THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING
IN MACHINE AND VEHICLE SYSTEMS

Attenuation of hand-held machine vibrations

Application of non-linear tuned vibration absorbers

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
Left: Operator working with original chisel machine in block of granite.
Top row: Evolution of the ATVA-damper from original machine to analytic modeling and finally to concept prototype.
Center: Finger blanching attack, Raynauds phenomena
Bottom row: Measured high frequent transient vibration from an impact wrench before and after ISO 5349 filtering.

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Abstract

Vibration injury in the hand-arm system from vibrating hand-held machines is one of the most common occupational health injuries and causes severe and often chronic nerve and vascular injuries to the operator. The objective with this study is to contribute to reduce hand-arm vibration injuries, first by developing methods of redesigning machines that will lower their vibration level and second to study how high frequency vibrations (HFV) above 1250 Hz effect the finger tissue which are not included in the current standard for risk evaluation.

Vibrating hand-held machines can be divided into two main groups depending of the source of the vibration. The first and largest group is machines where a reciprocating piston creates a vibration, e.g. breakers and hammer drills. The second group is rotating machines, e.g. grinders and polishing machines, where the vibration is generated by the unbalance force from the grinding disc or the rotating force from the friction between the abrasive pad and the work piece.

This study has mainly been focused on reciprocating machine vibrations by development of a method for attenuation of these vibrations based on tuned vibration absorbers (TVA). The TVA comprises of a spring and an additional mass connected to the main vibrating body that creates a counterforce to attenuate system vibration. The limitation of conventional TVA is that they are only effective in a narrow frequency span and thereby have a limited practical use.

However by introducing appropriate gap and spring preload into classical TVA makes the absorber dynamics nonlinear. It was shown that this kind of nonlinearity can lead to frequency auto tuning vibration attenuation. The efficiency of the developed technology called Auto Tuning Vibration Absorber (ATVA) was proven by simulations and the results were validated both in a dedicated test rig as well as in prototype machines. The results from a prototype machine showed that the vibration could be reduced by approx. 80 % and at the same time reduce the weight with 50 % compared to the original machine without affecting the efficiency.

The study of the mechanical effects from HFV on finger tissue was made by, first: development of measurement technique based on Micro Electrical Mechanical Systems (MEMS) accelerometers to be able to accurately measure vibrations up to 50 kHz, second: create a Finite Element (FE) model for simulating the pressure wave propagation into the finger tissue, and third: development of methods to reduce the HFV content in machines. Measurements show that there are high frequency high amplitude accelerations from impact tools and that these accelerations create a pressure wave that propagates into the tissue. It was also shown that there is a large potential to effectively reduce the HFV content in machines.

On the study of effects on finger tissue from HFV clearly indicated that there is potentially an increased risk that HFV will cause vibration injuries on humans and on biological tissues and that the current standard ISO 5349 underestimates the associated risks.

Key words: Nonlinear tuned vibration absorber, Hand-arm vibration, Vibration dynamics, Impact machine, High frequency vibration, Transient vibration, HAVS, White finger, Auto Tuned Vibration Absorber, ATVA, ISO 5349

Minskning av vibrationer i handhållna maskiner
Tillämpning av ickelinjära avstämda vibrationsdämpare

Avhandling för teknologie licentiatexamen i Maskin och Fordonssystem
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Sammanfattning

Vibrationsskador i händer och armar som orsakas av vibrationer från handhållna maskiner är en av de vanligaste arbetssjukdomarna och yttrar sig som allvarliga och ofta kroniska nerv- och kärlskador hos användaren med stort personligt lidande som följd. Syftet med detta arbete är att bidra till att minska vibrationsskadorna i samhället, dels genom att utveckla metoder som minskar maskiners vibrationer genom omkonstruktion och dels genom att studera hur högfrekventa vibrationer (HFV), över 1250 Hz, påverkar fingrets vävnad och som inte täcks in av dagens standard vid bedömning av risker.

Vibrerande handhållna maskiner kan delas in i två huvudgrupper beroende på typ av källa till vibrationerna. Den första och största gruppen är maskiner där en fram- och återgående kolv skapar vibrationen, t.ex. i en bilningsmaskin eller i en borrhammare. Den andra gruppen är de roterande maskinerna, t.ex. slipmaskiner och polermaskiner, där vibrationen genereras av obalansen i slipskivan eller den roterande kraften från slipmaterialets friktion mot underlaget.

Denna studie har huvudsakligen varit inriktad på de fram- och återgående maskinvibrationerna genom att utveckla en metod för reduktion av dessa vibrationer som baseras på avstämda vibrationsdämpare, Tuned Vibration Absorbers (TVA). TVA:n består av extra massa som är förbunden via en fjäder till huvudkroppen vars vibrationer skall minskas. Genom att stämma av extramassans resonansfrekvens kan denna fås att skapa en motkraft som effektivt minskar huvudkroppens vibrationer. Begränsningen i konventionella TVA är att de endast är effektiva i ett smalt frekvensområde och därigenom har en mycket begränsad praktisk användning i handhållna maskiner.

I detta arbete har visats att det effektiva frekvensområdet kan ökas väsentligt genom att införa en olinjär fjädringsväg för extramassan i TVA:n genom att införa ett glapp och en förspänning av fjädern vilket gör massornas dynamik olinjär. Det har visat sig att denna typ av olinjäritet kan väljas så att extramassans resonansfrekvens fås att automatiskt följa den pålagda kraftens frekvens på huvudmassan. Effektiviteten hos den utvecklade tekniken, som har kallats, Auto Tuning Vibration Absorber (ATVA), har visats genom simuleringar och resultaten har validerats både i en dedikerad testrigg såväl som i prototypmaskiner.

Studien av de mekaniska effekterna från HFV på fingervävnad bestod av tre delar. I del ett utvecklades mätteknik som utnyttjar Micro Electrical Mechanical Systems (MEMS) accelerometrar för att med hög precision kunna mäta vibrationer upp till 50 kHz samt uppmätning av HFV på olika maskiner. I del två skapades en Finita Element (FE) modell för simulering av tryckvågens spridning in i fingervävnaden. Del tre bestod av utveckling av metoder för reducering av HFV i maskiner. Mätningarna visade att det finns högfrekventa vibrationer med hög amplitud från slagverktyg och att dessa accelerationer skapar en tryckvåg som sprids in i vävnaden. Det visas också att det finns en stor potential att effektivt minska HFV i maskiner.
En litteraturstudie över biologiska och medicinska effekter från HFV indikerar att det sannolikt finns en risk att HFV orsakar vibrationsskador på människor och på biologisk vävnad och att den nuvarande standarden, ISO 5349, därmed troligen gravt underskattar riskerna.

To my dear family Ann-Marie, Ludvig & Alicia
and
J. P. Den Hartog whose book "Mechanical Vibrations" has inspired me through my entire career.

Preface

The research work was done between March 2013 and November 2017 at Division of Dynamics, Department of Mechanics and Maritime Systems, Chalmers University of Technology and at Swerea IVF. The project was set up as an Industrial PhD project.

The research work was carried out within several projects with objectives on reducing hand-arm vibration injuries. The projects have targeted both reducing the vibrations itself from hand-held machines but also to give an understanding of the mechanism of injury in the finger tissue from high frequent vibrations.
The main supervisor was Professor Viktor Berbyuk, Chalmers University of Technology and assistant supervisor was Docent Elis Carlström, Swerea IVF. I certainly would like to thank and acknowledge them for their great support given to me and for the patience and understanding when other tasks in work have come in conflict with this project.

I also would like to thank all personnel at Swerea IVF, Chalmers and participating companies in the project who have helped me. The master thesis students that have been involved have made valuable contribution and I have truly appreciated their curiosity and enthusiasm for the task. The patent text in Paper A has been written with technical support from Malina Haag at AWAPATENT.

Finally also a special thank to Bert Andersson for his excellent craftsmanship in manufacturing the prototypes.

The project was financial supported mainly by Vinnova, AFA Försäkring and companies within the Swedish stone industry to whom I am most thankful.

Göteborg, November 2017

HANS LINDELL

Thesis

This thesis includes an extended summary and the following appended papers:

Paper A

H. Lindell: Impact machine, Patent, WO 2015/193284 A1
Also published as: CA2951975A1, CN106715949A, EP3154746A1, US20170095920
https://encrypted.google.com/patents/WO2015193284A1

Paper B

Lindell, H., Berbyuk, V., Josefsson, M., and S. L. Grétarsson, (2015), "Nonlinear dynamic absorber to reduce vibration in hand-held impact machines", *In Proc. of the International Conference on Engineering Vibration,* Ljubljana, 7 - 10 September ; [editors Miha Boltežar, Janko Slavič, Marian Wiercigroch]. - EBook. - Ljubljana: Faculty for Mechanical Engineering, 2015 p. 1530-1539.

Paper C

Lindell H., Grétarsson S., Machens M. High Frequency Shock Vibrations and implications of ISO 5349 - Measurement of vibration, Simulating Pressure Propagation, Risk Assessment and Preventive Measures. *IFA Report 2017, Hand-arm vibration: Exposure to isolated and repeated shock vibrations – Review of the International Expert Workshop 2015 in Beijing. (Revised version)*

In addition to papers A, B and C the following reports have also been part of the research in this PhD project which is not included in the thesis:

1. Lindell H., Grétarsson S., Machens M., High Frequency Vibrations From Impact Tools – Measurement of Vibration and Simulating Pressure Propagation into Finger Tissue, *6:th American Conference on Human Vibration, Chicago, USA, 2016*
2. Lindell H., Zero Vibration Injuries – A Swedish Holistic Approach Fighting Vibration Injuries, *13:th Int. Conf. on Hand-Arm Vibrations, Beijing, China, 2015*
3. Lindell H., Berbyuk V., Grétarsson S., Josefsson M., Hand-Held Impact Machines with Nonlinear Tuned Vibration Absorber, *13:th Int. Conf. on Hand-Arm Vibrations, Beijing, China, 2015*
4. Lindell H., Grétarsson S., Machens M., Transient Vibrations from Impact Machines - Its relation to ISO 5349 and wave propagation in Human Tissue, ISO 5349 Standardization Workshop, *13:th Int. Conf. on Hand-Arm Vibrations, Beijing, China, 2015*
5. Lindell H., Berbyuk V., Gretarsson S., Josefsson M., Vibrationsreducering med avstämda vibrationsdämpare i icke resonant område, *Svenska Mekanikdagarna, Linköping, 2015*
6. H. Lindell: Impact Machine, Patent, EP13811461.6
   https://www.google.com/patents/EP2931481B1

The appended papers were prepared in collaboration with co-authors. The LS Dyna FE simulations in Paper C were made by Michael Machens and the MatLab code made by the author of this thesis has been extended by Josefsson M. and Grétarsson S. L. in their M.Sc. thesis.

Contents

PART I

Extended summary

1 Introduction

The objective with this work is to contribute to reduce hand-arm vibration injuries on humans from vibrating hand-held tools. The work has been divided into two main parts. The first part is the development of a generic method to reduce low frequency reciprocating machine vibrations covered by the standard ISO 5349 [1] (< 1250 Hz) in hand-held machines such as rock drills and breakers which is the dominant group of machines causing injuries. The second part is a study on how high frequency vibrations (HFV, >1250 Hz) effect the finger tissue which is not currently regulated in the standard ISO 5349.

The reason to also include HFV in the scope to reduce vibration injuries is that ISO 5349 is suspected to underestimate the risk from machines with high frequency and high amplitude vibrations. Examples of such machines are impact wrenches, dental drills, chisels etc.

2 Background and motivation

2.1 Hand-arm vibration injuries

Vibrations from hand-held tools cause injuries to the upper limb and is called the Hand Arm Vibration Syndrome (HAVS). It is one of the most common work related injury and has been known since the beginning of the 20:th century. The injury is often chronic and there is no effective cure. The symptoms of HAVS are painful and disabling disorders of the blood vessels, nerves and joints. Injuries to the nerves can include inability to do precision work such as button a shirt, feel the temperature of things and a tingling sensation in the fingers.
Injuries to the blood vessels cause attacks of finger blanching which is also known as “White Fingers” or in medical terms, Raynaud's phenomenon as seen in Figure 1. The blanched finger becomes numb and there is often pain associated when the blood returns. The knowledge about the mechanisms behind HAVS is not well known except that it is clearly related to vibration exposure.

Figure 1 Finger blanching from HAVS

In order to prevent workers from injuries there is a standard, ISO 5349, that describes how to measure machine vibration which was originally formulated in the mid 70:s. In an appendix, there is also a recommendation on maximum daily vibration exposure dose. The standard is limited to frequencies between 6.3 Hz and 1250 Hz and has a weighting filter that suppresses higher frequencies with

6 dB/octave which in practice results in an integration of the acceleration to velocity in the frequencies between 10 to 1000 Hz, Figure 2.

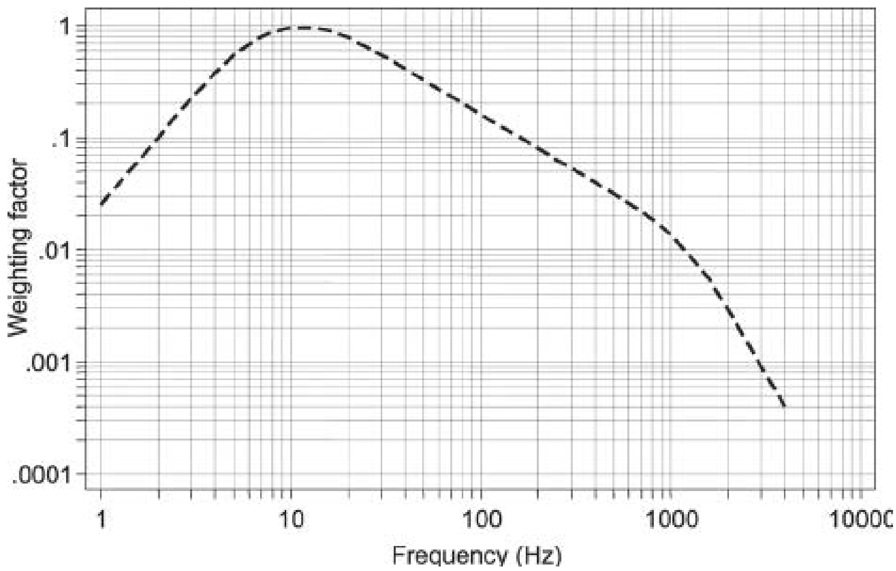

Figure 2 Frequency weighting curve for hand-transmitted vibration in ISO 5349-1:2001

2.2 Low frequency vibration

The sources of vibrations that expose the work force can be divided into different types of machines. In Figure 3, that is based on a report from the European Agency for Safety and Health at Work [2], page 27, there is an attempt to give an overview of the main types of machines and their vibration level measured according to ISO 5349 [1]. The vertical dotted line at 2.5 m/s^2 is the level where an operator can work 8 h/day with the tool without the employer has to offer medical examination to the employee as regulated in the European Machine Directive [3]. The machine types can based on their fundamental operating mechanism be divided into three main categories depending on the source of vibration. These machine types are:

- Reciprocating: This is the largest group where a piston is driven in a linear motion hitting a working tool such as a drill or chisel. The vibration in these tools originates from the reaction force from accelerating the piston and the frequency is usually between 15 and 60 Hz. Typical machines are breakers, rock drills and rammers.
- Rotating: This is the second largest group and where it is the unbalance in the working tool that is the main contribution to vibration. The unbalance also often varies during operation due to wear of the grinding wheel etc. Typical machines are angle and straight grinders and sanders.
- Combustion engine driven: These machines have more complex sources of vibration originating mainly from uncompensated forces from the piston/connecting rod/crankshaft system and the combustion. Typical machines are lawn mowers, chain saws and brush cutters.

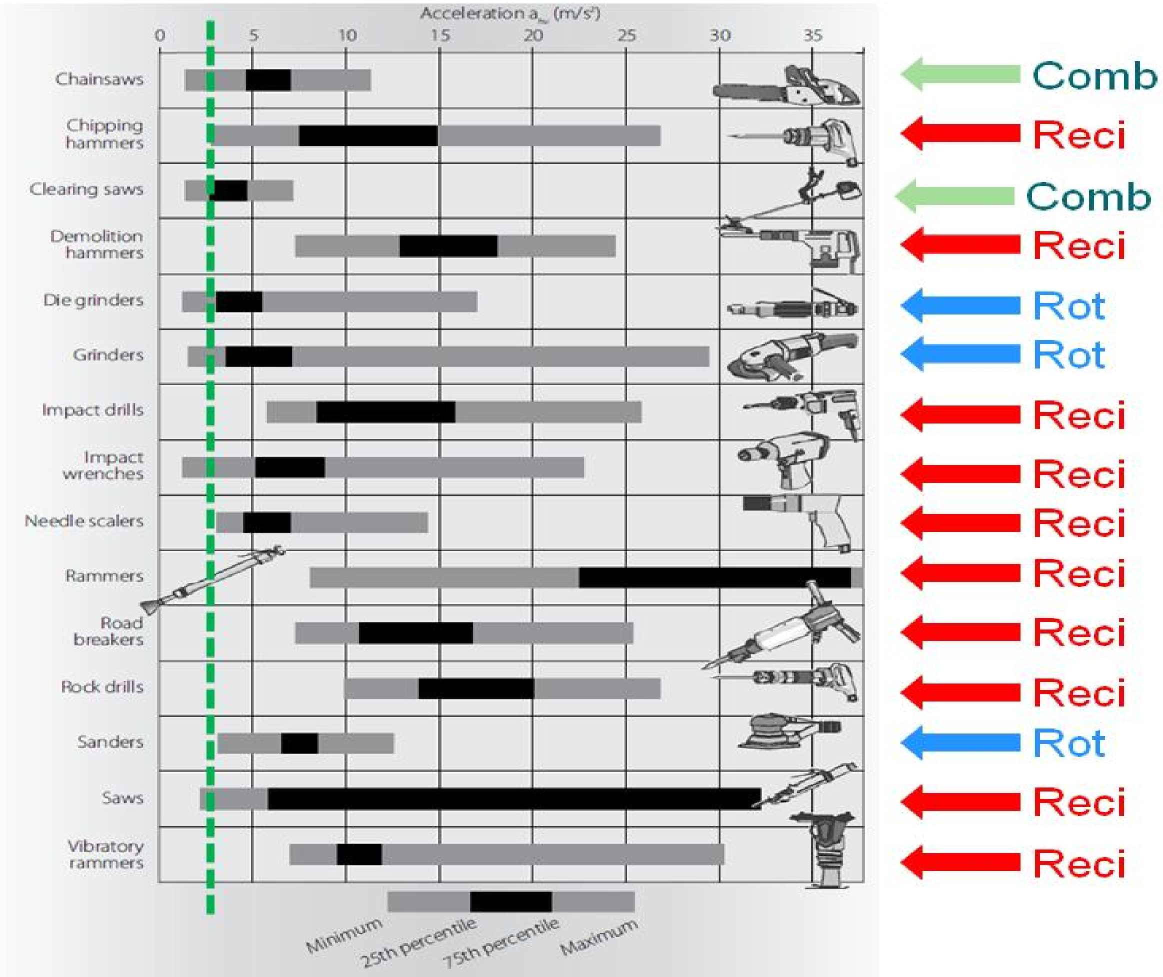

Figure 3 Machine types and their vibration level

The focus in this work has been to find a generic method that is robust and cost efficient to reduce vibrations from reciprocating machines which are the dominating sources of vibrations. As mentioned before, the vibration mainly originates from the reaction force from the piston hitting the work piece and the typical blow frequency range for these machines are between 10 and 60 Hz. There is also a contribution from the force from the process when the working tool, e.g. chisel, drill, is operating in the work piece, for instance a block of concrete or rock but these vibrations are usually not dominating but can contribute substantially but are highly depending on the specific operation.

For the group rotating machines with varying unbalances there are effective methods to achieve automatic and continuous balancing by using balancing rings with freely movable balls that compensate for unbalances in e.g. the grinding disc [4][5]. Also in the group of combustion engine driven machines are design solutions based on traditional vibration isolation that substantially reduces the vibration to the operator from chain saws and lawn mowers [6].

2.3 High frequency vibrations

The reason behind the design of the ISO 5349 weighting filter is that the finger is not equal sensitive to vibration at all frequencies. The highest sensitivity is found around 8 – 16 Hz and then declines and stops around 1250 Hz. The fundamental approach of ISO 5349 is that it is based on the assumption that the risk for damage follows the ability to sense the vibrations. This is also in line with the risk evaluation in the field of sound e.g. the dBA filter. This “What you can’t feel won’t hurt you.”

assumption is from a physical and mechanical view questionable. Even though the nerve sensor itself cannot sense a vibration it could still be damaged if exposed to a stimulus that is out of the sensing range. This is especially important in the case of vibration where the peripheral nerves are in almost direct contact with the stimulus. In the case of sound, the cochlear is to a large extent protected via the filter that the inner ear constitutes for sound at frequencies above the hearing range of 20 kHz.

The consequence of the weight function is that frequencies above 1250 Hz are not included in the standard for risk evaluation. This results in that potential risks for injuries from machines with high frequency vibrations are not considered at all.

However it has since long been suspected that HFV are likely to cause a significant amount of vibration injuries but there is inadequate understanding of the injury mechanisms. See Paper C and [7]–[12] for a more information on this topic. There is also no alternative standard assessing the risk associated with these vibrations. The current standard, ISO 5349, is by far the most dominant and constitutes the basis for almost all current directives and legislations such as the EU Vibration Directive. Neglecting potential risks from HVF results in that numerous occupational groups may be exposed to potential harmful vibrations without any regulations from worker's protection directives.

The authors of ISO 5349-1:2001, have clearly been aware of the problem with high frequency transient vibrations which is reflected in the scope where it states that it only covers vibrations within the frequency range of the octave bands from 8 Hz to 1000 Hz. Since there is currently no alternative standard for impact vibrations and there is awareness that ISO 5349 needs to be used for these anyway in the meanwhile it is also stated that it can only be used provisionally, quote, ***"Provisionally, this part of ISO 5349 is also applicable to repeated shock type excitation (impact)"*** and further:
"The time dependence for human response to repeated shocks is not fully known. Application of this part of ISO 5349 for such vibration is to be made with caution".

3 Research questions and objectives

3.1 Low frequency vibrations

The approach in this work to reduce vibrations from reciprocating machines is based on the technique with Tuned Vibration Absorber (TVA). These were invented by Frahm 1909 [13] and the function of them were described by Den Hartog [14]. The TVA comprises of an auxiliary mass that is connected to the main mass via a spring. The resonance of the auxiliary mass is tuned to the frequency of the exciting force and can be made to create a counter force of equal magnitude of the exciting force and thereby reduce the vibration of the main mass, Figure 4.

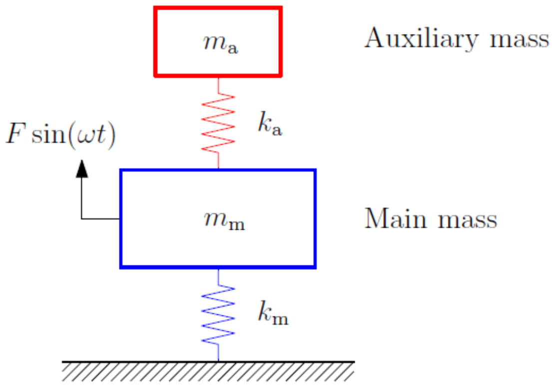

Figure 4, Tuned Vibration Absorber, [15] Unfortunately, the is TVA only effective in a very narrow frequency range. At higher frequencies, the auxiliary mass will start to make a phase shift towards 90 degrees relative the excitation force where it will go into resonance a greatly increase the vibrations of the main mass. At lower frequencies on the other hand the vibration reduction is rapidly decreasing. Due to the limited useful frequency range and the potential risk of increasing the vibration this technique cannot be used in hand-held machines. These machines have a relative large variation in operating frequency depending on factors like varying feed force, air supply pressure, manufacturing tolerances etc. Where TVA:s are frequently used are in tall buildings to suppress resonances excited from earthquakes but can also be found in applications where the frequency that needs to be suppressed is well known and do not vary. For example, it is used in the fuselage of the propeller aircraft SAAB 340 for suppressing vibrations from the blade passage frequency and is also found in cars for suppressing resonances in various structures. In most TVA:s found in practice a considerable amount of damping is introduced in order to achieve a more robust system for varying frequencies but to the cost of less vibration suppression.

The background behind the work performed at Swerea IVF with nonlinear TVA was initiated during an attempt to build a prototype machine of a pneumatic breaker for the stone industry based on a traditional linear TVA. The prototype was designed with a TVA based on an analytic model and manufactured. At the first attempt to run the machine the TVA was fitted with springs with a lower strength than intended since the correct springs were not available at the moment and there was no time to wait. When running the machine and varying the frequency by changing the air pressure it was found by serendipity that the vibrations were considerably lower in a much broader frequency range than what the analytic model had predicted. Even more surprising was that the expected phase shift where the TVA would instead increase the vibrations could not be reached at all. When disassembling the prototype, it was noticed that the springs in the TVA had been overloaded and permanently deformed and introducing a gap in the springs and thereby creating a nonlinearity in the TVA. What have happened was that the nonlinearity made the auxiliary mass to have a tuned frequency that would follow the frequency of the exaction force and thereby make it automatically tuning. Thereof is the reason for choosing the acronym ATVA which comes from Auto Tuning Vibration Absorber.

There are an extensive amount of earlier work made on nonlinear TVA, e.g. [16]–[19] but none of what have been found are directed towards the application on hand-held machines. Even more remarkably, no work has been found with the aim of controlling an excitation force out of the resonant area of the main system which is the case for hand-held machines. All literature found on nonlinear TVA are directed on reduction of resonance frequencies in structures.

There are also previous works with applying linear TVA to hand-held machines. One example is found in [20]. The machine was divided into two main parts consisting of the impact mechanism and a much lighter handle connected with a spring to the impact mechanism. Two TVA were then attached to the handle. The reason for using two TVA was to be able to suppress both the fundamental frequency and the first harmonic. The result showed a large reduction in vibration at the tuned frequency. However it is not described how the reduction would be affected by a variation in frequency. In the paper [21] the authors have also made a more general study on approaches on how to reduce vibrations from impact machines. However some of these solutions can be difficult to realize in practice.

The research questions in the field of TVA have mainly been the following:

- To what extent can the useful frequency range be widened by introducing nonlinearity in the TVA in hand-held machines?
- How can one explain the improved effective frequency range of the TVA by the introduced nonlinearities?

3.2 High frequency vibrations

The objectives with the study of HFV are to investigate the properties of vibrations on different machines and study how these vibrations are effecting the finger tissue. The results are intended to be used as input to the development of new regulations for HFV.

The research questions in the field of HFV have mainly been the following:

- What are the properties of the vibration when measured at high frequencies above >1250 Hz on different types of machines and how can they be measure accurately without effecting the measurement object?
- How will HFV from machines effect the finger tissue and to what degree will the skin texture of the finger prevent the HFV to propagate into the finger tissue?
- What are the possibilities to reduce the HFV by redesigning machines and thereby reduce HFV at the source?

4 Attenuation of low frequency vibrations

In order to reduce vibrations from reciprocating machines there is a need to find a robust solution to compensate for the reaction force from the piston that can work under varying frequencies and also needs to fulfill requirements such as, ability to function in harsh environment, low production cost and unaffected productivity.

Although impact machines have been used since the early 20th century, little has been changed in their fundamental design to date. Despite them being robust and efficient, vibration, noise, dust and poor ergonomics cause a large number of injuries to the operators. In Figure 5 is an example of a machine used for making holes for splitting stones with wedges. It is a pneumatic driven KV434 from Atlas-Copco and weighs 18 kg and has impact energy of 20 Joule. It consists in principal of a cast steel cylinder with a piston inside hitting the chisel. The interface to the operator is via two anti-vibration handles. The vibration is despite the handles approx. 20 m/s^2 hand-arm weighted acceleration according to ISO 5349 which gives a recommended daily exposure time that is only a fraction of an hour and regularly exceeded in daily use.

Figure 5 Chisel machine used in stone industry

The approach in this work on how to find a solution to reduce vibrations from these machines and fulfilling the other requirements is based on the traditionally TVA. However, the TVA will only function in a very limited frequency interval and this needs to be broadened in order to be used in practice.

This can be effectively solved by introducing a nonlinear spring on the auxiliary mass, see Paper A and Paper B. Figure 6 shows the engineering model together with a prototype of an auxiliary mass used in the prototype machine as seen in Figure 7.

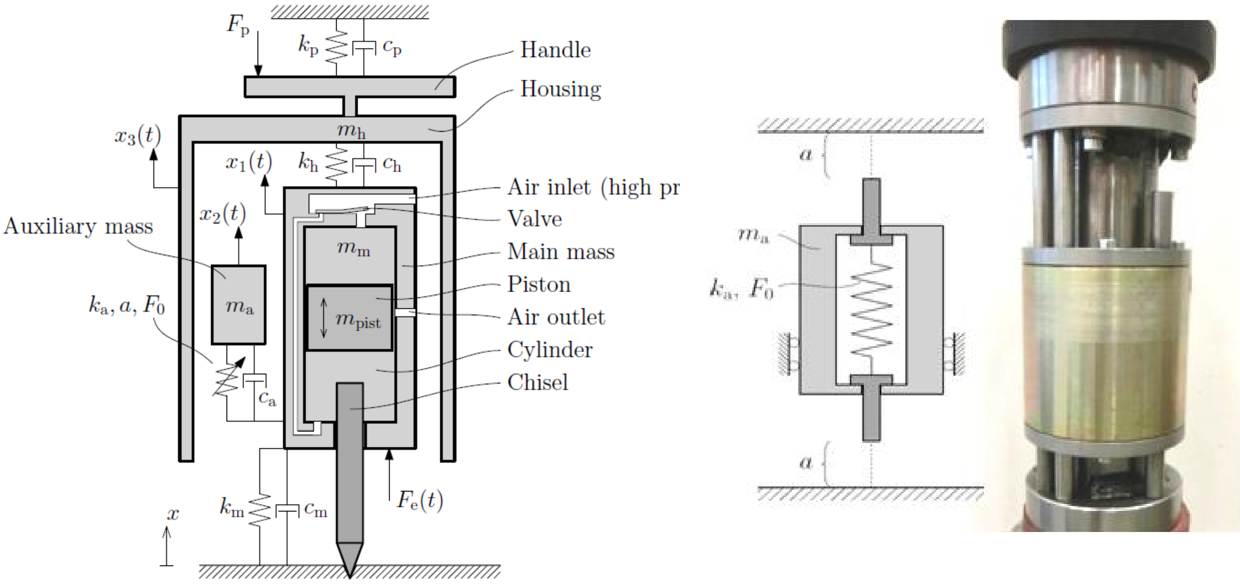

Figure 6 Left and middle, Engineering model of impact machine and auxiliary mass; Right, Prototype auxiliary mass

Figure 8 Prototype machine with ATVA

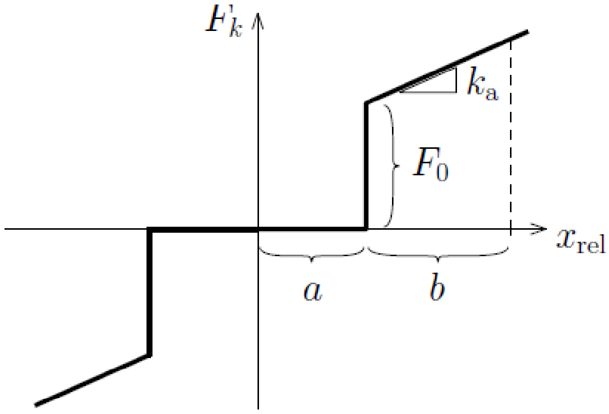

Figure 7 Nonlinear spring force in ATVA

The reduction of vibration in the prototype machine is made both by introducing ATVA and separating the machine into two parts, the impact mechanism and the user interface, and introduce a spring element in between as seen in the engineering model. The nonlinearity in the auxiliary mass system is made by introducing a gap and a pre-tension of the spring. This gives a force displacement characteristic as seen in Figure 8.

The engineering model has been simulated in a Matlab [15] [22] and the result on vibration in displacement in the handle as a function of frequency can be seen in Figure 9. Line A is the vibration with as built machine, line B is with spring installed between the two main parts, line C is with a linear TVA and line D finally is the vibration with a nonlinear TVA (ATVA).

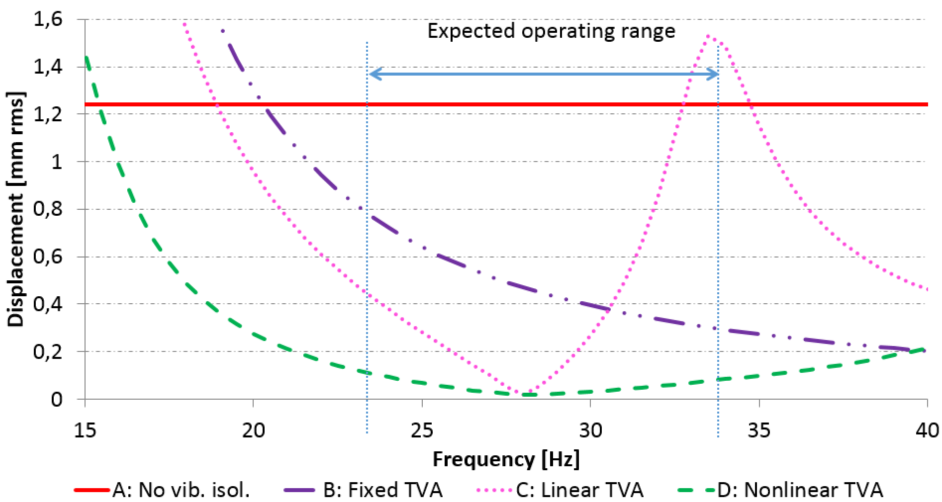

Figure 9 Vibration of the handle. Comparison between using the optimized Nonlinear TVA, the linear TVA tuned to 28 Hz, deactivated TVA (vibration isolation only) as well as no means of vibration reduction at all.

What can be found is that the ATVA greatly increases the effective frequency range and at the same time moves the resonance frequency above the operating area out of the potential operating range and thereby eliminates the risk of increasing the vibration.

The reason behind the increased frequency range is found by studying the phase relation between the excitation force and the auxiliary mass in Figure 10 and Figure 11 which is a close up. More details can be found in [22]. It can be seen that the phase of the nonlinear auxiliary mass remains around 180 deg, where it is effective in counter phase, between frequencies from 15 to 65 Hz. Whereas for the linear system the phase only stays around 180 deg between 22 to 31 Hz. It can also be seen that the resonance where the auxiliary mass increases the vibrations are drastically moved upwards from 32 to 72 Hz.

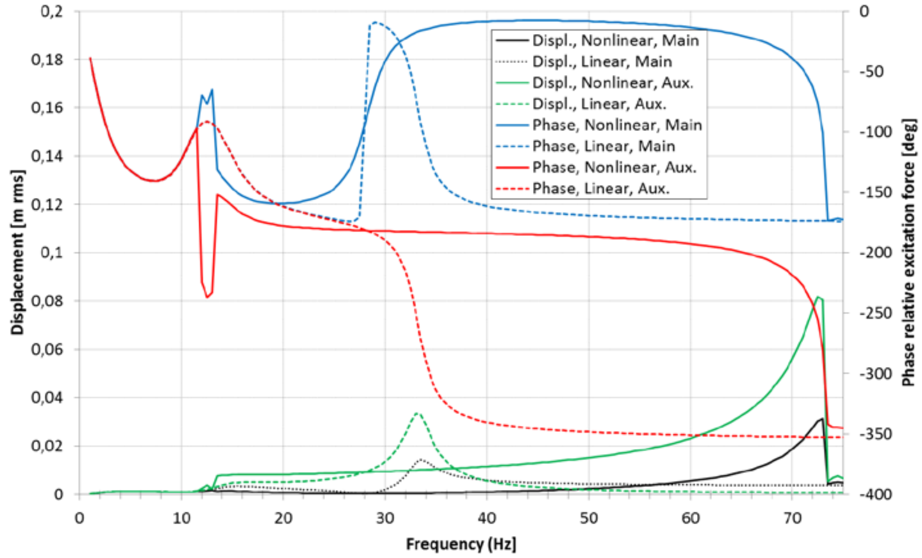

Figure 10 Phase and displacement for linear and nonlinear TVA

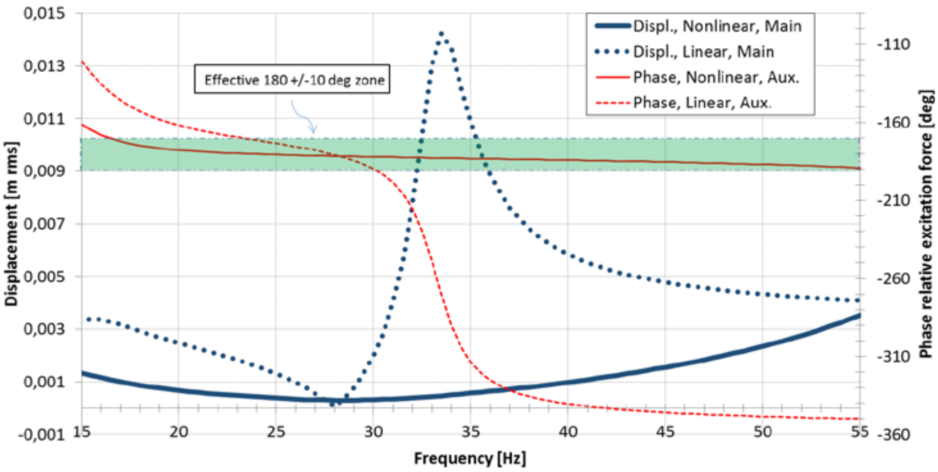

Figure 11 Phase and displacement for linear and nonlinear TVA, close up

The reason for the stable phase around 180 deg in turn is due to that when the excitation frequency increases in this mainly mass controlled system where the piston travels a constant distance gives a constant displacement to the main body regardless of the frequency. This will force the auxiliary mass to travel at higher speed and thereby compressing the spring deeper with increasing frequency. Since the spring is nonlinear and stiffening will the resonance frequency increase for the auxiliary mass with increasing excitation frequency. If then the nonlinearity is chosen in such a way that the resonance frequency increases with the same ratio as the excitation frequency the auxiliary mass can be made to automatically tune itself to the excitation frequency. In Figure 12 can it be seen how closely related the analytical calculated resonance frequency that is a function of the spring compression are to the excitation frequency. This is the motivation behind the name Auto Tuning Vibration Absorber or ATVA. More details on low frequency vibration attenuation by using ATVA can be found in the appended Papers A and B.

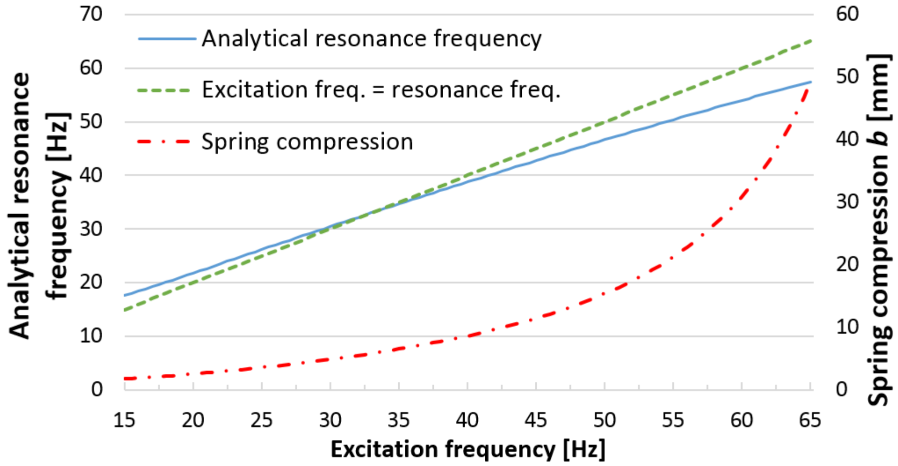

Figure 12 Analytical resonance frequency and spring compression versus excitation frequency

5 Measurement, modelling and reduction of high frequency vibrations

5.1 Measurement

In order to study the effects of HFV on finger tissue from hand-held machines it is of fundamental importance to be able to accurately measure the vibrations and there was a need to develop a measurement method. The measurement equipment that currently is used for hand-arm vibration measurements are most often based on piezoelectric accelerometers with housing in metal and are intended to measure vibrations up to a frequency of 1250 Hz. These accelerometers are difficult to use at much higher frequencies due the mass loading of the measurement object which often is a composite handle where they introduce resonance frequencies. Although there are dedicated shock piezoelectric accelerometers, e.g. Bruel&Kjaer, Model 8309, on the market they have a weight of several grams and need to be mounted to the test object with a treaded screw and thereby can seldom be used for hand-held machine vibrations. There are also optical systems as Laser Doppler Velociometers (LDV) but these are besides being very expensive often having a limited measurement displacement range which makes it difficult to use for field measurements.

However by taking use of the recent development of ultra light accelerometer technology based on Micro Electrical Mechanical Systems (MEMS) opens up new possibilities to study high frequency vibrations from hand-held machines. The advantage is that the weight of the sensor can be greatly reduced which ease the mounting and allows measurement on composite handles with much less loading effects. In this study, the accelerations were measured with a piezoresistive bridge shock MEMS accelerometer, model 3501A2060KG, from PCB, Figure 13. It weighs only 0.15 gram and has a 2 dB frequency range at 50 kHz and an amplitude range of 600,000 m/s^2. The resonance frequency of the accelerometer is 150 kHz.

Figure 13 MEMS accelerometer, PCB 3501 A2

The acceleration signal was anti-alias filtered with an analog 4:th order low pass Bessel filter at 200 kHz and sampled in the AD converter at 1 MHz. The signal was digitally low pass filtered at 50 kHz with a 6:th order Bessel low pass filter. The filter frequency of 50 kHz was chosen to prevent amplification from the accelerometer resonance at 150 kHz and ensure that the fixation method of the accelerometer would be rigid below that frequency. Bessel filters were chosen since they have a linear phase shift and thereby create minimum distortion of the time signal. The drawback is that they are not as sharp as Butterworth or Chebyshev filters. The mounting of the accelerometer to the machine handle is made by using Tack-it from UHU Patafix which is a high viscosity substance which makes the measurements easy to perform and was proved by experiments to gives a rigid connection to the measurement surface up to 50 kHz. See Paper C.

