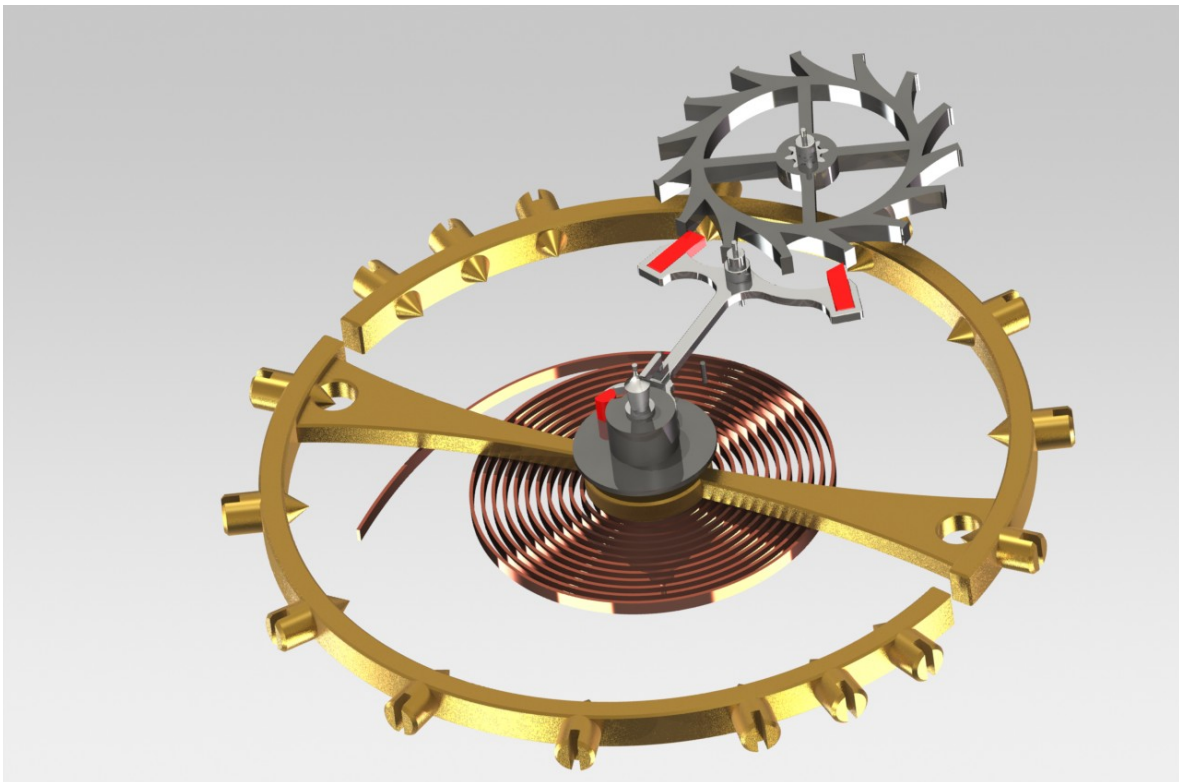

Design of a Compliant Escapement Mechanism

GRADUATION THESIS OF WOUT YPMA
IN COOPERATION WITH FLEXOUS AND TAG HEUER



CONFIDENTIAL

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Graduation Work

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Design of an Isofrequent Compliant Oscillator

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Abstract

Harmonic oscillators typically consist of a moving mass connect to the ground by linear spring. The accuracy of constancy of the frequency of these oscillators is limited by the linearity of the spring. By accepting nonlinearities in the spring element and compensating it with a transmission between the mass and spring, the accuracy of harmonic oscillators can be improved. We present a new method to design harmonic oscillators as we propose an oscillator with a nonlinear compliant transmission to compensate for the nonlinear spring. The results shows an isofrequent compliant oscillator with asymmetric oscillation behavior.

1 Introduction

Mechanical oscillators consist of a elastically deformable spring element and a movable mass. For harmonic oscillators there typically is a linear (direct) transmission between the movement of the mass (fig.1) and the deforming of the spring element and the linear stiffness spring.



Figure 1: Basic structure of a conventional harmonic oscillator

the equations of motion for this system is shown in equation 1. The solution for $\dot{x}_0 = 0$ is shown in equation 2.

$$m\ddot{x} + kx = 0 \quad (1)$$

$$x(t) = x_0 \cos \left(\sqrt{\left(\frac{k}{m}\right)} t \right) \quad (2)$$

$$f = \sqrt{\left(\frac{k}{m}\right)} \quad (3)$$

The key feature of such a harmonic oscillator is that the oscillation frequency (eqn.3) is independent of initial amplitude x_0 . These types of harmonic oscillators can be

found in many applications where accurate timekeeping is required and constant amplitude oscillation can not be guaranteed.

The constancy of the frequency is dependent on the constancy of the ratio $\frac{k}{m}$. Although keeping mass m constant requires no effort, the design and fabrication of a spring element with a constant stiffness k independent of amplitude x is an obstacle.

Compliant mechanisms gain their mobility from the deflection of flexible members rather than from sliding parts [5]. This gives compliant mechanisms considerable advantages over traditional rigid body joints. These advantages include part-count reduction, increased precision, no need for lubrication, no friction and reduced costs and wear.

Springs found in harmonic oscillators can be regarded as a compliant mechanism. The first application of these springs in horology was performed by Christiaan Huygens in the in the 17th century [3]. Since then, the constancy of the stiffness of these springs has been improved by trial and error but the basic design has remained the same [2]. The design of harmonic oscillators as of yet has not been attempted with nonlinear springs.

The goal of this paper is to present a novel and general design approach to create new oscillators with a constant frequency for a range of initial amplitudes. The approach is based on the acceptance of nonlinearity in the springs behavior. This changes the design of a harmonic oscilla-

tor from optimizing the linearity of the spring behavior, to a direct optimization of the constancy of the oscillation frequency over a range of initial amplitudes. This design problem will be solved using a nonlinear transmission using compliant mechanisms between a mass and a compliant nonlinear spring element. A common design strategy for compliant mechanisms, the Pseudo Rigid Body Model [4], will be used to design the transmission. With this method a rigid body linkage can be converted to a compliant mechanism by replacing revolute joints with thin compliant beams. The design will be used to design an oscillator with a constant frequency without the use of a linear spring. The design will be fabricated and tested to show the validity of the approach.

2 Method

The diagram in figure 2 illustrates the approach in this section in a few basic steps. The first step is to design the basic structure of the oscillator. Next the kinematics are optimized to get the desired transmission for a constant frequency. The third step is to convert a number of revolute joints to a compliant structure. The fourth and last step is to optimize this compliant structure to improve the overall performance.

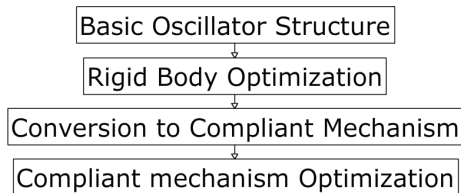


Figure 2: Design Approach for a Compliant Oscillator

2.1 Basic Structure

The basic structure for the oscillator in this paper consists of a rotating mass connected with a compliant Xbob joint. The straightline motion Xbob joint (fig 3a) was designed by Hubbard et al [6] and has nonlinearities in its force-deflection behavior. The connection is made by a six bar linkage, where the Xbob joint is represented by a slider joint with a spring in figure 3b.

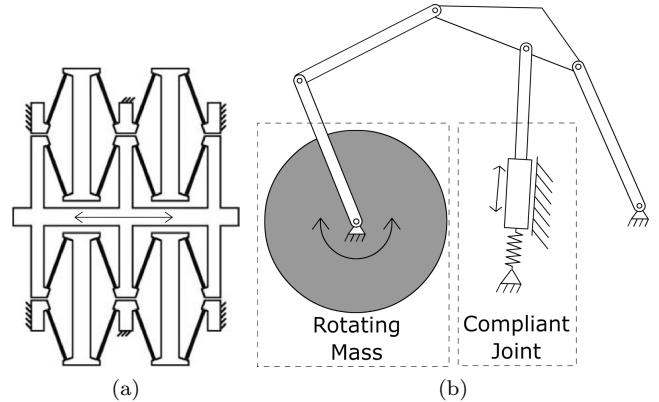


Figure 3: (a) the compliant Xbob joint (b) basic structure of the oscillator

The transmission of the rotation of the mass to the translation at the Xbob joint is achieved by a six bar linkage (including the Xbob joint). It is inherently nonlinear but can be tuned to fit a variety of transmission behaviors. For conventional harmonic oscillators it is rare to combine rotating kinetic energy with translational storage of strain energy. Because of the introduction of the transmission, the total mechanism has no bounds on the direction of the mass or the direction of deflection of the spring. Thus, it is possible to connect a kinetic energy reservoir with any type of elastic energy storage.

2.2 Rigid Body Optimization

Each revolute joint will lose energy in the form of friction, which will damp the oscillation. By using the Pseudo Rigid Body Model (PRBM) approach, the revolute joints are replaced by frictionless elastically deformable (compliant) members. This approach starts in this step of the design process by giving the relevant revolute joints a rotational stiffness. This means that the compliant Xbob joint is no longer the only storage element for strain energy.

In this paper all the joints that are not directly connected to the ground or mass (joint 3,4 and 5) are chosen to be become compliant. The resulting linkage is shown in figure 4

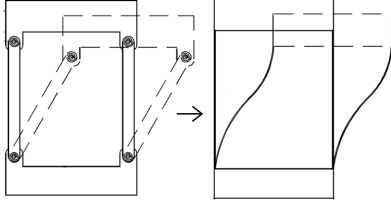


Figure 6: Conversion of rigid body linkage to compliant structure with PRBM

However, this conversion is only valid for a set of conditions concerning the direction of the forces applied to the elastic elements. In this paper this set of conditions is discarded and instead an additional step with optimization is taken to convert the linkage to a compliant mechanism. Both the length and the angle at which the elastic members are placed are optimized to further tune the transmission.

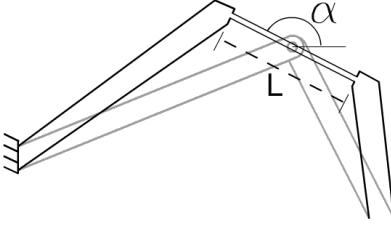


Figure 7: Conversion of a rigid body joint to compliant structure used in optimization

The optimization is performed in a similar way as indicated in figure 5, with the exception that the transmission is now calculated by the Finite Element Analysis program [1]. Also, the rotation of joints 3,4 and 5 is no longer minimized. Instead, the maximum stress in the structure during operation is extracted. By using a FEA program, the optimization now works with all the nonlinearities found in the compliant elements. The details of the optimization of the compliant structure are listed in table 2.

Table 2: Variables for Optimization

Pograms(s)	Matlab[7], Ansys[1]
Algorithm	Multi Objective Genetic Algorithm
Objective	Min(CV(RMSD(f))) for $60^\circ < \beta _{max} < 90^\circ$ Min(Stress in Structure)
Variables	$\alpha_3, L_3, \alpha_4, L_4, \alpha_5, L_5$
Bounds	$2 < L < 20$ $-180 < \alpha < 180$

3 Results

The linkage that results from the optimization discussed in section 2.2 is shown in figure 8.

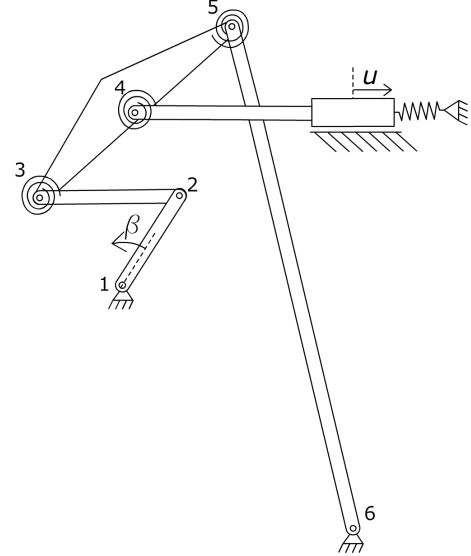


Figure 8: Linkage obtained from the Rigid Body Linkage Optimization

The transmission and the elastic energy level as a function of rotation β are shown in figure 9. The dynamic behavior is calculated by numerical integration of the Lagrange equation 5. The oscillation behavior for a range of initial amplitudes is shown in figure 10.

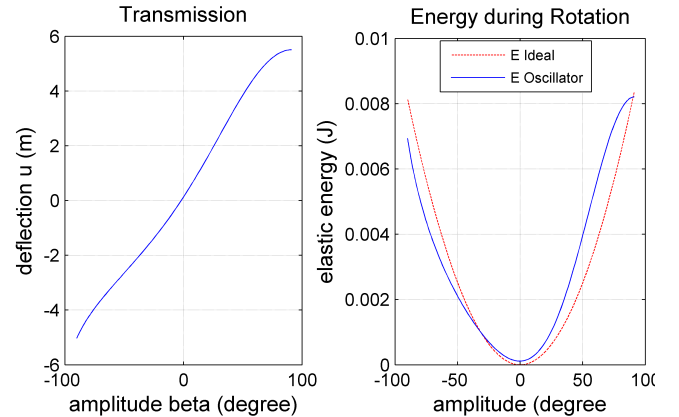


Figure 9: Static properties of Optimized Linkage (fig 8)

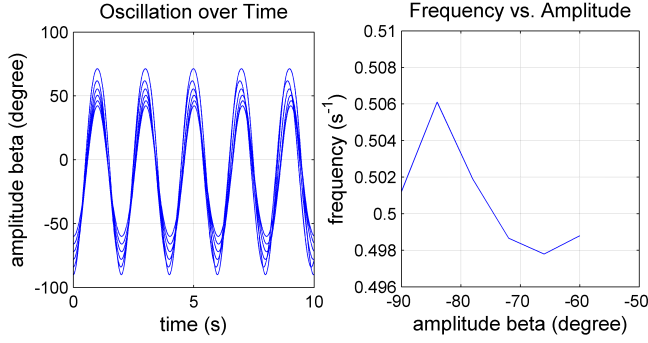


Figure 10: Dynamic properties of Optimized Linkage (fig.8)

Of all the results that were obtained, this linkage was chosen because of its low angular deflection for joints 3,4 and 5. The maximum deflection of these joints over the relevant range of β was 12.2° from relaxed stance. The CV(RMSD) (eqn 4) for this linkage was equal to 5.56×10^{-3} .

Before the conversion optimization is initiated, the entire mechanism is scaled to a desktop size and the frequency aim changed from 0.5Hz to 1.0Hz. This is done *before* the conversion process because scaling becomes more complicated when compliant elements are present.

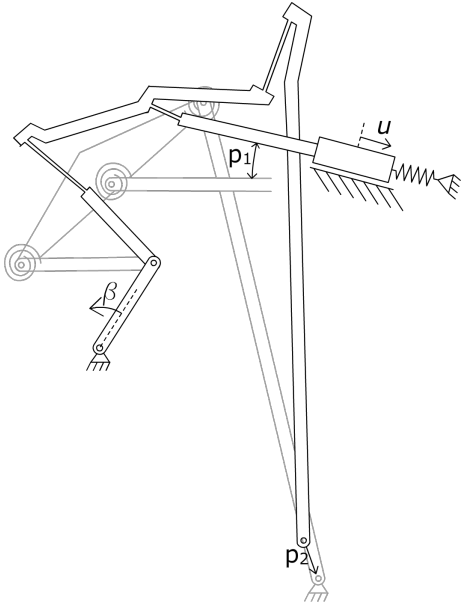


Figure 11: Oscillator obtained from the Compliant Conversion Optimization

The conversion process discussed in section 2.3 is performed using the mechanism from figure 8 as a base.

The Partially compliant mechanism that was obtained is shown in figure 11

The relaxed position of the rotation joints with stiffness is chosen to minimize the angular displacement. This will not necessarily be at the equilibrium stance of $\beta = 0$. Thus, the relaxed position of the entire mechanism will have to be pre-stressed before it can be assembled. In figure 11 the variables p_1 and p_2 indicate points where pre-stress must be applied

The transmission and the elastic energy level as a function of rotation β are shown in figure 12. The oscillation at different initial amplitudes is shown in figure 13.

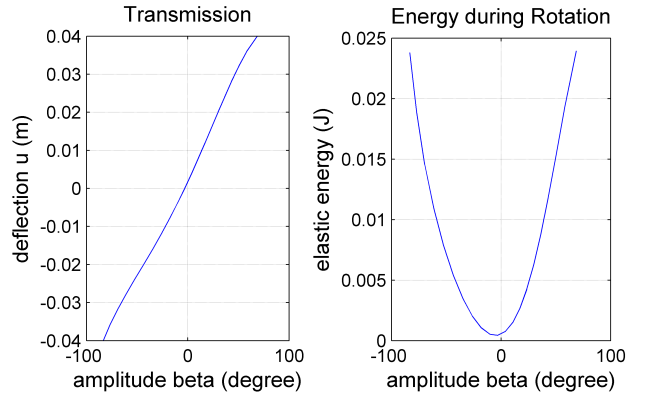


Figure 12: Static properties of the Partial Compliant Oscillator (fig 11)

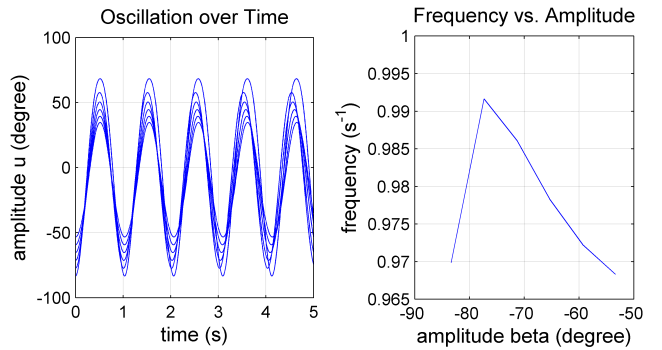


Figure 13: Dynamic properties of the Partial Compliant Oscillator (fig.11)

The CV(RMSD) (eqn.4) for this mechanism is equal to 9.64×10^{-3} .

3.1 Measurements & Validation

A prototype was fabricated with acrylic and spring steel to validate the internal model of the optimization. The prototype is fabricated by firstly laser cutting the rigid parts from 8mm thick acrylic. A slot with a length of 1mm is cut at the positions where a compliant part is to be connected. The compliant parts are cut from $10\mu\text{m}$ thick spring steel and are inserted and glued into the slots. The prototype is shown in figure 14.

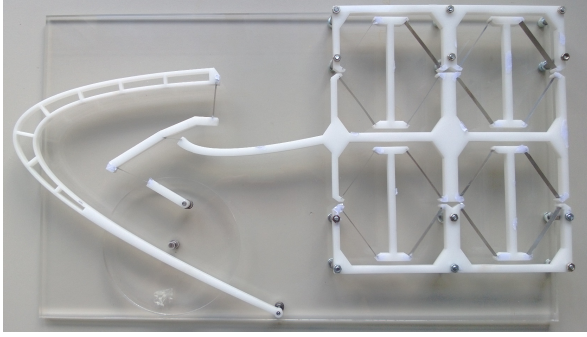


Figure 14: Prototype built to validate results of optimization

The force-deflection behavior has been measured with a loadcell and a precision stage. The transmission of rotation β to deflection u has been measured as well. The resolution for the measurement setup is shown in table 3.

Table 3: Specification for measurement setup

Parameter	Resolution
Force F (N)	$3mN$
Amplitude β (degree)	0.5°
Deflection u (mm)	$10\mu\text{m}$

The result is shown in figure 15, together with the predictions from the model.

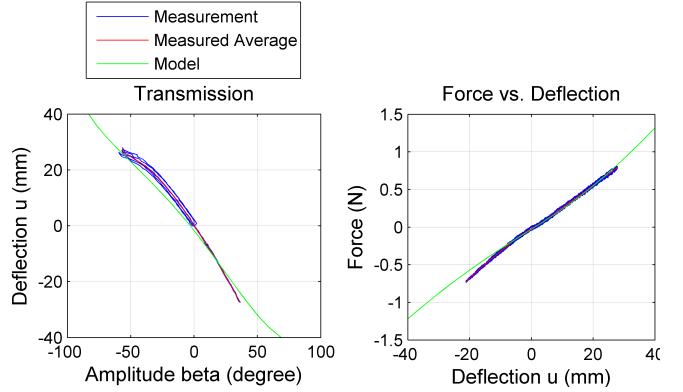


Figure 15: Measurement results together with optimization result

4 Discussion

4.1 Rigid Body

Conventional harmonic oscillator have a perfectly linear transmission because the linear spring and the mass are directly connected. The transmission in figure 9 does not show a strong linear relation between rotation β and deflection u . While stiffness k_2 is constant, the oscillator still shows a harmonic-like behavior with a CV(RMSD) of $5.56 * 10^{-3}$

This has been achieved by the added springs on joints 3,4 and 5. Each spring is another source for energy storage which makes them influence the dynamics of the mechanism. Notice the pre-stress that is applied to assemble the mechanism. This can be seen by the fact that the energy level of the oscillator in figure 9 is never equal to zero.

In figure 9 the energy distribution is also shown for an ideal conventional harmonic oscillator with the same mass and the same average frequency in the optimization. By comparing this harmonic oscillator with the result of the optimization, an interesting property arises: *The energy distribution is asymmetric.* This is a result of the nonlinear transmission as well as the added springs to joints 3,4 and 5. As a consequence, the oscillation is also asymmetric around $\beta = 0$. This is impossible for a conventional harmonic oscillators because of the direct (linear) connection between the mass and the spring element.

