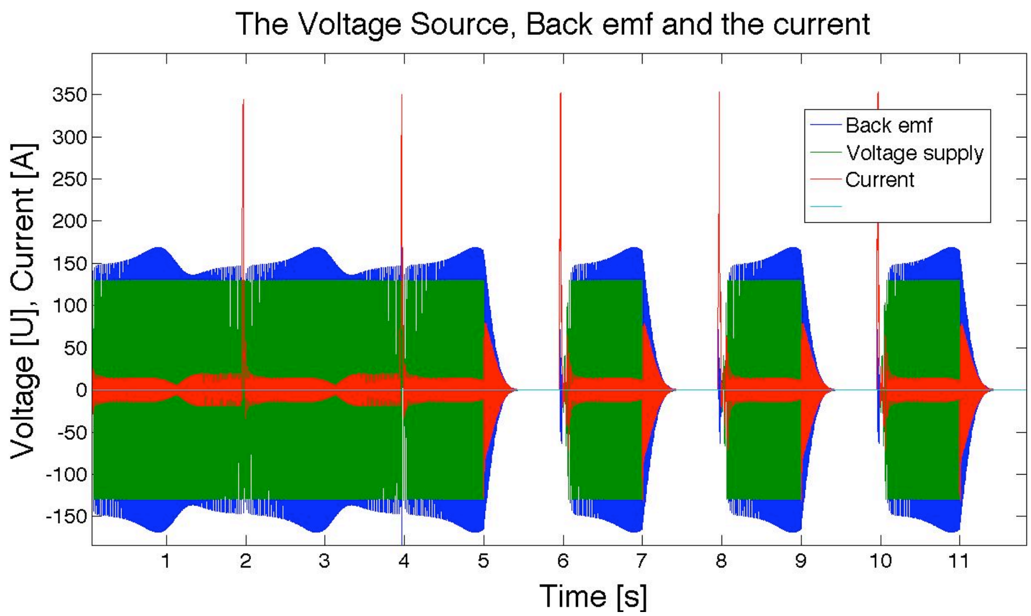
Simulations from solution a.