The time history of acceleration of the vibration from the handle of an impact wrench CP 734 made in aluminum is shown in Figure 14. The signal is shown both filtered at 30 kHz 6:th order Bessel and with the ISO 5349 filter. What can be found is that there are transient peaks with amplitudes in the region of 8 000 m/s^2. When filtering the signal with ISO 5349 in order to make risk estimation the peaks are reduced to a level of about 6 m/s^2. The reason for the large reduction is that the peaks have frequency content much higher than 1250 Hz and thereby heavily suppressed by the filter and will not be taken into account for risk evaluation. This results in a potential weakness of the risk estimation since the force that is subjected to the finger tissue from the vibrating machine is proportional to the acceleration.

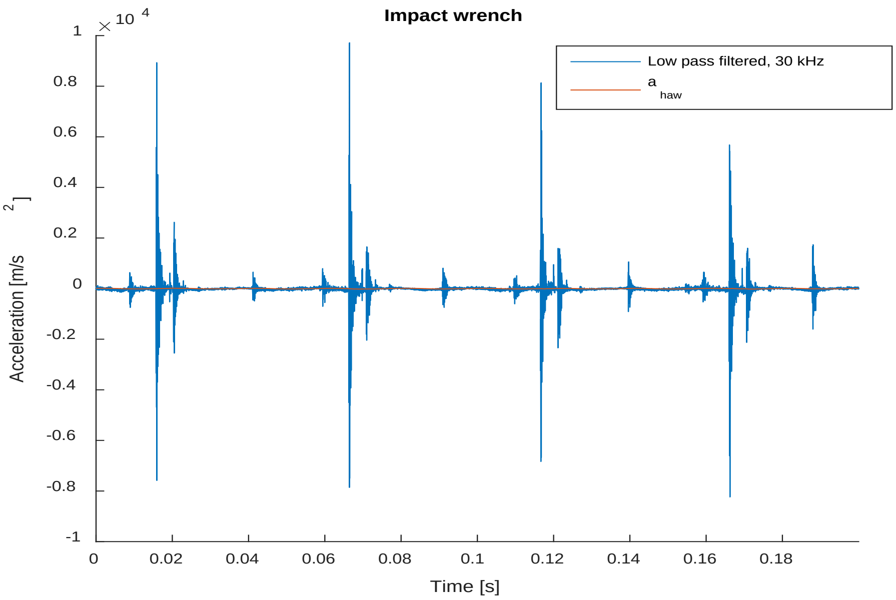

Figure 14 Vibration on impact wrench handle

5.2 Modelling of finger tissue

In chapter 5.1 it was shown that impact machines have high amplitude vibrations with a high frequency content. Although the acceleration amplitude is very high the displacement amplitude is very low due to the high frequency content. As an example, a sinusoidal vibration at 10 kHz with an amplitude of 10 000 m/s^2 peak, which is about of what is found on an impact machine, has a displacement amplitude of only 2,5 µm peak. Therefore has it commonly been arged that such a small displacement can not harm human tissue.

In order to study how transient vibrations affect the finger tissue a finite element (FE) model was developed. The objective with the model is to simulate how the vibrating surface of the machine interacts with the skin layers in the finger, and study how the created pressure wave propagates into the soft tissue of the finger. The simulation is described in more detail in paper C.

The prediction of wave propagation in viscous tissue material is modeled with a 2D plane strain finite element simulation model. It is solved by the multiphysics simulation program LS-DYNA, whereby the central difference method is adopted. The numerical simulation model consists of a finger model, discretized with 2-D plane strain continuum elements.

Initial simulations with a full 3D simulation model with relatively coarse discretization of the finger revealed that a 2D plane strain approach is valid at a distance of at least 25 mm from the tip of the finger in order to reach 2D plane strain conditions within the finger under the short period of the applied acceleration pulse.

The numerical simulation model of the finger includes the components of the human skin e.g. stratum corneum, living epidermis, dermis and subcutaneous tissue. The geometric properties of different skinlayers and the overall dimensions of the fingers are derived from findings published [23], [24]. Special attention is paid to the contour of the fingerprint where load introduction appears. The structure of the fingerprint of the skin is an important factor since it acts as a vibration isolator since it partly consists of air which is compressible in contrast to the material of the skin. Therefore an epoxy casting of the index finger fingerprint pushing on a plane plate with a force of 5 N was made. The casting was analyzed in a confocal microscope which built a 3D model of the finger print. The finger depth profile was then parameterized and described by five parameters representing the finger print profile. The exact dimensions are integrated into the simulation model, Figure 15.

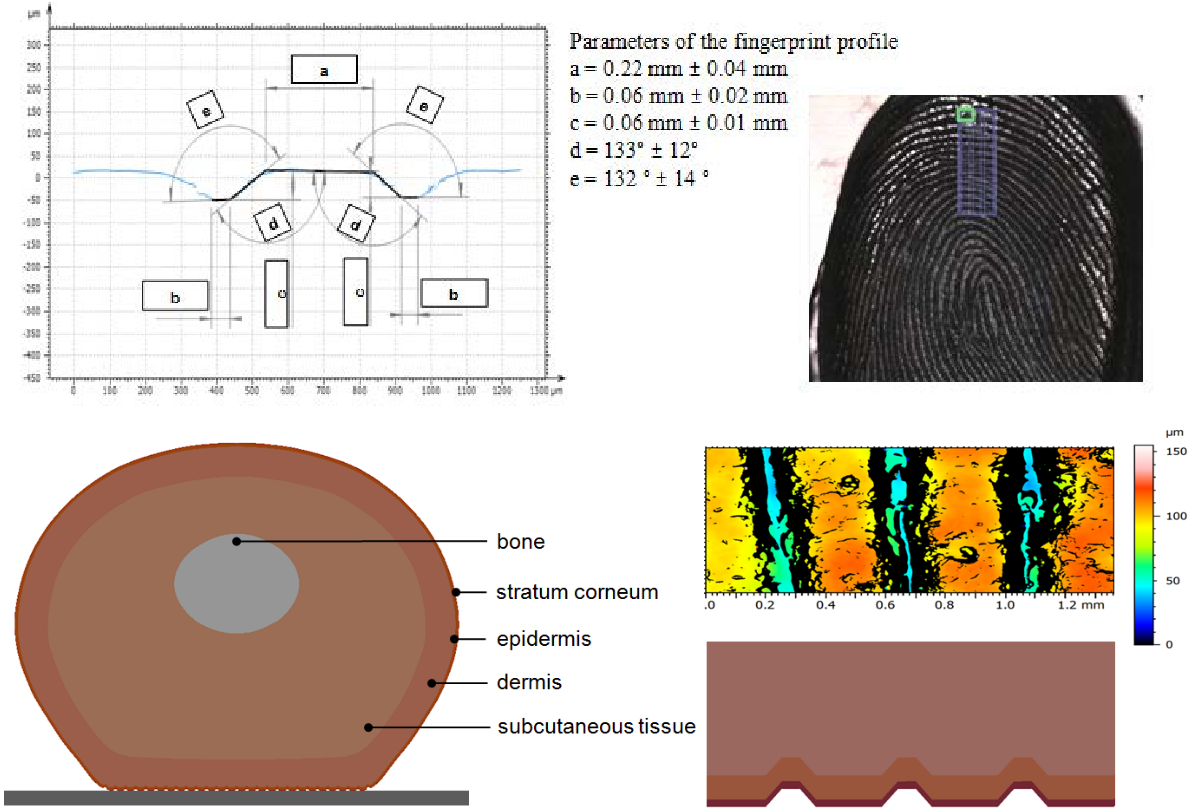

Figure 15 Numerical simulation model with experimentally validated geometry of the fingerprint

Published data on the mechanical properties of the skin layers stratum corneum, epidermis, dermis and subcutaneous tissue reveal differences in the order of magnitude, depending on test set-up, loading conditions, gender, age, location and environmental conditions was chosen from [24], [25]. However, the elastic material properties for dermis and subcutaneous tissue are taken from publications in [23] and the properties for bone material are taken over from [26]. In [27] a compression bulk wave speed of about 1500 m/s is listed for human skin from various investigations. Bulk modulus, density, the corresponding speed of sound and the shear modulus are listed in Table 1.

Component	Density [g/cm³]	Bulk-Modulus [MPa]	Shear-Modulus [MPa]	Soundspeed [m/s]
Stratum Corneum	1.04	2259.0	3.100	1500.0
Epidermis	1.04	2259.0	0.210	1500.0
Dermis	1.04	2259.0	0.080	1500.0
Subcutaneous Tissue	1.00	2161.0	0.034	1470.0
Bone	1.96	20070.0	7719.0	3200.0

Table 1 Material properties of the skin layers

The material response of the skin layers is time and history dependent and is therefore described by a viscoelastic constitutive model, based on exponential stress relaxation functions with shear relaxation behavior described in [28]. The viscoelastic material used in LS-DYNA utilizes the Zener model which is a configuration of a spring and spring-damper element in parallel.

$$G(t) = G_\infty + (G_0 - G_\infty)e^{-\beta t}$$

The viscoelastic behavior is described by the long term asymptotic shear modulus G_∞, the short term shear modulus G_0 and the stress relaxation time 1/ß. The stress relaxation time and long term shear modulus for subcutaneous tissue is taken from [29]. The long term shear modulus for the other tissue layers is adapted proportionally. The material data for the skin layers stratum corneum and epidermis stated in [23] is further refined by an experimental investigation of the finger and fingerprint distortion under compressive forces. The experimentally evaluated fingerprint geometry in both uncompressed and compressed state is used to validate the finite element simulation model in an inverse optimization approach. With this approach the shear modulus of the skin layers in the numerical simulation model are verified, see Figure 16.

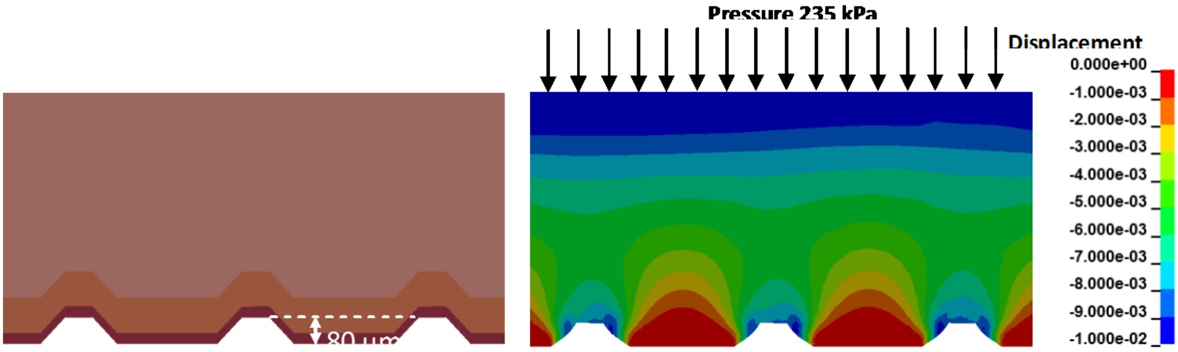

Figure 16 Unloaded geometry of the fingerprint (left) and numerical validation of the fingerprint distortion under constant pressure loading (right)

The volume or bulk viscosity of tissue material is hardly investigated in literature for frequencies below 1 MHz. The most comprehensive overview is published in [30] where the sound attenuation coefficient of human skin is defined to 0.35 dB/cm MHz and at least decreasing linearly towards lower frequencies. With this in mind and because of the short time period investigated the sound attenuation coefficient is disregarded in the present study and left as a topic for future research.

The metal plate acting on the fingerprint is accelerated by a single sinusoidal acceleration pulse characteristic for hand held tool vibrations with:

$$a(t) = \frac{A}{2}\left[1 + \cos(\pi + \frac{2\pi}{T}t)\right]$$
$$0 \le t \le T$$

Variant A: A=10 000 m/s^2 and T= 0.1 ms
Variant B: A=100 000 m/s^2 and T= 0.01 ms

The pressure is evaluated in centered position throughout the height of the different skin tissue layers to capture the transient propagation and subsequent reflection of the pressure waves, see Figure 17.

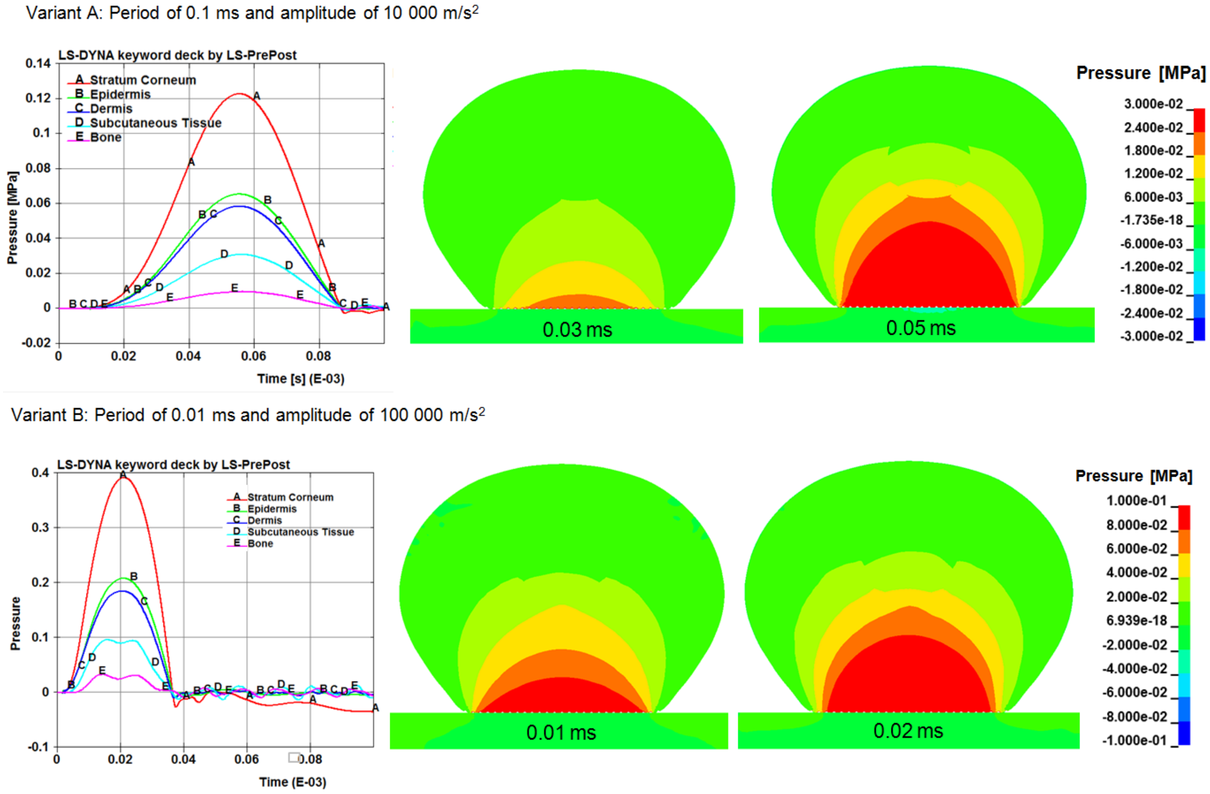

Figure 17: Pressure distribution after load initiation and corresponding pressure propagation in the different finger sections for Variant A and B.

The numerical simulation model unveils a significant pressure level in the finger under the transient acceleration pulse. For Variant A, a pressure of at least 0.6 bar is reached in the 3 outer tissue layers and for Variant B, the pressure level is increased to more than 2 bar in the outer three tissue layers. However, there is still further experimental research necessary to experimentally verify the viscoelastic material response of the different skin layers as publications over viscoelastic tissue material properties differ considerably.
Investigations in [31] and [32] on strain rate behavior of skin material revealed that elastic properties can increase significantly for strain rates in the same range as seen in the current simulation. Furthermore, in literature studies [33] a strong dependence of water content on elastic skin properties is found.

5.3 Reduction of HFV by redesign

Since there is a potential risk that HFV will increase the risk for vibration injury it is therefore of interest to investigate the potential to reduce these vibrations. It was found that the possibility to reduce HFV by redesign is high especially if it is done in the design stage of the tool.
In Paper C are two examples of preventive measures described, an impact wrench and an anvil in which design modifications accomplished large reductions on peak accelerations. The method used for reduction is based on introducing vibration isolation as close to the vibrating source as possible. The vibration isolation can preferable be made in foamed polymer to achieve an effective isolation even at higher frequencies.

6 Summary of appended papers

6.1 Paper A

This patent (Lindell H.: Impact Machine, Patent, US 201700.95920A1) is an extension of a preceding patent, EP13811461.6. It describes the effects of the widened effective frequency range by introducing nonlinearities. It describes a numerous of design variation and especially a variety where the spring is placed inside the ATVA mass. This gives an axially more compact design and also substantially increases the expected time of life for the spring since the relative compression in each working cycle is reduced due to that the spring can be made longer with the same total length of the ATVA.

6.2 Paper B

Where Paper A describes the effect of the ATVA on widening the useful frequency range and different design concepts, Paper B gives an explanation on why this effect occurs and the optimization of it. There is also a description on verification and experimental validation of the simulation model.

6.3 Paper C

This paper is included in the report from the work shop on "Shock Vibration" held at the 13:th International Hand-Arm Vibrations Conference in Beijing 2015 [34]. Paper D is a revised version of the included paper.

It describes issues in the field of HFV on how to measure the HFV content on machines, modeling the pressure wave propagation in finger tissue and a literature review on biological effects.

7 Conclusion

It was shown that there is a potential to substantially reduce vibrations from hand-held machines with reciprocating action by using the ATVA technique. Recent field tests with pneumatic impact breaker prototypes have also shown that the system is robust and is able to withstand the harsh environment in the stone industry.

It was shown that machines with impact mechanism such as impact wrenches have vibrations with high amplitude and high frequency accelerations, HFV, which are not taken into consideration by the current ISO 5349 standard for evaluation of risk. These vibrations can be measured accurately with ultra light MEMS accelerometers up to 50 kHz. In a simulation model, it is shown that these vibrations propagate into the finger tissue and creates a pressure wave. Finally, it was also shown that there is a large potential to significantly reduce HFV by redesign.

In summary, there is a large potential to reduce vibrations, both ISO 5349 weighted and HFV, from the vast majority of hand-held machines by redesign and thereby reduce one of the largest work related injuries that affect the working force.

In order to accomplish this is it important that the standard for estimating risks from vibration embraces all vibrations that affect the human in order to give an incentive for manufacturers to design tools with low vibrations. This is not the case for the current ISO 5349 with respect to HFV and there is need for improvement. Hopefully will this work contribute to the process in the development of a future improved standard for estimating risk for injury.

8 Summary of research findings

The specific research questions that have been raised in this work and the related findings are listed below. The first two are related to ATVA and the last three to HFV.

Q1: ***To what extent can the useful frequency range be widened by introducing nonlinearity in the TVA in hand-held machines?***

A1: By introducing nonlinearity in a TVA the useful frequency range can be extended substantially to a degree where the technique with ATVA can be practically implemented into hand-held machines which are rarely the case for a linear TVA. In the case of a hand-held pneumatic machine for the stone industry which is described in more detail in Paper A and B, simulations gave that the frequency span with an 85 % reduction was increased from 3 Hz to 17 Hz around the centre frequency of 28 Hz, see Figure 9. Measurements in test rig and on machine prototypes have confirmed the extended frequency range by the ATVA. However the level of reduction is less in the experiments than in the simulations. The experiments showed a reduction around 80 % whereas the simulations gave a reduction of more than 95 %. The reason for the discrepancy in not fully known but factors as non sinusoidal excitation force, variation in feed force and difficulty to estimate the coupling stiffness and dampening to the ground are suspected to contribute.

Q2: ***How can the improved effective frequency range of the TVA by the introduced nonlinearities be explained?***

A2: The linear TVA has a very narrow effective frequency range where the auxiliary mass is in counter phase to the excitation force. Slightly above this frequency will the auxiliary mass start to make a 90 deg phase shift and enter into a resonance frequency which will cause the vibration to increase.

However by introducing a nonlinear, stiffening spring the resonance frequency of the auxiliary mass of the ATVA can be made to automatically follow the variation in excitation force of the main mass. Also the resonance frequency with the 90 deg phase shift is also moved to a considerably higher frequency compared to the linear TVA. The reason behind is that the auxiliary mass is forced to travel at higher speed with increased frequency in order to be able to complete a cycle and thereby hits the spring with higher velocity and in turn compresses the spring to a higher degree. Since the system is made nonlinear and stiffening the resonance frequency will automatically follow the excitation frequency and thereby is an automatically tuning of the vibration absorber achieved which is given the acronym ATVA from Auto Tuning Vibration Absorber.

Q3: ***What are the properties of vibrations when measured at high frequencies above >1250 Hz, (HFV), on different types of hand-held machines and how to measure it accurately without effecting the measurement object?***

A3: It is shown that hand-held machines with an impact mechanism such as impact wrenches and chisel machines have vibrations with high amplitude and high frequency content and that these vibrations are not taken into account when estimating related risks for injuries with the current standard ISO 5349. The measured vibrations on the handle of an impact wrench up to 50 kHz in the time domain shows acceleration amplitudes in the region of 10 000 m/s^2. Measurement on the socket shows amplitudes in the order of 100 000 m/s^2.

Measuring HFV can be accurately made by taking use of ultra light MEMS accelerometers with an internal resonance frequency of 150 kHz and low pass filter the signal at 50 kHz. Due to the low weight and density of the accelerometer they can be attached to the measurement object by e.g. tack-it or hot melt glue and accurately measure vibrations up to 50 kHz.

Q4: ***How will HFV from machines effect the finger tissue and to what degree will the skin of the finger prevent HFV to propagate into the finger tissue?***

A4: It is shown in a FE simulation model that HFV is generating a pressure wave that is propagating into the finger tissue via the skin. The skin with its finger print consisting of epidermis with its outmost part, stratum corneum, will transfer the main part of the vibration into the finger tissue and only partly reduce the propagation of the wave.

Q5: ***What are the possibilities to reduce the HFV by redesign of machines and thereby reduce HFV at the source?***

A5: Prototypes on impact wrench, chisel machine and anvil have shown that there are possibilities to substantially reduce the HFV content by redesign. In the prototypes have the HFV content been reduced to a degree from 98 % for the anvil down to 80 % for the impact wrench.

The redesign have been made by introducing a vibration isolation layer as close to the source as possible and is a relative minor design modification that will not affect the performance of the machine and only marginally increase the production cost. The vibration isolation layer is made by foamed polymer in order to achieve effective vibration isolation over a broad frequency range.

9 Future research

In the area of ATVA there is a need to search for the limits of the technology and then work to extend them. There is also a need to study the feasibility of ATVA on other fields of use beyond hand-held machines.

In several cases the weight of the machine can be reduced in combination with reduced vibration level due to less stress on components and no need to reduce vibration by adding weight.

In the area of HFV, experimental studies on how they affect biomarkers and animal models would greatly improve the understanding on the affect on humans. Also a modeling of finger tissue and the effects from HFV inside the tissue would increase the understanding of the damage mechanism on the biological tissue. There is also a need to validate the simulation with experiments e.g. measurement of pressure inside finger tissue.

The hypothesis that HFV can be a part of causing HAVS was raised already in the 1980:s and has still not been properly answered. It is therefore important to continue to put effort on investigating the effect of HFV since it constitutes a large potential risk to workers but there is also an even larger potential in preventive measures by redesign of the tools if the hypothesis turns out to be correct.

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PART II: Appended papers A-C

Paper A

Impact Machine, Patent, US 201700.95920A1

US 20170095920A1

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(12) **Patent Application Publication** (10) **Pub. No.: US 2017/0095920 A1**

LINDELL (43) **Pub. Date: Apr. 6, 2017**

(54) **IMPACT MACHINE**

(71) Applicant: **SWEREA IVF AB**, Molndal (SE)

(72) Inventor: **Hans LINDELL**, Vastra Frolunda (SE)

(21) Appl. No.: **15/381,277**

(22) Filed: **Dec. 16, 2016**

Related U.S. Application Data

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Publication Classification

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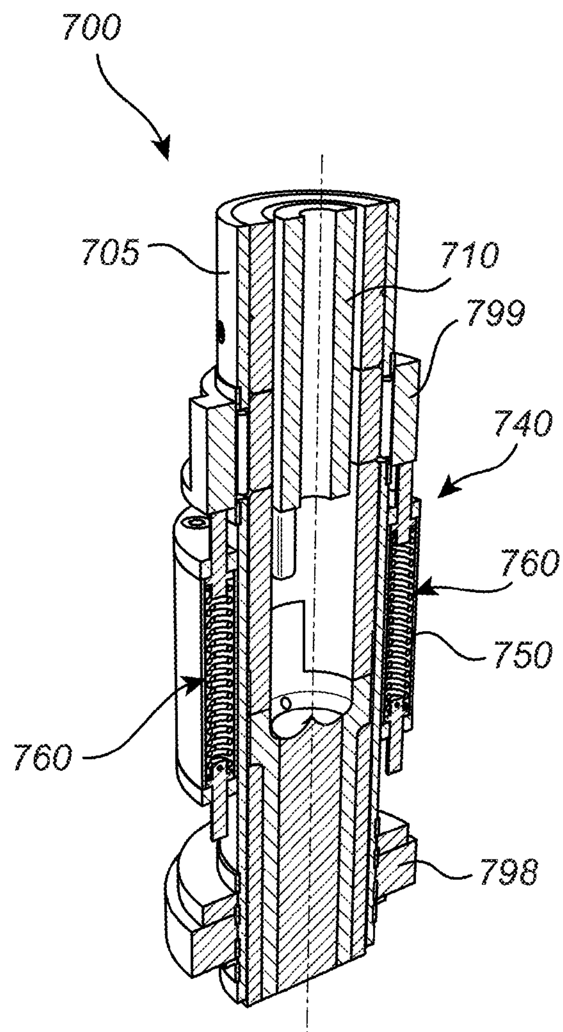
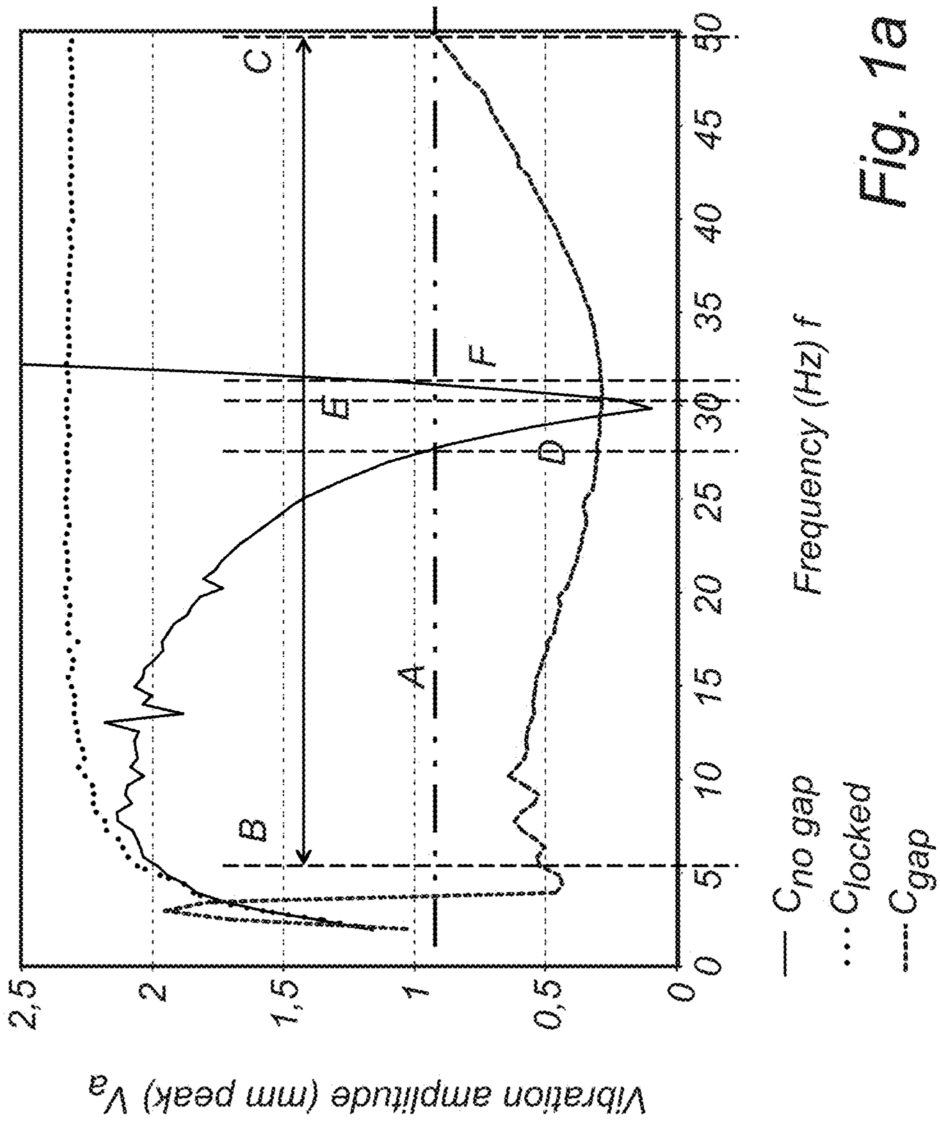
B25D 17/24 (2006.01)
F16F 7/116 (2006.01)

(52) **U.S. Cl.**
CPC ***B25D 17/24*** (2013.01); ***F16F 7/116*** (2013.01); *B25D 2217/0092* (2013.01); *B25D 2250/245* (2013.01); *B25D 2250/275* (2013.01)

(57) **ABSTRACT**

The invention relates to an impact machine which is adapted to perform a hammering operation on a surface or an object to be worked upon. In particular, a vibration reduction arrangement is attached to the housing and comprises a moveable counterweight, interacting with a motion reversing arrangement having a non-linear spring characteristics, such that the motion of the counterweight can be brought into a counter-acting movement in relation to the vibrations in the housing of the hammering element thus substantially reducing the vibrations. A spring action arrangement is arranged inside said counter weight, the counter weight being movable a first distance without actuating the spring action arrangement, and the counterweight comprises displaceable projecting member for actuating the spring action arrangement.




Vibration amplitude (mm peak) V_a
2,5
2
1,5
1
0,5
0
0
5
10
15
20
25
30
35
40
45
50
Frequency (Hz) f
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B
C
D
E
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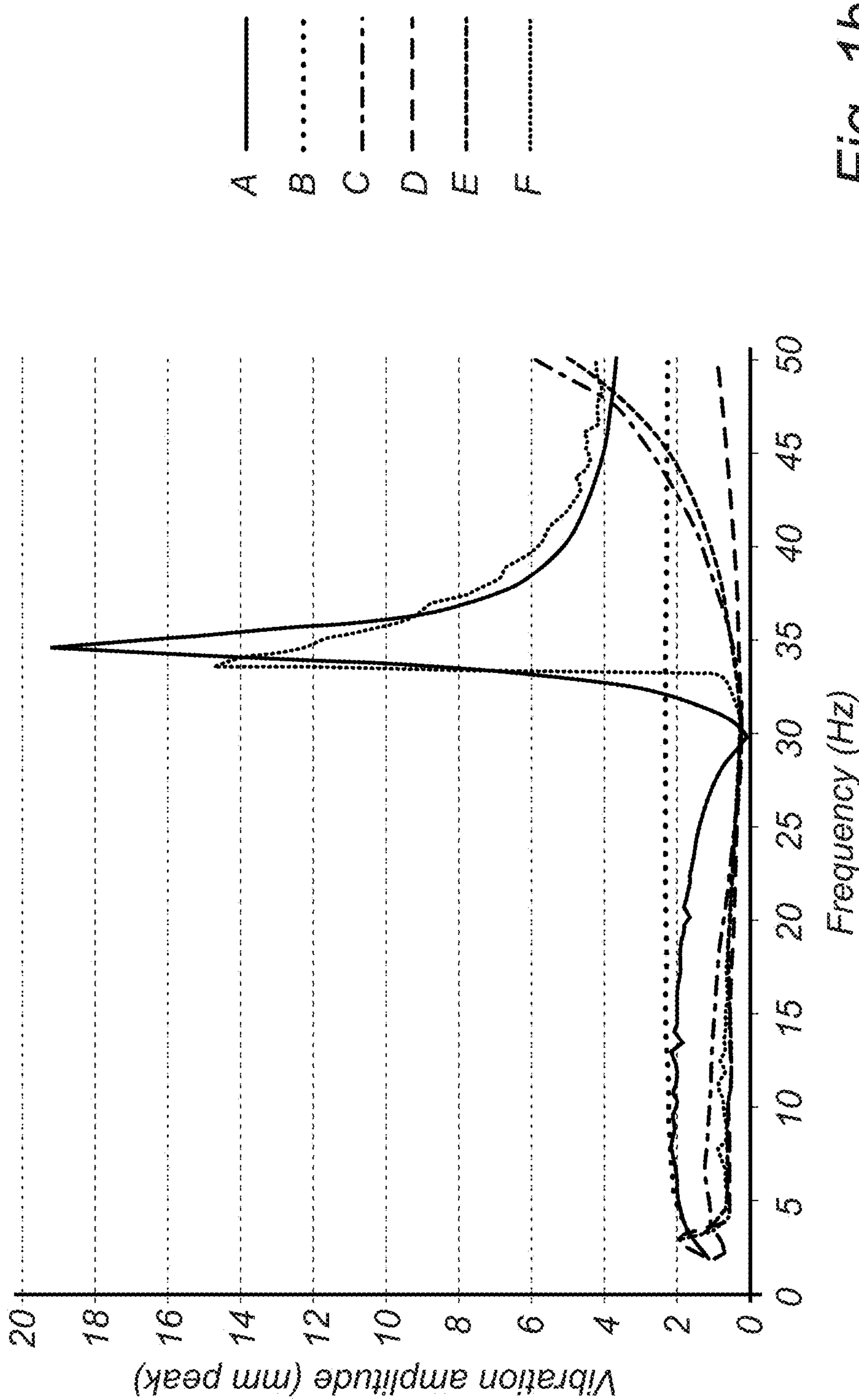
$C_{no\ gap}$
C_{locked}
C_{gap}

Fig. 1a


A
B
C
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E
F
Vibration amplitude (mm peak)
0
2
4
6
8
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12
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Frequency (Hz)
0
5
10
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50

Fig. 1b

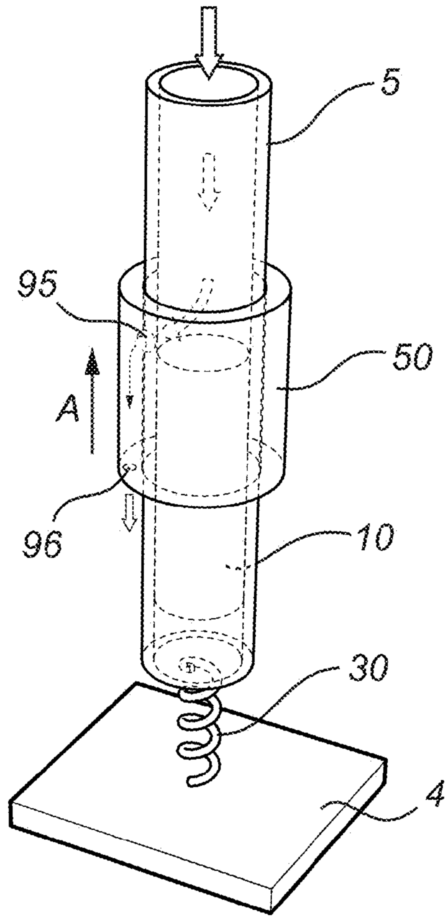


Fig. 2a

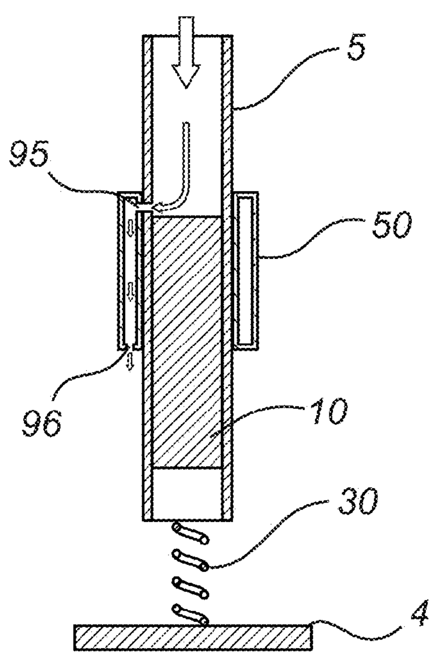
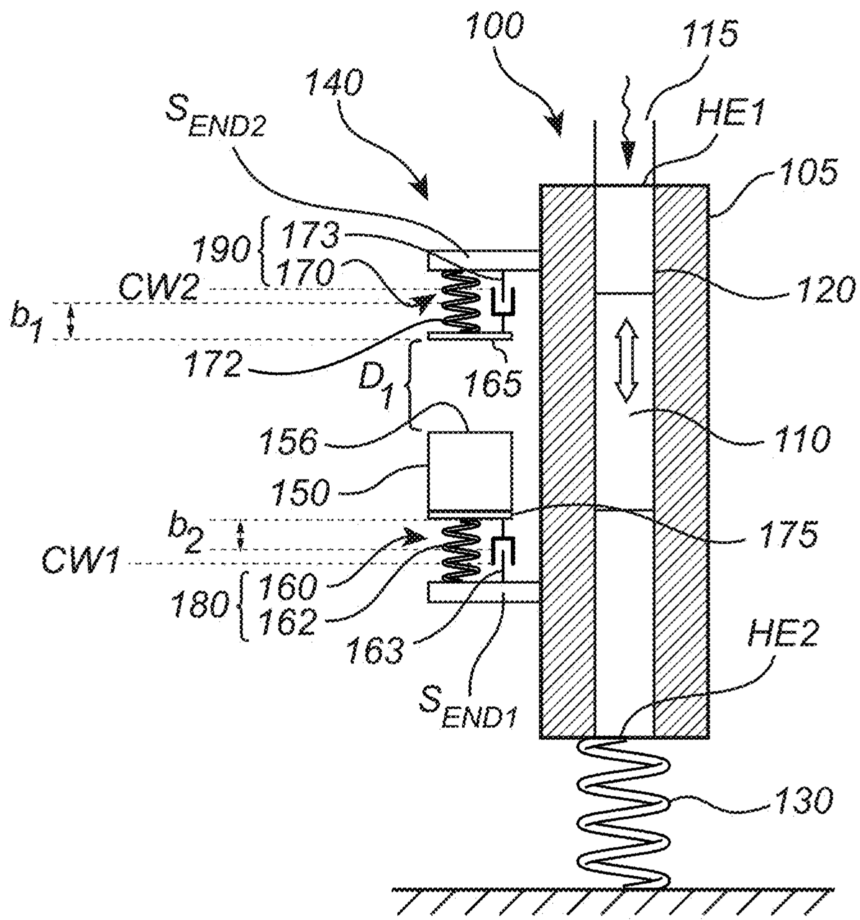


Fig. 2b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S_{END2}
HE1
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CW2
120
b_1
172
D_1
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110
150
175
b_2
CW1
160
180
162
163
HE2
S_{END1}
130

Fig. 3a

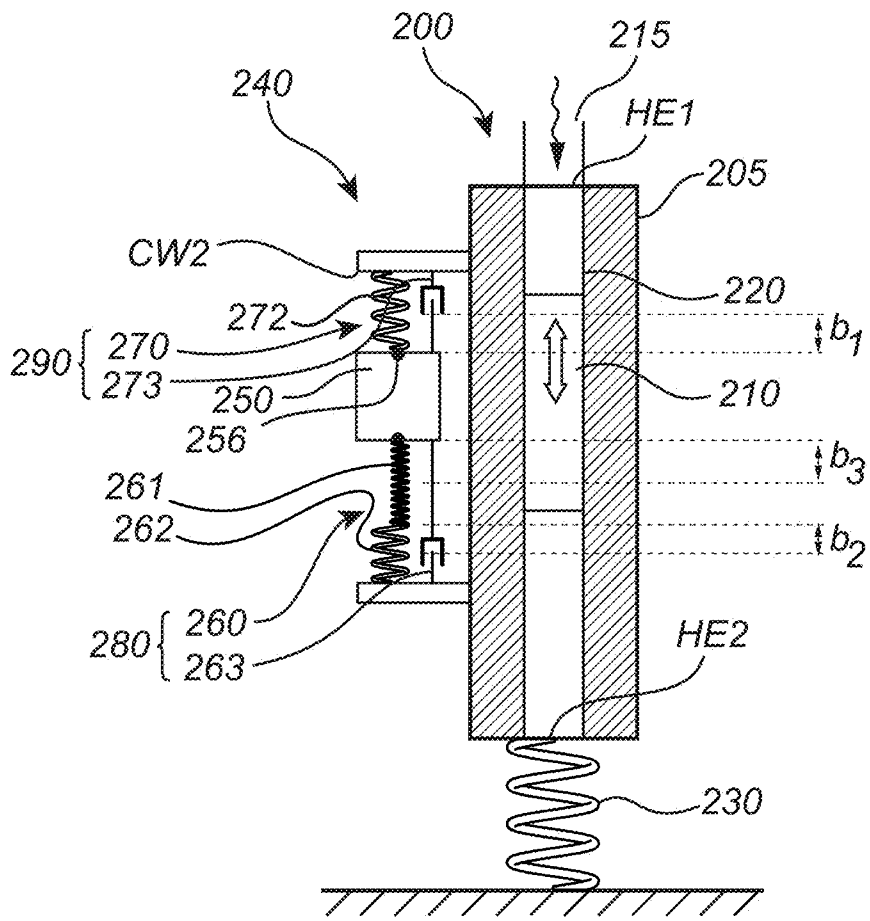

200
215
240
HE1
205
CW2
220
272
b_1
290
270
273
250
210
256
b_3
261
262
b_2
280
260
263
HE2
230