Until this point the entire linkage could have been seen as a linear torsion spring that is connected to the mass.

But now it is evident that indeed a new type of harmonic oscillator has been found that is no longer bound by symmetry.

The solution space for this optimization was limited by the resolution and bounds used in the genetic algorithm. By adjusting these parameters, the solution space can be increased and better solutions may be found. However, due to the numerical integration of the Lagrange equation, the evaluation of the objective function is slow and increasing the solution space will slowdown convergence even further.

4.2 Compliant Structure

The compliant structure that was found has a transmission and energy distribution that is very similar to the rigid body mechanism. The same nonlinear transmission and asymmetry in the energy distribution and oscillation is found.

However, the CV(RMSD) rose to a value of $9.64 * 10^{-3}$, an increase of 73%, indicating a decrease in performance. This can partly be explained by the fact that the Xbob joint was no longer considered to have a constant stiffness and instead was modeled completely in Ansys.

It can also be explained by a change in the kinematics. If the length of a compliant member is relatively short, the virtual rotation point of that member will lie close to the original rotation point of the joint. However, also the stresses inside that compliant member will increase. This means that a trade-off exists between maintaining the kinematics and minimizing the stresses. This means that a change in kinematics is almost unavoidable.

Adding variables that could adjust the center of the compliant member may correct for this change, but will also double the number of variables. For each iteration, the algorithm uses the FEA program Ansys to calculate the mechanisms transmission and force-deflection behavior which takes considerable time. Increasing the number of variables increase the number of iterations and also time needed for convergence.

4.3 Measurements & Validation

The results of the the measurements complies with the predictions of the model. The error of the measurements

is small enough to be attributed to manufacturing and measurement errors. Plastic deformation of the spring steel did occur at the maximum deflection of the Xbob joint. This is the reason that the prototype was only tested for 54.7% of the total range of deflection u . The amplitude β was tested for 57.6% of the predicted total range. The plastic deformation of the spring steel was not predicted by the Ansys model. Because of the limited range of motion of the prototype, dynamic testing was not possible.

The approach was designed in such a way that the inertia of the individual parts would be negligible compared to the rotating mass. This is not the case in the current prototype. This has the consequence that the oscillation frequency of the prototype is lower than the model predicts.

5 Conclusion

This paper proposes a method to design a partially compliant oscillator that isofrequent for a range of initial amplitudes. The basic idea is to accept nonlinearity in the force-deflection behavior of the spring element and try to compensate with a compliant structure. The oscillator is designed using two steps of optimization with a genetic algorithm which is also presented here. The first step optimizes the kinematics of a rigid body linkage. The second optimization converts the rigid body linkage to a partially compliant oscillator. The optimization will produce a wide variety options pending on the initial structure that was used.

The oscillator that was designed consists of a rotating mass and a prismatic compliant joint, called the Xbob. However, the approach can be used for any type of combination of a kinetic energy reservoir and an elastic potential energy reservoir.

An oscillator was obtained that has small deviations of the oscillation frequency over a range of initial amplitudes. The oscillator displayed an asymmetric oscillation due to the compliant nonlinear transmission. By no longer restricting the design of a constant frequency to a linear spring, the solution space has increased and new designs become possible.

6 Bibliography

References

- [1] Ansys. Academic research. *Release 14.0*, 2011.
- [2] A. H. C. Bourgeois, A. Perret. Spiral watch spring and its method of production, 2004.
- [3] K. Grimbergen. *Huygens and the advancement of time measurements*. 1278 edition, 2004.
- [4] L. L. Howell. *Compliant Mechanisms*. John Wley and Sons, Inc, New York, 2001.
- [5] L. L. Howell and A. Midha. A method for the design of compliant mechanisms with small-length flexural pivots. *Journal of Mechanical Design*, 116(1):280–290, 1994.
- [6] N. Hubbard, J. Wittwer, J. Kennedy, D. Wilcox, and L. L. Howell. A novel fully compliant planar linear-motion mechanism. *ASME Conference Proceedings*, 2004(46954):1–5, 2004.
- [7] Matlab. *version 7.11.0 (R2010b)*. The MathWorks Inc., 2010.

Literature Report

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Abstract

For many applications in engineering, a linear motion is desirable. This motion is often provided by a prismatic joints that consists of a sliding movement between surfaces. To reduce friction, backlash and remove the need for lubrication, a compliant mechanism can replace these joints. A drawback to compliant mechanisms is the stiffness of their structure. This report gives an inventory of linear motion compliant mechanisms and strategies to remove this stiffness (statically balancing the structure). A literature review on 1 degree of freedom compliant mechanisms and statically balanced compliant mechanisms has been performed. A classification from the results is made and discussed. Two compliant linear motion mechanisms and a balancing category were found to be superior in terms of a high range of motion vs size ratio. These conclusions present a foundation for the design of a new statically balanced linear motion compliant mechanisms.

1 Introduction

Compliant mechanisms gain their mobility from the deflection of flexible members rather than from sliding parts. This gives compliant mechanisms considerable advantages over traditional rigid body joints. These advantages include part-count reduction, increased precision, no need for lubrication, and reduced costs and wear (Howell, 2001).

Because compliant mechanisms rely on the deflection of flexible members, energy is needed to deform the structure. This introduces positive stiffness which means that force is needed to keep the mechanism in positions other than its initial undeformed stance (or other local minima of energy). In the field of precision engineering, where compliant mechanisms are mainly used for stages and manipulators, this becomes a problem. Constant actuation to keep the system in a standstill increases the energy consumption and decreases the precision of the system (Dunning et al., 2011). Because the mechanism absorbs part of the energy that comes into the system, the relationship between input and output is also distorted. In the medical field this means that information from force feedback diminishes when using compliant medical instruments (Sjoerdsma et al., 1997).

To overcome these problems, the mechanisms can be statically balanced. A statically balanced mechanism is a mechanism on which the energy flow of one or more potential energy storage elements are acting, such that the mechanism is in static equilibrium, throughout or at least a considerable part of its range of motion, rather than in a single point or a limited number of points only (Herder, 1998). Such a mechanism possesses constant potential energy and, therefore, if inertia is not considered, it can be moved without any operating effort, even though energy flows may be present within the mechanism.

Statically balancing already has many well known applications in gravity compensation. This can be done by the use of counter-weights. However, this increases the inertia of the system. An alternative is using elastic forces such as springs to compensate gravity (Nathan, 1985) (Streit and Shin, 1993). Another application of static balancing is the compensation of elastic forces. Examples of a system where elastic forces are balanced by springs are the balancing of the elastic forces of a prosthesis glove by springs (Herder, 1998) and the compensation by springs of the elastic forces that are accompanied in the operation of a compliant laproscopic gripper (Herder and Berg, 2000). The last two examples are called statically balanced compliant mechanisms (SBCMs) (Herder and Berg, 2000).

All these examples are instances where an existing mechanism is statically balanced. However, there are few statically balanced compliant joints known. (Morsch and Herder, 2010) designed a statically balanced rotational joints, but no examples exist of a statically balanced prismatic joint. Consequently designing a compliant mechanism, and certainly a statically balanced one, require much attention and must be tailor made for each situation. Statically balanced elements would create the possibility to build statically balanced compliant mechanisms, by combining several balanced elements without doing extended analyses and still obtain a compliant mechanism without zero force and stiffness.

Three categories of SBCMs can be distinguished (Herder and Berg, 2000). The first category consists of a compliant part and a conventional compensation mechanism, consisting of links, joints and springs. The second category is used to describe mechanisms that are fully compliant, but do use separate springs for the storage of the compensation energy. The third category contains fully compliant mechanisms that stores the compensation energy in its compliant members and thus use no separate springs. The fourth category is for mechanisms that can adapt to different load conditions, accommodating for the fact that compliant mechanisms behave different under loading then the unloaded situation. To take full advantage of the possibilities of compliant mechanisms, this study will only cover category three CMs.

The purpose of this literature study is to provide directions towards the design of a statically balanced compliant linear motion mechanism of the third category. To achieve this goal, (1) an overview of compliant linear motion mechanisms is made. Thereafter, (2) give an overview of SBCMs with 1 DoF and categorize them according to their balance strategy. Finally (3), The categories will be discussed and compared according to their performance. Based on the results, a combination between a compliant linear motion mechanism together with a strategy to balance the mechanism will be proposed.

2 Method

2.1 Search Method

The literature study is split in two parts. In the first part, a literature search is done to find compliant linear motion mechanisms. The second part consists of finding different SBCMs of the third category with 1 DoF and categorize them according to their balancing strategy.

The search for the Linear Motion Compliant Mechanisms consists of two sets of keywords: (1) compliant mechanisms and (2) linea motion. The keywords are listed in table 2.1. These key words were entered separately and in likely combinations in the search engines.

Table 2.1: Overview of sets and keywords used for the literature search

Set	Keywords
Compliant mechanisms	- Compliant, mechanism, flexure, elastic, monolithic - no joints
Linear Motion	- Straight, prismatic, linear, translation

For the search for SBCMs there are three sets of keywords: (1) compliant mechanisms, (2) statically balanced and (3) balance refinement. The third category was needed because the search with categories 1+2 yielded many gravity compensation systems which are not of interest for this report. The keywords are listed in table 2.2.

Table 2.2: Overview of sets and keywords used for the literature search

Set	Keywords
Compliant mechanisms	- Compliant, mechanism, flexure, elastic, monolithic - no joints
Static balance	- Static balance, neutral balance, stable, no force - force balance, force compensation - force cancellation, force free
balance refinement	-NOT(gravity)

The search engine SCOPUS was used to find relevant articles and conference proceedings. The results were filtered first by title. Secondly by reading the abstract and/or conclusion. The number of search results per search was confined to the first 1000 most relevant hits. Also, the references of the relevant articles were checked for additional relevant references that could weren't caught by the initial search.

2.2 Classification

To compare the mechanisms, it is beneficial to group similar designs together. This grouping could even out the error that is introduced by the authors level of expertise and reveal the value of a mechanism or strategy better. To accommodate the process, categories are chosen that group the mechanisms based on their design principles.

2.2.1 Compliant Linear Motion Mechanisms

The compliant linear motion mechanisms are firstly classified in two groups. The first group contains mechanisms with (1) distributed compliance. The second group contains mechanisms with (2) lumped compliance. If the width of any flexure in the mechanism is not at least 10 times smaller than the length of said flexure, the mechanism has lumped compliance (Howell, 2001). Otherwise the mechanism has distributed compliance. In figure 2.1 examples of both distributed (Cannon et al., 2005) and lumped compliance (Pavlovi and Pavlovi, 2009) are shown.

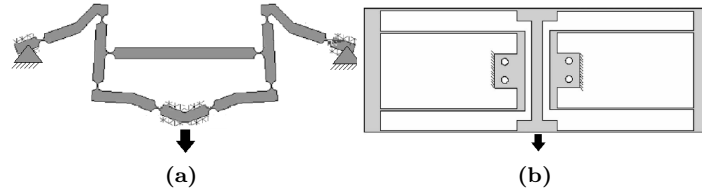


Figure 2.1: The difference in the level of compliance is categorized by (a) Lumped Compliance and (b) Distributed Compliance

For a large range of motion, flexures with lumped compliance can be modeled by rotational joint (Howell and Midha, 1994). This simplifies the design process for compliant mechanisms with lumped compliance. However, the use of lumped compliance also has its disadvantages. The drawbacks include high stress concentrations, low static and fatigue strength, and difficulty in manufacturing very thin and thick sections (Ananthasuresh and Kota, 1995).

The second classification is on basis of the characteristics of the mechanism. Three classes are distinguished, mechanisms that are (1)Stable, with one stable position, (2)Bistable, with two stable positions, and (3)Multistable, with three or more stable positions. A stable position indicates a (local) minimum of the potential or elastic energy inside the mechanism.

An overview of the classification for the compliant linear motion mechanisms is shown in table 2.3

2.2.2 Statically Balanced Compliant Mechanisms

SBCMs all have a range where the elastic energy inside the system is constant. However, the strategy to achieve this feature varies. One strategy is to connect a structure with positive stiffness to a structure with negative stiffness and pre-stress it to attain zero stiffness and force (=constant energy). Another

Table 2.3: Schematic representation of the classification for compliant linear motion mechanisms.

	Linear motion CMs		
level 1: Compliance	Distributed	Lumped	
level 2: Stability	Stable	Bistable	Multistable

would be to pre-stress the entire structure with a buckling force, thus putting the mechanism in a continuously equilibrium. To characterize the balancing strategies, categories are selected based on the pre-stressing action that is used to attain a constant energy.

Compliant mechanisms typically consist of elastically deformable beams. The degrees of freedom of the end of such beams is motion perpendicular to the beam. These beams can be pre-stressed in different directions relative to the beams degree of freedom. The SBCMs are classified in three groups, mechanisms with pre-stressing (1)axial or (2)off-axial to the degree of freedom of the beam or (3) torsional stressing, which means a pre-stressing in a rotational direction. A schematic overview of the classification is given in table 2.4.

Table 2.4: Schematic representation of the classification for SBCMs.

	SBCMs		
level 1: Stress Direction	Axial	Off-axial	Torsional

3 Results

3.1 Linear Motion Compliant Mechanisms

The literature search resulted in twelve devices. All the compliant mechanisms are fully compliant and perform a linear motion. Several mechanisms have one or two stable positions and only one has more than 2 stable positions (Chen et al., 2009). All were found to have at least one axis of in-plane symmetry. Relevant data for all mechanisms have been added to Appendix A.

A new class of fully compliant bistable mechanisms with the added benefit of integrated self-retraction has been developed by (Masters and Howell, 2003). This bistable switching device uses eight tensural pivots to achieve motion (fig 3.1a).

In (Kang et al., 2004) a compliant parallel mechanism is presented for the field of micro electro mechanical systems (MEMS). This mechanism uses ten flexural pivots to provide a linear motion, driven by a piezo actuator (fig 3.1b). Due to the over constrained design, the working range is limited.

A novel linear motion mechanism design was developed by (Hubbard et al., 2004). By connecting eight compliant versions of the Roberts four-bar approximate straight line mechanism, a compliant linear motion mechanism called X-bob is constructed (fig 3.1c).

in (Qiu et al., 2004) a design is presented for a compliant bistable switching device for MEMS application. The design consists of two curved centrally-clamped parallel beams (fig 3.1d).

In (Cannon et al., 2005) a compliant folded beam linear motion mechanism described for micro-scribing (fig 3.1e). The aim of this paper was the design of a very high axial to lateral stiffness ratio.

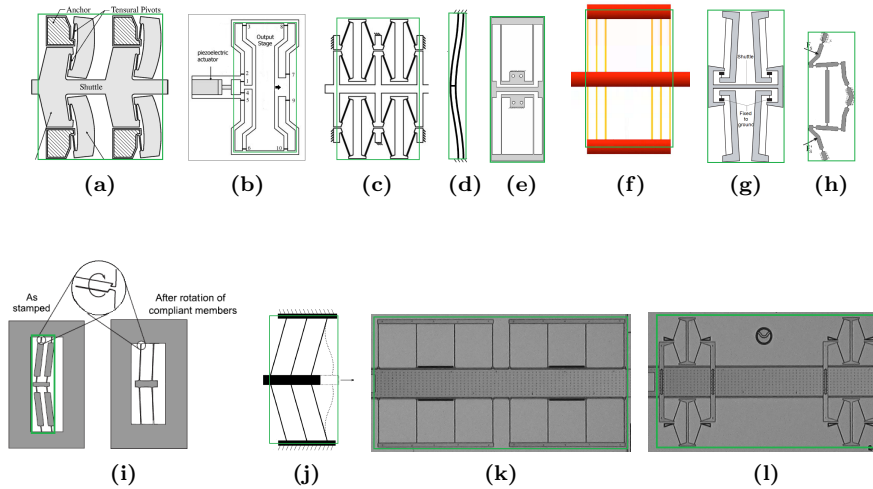


Figure 3.1: Twelve linear motion mechanisms; (a) Masters and Howell (2003), (b) Kang et al. (2004), (c) Hubbard et al. (2004), (d) Qiu et al. (2004), (e) Cannon et al. (2005), (f) Trease et al. (2005), (g) Chen et al. (2009), (h) Pavlovi and Pavlovi (2009), (i) Todd et al. (2010), (j) Holst et al. (2011), (k,l) Mackay et al. (2012)

In (Trease et al., 2005) the design of a compliant spatial linear motion mechanism is shown and analysed. However, a planar version of the design with similar behaviour is also available (fig 3.1f).

In (Chen et al., 2009) a fully compliant double tensural micromechanisms is shown with three stable positions (fig 3.1g). This mechanism comprises of both short and long flexures and can be used as a three-way switch.

(Pavlovi and Pavlovi, 2009) presents a design based on the compliant counterpart of a Watts linkage. To counteract the rotation of the output of the mechanism, two compliant Watts linkages are connected at the output. To improve on the rotational stiffness, two of the double Watts linkages are put in series. Finally to improve on the straight line error of the linkage, it is mirrored again. The complete mechanism consists of a total of eight Watts linkage (fig 3.1h).

(Todd et al., 2010) presents a new technique for fabricating compliant mechanisms from stamped metal sheets (fig 3.1i). The concept works by providing thinned segments to allow rotation of flexural beams 90 deg about their long axis, effectively providing a flexure as wide as the sheets thickness. A bistable compliant switch with this technique is described.

In (Holst et al., 2011) explores the deflection and buckling of fixed-guided beams used in compliant mechanisms (fig 3.1j). With a new method the behaviour of a bistable switching device is analysed.

In (Mackay et al., 2012), the Folded-beam and X-bob (fig 3.1k, 3.1l) were optimized for various metrics such as off-axis stiffness and stroke.

An overview with relevant data for the above described mechanisms is shown in table 3.1.

3.2 Statically Balanced Compliant Mechanisms

The literature search resulted in ten devices. All the compliant mechanisms have one DoF and except for (Rosenberg et al., 2010), all the mechanisms are fully compliant. Two mechanisms were found that had rotation instead of translation as its DoF. Relevant data for all mechanisms have been added to Appendix B.

(Stapel and Herder, 2004) present a feasibility study into the design of a fully compliant statically balanced grasper for minimal invasive surgery. The compliant grasper with positive stiffness is balanced

Table 3.1: Schematic representation of the classification for linear motion compliant mechanisms.

Compliance	Reference	Stability*	Size (mm)		Stroke (mm) δ
			x	y	
Distributed	(Masters and Howell, 2003)	B	0.29	0.33	0.0085
	(Hubbard et al., 2004)	S	190	260	44
	(Qiu et al., 2004)	B	3	0.06	0.12
	(Cannon et al., 2005)	S	22	7	4
	(Trease et al., 2005)	S	48	100	11.4
	(Todd et al., 2010)	B	55	6	3
	(Holst et al., 2011)	B	150	55	11
	(?) X-bob	S	1.85	1.48	0.5
	(?) Folded	S	1.88	0.912	0.5
Lumped	(Kang et al., 2004)	S	2.3	1.23	0.002
	(Chen et al., 2009)	M	0.27	0.55	0.042
	(Pavlovi and Pavlovi, 2009)	S	3.5	9	2.6

*S=Stable, B=Bistable, M=Multi-stable

by a compliant structure with negative stiffness.

In (Lange et al., 2008) the same problem is solved, but with the a technique called topology optimization. In topology optimization, for a defined objective an optimal design is systematically searched by a computer algorithm. Because of high stresses in the structure during operation, a prototype was never built.

In (Hoetmer et al., 2010) a building block approach is used for the design of SBCMs. It is assumed that a SBCM is composed of a functional segment with positive stiffness and a negative stiffness segment connected in series. The method is tested by applying it to balance a compliant gripper mechanism.

(Morsch and Herder, 2010) developed a generic zero stiffness compliant rotation joint. To this end, a conventional balanced mechanism, consisting of two pivoted bodies which are balanced with two zero-free-length springs, is taken as an initial concept. Then joints and springs are replaced with compliant structures.

(Rosenberg et al., 2010) presents a novel straight-line self-guiding statically-balanced mechanism which reflects on the advantages of lumped compliant mechanisms. The design is conceived with a constant energy approach for static balancing of mechanisms (Gallego and Herder, 2010). This design has one pin joint at the center which disqualifies it for a category three mechanism. However, because there is only one pin-joint and the design is original, it is still included in this part of the literature study.

In (Tolou et al., 2010) two different statically balanced compliant micro mechanisms were designed. In case I the mechanism is preloaded perpendicular to the stroke.

In case II the balancing is done collinear with the stroke. Also, the balancing is achieved without external preloading of the mechanism. Instead the mechanism build up of elastic energy is achieved by actuating it in the only DoF it has.

(Chen and Zhang, 2011) describe a near-zero-stiffness mechanism which combines two multistable mechanisms. As with (Tolou et al., 2010), the balancing is achieved without external preloading.

(Gallego and Herder, 2011) presents two cases where a static balance is achieved through the energy approach. In case I, an approach was implemented in the stiffness reduction of a typical compliant inverter by the application of a pre-load in order to reach a state of neutral stability for a small range of motion.