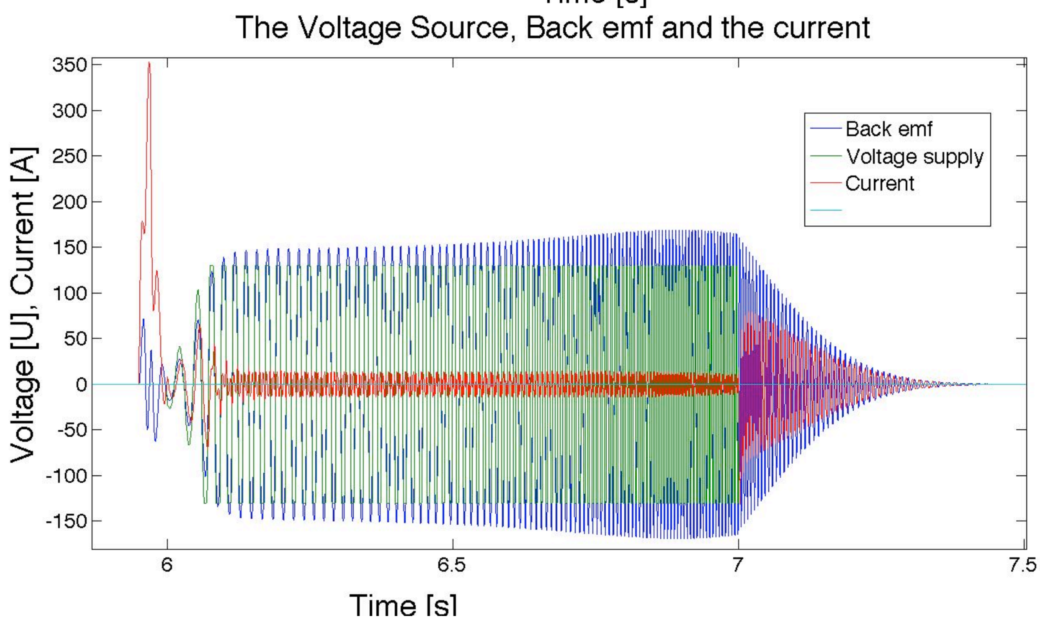




APPENDIX 15



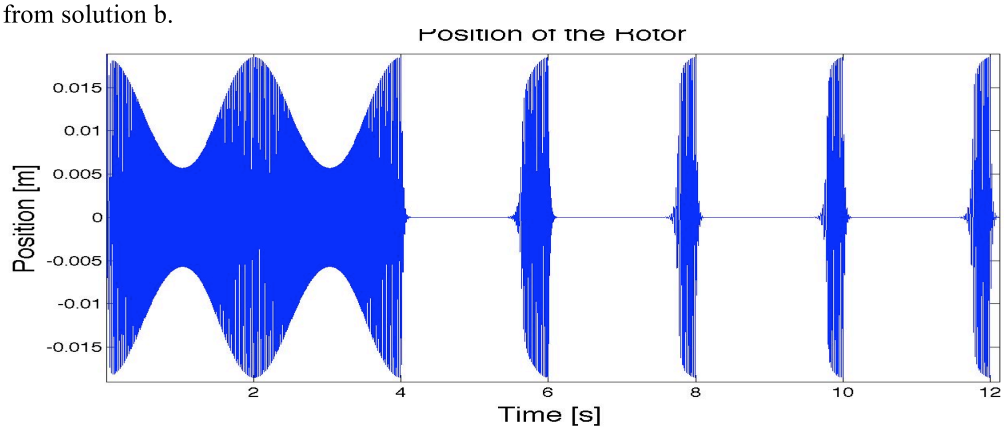
The Voltage Source, Back emf and the current
Voltage [U], Current [A]
Time [s]
Back emf
Voltage supply
Current