Fig. 3b

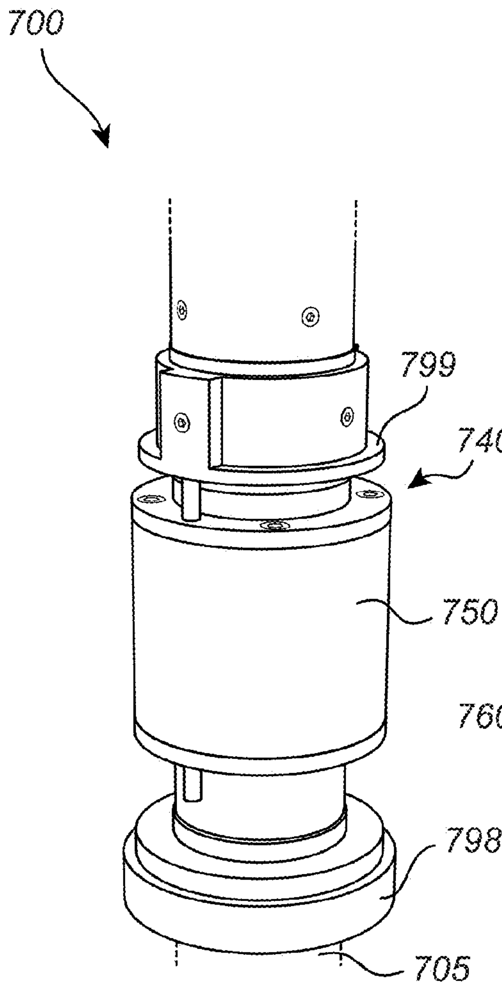


Fig. 4

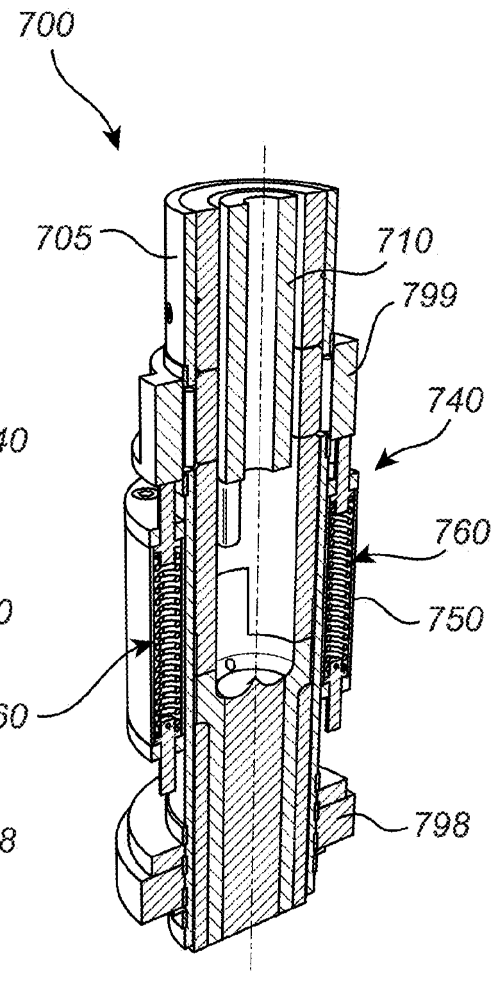


Fig. 5

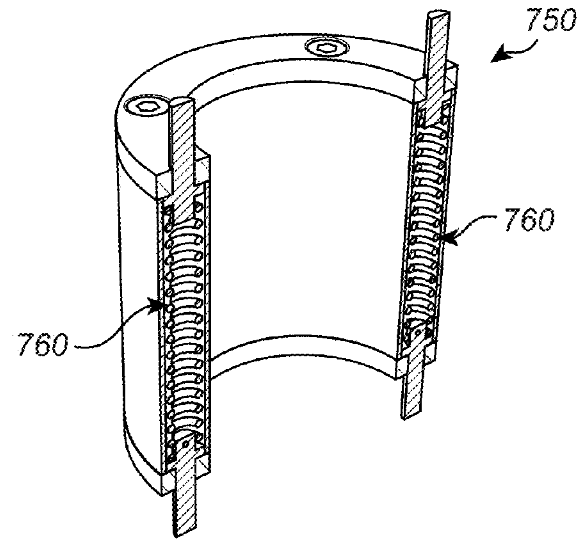


Fig. 6

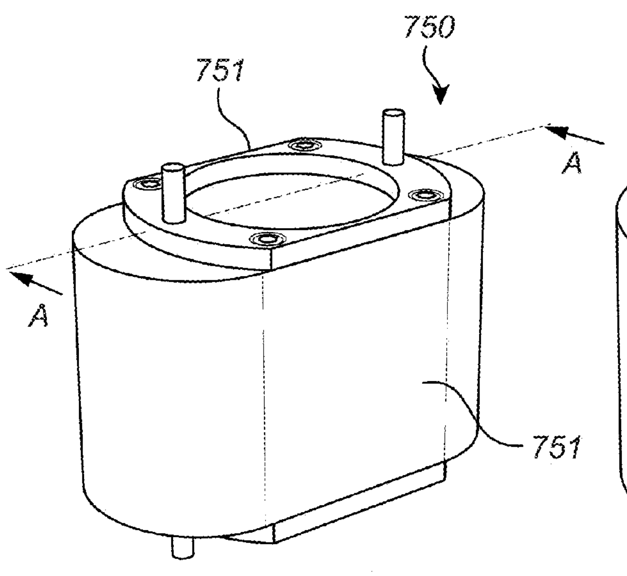


Fig. 7

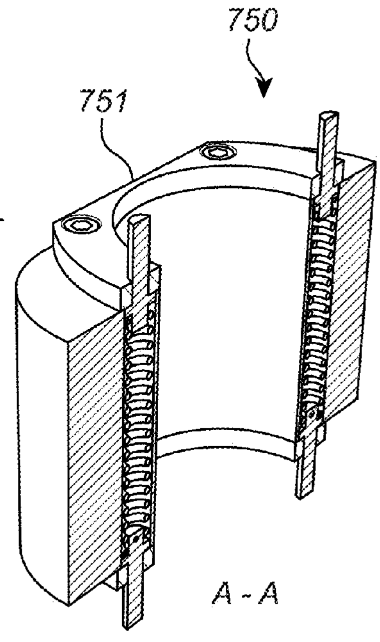


Fig. 8

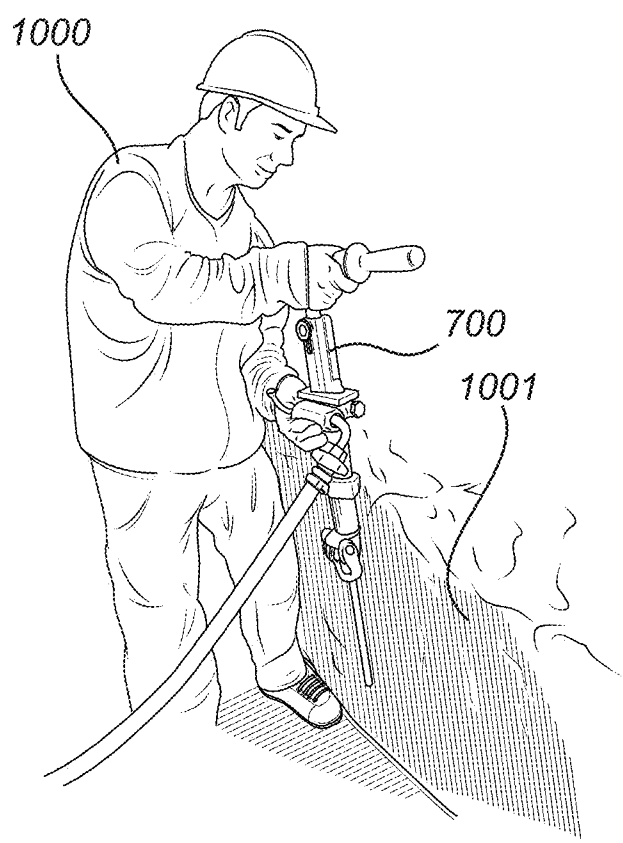


Fig. 9

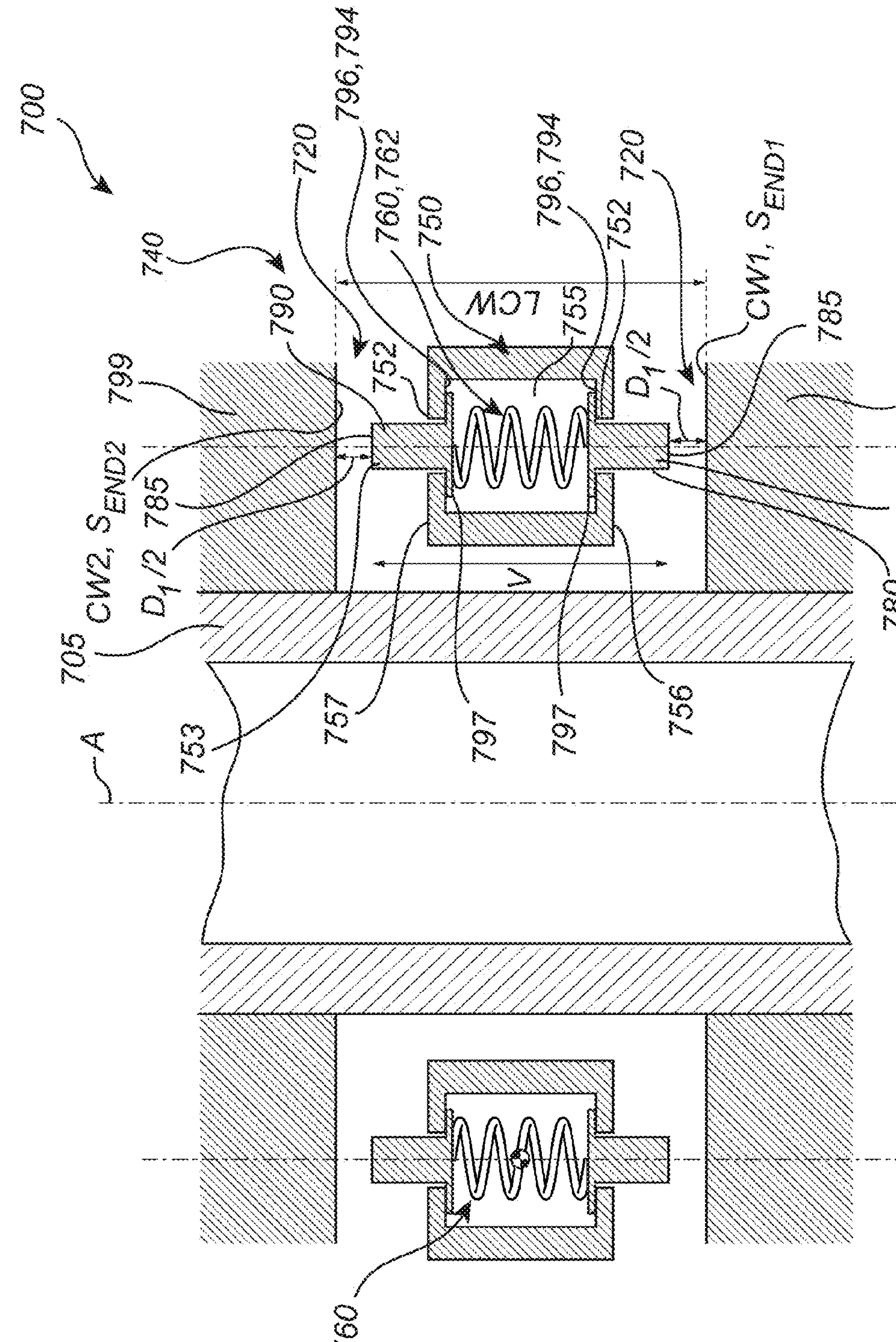

700
740
720
796,794
760,762
750
796,794
752
720
CW1, S_{END1}
LCW
755
752
790
752
799
$D_1/2$
785
798
785
CW2, S_{END2}
753
$D_1/2$
780
V
705
753
757
797
797
756
A
760

Fig. 10a

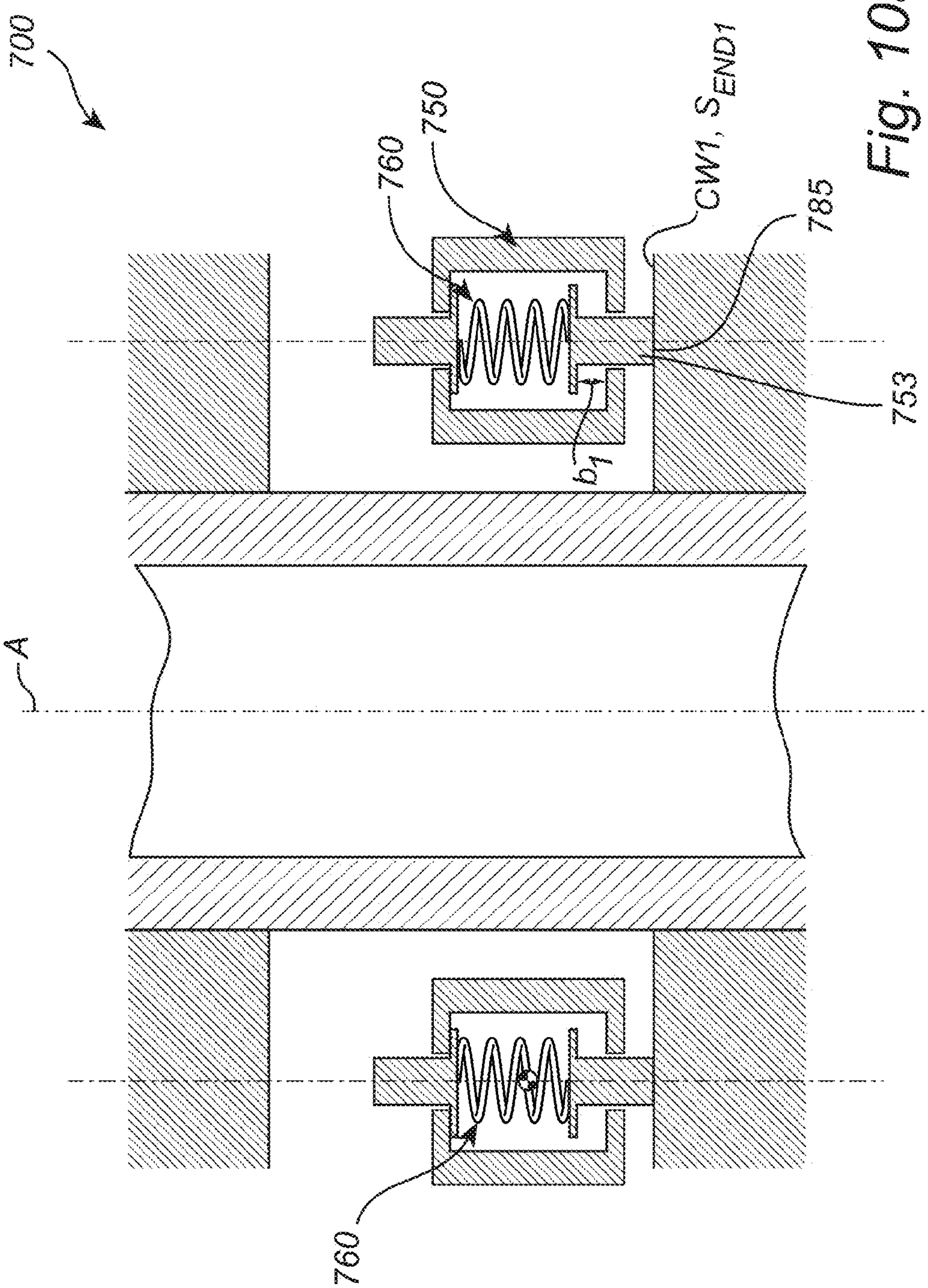
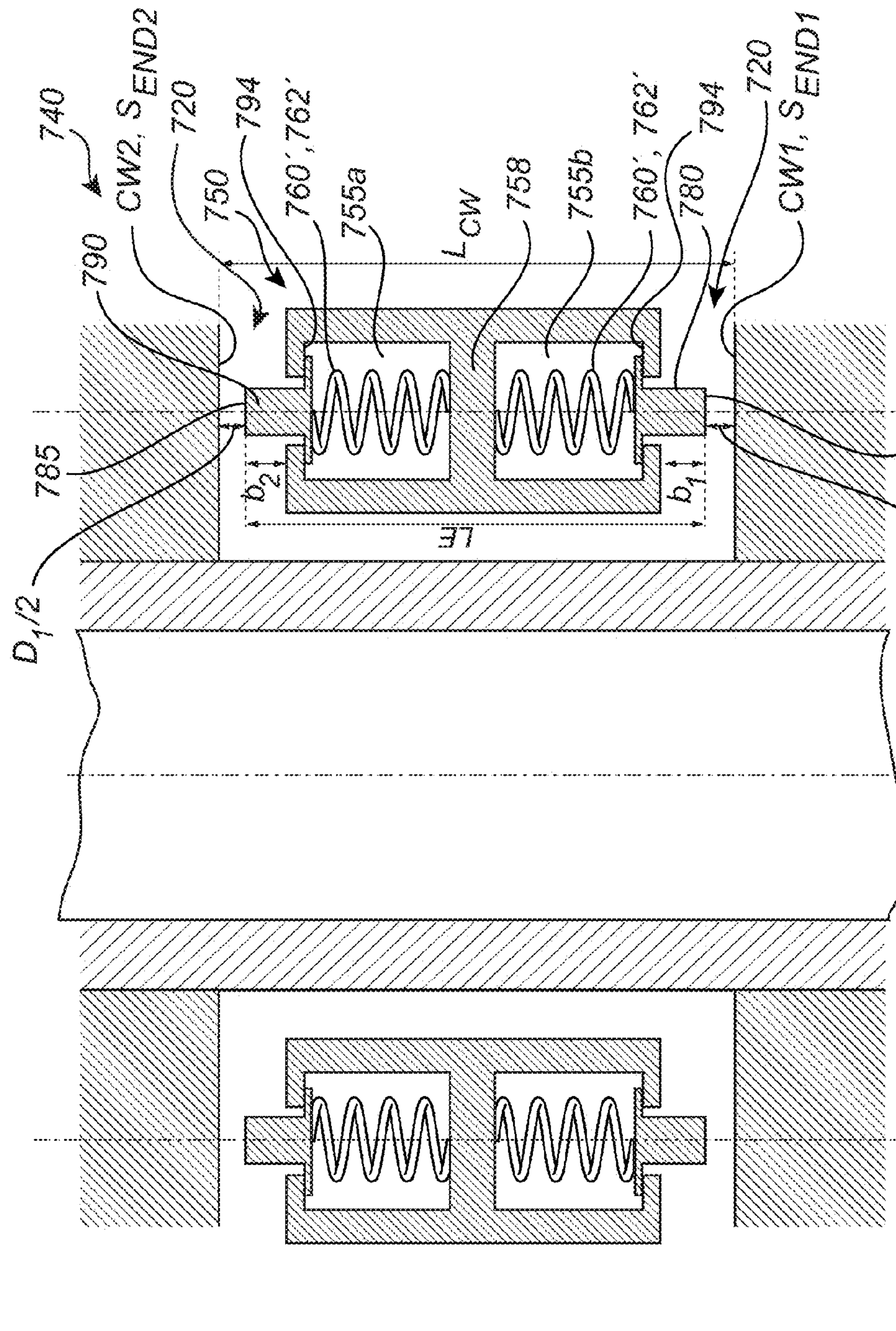
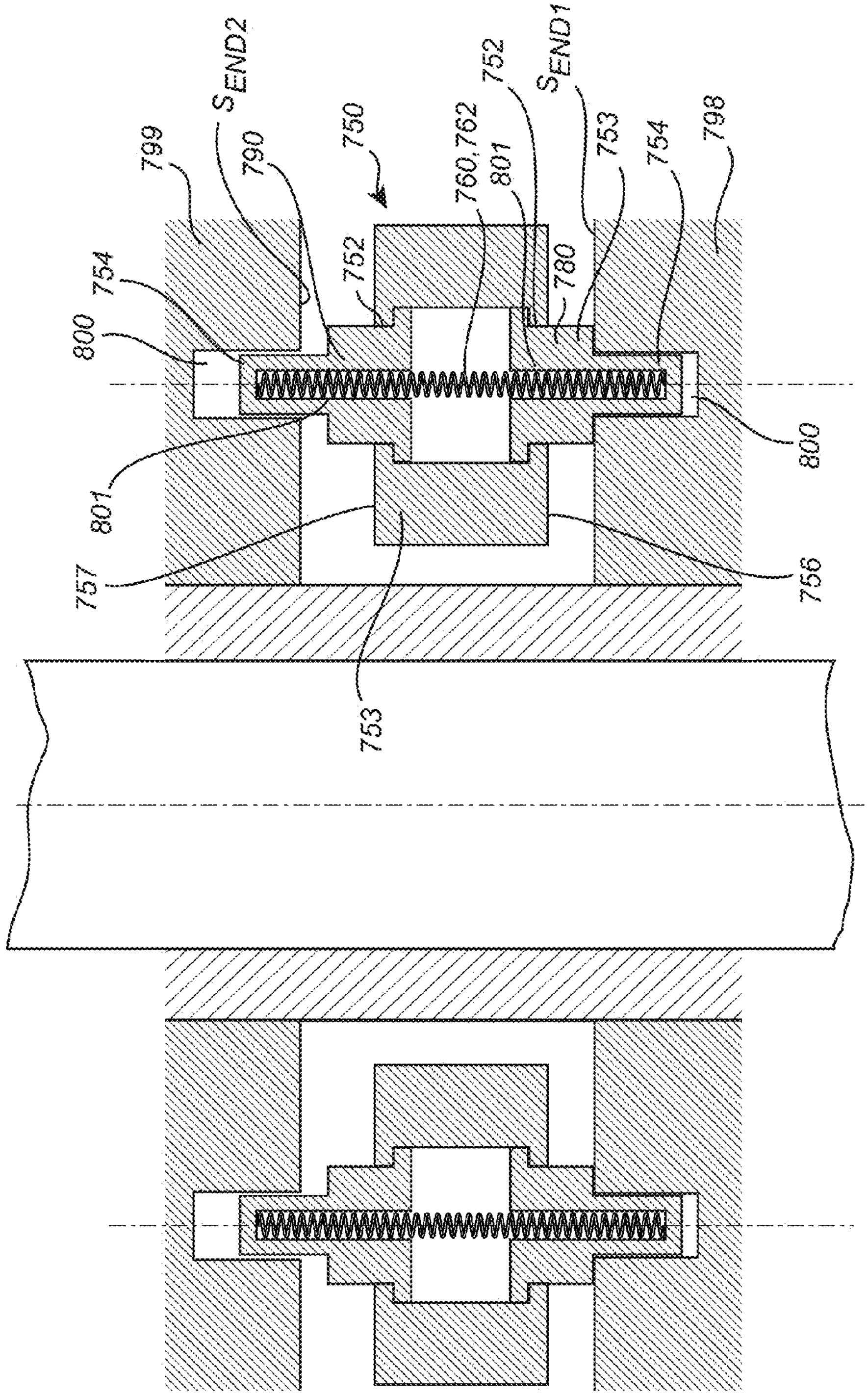


Fig. 10a´



740
CW2, S_{END2}
720
750
794
760′,762′
755a
L_{CW}
758
755b
760′, 762′
794
780
720
CW1, S_{END1}
790
785
b_2
b_1
LE
$D_1/2$
785
$D_1/2$

Fig. 10b



799
S_{END2}
790
754
800
752
750
760,762
801
752
S_{END1}
780
753
754
798
800
801
757
753
756

Fig. 11a

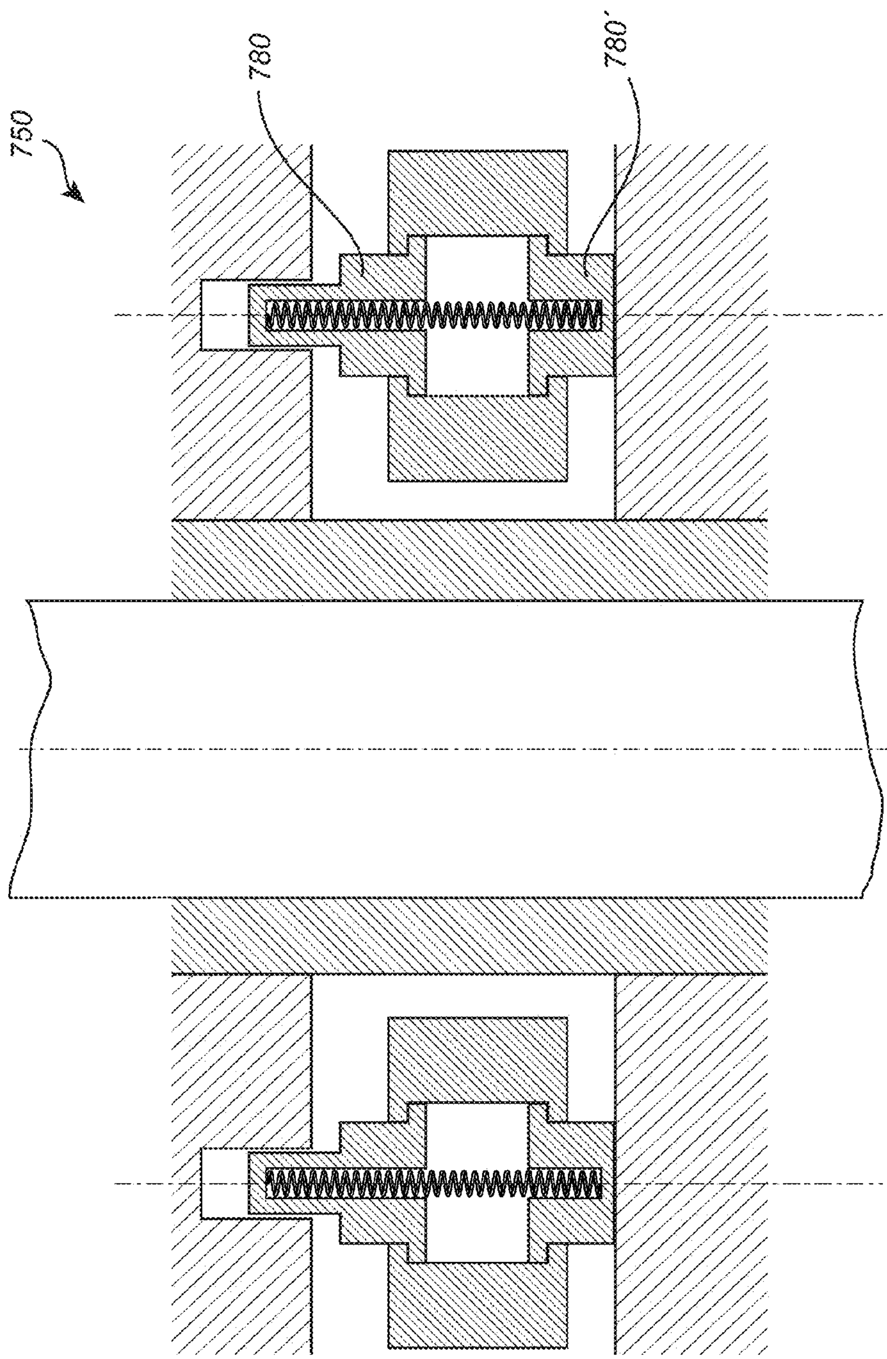


Fig. 11b

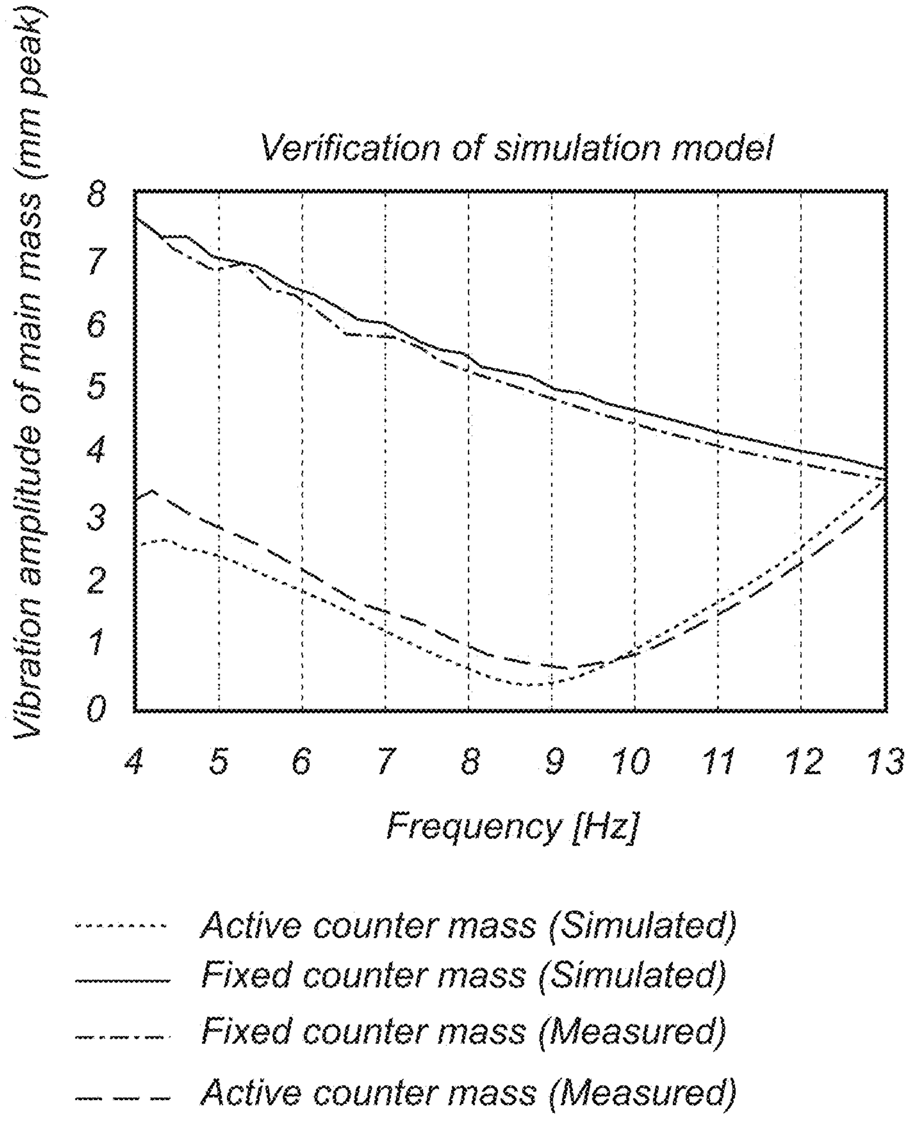

Verification of simulation model
Vibration amplitude of main mass (mm peak)
8
7
6
5
4
3
2
1
0
4
5
6
7
8
9
10
11
12
13
Frequency [Hz]
Active counter mass (Simulated)
Fixed counter mass (Simulated)
Fixed counter mass (Measured)
Active counter mass (Measured)

Fig. 12

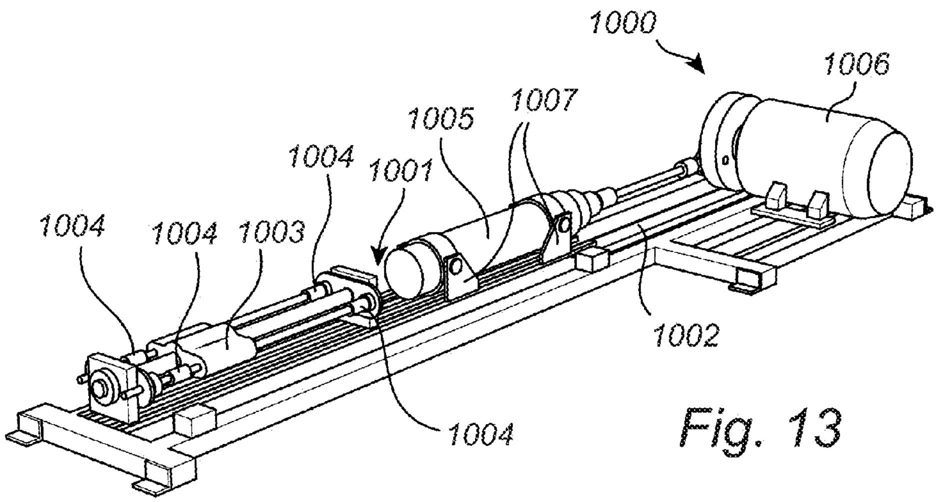

1000
1006
1007
1005
1004
1001
1004
1004
1003
1002
1004

Fig. 13

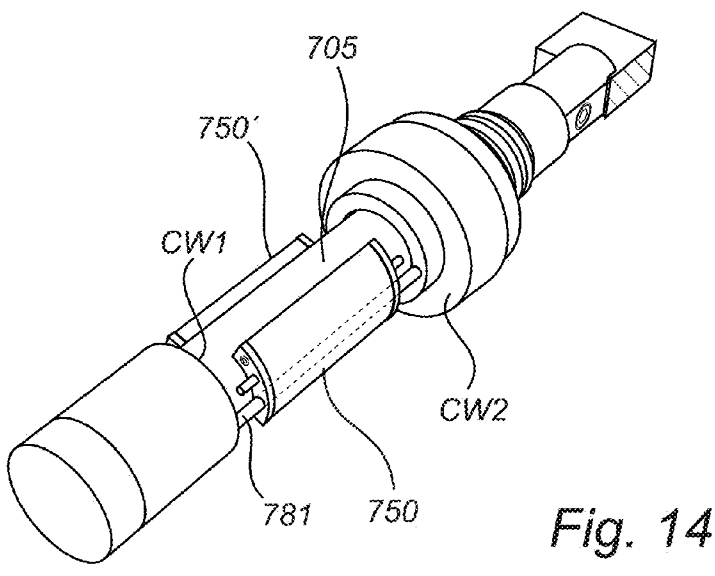

705
750′
CW1
CW2
781
750

Fig. 14

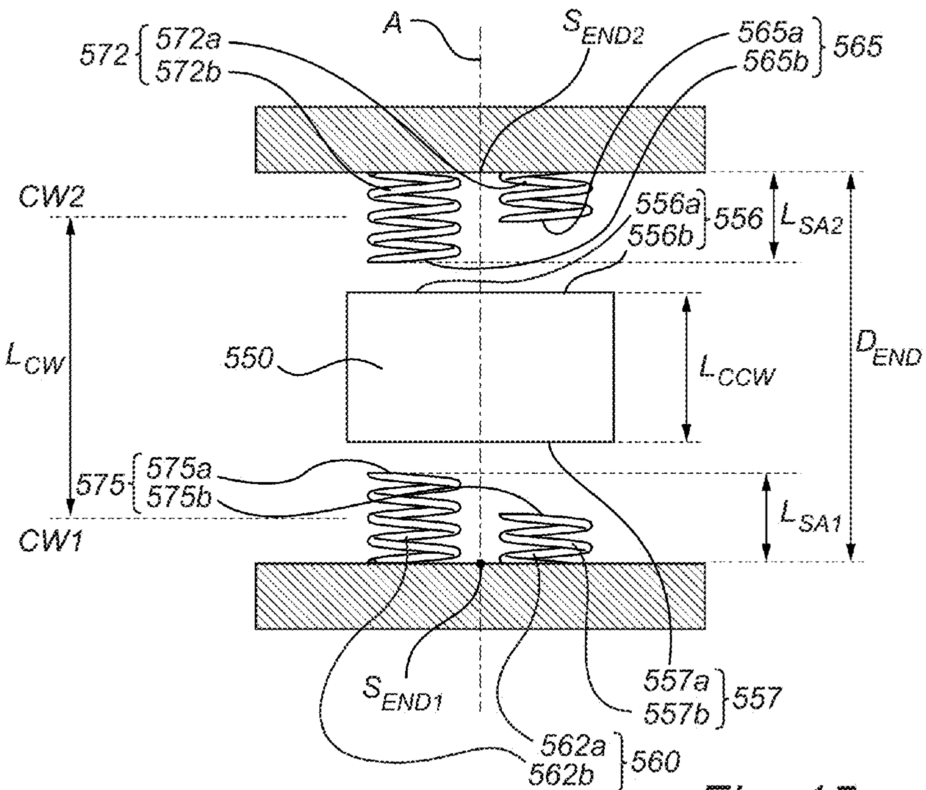


Fig. 15a

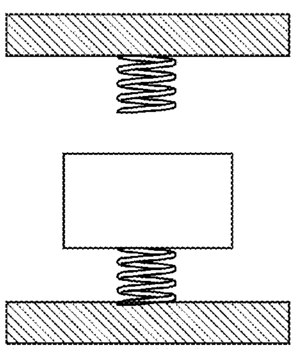


Fig. 15b

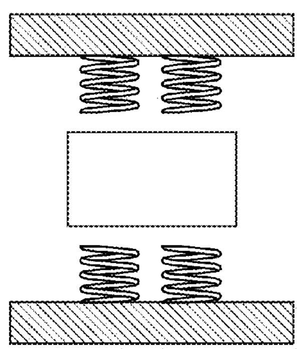


Fig. 15c

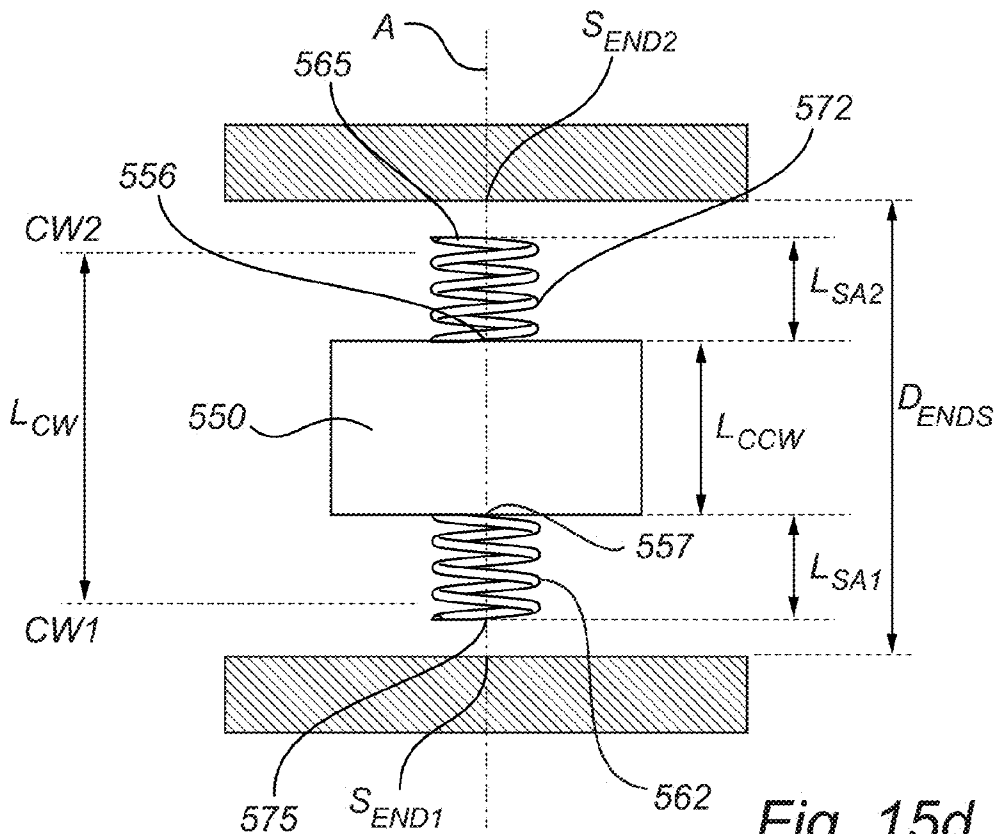
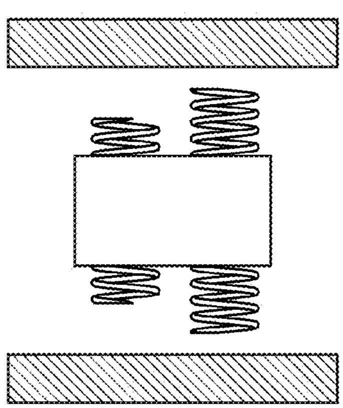