In case II, an approach was implemented into a building block approach where two blocks (four-bar linkages) were designed to exhibit a linear potential energy function, and doing so, by combining them

the total potential energy became constant. The blocks were modelled through the use of torsion springs so it will be possible to use the Pseudo-Rigid-Body model (Howell, 2001) to obtain the final fully compliant design.

An overview with relevant data for the above described mechanisms is shown in table 3.2.

Table 3.2: Schematic representation of the classification for SBCMs.

Stress direction	Reference	Residual force range (N)	Max. compensated force (N)	Size (mm)		balanced stroke (mm)
		f	F	x	y	δ
Axial	(Stapel and Herder, 2004)	0.037	15	23	8	0.56
	(Lange et al., 2008)	70	300	104	59.2	0.65
	(Hoetmer et al., 2010)	0.1	6.9	170	180	4.8
	(Morsch and Herder, 2010)	0.012Nm	0.25Nm	200	40	1.2rad
	(Tolou et al., 2010) Case II	0.002	0.035	1.84	0.095	0.05
	(Chen and Zhang, 2011)	0.9	7	177	173	7
Off-axial	(Tolou et al., 2010) Case I	0.0005	0.055	0.667	1.1	0.05
	(Gallego and Herder, 2011) Case I	0.16	1.4	54	59	3
Rotational	(Rosenberg et al., 2010)	0.12	4	1200	1800	2000
	(Gallego and Herder, 2011) Case II	0.235Nm	13.53Nm	3.3	2.1	2.3rad

4 Discussion

In this section, the mechanisms are compared and discussed with each other based on criteria. Many articles did not supply size, working range or information on the force-deflection characteristics. This makes it harder to make an objective comparison between mechanisms. When possible, an estimation for the mechanisms size was made.

The function of the linear motion CMs is to guide a straight linear motion. All mechanisms listed in table 3.1 fulfill this function. A second demand is a long stroke. The stroke is measured according to the metric given in (?), shown in formula 4.1. The metric represents the stroke (δ) with respect to the width (x) and height (y) of the system.

$$\delta^* = \frac{\delta}{\sqrt{x^2 + y^2}} \quad (4.1)$$

In figure 4.1 the metric for different linear motion CMs is shown. The results are grouped according their level of compliance.

The length of the stroke can be restricted by high stress in the system or the kinematics of the system. However, in (Pavlovi and Pavlovi, 2009) no stress calculations are shown or discussed. Therefore it is unknown whether this mechanism could achieve the claimed stroke and is excluded from further comparisons.

The mechanism with the distributed compliance and the shortest stroke is (Kang et al., 2004). A close examination of the mechanism shows that it is over constrained. This issue is resolved in the paper by stating that some flexural pivots should be regarded as two degree of freedom joints. However, the translational motion in a flexure is limited which would explain the short stroke ($\delta^* = 0.00077$) in the complete mechanism.

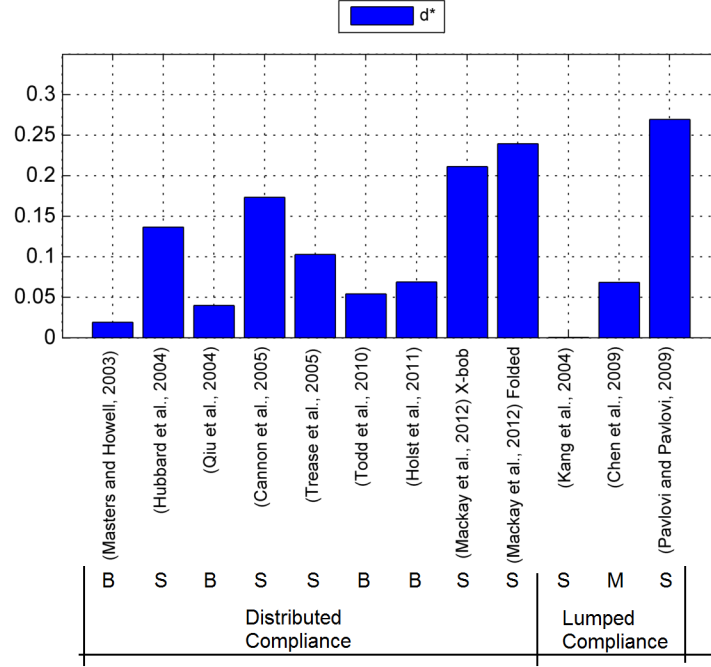


Figure 4.1: The ratios between translation and size for the linear motion CMs

When (Pavlovi and Pavlovi, 2009) is excluded from comparison, it can be seen that mechanisms with distributed compliance generally have a larger stroke. The mechanisms with only one stable position, all have longer strokes with respect to the bistable or multistable mechanisms.

The mechanisms discussed in (Mackay et al., 2012) are based on the designs in (Hubbard et al., 2004) and (Cannon et al., 2005). An increase in the stroke ratio of respectively 54% and 38% were achieved by size optimization. These two improved designs have the longest stroke to size ratios when the mechanism from (Pavlovi and Pavlovi, 2009) is neglected.

For the comparison between SBCMs another metric is added. It is shown in formula 4.2. This metric shows the portion of force that is balancing performance of the mechanism. It relates the forces that are present in the balanced system ($F_{balanced}$) to the forces that are present for the different parts of the system ($F_{unbalanced}$) over the entire balanced stroke of the mechanism.

(Gallego and Herder, 2010) and (Morsch and Herder, 2010) both have rotation as its DoF instead of translation. To make the metrics dimensionless, the stroke was multiplied by a quarter of the diagonal of mechanism's surface. However, a good comparison is still not possible since the metric is constructed in a different way compared to SBCMs with a translational DoF.

$$f^* = 1 - \frac{\max(F_{balanced}) - \min(F_{balanced})}{\max(|F_{unbalanced}|)} \quad (4.2)$$

For the comparison of the SBCMs, metric δ^* is also used. Now the stroke of the mechanism is taken as stroke of the mechanism where the it is balanced.

In figure 4.2 the results for the SBCMs are shown. The SBCMs are grouped according to their category.

Figure 4.2 shows that the highest average balancing performance per category is the axial stress category. However, for (Rosenberg et al., 2010) there is the issue of the pin-joint that was discussed in 3.2. Further research needs to be done to see if this mechanism can be adjusted to fit into the category

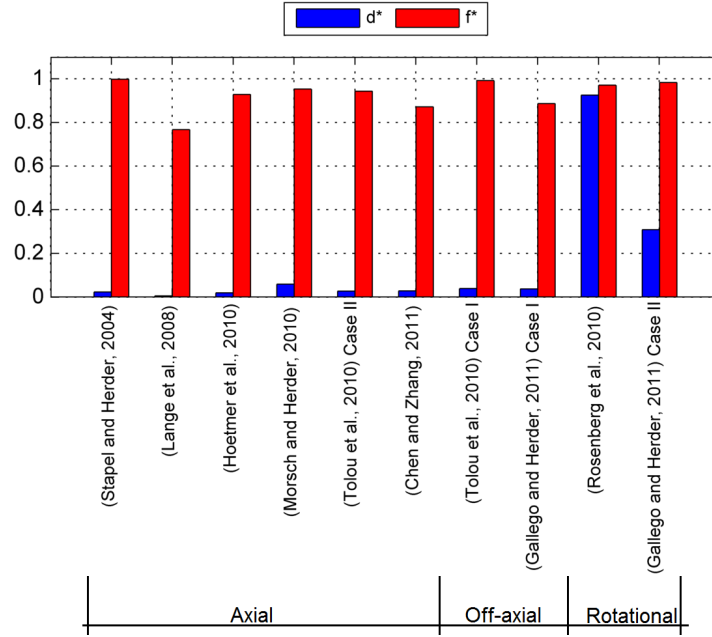


Figure 4.2: The ratios between translation and size and the ratios between remain force range and maximum force balanced

III CMs and still maintain its high stroke versus size ratio.

The mechanism with the highest compensation percentage is found in (Stapel and Herder, 2004). This mechanism acquires a value for f^* of 0.998. However, (Tolou et al., 2010) case I, (Hoetmer et al., 2010) and (Gallego and Herder, 2011) case II with 0.990, 0.986 and 0.982 are not far off. However, there is no clear (dis)advantage between a stress category and high balancing performance.

Figure 4.2 shows that the category with rotational stressing has the highest stroke. However, as mentioned before in section 3, (Rosenberg et al., 2010) does not belong to the category three SBCMs, further research should be conducted to find out if the pin joint could be replaced with a compliant structure. Also, the conversion from rotational to translational stroke for (Gallego and Herder, 2011) case II and (Morsch and Herder, 2010) gives a biased perception of their stroke.

figure 4.3 shows the balancing qualities of the SBCMs that have translation as its DoF instead of rotation ((Gallego and Herder, 2011) case II, (Morsch and Herder, 2010)) and also (Rosenberg et al., 2010) is excluded because of its pin joint.

From the set of SBCMs that have translation as its DoF and strictly belong to category three CMs, (Tolou et al., 2011) case I and (Gallego and Herder, 2011) case I present the largest stroke with a value of 0,039 and 0,038.

In 4.3 an advantage in terms of a high stroke can be seen for off-axially stressed SBCMs.

5 Conclusion

An overview, a classification and a discussion of existing linear motion compliant mechanisms and SBCMs with 1 DoF have been made towards the design of a new linear motion statically balanced compliant mechanism.

It was found that the compliant linear motion mechanisms with one stable position have a larger range of motion with respect to their size than mechanisms with two or more stable positions. It

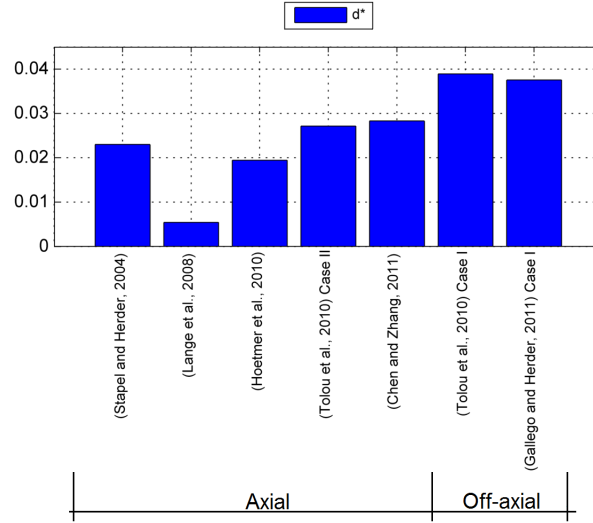


Figure 4.3: The ratios between translation and size for SBCMs

was also shown that the mechanisms with distributed compliance have a larger range of motion than mechanisms with lumped compliance.

Consequently it can be concluded that a statically balanced linear motion mechanism should be built with a stable linear motion mechanism and with distributed compliance as its base.

Two specific designs were found in literature that could fulfill this function: the X-bob mechanism and the Folded Suspension mechanism, both from (Mackay et al., 2012).

It was found that SBCMs that were off-axially stressed presented better range of motion with respect to their size than mechanisms that were axially stressed. No clear relation was found between the direction of stressing and the performance in terms of force compensation.

The balancing of a Folded mechanism or X-bob by pre-stressing them off-axially can result in a balanced compliant linear motion joint with a large stroke to size ratio.

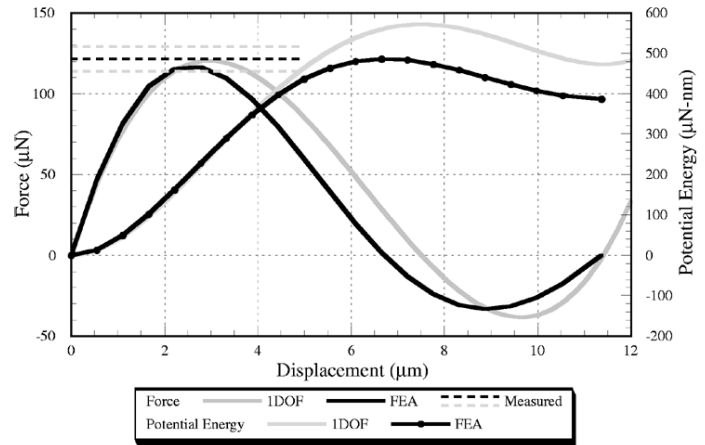
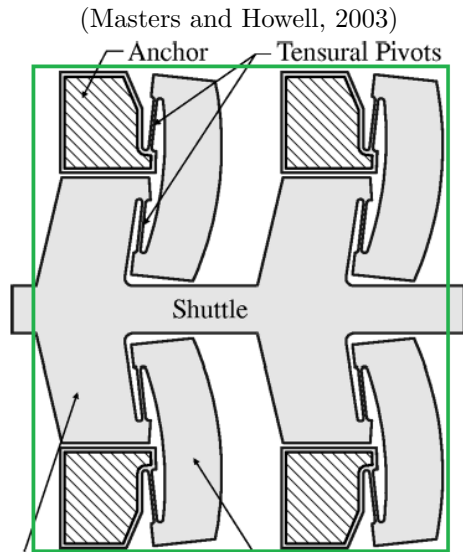
References

- Ananthasuresh, G. K. and Kota, S. (1995). Designing compliant mechanisms. *Mechanical Engineering*, 117(11):93–96.
- Cannon, B. R., Lillian, T. D., Magleby, S. P., Howell, L. L., and Linford, M. R. (2005). A compliant end-effector for microscribing. *Precision Engineering-Journal of the International Societies for Precision Engineering and Nanotechnology*, 29(1):86–94.
- Chen, G., Wilcox, D., and Howell, L. L. (2009). Fully compliant double tensural tristable micromechanisms (dttm). *Journal of Micromechanics and Microengineering*, 19(2):025011.
- Chen, G. and Zhang, S. (2011). Fully-compliant statically-balanced mechanisms without prestressing assembly: concepts and case studies. *Mechanical Sciences*, 2(2):169–174. MS.
- Dunning, A. G., Tolou, N., and Herder, J. L. (2011). Review article: Inventory of platforms towards the design of a statically balanced six degrees of freedom compliant precision stage. *Mech. Sci.*, 2(2):157–168. MS.
- Gallego, J. A. and Herder, J. (2010). Criteria for the static balancing of compliant mechanisms. ASME.

- Gallego, J. A. and Herder, J. (2011). Buckling as a new perspective on static balancing of mechanisms.
- Herder, J. and Berg, F. v. d. (2000). Statically balanced compliant mechanisms (sbcm's) an example and prospects. ASME.
- Herder, J. L. (1998). Design of spring force compensation systems. *Mechanism and Machine Theory*, 33(1-2):151–161.
- Hoetmer, K., Woo, G., Kim, C., and Herder, J. (2010). Negative stiffness building blocks for statically balanced compliant mechanisms: Design and testing. *Journal of Mechanisms and Robotics-Transactions of the Asme*, 2(4):041007–7.
- Holst, G. L., Teichert, G. H., and Jensen, B. D. (2011). Modeling and experiments of buckling modes and deflection of fixed-guided beams in compliant mechanisms. *Journal of Mechanical Design*, 133(5):051002–10.
- Howell, L. L. (2001). *Compliant Mechanisms*. John Wiley and Sons, Inc, New York.
- Howell, L. L. and Midha, A. (1994). A method for the design of compliant mechanisms with small-length flexural pivots. *Journal of Mechanical Design*, 116(1):280–290.
- Hubbard, N., Wittwer, J., Kennedy, J., Wilcox, D., and Howell, L. L. (2004). A novel fully compliant planar linear-motion mechanism. *ASME Conference Proceedings*, 2004(46954):1–5.
- Kang, B., Wen, J., Dagalakakis, N., and Gorman, J. (2004). Design of optimization for a parallel mems mechanism with flexure joints. *Proceedings of ASME Design Engineering Technical Conferences*, DETC2004/MECH-57455.
- Lange, D. d., Langelaar, M., and Herder, J. (2008). Design of a statically balanced compliant laparoscopic grasper using topology optimization. ASME.
- Mackay, A. B., Smith, D. G., Magleby, S. P., Jensen, B. D., and Howell, L. L. (2012). Metrics for evaluation and design of large-displacement linear-motion compliant mechanisms. *Journal of Mechanical Design*, 134(1):011008.
- Masters, N. D. and Howell, L. L. (2003). A self-retracting fully compliant bistable micromechanism. *Journal of Microelectromechanical Systems*, 12(3):273–280.
- Morsch, F. and Herder, J. (2010). Design of a generic zero stiffness compliant joint. ASME.
- Nathan, R. H. (1985). A constant force generation mechanism. *Journal of Mechanisms, Transmissions, and Automation in Design*, 107(12):508–512.
- Pavlovi, N. T. and Pavlovi, N. D. (2009). Compliant mechanism design for realizing of axial link translation. *Mechanism and Machine Theory*, 44(5):1082–1091.
- Qiu, J., Lang, J. H., and Slocum, A. H. (2004). A curved-beam bistable mechanism. *Journal of Microelectromechanical Systems*, 13(2):137–146.
- Rosenberg, E., Radaelli, G., and Herder, J. (2010). An energy approach to a 2dof compliant parallel mechanism with self-guiding statically-balanced straight-line behavior. ASME.
- Sjoerdsma, W., Herder, J. L., Horward, M. J., Jansen, A., Bannenberg, J. J. G., and Grimbergen, C. A. (1997). Force transmission of laparoscopic grasping instruments. *Minimally Invasive Therapy and Allied Technologies*, 6(4):274–278.
- Stapel, A. and Herder, J. (2004). Feasibility study of a fully compliant statically balanced laparoscopy grasper. ASME.
- Streit, D. A. and Shin, E. (1993). Equilibrators for planar linkages. *Journal of Mechanical Design*, 115(3):604.

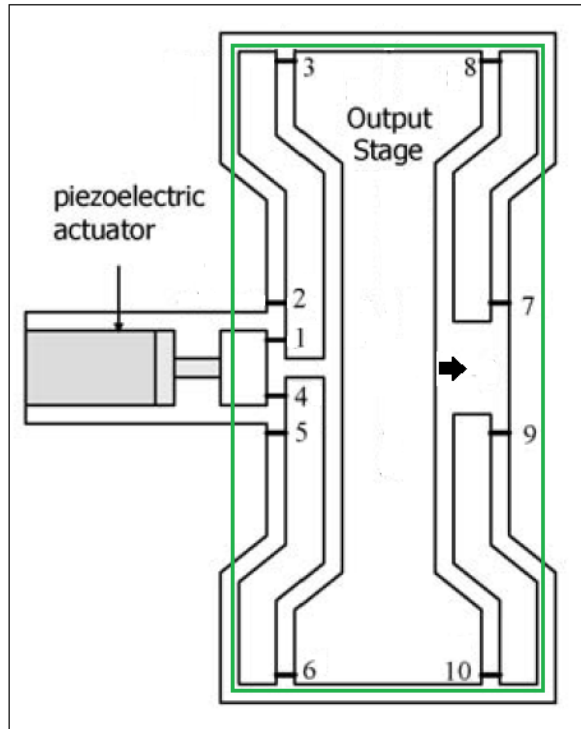
- Todd, B., Jensen, B. D., Schultz, S. M., and Hawkins, A. R. (2010). Design and testing of a thin-flexure bistable mechanism suitable for stamping from metal sheets. *Journal of Mechanical Design*, 132(7):071011–7.
- Tolou, N., Estevez, P., and Herder, J. (2011). Collinear-type statically balanced compliant micro mechanism (sb-cmm): experimental comparison between pre-curved and straight beams. *ASME Conference Proceedings*.
- Tolou, N., Henneken, V. A., and Herder, J. L. (2010). Statically balanced compliant micro mechanisms (sb-mems): Concepts and simulation. ASME.
- Trease, B. P., Moon, Y.-M., and Kota, S. (2005). Design of large-displacement compliant joints. *Journal of Mechanical Design*, 127(4):788.