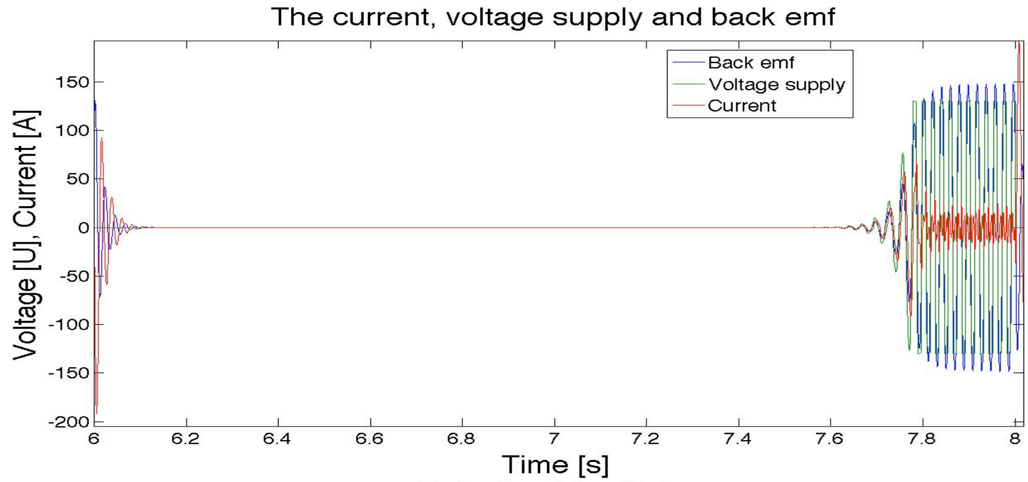


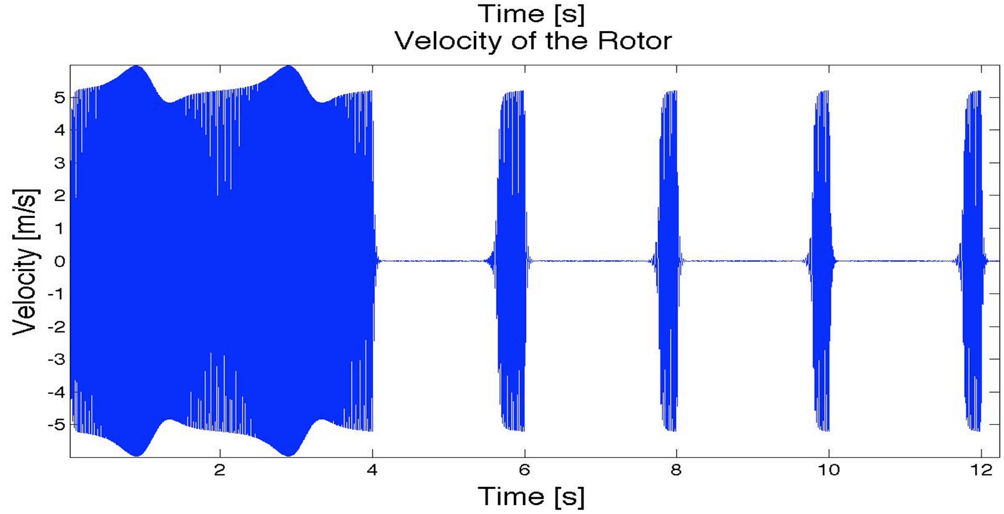
The Voltage Source, Back emf and the current
Voltage [U], Current [A]
Time [s]
Back emf
Voltage supply
Current

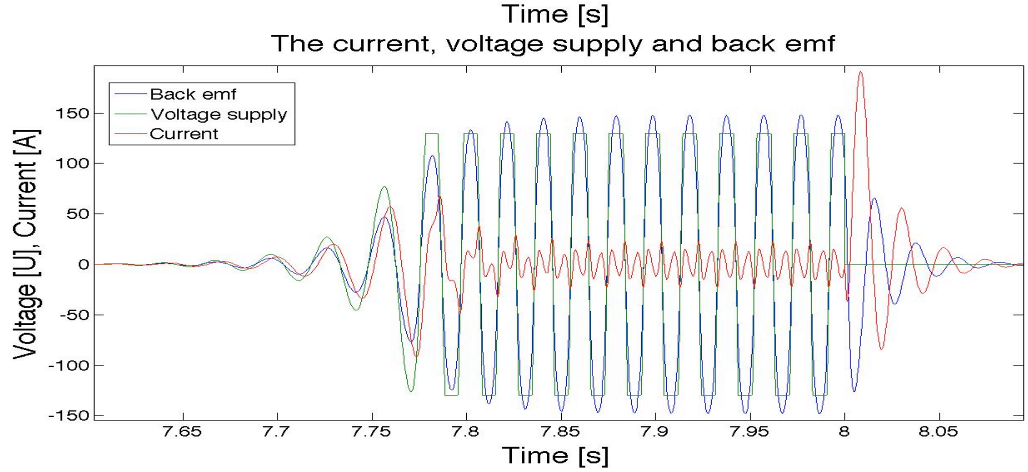
APPENDIX 16

Simulations from solution b.









WALL-MOUNTED VACON NXP

For the lower power range, Vacon NXP drives are available in a compact IP21 or IP54 enclosure. The Vacon NXP is one of the most compact and complete packages on the market, which has all the necessary components integrated within a single enclosure.

The wall-mounted units are equipped with internal EMC filtering, and the power electronics are integrated into an all-metal frame. The smaller frame sizes (FR4-FR6) have an integrated brake chopper as standard, and the 380-500 V units can be equipped with an integrated brake resistor. The larger frames (FR7-FR12) can be equipped with an integrated brake chopper as an option.