Fig. 15d



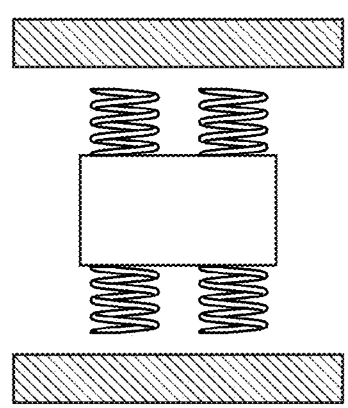


Fig. 15e

Fig. 15f

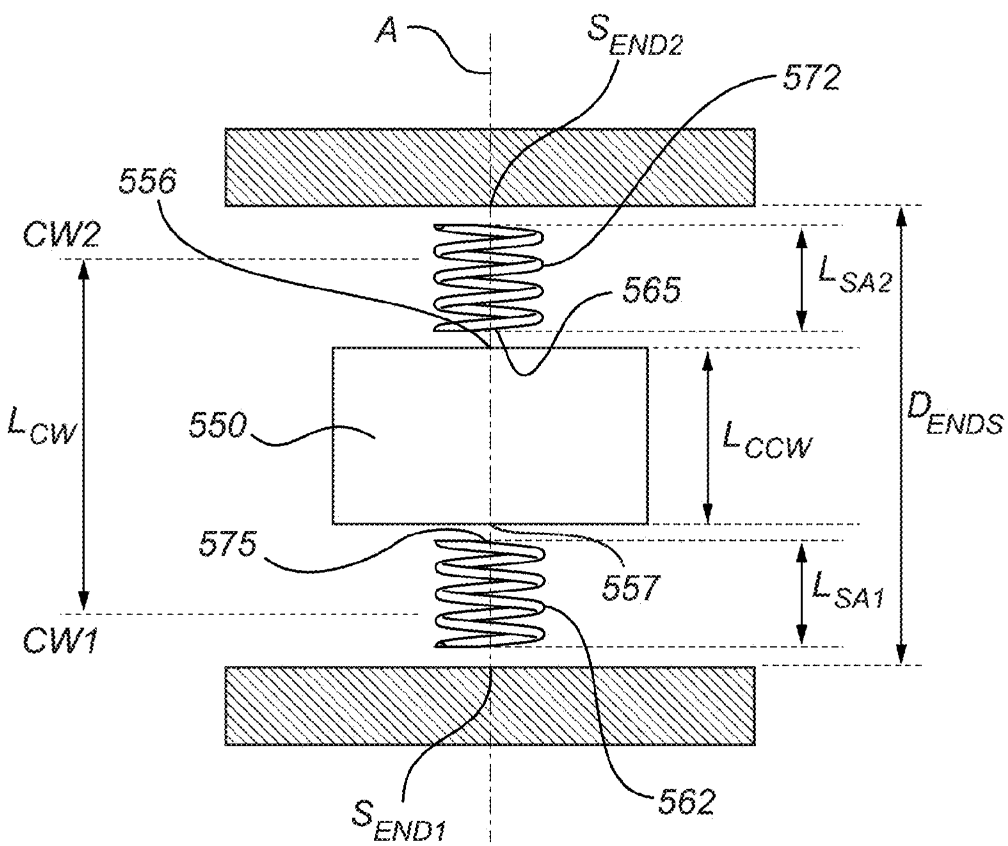

A
S_{END2}
572
556
CW2
L_{SA2}
565
L_{CW}
550
L_{CCW}
D_{ENDS}
575
557
L_{SA1}
CW1
S_{END1}
562

Fig. 15g

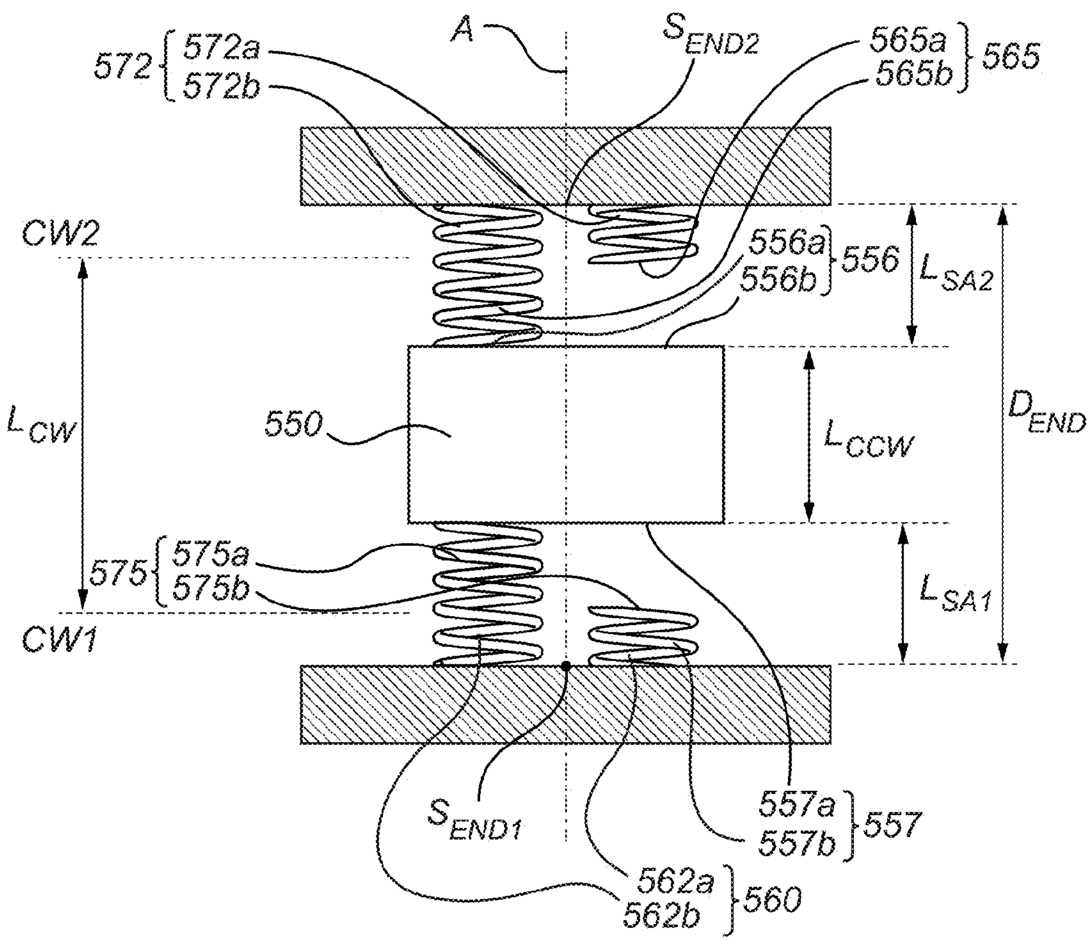

A
S_{END2}
572
572a
572b
565a
565b
565
CW2
556a
556b
556
L_{SA2}
L_{CW}
550
L_{CCW}
D_{END}
575
575a
575b
CW1
L_{SA1}
S_{END1}
557a
557b
557
562a
562b
560

Fig. 15h

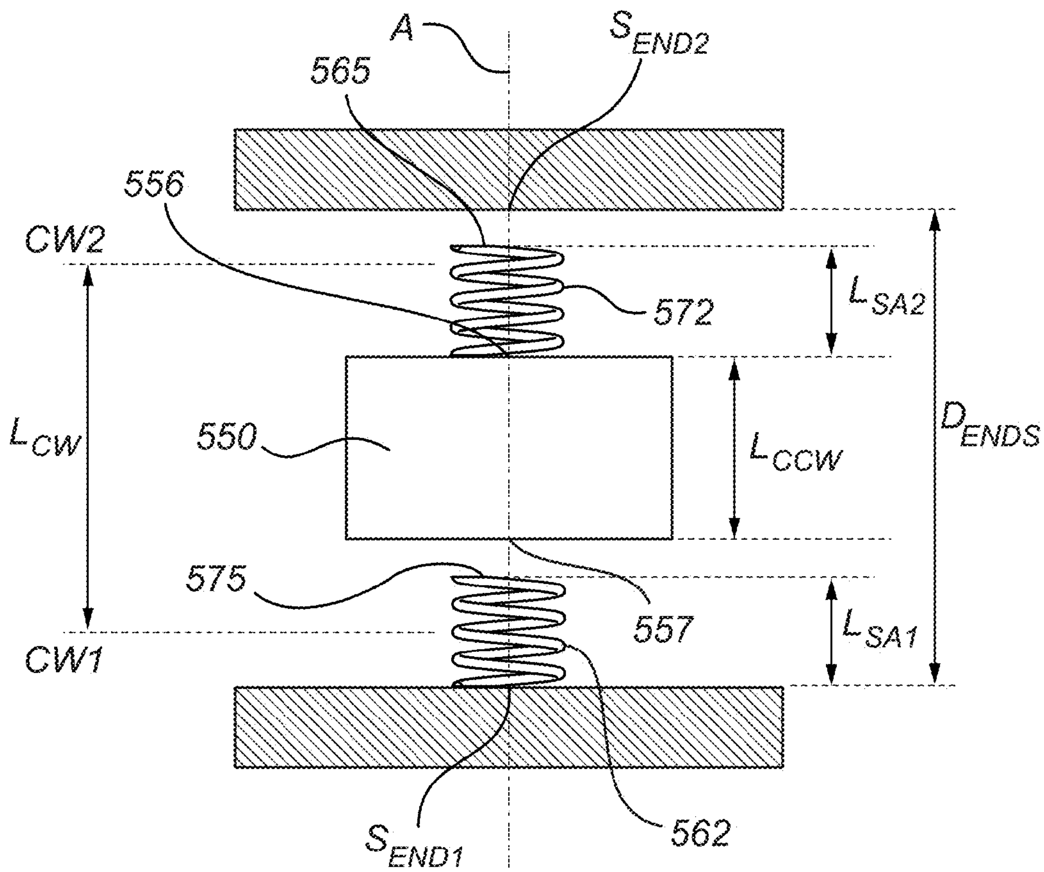

A
S_{END2}
565
556
CW2
572
L_{SA2}
L_{CW}
550
L_{CCW}
D_{ENDS}
575
557
L_{SA1}
CW1
562
S_{END1}

Fig. 15i

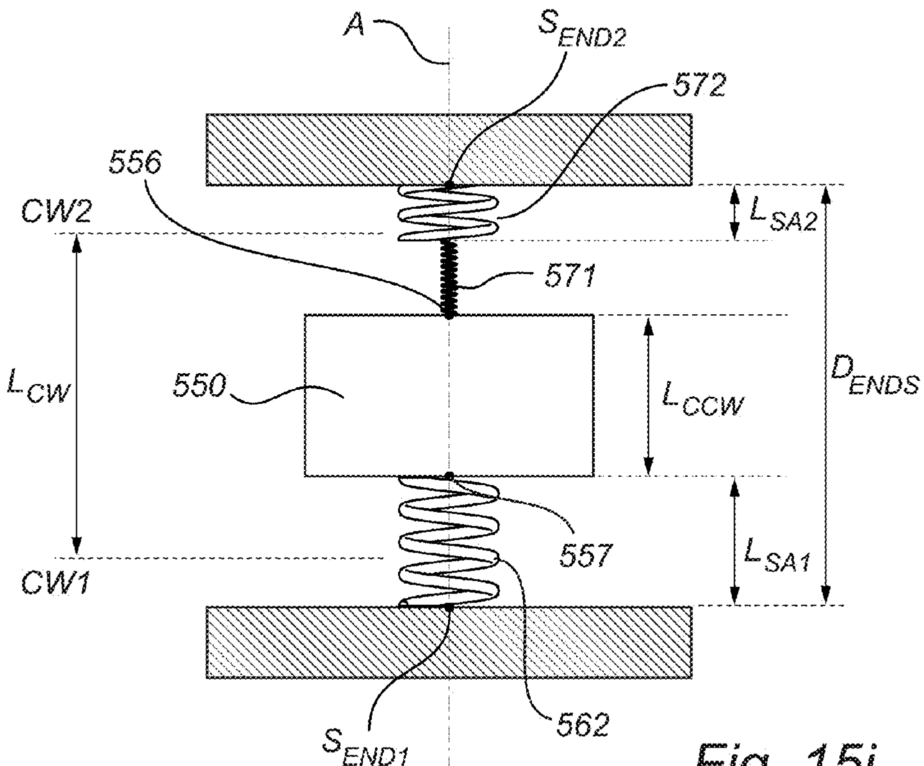


Fig. 15j

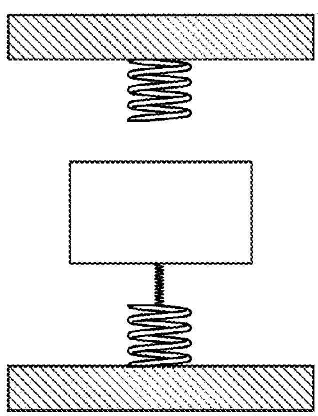


Fig. 15k

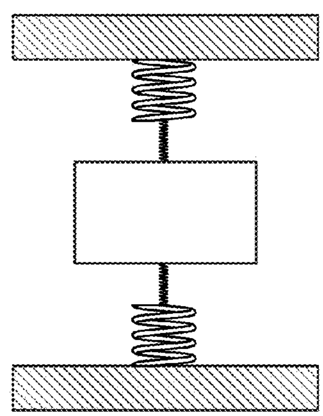


Fig. 15l

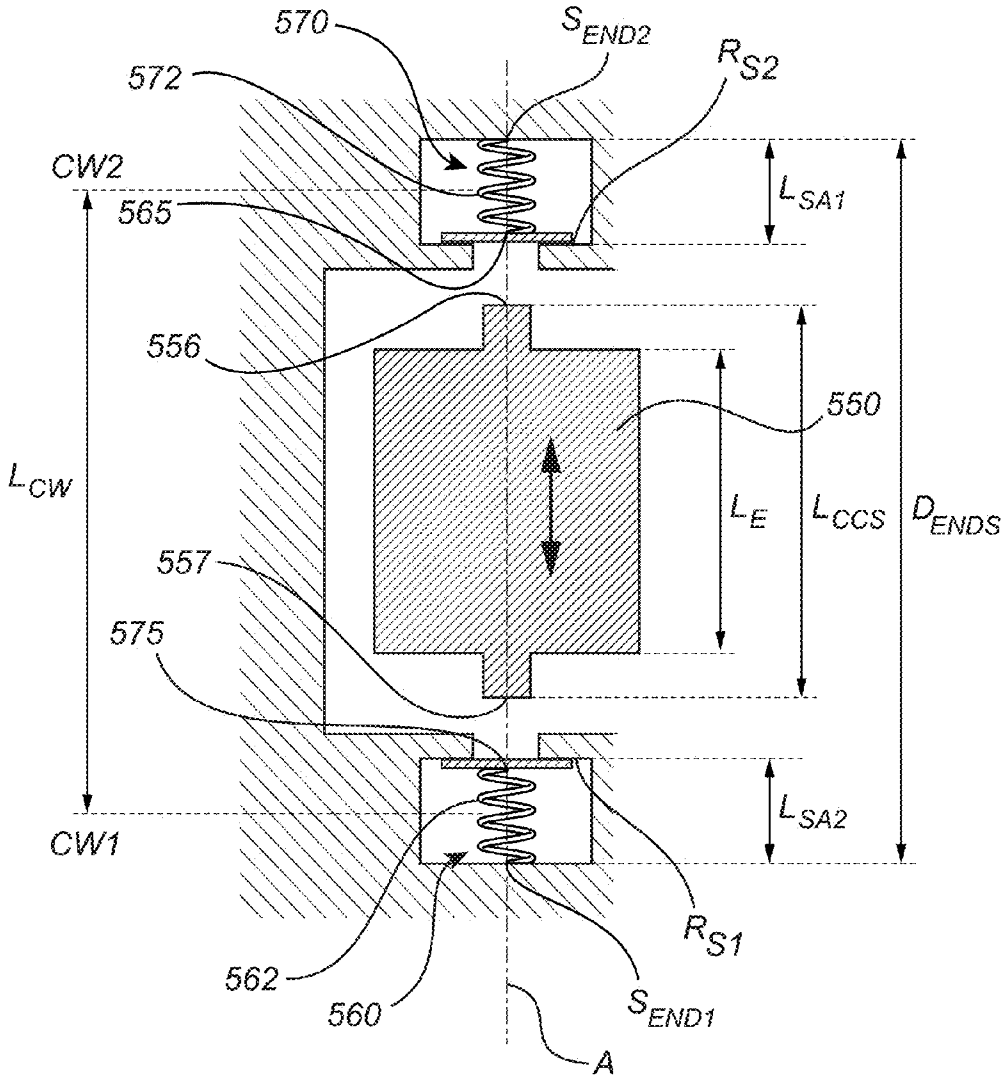

570
S_{END2}
572
R_{S2}
CW2
L_{SA1}
565
556
550
L_{CW}
L_E
L_{CCS}
D_{ENDS}
557
575
L_{SA2}
CW1
R_{S1}
562
560
S_{END1}
A

Fig. 16a

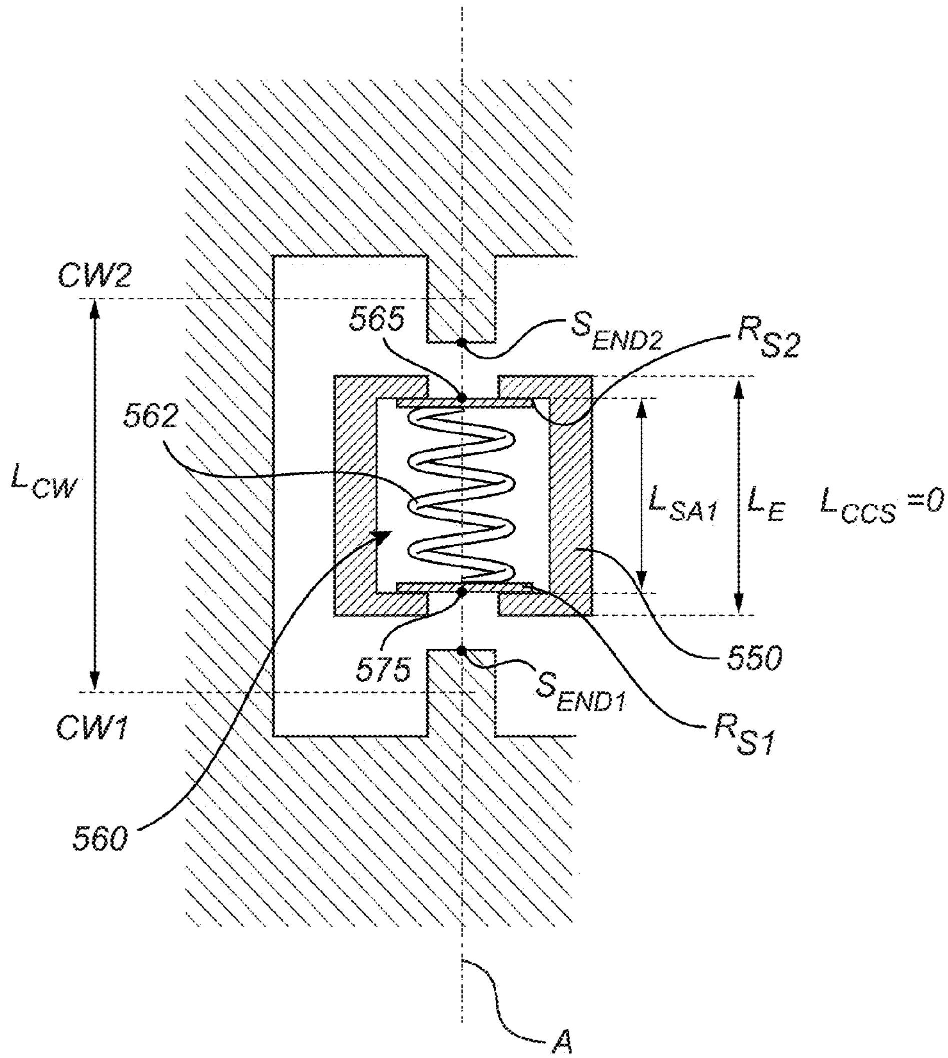

CW2
565
S_{END2}
R_{S2}
562
L_{CW}
L_{SA1}
L_E
$L_{CCS}=0$
550
575
S_{END1}
CW1
R_{S1}
560
A

Fig. 16b

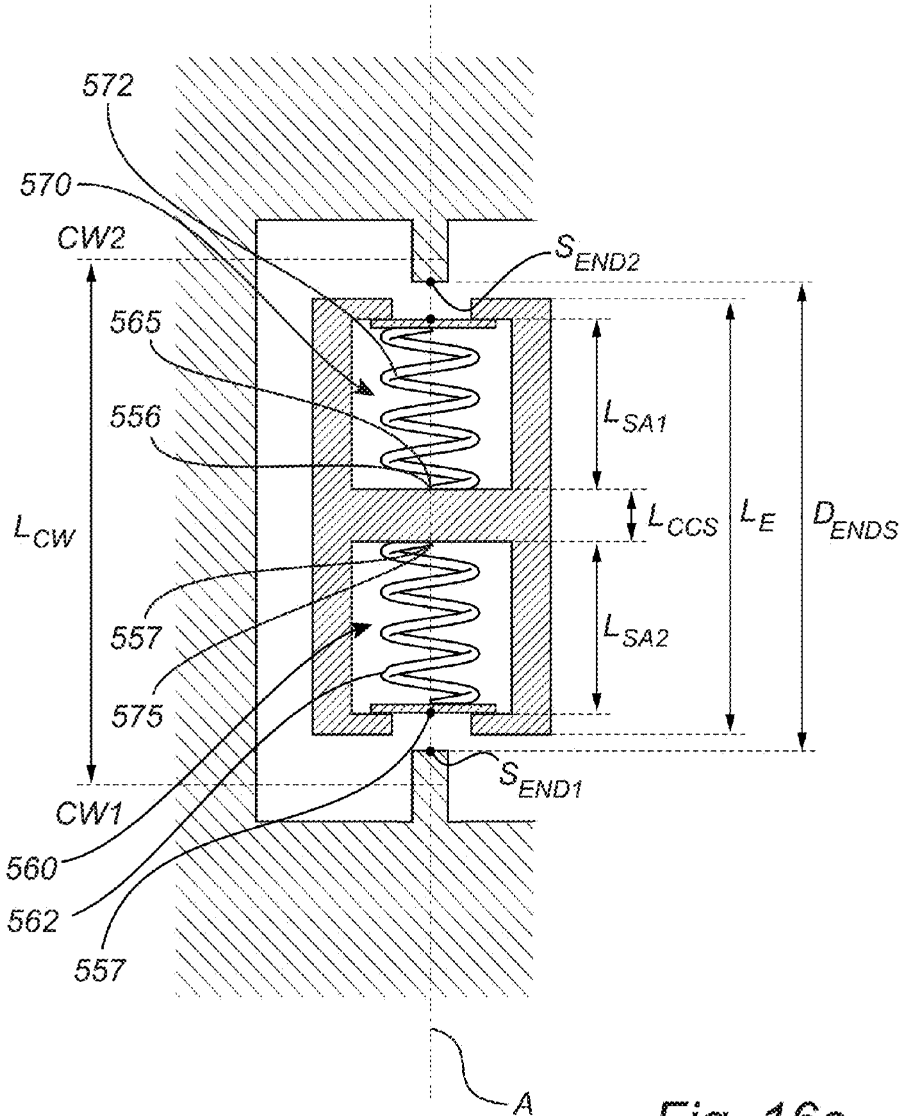

572
570
CW2
S_{END2}
565
556
L_{SA1}
L_{CW}
L_{CCS}
L_E
D_{ENDS}
557
L_{SA2}
575
S_{END1}
CW1
560
562
557
A

Fig. 16c

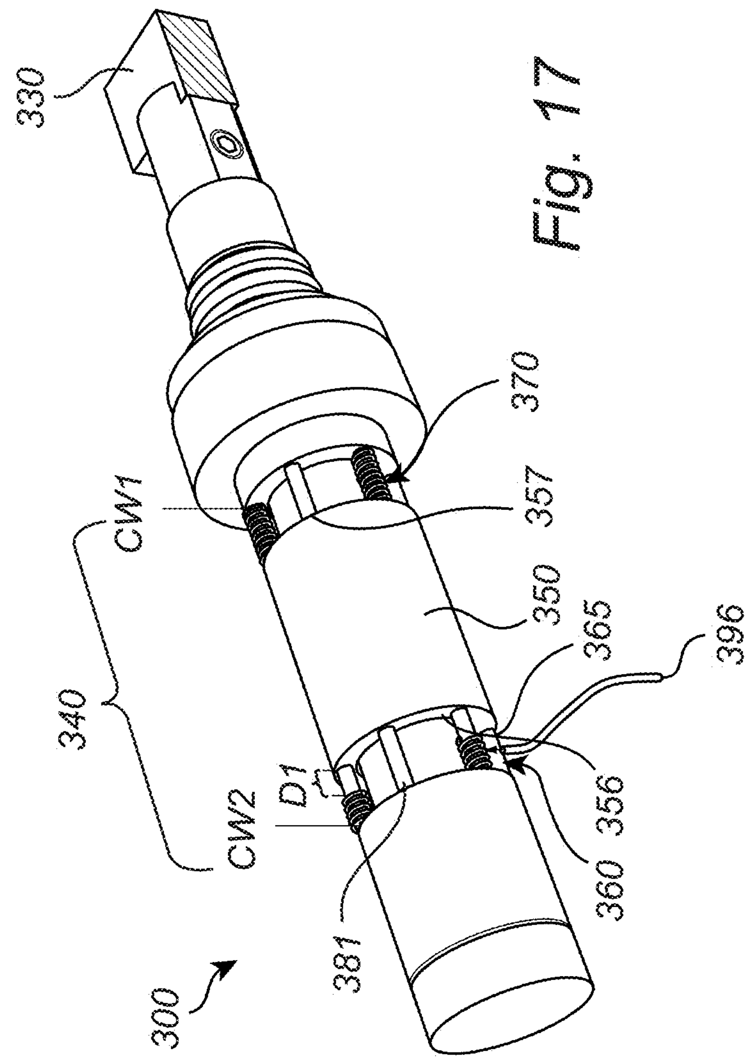


Fig. 17

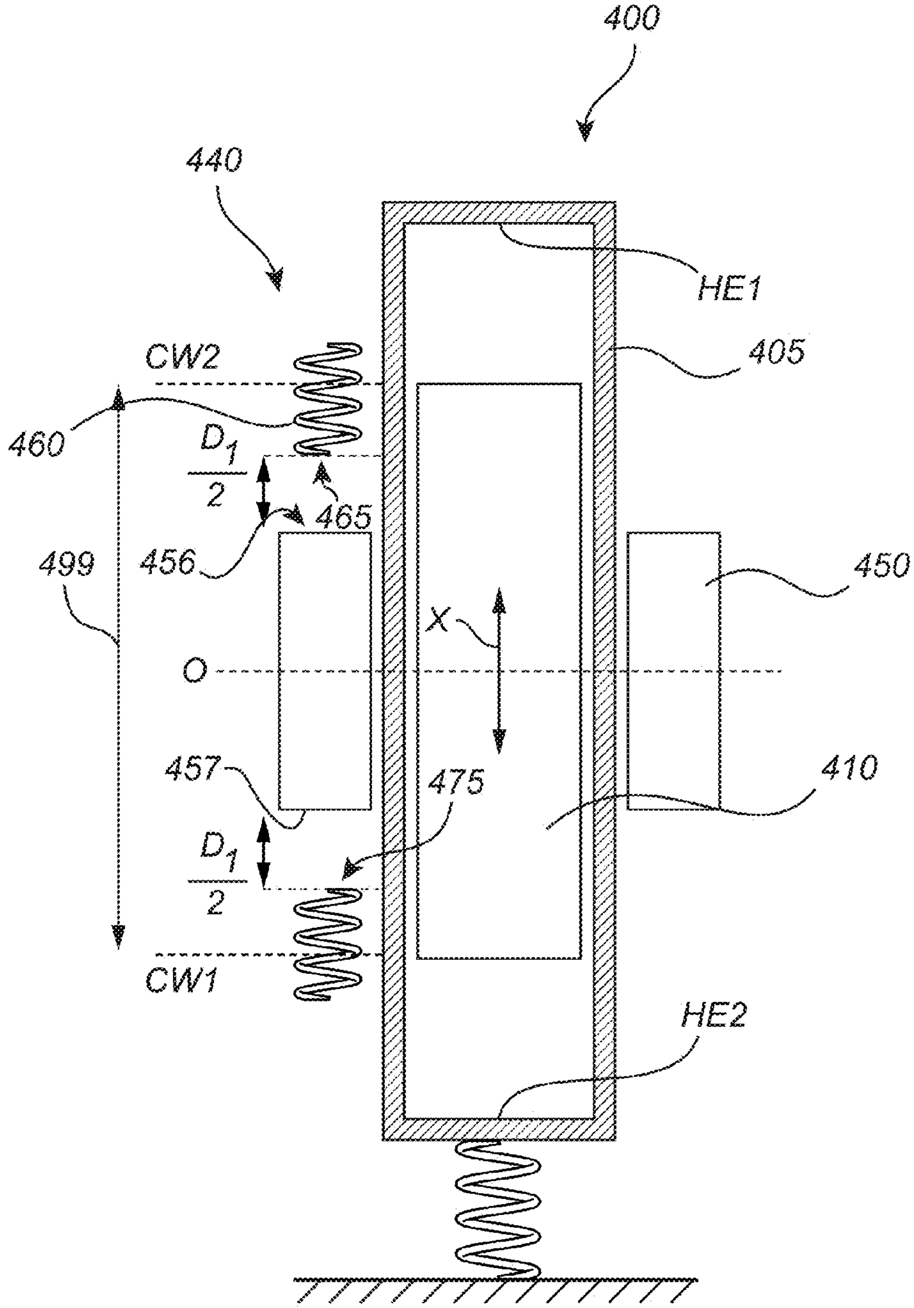

400
440
HE1
CW2
405
460
$\frac{D_1}{2}$
465
456
499
450
X
O
457
475
410
$\frac{D_1}{2}$
CW1
HE2

Fig. 18

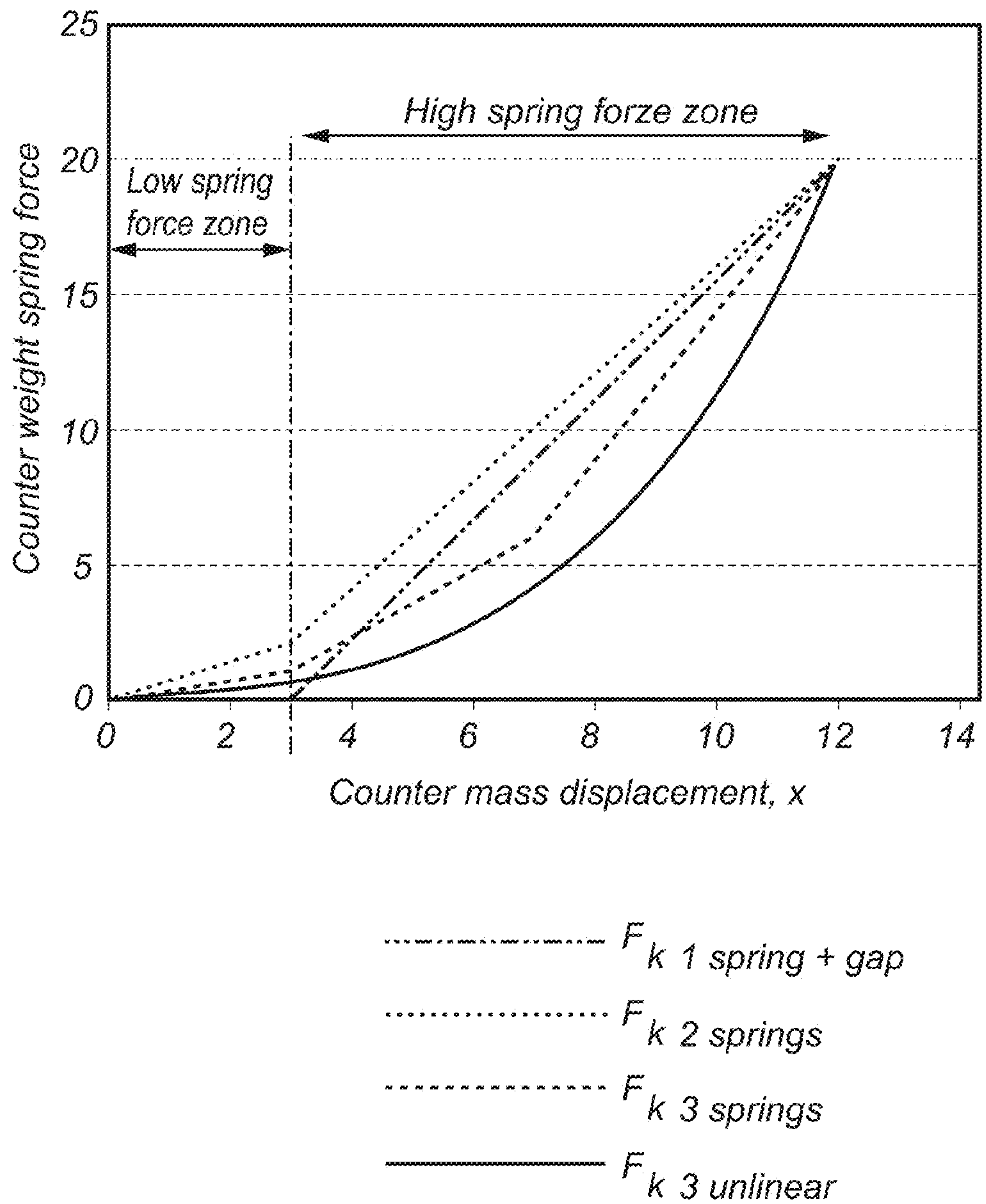

High spring forze zone
Low spring force zone
25
20
15
10
5
0
0
2
4
6
8
10
12
14
Counter weight spring force
Counter mass displacement, x
F_k 1 spring + gap
F_k 2 springs
F_k 3 springs
F_k 3 unlinear

Fig. 19

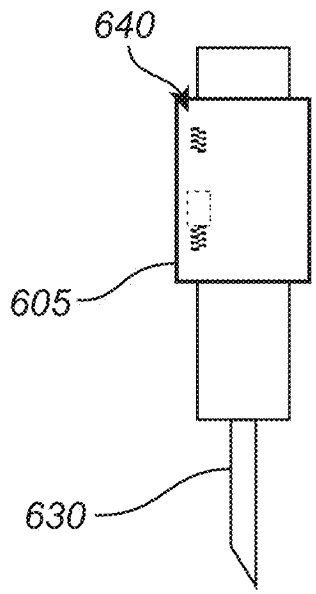


Fig. 20a

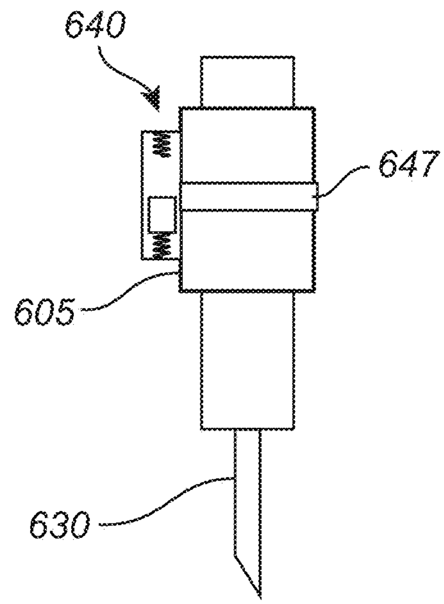


Fig. 20b

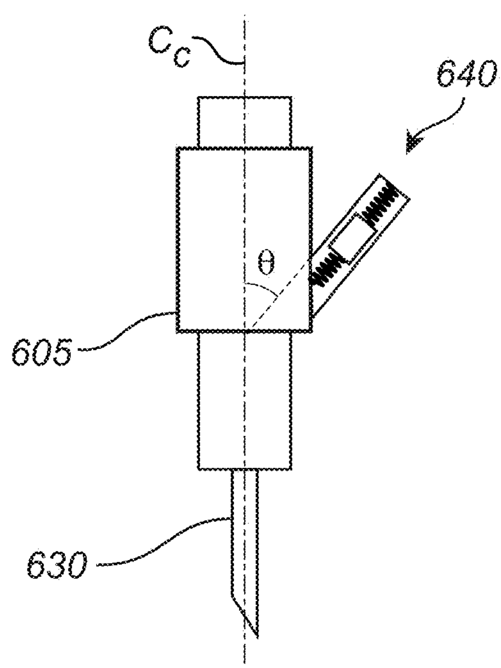


Fig. 20c

IMPACT MACHINE

FIELD OF THE INVENTION

[0001] The present invention relates to the field of impact machines, and more specifically to impact machines comprising a vibration reduction arrangement.

BACKGROUND AND SUMMARY OF THE INVENTION

[0002] Impact machines are frequently used in quarries, road construction and building construction applications in order to work hard surfaces, such as rock, concrete and pavements or softer materials such as asphalt. The invention can be used on machines such as rock drills, breakers, rammers and hammers which all have a similar impact mechanism.

[0003] A common design for impact machines comprises hydraulic, pneumatic, combustion engine or electrical actuating means and a moveable impact element transferring an impact force to a work tool attached to the housing. In operation, fatigue in a hard surface may be achieved by applying a continuous impact force from the work tool, such that the hard surface finally breaks. However, parts of the vibrations from the movements of the impact mechanism are transferred to the housing of the impact machine and to a connection point for either a manual handle or to a bracket for a machine attachment. The vibrations may thus result in body injuries if the tool is hand-held and in machine wear if the tool is attached to a machine.

[0004] An example of a vibration dampening mechanism for a hand-held machine tool is disclosed in the document US2012/0279741. The vibration damping mechanism comprises a movable counterweight arranged between two identical sets of compression springs. Each set comprises two springs of different lengths arranged in parallel; the longest spring is in constant contact with the counterweight, and the shortest spring is only in contact with the counterweight at high vibration amplitudes.

[0005] When the vibration amplitude of the counterweight is increased, the springs that are connected in parallel are both activated, such that the spring constants are added and the spring constant in the vibratory damping mechanism is increased. This shifts the natural frequency of the vibration damping means towards higher frequencies.

[0006] The purpose of the short springs arranged at a distance from the counterweight is to prevent striking of the mechanical limit position. Under normal operating conditions, the short springs will not be in contact with the counterweight.

[0007] The document DE102009046348 discloses another example of a vibration damping mechanism for a hand-held machine tool that comprises a counter acting weight. In order to provide a cost efficient method of producing the vibration damping mechanism, the geometry of a coil spring can be varied such that a central region of the spring has a certain weight that is sufficient to create a counter-acting weight. The central region is provided with a weight by arranging it like a tension spring with a large diameter and with compact windings. Alternatively, an external or internal counterweight can be attached to the central region to achieve a heavier counterweight. The central region is received between two elastic compression springs that are provided with stiff end-portions connected to a housing. However, the arrangement in DE102009046348 does not provide any means for extending the dampened frequencies range of the machine tool.

[0008] In view of the above, it is an object of the present invention to provide an improved vibration damping arrangement for an impact machine that is providing a vibration damping effect throughout a wide working frequency range of the machine and that also reduces the risk of sudden disruptions in the vibration damping effect. A further object is to provide a stronger feed-force of the impact machine, such that the weight of the hammering can be reduced. The inventor was recently given the assignment of trying to find a way to reduce the vibrations of the handles in hand held impact machine. As an example of a machine on the market he was given an impact machine, Atlas-Copco, KV434, having an impact energy of about 25 Joules, and a total weight of 12 kg whereof the impact element had a weight of 0.53 kg. The idea behind the vibration reduction arrangement of this machine, resided in making the machine sufficiently heavy such that the amplitude of the vibrations would be reduced due to the weight of the machine. However, still with this heavy weight, the machine generated very high vibrations in the range around region of 20 m/s^2, hand arm weighted acceleration.

[0009] In the beginning of the project, the inventor made an analysis of possible ways to lowering the vibrations in the handles, and came up with the following ideas:

[0010] (a) providing vibration isolation means (such as springs or shock absorbers) e.g. between the impact mechanism and the handles,

[0011] (b) using a actively controlled counterweight, which applies the substantially same force to the housing as the hammering element but in the opposite direction, or

[0012] (c) using a tuned vibration absorber.

[0013] Knowing that tuned vibration absorbers, such as the ones described by J. P. Den Hartog in "*Mechanical Vibratons*", 1985, or "*Shock and Vibration Handbook*", 1987, where a counterweight is coupled to an active weight by means of a spring, would only work within a very limited frequency interval; the inventor quickly gave up that idea. These types of vibration dampers are also referred to as "vibration reduction methods for narrow frequency ranges" in the following text, and are normally applied to devices having a very limited variation in working frequency, such as air plane engines or marine engines.

[0014] The reason for the inventor to abandon the idea of using vibration reduction methods for narrow frequency rages, was that the hand-held impact machines of the type the inventor was working with have a working frequency which varies through a wide frequency range, they normally presented a variation of 20-30% in working frequency and sometimes as high as 50%. The working frequency may vary due to e.g. instability in the control of the element e.g. due to instability in the compressed air supply to the machine, due to the operator holding the machine in varying angles in relation to the surface which is to be cut, due to a variation in the force by which the operator presses the machine against the surface, due to a varying surface hardness at different locations in the element which is to be cut, etc (the list is non-exhaustive).

[0015] However, as a comparative example the inventor wanted to give an illustration of how badly the "vibration reduction methods for narrow frequency rages" worked for impact machines. But when he was about to connect a counterweight to the housing of the impact machine by means of one spring at each end of the counterweight he realized that he did not have sufficiently strong springs at home to get the right working frequency. Not letting this stop him, he simply used the strongest once he had at hand.

[0016] As predicted, the dampening arrangement did not work at all when he turned the machine on. The machine just bounced around, being extremely difficult to control. Try to picture his surprise when, after a while, the bouncing suddenly stopped and there were hardly any vibrations in the handles at all. The inventor turned the machine off and then on again, and still the vibrations were almost eliminated. There was no reasonable explanation to this.

[0017] When he later disassembled the machine, he could see nothing unordinary except that the force of the counterweight had compressed the springs, such that there were gaps between the counterweight and the springs. Hence, he could see no other explanation, besides that it was the gap between the spring and the counterweight that had balanced the system. Encouraged by this, he started to simulate the system to see if his conclusions could be correct—and it turned out that they were.

[0018] After the inventor had analyzed his findings, he has come to the conclusion that the surprising vibration reducing effect may not only be achieved by arranging a gap between the counterweight and the springs, but also by means of alternative measures as will be described below.

[0019] The present invention is described in the independent claim, and advantageous embodiments of the present invention are defined in the dependent claims.

[0020] According to a first aspect of the present invention, there is provided an impact machine comprising:

[0021] a housing

[0022] a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position and a second hammering element position,

[0023] an impact receiving element attached to said housing,

[0024] actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

[0025] a vibration reduction arrangement attached to said housing, which comprises:

[0026] at least one counterweight distributed around said hammering element and being displaceable in a first axial direction between a respective first counterweight position and a respective second counterweight position in response to the hammering action of said hammering element, —a respective first motion reversing mechanism for each one of said at least one counterweight, each respective first motion reversion mechanism comprising a first spring-action arrangement being arranged to reverse the direction of motion of a respective one of said at least one counterweight, wherein

[0027] each one of said at least one counterweight is arrangeable at a position located between said respective first counterweight position and said respective second counterweight position from which position each one of said at least one counterweight is moveable a first distance extending in said first axial direction without actuating said first spring-action arrangement; and wherein

[0028] the spring action arrangement of said respective first motion reversing mechanism is arranged inside said respective one of said at least one counterweight,

[0029] each of said respective first motion reversing mechanism further comprises a first end surface attached to said housing and arranged adjacent to said respective first counterweight position and

[0030] each one of said at least one counterweight comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective spring action arrangement and arranged between said respective spring action arrangement and said first end surface in said first axial direction, wherein when any of said at least one counter weight is arranged in said respective first counterweight position:

[0031] said engagement surface and said first end surface are pressed against each other,

[0032] said at least one spring-action arrangement is actuated.

[0033] In essence the inventors have realized that the invention relates to an impact machine which is adapted to perform a hammering operation on a surface or an object to be worked upon.

In particular, a vibration reduction arrangement is attached to the housing and comprises a moveable counterweight, interacting with a motion reversing arrangement having a non-linear spring characteristics, such that the motion of the counterweight can be brought into a counter-acting movement in relation to the vibrations in the housing of the hammering element thus substantially reducing the vibrations.

[0034] Generally for this invention it is desirable to minimize the damping, so that the velocity of counterweight in the opposite direction is as high as possible. If possible even a negative damping is desirable, and this can be achieved e.g. via compressed air. For example the damping at rated power is between −25% and +25% of critical damping. Alternatively the lower end may be −15%, −10%, −5%, −1% or −0.1% or 0% of critical damping. Additionally or alternatively the higher end may be 15%, 10%, 5%, 1% or 0.1% or 0% of critical damping.