A Linear Motion Compliant Mechanisms Data



<i>Author(s)</i>	Masters, N. D. and Howell, L. L
<i>Paper</i>	A self-retracting fully compliant bistable micromechanism
<i>Width</i>	0.29mm
<i>Height</i>	0.33mm
<i>Stroke</i>	0.0085mm
<i>Stability</i>	Bistable, 2 stable points
<i>Compliance</i>	Distributed

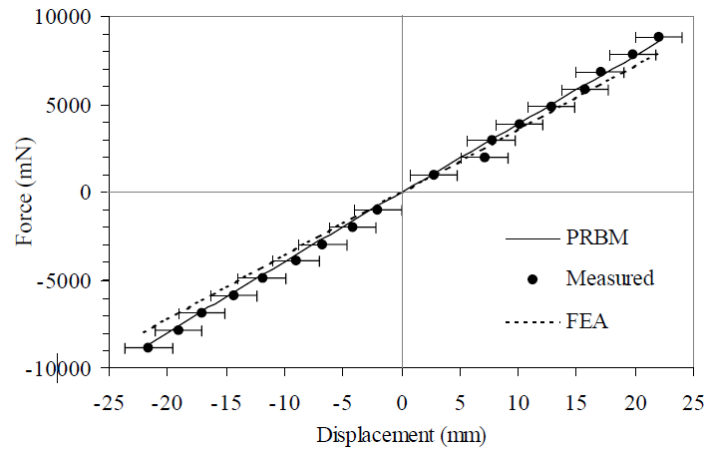
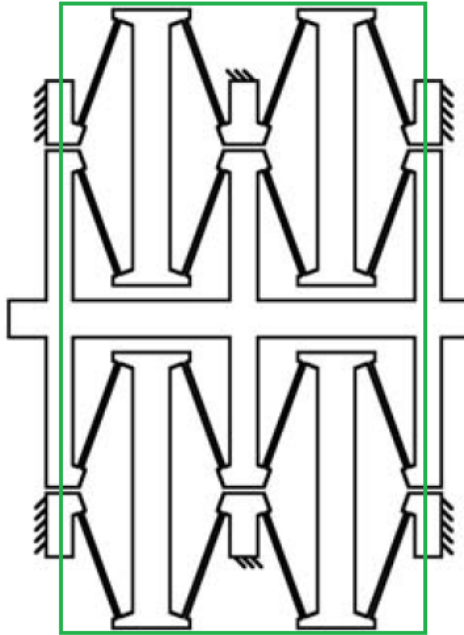
(Kang et al., 2004)



no force-deflection figure available

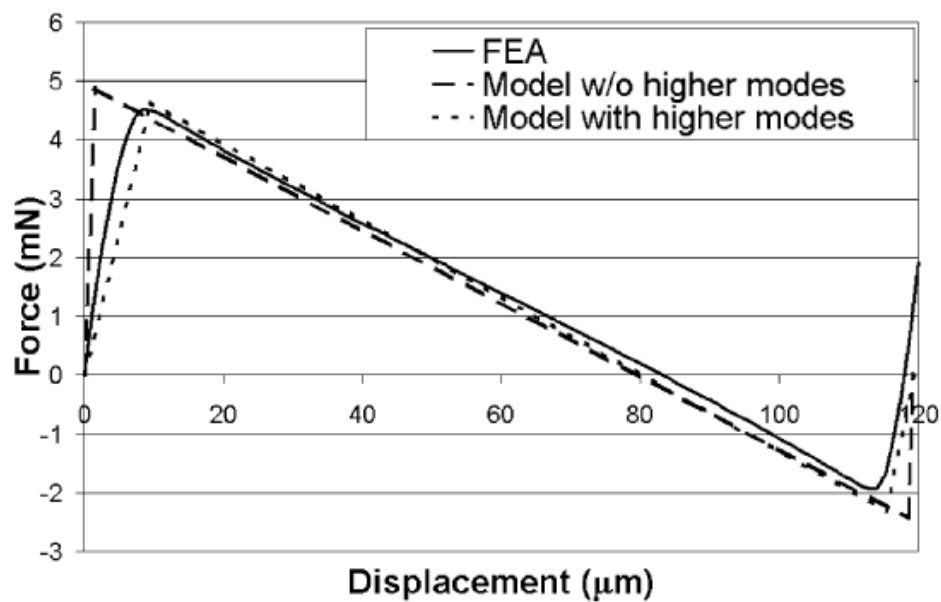
<i>Author(s)</i>	Kang, B.H. and Wen, J.T. and Dagalakis, N.G. and Gorman, J.J.
<i>Paper</i>	Design of Optimization for a parallel Mems Mechanism with Flexure Joints
<i>Width</i>	2.3mm
<i>Height</i>	1.23mm
<i>Stroke</i>	0.002mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Lumped

(Hubbard et al., 2004)



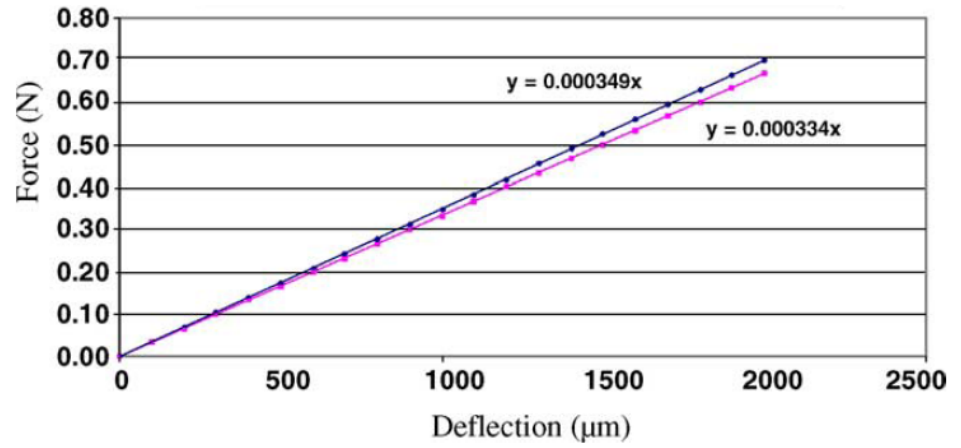
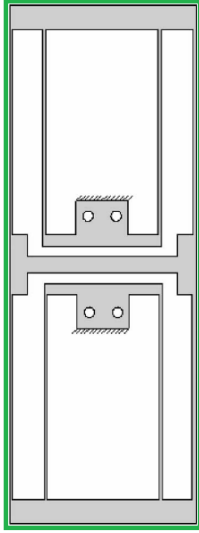
<i>Author(s)</i>	Hubbard, N.B. and Wittwer, J.W. and Kennedy, J.A. and Wilcox, D. and Howell, L. L.
<i>Paper</i>	A Novel Fully Compliant Planar Linear-Motion Mechanism
<i>Width</i>	190mm
<i>Height</i>	260mm
<i>Stroke</i>	44mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Distributed

(Qiu et al., 2004)



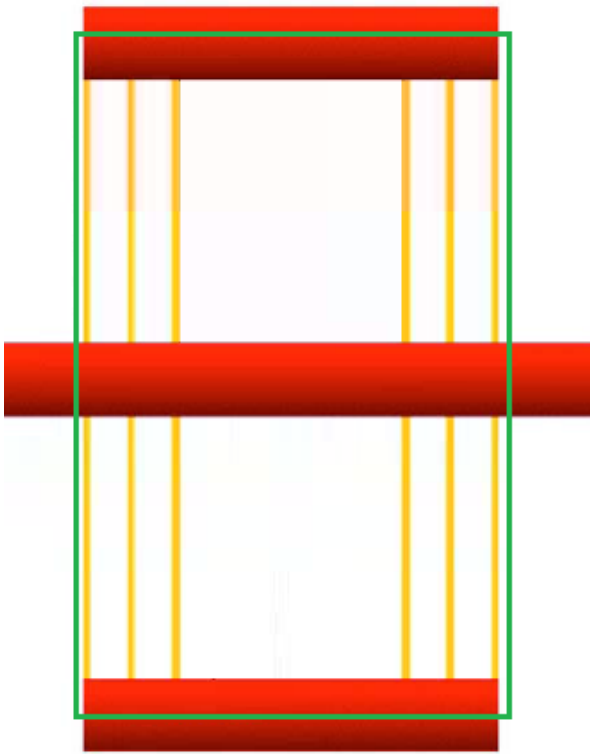
<i>Author(s)</i>	Qiu, J. and Lang, J. H. and Slocum, A. H.
<i>Paper</i>	A curved-beam bistable mechanism
<i>Width</i>	3mm
<i>Height</i>	0.06mm
<i>Stroke</i>	0.12mm
<i>Stability</i>	Bistable, 2 stable points
<i>Compliance</i>	Distributed

(Cannon et al., 2005)



<i>Author(s)</i>	Cannon, B. R. and Lillian, T. D. and Magleby, S. P. and Howell, L. L. and Linford, M. R.
<i>Paper</i>	A compliant end-effector for microscribing
<i>Width</i>	22mm
<i>Height</i>	7mm
<i>Stroke</i>	4mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Distributed

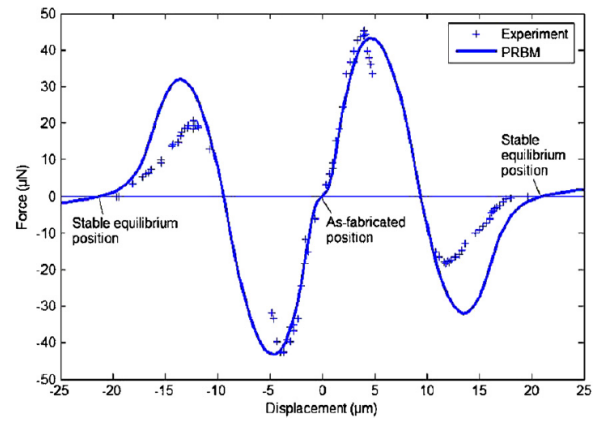
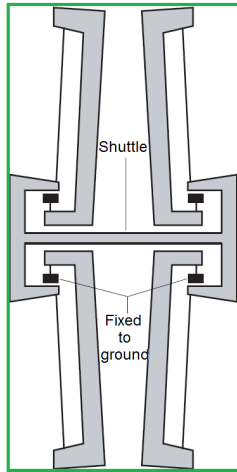
(Trease et al., 2005)



No force-deflection figure available

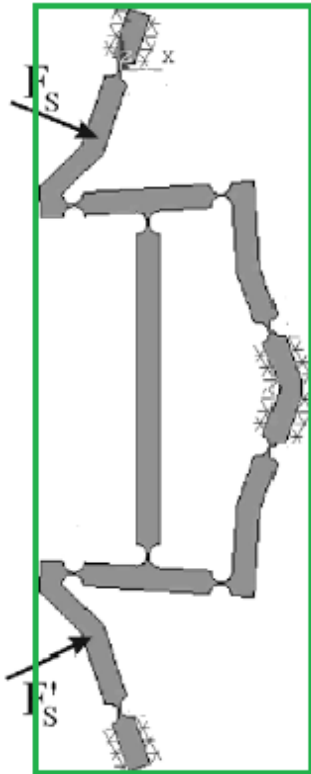
<i>Author(s)</i>	Trease, Brian P. and Moon, Yong-Mo and Kota, Sridhar
<i>Paper</i>	Design of Large-Displacement Compliant Joints
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<i>Stroke</i>	11.4mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Distributed

(Chen et al., 2009)



<i>Author(s)</i>	Chen, G. and Wilcox, D. and Howell, L. L.
<i>Paper</i>	Fully compliant double tensural tristable micromechanisms (DTTM)
<i>Width</i>	0.27mm
<i>Height</i>	0.55mm
<i>Stroke</i>	0.042mm
<i>Stability</i>	Multistable, 3 stable points
<i>Compliance</i>	Lumped

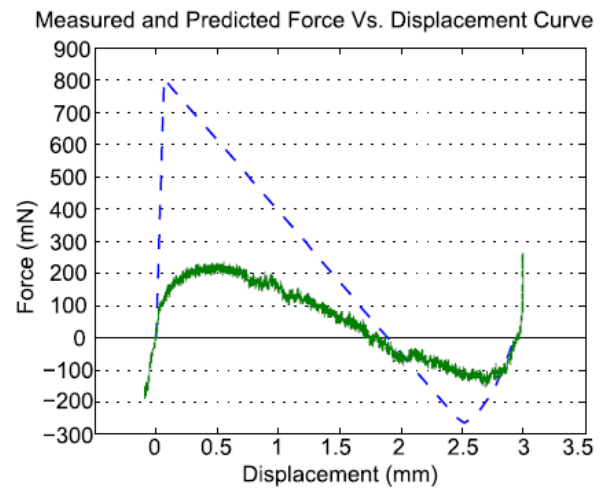
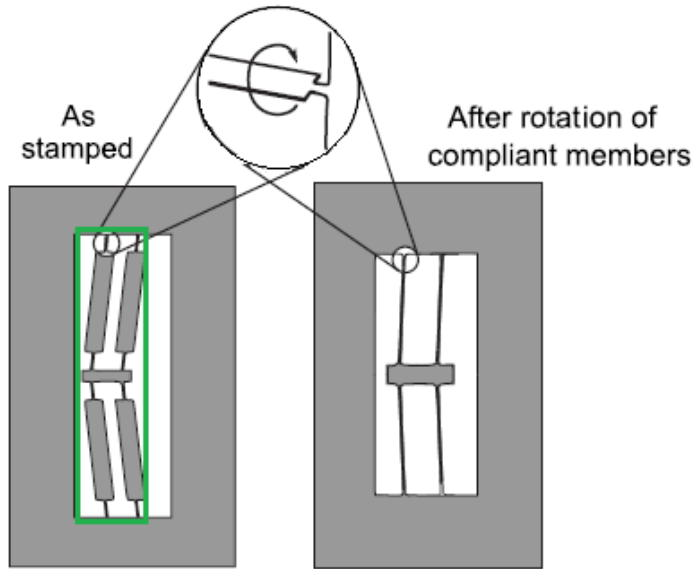
(Pavlovi and Pavlovi, 2009)



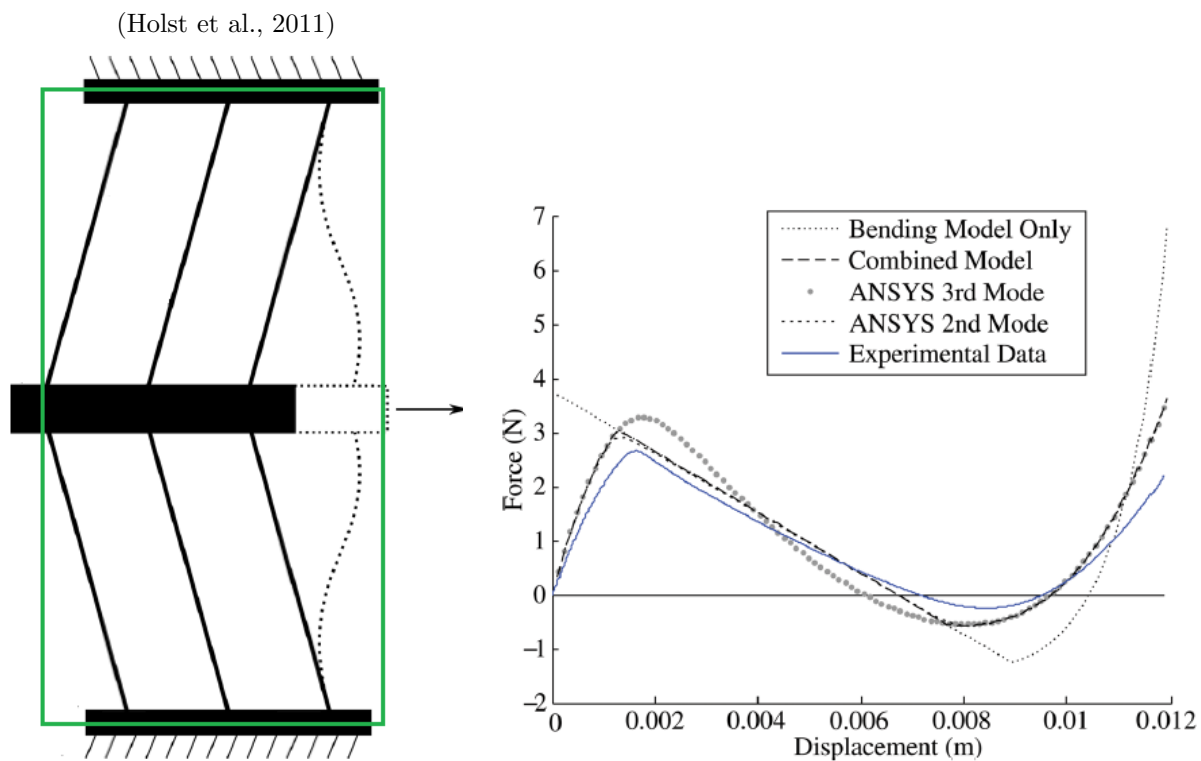
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<i>Author(s)</i>	Pavlovi, Nenad T. and Pavlovi, Nenad D.
<i>Paper</i>	Compliant mechanism design for realizing of axial link translation
<i>Width</i>	3.5mm
<i>Height</i>	9mm
<i>Stroke</i>	2.57mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Lumped

(Todd et al., 2010)

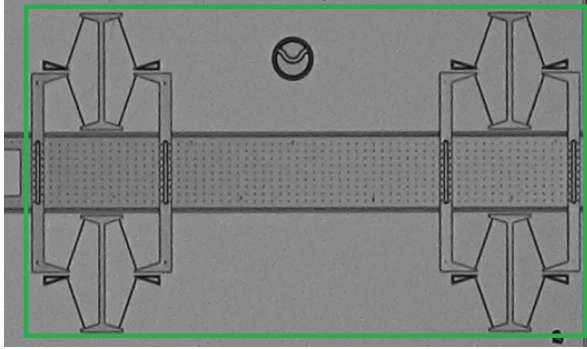


<i>Author(s)</i>	Todd, B. and Jensen, B. D. and Schultz, S. M. and Hawkins, A. R.
<i>Paper</i>	Design and Testing of a Thin-Flexure Bistable Mechanism Suitable for Stamping From Metal Sheets
<i>Width</i>	55mm
<i>Height</i>	6mm
<i>Stroke</i>	3mm
<i>Stability</i>	Bistable, 2 stable points
<i>Compliance</i>	Distributed



<i>Author(s)</i>	Holst, G. L. and Teichert, G. H. and Jensen, B. D.
<i>Paper</i>	Modeling and Experiments of Buckling Modes and Deflection of Fixed-Guided Beams in Compliant Mechanisms
<i>Width</i>	150mm
<i>Height</i>	55mm
<i>Stroke</i>	11mm
<i>Stability</i>	Bistable, 2 stable points
<i>Compliance</i>	Distributed

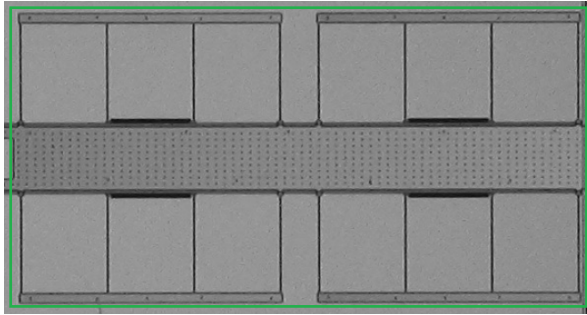
(Mackay et al., 2012) X-bob



No force-deflection figure available

<i>Author(s)</i>	Mackay, A. B. and Smith, D. G. and Magleby, S. P. and Jensen, B. D. and Howell, L. L.
<i>Paper</i>	Metrics for Evaluation and Design of Large-Displacement Linear-Motion Compliant Mechanisms
<i>Width</i>	1.85mm
<i>Height</i>	1.48mm
<i>Stroke</i>	0.5mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Distributed

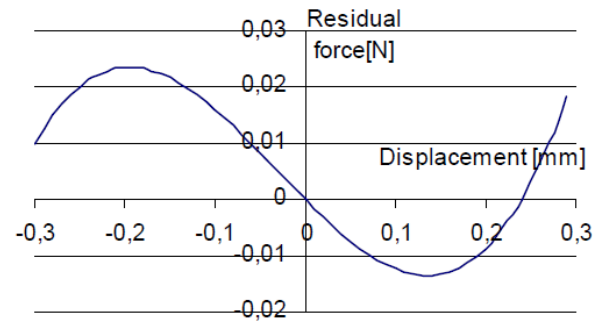
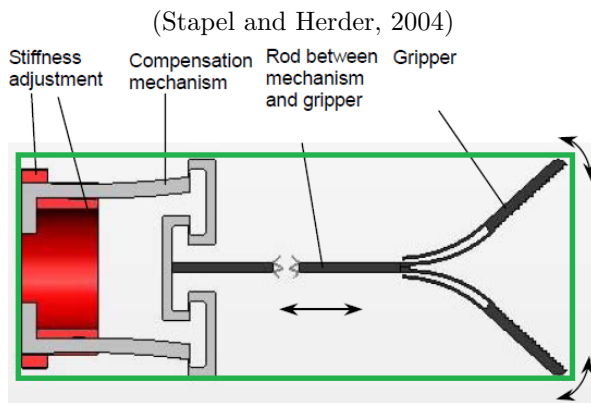
(Mackay et al., 2012) Folded



No force-deflection figure available

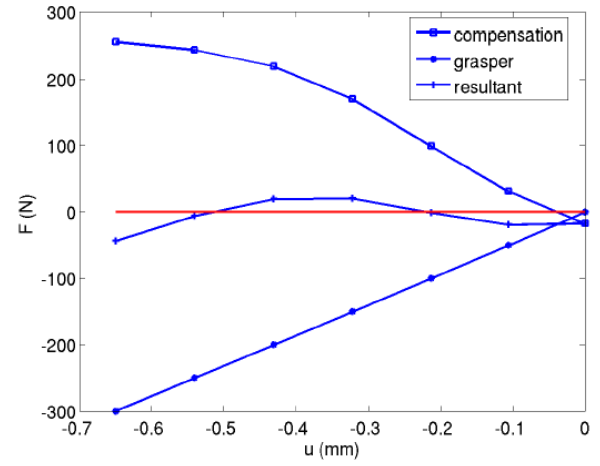
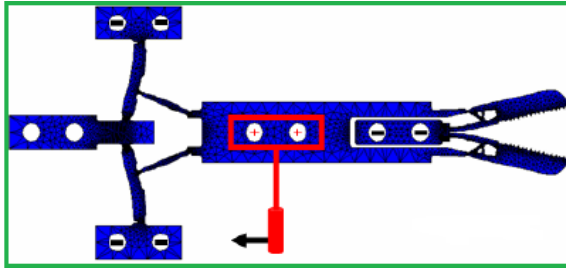
<i>Author(s)</i>	Mackay, A. B. and Smith, D. G. and Magleby, S. P. and Jensen, B. D. and Howell, L. L.
<i>Paper</i>	Metrics for Evaluation and Design of Large-Displacement Linear-Motion Compliant Mechanisms
<i>Width</i>	1.88mm
<i>Height</i>	0.912mm
<i>Stroke</i>	0.5mm
<i>Stability</i>	Stable, 1 stable point
<i>Compliance</i>	Distributed