Mains voltage 208—240 V, 50/60 Hz, 3~

AC drive type	Loadability					Motor shaft power		Frame size	Dimensions and weight W*H*D (mm)/ kg
	Low (+40°C)		High (+50°C)		Maximum current I_S	230 V supply			
	Rated continuous current I_L (A)	10% overload current (A)	Rated continuous current I_H (A)	50% overload current (A)		10% overload P (kW)	50% overload P (kW)		
NXP 0003 2 A 2 H 1 SSS	3.7	4.1	2.4	3.6	4.8	0.55	0.37	FR4	128*292*190/ 5
NXP 0004 2 A 2 H 1 SSS	4.8	5.3	3.7	5.6	7.4	0.75	0.55	FR4	128*292*190/ 5
NXP 0007 2 A 2 H 1 SSS	6.6	7.3	4.8	7.2	9.6	1.1	0.75	FR4	128*292*190/ 5
NXP 0008 2 A 2 H 1 SSS	7.8	8.6	6.6	9.9	13.2	1.5	1.1	FR4	128*292*190/ 5
NXP 0011 2 A 2 H 1 SSS	11	12.1	7.8	11.7	15.6	2.2	1.5	FR4	128*292*190/ 5
NXP 0012 2 A 2 H 1 SSS	12.5	13.8	11	16.5	22	3	2.2	FR4	128*292*190/ 5
NXP 0017 2 A 2 H 1 SSS	17.5	19.3	12.5	18.8	25	4	3	FR5	144*391*214/ 8.1
NXP 0025 2 A 2 H 1 SSS	25	27.5	17.5	26.3	35	5.5	4	FR5	144*391*214/ 8.1
NXP 0031 2 A 2 H 1 SSS	31	34.1	25	37.5	50	7.5	5.5	FR5	144*391*214/ 8.1
NXP 0048 2 A 2 H 1 SSS	48	52.8	31	46.5	62	11	7.5	FR6	195*519*237/ 18.5
NXP 0061 2 A 2 H 1 SSS	61	67.1	48	72	96	15	11	FR6	195*519*237/ 18.5
NXP 0075 2 A 2 H 0 SSS	75	83	61	92	122	22	15	FR7	237*591*257/ 35
NXP 0088 2 A 2 H 0 SSS	88	97	75	113	150	22	22	FR7	237*591*257/ 35
NXP 0114 2 A 2 H 0 SSS	114	125	88	132	176	30	22	FR7	237*591*257/ 35
NXP 0140 2 A 2 H 0 SSS	140	154	105	158	210	37	30	FR8	291*758*344 / 58
NXP 0170 2 A 2 H 0 SSS	170	187	140	210	280	45	37	FR8	291*758*344 / 58
NXP 0205 2 A 2 H 0 SSS	205	226	170	255	336	55	45	FR8	291*758*344 / 58
NXP 0261 2 A 2 H 0 SSF	261	287	205	308	349	75	55	FR9	480*1150*362/ 146
NXP 0300 2 A 2 H 0 SSF	300	330	245	368	444	90	75	FR9	480*1150*362/ 146

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Notes on hydraulic design of wave energy harvesting system

Introduction

The basic concept to test wave energy harvesting in the lab is

- To have a resonator as the core harvesting device
- The resonator is equipped with two gas springs
- The gas springs are driven by a hydraulic system

The following is some notes on the hydraulic system design.