[0035] In more detail an improved vibration reducing arrangement may be achieved by providing a motion reversing arrangement having a low force zone and a high force zone, wherein the low force zone is activated before the high force zone when the motion of the counterweight is decelerated and reversed. The spring coefficient of the low force zone may correspond to a first spring coefficient (k_1) which e.g. is $-k_{trad} \le k_1 \le k_{trad}/2$, and the spring coefficient of said high force zone e.g. is $k_{trad}/5 \le k_2 \le 30 * k_{tra}$ when at least one member of said motion reversing arrangement is prestressed or biased. Alternatively, the spring coefficient of said high force zone is preferably $2 * k_{trad} \le k_2 \le 30 * k_{trad}$ when said motion reversing arrangement is not prestressed or biased. k_{trad} is determined from the following formula

$$F_{res} = \frac{1}{2\pi} \sqrt{\frac{k_{trad}}{m}}$$

F_{res} being the resonance frequency of the impact machine at rated power, and m the weight of the counterweight,

[0036] Furthermore, two motion reversing arrangement may be provided, one on each side of the counterweight, wherein the sum of the lengths the respective low force zones (length of low force zone 1+of low force zone 2) preferably is at least 30% or at least 40% of the distance between said first counterweight position and said second counterweight position. When $k_1=0$, said low force zone is a gap, when $k_1 \neq 0$ said low force zone may comprise one or more spring action members having a resulting first spring coefficient k_1. Moreover, the motion reversing mechanism may be attached to said counterweight and/or an end surface, or being loose (i.e. not attached to anything) as long as the described low and high force zones are provided. As stated above the at least one counterweight is distributed around said hammering element. This may mean that the counterweight is only one counterweight which fully surrounds said hammering element or alternatively if there are several counterweights, they may be evenly distributed around said hammering element.

[0037] The term "without actuating" in: "said at least one counterweight being arrangeable at a position located between said first counterweight position and said second counterweight position from which position said at least one counterweight is moveable a first distance extending in said first axial direction without actuating said at least one spring-action arrangement" means that the counterweight can move said first distance without the at least one spring-action arrangement is being compressed. I.e. an influence on the at least one spring-action arrangement by the counterweight moving said first distance does not makes the at least one spring-action arrangement actuated. The engaging surface can also be called connecting surface. More details may be found below.

[0038] The counterweight is normally not filled with oil or other liquids for damping purposes. Oil can however be used for lubrication purposes.

[0039] According to at least one exemplary embodiment said vibration reduction arrangement further comprises:

[0040] a respective second motion reversing mechanism for each one of said at least one counterweight, each respective second motion reversion mechanism comprising a second spring-action arrangement being arranged to reverse the direction of motion of a respective one of said at least one counterweight, and

[0041] the spring action arrangement of said respective second motion reversing mechanism is arranged inside said respective one of said at least one counterweight,

[0042] each of said respective second motion reversing mechanism further comprises a second end surface attached to said housing and arranged adjacent to said respective second counterweight position and

[0043] each one of said at least one counterweight comprises a second projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective second spring action arrangement and arranged between said respective spring action arrangement and said second end surface in said first axial direction (A) wherein

when any of said at least one counter weight is arranged in said respective second counterweight position:

[0044] said engagement surface of said second projecting member and said second end surface are pressed against each other,

[0045] said engagement surface of said second projecting member is displaced relative a center of gravity of said counterweight compared to when said counterweight is arranged in a position where said engagement surface of said second projecting member and said second end surface are separated from each other, and

[0046] said second spring-action arrangement is actuated.

[0047] According to at least one exemplary embodiment said spring action arrangement of said first motion reversing mechanism is separated from said spring action arrangement of said second motion reversing mechanism.

[0048] According to at least one exemplary embodiment said spring action arrangement of said first motion reversing mechanism and said spring action arrangement of said second motion reversing mechanism is one and the same.

[0049] According to at least one exemplary embodiment said spring action arrangement of said first motion reversing mechanism comprises a first spring action member.

[0050] According to at least one exemplary embodiment said spring action arrangement of said second motion reversing mechanism comprises a second spring action member.

[0051] According to at least one exemplary embodiment said spring action member of said first motion reversing mechanism is separated from said spring action member of said second motion reversing mechanism.

[0052] According to at least one exemplary embodiment said spring action member of said first motion reversing mechanism and said spring action member of said second motion reversing mechanism is one and the same. According to at least one exemplary embodiment said first spring action member is prestressed, and has a first spring characteristics (k_1) within the interval $k_{trad}/5 \leq k_1 \leq 30*k_{trad}$. Alternatively said first spring action member is not prestressed, and has a first spring characteristics (k_1) within the interval $2*k_{trad} \leq k_1 \leq 30*k_{trad}$.

[0053] According to at least one exemplary embodiment said second spring action member is prestressed, and has a first spring characteristics (k_1) selected such that the resulting spring characteristics of the two spring actions members is within the interval $k_{trad}/5 \leq k_1 \leq 30*k_{trad}$, when one or both are prestressed. Alternatively, neither of the spring actions members are prestressed and the resulting spring characteristics of the two spring actions members is within the interval $2*k_{trad} \leq k_1 \leq 30*k_{trad}$.

[0054] According to at least one exemplary embodiment said motion reversion mechanism comprises four spring-action arrangements distributed around said hammering element.

[0055] According to at least one exemplary embodiment said counterweight comprises two spring-action arrangements inside said counterweight.

[0056] According to at least one exemplary embodiment said two spring-action arrangements are identical.

[0057] According to at least one exemplary embodiment said counterweight further comprising restricting means adapted to restrict the movement of said projecting member in the first axial direction and/or in a direction opposite thereto.

[0058] According to at least one exemplary embodiment said restricting means comprises at least one first retaining surface attached to said counterweight, and said projecting member further comprises at least one flange, wherein said retaining surface restricts the motion of said flange in the first axial direction and/or in a direction opposite thereto.

[0059] According to at least one exemplary embodiment said restricting means further comprises a second retaining surface adapted to restrict the movement of said second projecting member in said second axial direction and/or in a direction opposite thereto, and said spring action member is biased by said first retaining surface and said second retaining surface.

[0060] According to at least one exemplary embodiment said vibration reduction arrangement is arranged around said housing, such that said at least one counterweight is rotatable about a central longitudinal axis of said housing, coaxial with said first axial direction.

[0061] According to at least one exemplary embodiment said impact machine further comprises counterweight guiding means arranged to cause said counterweight to move in a linear direction between said first counterweight position and said second counterweight position.

[0062] According to at least one exemplary embodiment when said at least one counterweight is only one counterweight, said counterweight fully surrounds said hammering element.

[0063] According to at least one exemplary embodiment when said at least one counterweight comprises of two or more counterweights, said counterweights are evenly distributed around said hammering element.

[0064] According to at least one exemplary embodiment said counterweight comprises an outer truncated elliptical cross-section which is perpendicular to said first axial direction.

[0065] According to at least one exemplary embodiment said at least one spring-action arrangement further comprises a first spring-action member and a second spring-action member arranged in parallel in said first axial direction. What is stated herein about the spring coefficient in a system having non-parallel spring members, may also be applied to the resulting spring coefficient of a system having two or more parallel spring action members.

[0066] According to at least one exemplary embodiment the first distance (D_1) is at least 20%, or at least 40%, or at least 60% or at least 70% or at least 80% of the distance between the first and the second counterweight positions. According to at least one exemplary embodiment a first spring action member and a second spring action member arranged in parallel, wherein said first spring coefficient of said first spring-action member is lower than said second spring coefficient of said second spring-action member, and wherein said first spring coefficient applies to a distance corresponding to at least 10% or at least 15% or at least 20% or at least 25% of a distance between said first and said second counterweight position; and said second spring coefficient applies to a remaining distance between said first and said second counterweight position.

[0067] According to at least one exemplary embodiment said impact machine further comprises hammer element guiding means arranged to cause said hammering element to move in a linear direction between said first hammering element position and said second hammering element position.

[0068] According to at least one exemplary embodiment said impact receiving element is a work tool.

[0069] According to at least one exemplary embodiment said impact machine is handheld.

[0070] According to at least one exemplary embodiment said impact machine is arranged to be attached to a machine, preferably a construction machine such as an excavator, backhoe loader or skid steer loader.

[0071] According to at least one exemplary embodiment the weight of the hammering element H corresponds to between 20% and 300% of the weight m of the counterweight.

[0072] According to at least one exemplary embodiment an impact machine comprises:

[0073] a housing,

[0074] a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position and a second hammering element position,

[0075] an impact receiving element attached to said housing,

[0076] actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

[0077] a vibration reduction arrangement attached to said housing, which comprises:

> **[0078]** a counterweight distributed around said hammering element and being displaceable in a first axial direction between a first counterweight position and a second counterweight position in response to the hammering action of said hammering element, —a first motion reversing mechanism comprising a first spring-action arrangement being arranged to reverse the direction of motion of said counterweight,

wherein

[0079] said counterweight is arrangeable at a position located between said first counterweight position and said second counterweight position from which position said counterweight is moveable a first distance extending in said first axial direction (A) without actuating said spring-action; and wherein

[0080] the spring action arrangement of said motion reversing mechanism is arranged inside said counterweight,

[0081] said first motion reversing mechanism further comprises a first end surface attached to said housing and arranged adjacent to said first counterweight position and

[0082] said counterweight comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said spring action and arranged between said spring action arrangement and said first end surface in said first axial direction wherein when said counter weight is arranged in said first counterweight position:

[0083] said engagement surface and said first end surface are pressed against each other, and

[0084] said at least one spring-action arrangement is actuated.

[0085] According to at least one exemplary embodiment an impact machine comprises:

[0086] a housing

[0087] a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position and a second hammering element position,

[0088] an impact receiving element attached to said housing,

[0089] actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

[0090] a vibration reduction arrangement attached to said housing, which comprises:

[0091] a first number of counterweights arranged evenly distributed around said hammering element, each counterweight being displaceable in a first axial direction between a respective first counterweight position and a respective second counterweight position in response to the hammering action of said hammering element, —a first number of motion reversing mechanisms, each comprising a first spring-action arrangement being arranged to reverse the direction of motion of a respective one of said first number of counterweights,

wherein

[0092] said each one of said first number of counterweights is arrangeable at a position located between said respective first counterweight position and said respective second counterweight position from which position each one of said at least one counterweight is moveable a first distance extending in said first axial direction without actuating said at least one spring-action arrangement; and wherein

[0093] said first spring action arrangement of each first motion reversing mechanism is arranged inside said respective one of said first number of counterweights,

[0094] each of said respective first motion reversing mechanism further comprises a respective first end surface attached to said housing and arranged adjacent to said respective first counterweight position and

[0095] each one of said at least first number of counterweights comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective spring action arrangement and arranged between said respective spring action arrangement and said first end surface in said first axial, wherein when any one of said first number of counter weights is arranged in said respective first counterweight position:

[0096] said engagement surface of said counterweight and said respective first end surface are pressed against each other,

[0097] said engagement surface is displaced relative a center of gravity of said counterweight compared to when said counterweight is arranged in a position where said engagement surface and said first end surface are separated from each other, and

[0098] said at least one spring-action arrangement is actuated.

[0099] The present invention provides the advantage of enabling a substantial decrease in the weight of an impact machine, lower vibration amplitude and an extended frequency range of low vibration amplitude. The force from the counterweight on the housing also creates a feed force that improves the efficiency of the machine.

[0100] The counterweight moves in a counter-phased movement in relation to the direction of a hammering element, where the travel distance of the counterweight is restricted to a maximum counterweight displacement distance between the first counterweight position and the second counterweight position measured when the machine is operated at rated power. A movement beyond these points is thus not possible, or normally not possible, when the machine is operated at rated power. Hence, depending on e.g. the impact forces, the travel distance of the counterweight may be equal to the maximum counterweight displacement distance, or shorter.

[0101] According to said first aspect of the invention the counterweight comprises a projecting member, i.e. a member which, when said counterweight is arranged in said first or second counter weight position, is displaced relative a center of gravity of said counterweight, compared to when said counterweight is arranged in a position where said engagement surface and said first end surface are separated from each other. In other words, when said spring action member is actuated, the projecting member is displaced relative a center of gravity of said counterweight, compared to when said spring action member is unactuated. In these cases, the motion of the center of gravity could preferably be considered when determining the maximum center of gravity displacement distance. I.e. the maximum center of gravity displacement distance equals the distance that the center of gravity of the counterweight travels or passes between the two end positions of the counter weight, when the machine is operated at rated power. In analogy, the center of gravity can also be considered when determining the first center of gravity position. In other words, said first center of gravity position corresponds to that position of the counter weight where the center of gravity of said counterweight is arranged furthest along a first counterweight displacement direction; and said second center of gravity position corresponds to that position where the center of gravity of said counterweight is arranged furthest along a direction opposite to said first counterweight displacement direction, wherein said counterweight displacement direction is equal to said first axial direction (A) or a direction opposite thereto.

[0102] The motion reversing mechanisms may be limited in their axial movement by end surfaces, which define fixed surfaces inside the vibration reduction arrangement. Furthermore, they may serve as abutment surface or attachment surfaces for the motion reversing mechanisms. The vibration reduction arrangement may comprise an end surface on each side of the counterweight.

[0103] In relation to this invention, the term first spring-action arrangement includes all spring-action members that take part in the reversing of the direction of motion of said counterweight, and which spring-action members are arranged, between a contact surface of the counterweight and a first end-surface.

[0104] Further, in relation to this invention, the term second spring-action arrangement includes all spring-action members that take part in the reversing of the direction of motion of the counterweight, and which are arranged between said counterweight and said second end surface.

[0105] In relation to this invention, the term discontinuous spring-action force implies a discontinuous change in the force acting upon the counterweight over the maximum counterweight displacement distance, between the first counterweight position and the second counterweight position. Alternatively, a discontinuous spring-action force may be provided by a combination of spring-action members, with different spring coefficients, e.g. first connected to each other in series and then in connection with the counterweight. The spring-action member applies a force on a contact surface of the counterweight. Alternatively, a discrete combination of a spring-action member arranged at a first distance from (i.e. not physically attached to) the counterweight may provide for the discontinuous spring-action force. In the latter example, there is no applied spring-action force on the counterweight throughout the first distance.

[0106] When e.g. one non-linear spring abut the counterweight on each end this may be referred to a non-linear spring-action. A non-linear spring may also be referred to as an unlinear spring.

[0107] In relation to the present invention, the length of the intermediate distance is determined when the impact machine, as well as the motion reversing mechanism(s), as well as the spring-action member(s) are at rest, i.e. the spring-action member(s) is/are neither compressed nor extended, i.e. the spring-action members are unactuated. In the case of a biased or prestressed spring-action member, the definition of unactuated implies that the spring-action member is only being subject to the inherent biasing force.

[0108] Normally, the intermediate distance is most easily determined when the vibration reduction arrangement is positioned with the counterweight's travel direction coinciding with the horizontal plane. Moreover, provided that the spring-action members acting upon the counterweight are at rest, the first distance which hereinafter is also referred to as the gap or the intermediate distance, is defined as a distance between the first and the second end-surface, through which distance the counterweight is freely moveable without actuating the spring-action members. In other words, the counterweight is freely moveable and the spring-action members are left unactuated. In yet other words, the counterweight is freely moveable while all the spring-action members are at rest or unactuated.

[0109] The distance of the gap D_1 can be calculated as the distance between the first and the second counterweight position subtracted by the exterior length (maximum axial length) of the counterweight **750** including the extention of the projecting members from the outer surfaces of the counterweight to the respective engaging surfaces.

[0110] The term first axial direction is the axial travel direction of the counterweight. The direction is parallel with the axial extensions of the spring action members.

[0111] In relation to this invention the term engaging surface, also called contacting surface, of a projecting member refers to the surface of the projecting member, which is in contact with a first or a second end surface when the spring-action member is being compressed.

[0112] The purpose of the spring-action member is to reverse the motion of the counterweight. Within the scope of the present invention, there are numerous ways of selecting and designing a spring-action member, whereby the invention should not be limited to any particular type. However, depending on the type of spring-action member, different parameters/coefficients may be used to specify the motion reversing capabilities of the spring-action member. In particular, the expression spring characteristics should be interpreted in a wide sense. Some embodiments of the spring-action member present a linear elasticity and may therefore be characterized in terms of a linear spring coefficient k according to Hook's law. Other embodiments of the spring-action member may include materials which present a spring characteristics corresponding to a non-linear spring coefficient. Still other embodiments of the spring-action member may include materials which present a spring characteristics corresponding to a combination of a spring coefficient k and a dampening coefficient c, such as rubber, solid or foamed polyurethane, etc. (The list in non-exhaustive)

[0113] Hence, spring-action members may include materials which present a non-linear elasticity, like for instance rubber, steel material, non-linear springs or air cushions. Further, a spring-action member hereby refers to any type of member, which is capable of providing a motion reversing action on the counterweight. In addition, the spring-action member may have various geometric shapes, like a coil spring, rubber ball or a surface, such as a plate.

[0114] The advantages include that the vibrations from the impact machine are dampened for a large working frequency range and the absence of a sudden vibration increase enables the vibration reduction arrangement to exclude a safety zone; as well as that many different elements may be applied, giving rise to a vast design freedom.

[0115] The force from the counterweight on the housing also creates a feed force that improves the efficiency of the machine.

[0116] Generally, all terms used in the claims are to be interpreted according to their ordinary meaning in the technical field, unless explicitly defined otherwise herein. All references to "a/an/the [element, device, component, means, step, etc]" are to be interpreted openly as referring to at least one instance of said element, device, component, means, step, etc., unless explicitly stated otherwise.

[0117] Other objectives, features and advantages of the present invention will appear from the following detailed disclosure, as well as from the drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0118] These and other aspects will now be described in more detail with reference to the appended drawings, in which exemplary embodiments of the present invention are shown, wherein:

[0119] FIGS. **1***a* and **1***b* are diagrams illustrating the correlation between working frequency and vibration amplitude for simulated vibration arrangements,

[0120] FIGS. **2***a* and **2***b* are schematic views of exemplary impact machine embodiments in perspective and in cross section respectively,

[0121] FIGS. **3***a* and **3***b* are schematic cross-sectional views of exemplary impact machine embodiments not covered by the claims,

[0122] FIG. **4** is a schematic perspective of a part of the impact machine according to the invention,

[0123] FIG. **5** is a cross-sectional schematic view of the part of the impact machine in FIG. **4**.

[0124] FIG. **6** is a cross-sectional schematic view of the counterweight in FIGS. **4** and **5**.

[0125] FIG. **7** is a schematic perspective of another exemplary embodiment of a counterweight according to the invention.

[0126] FIG. **8** is a cross-sectional schematic view of the counterweight in FIG. **7**.

[0127] FIG. **9** shows the impact machine according to the invention in use.

[0128] FIG. **10***a* shows a cross-sectional schematic view of the counterweight and its arrangement in FIG. **5**

[0129] FIG. **10***a*' shows the counterweight in FIG. **10***a* in a first counterweight position.

[0130] FIG. **10***b* shows a cross-sectional schematic view of another exemplary embodiment of the counterweight and its arrangement covered by the claims.

[0131] FIGS. **11***a*-**11***b* show cross-sectional schematic views of other exemplary embodiment of the counterweight and its arrangement according to the invention.

[0132] FIG. **12** shows a diagram showing verification of simulation model.

[0133] FIG. **13** shows a schematic perspective view of a test rig.

[0134] FIG. **14** is a schematic perspective view of an impact machine according to a further exemplary embodiment according to the invention.

[0135] FIGS. **15***a*-**15***l* are schematic perspective views of other exemplary embodiments not covered by the claims,

[0136] FIGS. **16***a*-**16***c* are schematic perspective views of exemplary embodiments including biased spring-action members, not covered by the claims

[0137] FIG. **17** is a schematic perspective, of an impact machine according to a further exemplary embodiment not covered by the claims,

[0138] FIG. **18** is a cross-sectional schematic view of a impact machine, in which geometry of the vibration reduction arrangement, not covered by the claims, is illustrated in detail,

[0139] FIG. **19** is a diagram describing examples of the spring-action characteristics as a function of the counterweight displacement,

[0140] FIGS. **20***a* to **20***c* are schematic perspective views of exemplary mounting positions for the vibration reduction arrangement not covered by the claims.

DETAILED DESCRIPTION

[0141] It should be noted that the illustrated embodiments by no means limits the scope of the present invention. In particular, the motion-reversing mechanism is described and illustrated as a coil spring. However, coil springs should only be seen as a representation of a possible spring-action member. In relation to this invention, spring-action members should include any member, which is capable of elastically reversing the moving counterweight in a vibration reduction arrangement. Like features are denoted with the same reference number.

[0142] FIG. **1***a* and FIG. **1***b* show diagrams with a representative image of the correlation between a working frequency f for an impact machine and the amplitude of transferred vibrations V_a to a handle or an attachment point to the impact machine. Notably, the diagrams in FIGS. **1***a* and **1***b* illustrate examples of variations in the vibration reduction effect for a machine with several different vibration reduction arrangements. The graphs were generated in the multi-body simulation program RecurDyn®, by Functionbay®. All simulations have been further validated in the simulation software Matlab®, by Math Works®, and also verified by laboratory test with mechanical components.

[0143] FIGS. **2***a* and **2***b* show a schematic view of the simulation model in the program RecurDyn® that was designed for the simulations according to FIGS. **1***a* and **1***b*. The schematic model comprises a moveable hammering element **10**, a counterweight **50**, a hammering element housing **5** and an impact receiving element **30** in contact with the ground **4**. The counterweight is moveable in a first axial direction A. The impact receiving element is given the function of a spring, as it provides a resilient movement in relation to the surface and simulate the effect of a work tool. The impact receiving element may also be referred to as an implement.

[0144] Now referring to FIG. **1***a*. In the simulations, the spring characteristics of the counterweight **50** are applied as forces acting on the housing **5**. For an instance in time where the counterweight **50** is at its respective end position, the force acting on the housing **5** is $F_{CW}=kx+c\dot{x}$ from the counterweight, and $F_{HE}=m\ddot{x}$ is the force from the hammering element. The force F_s from the impact receiving element **30** is determined by $F_S=Kx+C\dot{x}$. F_{HE} and F_{CW} are being arranged in the opposite directions.

The following parameters were entered in the program:

[0145] Spring coefficient for counterweight **50** set to: k=300 N/mm

[0146] Damping coefficient counterweight **50** set to: c=0.1 N*s/m

[0147] Spring coefficient against surface **4** set to: K=0.5 N/mm

[0148] Damping coefficient against surface **4** set to C=1001 N*s/m

[0149] Weight of hammering element housing **5** set to: M=3000 gram

[0150] Weight of counterweight **50** set to: m=1000 gram

[0151] Weight of hammering element **10** set to: H=300 gram

[0152] Amplitude of hammering element **10** set to: s=60 mm peak to peak, sinus-shaped.

[0153] Variable simulation parameter: frequency (f) and gap

The distinctive simulation curves according to FIG. **1** were achieved by entering the following specific constraints:

[0154] C_{gap}: Gap D_1 is fixed to 15 mm.

[0155] $C_{no\ gap}$: Gap D_1 is fixed to 0 mm

[0156] C_{locked}: Counterweight **50** is fixedly connected to the housing **5** of the hammering element.

[0157] A first simulation C_{locked} illustrates the vibrations of the hammering element housing **5** under condition that the counterweight is locked/immovable in relation to the housing of the hammering element. This graphic may serve as reference to the other two configurations in the diagram, as it illustrates an un-dampened impact machine. As the impact machine is shifted through the operating range of 2 Hz to 50 Hz, the vibration amplitude V_a plateaus just above 2 mm after 10 Hz in this example.

[0158] A second simulation $C_{no\ gap}$, illustrates the vibrations under condition that the counterweight is moveable in relation to the hammering element. The $C_{no\ gap}$, is a typical illustration of the efficiency from a "narrow range vibration reduction arrangement". As the working frequency of the impact machine is increased, the vibration amplitude V_a gradually decreases. The vibration reduction effect continues, whereby a minimum vibration amplitude V_a equals 0.1 and is achieved at 30 Hz. Notably, by achieving a vibration amplitude below at least 1 mm, in this example this opens up a possibility to design the total machine system which handles keep a vibration amplitude that meets the health and safety standards. 1 mm is just given as example when illustrating how different parameters affect the systems. If desired this limit can be set to another value for example to meet health and safety standards. For this configuration $C_{no\ gap}$, the useful frequency range of the impact machine thus lies within the range between the points D to F (27 to 32 Hz), which corresponds to a narrow interval of only 5 Hz. Moreover, a variation of 5 Hz is very a narrow interval in relation the normal/typical variations of impact machines' working frequency. Additionally, due to the steep increase in vibration at point F, where a considerable amplification of the vibration occurs, a safety zone is required, whereby the useful working frequency range is even further reduced to the range between points D to E (27 to 30 Hz), corresponding to an even narrower interval of 3 Hz. Another draw-back with this configuration is that, without safety zone and as the machine shifts out from the useful frequency range, the vibration amplitude drastically changes in such a way that a user of the impact machine would be unprepared to the sudden increase in vibrations from the impact machine.

[0159] A third simulation C_{gap} illustrates a vibration reduction effect according to an embodiment of the present invention. The counterweight is moveable in a counter-phased manner in relation to the hammering element and travels through a distance/gap where the counterweight is freely moveable and where the springs used for deaccelerating the counterweight are unattached. A significant change in the graphic is thus visible, whereby the vibration amplitude is first increasing then transitioning to a significantly lower level than for the two previous examples. Compared to the draw-backs of the configuration $C_{no\ gap}$, the illustrated embodiment of the present invention C_{gap}, achieves an efficient vibration reduction effect throughout a substantially larger frequency range between the points B and C (6 and 49 Hz), when a reduction of 1 mm is required. Another benefit from the embodiment of the present invention is that even if the vibration amplitude increases outside the useful frequency range, the increase is rather moderate such that it may be anticipated by a user of the impact machine, whereby a safety zone is not needed.

[0160] FIG. **1***b* illustrates a second set of simulations based on essentially the same impact mechanism parameters as in the example illustrated in FIG. **1***a*. The goal of the second set of simulations was to see how variations of gap length, spring coefficients and prestressing/biasing would affect the vibration, in order to better optimize the benefits.

[0161] For reference, there are two simulation setups A and B which are equal to $C_{no\ gap}$ and C_{locked} hence they respectively illustrate a more traditional setup with no gap and a setup with a locked counterweight. Traditionally, the spring has been designed based on the correlation to the counterweight's resonance frequency given by the equation (which is the same as stated on page 7):

$$F_{res} = \frac{1}{2\pi}\sqrt{\frac{k_{trad}}{m}}$$

[0162] For setups C and D, a gap of 10 mm for setup C and a gap of 15 mm for setup D have been introduced. Furthermore, the spring coefficient k has been increased compared to the traditional setup so that it is approximately 3 times higher for setup C and almost 10 times higher for setup D in order to give optimum performance at 30 Hz.

[0163] For setups E and F pretension loads for the spring action members were added to the simulation. Both setup E and F have a gap of 15 mm, similarly to setup D. Setup E was setup with a prestressed load of 300 N and the spring coefficient was roughly 3 times higher than for setup A. Simulation F was setup with a prestressed load of 400 N and the spring coefficient was roughly ⅘ of the traditional setup used in A. The results from the simulation set show promising results for all four setups C-F. Setup C in this second simulation set lies below 1 mm peak in the frequency range 14 Hz-37 Hz and is more efficient than setup B all the way up to 44 Hz. Setup D has a very small peak, <2 mm, below 5 Hz, but lies below the 1 mm level up to 60 Hz, giving an effective range from 4 Hz to at least 60 Hz. Setup E has a small peak at 3 Hz similar to setup D and then shows an effective range below the 1 mm limit to 39 Hz. Setup F has a steep increase in amplitude at 34 Hz in a similar manner to the traditional no gap setup A. However, setup F has an effective range almost down to 4 Hz. The reason for the drastically increased frequency stability is due to the fact that increased amplitude of the counterweight gives an increase in the resonance frequency as a result of the strong nonlinearity. This is believed to make the counterweight adjust its amplitude so that the resonance frequency will be optimized for the disturbing frequency. Including safety zones that should still be in the range 5 Hz to 30 Hz. The expression for the resonance frequency for the systems without pretension load and infinite weight of the housing is then given or may be selected by:

$$f_{res} = \frac{b}{2D_1 + 2\pi b}\sqrt{\frac{k}{m}}$$

[0164] Where b corresponds to the compressed distance of the spring. From the equation it can be seen that when the amplitude increases and b with it, the resonance frequency is increased. Preferably, the gap D_1 and spring coefficient k can be chosen so that the resonance frequency is close to the working frequency of the machine where the vibrations needs to be reduced. The compressed distance of the spring b is given by calculating the momentum of the impact weight that excites the system. This momentum should be the same that the spring action member absorbs from the counterweight via the endsurfaces, when the spring action member is arranged in the counterweight, and b can be calculated from that. This is a good approximation for a well functioning system with low vibration amplitude of the housing.

[0165] Setups E and F showed that advantages can be achieved by prestressed loads on the springs. Setup E has the same spring coefficient as used in setup C, but a gap of 15 mm. The vibration peaks are almost 0.5 mm lower for setup Eat 14 Hz, where setup C crosses the 1 mm limit.

[0166] A prototype machine was constructed to test the concept with tuned weight damper. The prototype was based on a redesigned construction of Atlas Copco KV 434. The tests performed are described in the following section. The prototype was constructed according to one embodiment of the invention and the test parameters during the test were:

[0167] Counterweight weight 930 g

[0168] Hammering element housing weight 4200 g

[0169] The tests were performed with three setups, in one setup the prototype did not have a counterweight, this setup should represent a reference machine and is called only "No counterweight". A second setup had a counterweight which was blocked so that it does not move relative to the prototype's housing. Essentially, the blocked counterweight adds weight to the prototype, which added weight dampens the vibrations. For the third setup the counterweight was free to move in reverse phase to the hammering element. The following settings were implemented for the spring action to the counterweight.

[0170] Spring stiffness 100 000 N/m

[0171] Gap 15 mm

[0172] In order to achieve effective vibration reduction the machine was divided into two functional parts: first a suspended weight that contains the impacting mechanism and a tuned vibration absorber comprising the counterweight, and second a housing with the interface to the operator. Vibration isolation between the suspended weight and the housing is applied in the axial, radial and rotational direction in order to handle the vibrations that still remain after the tuned absorber. Care had to be taken not to compromise with the ability to accurately control the machine. The vibrations on the handles of the machines were measured in a test rig, which yielded the same characteristics as described in ISO 8662-5. A three axis Dytran 3053B2 accelerometer with mechanical filter was used to measure the vibrations on the handles. The handle acceleration is measured in weighted vector sum hand-arm acceleration and the signals were analyzed in Labview. Vibration measurements on the hammering element housing were done with a laser displacement sensor, Contrinex, LAS-5050L, and the counterweight was measured with stroboscopic light and a steel scale. The tuned vibration absorber produced a reduction of 68% from 8.4 to 2.7 m/s2 on the handle. The stability of the operation of the tuned vibration absorber was tested by varying the air pressure to the machine from 3 to 7 Bar as well as varying the feed force from −110 N to 450 N. It was found that the vibration level varied between 2.2 and 3.6 m/s2 on the handle. An analysis of the behavior of the counterweight and how it affects the vibrations of the suspended weight was also carried out. The suspended weight displacement was 1.9 mm peak to peak while the counterweight displacement was 30.4 mm peak to peak. From those results it can be calculated that the generated peak force from the counterweight reached 684 N providing the movement of the weight is sinusoidal. For reference, the suspended weight displacement with the counterweight removed was 6.4 mm peak to peak, and 5.2 mm with the counterweight blocked. The general behavior of the vibration absorber corresponds well to the simulation with respect to a high stability of the vibration reduction over a wide frequency range and varying feed force which is the main issue. The discrepancy is mainly due to the simplified model of the excitation force which is represented as a sinusoidal force in the simulation model.

[0173] FIG. **3***a* illustrates an impact machine **100** which comprises a moveable hammering element **110** arranged inside a housing **105**. The hammering element **110** is displaceable between a first hammering position HE**1** and a second hammering position HE**2** by actuating means **115**. Examples of actuating means **115** may include a pneumatic, an electric, combustion engine or a hydraulic blow force. An implement **130**, such as a work tool is attached to the impact machine **100** in such a way that the blow force from the moveable hammering element **110** is transferred to the implement **130** in the second hammering element position H**2**. A vibration reduction arrangement **140**, not covered by the claims, is attached to the housing **105**. The vibration reduction arrangement **140** comprises a counterweight **150**, which is displaceable between a first counterweight position CW**1** and a second counterweight position CW**2** in response to the hammering action of the hammering element **110**. In the first CW**1** and the second CW**2** counterweight positions, spring-action arrangements **160**, **170** are arranged to receive and reverse the motion of the counterweight **150**. In the illustrated example, the first spring-action arrangement **160** and a second spring-action arrangement **170** each consist of one spring action member **162**, **172**. The spring-action arrangements **160**, **170** are limited in their axial movement by a first S_{end1} and a second end-surface S_{end2} which are parts of a housing of the vibration reduction arrangement **140**. In the illustrated example, the spring-action member **162** is attached to the first end-surface S_{end1} and the second spring-action member **172** is attached to the second end-surface S_{end2}. When the vibration reduction arrangement **140** is in use, the spring-action members **162**, **172** are compressed with a compression length b_1 and b_2 respectively. The compression lengths b_1 and b_2 can be varied depending on the force applied to the spring-action members **162**, **172**. The maximum compression length is achieved under operating conditions when the counterweight **150** reaches the first CW**1** or the second counterweight position CW**2**. Specifically, normal operating conditions are defined at the rated power and according to applicable ISO standards if any, such as ISO 8662-2 for hand-held portable power tools. In the embodiment illustrated in FIG. **3***a*, provided that the machine is at rest, a spring-action member **160** is arranged with its contact endpoint **165** at a distance D_1 from a contact surface **156** of the counterweight **150**. In relation to FIG. **3***a*, the first counterweight position CW**1** and the second counterweight position CW**2** define the maximum counterweight displacement distance. In these positions CW**1**, CW**2** the spring-action members **160**, **170** are compressed. In addition, damping units **180**, **190** with a damping ratio of less than 0.3 are arranged in parallel with each spring-action arrangements **162**, **172**. This may be a separate reduction component, or a natural reduction effect from the spring-action member itself.

[0174] In particular, the damping ratio can be determined by the following relationship:

$$\zeta = \frac{c}{2\sqrt{mk}}.$$

Wherein:

[0175] ζ: Damping ratio

m: weight

c: damping coefficient

k: spring coefficient

[0176] FIG. **3***b* illustrates another example of a vibration reduction arrangement which is not covered by the claims, which is arranged as the embodiment presented in FIG. **3***a*, except from that the distance D_1 has been replaced by a spring-action member **261** with a lower spring coefficient than a second spring-action **262** member. The spring-action member **272** is not fixedly connected to the counterweight **250**. Had the spring-action member **272** been fixedly connected to the counterweight, the spring characteristics of this spring-action member **272**, would reduce the effect of the weak spring and thus the vibration dampening effect. This combination of spring-action members provides for a discontinuous change in the force acting upon the counterweight **250**. In addition, damping units **263**, **273** with a damping ratio of less than 0.3 may be arranged in parallel with each spring-action arrangement **260**, **270**. This may be a separate reduction component, or a natural reduction effect from the spring-action member itself. Alternatively, the spring action member at said second counterweight position may be attached to the counterweight, but not to said second end surface (S_{end2})

[0177] Returning to FIGS. **2***a* and **2***b* for reference, a means of initiating the reversed phase movement of the counterweight according to one exemplary embodiment will be described. As the hammering element **10** may be displaced by means of for example pneumatic actuation or hydraulic actuation there is a fluid flow into the housing **5**, preferably flowing into the distal end of the housing relative the implement **30**. The housing may be provided with an outlet **95** arranged in the wall of the housing **5** so that as the hammering element **10** is displaced towards the implement **30** by the fluid, the outlet **95** is exposed. The pressurized fluid will be conducted through the outlet **95**, which is preferably arranged at a location where it opens into the counterweight **50** at least when it is in its neutral position. According to this exemplary embodiment the counterweight is arranged with a chamber conducting the pressurized fluid to a nozzle **96** which is preferably arranged at the proximal end of the counterweight, relative to the implement. The pressurized fluid will bring the counterweight into an initiated counter-phased movement by a boost of fluid through the nozzle **96**. While a flow arrangement and boost according to this exemplary embodiment is not essential for the impact machine to work results from tests have shown that the boost does reduce the time it takes for the counterweight to reach the desired counter phased movement. The flow is preferably a gas flow and most preferably an air flow, as this may be released freely. However, it may be possible to arrange a recirculation system so that other fluids may be used. It should be noted that there may be more than one outlet **95** and/or more than one nozzle **96**, for example to change the flow rate or to direct or balance the boost. Furthermore, in an exemplary embodiment where the impact machine is electrically actuated the reversed phase movement of the counterweight may be initiated by for example an electromagnetic actuator.

[0178] FIGS. **4** and **5** show a part of an impact machine **700**, similar to the one in FIG. **3***a* comprising a hammering element housing **705** with a longitudinally moveable hammering element **710** displaceable between a first hammering position and a second hammering position by actuating means (not shown) such as a pneumatic force, a hydraulic force, an electric engine or a combustion engine. Around the hammering element housing **705** is a vibration reduction arrangement **740** comprising a counterweight **750** and the counterweight **750** is being displaceable in a first axial direction A between a first counterweight position CW**1** and a second counterweight position CW**2** in response to the hammering action of said hammering element (not shown, however shown and described in regard to FIG. **10***a*). In the counterweight **750** two spring-action arrangements **760** are arranged, and the counterweight **750** is moveable a first distance $D_1/2$ from the centre position extending in said first axial direction A without actuating the spring-action arrangements **760**. The counterweight is also movable $D_1/2$ in the opposite direction without actuating the spring-action arrangements **760** (see FIG. **10***a*). According to this example the counterweight **750** has a cylindrical shape and the two spring-action arrangements **760** are arranged at equal distances from each other around the counterweight **750**. The counterweight is not limited to have only two spring-action arrangements **760**, there may be more, for example four spring-action arrangements **760**, or any integer in the interval of 1-20, which are equally distributed around the hammering element housing **705**. A first disc **798** comprising a first end surface (S_{END1}) is attached to said housing and arranged adjacent to said first counterweight position (CW**1**) and a second disc **799** comprising a second end surface (S_{END2}) is attached to said housing and arranged adjacent to said first counterweight position (CW**2**), see FIG. **10***a* and the related description, and when said spring action arrangement **760** engages said first end surface (S_{END1}) or said second end surface (S_{END2}) the spring-action arrangements (**760**) are being compressed.

[0179] FIG. **6** shows a cross-section of the counterweight **750** with its spring-action arrangements **760** in perspective. The counterweight **750** is not limited to having a cylindrical shape.

[0180] FIG. **7** and FIG. **8** show an alternatively shape where the counterweight **750** has an outer surface which approximates a truncated ellipse, i.e. it has cross-section perpendicular to said first axial direction which approximates a truncated ellipse. In more detail, a truncated ellipse is an ellipse where a section of the longest side has been cut off such that it is flat. In other words, it has one or two flat sides **751**, which each is positioned on the longest side of the somewhat elliptical shaped counterweight so that the it gets it truncated shape. This is advantageous, if for example, there is limited space when using the impact machine.

[0181] FIG. **9** shows such a situation. The user **1000** has to remove material with the impact machine **700** from a stone wall **1001** and part of the stone wall **1001** is in the way. By having a flat side on the counterweight the user can get closer to the stone wall **1001** with the impact machine **700**.

[0182] FIG. **10***a* shows a schematic cross-section of the impact machine **700** and the counterweight **750** in FIG. **5**. The vibration reduction arrangement **740** is attached to the housing **705**. The vibration reduction arrangement **740** comprises a counterweight **750**, which is displaceable between a first counterweight position CW**1** and a second counterweight position CW**2** in response to the hammering action of the hammering element (not shown).

[0183] Furthermore, the vibration reduction arrangement **740** has a motion reversing mechanism **720**, comprising a first end surface S_{END1} which is located adjacent to the first counterweight position CW**1** and a second end surface S_{END2} which is located adjacent to the second counterweight position CW**2**. Here, the first end surface S_{END1} is part of the first disc **798** and the second end surface S_{END2} is part of the second disc **799**.

[0184] The first end surface S_{END1} and the second end surface S_{END2} are located adjacent to the first CW**1** and second counterweight position CW**2** respectively, so that they are longitudinally opposite each other. Alternatively to the first and second discs **798**, **799** the first S_{END1} and second end surfaces S_{END2} are formed on the housing **705**. The counterweight **750** may comprise at least one cavity **755** and is provided with one or more openings **752** facing the first counterweight position CW**1** and/or the second counterweight position CW**2**, respectively. The cavity is normally not filled with oil or other liquids for damping purposes. Oil can however be used for lubrication purposes. Each opening **752** runs from the at least one cavity **755** to an outer surface, i.e. outer surfaces **756**, **757** which on the counterweight **750** is the respective surface proximal to the respective counterweight position CW**1**, CW**2**. Inside the counterweight **750** two spring-action arrangements **760**, are provided. A first **780** and a second projecting member **790** are arranged adjacent to the spring action arrangement **760** on opposite sides of the spring action arrangement **760**. In the embodiment illustrated in FIG. **10***a* the projecting members **780**, **790** are fixed to the spring action arrangement **760** but the connection may also be a non-fixed surface abutment. The spring action arrangement **760** is centered and biased between the projecting members **780**, **790** and the projecting members **780**, **790** are formed so that they are longitudinally displaceable inside the counterweight **750**, with a portion **753** of the projecting member **780**, **790** extending through the openings, projecting beyond the outer surfaces **756**, **757** towards the respective end surface S_{END1}, S_{END2}. On the inside of the counterweight **750** restricting means **796** are arranged comprising at least one retaining surface **794**. The projecting member **780**, **790** each further comprises a flange **797**. The retaining surface **794** restricts the motion of said flange **797** in the first axial direction (A) or in a direction opposite thereto, respectively.