B Statically Balanced Compliant Mechanisms Data



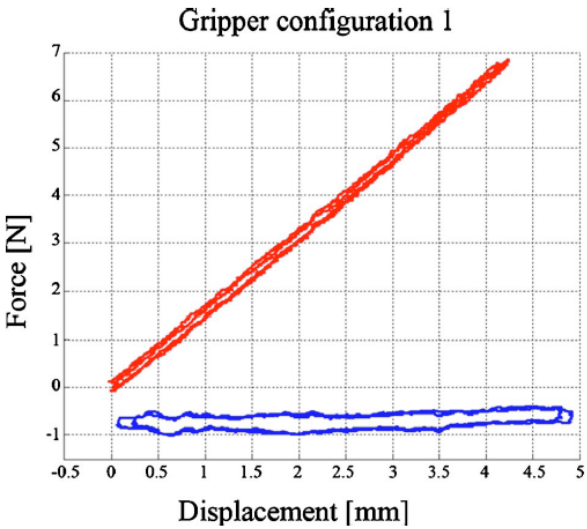
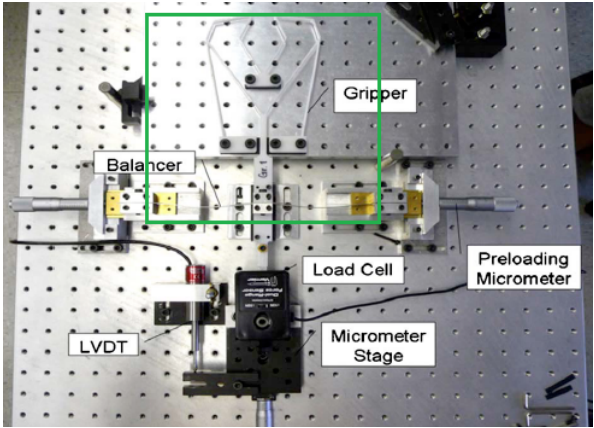
<i>Author(s)</i>	Stapel, A. and Herder, J.
<i>Paper</i>	Feasibility study of a fully compliant statically balanced laparoscopy grasper
<i>Width</i>	23mm
<i>Height</i>	8mm
<i>Stroke</i>	0.56mm
<i>Source of force data</i>	Finite Element analysis
<i>Residual force range</i>	0.037N
<i>Max. compensated force</i>	15N

(Lange et al., 2008)



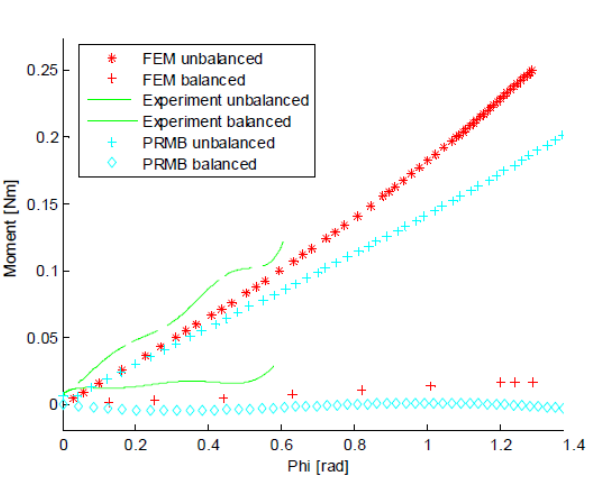
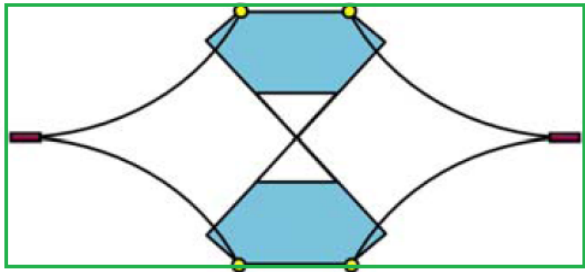
<i>Author(s)</i>	Lange, D.J.B.A de and Langelaar, M. and Herder, J.
<i>Paper</i>	Design of a statically balanced compliant laparoscopic grasper using topology optimization
<i>Width</i>	104mm
<i>Height</i>	59.2mm
<i>Stroke</i>	0.65mm
<i>Source of force data</i>	Finite Element analysis
<i>Residual force range</i>	70N
<i>Max. compensated force</i>	300N

(Hoetmer et al., 2010)



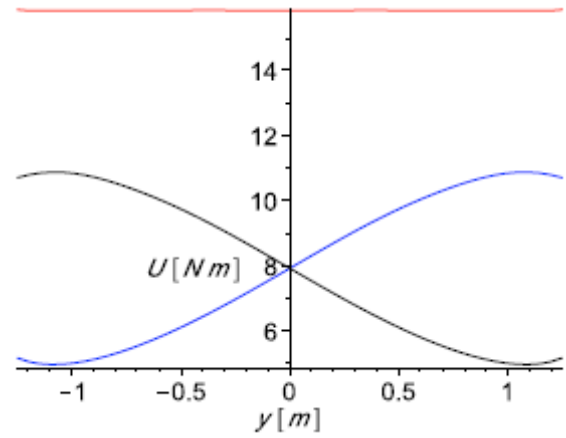
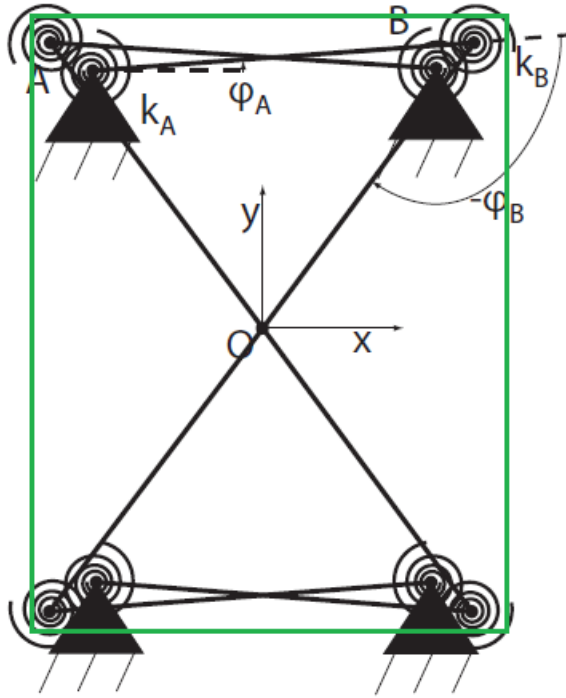
Author(s)	Hoetmer, K. and Woo, G. and Kim, C. and Herder, J.
Paper	Negative Stiffness Building Blocks for Statically Balanced Compliant Mechanisms: Design and Testing
Width	170mm
Height	180mm
Stroke	4.8mm
Source of force data	Testing
Residual force range	0.1N
Max. compensated force	6.9N

(Morsch and Herder, 2010)



Author(s)	Morsch, F.M. and Herder, J.
Paper	Design of a Generic Zero Stiffness Compliant Joint
Width	200mm
Height	40mm
Stroke	1.2rad
Source of force data	Finite Element analysis
Residual force range	0.012Nm
Max. compensated force	0.25Nm

(Rosenberg et al., 2010)



Author(s)

Rosenberg, E.J. and Radaelli, G. and Herder, J.

Paper

An Energy Approach to a 2DOF Compliant Parallel Mechanism with Self-guiding
Statically-Balanced Straight-Line Behavior

Width

1200mm

Height

1800mm

Stroke

2000mm

Source of force data

Pseudo Rigid Body Model

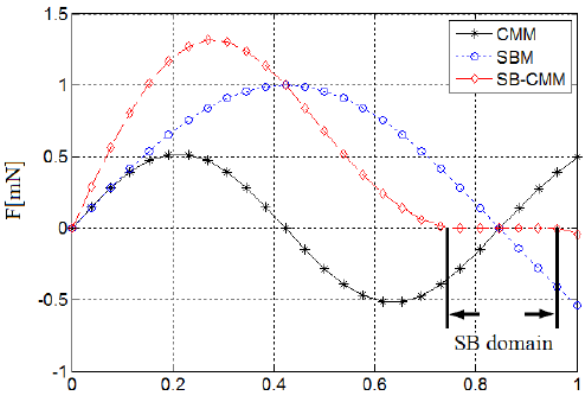
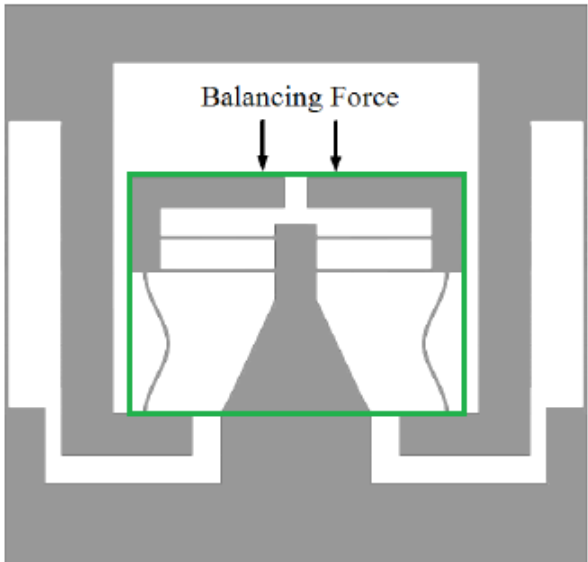
Residual force range

0.12N

Max. compensated force

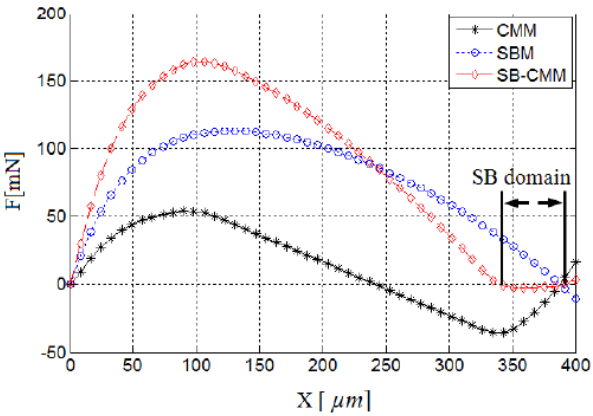
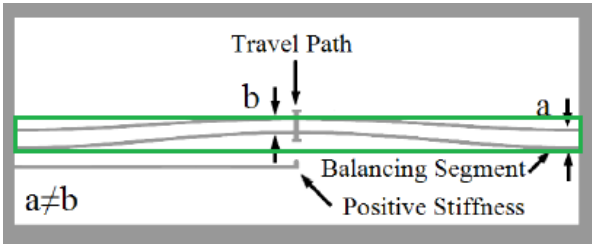
4N

(Tolou et al., 2010) Case I



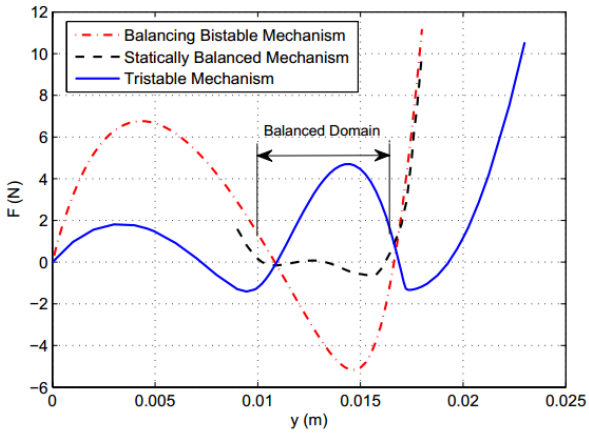
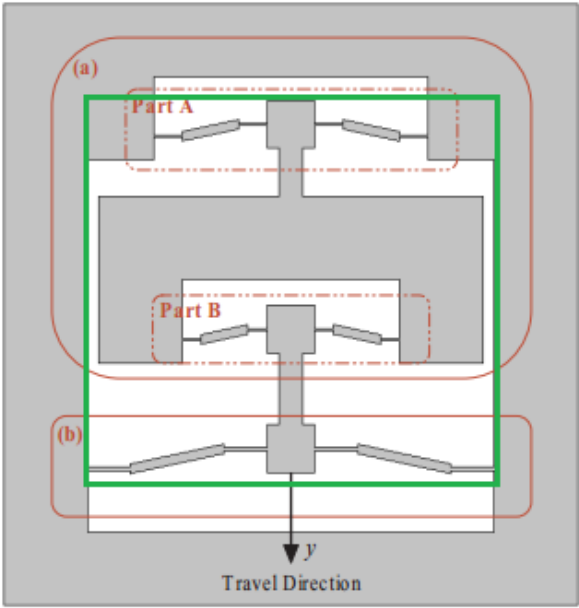
Author(s)	Tolou, N. and Estevez, P. and Herder, J.
Paper	Collinear-type statically balanced compliant micro mechanism (SB-CMM): experimental comparison between pre-curved and straight beams
Width	0.667mm
Height	1.1mm
Stroke	0.05mm
Source of force data	Finite Element analysis
Residual force range	0.0005N
Max. compensated force	0.055N

(Tolou et al., 2010) Case II



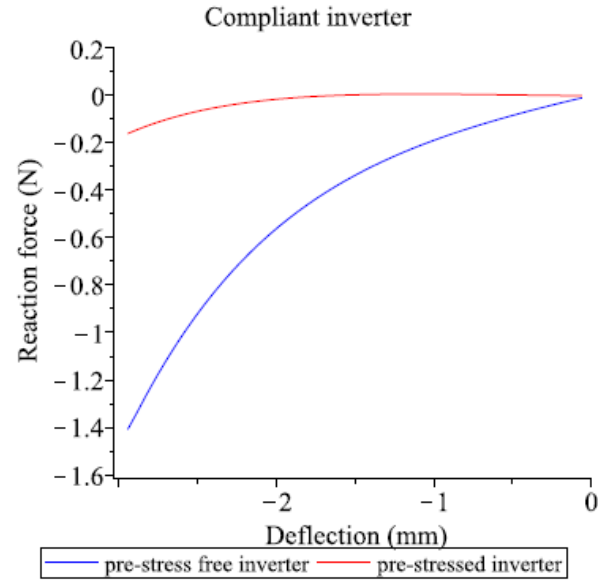
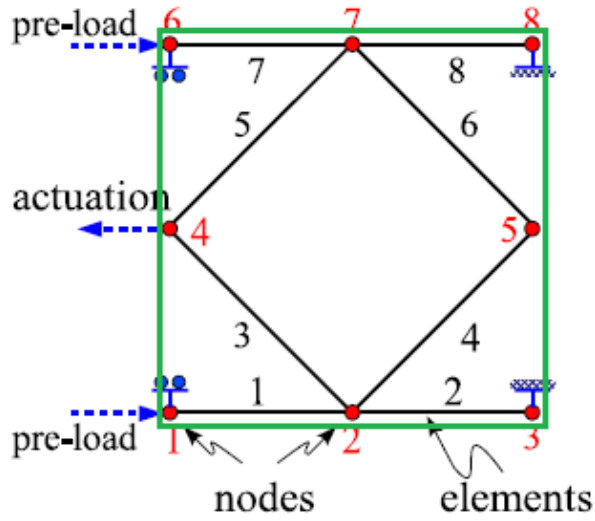
<i>Author(s)</i>	Tolou, N. and Estevez, P. and Herder, J.
<i>Paper</i>	Collinear-type statically balanced compliant micro mechanism (SB-CMM): experimental comparison between pre-curved and straight beams
<i>Width</i>	1.84mm
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<i>Residual force range</i>	0.002N
<i>Max. compensated force</i>	0.035N

(Chen and Zhang, 2011)



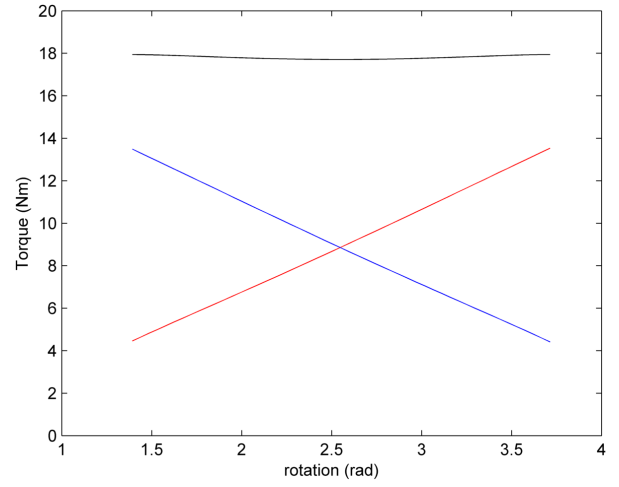
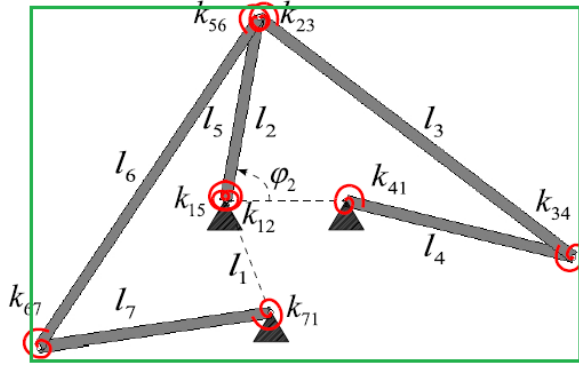
Author(s)	Chen, G. and Zhang, S.
Paper	Fully-compliant statically-balanced mechanisms without pre-stressing assembly: concepts and case studies
Width	177mm
Height	173mm
Stroke	7mm
Source of force data	Finite Element analysis
Residual force range	0.9N
Max. compensated force	7N

(Gallego and Herder, 2011) Case I



Author(s)	Gallego, J. A. and Herder, J.
Paper	Buckling as a new perspective on static balancing of mechanisms
Width	54mm
Height	59mm
Stroke	3mm
Source of force data	Finite Element analysis
Residual force range	0.16N
Max. compensated force	1.4N

(Gallego and Herder, 2011) Case II



<i>Author(s)</i>	Gallego, J. A. and Herder, J.
<i>Paper</i>	Buckling as a new perspective on static balancing of mechanisms
<i>Width</i>	3.3
<i>Height</i>	2.1
<i>Stroke</i>	2.3rad
<i>Source of force data</i>	Pseudo Rigid Body Model
<i>Residual force range</i>	0.235Nm
<i>Max. compensated force</i>	13.53Nm

General Report

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1 Conceptual design

A conceptual design can be constructed in more ways than one. Two options were chosen to get a good sense of what function the escapement should fulfill. The first option is to treat the escapement as a black box, it's called a Top Down approach and will be discussed in section 1.1. The second option is to look at existing designs and acquire their functions and then unite them with a general function that covers all of the different escapements. This approach is called the Bottom Up approach and is used in section 1.2.

1.1 Top Down

The escapement is considered to be a black box with an input and an output. In accordance with the input and output, a main function can be acquired. This general function can be translated into more specific functions on the base of certain cases of the output signal to solve the black box problem. The process is shown in figure 1.1. In addition, its content is discussed in more detail.

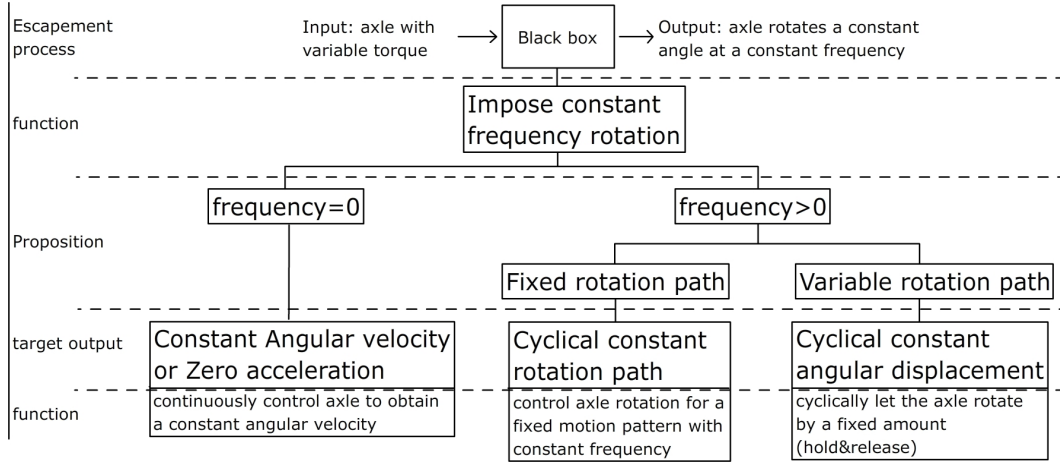


Figure 1.1: Decomposition of the general function for different cases

The blackbox has an input of a torque on an axle, produced by a spring. Because the spring unwinds during operation, the torque decays making the input a variable torque. The output of the system must be an axle that rotates a constant angle at a constant frequency. These angle and frequency are independent of the input torque. A logical conclusion is that the general function of the escapement is to *Impose constant frequency rotation*.