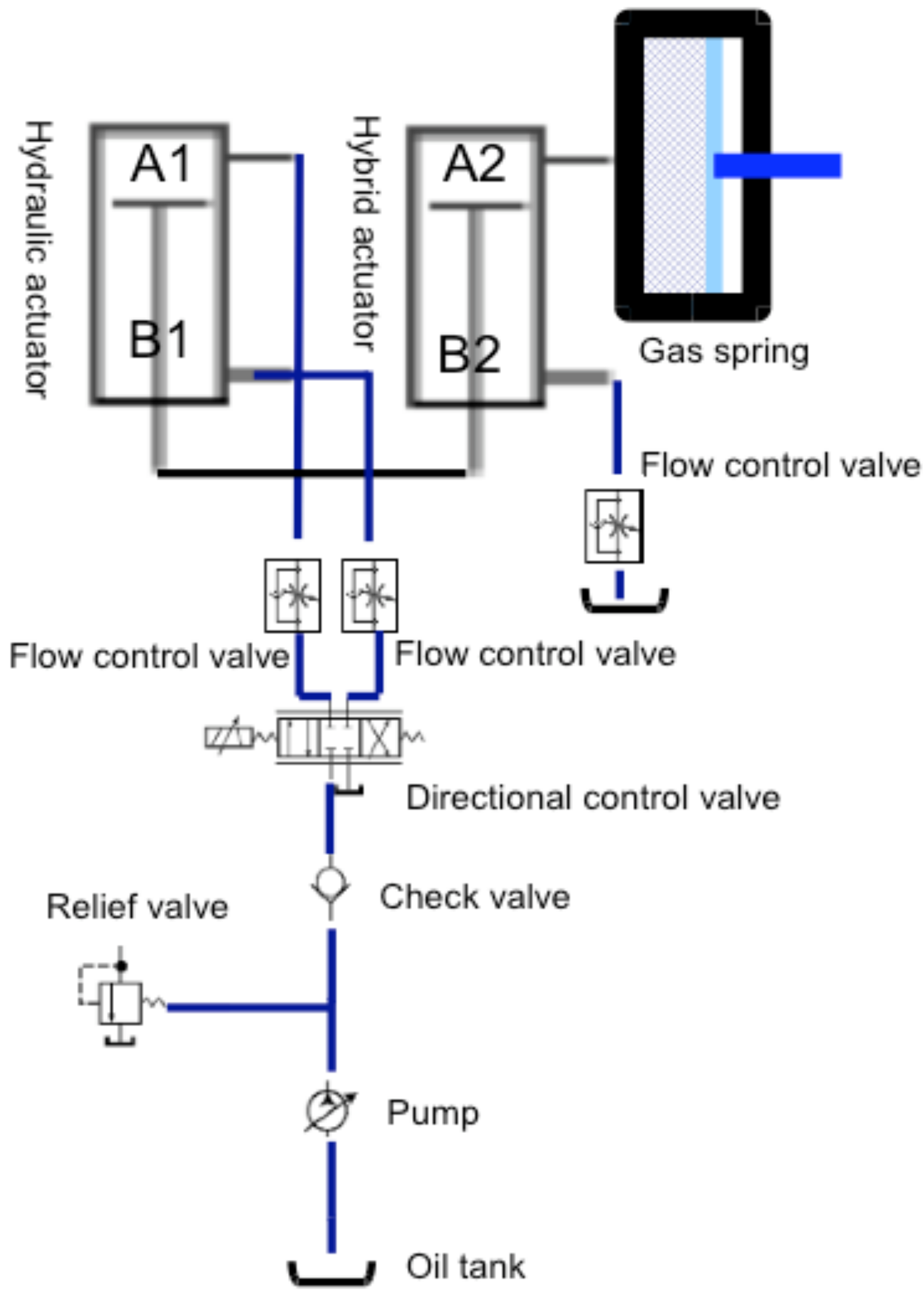


Figure 1: Configuration I of hydraulic system

Two configurations

Two configurations were planned, as shown in Figure 1: Configuration I of hydraulic system and Figure 2: Configuration II of hydraulic system. The differences between two configurations include

- Configuration I: two actuators included on each side to drive one gas spring
  - one hydraulic actuator
  - one hybrid actuator
- Configuration II: only one hybrid actuator is used on each side to drive a gas spring

The same operating requirement applies to both configurations: to keep the force balanced on the piston of hybrid actuator in between gas pressure and hydraulic pressure.

Configuration I

Description

As shown in Figure 1, Configuration I includes two actuators on each side of the resonator to drive one gas spring. One is a hydraulic actuator, both chambers (A1 & B1) of which are filled with oil. The other one is a hybrid actuator, on chamber (A2) of which is filled with gas and the other chamber (B2) is filled with oil.

Pressurized oil from the hydraulic pump is introduced into chamber A1 / B1 of the hydraulic cylinder to produce the force needed to keep the piston of the hybrid actuator in a stable position while resonating.

Pros

- Should be easy to control
- Possible for the hydraulic system to work at a lower pressure (refer to the calculation below)

Cons

- Synchronized movement of two pistons, may cause unbalance of force

Components list

No.	Name	Qty	Remarks
1	Oil tank	1	
2	Pump + motor	1	
3	Relief valve	1	
4	Check valve	1	Non-return valve
5	Directional valve	1	Proportional valve / Servo-control valve
6	Flow control valve	3	Used to adjust the speed of piston
7	Hydraulic actuator	1	
8	Hybrid actuator	1	The seal has to be able to seal high pressure gas

Basic calculations

For a hydraulic system, the basic calculation includes

- Flow rate for each component, including piping
- Pressure that each components may withstand
- Velocity of the piston

The connection between two actuator is that the force needed by the hybrid actuator to hold the gas spring in position has to be supplied by the hydraulic actuator.