[0185] The embodiment illustrated in FIG. **10***a* shows the projecting members **780**, **790** which extend the same length in the direction of the respective end surface S_{END1}, S_{END2}. However, each projecting member **780**, **790** may be formed with an individual extension length, for example to provide a non-linearity. Relative to the counterweight **750**, the projecting member **780**, **790** comprises an engaging surface **785** at the distal end. The engaging surfaces **785** of each projecting members **780**, **790** are contacting surfaces which each contacts the respective end surfaces S_{END1}, S_{END2} and the engaging surfaces **785** may hence be named contacting surfaces **785**. The counterweight **750** is shown as being arranged in the middle between the first counterweight position CW**1** and the second counterweight position, and there is a total gap having a distance D**1** between the engaging surfaces **753** of the projecting members **780**, **790** which are comprised in the counterweight **750** and the respective end surfaces S_{END1}, S_{END2} between which the counterweight **750** is movable. Hence, there is a gap, i.e. a distance of $D_1/2$ between the engaging surface **753** of the projecting member **780** and the end surfaces S_{END1} and a second gap of $D_1/2$ between the engaging surface **753** of the projecting member **790** and the end surfaces S_{END2}. When in use the counterweight **750** is movable between the first CW**1** or the second counterweight position CW**2** in reaction to the movement of the hammering element, and consequently the movement of the housing **705** may be reduced. For example, as the counterweight **750** approaches the first counterweight position CW**1** the engaging surface **785** of the projecting member **780** engages with the first end surface S_{END1} and compresses the spring-action arrangement **760**, i.e. the spring-action arrangement **760** is actuated, (see FIG. **10***a*') the counterweight continues towards S_{END1} until it reaches its turning point where at the motion of the counterweight **750** is reversed. In FIG. **10***a*' the counterweight is in the first counterweight position CW**1** and when said counterweight **750** is arranged in said first counterweight position CW**1** said engagement surface **785** and said first end surface S_{END1} are pressed against each other, said engagement surface **785** is displaced relative a center of gravity of said counterweight compared to when said counterweight is arranged in a position where said engagement surface **785** and said first end surface S_{END1} are separated from each other (as in FIG. **10***a*), and said at least one spring-action arrangement **760** is actuated.

[0186] When the counterweight **750** is in the first CW**1** and the second CW**2** counterweight positions respectively, the motion reversing mechanism **720** is arranged to receive and reverse the motion of the counterweight **750**. In the illustrated example, the spring-action arrangement **760** comprises one spring action member **762**. The spring-action member **762** is limited in its axial movement by a first and a second retaining-surface **794** which are parts of the counterweight **750**. In the illustrated example, the spring-action member **762** is attached to the projecting members **780**, **790**, which when at a distance D**1** from either counterweight position CW**1**, CW**2** are retained by the first and second retaining-surface **794** so that the spring-action member **162** is pretensioned. Normally, the longer the spring action member **762** the better, as this increases the lifetime of the spring action member.

[0187] The counterweight in FIG. **10***a* shows two spring-action arrangements **760**, one on each side of the hammering element. **2**. However it may be advantageous to have a higher number of thinner spring action members **762**, when a small radius of the counterweight is desired.

[0188] FIG. **10***b* illustrates a cross-section of an embodiment in which the counterweight **750** is provided with more than one cavity **755***a*, **755***b*. The cavities **755***a*, **755***b* are formed analogously to the at least one cavity illustrated in FIG. **10***a*, but separated perpendicularly relative to the longitudinal direction by a separating member **758** comprised in the counterweight **750**. Each cavity **755***a*, **755***b* is provided with a spring action arrangement **760'** comprising one or more spring action member **762'** each. The one or more spring action members **762'** being connected to one of the projecting members **780**, **790** each. The arrangement with more than one cavity **755***a*, **755***b* provides the possibility to have individual pretension, individual spring action members **762'** that may have different spring characteristics, etc. When the vibration reduction arrangement **740** is in use, the spring-action members **762'** are compressed with a compression length b_1 and b_2 respectively. The compression lengths b_1 and b_2 can be varied depending on the force applied to the spring-action members **762'**. Analogously, for the embodiment illustrated in FIG. **10***a*, the spring action member **762** is compressed with a compression length b1 at each end. The maximum compression length is achieved under operating conditions when the counterweight **750** reaches the first CW**1** or the second counterweight position CW**2**. Specifically, normal operating conditions are defined at the rated power and according to applicable ISO standards, such as ISO 8662-2 for hand-held portable power tools or in analogy with this.

[0189] In the example illustrated in FIG. **10***b*, provided that the impact machine is at rest, the motion reversing mechanism **720** is arranged with its engaging surface **785** at a distance $D_1/2$ from the end surface S_{END1}. In relation to FIG. **10***b*, the first counterweight position CW**1** and the second counterweight position CW**2** define the maximum counterweight displacement distance.

[0190] With reference to FIGS. **10***a* and **10***b* the projecting members **780** are dimensioned in view of the openings of the counterweight **750** such that they may move longitudinally relative to the counterweight, as the spring action members **762**, **762'** are compressed. In the illustrated examples the projecting members **780**, **790** are retained by the first and second retaining-surface **794** of the counterweight **750** when the spring action members **762**, **762'** are least compressed. In the counterweight positions CW**1**, CW**2** the spring-action members **762**, **762'** are compressed, and the projecting members **780**, **790** are dimensioned so that its engaging surfaces **785** are extending at least longitudinally beyond the first **758** and second retaining-surface **759** of the counterweight **750**, or beyond the outer surface **756**, **757** of the counterweight **750**. The distance of the gap D_1 can be calculated as the distance Lcw between the first CW**1** and the second counterweight position CW**2** subtracted by the exterior length V or LE (maximum axial length) of the counterweight **750** including the extention of the projecting members **780** from the outer surfaces **756**, **757** to the respective engaging surfaces **785**.

[0191] FIG. **11***a* shows another exemplary embodiment of the counterweigh **750** which is similar to the one shown in FIG. **10***a* except that the projecting members **780**, **790**, which are formed so that they are longitudinally displacable inside the counterweight **750**, with a portion **753** of the projecting member **780**, **790** extending through the openings **752** of the counterweight, projecting beyond the outer surfaces **757**, **756** towards the respective end surface S_{END1}, S_{END2}, each comprises a further projecting portion **754**. The projecting portions **754** are each arranged in a hole **800** of the first disc **798** and the second disc **799** respectively. When the projecting members **780**, **790** are being longitudinally displaced the projecting portions **754** are longitudinally displaceable inside the holes **800**. Each projecting members **780**, **790** further comprises a hole **801** into which the spring action member **762** extends into. Preferably, a larger part of the spring-action members are arranged inside the counterweight. Preferably 75% is arranged inside the counterweight, more preferably 50%. The spring action member illustrated in FIG. **11***a* has just started to be compressed, and the counterweight **750** will continue its motion towards the turning point, which causes a compression of the spring action member. Preferably, bur not necessary, before the outer surface **756** contacts the first end surface S_{END1}, the counterweight will have reached its turning point and the direction of motion of the counterweight will be reversed.

[0192] FIG. **11***b* shows another exemplary embodiment of the counterweigh **750** where one of the projecting members **790** has a the same design the projecting members **790** in FIG. **11***a* and the other projecting member **780** has a the same design as the projecting member **780** shown in FIG. **10***a*, except that the spring action members **762** protrudes into an hole **802** in the projecting member **780**.

[0193] The vibration reduction arrangements described above in regard to FIGS. **4**-**11***b*. has to be optimized in order to get as large vibration absorbation as possible. This can be done in three steps:

[0194] Step 1: Insert the Following Input Data

[0195] Main mass (M) excluding the hammering element, the impact receiving element and the counterweight.

[0196] It is used for calculating the force in alternative two and the optimisation in the last step.

[0197] Hammering element mass ($m_{hammering\ element}$)

[0198] Counterweight mass (m)

[0199] Main operating frequency (f)

[0200] Lowest operating frequency (f_{min})

[0201] Highest operating frequency (f_{max})

[0202] (The frequencies are usually known from a prototype machine. (f_{max}) and (f_{max}) are consequences from tolerances in the manufacturing process, different air pressures in use and feed force and the impact receiving element

[0203] Excitation in terms of momentum from hammering element movement on main mass gives the spring compression "b"

[0204] Alt 1: Calculation from hammering element mass, displacement and frequency. The compression distance (b) of the spring is determined by equation (2) below.

[0205] Alt 2: Calculation from measured vibrations on machine. The compression distance (b) of the spring is determined by equation (3) below.

[0206] The force F and the compensation (b) can be used during the optimisation in order to increase the optimisation.

[0207] Step 2: Determine Start Values for the Tuned Mass System Optimization

Select a combination of k, a and F_0 that gives a resonance frequency close to f by the following expression that gives an approximate resonance frequency for the tuned mass.

$$f := \sqrt{\frac{k \cdot b^2}{m \cdot (4 \cdot a + 2 \cdot \pi \cdot b)^2} + \frac{F_0 \cdot b}{8 \cdot m \cdot (2 \cdot b + a)^2}} \qquad \text{Equation (1)}$$

Where:

[0208] a is the gap from the central position

F_0 is the spring pretension

b is the compression of the spring under operation at rated power. For a hammering element or main mass with a sinusoidal movement b is given by the expression.

$$b := \frac{-F_0}{k} + \sqrt{\frac{F_0^2 \cdot b}{k^2} + \frac{m_{piston}^2 \cdot v_{piston\_peak}^2}{k \cdot m}} \qquad \text{Equation (2)}$$

Where piston=hammering element

$$b := \frac{-F_0}{k} + \sqrt{\frac{F_0^2 \cdot b}{k^2} + \frac{M^2 \cdot v^2}{k \cdot m}} \qquad \text{Equation (3)}$$

Where $v_{hammering\ element}$ is the maximum velocity of the hammering element.

The equation is based on that the hammering element and the counterweight gives equal impulse to the main mass.

[0209] Step 3: Optimization of Tuned Mass System

The intention is to find the best combination of parameters that gives the lowest vibration amplitude in the frequency interval between f_{min} and f_{max}.

Alt 1: Optimization in simulation model. Design a simulation model that represents the mechanical system. Preferably the model of the system is built in a mathematical simulation program such as Matlab, Octacve etc. or in a Multi Body

Simulation program (MBS) such as Recurdyn, ADAMS etc. The parameters are then preferably optimized by he method of Design Of Experiment (DOE) or using built in optimization functions in the software packages.

Alt 2: Experimental optimization. The optimization is done by changing the parameters on a physical prototype machine This is preferably done by the method of Design Of Experiment (DOE).

[0210] The optimization gives information on the combination of play, spring rate, spring pretension, counterweight etc. Which gives the lowest value on a target function which shall be minimized. The target function is defined as the area under the vibration curves in FIG. **1***b* for the Main mass (M) between the lowest operating frequency (f_{min}) and the highest operating frequency (f_{max}).

[0211] The term Design of Experiments may be interpreted as: "in statistics, these terms are usually used for controlled experiments. Formal planned experimentation is often used in evaluating physical objects, chemical formulations, structures, components, and materials. Other types of study, and their design, are discussed in the articles on computer experiments, opinion polls and statistical surveys (which are types of observational study), natural experiments and quasi-experiments (for example, quasi-experimental design). In the design of experiments, the experimenter is often interested in the effect of some process or intervention (the "treatment") on some objects (the "experimental units"), which may be people, parts of people, groups of people, plants, animals, etc. Design of experiments is thus a discipline that has very broad application across all the natural and social sciences and engineering."

[0212] The diagram in FIG. **12** is a verification of the Matlab simulation model described above with experimental results and shows a good compliance. A test rig where designed and manufactured carefully so relevant parameters could be controlled. FIG. **13** shows the used test rig **1000**. It comprises of a main mass **1001** that can slide with low friction on a supporting structure **1002**. On the main mass **1001** there is a counterweight **1003** that moves with low friction between springs **1004**. The excitation force is created by a piston in side a cylinder **1005** driven by an electrical motor **1006**. The piston is connected to the main mass **1001** with brackets **1007** and the force created from the piston is measured by a force sensor (not shown). The vibrations of the main mass **1001** and counterweight **1003** are measured with laser displacement sensors. With the test rig **1000**, the parameters of force, frequency, gap, spring coefficients, main mass, counterweight can be controlled. The test were made for two setups, one with the counterweight fixed to the main mass and one with the counterweight active. The results are compared to the simulated results in FIG. **12**.

Parameters were chosen for an optimum performance at 9 Hz.

The input data for the experiment and simulation were:

M = 4.8 kg	main mass
m = 1.5 kg	counterweight mass
k = 14800 N/m	spring coefficient counterweight
K = 100 N/m	spring coefficient main mass
C = 100 Ns/m	damping main mass
c = 0.1 Ns/m	damping counterweight
a = 9 mm	gap from neutral position

The excitation force in the simulation model were: F=(1.55*omega−20)*sin(omega*t) and this is an estimation of the measured input force.

[0213] FIG. **12** shows that the vibration amplitude of the main mass for the fixed counterweight (in FIG. **12** the counterweight is called counter mass), both when simulated and measured decreases almost linear when the working frequency increases. When the frequency is 4 Hz, the vibration amplitude is around 7.5 mm and when the frequency is 13 Hz, the vibration amplitude is around 3.5 mm peak. When the counterweight is active both the simulated and measured values follows almost the same curve. At a frequency of 4 Hz the amplitude is around 2.5-3 mm peak and it decreases as the frequency gets higher until it reach around a frequency of 9 Hz where the vibration amplitude is below 1 mm peak. The vibration amplitude increases then again as the frequency gets higher.

An example of the development method for the dimensioning of the vibration reduction arrangements described above comprises, as stated above, the steps of providing the input parameters, determining start values for the variable parameters and optimizing the vibration reduction arrangement. In the example case the input variables are as follows:

[0214] The main mass weighs 8 kg, which excludes the hammering element mass, counter weight and the impact receiving element.

[0215] The hammering element mass is 0.5 kg.

[0216] The counterweight has a total mass of 1 kg, (if there are several separate counterweights the weight is the total weight of all counterweights.)

[0217] From the development of the impact machine it is known that the main operating frequency f is 30 Hz. To accommodate variations in manufacturing tolerances, pneumatic pressure, applied force, etc. there is a variation around the main operating frequency resulting in an operating frequency range with a minimum frequency f_{min} 25 Hz and a maximum frequency f_{max} 35 Hz.

[0218] In order to improve the optimization process the excitation momentum on the main mass may be calculated and used as an input; Two ways of calculating this is to for example calculate the excitation momentum from the displacement and frequency of the hammering element, or to calculate the excitation force F from vibrations measured on the impact machine.

[0219] A goal of this exemplary development process is to determine optimal dimensions for the spring constant(s) k, the gap a and the spring pretension F_0. Where a is the distance D**1** when the counterweight is in the central position. Starting values of these parameters are given by the following expression for an approximate value for the operating frequency:

$$f := \sqrt{\frac{k \cdot b^2}{m \cdot (4 \cdot a + 2 \cdot \pi \cdot b)^2} + \frac{F_0 \cdot b}{8 \cdot m \cdot (2 \cdot b + a)^2}}$$

The equations are based on that the main mass is standing still

[0220] The compression of the spring b under operation is given by an value which is being measured during a non-linear movement. This movement is then fed into the simulation model and optimized therefrom. However, for most cases an assumption regarding a sine-wave movement is an acceptable approximation.

[0221] The optimization is then performed by the method of Design of Experiments.

[0222] FIG. **14** shows a schematic perspective view of a part of an impact machine according to another exemplary embodiment of the invention. The counterweight **750** is arranged only partly around the housing **705** of the hammering element. A second counterweight **750'** is arranged on the opposite side of the impact machine. The counterweights **750**, **750'** are moveable between the first counterweight position CW**1** and the second counterweight position CW**2** in response to the impact action of the hammering element. Each counterweight **750**, **750'** is guided between the two positions CW**1** and CW**2** by means of counterweight guiding means **781**. In this embodiment, each guiding means **781** is internally arranged within the counterweight **750**, **750'**. However, other possible modifications may include external guiding means, such as an external housing, which encloses and guides the counterweight by its periphery. The counterweights may have a cross-sectional internal design as disclosed in FIGS. **4-8** and **10***a*-**11***b* however, each spring-action arrangement is instead arranged in a separate counterweight, they will however work in the same way.

[0223] Now referring to FIGS. **15***a*-**15***l*, which schematically illustrate other examples of possible embodiments, not covered by the claims. In common for all embodiments according to FIGS. **15***a*-**15***h* is that the counterweight **550** is provided with an axial length Lccw between the contact surfaces **556**, **557** of the counterweight **550**. Furthermore, the first spring-action arrangement **560** is limited in its axial movement by a first end surface S_{end1} and the second spring-action arrangement **572** is limited in its axial movement by a second end surface S_{end2}. The first spring action arrangement **560** has an unactuated length L_{SA1}, and the second spring action arrangement **572** has an unactuated length L_{SA2}. The unactuated length for the first spring action arrangement **560** is the distance between the end surface S_{end1} and the contact endpoint **565**. The unactuated length for the second spring action arrangement **572** is the distance between the surfaces S_{end2} and the contact endpoint **575**. The distance D_{end} between the first S_{end1} and the second end surface S_{end2} is the total axial distance inside the vibration reduction arrangement. The counterweight **550** is axially moveable in a first axial direction A, between a first CW**1** and a second counterweight position CW**2**, in which the spring-action arrangements **560**, **572** are compressed. Specifically, in the embodiments illustrated in FIGS. **15***a*-**15***h*, the length of the gap D_1 can be calculated as:

$$D_1=D_{ends}=L_{ccw}-L_{SA1}-L_{SA2}$$

[0224] FIG. **15***a* illustrates that the first **560**, and the second spring-action arrangements **572** can include a first **562***a*, **572***a* and a second spring-action member **562***b*, **572***b* arranged in parallel as pair at each counterweight position CW**1**, CW**2**. The unactuated axial length of the first spring-action member **572***a*, **762***a* can be shorter than the axial length of the second spring-action member **572***b*, **562***b* included in a pair. However, the unactuated lengths of the first **560** and the second **572** spring-action arrangements L_{SA1}, L_{SA2} are defined by the longest spring-action member. FIG. **15***b* illustrates that the first **560** and the second **572** spring-action arrangements can comprise a single spring-action member. FIG. **15***c* illustrates that the first **560** and the second spring-action arrangements **572** may include two spring-action members of the same length and which are arranged in parallel as pair at each counterweight position.

[0225] Now referring to FIG. **15***d*, which illustrates an embodiment spring-action arrangements are fixedly attached to the counterweight **550**. However, the first S_{end1} and the second end surfaces S_{end2} serve as abutment surfaces in this case. Furthermore, FIGS. **15***e* and **15***f* show that it is also possible to modify the arrangement in FIG. **15***d* by arranging springs in parallel with the same or with different lengths.

[0226] Now referring to FIG. **15***g*, which illustrates an embodiment, in which the first **560** and the second **572** spring-action arrangements consist of loose spring-action members **562**, **572** arranged on each side of the counterweight **550**, such that they are only abutting the end surfaces and an the counterweight **550**.

[0227] FIG. **15***h* illustrates a still further embodiment, in which a combination of a spring-action arrangement **560** attached to the counterweight **550** and a spring-action arrangement **572** attached to an end surface S_{end1} is possible.

[0228] FIG. **15***h* illustrated another possible embodiment wherein a first spring action arrangement **560** comprises a first spring-action member **562***a* and a second spring-action member **562***b* with different lengths. The spring action members **562***a*, **562***b* are arranged in parallel with each other at a first counterweight position CW**1**. Furthermore, the longest spring-action member may be fixedly attached to the counterweight **550**. Alternatively, the longest spring-action member may be abutting the counterweight **550** in an unactuated condition.

[0229] FIG. **15***j* illustrates an embodiment where the gap D_1 has been replaced by a weak spring-action member **561**. The gap D_1 is thus represented by the unactuated length of the weak spring-action member **561** and can be calculated from the same relationship as for the FIGS. **15***a*-**15***h*, or by simply measuring the length of the weak spring action member **561** itself.

[0230] Additionally, other examples of constructions, are possible as variants or as complements For example, the first spring-action arrangement inside the counterweight may comprise more than one spring-action member, arranged in parallel or in series with the first spring-action member, in a similar manner to the examples described. Furthermore, the first spring-action arrangement inside the counterweight may be complemented by a second spring-action arrangement according to one of the illustrated examples, which are outside the counterweight. For instance, as illustrated in FIG. **15***k*, another embodiment may include a combination of a gap D_1 and a combination of at least two spring-action members **561**, **562** with different spring coefficients, connected to each other in series at a first counterweight position CW**1**. At the second counterweight position CW**2**, a single spring-action member may be arranged. The counterweight **550** is not fixedly attached to the spring-action members. FIG. **15***l* illustrates still another possible embodiment which comprises at least two spring-action members with different spring coefficients connected in series at each counterweight position CW**1**, CW**2**, whereby the counterweight **550** is fixedly connected to the spring-action members.

[0231] As schematically illustrated in FIGS. **16***a*-**16***c*, other embodiments not covered by the claims may include biased spring-action arrangements. The length of the gap D_1 can be calculated from the same relationship as for the FIGS. **15***a*-**15***h*.

[0232] As illustrated in FIG. **16***a*, a way to bias a spring-action arrangement is to retain it in a partially compressed state. The biased spring-action members **562**, **572** are arranged in compartments defined by an end surface S_{end1}, S_{end2} and a retaining surface R_{S1}, R_{S2}. The contact surfaces **556**, **557** of the counterweight **550** may be adapted such that the axial length Lccs of the counterweight **550** between the counterweight contact surfaces **556**, **557** is longer than an exterior length L_E of the counterweight **550**.

[0233] As illustrated in FIG. **16***b*, the spring-action arrangement **560** can be built-in inside the counterweight **550**. The retaining surfaces are arranged inside the counterweight **550**. The end surfaces S_{end1}, S_{end2} are provided with a protruding portion such that an abutment with the spring-action arrangement **560** is possible. The protruding portions are dimensioned such that they only contact the spring action arrangement without hindrance or contact with other parts. In this embodiment, a single spring-action member **562** may be arranged inside the counterweight **550**, with an axial extension between the first R_{SA1} and the second retaining surfaces R_{SA2}. The length of the counterweight **550** in the direction between the contact points is null.

[0234] For embodiments with built-in spring action members, the distance of the gap D_1 can be calculated as the distance Lcw between the first CW**1** and the second counterweight position CW**2** subtracted by the exterior length L_E (maximum axial length) of the counterweight **550**.

[0235] As illustrated in FIG. **16***c*, a first **560** and a second spring-action arrangement **570** may be arranged inside the counterweight **550** and are separated by a wall inside the counterweight **550**. The first **560** and the second spring-action arrangements **570** are retained by the wall inside the counterweight **550** at a first distal end and at a first retaining surfaces at the second distal end. The length of the counterweight in the direction between the contact points is null. The list of possible embodiments and modifications is non-exhaustive.

[0236] FIG. **17** shows a schematic perspective view of an impact machine according to another exemplary embodiment not covered by the claims. As shown in FIG. **17**, the counterweight **350** has a cylindrical shape and is arranged around the housing **305** of the hammering element **310**. The counterweight **350** is moveable between the first counterweight position CW**1** and the second counterweight position CW**2** in response to the impact action of the hammering element **310**. The counterweight **350** is guided between the two positions CW**1** and CW**2** by means of counterweight guiding means **381**. In this embodiment, the guiding means **381** are internally arranged within the counterweight **350**. However, other possible modifications may include external guiding means, such as an external housing, which encloses and guides the counterweight by its periphery. Motion reversing mechanisms **380**, **390** are arranged at the first counterweight position CW**1** and the second counterweight position CW**2**. In this embodiment, the motion reversing mechanisms **380**, **390** comprise two coil springs arranged at each counterweight position CW**1**, CW**2**. A gap D_1 is arranged between a first counterweight position CW**1** and the second counterweight position CW**2**. In particular the gap D1 is illustrated in FIG. **15***a* as the distance between a contact surface **356** of the counterweight **350** and a contact surface **365** of the spring-action member **362**.

[0237] The working principle of an impact machine, as disclosed in the appended drawings, is to apply a reoccurring force upon the hammering element **110**, **210**, **310** such that the hammering element **110**, **210**, **310** performs a periodical movement between a first HE**1** and a second HE**2** hammering element position. A reoccurring force on the hammering element **110**, **210**, **310** is achieved by actuating means **115**, **215**, **315**. Typically, hand-held impact machines operate with hydraulic, combustion engine or electric power, while larger machine-mounted impact devices are powered with hydraulics or pneumatics. The purpose of the actuating means **115**, **215**, **315** is to transfer impact energy to the hammering element **110** by either electric/mechanical, hydraulic or pneumatic means such that an impact force is applied to the hammering element. An example how this may be done is to periodically supply air or hydraulic fluid to and from an expandable chamber, which in turns transfers the force to the hammering element **110**, **210**, **310**. In the second position HE**2**, the hammering element **110**, **210**, **310** is in mechanical contact with an implement **130**, **230**, **330**, such as a work tool, whereby the hammering element **110**, **210**, **310** transfers a blow force to the implement **130**, **230**, **330**. Correspondingly, the function of the vibration reduction arrangement **140**, **240**, **340** is to bring the counterweight **150**, **250**, **350** into a counter-phased movement in relation to the hammering element **110**, **210**, **310**, such that the vibrations from the hammering element **110**, **210**, **310** are counter-acted by the vibrations from the counterweight movement. The implement/work tool **130**, **230**, **330** may be positioned in any direction, whereby the movement of the counterweight is aligned with the hammering element. In order to bring the counterweight **150**, **250**, **350** into the right counter-phased movement, the travel length and the spring coefficients cause the counterweight **150**, **250**, **350** to move according to a particular resonance frequency. However, it may be an advantage to rapidly bring the counterweight **150**, **250**, **350** into the right frequency so that the reduction effect may be instantaneously created when the impact machine is turned on. This may be achieved by applying an essentially instantaneous force on the counterweight, whereby the exhaust of pressurized air or liquid from the housing **105**, **205**, **305** of the hammering element may be used. A first motion reversing mechanism **160**, **260**, **360** is located at the second counterweight position (CW**2**) and a second motion reversing mechanism **170**, **270**, **370** is located at the first counterweight position. In particular, the motion reversing mechanisms **160**, **170**; **260**, **270**; **360**, **370** receive and reverse the counterweight **150**, **250**, **350** as it moves between the first CW**1** and second counterweight position CW**2**.

[0238] FIG. **18** shows a schematic cross-sectional view of the principle components of the impact machine and the vibration reduction arrangement according to an exemplary embodiment, not covered by the claims. A dampening arrangement comprises a cylindrical counterweight **450** which encloses the housing **405** of the hammering element **410**. The vibration reduction arrangement **440** may be dimensioned based on the impact machine's typical working frequency f_{work} and the related vibration amplitude V_a. The weight H of the hammering element **410** and the distance between the first HE**1** and the second HE**2** hammering element position are thus typically known parameters. In addition, the weight M of the hammering element housing

405 M is a known parameter. Further, FIG. **18** may be used for exemplifying how the gap D_1 may be determined and for the embodiment illustrated in FIG. **16**. Firstly, the vibration reduction arrangement **440** is arranged such that both spring-action arrangements **460**, **470** are uncompressed, thereafter the counterweight **450** is arranged mid-way between the contact surfaces **465**, **475** of the respective spring-action arrangements **460**, **470**. In a second step, the gap D_1 is determined.

[0239] Notably, the gap D_1 corresponds to a distance Dends between the contact surface **465** of the first spring-action arrangement **470** and a contact surface of the counterweight **456**, and a second distance $D_1/2$ between a contact surface **475** of a second spring-action arrangement **470** and a second contact surface **457** of the counterweight **450**. It is also provided that the first **460** and the second spring-action arrangement **470** are arranged on opposite sides of the counterweight **450**. The principal concept of the vibration reduction arrangement is to counter-act the vibrations from the hammering element **410**. By consequence, the maximum force F_{HE} for the hammering element should therefore be essentially the same as the maximum force for the counterweight F_{CW}. In particular, the accelerating forces acting on the counterweight **450** and the hammering element **410** should be equal such that the counter force on the hammering element housing **405** is the same. The counterweight's **450** movement is restricted to a maximum counterweight displacement distance between a first counterweight position CW**1** and a second counterweight position CW**2**. A movement beyond these points is thus not possible. Hence, depending on e.g. the hammering element **410** acceleration forces, the travel distance of the counterweight **450** may be at maximum travel distance, or shorter.

[0240] Hence, the distance between the first CW**1** and the second counterweight position CW**2** and the weight of the counterweight **410** may be varied and adapted to geometric and weight constraints of the impact machine. As in the practical example given in relation to the diagram in FIG. **2**, the weight of the hammering element is 0.3 kg and the weight of the counterweight is 1 kg. The total weight of the hammer element housing is 3 kg.

[0241] The next step may be to select a resonance frequency f_{res} of the counterweight. Based on a predetermined/desired working frequency of the impact machine, a resonance frequency of the counterweight is selected so that it lies sufficiently far above the predetermined working frequency of the impact machine so that the dampening effect is good at the working frequency. In this particular example, based on that the normal operating frequency of the machine is 30 Hz, the resonance frequency is selected to 32 Hz. Following, based on the selected counterweight resonance frequency f_{res} the spring coefficient k for the counterweight may be determined from the following relationship:

$$f_{res} = \frac{b}{2D_1 + 2\pi b}\sqrt{\frac{k}{m}}$$

[0242] Where:

[0243] m: weight of the counterweight **300** (grams)

[0244] f: resonance frequency of the counterweight (Hz)

[0245] b: compressed distance of the spring(s)

In cases where there is no gap in the arrangement the coefficients are advantageously chosen by simulation.

[0246] FIG. **19** illustrates a diagram with four different spring-action member embodiments, which present a non-linear spring-action force as a function of the counterweight displacement x. The inclination of the curves may be referred to as a spring coefficient k. FIG. **16** geometrically shows the counterweight displacements x in relation to an embodiment, not covered by the claims $F_{k\ gap}$. Notably, as geometrically presented in FIG. **18**, the starting point (at 0 mm) as illustrated in FIG. **16** relates to the position O, whereas the endpoint (at 12 mm) always relates to the position CW**2** for all spring-action member embodiments. Moreover, the position O can be seen as when the counterweight **450** is in equilibrium, when the applied spring-action force is null.

[0247] Now referring to FIG. **19**, a first configuration $F_{2\ springs}$ comprises a counterweight in contact with two spring-action members, connected in series at one side of the counterweight in its travel direction.

[0248] A second configuration $F_{1\ spring+Gap}$ comprise a counterweight is in contact with one spring-action members, but where a gap is introduced in the travel path of the counterweight. In the second configuration, it should be noted that the gap is actually only half a gap $D_1/2$, which is geometrically illustrated in FIG. **18**. A third configuration $F_{3\ springs}$ relates to an embodiment of the present invention, where the counterweight is in contact with three spring-action members connected in series.

[0249] Due to the change in spring coefficient and depending on the stiffness of the spring-action members, the counterweight is being subject to two zones with a different spring coefficient: a "low spring force zone" and a "high spring force zone". The low spring-force zone corresponds to the beginning of the compression of a spring-action member, where the spring-action force on the counterweight is at its lowest level. The high spring-force zone corresponds to the end of the compression of a spring-action member, where the spring-action force on the counterweight is at its highest level. It has been realized in the context of the present invention that in order to achieve an efficient vibration reduction effect for a wide working frequency range, the length of the low spring-force zone may correspond to at least 25% of the length of the high spring-force; and the average spring coefficient in the low spring-force zone should be lower than 50% of the spring coefficient in the high spring-force zone. Another discovery from the experimental studies is that if the embodiment includes a gap, the distance of the gap may be selected to around 30% of the total travel distance between first CW**1** and second counterweight position CW**2**.

[0250] In the relationship above, k stands for the total spring coefficient acting upon the counterweight in the "low" and "high" spring-action force. If the spring-action member is a combination of two spring-action members connected in series, then the spring coefficient for each spring-action member k_1 and k_2 should sum up to a total, i.e. an equivalent spring coefficient k_{ekvl} according to the following equation:

$$\frac{1}{k_{ekv}} = \frac{1}{k_1} + \frac{1}{k_2}$$

[0251] FIGS. **20***a* to **20***c* illustrate different embodiments relating to the mounting of the vibration reduction arrangement, however not covered by the claims. FIG. **20***a* shows an internally mounted vibration reduction arrangement **640**. In this example, the vibration reduction arrangement **640** is integrated into the impact machine and may be mounted inside a housing **605** of the impact machine. As illustrated, the vibration reduction arrangement **640** may be centered around the displacement axis of the hammering element.

[0252] FIG. **20***b* shows another possible embodiment, not covered by the claims, where the vibration reduction arrangement **640** is externally mounted around the housing **605** of the impact machine an attached thereto with fastening means **647**. An advantage of this arrangement is that it may be used as an attachment. The vibration reduction arrangement **640** may have a symmetrical and distributed counterweight around the circumference of the housing of the impact machine.

[0253] FIG. **20***c* shows another possible embodiment, not covered by the claims where the counterweight's travel direction is off-set at an angle θ in relation to the travel direction of the hammering element. This embodiment may bring advantages if there is need to reduce vibrations in both vertical and horizontal direction.

EXEMPLYFING EMBODIMENTS

Embodiment 1

[0254] An impact machine (**100**; **200**) comprising:

[0255] a housing (**105**; **205**)

[0256] a hammering element (**110**; **210**) arranged inside said housing (**105**; **205**), said hammering element (**110**; **210**) is displaceable between a first hammering element position (H**1**) and a second hammering element position (H**2**),

[0257] an impact receiving element (**130**; **230**) attached to said housing (**105**; **205**),

[0258] actuating means (**115**; **215**) arranged to cause said hammering element (**110**; **210**) to perform a hammering operation on said impact receiving element (**130**; **230**),

[0259] a vibration reduction arrangement (**140**; **240**) attached to said housing (**105**; **205**), which comprises:

[0260] a counterweight (**150**; **250**) being displaceable in a first axial direction (A) between a first counterweight position (CW**1**) and a second counterweight position (CW**2**) in response to the hammering action of said hammering element (**110**; **210**),

[0261] at least one motion reversing mechanism (**180**; **280**) each of said motion reversion mechanism comprising at least one spring-action arrangement (**160**; **260**), each of said at least one spring-action arrangements (**160**; **260**), being arranged to reverse the direction of motion of said counterweight (**150**; **250**),

wherein

[0262] said counterweight (**150**) is arrangeable at a position located between said first counterweight position (CW**1**) and said second counterweight position (CW**2**) from which position said counterweight (**150**) is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said at least one spring-action arrangement (**160**);

or wherein

[0263] said vibration reduction arrangement (**240**) further comprises a first end surface (S_{End1}), said at least one spring action arrangement (**260**) being arranged between said counterweight (**250**) and said first end surface (S_{End1}), said at least one spring action arrangement (**260**) comprising a first spring-action member (**261**) attached to said counterweight (**250**), and a second spring-action member (**272**) arranged in series with said first spring-action member (**261**) in said first axial direction (A) and being attached to said first end surface (S_{End1}) and said first spring action member (**261**); said first spring-action member (**261**) having a first spring characteristics comprising a first spring coefficient (k_1) within the interval $-k_{trad} \leq k_1 \leq k_{trad}/2$ and $k_1 \neq 0$, and the second spring-action member (**262**) having a second spring characteristics comprising a second spring coefficient (k_2) within the interval $k_{trad}/5 \leq k_2 \leq 30^* k_{trad}$, and k_{trad} is determined from the following formula

$$F_{res} = \frac{1}{2\pi}\sqrt{\frac{k_{trad}}{m}}$$

F_{res} being the resonance frequency of the impact machine at rated power, and m the weight of the counterweight (**250**), or wherein

[0264] said vibration reduction arrangement (**540**) further comprises a first end surface (S_{End1}), said at least one spring action arrangement (**560**) being arranged between said counterweight (**550**) and said first end surface (S_{End1}), said at least one spring action arrangement (**560**) comprising a first spring-action member (**562***a*) and a second spring-action member (**562***b*), a first end of said first spring action member (**562***a*) and second spring-action member (**562***b*), respectively, is attached to said counterweight (**550**); a second end of said first spring action member (**562***a*) and second spring-action member (**562***b*), respectively, are attached to said first end surface (S_{End1}), said first (**562***a*) and second spring members (**562***b*) being arranged in parallel with each other in said first axial direction (A), wherein the first spring-action member (**562***a*) having a first spring characteristics comprising a first spring coefficient (k_1) being arranged within the interval $-k_{trad} \leq k_1 \leq k_{trad}/2$ and $k_1 \neq 0$, and the second spring-action member (**562**) having a second spring characteristics comprising a second spring coefficient k_2, arranged within the interval $k_{trad}/5 \leq k_2 \leq 30^* k_{trad}$, and k_{trad} is determined from the following formula

$$F_{res} = \frac{1}{2\pi}\sqrt{\frac{k_{trad}}{m}}$$

F_{res} being the resonance frequency of the impact machine at rated power, and m the weight of the counterweight (**550**).

Embodiment 2

[0265] The impact machine according to embodiment 1, wherein said counterweight (**550**) is arrangeable at a position located between said first counterweight position (CW**1**) and said second counterweight position (CW**2**) from which position said counterweight (**550**) is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said at least one spring-action arrangement (**160**) and said at least one spring action arrangement (**560**) being arranged inside said counterweight (**550**), and said vibration reduction arrangement further comprises:

[0266] a first end surface (S_{End1}) arranged adjacent to said first counterweight position (CW**1**) and

[0267] a second end surface (S_{End2}) arranged adjacent to said second counterweight position (CW**2**);

said first end surface (S_{End1}) is arranged to receive said at least one spring action arrangement (**560**) when in motion towards said first counterweight position (CW**1**); and

said second end surface (S_{End2}) is arranged to receive said at least one spring action arrangement when in motion towards said second counterweight position (CW**1**).

Embodiment 3

[0268] The impact machine according to embodiment 2, wherein said at least one spring action arrangement (**560**) comprises a first spring action member, which first spring action member is prestressed, said first spring action member having a first spring characteristics (k_1) within the interval $k_{trad}/5 \leq k_1 \leq 30^* k_{trad}$.

Embodiment 4

[0269] The impact machine according to embodiment 1, wherein said counterweight (**150**) is arrangeable at a position located between said first counterweight position (CW**1**) and said second counterweight position (CW**2**) from which position said counterweight (**150**) is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said at least one spring-action arrangement (**160**) and wherein said vibration reduction arrangement (**140**) further comprises:

[0270] a first end surface (S_{End1}) arranged adjacent to said first counterweight position (CW**1**), and

[0271] a second end surface (S_{End2}) arranged adjacent to said second counterweight position (CW**2**); and

said at least one motion reversing mechanism comprises a first motion reversing mechanism and a second motion reversing mechanism, and the at least one spring action arrangement of said first motion reversing mechanism is arranged between said counterweight and said first end surface (S_{End1}), and the at least one spring action arrangement of said second motion reversing mechanism is arranged between said counterweight and said second end surface (S_{End2}),

Embodiment 5

[0272] The impact machine according to embodiment 4, wherein said first spring action arrangement is attached to said first end surface (S_{End1}), and said first spring action arrangement is arranged to receive said counterweight when in motion towards said first counterweight position (CW**1**).

Embodiment 6

[0273] The impact machine according to embodiments 4 or 5, wherein said second spring action arrangement (**170**) is attached to said second end surface (S_{End1}), and said first spring action arrangement (**160**) is arranged to receive said counterweight (**150**) when in motion towards said first counterweight position (CW**1**).

Embodiment 7

[0274] The impact machine according to embodiments 4 or 6, wherein said first spring action arrangement (**260**) is attached to said counterweight, and said first end surface is arranged to receive said first spring arrangement when said counterweight is in motion towards said first counterweight position (CW**1**).

Embodiment 8

[0275] The impact machine according to embodiments 4, 5 or 7, wherein said second spring action arrangement is attached to said counterweight, and said second end surface is arranged to receive said second spring arrangement when said counterweight is in motion towards said first counterweight position (CW**1**).

Embodiment 9

[0276] The impact machine according to embodiments 4, 6 or 8, wherein said first spring action arrangement is arrangeable at a position located between counterweight and said first end surface from which position said first spring arrangement is moveable a first distance (D**1**) extending in said first axial direction without actuating said first spring-action arrangement.

Embodiment 10

[0277] The impact machine according to embodiments 5, 7 or 9, wherein said second spring action arrangement is arrangeable at a position located between counterweight and said second end surface from which position said second spring arrangement is moveable a first distance (D**1**) extending in said first axial direction without actuating said second spring-action arrangement.

Embodiment 11

[0278] The impact machine according to any one of the embodiments 4-10, wherein said first spring action arrangement comprises a first spring action member, which first spring action member is biased, and/or wherein said second spring action arrangement comprises a second spring action member, which second spring action member is biased.

Embodiment 12

[0279] The impact machine according embodiment 1, wherein said first spring-action member and a second spring-action member (**272**) are arranged in series or parallel in said first axial direction (A); said first spring-action member (**261**) having a first spring coefficient (k_1) within the interval $-k_{trad} \leq k_1 \leq k_{trad}/2$ and $k_1 \neq 0$, the second spring-action member (**262**) having a second spring coefficient (k_2) within the interval $2^* k_{trad} \leq k_2 \leq 30$ and the second spring member is not prestressed.

Embodiment 13

[0280] The impact machine according to embodiment 1, wherein said first spring-action member and a second spring-action member (**272**) are arranged in series or parallel in said first axial direction (A); said first spring-action member (**261**) having a first spring coefficient (k_1) within the interval $-k_{trad} \leq k_1 \leq k_{trad}/2$ and $k_1 \neq 0$, the second spring-action member (**262**) having a second spring coefficient (k_2) within the interval $k_{trad}/5 \leq k_2 \leq 30$ and the second spring member is prestressed.

Embodiment 14

[0281] The impact machine according to any one of the embodiments 2-11, wherein the spring coefficient k for the spring-action arrangement acting on the counterweight (**150**; **250**) may be determined from the following formula with a deviation from the calculated value of less than 50%, or less than 30%, or less than 20% or less than 10%; and wherein f is the resonance frequency of the impact machine at rated power, k is the spring coefficient, m is the weight of the counterweight (**150**; **250**), D_1 is said first distance and b is the compression distance of the at least one spring arrangement of said first motion reversing mechanism (**160**; **260**).

$$f = \sqrt{\frac{k}{m}} * \left(\frac{b}{2\pi b + 2D_1}\right)$$

Embodiment 15

[0282] The impact machine according to anyone of embodiments 1-11 or 14, wherein the first distance (D_1) is at least at least 20%, or at least 40%, or at least 60% or at least 70% or at least 80% or at least 90% of the distance between the first (CW**1**) and the second (CW**2**) counterweight positions.

Embodiment 16

[0283] The impact machine according to anyone of embodiments 1, 12 or 13, wherein a first spring action member (**261**) and a second spring action member (**262**) arranged in series or in parallel, wherein the first spring coefficient of the first spring-action member (**261**) is lower than the second spring coefficient of the second spring-action member (**262**), and wherein that the first spring coefficient applies to a distance corresponding to at least 10% or at least 15% or at least 20% or at least 25% of the distance between the first CW**1** and the second CW**2** counterweight position; and the second spring coefficient applies to a remaining distance between the first CW**1** and the second counterweight position CW**2**.

Embodiment 17

[0284] The impact machine according to embodiment 16, wherein the first spring coefficient is at least 50% lower than the second spring coefficient.