The output frequency can be equal to zero or have any other positive value. By making this distinction, two different cases arise.

- If the frequency is equal to zero, then an instantaneous constant velocity must be maintained. For this case the function changes to '*Continuously control axle to achieve a constant angular velocity*'.
- If the frequency is greater than zero, then the velocity of the axle is not constant, but the angular rotation profile is cyclical. This case can again be separated in two cases; either the speed profile of the axle is controlled and thus constant for every cycle independent of the input torque or

the speed profile is uncontrolled and dependent of the input torque. In both cases the angular displacement remains constant for each cycle.

- If the speed profile is to be fixed for every cycle, the function changes into '*Control axle rotation speed for a fixed motion pattern with constant frequency*'. For this case the speed needs to be controlled so it is not affected by the variable torque from the input
- If the speed profile can be variable for every cycle, the function changes into '*Let the axle rotate by a fixed amount at a constant frequency*'. This case is different because there is no need to monitor or control the angular velocity pattern of the axle.

In the next three sections, the implications of the three functions will be discussed.

1.1.1 Constant Angular Velocity

If the frequency of the output is equal to zero, then the rotation speed must be instantaneously constant during operation. Constant control must be used to maintain zero acceleration of the axle. This control could be feedback or feedforward control.

- *feedback control* - Feedback control adjusts the system based on its current state. This means that however fast the control system may be, it will always need time to adjust and never be fast enough to adjust the rotation speed to the desired instantaneously constant angular velocity.
- *feedforward control* - Feedforward control uses predefined knowledge of the system to acquire its required control behavior. The accuracy of this control is highly dependent on the accuracy of the predefined knowledge of the system. Because there are always unknown effects in any system, feedforward can not control the axle to give an output of an instantaneously constant angular velocity.

This shows that precise control of the angular velocity is not a reasonable option for an escapement mechanism.

1.1.2 Cyclical constant rotating motion

For this option, in addition to controlling the angular velocity, a track of time needs to be kept to ensure a cyclical behavior with a constant frequency. The option of an output with cyclical constant rotation motion is not preferred for the same reason given in section 1.1.1 on the control of the angular velocity.

1.1.3 Cyclical constant angular displacement

To fulfill the output of a cyclical constant angular displacement, there are two elements that need to be controlled. The first element that needs to be controlled is the frequency. The second is the a constant angle of rotation displacement. Notice that this means that only the endpoint of the rotation needs to be secured, the speed or path that the rotation takes to get there is of no importance. Both elements can be controlled and this makes the it a viable option to make an escapement with an output in the form of a *cyclical constant angular displacement*.

1.2 Bottom Up

Current escapement mechanisms can generally be divided into two parts, the anchor and the oscillator. The function of the escapement can be divided into multiple subfunctions for both the anchor and oscillator. These subfunctions have to meet certain criteria for the total system to be a successful escapement. An overview of the different functions and criteria is given in figure 1.2. In addition, the different functions and criteria are discussed in more detail and quantified if needed.

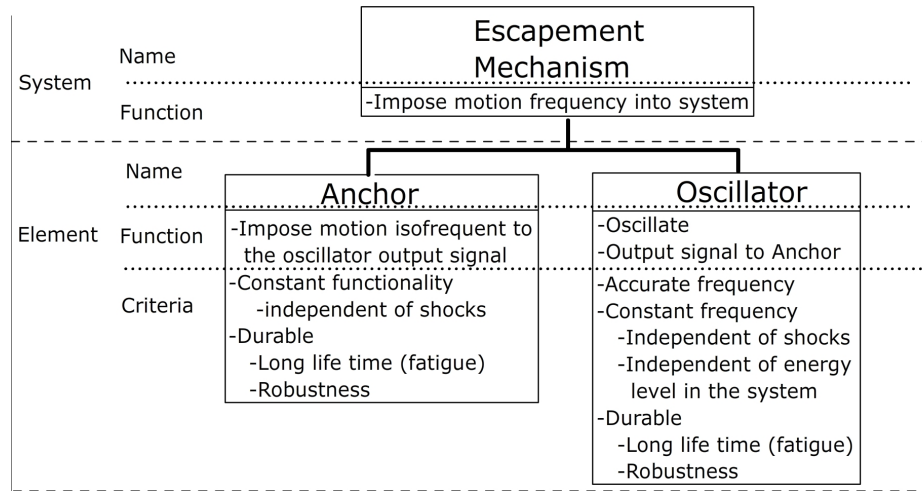


Figure 1.2: Decomposition of the system into functions and criteria

Functions

- *Impose motion isofrequent to the oscillator output signal* - The anchor locks and unlocks the escape wheel at the exact frequency that is indicated by the output signal of the oscillator.
- *Oscillate* - The origin of the frequency of the escapement movement comes from an oscillating mass. This means that the oscillating behavior defines the timekeeping capability of the watch.
- *Output signal* - The oscillator needs to present the oscillating behavior as signal to the anchor in order for the anchor to perform its function at the correct frequency.

Criteria

Criteria	<i>Constant functionality (Independent of shocks)</i>
How to determine	Measure the both the frequency of the imposed motion on the system and the frequency of the signal given by the oscillator over a period of one minute while a shock of 150g is applied to the system and compare the resulting frequencies.
Targets	The measured frequencies must be equal to ensure resilience to shocks
Rationale	The oscillator will be placed inside the casing of a wristwatch. A wristwatch is often subjected to shocks when it falls or by fast movements of the arm. These conditions may not affect the accuracy of the oscillator frequency

Criteria	<i>Accurate frequency</i>
How to determine	Measure the frequency of the oscillator with a high precision stroboscope
Targets	Measure a constant frequency of $4\pm 0.0004\text{Hz}$
Rationale	The accuracy of the watch is dependent on the accuracy of the frequency of the escapement
Criteria	<i>Constant frequency (independent of shocks)</i>
How to determine	Measure the frequency of the oscillator with a high precision stroboscope while an acceleration of 150g is applied to the escapement mechanism
Targets	Measure a constant frequency of $150\pm 0.0004\text{Hz}$
Rationale	The oscillator will be operate inside a wristwatch. A wristwatch is often subjected to shocks when it falls or by fast movements of the arm. These conditions may not affect the accuracy of the oscillator frequency
Criteria	<i>Constant frequency (Independent of energy level)</i>
How to determine	Measure the frequency of the oscillator with a high precision stroboscope over a range of 240 oscillations (1 minute) with a starting energy level of the oscillator from 80% to 120% with steps of 4%. In this case 100% energy of the oscillator is the design aim of the average energy inside the system.
Targets	Measure a constant frequency of $4\pm 0.0004\text{Hz}$
Rationale	The oscillator transfers a signal to the anchor which will take energy that must be replenished. The energy source that supplies this energy may have problems keeping the energy level inside the system perfectly level throughout operation. It is vital that these fluctuations in the energy level of the oscillator does not compromise the accuracy of the frequency
Criteria	<i>Durable (life time)</i>
How to determine	Use a model of the structure in a Finite Element Analysis (FEA) program and run a fatigue analysis to acquire the number of cycles before the survival rate drops below 95%
Targets	The amount of cycles must be higher than $6.3 * 10^8$ cycles (5 years at 4Hz)
Rationale	The average life expectancy of the oscillator should be as high as possible. Five years of continuous operation is adequate.
Criteria	<i>Durable(robustness)</i>
How to determine	Dynamic analysis of the structure using FEM model to determine whether stresses occur that might induce plastic deformation or even fractures occur at shocks of 5000g
Targets	Stresses must remain below the level of plastic deformation throughout the analysis
Rationale	The wristwatch in which the oscillator is placed is subjected to shocks. In addition to keeping its functionality, it is even more vital that no damage is done to the oscillator or its surrounding parts

An overview of the criteria and their subsequent quantifications is shown in table 1.1.

Table 1.1: Overview of the specifications for the escapement mechanism

Criteria	
Accurate frequency	$4 \pm 0.0004 \text{ Hz}$
Independent of shocks	$4 \pm 0.0004 \text{ Hz}$ at 150g acceleration
Independent of energy level	$4 \pm 0.0004 \text{ Hz}$ at energy level of $50\% < E_{design} < 100\%$
Life time	95% chance of survival after $6.3 \cdot 10^8$ oscillations
Robustness	no plastic deformation or fracture at 5000g acceleration

1.2.1 Oscillator

The functions and criteria set for the oscillator in section 1.2 need to be met to obtain a successful design. For each function and criteria, there are multiple ways of achieving this goal. To this end, strategies are proposed in figure 1.3. In addition the strategies are discussed in more detail.

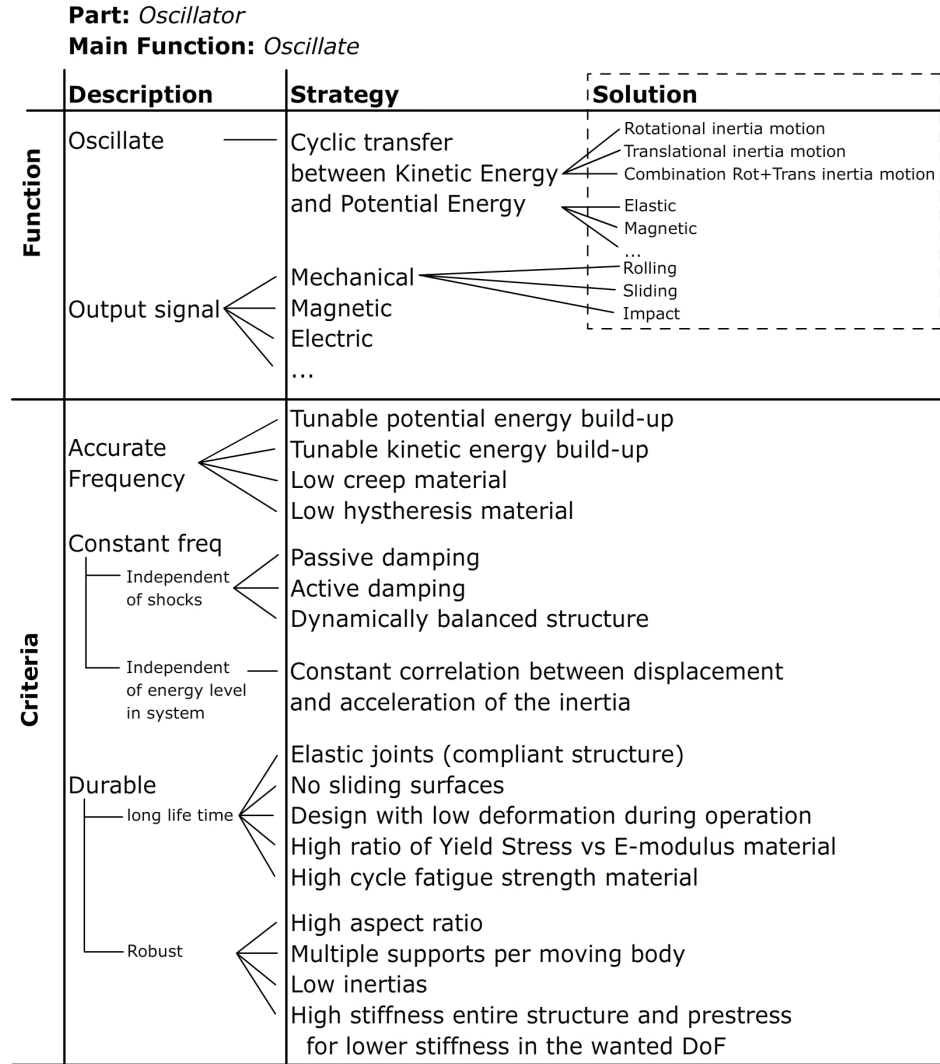


Figure 1.3: Design strategies for the oscillator

Oscillate

- *Cyclic transfer between kinetic and potential E* - An oscillator consists of a potential and a kinetic energy reservoir. If these reservoirs are linked for energy transfer and the initial energy level is unequal to zero, then the energy levels inside the reservoirs will oscillate at a certain frequency. Viable solutions for potential energy reservoirs are elastic, magnetic. Possible kinetic energy reservoirs include rotational and translational inertia motion or a combination of these two.

Output signal

- *Mechanical, Magnetic, Electric* - The signal that carries the frequency signature of the oscillator can be transferred in the form of mechanical connection, magnetic or electric. More options exist but are not viable for this design. A solution space is added for the Mechanical option.

Accurate Frequency

- *Tunable potential energy buildup* - The fabrication process is likely to introduce errors, this issue can be resolved by tuning the energy buildup of the potential energy. For instance, if the potential energy reservoir consists of spring, the energy buildup can be tuned by removing coils, changing its force-displacement behavior.
- *Tunable kinetic energy buildup* - Another way to resolve fabrication errors is to adjust the buildup of kinetic energy. This can be done by adjusting the inertia of the system. For instance, if the kinetic energy reservoir consists of a rotating mass, the energy buildup can be tuned by displacing mass towards or away from the center of rotation, changing the inertia around the rotation axis.
- *Low creep material* - Solid materials have the tendency to deform (creep) under the influence of stresses. If the structure deforms, it is likely that the frequency of the oscillator will be affected. A material with low creep is therefore needed.
- *Low hysteresis material* - Hysteresis during elastic deflection causes energy to dissipate due to internal friction in the material. To keep a constant frequency, it is preferable to keep a constant level of energy inside the system. Therefore a material is preferred with low hysteresis.

Constant frequency (Shocks)

- *Passive damping* - Shocks impose motion on bodies with inertia, thus increasing or decreasing the energy level inside the system. If these shocks are not addressed it is likely that a shock can drain or overflow the kinetic energy reservoir. The transfer of energy to the bodies can be stopped by absorbing the incoming energy with damping by passive elements like isolators.
- *Active damping* - Shocks imposed on the system could be damped by active damping. This means dissipating energy from the system by a controlled actuator.
- *Dynamic balancing* - A 1 DoF structure can be designed in such a way that during the movement of the system, the center of mass stays at exactly the same place. If this system is subjected to a high acceleration shock, then the shock will exert a force on the system, trying to move its center of mass. Because the center of mass can not move in this structure by movement of the individual parts, the system's movements are unaffected by the shock.

Constant frequency (Energy)

- *Constant correlation between displacement and acceleration of the main inertia* - For an oscillator the frequency is independent of the amplitude (or energy level) if the correlation between the inertia and stiffness is constant and independent of time, displacement and velocity. Notice however, this does not explicitly imply that inertia and stiffness have to be constant.

Durable (life time)

- *Compliant mechanism* - Compliant mechanisms are flexible mechanisms that transfer an input force or displacement to another point through elastic body deformation. Compliant structures are known to have higher life span than structures with rigid joints.

- *No sliding surfaces* - Sliding surfaces introduce contribute to wear and thus shorten the life span of the system.
- *Design with low deflection during operation* - High deflections introduce high stresses which influence the number of cycles a structure can withstand.
- *High ratio of Yield stress vs E-modulus material* - A high ratio of yield stress vs E-modulus means that a material can be deflected more before elastic deformation occurs.
- *High cycle fatigue strength material* - A high cycle fatigue strength material can withstand more cycles at higher stresses.

Durable (Robustness)

- *High aspect ratio* - A high aspect ratio gives the structure the ability to withstand higher out of plane forces before plastic deformation occurs.
- *Multiple supports per moving body* - By supporting a body at multiple points, forces can be transfered to the casing of the watch at more points, thus dividing the total force and lower the stress at each support.
- *High stiffness entire structure and pre-stress for lower stiffness in a desired DoF* - A strategy to get a robust structure is to first create a structure that is stiff in all DoFs and then adding negative stiffness in a desired DoF.

1.2.2 Anchor

The functions and criteria set for the anchor in section 1.2 need to be met to get a successful design. For each function and criteria, there are multiple ways of achieving this goal. To this end, strategies are proposed in figure 1.4. In addition the strategies are discussed in more detail.

Part: Anchor			
Main Function: Impose motion isofrequent to the oscillator output signal			
	Description	Strategy	Solution
Function	Impose motion — isofrequent to the oscillator output	Stop, hold and release the escape wheel	
Criteria	Constant funct └ Independent of shocks Durable └ long life time Robust	Passive damping Active damping Dynamically balanced structure Stiffness in positions Kinematically lock and release movement in positions (draw) Low inertia Elastic joints (compliant structure) No sliding surfaces Design with low deformation during operation High ratio of Yield Stress vs E-modulus material High cycle fatigue strength material High aspect ratio Multiple supports per moving body High stiffness entire structure and prestress for lower stiffness in the wanted DoF	

Figure 1.4: Design strategy for the anchor

Impose motion isofrequent to the oscillator output

- *Stop, hold and release* - To impose the correct motion, the anchor should stop, hold and release the escape wheel at a frequency that is equal to that of the signal from the oscillator. There are several ways of implementing this strategy. The most probable is a mechanical or magnetic solution.

Constant functionality

- *Passive damping* - Shocks impose motion on bodies with inertia, thus it can conflict with imposing the right motion on the escape wheel. The transfer of energy to the bodies can be stopped by absorbing the incoming energy with damping by passive elements like isolators.
- *Active damping* - Shocks imposed on the system could be damped by active damping. This means absorbing the shock from the system by a controlled actuator.
- *Dynamic balancing* - A 1 DoF structure can be designed in such a way that during the movement of the system, the center of mass stays at exactly the same place. If this system is subjected to a high acceleration shock, then the shock will exert a force on the system, trying to move its

center of mass. Because the center of mass can not move in this structure by movement of the individual parts, the system is unaffected by the shock.

- *Positive stiffness in positions* - The functionality of the system is dependent on switching between positions. By giving the positions positive stiffness, minor forces on the system will not result in a change of positions but only a small oscillation.
- *kinematically lock and release movement in positions* - The functionality of the system can be partly secured by kinematically restrict movement temporarily in certain positions. In existing escapement mechanisms, this is done by 'draw'.
- *Low inertia* - Shocks transfer less energy to a low inertia body. By keeping the inertias of the anchor low, shocks have less influence on this part.

2 Oscillator design

There are two functions that need to be fulfilled by the oscillator. The first function is to oscillate at a constant frequency and the second is to present an output signal of the constant frequency to the next mechanism.

The output signal is that is given is dependent on the mechanism that is supposed to receive it and also on the oscillation behavior. Also, the accuracy of the complete escapement mechanism is dependent on the accuracy of the oscillator. For those reasons, the oscillating function is considered to be the key function of the oscillator part and the output signal function will be of later importance.

2.1 Oscillation function

In section 1.2 it was already stated that the strategy to fulfill oscillation function is to create '*a cyclic transfer between kinetic energy and potential energy*'. For a wristwatch there are only few viable options for potential and kinetic energy reservoirs. The viable options are listed in table 2.1

Potential energy	Kinetic energy
Elastic energy	Rotational energy
Magnetic energy	Translational energy

Table 2.1: Viable options of potential and kinetic energy reservoirs

To further reduce the solution space, the option of magnetic energy as a reservoir for potential energy is discarded. Although magnetic energy could function inside a wristwatch, the author is specialized in elastically deforming mechanisms.

The remaining option of elastic energy can be further divided in rotational and translational elastic deformation. Four combinations between potential energy and kinetic energy remain as possibilities to fulfill the oscillation function. An overview of these combinations and a simple embodiment is give in figure 2.1. The translational and rotational elastic deformation is illustrated as a normal spring and a torsion spring.

Combinations 1 through 4 have properties that are inherent to their setup shown in fig 2.1. These properties give insight into which combination is the most viable candidate to function as an oscillator inside a wristwatch.

2.1.1 frequency dependency to energy level

Combinations 1 and 4 have a direct connection between the movement of the inertia and the elastic deformation. If elastic deformation force is a linear function of the displacement and the inertia is independent of position and time, then the equation of motion looks like equation 2.1.

$$I\ddot{q}(t) + kq(t) = 0 \quad (2.1)$$

Where I is the inertia, $q(t)$ is the deflection of the elastic deformation and K is a constant. This equation of motion has a solution of the form of formula 2.2.

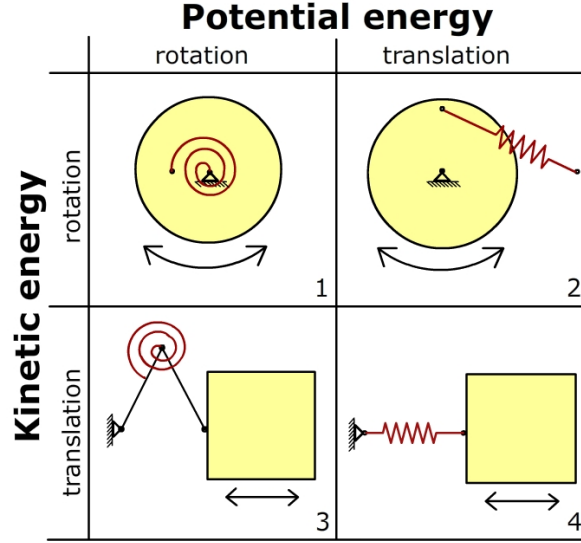


Figure 2.1: Conceptual possibilities to fulfill the oscillation function of the oscillator

$$q(t) = A \sin \left(\sqrt{\frac{k}{I}} t \right) + B \cos \left(\sqrt{\frac{k}{I}} t \right) \quad (2.2)$$

Where A and B are constants, dependent on the initial conditions $q(0)$ and $\dot{q}(0)$. Notice that for this setup the frequency of the oscillator is automatically independent of the initial conditions (energy level) of the system.