Hydraulic actuator

Description	Sym.	Value	Unit	Notes
Diameter of piston	D_h	50.00	mm	
Area of piston	A_{h1}	1963.50	mm^2	
Pressure in Chamber A1	P_{h1}	12.5	MPa	
Hydraulic force on piston	F_h	24543.69	N	
Stroke length of piston	L_h	250.00	mm	
Period	T	2	s	Based on typical wave period
Piston speed	v_p	250.00	mm/s	
Flow rate	Q_1	29.45	l/min	
Diameter of piston rod	d_h	30.00	mm	
Area of piston rod	A_{hr}	706.86	mm^2	
Differential area of piston and rod	A_{h2}	1256.64	mm^2	
Pressure in Chamber B1	P_{h2}	19.53	MPa	

Hybrid actuator

Description	Sym.	Value	Unit	Notes
Diameter of piston	D_g	32.00	mm	
Area of piston	A_{g1}	804.25	mm^2	
Pressure in Chamber A2	P_{g1}	30	MPa	Maximum gas spring pressure
Hydraulic force on piston	F_g	24127.43	N	
Stroke length of piston	L_g	250.00	mm	
Diameter of piston rod	d_h	20.00	mm	
Area of piston rod	A_{gr}	314.16	mm^2	
Differential area of piston and rod	A_{g2}	490.09	mm^2	
Pressure in Chamber B2	P_{2g}	49.23	MPa	

It is shown in the table that the main hydraulic system is operated with a pressure lower than 20MPa and flow rate of 30 l/min. It is a normal high pressure hydraulic system. The components are easy to be obtained.

Configuration II

Description

As shown in Figure 2, Configuration II includes only one hybrid actuator on each side of the resonator to drive a gas spring. The hybrid actuator has one chamber (A2) filled with gas and the other (B2) one filled with oil.
Pressurized oil from the hydraulic pump is introduced into chamber B2 of the hybrid actuator and supplies the force needed to keep the piston in a stable position while resonating.