Embodiment 18

[0285] An impact machine according to anyone of the preceding embodiments, further comprising hammer element guiding means (**120**, **220**, **320**) arranged to cause said hammering element (**110**, **210**, **310**) to move in a linear direction between said first hammering element position (HE**1**) and said second hammering element position (HE**2**).

Embodiment 19

[0286] An impact machine according to anyone of the preceding embodiments, further comprising counterweight guiding means (**381**) arranged to cause said counterweight (**350**) to move in a linear direction between said first counterweight position (CW**1**) and said second counterweight position (CW**2**).

Embodiment 20

[0287] An impact machine according to any one of the preceding embodiments, wherein said impact receiving element (**130**; **230**) is a work tool.

Embodiment 21

[0288] An impact machine according to anyone of the preceding embodiments, wherein said impact machine (**100**; **200**) is handheld.

Embodiment 22

[0289] An impact machine according to anyone of the preceding embodiment, wherein said impact machine is arranged to be attached to a machine, preferably a construction machine such as an excavator, backhoe loader or skid steer loader.

Embodiment 23

[0290] An impact machine according to anyone of the preceding embodiments, wherein the weight of the hammering element H corresponds to between 20% and 300% of the weight m of the counterweight (**150**; **250**).

[0291] The skilled person will realize that the present invention by no means is limited to the described exemplary embodiments. The mere fact that certain measures are recited in mutually different dependent claims does not indicate that a combination of these measures cannot be used to advantage. Moreover, the expression "comprising" does not exclude other elements or steps. Other non-limiting expressions include that "a" or "an" does not exclude a plurality and that a single unit may fulfill the functions of several means. Any reference signs in the claims should not be construed as limiting the scope. Finally, while the invention has been illustrated in detail in the drawings and in the foregoing description, such illustration and description are considered to be illustrative or exemplary and not restrictive; the invention is not limited to the disclosed embodiments.

1. An impact machine comprising:

a housing

a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position (HE**1**) and a second hammering element position (HE**2**),

an impact receiving element attached to said housing,

actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

a vibration reduction arrangement attached to said housing, which comprises:

at least one counterweight distributed around said hammering element and being displaceable in a first axial direction (A) between a respective first counterweight position (CW**1**) and a respective second counterweight position (CW**2**) in response to the hammering action of said hammering element,

a respective first motion reversing mechanism for each one of said at least one counterweight, each respective first motion reversion mechanism comprising a first spring-action arrangement being arranged to reverse the direction of motion of a respective one of said at least one counterweight, wherein

each one of said at least one counterweight is arrangeable at a position located between said respective first counterweight position (CW**1**) and said respective second counterweight position (CW**2**) from which position each one of said at least one counterweight is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said first spring-action arrangement; and wherein

the spring action arrangement of said respective first motion reversing mechanism is arranged inside said respective one of said at least one counterweight,

each of said respective first motion reversing mechanism further comprises a first end surface (S_{END1}) attached to said housing and arranged adjacent to said respective first counterweight position (CW**1**) and

each one of said at least one counterweight comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective spring action arrangement and arranged between said respective spring action arrangement and said first end surface (S_{END1}) in said first axial direction (A) wherein

when any of said at least one counter weight is arranged in said respective first counterweight position:

said engagement surface and said first end surface (S_{END1}) are pressed against each other, and

said at least one spring-action arrangement is actuated.

2. The impact machine according to claim **1**, wherein said vibration reduction arrangement further comprises:

a respective second motion reversing mechanism for each one of said at least one counterweight, each respective second motion reversion mechanism comprising a second spring-action arrangement being arranged to reverse the direction of motion of a respective one of said at least one counterweight, and

the spring action arrangement of said respective second motion reversing mechanism is arranged inside said respective one of said at least one counterweight,

each of said respective second motion reversing mechanism further comprises a second end surface (S_{END2}) attached to said housing and arranged adjacent to said respective second counterweight position (CW**2**) and

each one of said at least one counterweight comprises a second projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective second spring action arrangement and arranged between said respective spring action arrangement and said second end surface (S_{END2}) in said first axial direction (A) wherein

when any of said at least one counterweight is arranged in said respective second counterweight position (CW**2**):

said engagement surface of said second projecting member and said second end surface (S_{END2}) are pressed against each other,

said engagement surface of said second projecting member is displaced relative a center of gravity of said counterweight compared to when said counterweight is arranged in a position where said engagement surface of said second projecting member and said second end surface (S_{END2}) are separated from each other, and

said second spring-action arrangement is actuated.

3. The impact machine according to claim **2**, wherein said spring action arrangement of said first motion reversing mechanism is separated from said spring action arrangement of said second motion reversing mechanism.

4. The impact machine according to claim **2**, wherein said spring action arrangement of said first motion reversing mechanism and said spring action arrangement of said second motion reversing mechanism is one and the same.

5. The impact machine according to claim **1**, wherein said spring action arrangement of said first motion reversing mechanism comprises a first spring action member.

6. The impact machine according to claim **2**, wherein said spring action arrangement of said second motion reversing mechanism comprises a second spring action member.

7. The impact machine according to claim **6**, wherein said spring action member of said first motion reversing mechanism is separated from said spring action member of said second motion reversing mechanism.

8. The impact machine according to claim **6**, wherein said spring action member of said first motion reversing mechanism and said spring action member of said second motion reversing mechanism is one and the same.

9. The impact machine according to claim **5**, wherein said first spring action member is prestressed, and has a first spring characteristics (k_1) within the interval $k_{trad}/5 \leq k_1 \leq 30^*k_{trad}$

10. The impact machine according to claim **6**, wherein said second spring action member is prestressed, and has a first spring characteristics (k_1) within the interval $k_{trad}/5 \leq k_1 \leq 30^*k_{trad}$.

11. An impact machine according to claim **1**, wherein said motion reversion mechanism comprises four spring-action arrangements distributed around said hammering element.

12. An impact machine according to claim **1**, wherein said counterweight comprises two spring-action arrangements inside said counterweight.

13. An impact machine according to claim **12**, wherein said two spring-action arrangements are identical.

14. The impact machine according to claim **1**, wherein said counterweight further comprises restricting means adapted to restrict the movement of said projecting member in the first axial direction (A) and/or in a direction opposite thereto.

15. The impact machine according to claim **14**, wherein said restricting means comprises at least one first retaining surface attached to said counterweight, and said projecting member further comprises at least one flange, wherein said retaining surface restricts the motion of said flange in the first axial direction (A) and/or in a direction opposite thereto.

16. The impact machine according to claim **2**, wherein

said counterweight further comprises restricting means adapted to restrict the movement of said projecting member in the first axial direction (A) and/or in a direction opposite thereto,

said restricting means comprises at least one first retaining surface attached to said counterweight, and said projecting member further comprises at least one flange,

said retaining surface restricts the motion of said flange in the first axial direction (A) and/or in a direction opposite thereto,

said restricting means further comprises a second retaining surface adapted to restrict the movement of said second projecting member in said second axial direction and/or in a direction opposite thereto, and

said spring action member is biased by said first retaining surface and said second retaining surface.

17. An impact machine according to claim **1**, wherein said vibration reduction arrangement is arranged around said housing, such that said at least one counterweight is rotatable about a central longitudinal axis of said housing, coaxial with said first axial direction (A).

18. An impact machine according to claim **1**, further comprising counterweight guiding means arranged to cause said counterweight to move in a linear direction between said first counterweight position (CW**1**) and said second counterweight position (CW**2**).

19. An impact machine according to claim **1**, wherein when said at least one counterweight is only one counterweight, said counterweight fully surrounds said hammering element.

20. An impact machine according to claim **1**, wherein when said at least one counterweight comprises of two or more counterweights, said counterweights are evenly distributed around said hammering element.

21. An impact machine according to according to claim **19**, wherein said counterweight comprises an outer truncated elliptical cross-section which is perpendicular to said first axial direction.

22. The impact machine according to claim **1**, wherein said at least one spring-action arrangement further comprises a first spring-action member and a second spring-action member arranged in parallel in said first axial direction (A).

23. The impact machine according to claim **1**, wherein the first distance (D_1) is at least 20%, or at least 40%, or at least 60% or at least 70% or at least 80% of the distance between the first (CW**1**) and the second (CW**2**) counterweight positions.

24. The impact machine according to claim **22**, wherein a first spring action member and a second spring action member are arranged in parallel, wherein said first spring coefficient of said first spring-action member is lower than said second spring coefficient of said second spring-action member, and wherein said first spring coefficient applies to a distance corresponding to at least 10% or at least 15% or at least 20% or at least 25% of a distance between said first (CW**1**) and said second (CW**2**) counterweight position; and said second spring coefficient applies to a remaining distance between said first (CW**1**) and said second counterweight position (CW**2**).

25. An impact machine according to claim **1**, further comprising hammer element guiding means arranged to cause said hammering element to move in a linear direction between said first hammering element position (HE**1**) and said second hammering element position (HE**2**).

26. An impact machine according to claim **1**, wherein said impact receiving element is a work tool.

27. An impact machine according to claim **1**, wherein said impact machine is handheld.

28. An impact machine according to claim **1**, wherein said impact machine is arranged to be attached to a machine, preferably a construction machine such as an excavator, backhoe loader or skid steer loader.

29. An impact machine according to claim **1**, wherein the weight of the hammering element H corresponds to between 20% and 300% of the weight m of the counterweight.

30. An impact machine comprising:

a housing

a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position (HE**1**) and a second hammering element position (HE**2**),

an impact receiving element attached to said housing,

actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

a vibration reduction arrangement attached to said housing, which comprises:

a counterweight distributed around said hammering element and being displaceable in a first axial direction (A) between a first counterweight position (CW**1**) and a second counterweight position (CW**2**) in response to the hammering action of said hammering element,

a first motion reversing mechanism comprising a first spring-action arrangement being arranged to reverse the direction of motion of said counterweight,

wherein

said counterweight is arrangeable at a position located between said first counterweight position (CW**1**) and said second counterweight position (CW**2**) from which position said counterweight is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said spring-action arrangement; and wherein

the spring action arrangement of said motion reversing mechanism is arranged inside said counterweight,

said first motion reversing mechanism further comprises a first end surface (S_{END1}) attached to said housing and arranged adjacent to said first counterweight position (CW**1**) and

said counterweight comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said spring action arrangement and arranged between said spring action arrangement and said first end surface (S_{END1}) in said first axial direction (A) wherein

when said counter weight is arranged in said first counterweight position:

said engagement surface and said first end surface (S_{END1}) are pressed against each other, and

said at least one spring-action arrangement is actuated.

31. An impact machine comprising:

a housing

a hammering element arranged inside said housing, said hammering element is displaceable between a first hammering element position (HE**1**) and a second hammering element position (HE**2**),

an impact receiving element attached to said housing,

actuating means arranged to cause said hammering element to perform a hammering operation on said impact receiving element,

a vibration reduction arrangement attached to said housing, which comprises:

a first number of counterweights arranged evenly distributed around said hammering element, each counterweight being displaceable in a first axial direction (A) between a respective first counterweight position (CW**1**) and a respective second counterweight position (CW**2**) in response to the hammering action of said hammering element,

a first number of motion reversing mechanisms, each comprising a first spring-action arrangement being arranged to reverse the direction of motion of a respective one of said first number of counterweights,

wherein

said each one of said first number of counterweights is arrangeable at a position located between said respective first counterweight position (CW**1**) and said respective second counterweight position (CW**2**) from which position each one of said at least one counterweight is moveable a first distance (D**1**) extending in said first axial direction (A) without actuating said at least one spring-action arrangement; and wherein

said first spring action arrangement of each first motion reversing mechanism is arranged inside said respective one of said first number of counterweights,

each of said respective first motion reversing mechanism further comprises a respective first end surface (S_{END1}) attached to said housing and arranged adjacent to said respective first counterweight position (CW**1**) and

each one of said at least first number of counterweights comprises a first projecting member, which projecting member comprises an engaging surface, which engaging surface is connected to said respective spring action arrangement and arranged between said respective spring action arrangement and said first end surface (S_{END1}) in said first axial direction (A), wherein

when any one of said first number of counter weights is arranged in said respective first counterweight position:

said engagement surface of said counterweight and said respective first end surface (S_{END1}) are pressed against each other,

said engagement surface is displaced relative a center of gravity of said counterweight compared to when said counterweight is arranged in a position where said engagement surface and said first end surface (S_{END1}) are separated from each other, and

said at least one spring-action arrangement is actuated.

* * * * *

Paper B

Nonlinear Dynamic Vibration Absorber to Reduce Vibration in Hand-held Impact Machines.

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NONLINEAR DYNAMIC ABSORBER TO REDUCE VIBRATION IN HAND-HELD IMPACT MACHINES

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Abstract. *Hand-held impact machines have been used since the early 20th century. Up to day still a little has been changed in their fundamental design. Despite them being robust and efficient, vibration, noise, dust and poor ergonomics cause a large number of injuries to the operators, especially in the stone industry. In this paper, vibration dynamics of a pneumatic hand-held impact machine (HHIM) equipped with a nonlinear tuned vibration absorber (NLTVA) is in focus. The considered HHIM has the same parameters as the existing conventional one with respect to piston weight, impact energy and operating frequency. The paper presents the mathematical and computational models of vibration dynamics of the HHIM with NLTVA as well as the experimental set-up developed for model validation. The advantage of using a nonlinear spring is shown and how the effective vibration reduction frequency of the NLTVA can be made to follow the excitation frequency and thereby in principal achieving the same conditions as is the case for tuned pendulum absorbers for torsional vibrations where the resonance frequency follows the engine order. Using the verified and validated computational model, the sensitivity analysis of vibration dynamics has been performed in a broad set of feasible operational scenarios with respect to structural parameters of the HHIM equipped with NLTVA. Sensitivity analysis results have formed the base for optimisation of the parameters of NLTVA. It was shown by simulations as well as it was proved by experiment that the vibration of the HHIM equipped with NLTVA can be reduced significantly in a broader range of operating frequencies compared to the vibration of the same machine equipped with linear tuned vibration absorber. The results obtained confirm the possibility to design a user friendly low vibration impact machine efficiently operating in a broad frequency range. Optimised NLTVA combined with vibration isolation has shown to significantly reduce the vibration on the operator.*

1 INTRODUCTION

Vibration exposure from hand-held impact machines (HHIM) such as rock drills, rammers and breakers with a reciprocating action is a major cause for injuries to the workers in the industry. In order to improve the work environment in the stone industry, a project was started with the objectives to redesign the tools to achieve low vibration as well as improved ergonomics, dust removal and reduced noise while maintaining productivity [1]. Redesign of current hand-held pneumatic impact machines can reduce the vibration level and thereby reduce injuries to workers. Hand-arm vibration injury, often called Hand-Arm Vibration Syndrome (HAVS), is one of the most common reasons for work related injuries among this group of workers in the industry.

Although impact machines have been used since the early 20th century, little has been changed in their fundamental design to date. Despite them being robust and efficient, vibration, noise, dust and poor ergonomics cause a large number of injuries to the operators. Previous work has been done to reduce vibrations from these machines, some of which have been patented [2, 3]. One approach is to use traditional linear tuned vibration absorbers (TVA) invented in 1909 by Frahm and described by Den Hartog [4]. However, this technology is to a large extent limited in practical use on this application since it is only effective in a narrow frequency range. At higher frequencies, the TVA will instead increase the vibration and at lower frequencies will the effect rapidly decrease.

A thorough analysis of the dynamics of the HHIM together with an implementation of a TVA in the handle of the machine in combination with a vibration isolated handle is described in [5]. More general aspects of the NLTVA are analysed in [6] which show how different parameters influence the stability of the overall system.

However, by introducing nonlinear spring characteristic of the auxiliary mass, the effective frequency range can be greatly increased and thereby can the technology be effectively implemented to this kind of machines [7]. As a result, a new generation of impact machines is developed by approaching the redesign from a user perspective, and starting adhering to strict conditions of low vibration, noise and dust as well as sound ergonomics.

The objective of this study has therefore been to develop a user friendly low vibration impact machine using NLTVA together with integrated vibration isolation. A HHIM with a NLTVA combined with vibration isolation has shown to significantly reduce the vibration on the operator from 20 m/s^2_{haw} to 2.7 m/s^2_{haw} [1].

2 ENGINEERING MODEL OF HHIM

The pneumatic HHIM in question consists of the following machine functional components (MFCs): the housing of mass m_h, the main mass m_m, the piston of mass m_{pist} moving in the cylinder, the NLTVA with auxiliary mass m_a, the chisel, and the handle. The detailed engineering model of the HHIM is depicted in Figure 1. It is assumed that the MFCs are under the action of the external loads modelled by the exciting force F_e applied to the main mass and the internal loads modelled by the forces exerted by springs and dampers. The effective mass of the hand-arm system is quite small and is neglected in this study. The following additional notations are in use: k_m, k_a, k_h, k_p are the stiffness between main mass and ground, main mass and auxiliary mass, main mass and housing, and the hand-arm stiffness representing the operator, respectively; c_m, c_a, c_h, c_p are the damping between main mass and ground, main mass and auxiliary mass, main mass and housing, and the hand-arm damping representing the operator, respectively; a and F_0 are the half gap length and the spring preload, respectively, characterising the TVA nonlinearity.

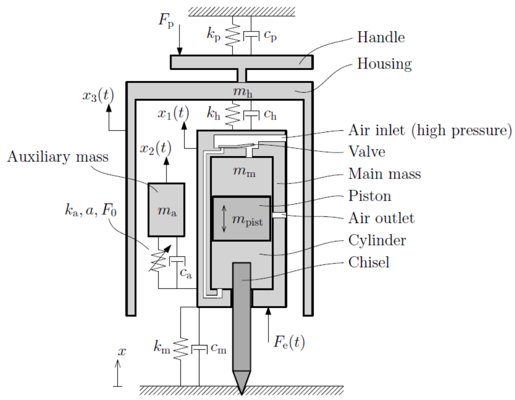

Figure 1: Engineering model of the HHIM.

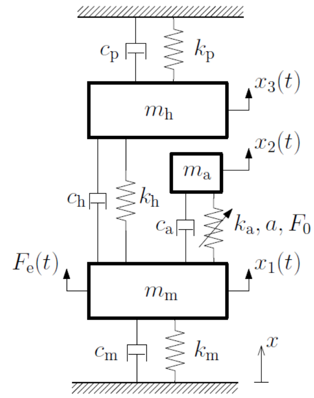

Figure 2: Sketch of the HHIM as 3-DoF vibrating system.

Parameter values used in the modelling are introduced in Table 1, except for k_a, a, F_0 which are subject to optimisation. All values, except the three masses and k_h, have been estimated. For example, hand-arm stiffness k_p was taken from [8, 9] and hand-arm damping c_p was taken from [9]. Auxiliary damping c_a is kept as low as possible in the HHIM for maximum performance of the NLTVA. It is therefore given a low value in the model. It has been noted that all parameters pertinent to the coupling between main mass and housing as well as the couplings to ground have a minor influence on system response for excitations near nominal operating frequency since that is well above the first two resonance frequencies of the system [10]. Amplitude of the exciting force F_e in a HHIM prototype has previously been measured to be 351 N at an operating frequency of 26.1 Hz [10], which is utilised in this model. It was also concluded that a sinusoidal force is a good approximation, and its amplitude is assumed to increase quadratic with operating frequency since the displacement of the piston is constant.

	Masses [kg]			Stiffness coefficients [kN/m]			Viscous damping coefficients [Ns/m]			
Parameter	m_m	m_a	m_h	k_m	k_h	k_p	c_m	c_a	c_h	c_p
Nominal value	2.7	1	3.1	500	14000	1000	100	1	20	60

Table 1: Model parameters.

3 MATHEMATICAL AND COMPUTATIONALTIONAL MODELS

With described engineering model, the HHIM is considered as a vibrating system with three degrees of freedom, the sketch of which is depicted in Figure 2. By introducing generalized coordinates x_1, x_2, x_3 the equations of motion of the system can be written as follows

$$m_m\ddot{x}_1 + (c_m + c_h)\dot{x}_1 - c_h\dot{x}_3 + (k_m + k_h)x_1 - k_h x_3 = F_e(t) + F_k(\mathbf{x}) + F_c(\mathbf{x},\dot{\mathbf{x}}) - m_m g \tag{1}$$

$$m_a\ddot{x}_2 = -F_k(\mathbf{x}) - F_c(\mathbf{x},\dot{\mathbf{x}}) - m_a g \tag{2}$$

$$m_h\ddot{x}_3 - c_h\dot{x}_1 + (c_h + c_p)\dot{x}_3 - k_h x_1 + (\mathrm{k}_h + k_p)x_3 = -m_h g \tag{3}$$

Here in Eqs. (1-3), $\mathbf{x} = [x_1, x_2, x_3]^T$, $\dot{\mathbf{x}} = [\dot{x}_1, \dot{x}_2, \dot{x}_3]^T$ are the vectors of generalized coordinates and velocities, respectively. $F_k(\mathbf{x})$ and $F_c(\mathbf{x},\dot{\mathbf{x}})$ are nonlinear spring force and damping force exerted by the NLTVA and applied to the main mass. g is acceleration due to gravity.

Different expressions for $F_k(\mathbf{x})$ and $F_c(\mathbf{x},\dot{\mathbf{x}})$ can be chosen and thus different concepts of the TVA can be studied. The following representations for these forces are proposed

$$F_k(\mathbf{x}) = \begin{cases} F_0 + k_\mathrm{a}(x_2 - x_1 - a) & \text{if } x_2 - x_1 > a \\ -F_0 - k_\mathrm{a}(-x_2 + x_1 - a) & \text{if } x_2 - x_1 < -a \\ 0 & \text{else} \end{cases} \tag{4}$$

$$F_c(\mathbf{x},\dot{\mathbf{x}}) = c_\mathrm{a}(\dot{x}_2 - \dot{x}_1) \tag{5}$$

Here in the expressions (4) and (5), the parameters a, F_0, k_a, and c_a are the half gap length, the spring preload, the stiffness between main mass and auxiliary mass and the damping between main mass and auxiliary mass of the NLTVA.

The following direct dynamics problem for the HHIM is formulated.

Problem A. Let the engineering model of the HHIM, the exciting force applied to the main mass $F_\mathrm{e}(t)$, $t \in [0, t_\mathrm{f}]$ and the internal loads modelled by the forces exerted by springs and dampers of the NLTVA $F_k(\mathbf{x})$ and $F_c(\mathbf{x},\dot{\mathbf{x}})$ be given. It is required to determine the vectors of generalized coordinates $\mathbf{x} = [x_1, x_2, x_3]^\mathrm{T}$ and generalized velocities $\dot{\mathbf{x}} = [\dot{x}_1, \dot{x}_2, \dot{x}_3]^\mathrm{T}$ of the steady-state performance of the HHIM that satisfy the equations of motion Eqs. (1-3) and arbitrary prescribed initial conditions $\mathbf{x}(0) = \mathbf{x}_0$, $\dot{\mathbf{x}}(0) = \dot{\mathbf{x}}_0$. Note that the final state of the machine as well as the final time instant are not prescribed, i.e. the components of the vectors $\mathbf{x}(t_\mathrm{f}), \dot{\mathbf{x}}(t_\mathrm{f})$ and the final time instant t_f are free. All structural parameters of the machine including parameters of the NLTVA are assumed to be given.

In order to solve *Problem A*, a computational model of the HHIM in question has been developed and implemented in MATLAB©. The core of the computational model is the numerical algorithm of the solution of the initial value problem for Eqs. (1-3) based on MATLAB function *ode45* with the value of both the relative and absolute error tolerances equal to 10^{-8}.

4 MODEL VERIFICATION AND VALIDATION

4.1 Model verification

To ensure correctness and accuracy of numerical solutions obtained from the computational model, a verification of the model is done by comparing its results with reference results.

Firstly, the modelled system was configured as a linear system by setting both the gap a and preload F_0 in Eq. (4) to zero. Perfect correlation was seen between the numerical solution and the analytical solution of the same system, with respect to displacements, velocities, accelerations and phase angles. Secondly, simulations were run for the unforced undamped auxiliary system, i.e. the auxiliary mass restricted by the nonlinear spring described by Eq. (4), initiated from a spring compression b. Since that system is not dissipative, the auxiliary mass should oscillate with the same amplitude indefinitely. The said system was simulation for 20 seconds for several different configurations of the parameters k_a, a, F_0 and b. The undamped resonance frequency has been derived analytically in [10] and is given by

$$f_\mathrm{res} = \sqrt{\frac{k_\mathrm{a}}{m_\mathrm{a}}}\left[2\pi - 4\arcsin\left(\frac{F_0}{F_0 + k_\mathrm{a}b}\right) + 4a\sqrt{\frac{k_\mathrm{a}}{k_\mathrm{a}b^2 + 2F_0 b}}\right]^{-1} \tag{6}$$

4.2 Model validation

Validations of the computational models have been made in two experimental set-ups, a dedicated test rig [11] and in a pneumatic HHIM prototype. In the test rig, all relevant parameters could be controlled and measured and most important, the excitation frequency could be varied from stand still up to 18 Hz. The test rig employed the TVA nonlinearity created by a gap but no preload in this test. In the HHIM prototype on the other hand, all parameter were close to a real HHIM although the exciting force frequency could not be varied easily and parameters as the stiffness to ground, damping and exciting force needed to be estimated or measured indirectly.

Validation in test rig

In the test rig, the force is generated from an electric motor to a pneumatic piston connected to a force transducer that is attached to the main mass, Figure 3. The force can be varied to frequency by the motor speed and to amplitude by restricting the air openings in the pneumatic cylinder or changing the stroke length. Displacement of the main and auxiliary mass is measured by laser displacement sensors. In the experiments, the force is increasing linearly with a peak of 45 N at the frequency 18 Hz. The set-up of the system is chosen so that the main mass has a resonance frequency at 2 Hz and the auxiliary mass is tuned to give the lowest vibration of the main mass at 9 Hz.

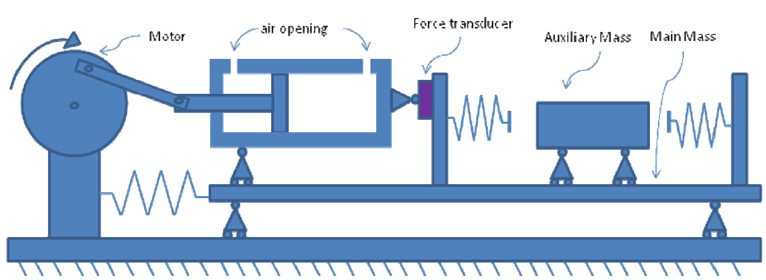

Figure 3: Test rig.

The results from the experiments compared to the simulations are shown in Figure 4. It is clearly seen that there is a substantial reduction in vibration amplitude of the main mass when the NLTVA is active in a broad frequency range. The deviations are mainly believed to be due to that the experimental exciting force is not purely sinusoidal but has a substantial degree of harmonic content. There are also uncertainties in the spring force for the auxiliary mass since it was compressed partly beyond its specifications. The exciting force in the simulation model was purely sinusoidal.

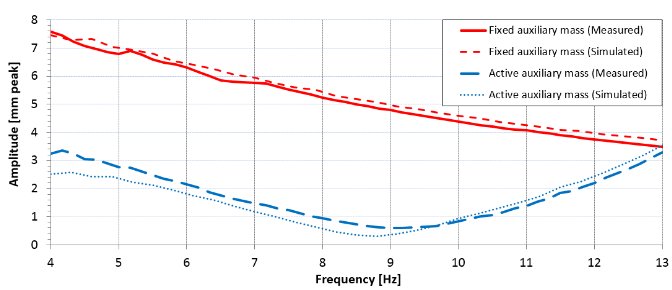

Figure 4: Experimental and simulated results of main mass vibration for active and inactive auxiliary mass.

Validation using the prototype

Since the working frequency and the amplitude of the exciting force in the prototype cannot be varied, gap length a was varied to see if the model would predict the changes in reality. To be able to measure the motion of the auxiliary mass, the housing was removed during measurements and simulations. Since the operating frequency varied slightly between measurements, the frequency from each measurement was used as the excitation frequency in respective simulation. The gap lengths used were 3.2, 6.1 and 9.7 mm and corresponding measured operating frequencies were 26.5, 25.8 and 25.4 Hz, respectively. The simulated and measured root mean square (RMS) displacements for different gap lengths are compared in Figure 5. RMS response of the experimental data was calculated after applying a 5th order high-pass Butterworth filter with a cutoff frequency of 6.5 Hz. The comparison shows very similar results, especially when taking into account that many of the parameters in the prototype cannot be measured very accurately. The model predicts the change in vibration level of both the auxiliary and main mass well.

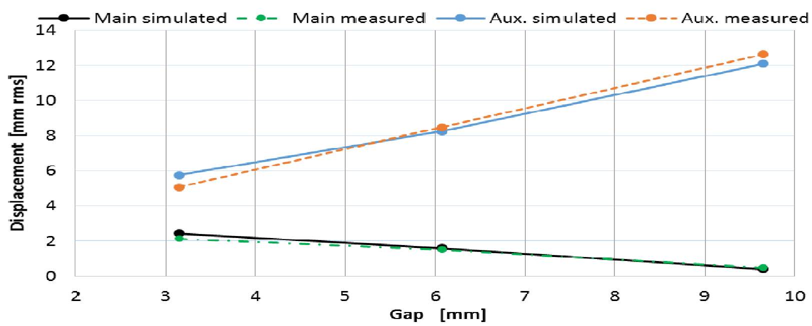

Figure 5: Experimental and simulation results for the prototype when the gap length is varied.

5 SENSITIVITY ANALYSIS AND OPTIMISATION

To investigate how the three NLTVA parameters k_a, a and F_0 that are subject to optimisation affects the vibration suppression, each parameter was varied around its nominal value. The results from varying single parameters separately can be seen in Figure 6.

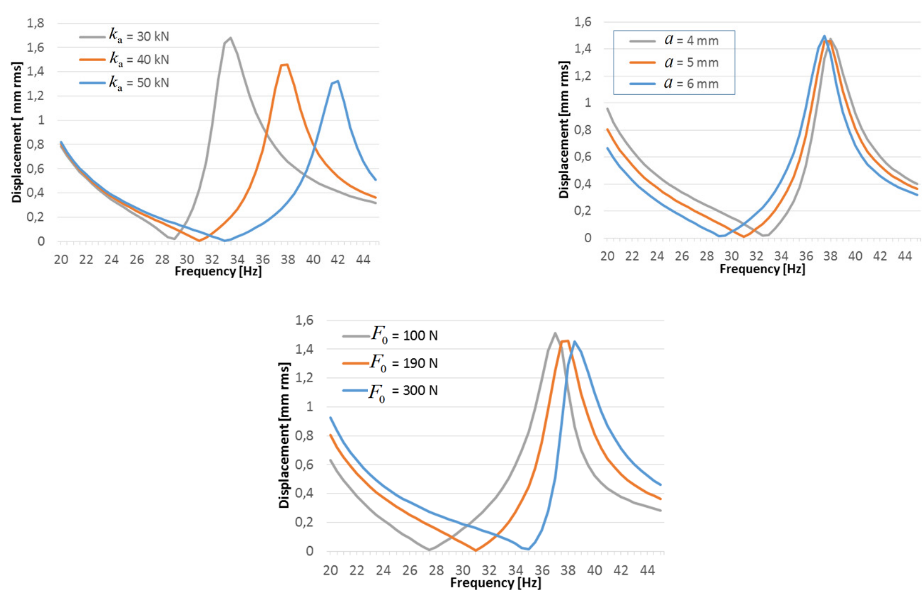

Figure 6: Vibration of the handle for different values of the NLTVA parameters. Only one parameter is varied at a time while other parameters are kept at their nominal values; k_a = 40 kN/m, a = 5 mm, F_0 = 190 N.

Figure 6 clearly shows that any of the NLTVA parameters can be adjusted to change the frequency that has maximum suppression, i.e. the tuned frequency. When the stiffness k_a is increased, the tuned frequency moves upwards and vice versa if the stiffness is lowered. The unwanted resonance above the tuned frequency is also moved considerably upwards in frequency when the stiffness is increased, and vice versa when the stiffness is decreased. The resonance is however not affected significantly when the gap length a and preload F_0 are changed. This suggests that an optimal result would be a high stiffness k_a to move the unwanted resonance to a higher frequency, combined with a lowered preload F_0 and/or an increased gap length a to keep the minimum, i.e. the tuned frequency, around 28 Hz.

The following optimisation problem is formulated in order to optimise the proposed NLTVA in the considered HHIM:

$$\left(k_{\mathrm{a}}, a, F_0\right)_{\mathrm{optim}} = \underset{k_{\mathrm{a}}, a, F_0}{\operatorname{argmin}}\ L\left(k_{\mathrm{a}}, a, F_0\right) \tag{7}$$

Here in Eq. (7), L is the objective function and k_a, a and F_0 are the NLTVA parameters subject to optimisation. The objective function is defined as the area under a weighted response curve in a frequency range that spans ±15 % of the nominal operating frequency, 28 Hz, to account for variations.

$$L(k_{\mathrm{a}}, a, F_0) = \int_{23.8\ \mathrm{Hz}}^{32.2\ \mathrm{Hz}} \dot{x}_{3,\mathrm{RMS}}(f, k_{\mathrm{a}}, a, F) \cdot W(f)\ \mathrm{d}f \tag{8}$$

In Eq. (8), $\dot{x}_{3,\mathrm{RMS}}$ is the velocity RMS response at the handles, computed by the numerical model, and $W(f)$ is a weighting function that provides priority of minimisation at the nominal operating frequency, 28 Hz, but still lets the response at the frequency bounds 23.8 Hz and 32.2 Hz be accounted for. The weighting function is a normal distribution centred at the nominal frequency with a standard deviation of 2.83 Hz. Choosing this standard deviation makes the weighting function being three times as large at the nominal frequency as at the frequency bounds. In the optimisation routine, the frequency range of interest, $f \in [23.8,\ 32.2]\ \mathrm{Hz}$, is discretised into 15 points such that the density of points is proportional to the weighting function $W(f)$. The objective function is computed by summing the velocity RMS responses evaluated at the discrete frequencies. The minimisation problem in Eq. (7) is solved in MATLAB with *lsqnonlin*, a nonlinear least squares solver implementing the trust-region-reflective method. During optimisation, all parameters are restricted by a lower bound of zero and upper bounds which are 500 kN/m for stiffness k_a, 40 mm for gap a and 500 N for preload F_0. To increase the chance of reaching the global optimum sought in Eq. (7), multiple starts are run, each with an initial guess randomised uniformly within the parameter bounds.

Optimisation results

The optimisation has been run using over 250 randomised starts. About half of the optimisation runs resulted in parameters that gave very low objective function. The parameters that gave the lowest found objective function value were k_a = 164.23 kN, a = 10.21 mm and F_0 = 98.2 N. The parameters are as expected considering the findings of the sensitivity study. The stiffness is very high, which places the unwanted resonance frequency far above the operating frequency, and the increased gap length a moves the tuned frequency down to 28 Hz. Figure 7 compares the response of the handle between using the optimised NLTVA and the traditional linear TVA. The suppression band with the optimised NLTVA is much wider than with the linear TVA, clearly demonstrating the advantage of using a NLTVA.

It is thus shown that there is a substantial advantage in using a nonlinear spring characteristic instead of the traditional linear spring for the auxiliary mass in order to achieve a broader useful frequency range and thereby a more robust system. The core reason for the improvement can be explained to a large extent by that the phase of the auxiliary mass remains close to 180° in relation to the exciting force over a very broad frequency range, Figure 8 and Figure 9. The counterphase of the auxiliary mass is maintained because the resonance frequency of the auxiliary system to a large extent follows the increasing excitation frequency, Figure 10. The reason for this in turn is that the displacement amplitude of the auxiliary mass increases with higher excitation frequencies, which is also seen in Figure 10, and, since the auxiliary spring characteristic from the optimised k_a , a and F_0 is nonlinear stiffening, the resonance frequency of the auxiliary system will then increase.

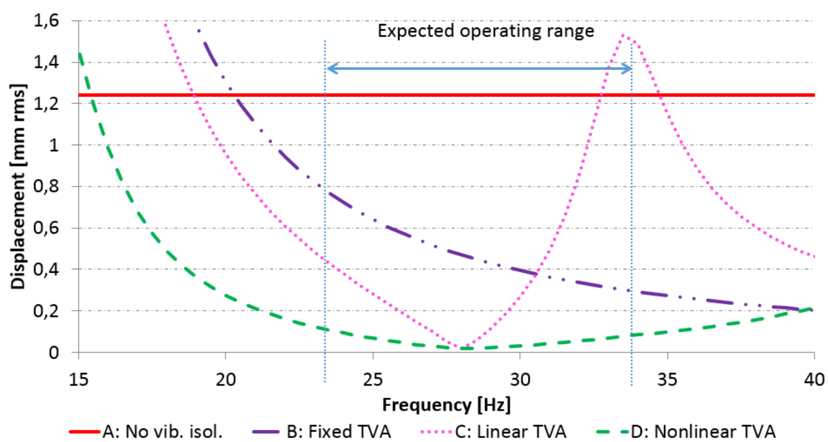

Figure 7: Vibration of the handle. Comparison between using the optimised NLTVA, the LTVA tuned to 28 Hz, deactivated TVA (vibration isolation only) as well as no means of vibration reduction at all.

In Figure 8 and the close-up in Figure 9, the phase and amplitude are compared between using a linear and a nonlinear TVA that are both optimised to reduce vibrations in the area around 28 Hz of the system.

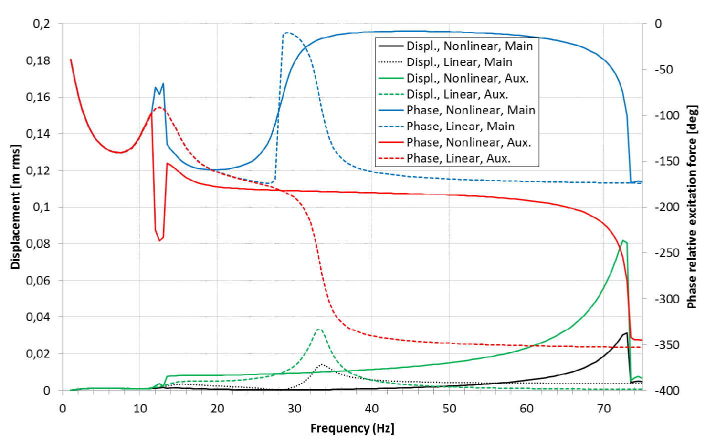

Figure 8: Phase and displacement for NLTVA and LTVA.

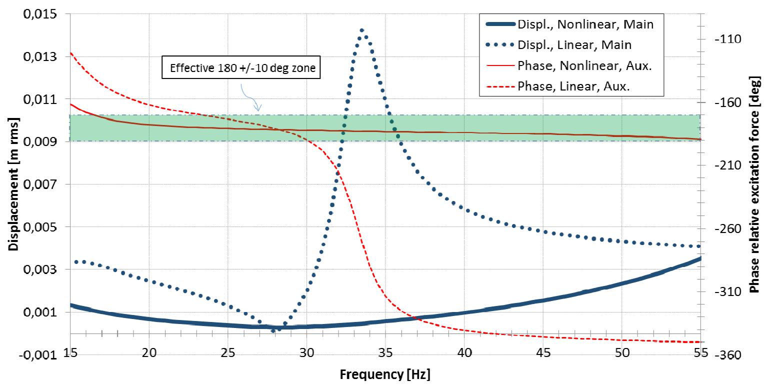

Figure 9: Phase and displacement for NLTVA and LTVA, close-up.

What can be seen from these figures is that the linear auxiliary mass is only in counterphase to the excitation up to about 30 Hz. Just above 30 Hz, there is a phase shift and a severe amplification of the vibration. In the nonlinear system on the other hand, the auxiliary mass maintains the counterphase and reduces vibration even up to about 60 Hz. The resonance introduced by adding the TVA to the system is considerably far above and on a safe distance from the tuned frequency in the nonlinear case, making the effectiveness of the NLTVA very insensitive to varying excitation frequency.

In Figure 10, it is shown how the resonance frequency of the auxiliary mass is increasing with increased excitation frequency. The analytical resonance frequency is given by Eq. (6) and is a system where the exciting force is increasing quadratic with increased frequency. This condition is valid for most mechanical systems which vibrations are mass controlled and where there is a constant displacement of moving components over the operating frequency.

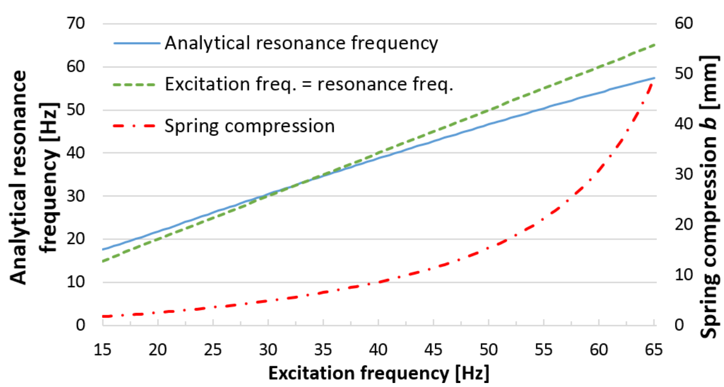

Figure 10: Analytical resonance frequency and spring compression versus excitation frequency.

The relation that the resonance frequency is increasing with excitation frequency can also be found in the closely related “sister” technology with tuned pendulum torsion absorbers [4]. The tuned pendulums are used to create torque on a shaft to reduce torsional vibrations, e.g. in the shaft from a combustion engine. In the tuned pendulum system, the resonance frequency is proportional to the rotating speed which means that the resonance frequency will follow the engine order and thereby have an effective vibration reduction throughout the whole speed range.

6 CONCLUSIONS

A 3-DoF computational model for a pneumatic HHIM utilising a NLTVA in combination with vibration isolation was developed and implemented in MATLAB. The numerically modelled HHIM is excited by a harmonic force exerted by a piston reciprocating with constant amplitude. The model was verified against analytical results and validated against experimental data from a dedicated test rig as well as from a HHIM prototype.

Sensitivity analysis and optimisation of the three NLTVA parameters; stiffness, gap and spring preload, have performed based on the numerical model. The obtained results showed that, for the considered HHIM, the optimised NLTVA is superior to a linear TVA tuned to the same frequency, especially in terms of its insensitivity to variations in operating frequency. In combination with vibration isolation, the optimised NLTVA can significantly and reliably reduce vibration on the operator in a broad range of operating frequencies.