For combinations 2 and 3 the transmission between movement of the inertia and the elastic deformation is non-linear. As a result some form of a non-linear differential equation arises as the equation of motion, shown in equation 2.3

$$I\ddot{q}(t) + F(q(t)) = 0 \quad (2.3)$$

Where $F(q(t))$ is the force as a result of the elastic deformation which is a non-linear function of the displacement $q(t)$ of the inertia. For non-linear differential equations the solution can seldom be given explicitly, but the frequency will be dependent of the initial conditions (energy level). In order for these combinations to fulfill the criteria of having a constant frequency independent of the energy level, the inertia and/or stiffness needs to compensate for the non-linearity of the transmission, or a linear transmission needs to be designed

2.1.2 frequency dependency to shocks

For combinations 1 and 2, motion of the inertia does not shift the center of mass for any possible motion. This means in reverse that any shock (that tries to move the center of mass), will not influence the motion of the inertia. This property automatically satisfies the criteria of '*independence to shocks*'.

For combinations 3 and 4, the movement of the inertia does shift the center of mass. This means that shocks in the direction of motion can directly influence the systems movements. To counteract this problem, a configuration of inertias needs to be designed to solve this issue.

2.1.3 Length of stroke for elastic member

For combination 1 and 4 the movement of the inertia is directly coupled to the inertia. Because an elastic member (or compliant mechanism) typically has a limited stroke, the range of motion for the inertia will be restricted. This means that the maximum energy inside for equal inertias will be smaller for combinations 1 and 4, than for combinations 2 and 3. Combinations 1 and 4 must be compensated by either scaling up the inertia, or creating a transmission between the elastic member and the inertia.

2.2 Evaluation

The results of the comparison is shown in table 2.2.

	Kinetic energy*	Potential energy*	f independent of energy	f independent to shocks	Small stroke of elastic member
1	R	R	+	+	-
2	R	T	-	+	+
3	T	R	-	-	+
4	T	T	+	-	-

*R=rotational, T=translational

Table 2.2: Overview of the properties of different combinations between kinetic en potential energy reservoirs.

From this table it can be concluded that combination 1 and 2 have an advantage of the others because for both solutions there is only one criteria that is not satisfied.

- Combination 1 would need to be scaled up or a transmission should be implemented for it to work with elastic members and at the same time be able to keep the same energy level. Scaling of the device is out of the question since there is a size restriction for the design. Adding a transmission would be possible, but it would have to be a constant transmission in order to keep the frequency independent of the energy level.
- To create a constant frequency at different energy levels for concept 2, a non-linear stiffness behavior must be found that in combination with the non-linear transmission creates a linear system, or a linear transmission must be found that can be combined with a linear stiffness to create an overall linear system.

For the reasons stated above, it is decided to design the oscillator with translational elastic potential energy and a rotational kinetic energy.

3 Transmission design

The transmission has the sole function of connecting translational movements and forces of the elastic element to the rotational movements and moments of the inertia wheel. It is assumed that the inertia remains constant throughout rotation. This leaves two possible behavior combinations for the transmission and the elastic element.

- A linear transmission with a linear transfer between rotation and translation, together with an elastic element with a linear relation between force and translation. Together, a linear transmission and spring will automatically create a rotational spring with linear characteristics.
- A non-linear transmission, together with an element with non-linear spring characteristics. The combination of these two elements are designed such that together, they represent a rotational spring with linear characteristics.

The first option has the problem that a perfectly linear spring does not exist. But also the later can become a problem. Although non-linear elastic behavior can be achieved, it is hard and not always possible to achieve the desired non-linear characteristic. It is expected that the best option is to use a slightly nonlinear spring, compensated with a nonlinear transmission. However, no options are excluded at this stage.

3.1 Wrapped Cam Transmission

Single Wrapped Cam mechanism

By continuously varying the radius of a cam, a perfect linear transmission can be realized that can rotate over 180 degrees. A mathematic approach to design these wrapped cams is presented by Tidwell et al. (1994). This approach is refined in Kilic et al. (2012) to be made suitable for the current design problem. With a given force-deflection behavior and a required torque-rotation behavior, a wrapped cam can quickly be designed with the correct transmission. In figure 3.1 an oscillator equipped with such a wrapped cam is shown.

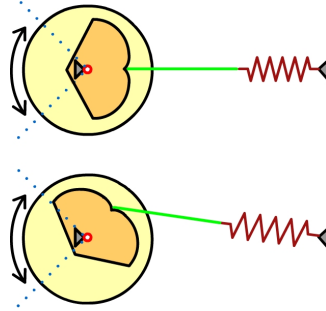


Figure 3.1: Single wrapped cam mechanism

The green line represents the part that wraps around the cam. It is modeled to have negligible bending stiffness. The approach to design the wrapped cam is based on the equation for virtual work (eqn 3.1) and is explained in further detail below.

$$\int_0^u F(u + u_t) du = \int_0^\alpha G(\alpha) d\alpha \quad (3.1)$$

Equation 3.1 requires that the work related to the characteristic function, F , is equal to the work related to the characteristic function, G . If functions F and G are elementary mathematical functions, $u(\alpha)$ can be determined explicitly. Since a mechanism that satisfies $u(\alpha)$ is required, this second scenario is a mechanism synthesis. In figure 3.2 the variables are indicated that are needed to design the cam.

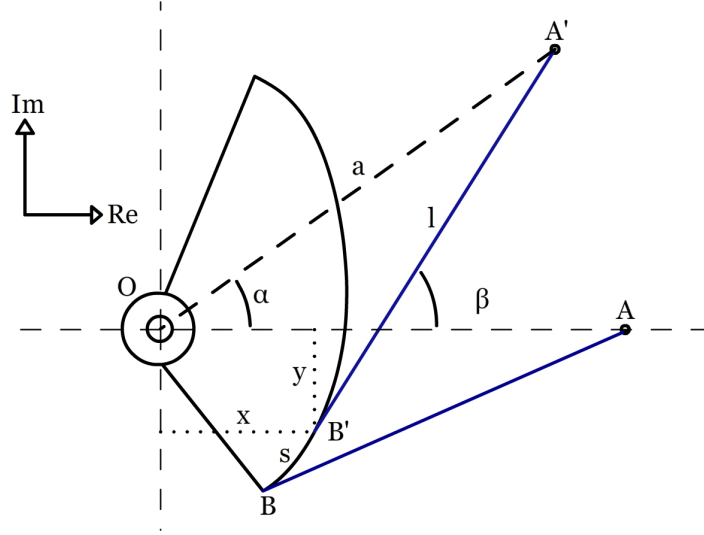


Figure 3.2: The parameters and variables used for the wrapped cam mechanism

In figure 3.2 the attachment for the elastic element is rotated around the rotation axis of the cam. This makes computation to obtain the function for the radius of the cam less complicated. When the cam is rotated by an angle of α , the elongation of the string will be

$$u = s + l - r(\beta - \alpha) - s_0 - l_0 \quad (3.2)$$

where the subscript 0 refers to the initial values of the variables. Due to the tangency condition,

$$dx = ds \cos(\beta) dy = ds \sin(\beta) \quad (3.3)$$

The loop closure equation in complex form is given in equation 3.4 as:

$$x + iy + le^{i\beta} = ae^{i\alpha} \quad (3.4)$$

With these equations, the resulting cam profile can be calculated,

$$\frac{du}{da} = a \sin(\beta - \alpha) \quad (3.5)$$

$$\beta = \sin^{-1}\left(\frac{du}{ada\alpha}\right) + \alpha \quad (3.6)$$

$$l = a \cos(\beta - \alpha) / \left(\frac{1}{a \cos(\beta - \alpha)} \frac{d^2 u}{d\alpha^2} + 1 \right) \quad (3.7)$$

$$x = a \cos(\alpha) - l \cos(\beta) \quad (3.8)$$

$$y = a \sin(\alpha) - l \sin(\beta) \quad (3.9)$$

The rotation of the inertia elongates the elastic member and initially also rotates the line along which the elastic member exerts force. This means that rotation of the inertia increases both the force exerted by the elastic member and also the arm that the force makes with respect to the rotation point of the inertia. For small rotations of the inertia, the buildup of torque by the arm multiplied by the force of a non pre-stressed elastic element, is not high enough for reasonable radii of the cam. For that reason, the elastic element needs to be pre-stressed in order to achieve linearity over the entire range of motion.

This transmission has the added benefit of being able compensate for nonlinearities in spring behavior.

Double Wrapped Cam mechanism

To avoid the need for pre-stress, a double wrapped cam mechanism can be used. This wrapped cam has a constant radius which creates a constant transfer between the force of the elastic element and the torque it creates around the inertia's axis. In figure 3.3 an oscillator equipped with such a wrapped cam is shown.

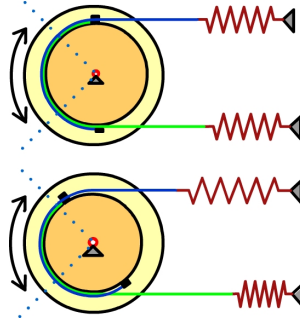


Figure 3.3: Double wrapped Cam Mechanism

Because the green wrapping element has negligible bending stiffness, it will automatically buckle if it is subjected to compression forces. For that reason an antagonistic pair of springs is needed to acquire a moment in both directions.

3.2 Linkage Transmission

The conversion of a linear motion to a rotation motion can be achieved by a linkage mechanism. Unlike the wrapped cam mechanism, there is no method to directly determine the best transmission. Optimization is used to come up with solutions. At this point, the exact characteristics of the spring are not yet known. It is well known that the transmission of any given linkage is likely to be nonlinear. For that reason, the aim of this linkage design is set to have a linear transmission.

3.2.1 Concept Generation

The linkage concept generation was conducted with the help of Gruebler's equation (eqn 3.10).

$$F = 3(n - 1) - 2l \quad (3.10)$$

Where:

- F = total degrees of freedom in the mechanism
- n = number of links (including the frame)
- l = number of lower pair joints (one degree of freedom)

In general, Gruebler's equation is used to determine the number of degrees of freedom for a linkage, but it can also predict what combination of number of links and joints result in a system with a required degrees of freedom. If the elastic element is represented by a prismatic joint and the system can have only 1 degree of freedom, then there are limited possibilities in the total number links and joints. In table 3.1 the possible combinations are listed with an increasing number of links.

Table 3.1: Possible number of joints and links for a 1DoF system

n (=Links)	l (=Joints)	F (=DoF)
2	1	1
4	4	1
6	7	1
8	10	1
10	13	1

The number of possible linkage setups increase exponentially. For a six bar linkage with one prismatic joint, there are already 19 viable setups. For that reason, only four and six bar linkages are considered.

The two possible setups for a fourbar linkage are shown in fig 3.4.

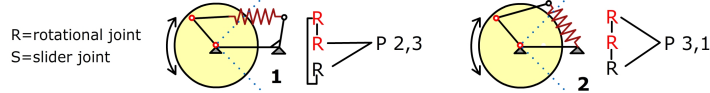


Figure 3.4: Linkage with four links and four joints

There are many possibilities for a six bar linkage but not all are viable for this design problem. An approach was used to guarantee a complete set of concepts. First a chain of only rotational (R) joints is established, then a dyad with the remaining prismatic (P) and rotation joints is added. In figures 3.5, 3.6 and 3.7 the resulting linkages are shown.

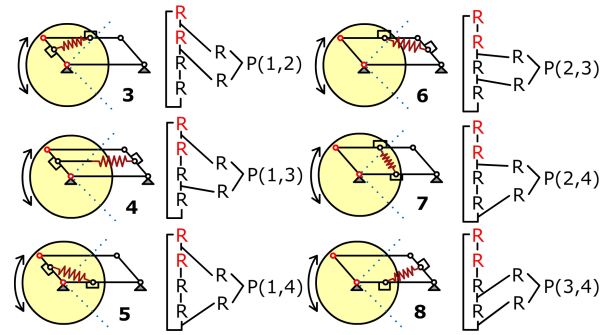


Figure 3.5: Four bar loops with a -R-P-R- Dyad

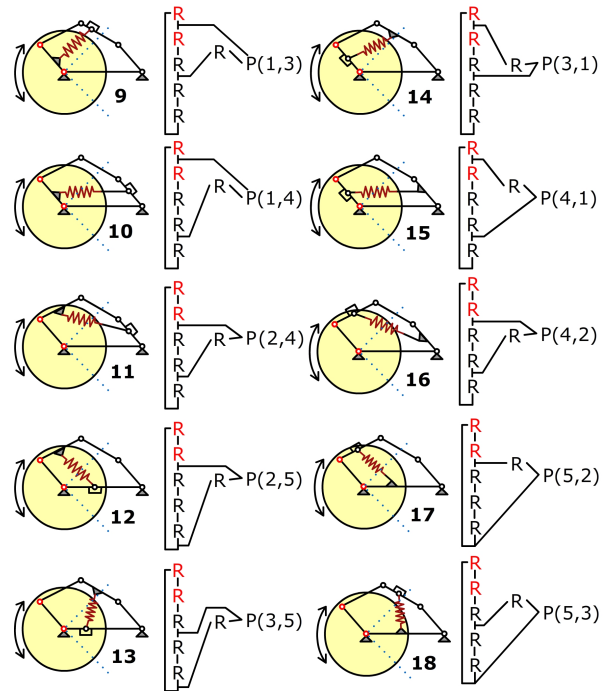


Figure 3.6: Five bar loops with a -R-P- Dyad

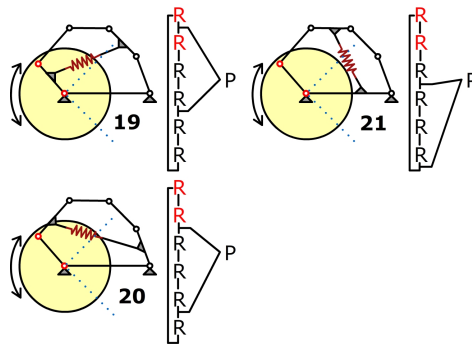


Figure 3.7: Six bar loops with a -P- Dyad

3.2.2 Evaluation

To select appropriate candidates for optimization, the linkages are evaluated based on their theoretical maximum suitability for this design project.

- *Linearity of Transmission* - Achieving linearity between rotation and translation is the primary objective for this transmission. A linear transmission secures a constant frequency independent of the amplitude. The linkages need to be optimized before the exact linearity of the transmission can be acquired. However, it is possible to estimate what the maximum linearity theoretically can be achieved for each concept.
- *Max Angular Rotation (excluding Inertia Axis)* - Compliant joints typically have a restriction on the maximum rotation before plastic deformation occurs. Lower maximum rotations result in a more viable compliant counterpart. The concepts are evaluated by this criterion in a theoretical sense.
- *Constant Inertia* - The inertia of the total system may become non-constant if the spring element moves a lot during operation. If the inertia is non-constant, then a linear transmission no longer secures a constant frequency independent of amplitude. This criterion evaluates whether the elastic element does not move, rotates or rotates and translates during operation.

In table 3.2 the concept linkages are listed and judged according to criteria. In table 3.2, a positive sign indicates that the concept is superior for a criterion, a zero that it is average and a negative sign means that it is inferior compared to other concepts.

Table 3.2: Comparison of the different four and six bar linkages

	Linearity	Angular Deflection	Constant Inertia	total
1	-	-	-	---
2	-	-	+	--+
3	+	+	-	++-
4	+	+	-	++-
5	-	+	0	-+0
6	+	+	-	++-
7	-	+	0	-+0
8	+	+	0	++0
9	+	+	+	+++
10	-	-	+	--+
11	+	+	-	++-
12	-	-	0	--0
13	+	+	0	++0
14	+	+	0	++0
15	-	-	0	--0
16	+	+	-	++-
17	-	-	+	--+
18	+	+	+	+++
19	-	-	+	--+
20	-	-	-	---
21	-	-	+	--+

From table 3.2 it can be seen that the six bar linkages 9 and 18 are best suited for this design problem. Notice that 9 is the kinematic inverse of 18. Because the elastic element moves less in concept 18, its counterpart 9 is discarded. To ensure the inferiority of the simpler linkages, linkage 2 was also included in the optimization procedures.

The chosen linkages can be constructed with three different degrees of compliant design: non compliant, partially compliant and fully compliant design. For each category, an optimization routine was used to obtain appropriate dimensions for the links.

3.3 Rigid Body Linkage

For the non-compliant mechanism, rigid body joints are used for every rotational joint. Only the elastic element remains compliant in order to store the potential energy. This means that the maximum angular rotation is no longer a constraint, thus for this optimization, the only objective is the linearity of the transmission.

The optimization was done on a dynamic model of linkages implemented in Matlab. The inputs to the model define the geometry of the linkage and the output is the correlation coefficient between rotation of the inertia and linear deflection of the elastic element. By using the dynamic instead of static modeling, the singularities of the linkages are easily identified. Because the high amount of variables and also the unpredictability of the result, a genetic algorithm optimization is used.

After the optimization, the linearity of the transmission was not strong enough to create a pure harmonic oscillator. This is due to the fact that the transmission remains an approximation of a linear system, but is unable to exactly match its behavior. For that reason the optimization is adjusted to check the frequency at five different initial amplitude. Although this approach decreases the speed of optimization, it yields better results. The aim for the frequency was to obtain a oscillation with a frequency of 0.5 hz for every initial amplitude. In figure 3.8 the neutral stance of the resulting four bar linkage is shown. In figure 3.9 the behavior of this linkage is presented.

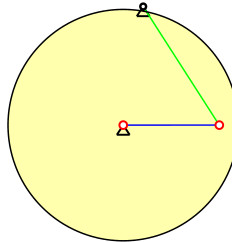


Figure 3.8: Appearance of the optimized four bar linkage. The red circles indicates a rotational joint attached to the inertia wheel. The black circles indicate a rotational joint. The black triangle indicates a fixation to the outside world. The green line indicates the elastic element that functions as a prismatic (sliding) joint.

From figure 3.9 it can be seen that the storage of potential energy is not symmetric around the amplitude of zero rad as would be the case for a harmonic oscillation. This means that the transmission between the rotation and translation is not constant throughout the rotation of the inertia. In figure

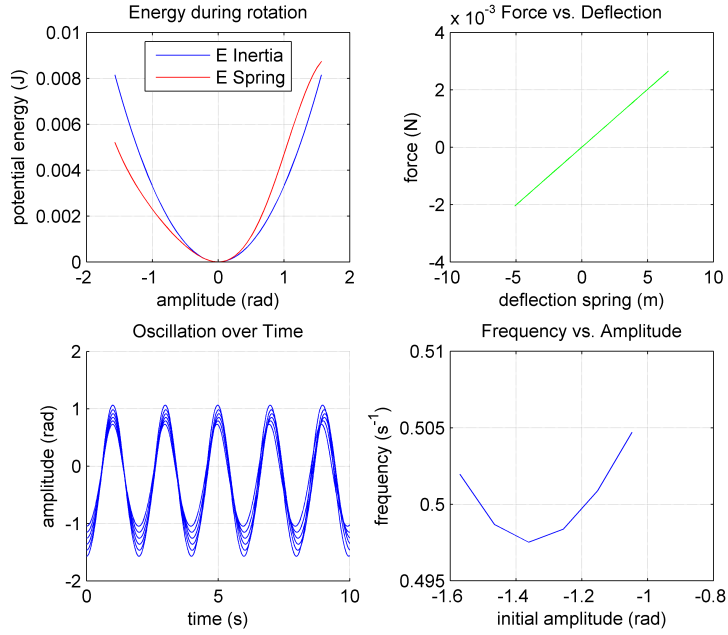


Figure 3.9: Performance of optimized four bar linkage. The standard deviation of the mean frequency is $2.70 \times 10^{-3} s^{-1}$. This achieved by having a non-symmetric oscillation caused by a non-symmetric, but on both sides linear, transmission.

3.10 the transmission between rotation of the inertia and translation of the spring is shown. It can be seen that on both sides there is indeed an approximate linear correlation between rotation and translation, but not with an equal slope which causes the unsymmetrical oscillation.

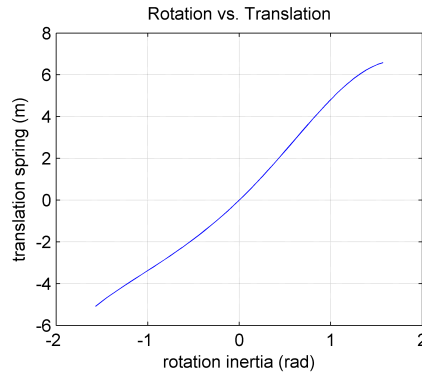


Figure 3.10: Correlation between rotation of the inertia and translation of the spring connection for the optimized four bar linkage

In figure 3.11 the neutral stance of the resulting six bar linkage is shown. In figure 3.12 the behavior of this linkage is presented.