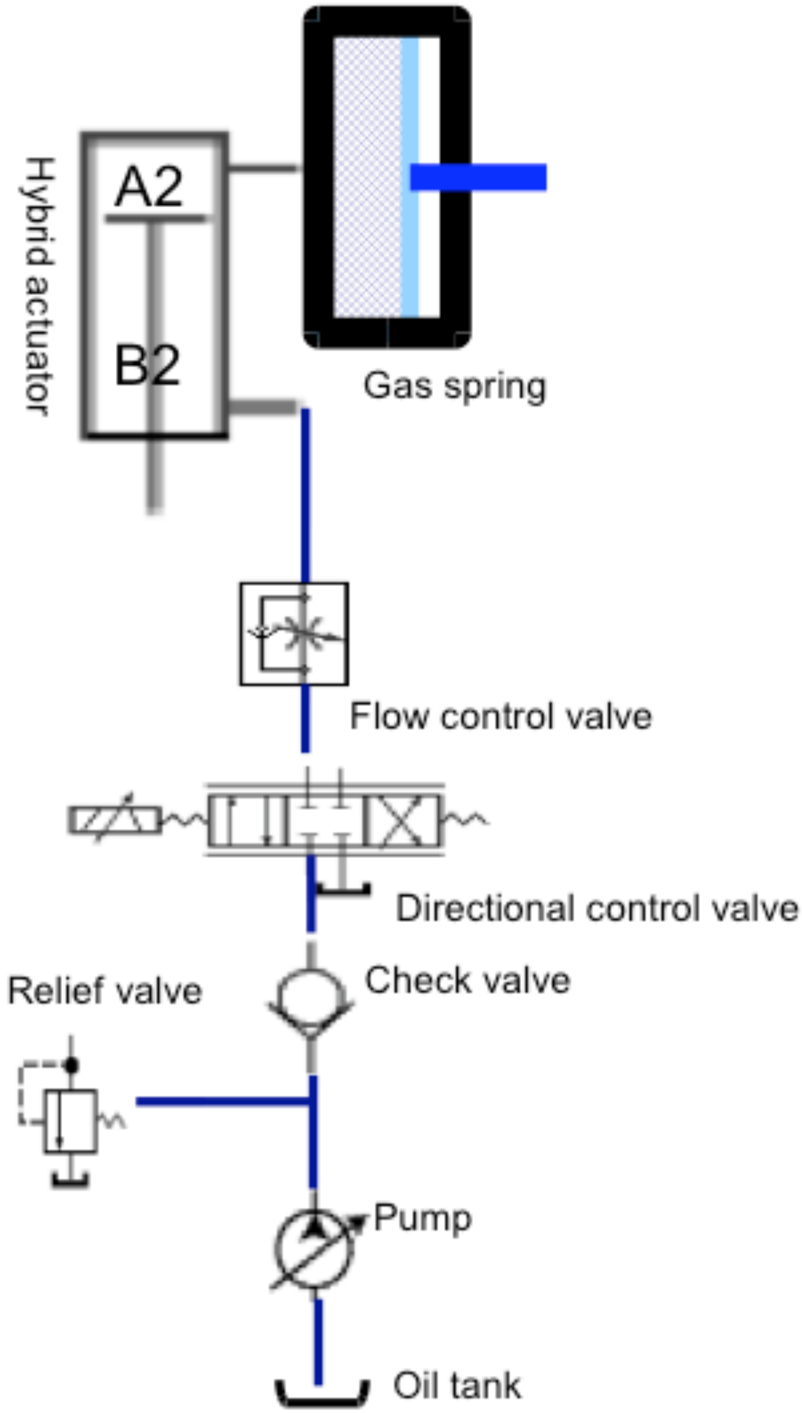


Figure 2: Configuration II of hydraulic system

Pros

- The system is simpler.
- Avoid unbalance force / misalignment of the piston rod

Cons

- Relative higher operating pressure of hydraulic system
- Higher requirement to the *Directional Control Valve* to follow the high frequency pressure

variance of the gas spring

Components list

The components needed for this configuration is similar to that of Configuration I. The differences are

- Only one *Flow Control Valve* is needed, instead of three as in Configuration I
- The hydraulic actuator is not needed.

Basic calculations

There is only one actuator in the system, so the calculation is simplier.

Hybrid actuator

Description	Sym.	Value	Unit	Notes
Diameter of piston	D_g	32.00	mm	
Area of piston	A_{g1}	804.25	mm^2	
Pressure in Chamber A2	P_{g1}	30	MPa	Maximum gas spring pressure
Hydraulic force on piston	F_g	24127.43	N	
Stroke length of piston	L_g	250.00	mm	
Diameter of piston rod	d_h	20.00	mm	
Area of piston rod	A_{gr}	314.16	mm^2	
Differential area of piston and rod	A_{g2}	490.09	mm^2	
Pressure in Chamber B2	P_{2g}	49.23	MPa	