ACKNOWLEDGEMENTS

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Paper C

High Frequency Shock Vibrations and implications of ISO 5349 - Measurement of vibration, Simulating Pressure Propagation, Risk Assessment and Preventive Measures

Report from Workshop on Single Shocks at 13:th Int. Conf. on Hand-Arm Vibration, Oct. 16, 2015, Beijing, China

High Frequency Shock Vibrations and Implications of ISO 5349

- Measurement of Vibration, Simulating Pressure Propagation, Risk Assessment and Preventive Measures

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Summary

This report covers the material presented at the Workshop on Single Shocks at 13:th Int. Conf. on Hand-Arm Vibration, Oct. 16, 2015, Bejing, China. Parts of the content have been updated with recent findings from research at Swerea, when applicable.
High frequency shock vibrations above 1250 Hz are likely to cause a significant amount of vibration injuries [1 - 6], but there is inadequate understanding of the injury mechanisms. There is also no standard assessing the risk associated with these vibrations. The current standard, ISO 5349, that all regulations and legislation are based upon such as the EU Vibration Directive covers only vibrations up to 1250 Hz. Frequencies above this are not considered at all. This results in that several occupational groups are exposed to potential harmful vibrations that are not regulated by any worker's protection directives. Examples of major occupational groups exposed to high frequency vibrations are users of impact wrenches for assembly and repair of vehicles and personnel in the dental sector.

The objectives with this study are; **first**, to measure and compare vibrations from impact tools with tools with a continuous vibration and study how the weighting filter in ISO 5349 affects the acceleration; **second**, study how high frequency impact vibration is transmitted into the finger tissue via the skin by using a FE-model; **third**, a brief review of literature on how high frequency vibrations affect red blood cells and Hand-Arm Vibration Syndrome (HAVS) prevalence and; **fourth**, shown that the high frequency vibrations can be significantly reduced by redesign in common machines.

The conclusions are:

- Acceleration can be measured accurately up 50 kHz with newly developed ultra light MEMS accelerometers.
- Impact machines generate high amplitude high frequency transient accelerations.
- High frequency accelerations from impact machines are nearly eliminated by the weighting filter in ISO 5349 and thereby disclosed from risk evaluation.
- Transient vibrations from machines generate a shock wave that propagates into the finger tissue.
- The epidermis layer of the finger has a relatively small attenuation.
- There are several studies that show that ISO 5349 underestimates the risk for HAVS from transient vibrations.
- Transient vibrations can be substantially reduced by redesign of machines.
- ISO 5349 cannot be used for estimation of injuries from High Frequency Vibrations (HFV) and was never intended to which is clearly stated in the scope.

- There is a need for an amendment to ISO 5349 covering HFV to create an incentive for tool manufacturers and users to reduce the vibration levels.

The intent is that with an increased understanding on how high frequency vibrations from machines are interacting with biological tissue will emphasize the need to regulate these and establish a standard for these vibrations and thereby create an incentive for machine producers and users to reduce the high frequency vibrations.

1 Introduction

High frequency vibration of 1250 Hz is likely to cause a significant part of vibration injuries [1 - 6] but there is a lack of understanding of the injury mechanisms as well a functioning standard for measurement of high frequency vibrations. Today's standard, ISO 5349, that all regulations and legislation are based on only covers vibrations up to 1250 Hz. Frequencies above this is not considered at all. This means that major occupational groups are exposed to harmful vibrations that are neither measured nor regulated by either the EU Directives or the Work Environment Authority. Examples of major occupational groups exposed to high frequency vibrations are users of impact wrenches at assembly and repair of vehicles and personnel in dental industry.
For vehicle repairer workers, the impact wrenches are one of their most used tools which emit high impact vibrations. An impact wrench has typically an impact blow rate of 20 Hz which is well within the ISO 5349 range, but each stroke creates a pulse of high frequency vibrations with very high acceleration levels of several thousands of m/s^2. Since the pulse of high frequency vibration is short these vibrations are often called for transient. Several medical studies [1; 2] have since long pointed out that ISO 5349 greatly underestimates the risk of Hand Arm Vibrations Syndrome (HAVS) at workers subjected to transient vibrations.
Dentists, dental technicians and dental hygienists work with tools that expose them to high frequency vibrations. A dental drill typically rotates up to 400 000 rpm i.e. vibrates with a frequency of about 7000 Hz. In a recent article [6] it was revealed that the high-frequency vibrations from the dental drills often cause vibration injuries to the dentists who have worked with repair of teeth in spite of that the ISO 5349 vibration being almost zero.
There have also been studies on how transient vibrations affect red blood cells [4; 5] and rat tails [3]. The results have shown that there is a very large negative impact on cells and tissues.

The authors of ISO 5349-1:2001, have clearly been aware of the problem with high frequency transient vibrations which is reflected in the scope where it says that it only covers vibrations within the frequency range of the octave bands from 8 Hz to 1000 Hz. It also states in the Scope: **"Provisionally, this part of ISO 5349 is also applicable to repeated shock type excitation (impact)"** and further **"The time dependence for human response to repeated shocks is not fully known. Application of this part of ISO 5349 for such vibration is to be made with caution"**. There is no doubt that the effects from high frequency transient vibrations are not fully covered by ISO 5349 but it still forms the basis for risk evaluation on these machine categories.

The objective for this study was to describe how the vibrating surface of hand-held tools affects the finger tissue with respect to pressure wave propagation. This was done by developing a computational model implemented in LS-Dyna. The acceleration input parameters were taken from measurement on a hand-held, impact wrench.

2 Study of “transient” and “continuous” acceleration above 1250 Hz

2.1 Measuring high frequency and high amplitude acceleration

Resent development of accelerometer technology with Micro Electrical Mechanical Systems (MEMS) has open new possibilities to study high frequency vibrations from hand-held machines. The advantage is that the weight of the sensor can be greatly reduced which ease the mounting and allows measurement on polymer handles. The low mass allows it to be used on low mass structures without the loading effects on vibration properties of small structures. In this study the accelerations were measured with a piezoresistive bridge shock MEMS accelerometer, model 3501A2060KG, from PCB. It weighs 0.15 gram and has a 2 dB frequency range at 50 kHz and an amplitude range of 600,000 m/s^2. The resonance frequency of the accelerometer is 150 kHz. The acceleration signal was anti-alias filtered with an analog 4:th order low pass Bessel filter at 200 kHz and then sampled in the AD converter at 1 MHz. The signal was then digitally low pass filtered at 30 kHz with a 6:th order Bessel low pass

Figure 1 MEMS accelerometer

filter. The filter frequency of 30 kHz was chosen both to prevent amplification from the accelerometer resonance at 150 kHz and ensure that the fixation method of the accelerometer would be rigid below that frequency. Bessel filters were chosen since they have a linear phase shift and thereby create minimum distortion of the time signal.

In order to investigate the influence of the fixation method of the accelerometer to the machine surface a test rig was developed that gives a high peak acceleration similar to that from an impact wrench with high repeatability. The accelerometer was attached to the surface in three ways, plastic melt glue, accelerometer wax and tack-it from UHU Patafix. Also different temperatures from 10 C to 40 C were tested.
It was found that up to 30 kHz the difference between the methods were small, estimated below 10 %. Due to its simplicity tack-it was used in this study.
Note: Recent development of the measurement technique shows that accurate measurements can be made up 50 kHz but the results in this study are filtered at 30 kHz.

Figure 2 Test rig for accelerometer mounting evaluation

2.2 Comparison of accelerations induced by an impact machine and by a grinder and the influence of ISO 5349 filtering

In order to study the difference between a machine with impact vibrations and a machine with continuous vibrations measurements were performed at the handles on a CP734 impact wrench and a IR88V85 straight grinder. Both machines are pneumatic with metal handles.

Figure 3 Impact wrench CP734 and Ingersoll Rand 88V85 grinder

The accelerations are presented below both before and after ISO 5349 filtering.
What can be seen is that the impact wrench has very high acceleration peaks around 10 000 m/s^2 and that the ISO 5349 weighting almost eliminates these peaks. The grinder has peak vibration at about 500 m/s^2 which is 1/20 of the impact wrench. Note that the grinder vibration is close to the noise level of the measurement system which is about 100 $m/s^2{}_{peak}$. Both machines have an ISO 5349 weighted acceleration of about 5 m/s^2 measured in three axes, and thereby, they should have the same associated risk for HAVS.

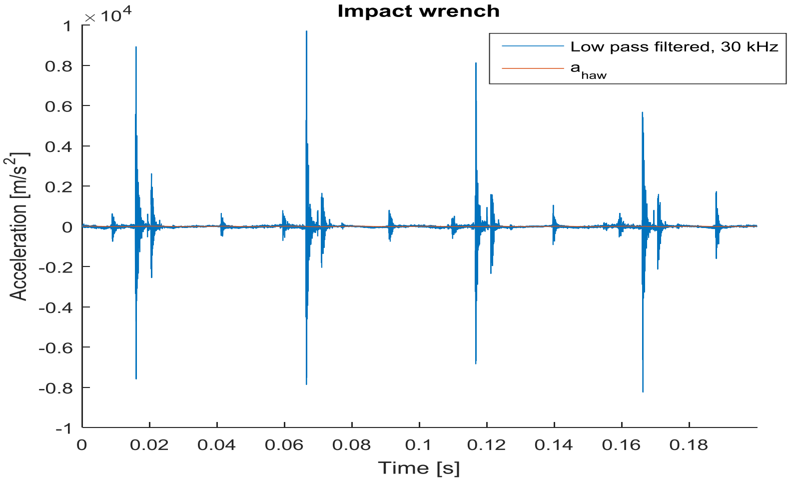


Figure 4 Impact wrench vibration before and after ISO 5349 filtering

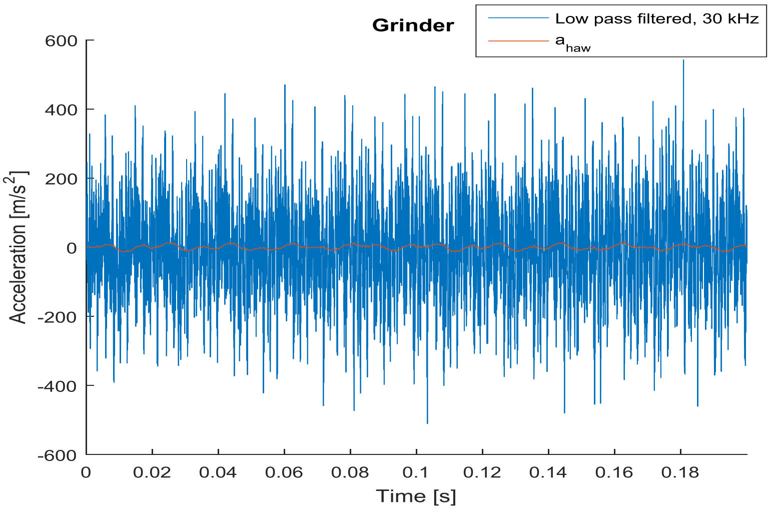


Figure 5 Grinder vibration filtered at 30 kHz and with ISO 5349 filtering

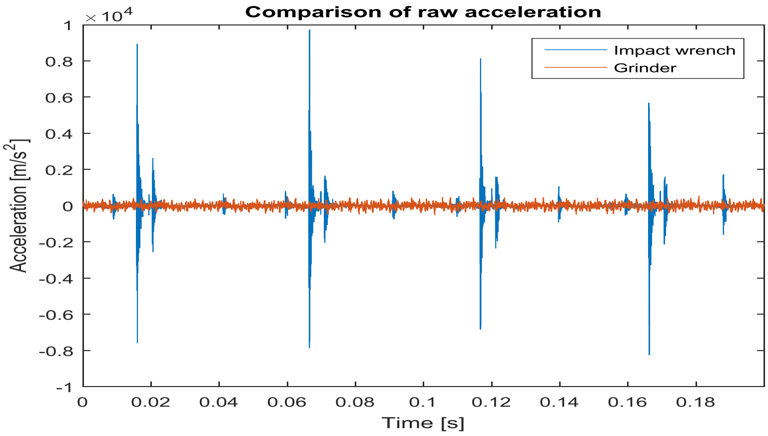


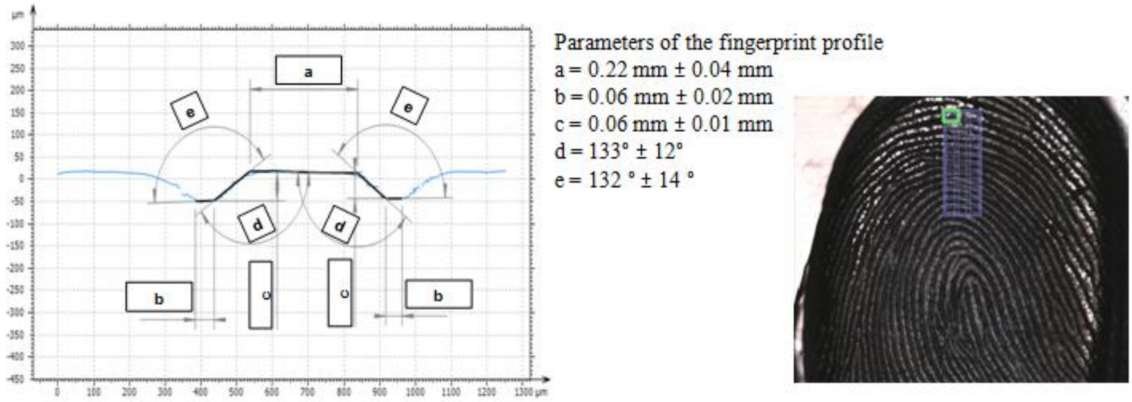
Figure 6 Grinder and impact wrench vibrations filtered at 30 kHz

3 Simulation of pressure propagation into finger tissue

In order to study how transient vibrations affect the finger tissue a finite element (FE) model was constructed. The objective with the model is to simulate how the vibrating surface of the machine interacts with the skin layers in the finger, and study how the created pressure wave propagates into the soft tissue of the finger.

The prediction of wave propagation in viscous tissue material is modeled with a 2D plane strain finite element simulation model. It is solved by the multiphysics simulation program LS-DYNA, whereby the central difference method is adopted. The numerical simulation model consists of a finger model, discretized with 2-D plane strain continuum elements. Initial simulations with a full 3D simulation model with relatively coarse discretization of the finger revealed that a 2D plane strain approach is valid at a distance of at least 25 mm from the tip of the finger in order to reach 2D plane strain conditions within the finger under the short period of the applied acceleration pulse.

The numerical simulation model of the finger includes the components of the human skin e.g. stratum corneum, living epidermis, dermis and subcutaneous tissue. The geometric properties of the different skin layers and the overall dimensions of the finger are derived from findings published in [7] and [8]. Special attention is paid to the contour of the fingerprint where load introduction appears. The structure of the fingerprint of the skin is an important factor since it acts as a vibration isolator since it partly consists of air which is compressible in contrast to the material of the skin.
Therefore an epoxy casting of the index finger fingerprint pushing on a plane plate with a force of 5 N was made. The casting was analyzed in a confocal microscope which built a 3D model of the finger print. The finger depth profile was then parameterized and described by five parameters representing the finger print profile. The exact dimensions are integrated into the simulation model, see figure 7.



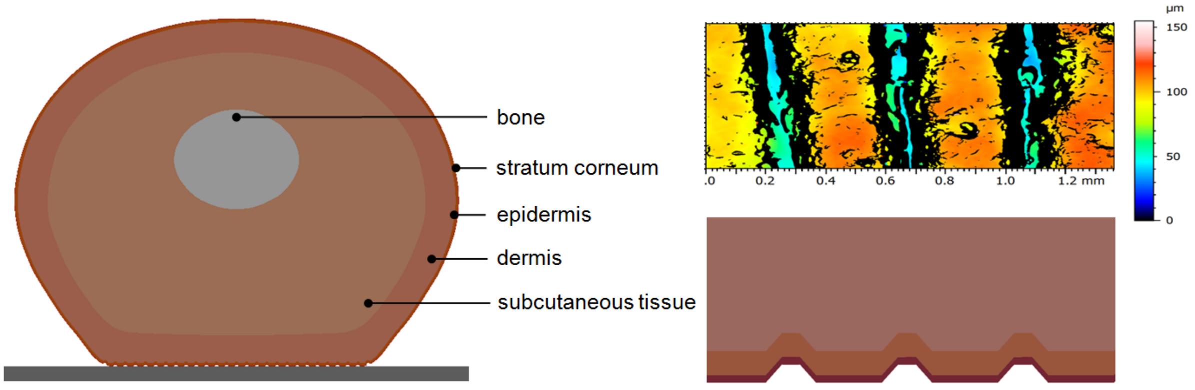


Figure 7: Numerical simulation model with experimentally validated geometry of the fingerprint

Published data on the mechanical properties of the skin layers stratum corneum, epidermis, dermis and subcutaneous tissue reveal differences in the order of magnitude, depending on test set-up, loading conditions, gender, age, location and environmental conditions was chosen from [8; 9]. However, the elastic material properties for dermis and subcutaneous tissue are taken from publications in [7] and the properties for bone material are taken over from [10]. In [11] a compression bulk wave speed of about 1500 m/s is listed for human skin from various investigations. Bulk modulus, density, the corresponding speed of sound and the shear modulus are listed in table 1.

Component	Density [g/cm^3]	Bulk-Modulus [MPa]	Shear-Modulus [MPa]	Soundspeed [m/s]
Stratum Corneum	1.04	2259.0	3.100	1500.0
Epidermis	1.04	2259.0	0.210	1500.0
Dermis	1.04	2259.0	0.080	1500.0
Subcutaneous Tissue	1.00	2161.0	0.034	1470.0
Bone	1.96	20070.0	7719.0	3200.0

Table 1: Material properties of the skin layers

The material response of the skin layers is time and history dependent and is therefore described by a viscoelastic constitutive model, based on exponential stress relaxation functions with shear relaxation behavior described in [12]. The viscoelastic material used in LS-DYNA utilizes the Zener model which is a configuration of a spring and spring-damper element in parallel.

$$G(t) = G_\infty + (G_0 - G_\infty)e^{-\beta t}$$

The viscoelastic behavior is described by the long term asymptotic shear modulus G_∞, the short term shear modulus G_0 and the stress relaxation time 1/ß. The stress relaxation time and long term shear modulus for subcutaneous tissue is taken from [13]. The long term shear modulus for the other tissue layers is adapted proportionally.
The material data for the skin layers stratum corneum and epidermis stated in [7] is further refined by an experimental investigation of the finger and fingerprint distortion under compressive forces. The experimentally evaluated fingerprint geometry in both uncompressed and compressed state is used to validate the finite element simulation model in an inverse optimization approach. With this approach the shear modulus of the skin layers in the numerical simulation model are verified, see figure 8.

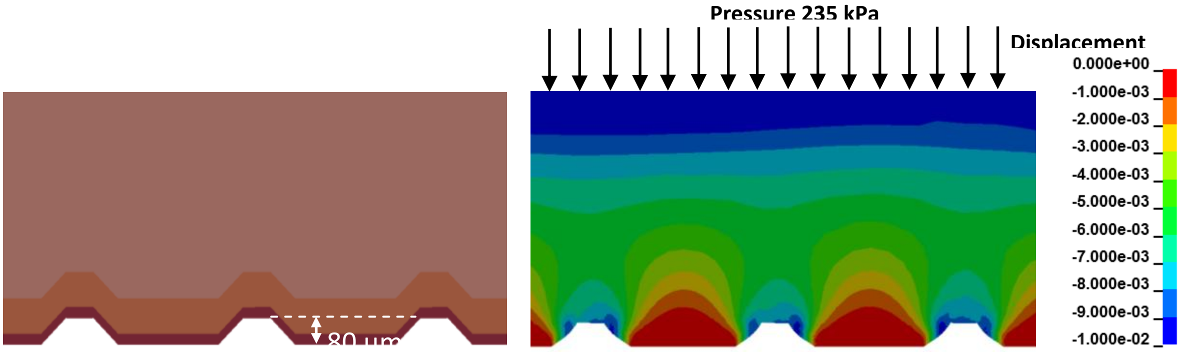


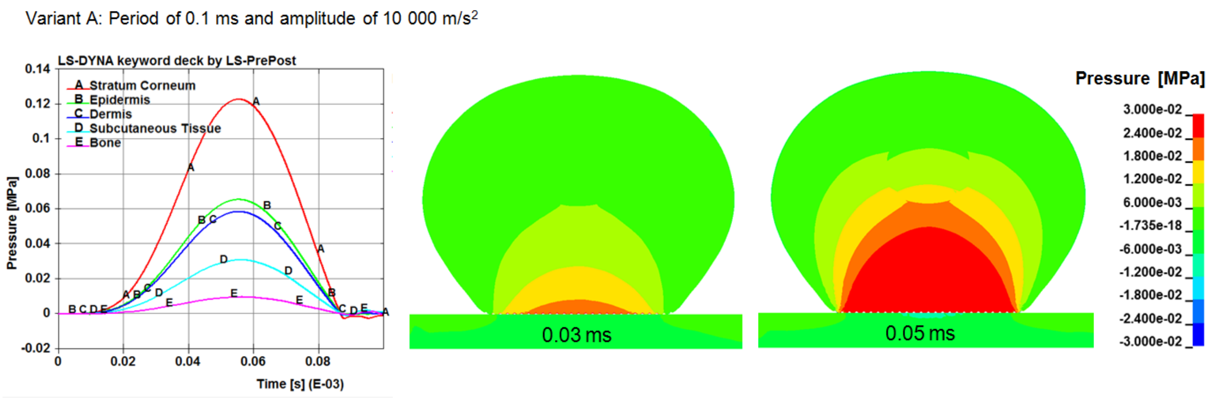
Figure 8: Unloaded geometry of the fingerprint (left) and numerical validation of the fingerprint distortion under constant pressure loading (right)

The volume or bulk viscosity of tissue material is hardly investigated in literature for frequencies below 1 MHz. The most comprehensive overview is published in [14] where the sound attenuation coefficient of human skin is defined to 0.35 dB/cm MHz and at least decreasing linearly towards lower frequencies. With this in mind and because of the short time period investigated the sound attenuation coefficient is disregarded in the present study and left as a topic for future research.

The metal plate acting on the fingerprint is accelerated by a single sinusoidal acceleration pulse characteristic for hand held tool vibrations with:

A: Period of 0.1 ms and amplitude of 10 000 m/s^2
B: Period of 0.01 ms and amplitude of 100 000 m/s^2

The pressure is evaluated in centered position throughout the height of the different skin tissue layers to capture the transient propagation and subsequent reflection of the pressure waves, see figure 9.



Variant B: Period of 0.01 ms and amplitude of 100 000 m/s²

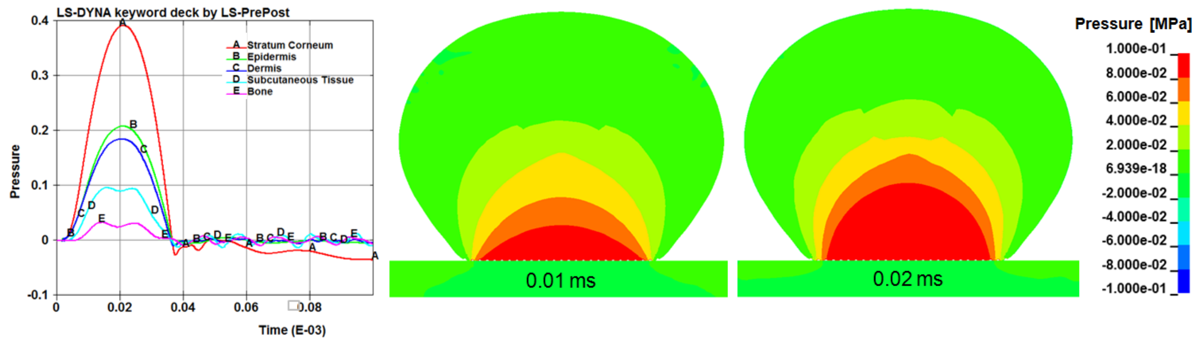


Figure 9: Pressure distribution after load initiation and corresponding pressure propagation in the different finger sections for Variant A and B.

The numerical simulation model unveils a significant pressure level in the finger under the transient acceleration pulse. For Variant A a pressure of at least 0.6 bar is reached in the 3 outer tissue layers and for Variant B the pressure level is increased to more than 2 bar in the outer three tissue layers. However, there is still further experimental research necessary to experimentally verify the viscoelastic material response of the different skin layers as publications over viscoelastic tissue material properties differ considerably.
Investigations in [15] and [16] on strain rate behavior of skin material revealed that elastic properties can increase significantly for strain rates in the same range as seen in the current simulation. Furthermore in literature studies [17] a strong dependence of water content on elastic skin properties is found.

4 Transient vibrations and effect on biological material

In order to investigate if transient vibrations have any effect on biological material there have been studies on exposing red blood cells in vitro and also rat tail in vivo to transient vibrations. Results from three studies are presented in this chapter as examples on findings where transient vibrations have shown to cause severe damages to biological material. It has not the intention to be a comprehensive review, and there are substantially more publications available in this field.

4.1 Study 1: Transient vibration from impact wrenches: Vibration negative effect on blood cells and standards for measurement

The first study [4] was published in 1998 where cow blood was placed in containers on the handles of an impact wrench and on a straight grinder. There was also a container on the socket of the impact wrench.

What was found was that there was four times higher degree of damaged red blood cells on the impact wrench handle than on the grinder handle in despite of that the ISO 5349 vibration on the impact wrench handle was three times lower. And even more, since ISO 5349 estimates the risk by taking the weighted acceleration in square the associated risk should have been 9 times higher on the grinder handle then on the impact wrench handle.

On the impact wrench socket there was a complete destruction of red blood cells.

These results clearly indicate that ISO 5349 weighted acceleration do not correlate to the amount of damaged red blood cells. Instead is the peak acceleration a much better predictor of the damage.

Figure 11 Blood container on impact wrench socket, handle and grinder handle

Test case	Peak vibration amplitude, (m/s^2)	Measured ISO5349 vibration, (m/s^2)	Lysis (%) after 15 min exposure
Impact wrench handle	15 000	2.2	0.4
Impact wrench socket	> 30 000	10	100
Grinder handle while grinding	1 000	7.1	0.1

Table 2 Results of damaged red blood cells after 15 min exposure

4.2 Study 2: Effect of impulsive vibration on red blood cells in vitro

The second study [5] was published in 2005 where red blood cells were subjected to transient vibrations in a test rig. The result was similar to the first study, and it was shown that red blood cells were damaged by the transient vibrations and that the degree was depending on the acceleration level and exposure time.

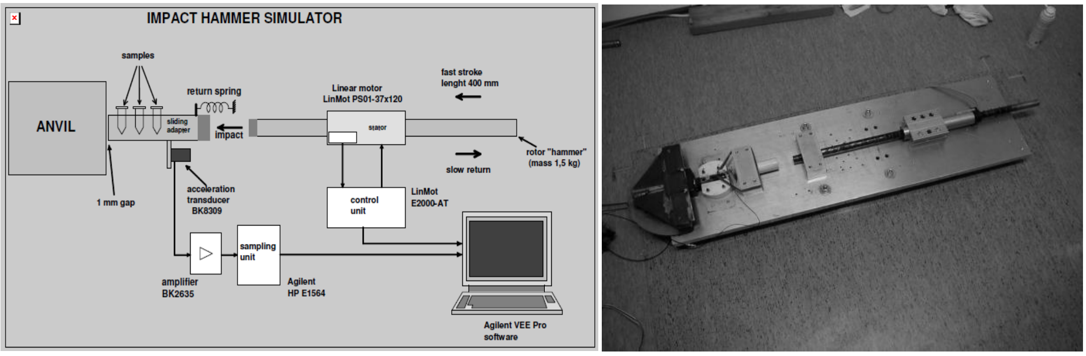


Figure 12 Test setup for red blood cells

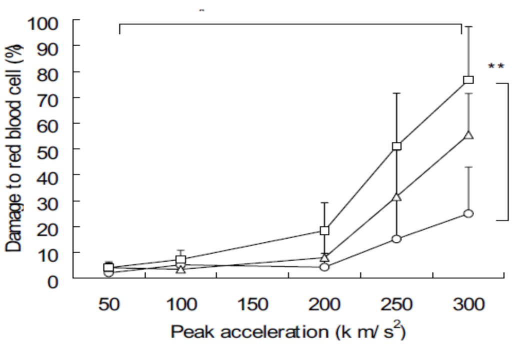


Figure 4. Means and standard deviations of the damage of red blood cells caused by impulse vibration for three exposure durations. (circle, triangle, and square = exposure duration of 10, 20 and 30 minutes, respectively; * P<0.0001, among five peak accelerations; ** P<0.0001, among three exposure durations).

Figure 13 Degree of damaged red blood cells from vibration

4.3 Study 3: Vibration from a riveting hammer causes severe nerve damage in the rat tail model

This study [3] is made on a rat tail model in vivo where the rat tail was exposed to transient vibration from a dedicated test rig with the intention to have the vibration level similar to a bucking bar. Recent measurements made on the test rig after the study was published showed that the ISO 5349 vibration was 9 m/s^2 and the peak acceleration is in the region of 100 000 m/s^2 measured up to 50 kHz. This is a vibration that is similar to what is found on bucking bars, impact wrench sockets and chisels.

The tails were exposed to a single 12 minute vibration. Immediate after stopping vibration there was damage to nerve endings in the skin, mast cell degranulation and hypersensitivity to thermal stimulation. Four days after stopping vibration, the nerve endings had become disrupted, indicating that the single vibration insult triggered a destruction process.

The result from the study is summarized in: “Shock-wave vibration causes severe nerve damage. Frequency weighting seriously underestimates the risk of nerve injury with impact tools.”

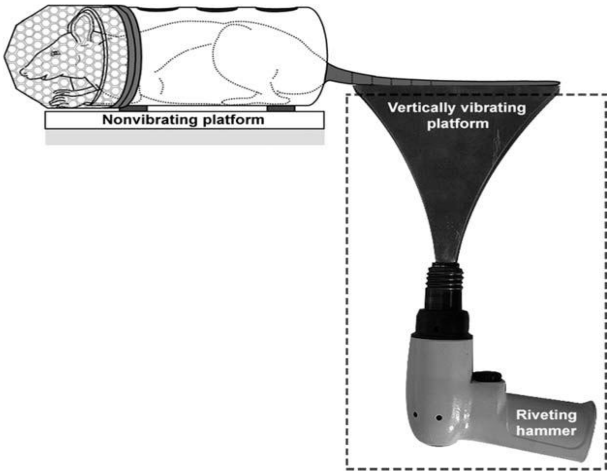


Figure 14 Test rig for rat tail

5 Transient vibrations and HAVS prevalence

There are several studies emphasizing the increase risk for HAVS from machines with transient vibrations. The relationship is examined closer in this section.

5.1 Vibration from riveting tools in the frequency range 6 Hz-10 MHz and Raynaud's phenomenon

The first study [2] conducted in 1986 investigated the prevalence of Raynaud's phenomenon, which is finger blanching and part of HAVS, among workers in the aircraft industry in Sweden. In this industry the main tools used were riveting hammers and bucking bars for assembling the aircraft structures. Fastening a rivet takes only a second so the cumulative exposure time per day is low. However, the acceleration from these tools is very high, reaching acceleration peaks in the region of 100,000 m/s^2.

The cohort was 288 riveters with more than 10 years of work exposure.

The average exposure time was 1 minute/day, and the ISO 5349 vibration level were 10 m/s^2 for the rivet hammer and 11 m/s^2 for the bucking bar.

According to the ISO 5349 risk estimation, there would be very little risk for HAVS, but the prevalence of Raynaud's phenomenon was 50%.
The conclusion is that ISO 5349 does not accurately estimate the risk from using these tools.

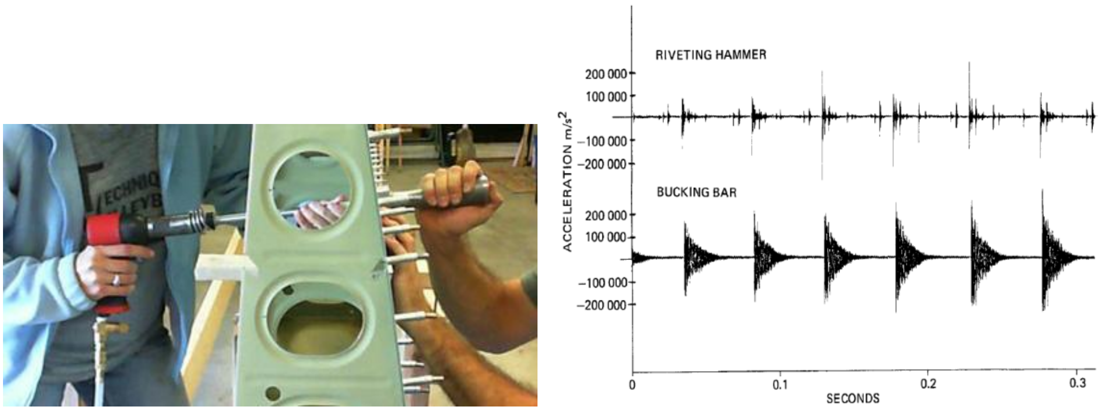


Figure 15 Riveting hammer (left) and bucking bar tool (right) and corresponding accelerations

5.2 Hand-arm vibration syndrome in Swedish car mechanics

The second study [1] on HAVS prevalence was made in 2003 on car mechanics in Sweden. The main tool used were impact wrenches with an ISO 5349 weighted acceleration of 3,5 m/s^2 but with high transient vibrations. The average exposure time was only 10 minutes/day but the prevalence of neurological symptoms based on the Stockholm Workshop scale varied from 8 - 55 %, depending of years of work exposure. The neurological disease is by far much greater than predicted by ISO 5349 calculations.

These 3 studies support the conclusion that ISO 5349 markedly underestimates the risk from tools with transient vibrations.

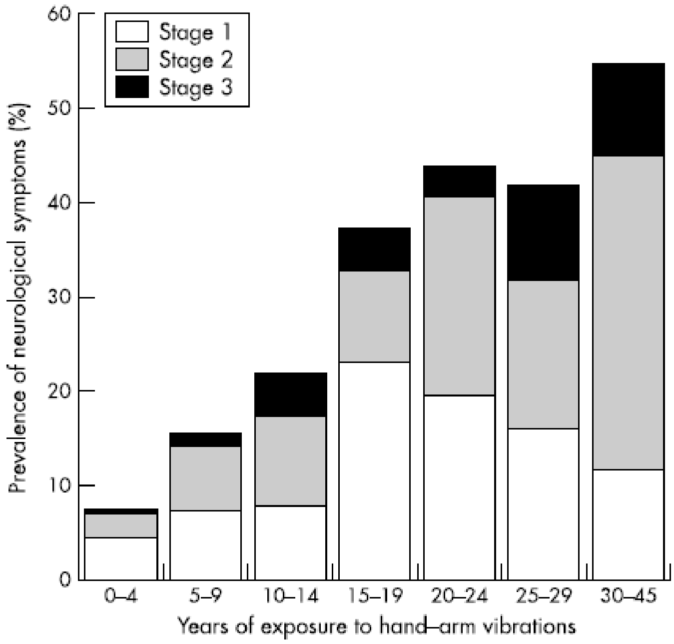


Figure 5 Prevalence of neurological symptoms in different stages according to the Stockholm Workshop scale,[7] in 801 car mechanics, as function of exposure time (in years, information lacking in five mechanics) to hand-arm vibrations. The total number of workers with symptoms and/or signs at the clinical examination was 184, including 20 workers with other conditions that could have contributed to the symptoms (see "Results").

Figure 16 Prevalence on neurological symptoms

6 Preventive measures to reduce transient vibration

If transient vibrations are considered an increased risk for vibration injury, is it possible to reduce these vibrations? The answer is absolutely yes. The possibility to reduce high frequency vibrations by technical modifications is very good, especially if done at the design stage of the tool.

Below are two examples of preventive measures in which technical modifications accomplished large reductions on peak accelerations.

6.1 Modified anvil used in assembly lines

In the assembly line for heavy vehicles, impact wrenches are frequently used to tighten screw joints, and there is 33 % prevalence of HAVS according to internal health reports.

The nuts used are often purposely deformed to avoid loosening when subjected to vibrations. This means that there is a relatively high torque needed to tighten the joint. The tightening of the joint is made by an impact wrench and an anvil holding the nut. The anvil used were ordinary open end and box wrenches.

The vibrations on the unmodified anvil was showed an ISO 5349 vibration of 13 m/s^2 (ISO 5349 weighted acceleration), and peak accelerations reached 8000 m/s^2.

By redesigning an anvil with an internal vibration isolation layer, the ISO 5349 vibration was reduced to 6 m/s^2 with 150 m/s^2 peak accelerations

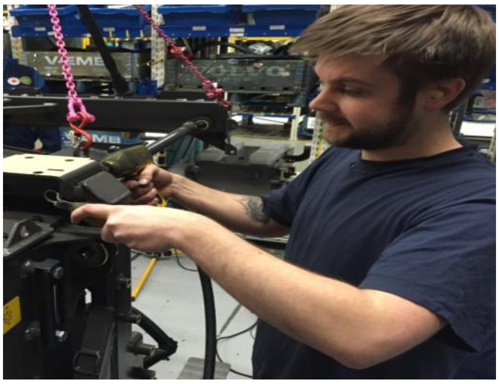


Figure 17 Tightening of screw joints with impact wrench and anvil

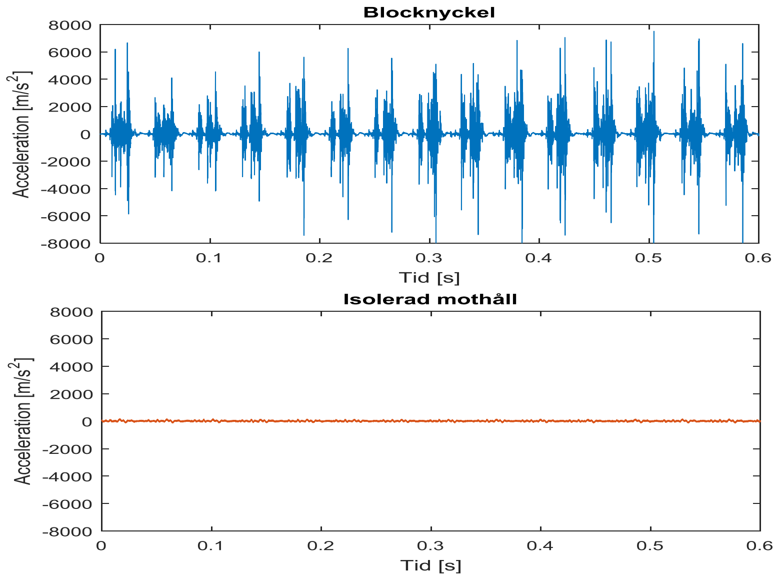


Figure 18 Vibration on original wrench anvil (top) and vibration isolated (below)

Figure 19 Vibration isolated anvil

6.2 Modified impact wrench for repair

Impact wrenches are common in car repair shops and represent the main vibrating tool used by mechanics. Each exposure time is short and takes typically less than a second to tighten or loosen a nut. The ISO 5349 vibrations are in the region of 5 m/s^2 but they all show very high transient vibrations.

The modified machine described below has a redesigned main bearing fitted with a vibration isolation component that prevents both the impacts between the socket and the screw and the internal flywheel and clutch from being directly transmitted to the casing and aluminum handle of the machine.

This modification reduced peak acceleration from 7000 m/s^2 to 800 m/s^2 while maintaining the same work efficiency. The ISO 5349 vibrations were unchanged and remained at 4.5 m/s^2.

Figure 20 Vibration reduced impact wrench

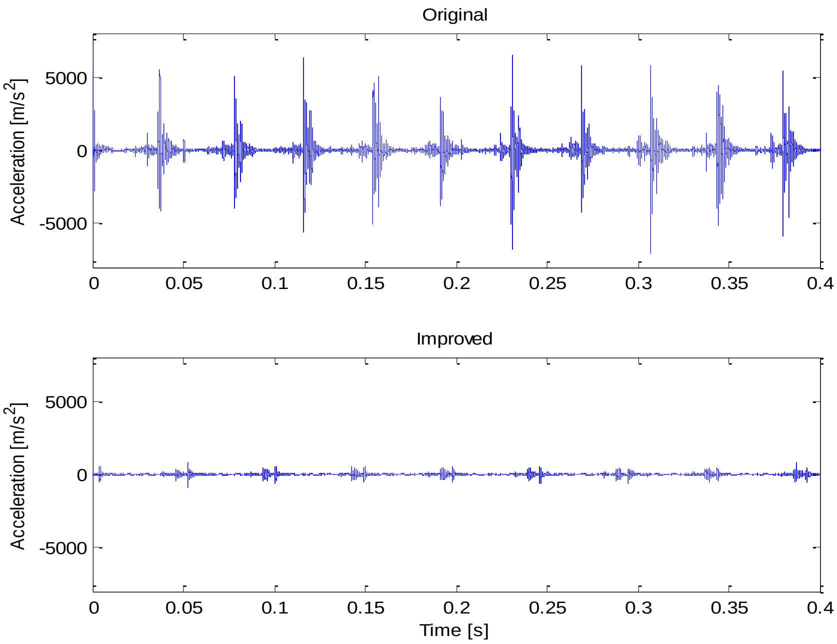


Figure 21 Vibration on original impact wrench and improved

7 Conclusion and discussion

The conclusions from these studies are the following:

- Acceleration can be measured accurately up 50 kHz with newly developed ultra light MEMS accelerometers.
- Most of impact machines generate high amplitude, high frequency transient accelerations.
- High frequency accelerations from impact machines are nearly eliminated by the weighting filter in ISO 5349 and thereby disclosed from risk evaluation.
- Transient vibrations from machines generate a shock wave that propagates into the finger tissue.
- The epidermis layer of the finger has a relatively small attenuation.
- There are several studies that show that ISO 5349 underestimates the risk for HAVS from transient vibrations.
- Transient vibrations can be substantially reduced by redesign of machines.
- ISO 5349 cannot be used for estimation of injuries from HFV and was never intended to do so, and this is clearly stated in the standard.
- There is a need for an amendment to ISO 5349 covering HFV to create an incentive for tool manufacturers and users to reduce the vibration levels.

An opportunity to better address the associated risk from transient vibrations is to study and learn from how related disciplines handle transient, impulse stimulus. There are considerable similarities with regulation for areas such as, impulse noise, head impacts, fragile goods, and material fatigue. What all these areas have in common is that they mainly study the affect of the stimuli in the time domain and not in the frequency domain as is the case for ISO 5349.

8 Acknowledgments

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