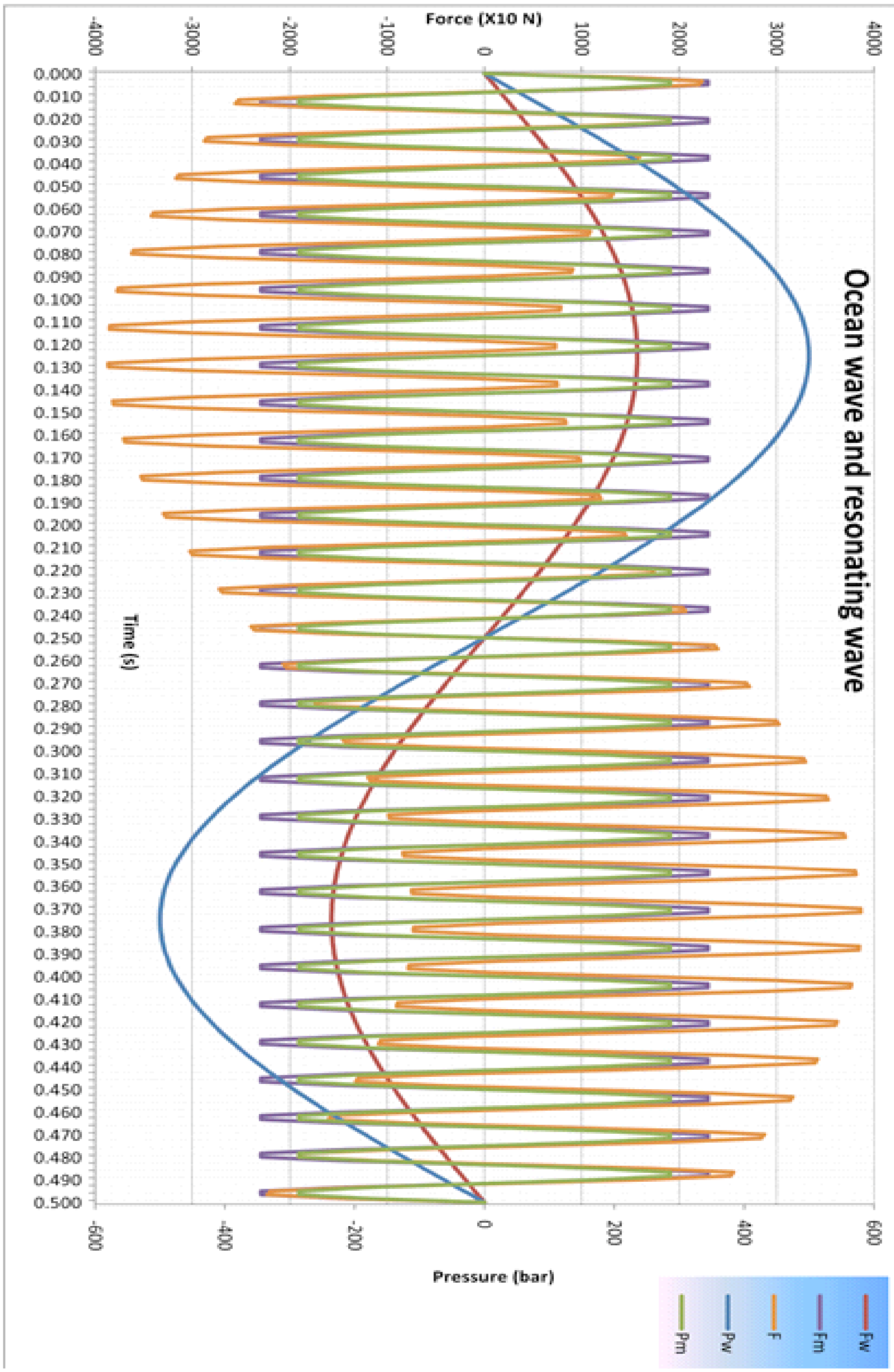
It is shown from the calculation that the hydraulic system has to operate with maximum working pressure up to 50 MPa.

Discussions

- An accumulator maybe needed to compensate the pressure fluctuation in the hybrid actuator.
- How to synchronize the wave movement and resonating movement if the piston of hybrid actuator has to keep still?
- Composition of waves
  - Ocean wave simulation
    - $p_w(t) = P_w \cdot \sin(\omega_w t + \psi_w)$
  - Resonating wave
    - $p_r(t) = P_r \cdot \sin(\omega_r t + \psi_r)$
  - General wave
    - $p(t) = p_r(t) + p_w(t)$
- Pressure control or flow (speed / movement) control?
  - It is a pressure control system if the piston of hybrid actuator has to keep still.
    - Big range pressure control is another challenge
  - It is a flow control system if the piston of hybrid actuator has to move as ocean wave.
    - The maximum flow rate is moderate. So the range of fluctuation is quite limited and should be easy to realize.
- Frequency of wave is different from the resonance frequency of resonator
  - how the pistons move?
    - When the resonator is resonating, the pressure of gas spring is changing in the frequency of resonating, but not the pressure in the hydraulic actuator?
- Load on the piston of hydraulic actuator?
  - Why to have a load on piston?
    - To compress or decompress the gas?

Following is a simulation of the pressure and force of Configuration II. Assume that the frequency of ocean wave is 2 Hz, and resonating frequency is 60Hz.

- F_w: force of ocean wave (simulated by hydraulic system)
- F_m: gas spring force of resonator
- F: general force from hydraulic system
- P_w: hydraulic pressure needed to simulate the ocean wave, 0 – 500 bar
- P_m: gas spring pressure of resonator, 0 – 300 bar



Ref: Ph.D Pingju Li, Resonator AS