From figure 3.12 it can be seen that the storage of potential energy is again not symmetric around the

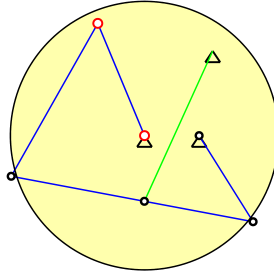


Figure 3.11: Appearance of the optimized six bar linkage. The red circles indicates a rotational joint attached to the inertia wheel. The black circles indicate a rotational joint. The black triangle indicates a fixation to the outside world. The green line indicates the elastic element that functions as a prismatic (sliding) joint.

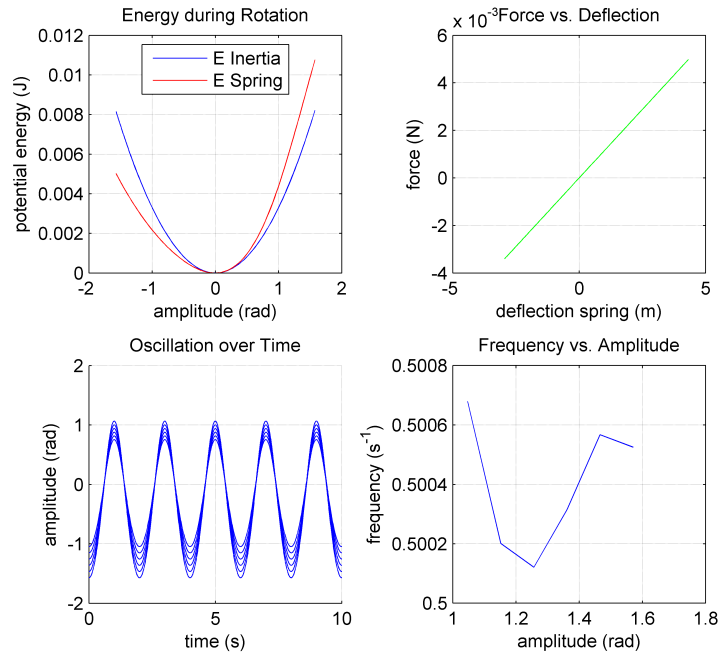


Figure 3.12: Performance of optimized six bar linkage. The standard deviation of the mean frequency is $0.222 \times 10^{-3} s^{-1}$ which is lower than for the four bar linkage (fig 3.8). This achieved by a non-symmetric oscillation caused by a non-symmetric, but on both sides linear, transmission.

amplitude of zero rad. In figure 3.13 the correlation between rotation of the inertia and translation of the spring is shown. It can be seen that the linearity on both sides is more strongly than for the four bar linkage. This causes the frequency to be more constant for the six bar linkage.

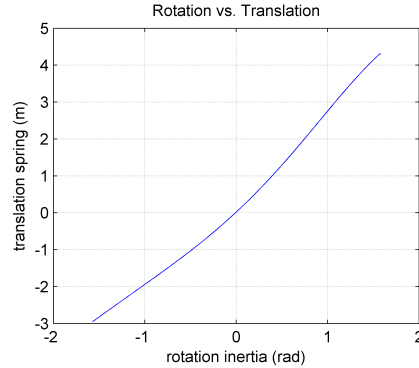


Figure 3.13: Correlation between rotation of the inertia and translation of the spring connection for the optimized six bar linkage

The optimization procedure shows that the evaluation of the linearity criterion in table 3.2 was correct. As predicted the six bar linkage based on concept 18 shows a much more linear correlation between translation and rotation then the four bar linkage based on concept 2.

3.3.1 Measurements and Validation

To test the validity of the matlab model, a large scale prototype was built which could show could prove the linearity of the transmission for the six bar linkage design of figure 3.11.

If it can be shown that the transmission matches the results of the optimization, the oscillation will be assumed harmonic. This assumption makes the measurements easier. Instead of measuring oscillation over time where damping creates an error, a force deflection measurement can suffice where friction creates a less significant error. An overview of the setup is shown in figure 3.14.

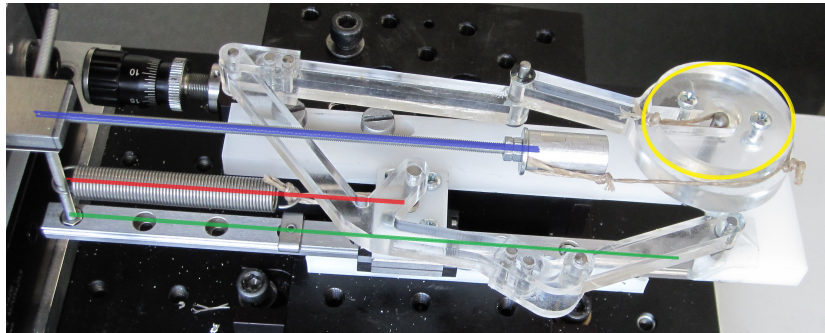


Figure 3.14: The measurement setup used to validate the Matlab model.

The spring element (red) with a constant stiffness is connected in such a way that it is pre-stressed a minimal amount at an amplitude of 1,5 radians. To maintain a pure translation along a fixed axis, the attachment to the linkage is put on a slider (green) parallel to both the required path and the spring. The inertia is replaced with a constant radius cam (yellow). The rope around the cam is pulled and both the force and deflection are measured with with sensors (blue) connected to Labview.

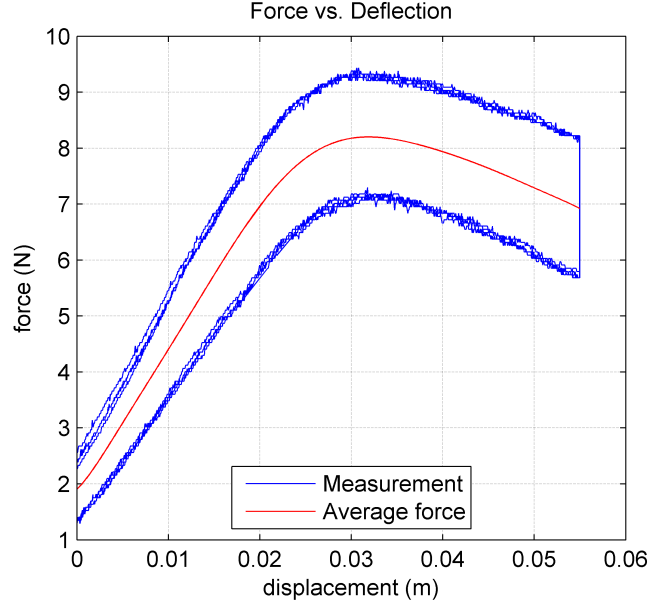


Figure 3.15: The measurement data of 3 consecutive two way measurements of the transmission and the average fit

To show the linearity of the transmission the rotation of the inertia and the deflection of the spring element need to be calculated. The rotation of the inertia can be calculated with the displacement of the rope and the radius of the cam. The the law of conservation of energy (eqn 3.11) allows the calculation of the displacement for the spring.

$$\int_0^a u_{rope} f_{rope} du = \frac{1}{2} u(a)^2_{spring} k_{spring} \quad (3.11)$$

From the data in figure 3.15 the energy level inside the spring can be calculated for different rotation amplitudes and also the transmission can be plotted. In figure 3.16 the result of the measurement is shown together with the results from the matlab simulation.

There is a deviation visible between the two signals, but it is likely due to fabrication error or incorrect spring data. The measurements therefor shows that the Matlab code is valid.

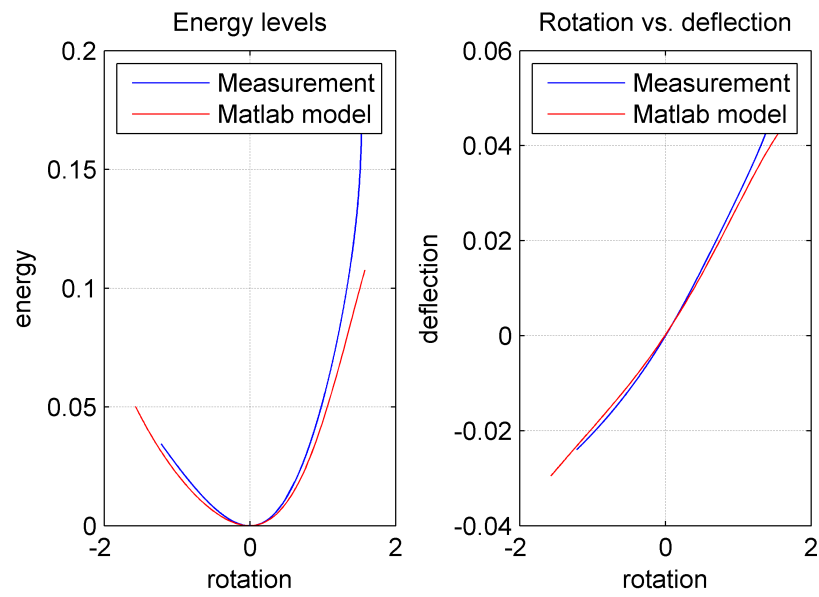


Figure 3.16: Comparisson between measurements and Matlab model. This figure shows the validity of the Matlab model. Deviations are likely caused by manufacturing errors

3.4 Partially Compliant Linkage

A partial compliant version of linkage 18 from figure 3.6 can be superior to the rigid linkage in terms of weight, number of parts and possibly even accuracy. A linkage is optimized wherein all the joints indicated in green in figure 3.17 are designed with compliant joints.

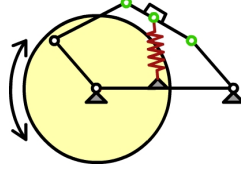


Figure 3.17: Indication of the compliant joints, indicated in green, for a partially compliant linkage

These joints were chosen because they move through space unattached to the housing of the watch or the inertia wheel. This means that the inertia of the joints generally doesn't add linearly to the movement of the system and thus influences the constancy of the frequency. By making these joints compliant the inertia of the joints decreases and reduces the error they pose on the system.

The optimization procedure used in section 3.3 is expanded to include the behavior of the compliant joints. The compliant joints are modeled using a technique called the Pseudo Rigid Body Model (PRBM) described by Howell and Midha (1994). This technique replaces the compliant joint with a rigid link joint and a torsion spring. This returns the optimization of a compliant linkage to the optimization of a rigid body linkage with added torsion springs which is computationally less demanding compared to Finite Element Analysis (FEA).

Because the stiffness and the pretension of all the compliant joints can be added to the optimization procedure to increase the number of variables, the design space increases. However, since compliant joints also have limitations on the maximum angular rotation, the design space also decreases. It is therefore unknown whether a solution exists that can match the performance of the rigid linkage design.

3.4.1 Balancing Strategies

There are several strategies to deal with the compliant joints' stiffness within the system. The strategies can be described in terms of the combined stiffness that the compliant joints transfer to the rotation joint of the inertia. Three strategies are listed in table 3.3 together with the consequence that the strategy imposes.

Table 3.3: Strategies for partially compliant linkage transmission

Stiffness at Inertia Wheel	Residual Rotation Stiffness	Influence on linearity
Low Positive	$k_{init} > k_{res} > 0$	low
High Positive	$k_{res} < 0$	high
Low Negative	$k_{res} > k_{init}$	low
High Negative	$k_{res} \gg k_{init}$	high

- **Stiffness at Inertia Wheel** - The stiffness in the compliant rotational joints is modeled to have a constant positive stiffness. Each joint has a transmission to the rotation axis of the inertia

wheel. Thus, the constant stiffness of the joint can be modeled as a continuous stiffness at the rotation axis of the inertia wheel. The height and sign of this stiffness is considered to purely be the result of the stiffness of the compliant joint, its transmission and the pre-stressing of the joints.

- **Residual Rotation Stiffness** - The stiffness that the compliant joints transfer to the rotation axis of the inertia wheel has to be taken into account for the placement of the compliant prismatic joint. The implied stiffness of the compliant rotation joints can be subtracted from the stiffness that was initially required to get a harmonic oscillation with the correct frequency.
- **Influence on linearity** - The transmission between the compliant rotation joints and the axis of the inertia wheel is likely to be non-linear. This means that the required residual rotation stiffness will become non-linear as well. This added non-linearity is likely to be successfully modeled by the compliant prismatic joint.

From table 3.3 it can be seen that having a high positive or negative stiffness influence of the compliant rotational joints results in a high influence on the linearity of the transmission which is undesirable.

A high residual stiffness generally implies that the stiffness of the prismatic joint is higher as well. For an oscillator at the scale of a watch, the stiffness scales down slower than the inertia. This means that the prismatic compliant joint will be relatively stiffer. Therefore a required high residual stiffness is considered to be advantageous. It is however not clear if a low negative stiffness is easily transferred from compliant rotation joints positive.

For these reasons, a partially compliant linkage with a transfer of low positive and low negative stiffness from the rotation joints will be optimized

The stiffness of the compliant rotational joints is implemented dimensionless as the ratio with respect to the required stiffness of the total system. Optimizations were run for different ratio's ranging from 0.2 to 2. With these values, the residual stiffness required at the rotation axis of the inertia wheel complies with the bounds set in table 3.3.

3.4.2 Positive Stiffness Rotation Joint Transmission

The goal of the optimization is to obtain a the partially compliant linkage with a linear transmission and where the positive stiffness of the compliant joints have a positive stiffness with respect to the rotation axis of the inertia, preferably with low rotation angles of the compliant joints.

For an optimization where the maximum rotation angles of the compliant joints are low, the neutral angles of the compliant rotation joints are automatically placed in the middle of the total range of rotation. In most cases this corresponds to the position where the inertia wheel is at an angle of zero degrees. Therefore, in general no additional measures need to be taken to ensure that the compliant rotational joints transfer a positive stiffness to the rotational axis of the inertia wheel. Special cases will be discussed at the end of this section.

In figure 3.18 the optimized partially compliant linkage is shown. Notice the similarity between the partially compliant linkage (fig 3.18) and the rigid body linkage (fig 3.11). This linkage was found with a stiffness ratio k of 1.6 and a maximum rotation of 28 degrees.

The performance of this linkage is shown in figure 3.19

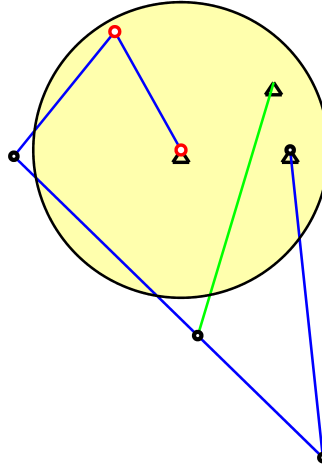


Figure 3.18: Geometry of the optimized Partially Compliant Six Bar Linkage. The maximum rotation of the compliant joints is 28 degrees.

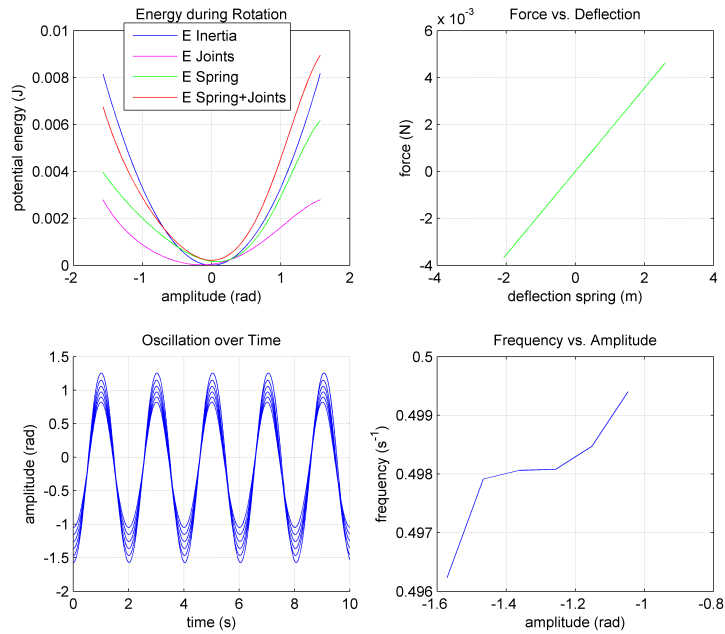


Figure 3.19: Performance of the optimized Partially Compliant Six Bar Linkage. The standard deviation of the frequency is $1.032 \times 10^{-3} s^{-1}$.

3.4.3 Negative Stiffness Rotation Joint Transmission

In figure 3.20 the optimized partially compliant linkage is shown where the optimization forced the transmission to have negative stiffness. Notice from figure 3.21 that because of the negative stiffness of the transmission, the maximum energy of the xbob joint had to increase.

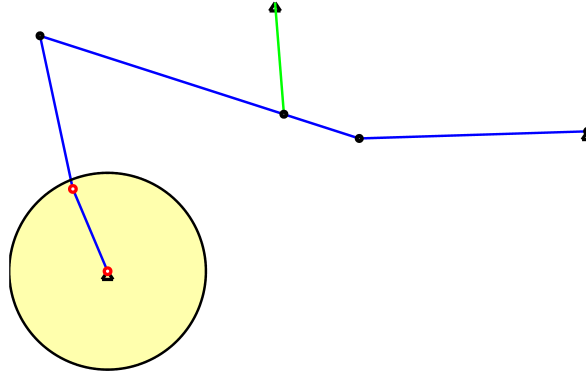


Figure 3.20: Geometry of the optimized Partially Compliant Six Bar Linkage. The maximum rotation of the compliant joints is 19 degrees.

The performance of this linkage is shown in figure 3.21

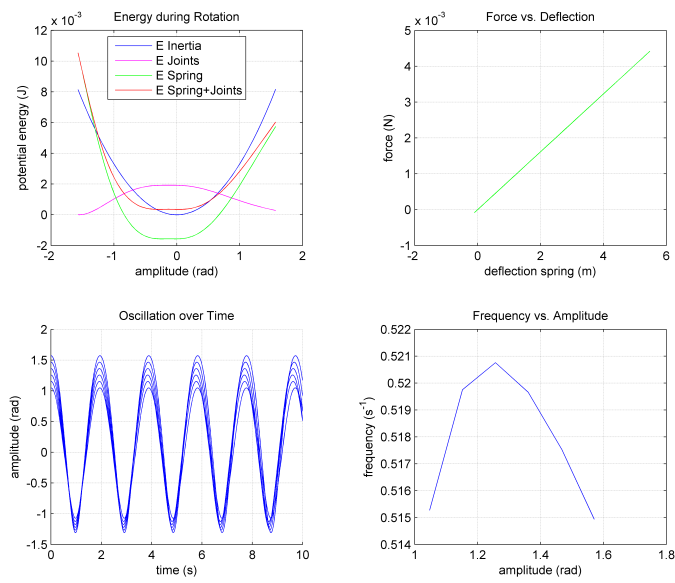


Figure 3.21: Performance of the optimized Partially Compliant Six Bar Linkage. The standard deviation of the frequency is $2.468 * 10^{-3} s^{-1}$.

3.5 Fully Compliant Linkage

The revolute joint of the inertia undergoes a large rotation making it difficult to convert into a compliant joint. During operation, this high angular rotation is divided amongst all the other joints. This means that the revolute joints in the fourbar linkage based concepts from figures 3.5 and 3.4 will in general undergo higher rotations than the five and six bar linkages from figures 3.6 and 3.7.

Even though it is possible to design fully compliant linkages based on these concepts, no solutions were found that show enough potential for a good oscillator. Either the frequency was highly dependent on the amplitude, or the angular rotation of a revolute joint would exceed 30 degrees. By creating a finer grid for the optimization solution space, no significant improvements were made. Even though it cannot be ruled out that a solution exists, the optimization was not successful in finding one. For that reason, a fully compliant design was no longer pursued.

3.6 Conclusion

Although a transmission involving wrapping cam mechanisms presents the chance to make a perfect oscillator in theory. Building such a system at the scale of a wristwatch component brings many problems. Continuing with the linkages could yield good oscillators. The use of compliant members is superior to revolute joints. Also, the fabrication of compliant mechanisms has already been successful for even smaller scales than is required for wristwatch applications. For instance, Shie and Huang (2010) was successful in the fabrication of a fully compliant mechanism to grip individual cells.

The performance of these compliant oscillators will depend on the impact of the conversion from a linkage to a compliant mechanism. But it also depends on the improvement that can be realized by additional optimization of the compliant design with the help of FEA programs such as ANSYS (2011). These factors depend mostly on the linkage that is chosen for the conversion and the variables chosen for the post-conversion optimization.

Additional optimization of the frequency variation can hopefully increase the performance of the compliant design. However, the basic kinematics will remain the same and thus the optimization will have little improvement for the maximum rotation.

For the reasons stated above, the choice was made to continue with a partially compliant transmission. A solution was chosen where the angular deflection is low but the performance is still mediocre. It is unknown if the performance can be improved with the post-conversion optimization. The selected linkage is shown in figure 3.22. The performance of this linkage is shown in figure 3.23.

This linkage has the advantage of a maximum angular deflection of only 12 degrees. This low value is much more suited for this step because it makes the conversion to a compliant mechanism smoother since compliant joints with lower maximum angular deflections can be used.

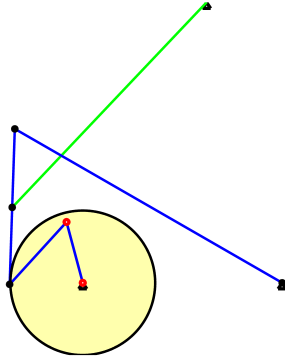


Figure 3.22: Geometry of the optimized Partially Compliant Six Bar Linkage. The maximum rotation of the compliant joints is 12 degrees.

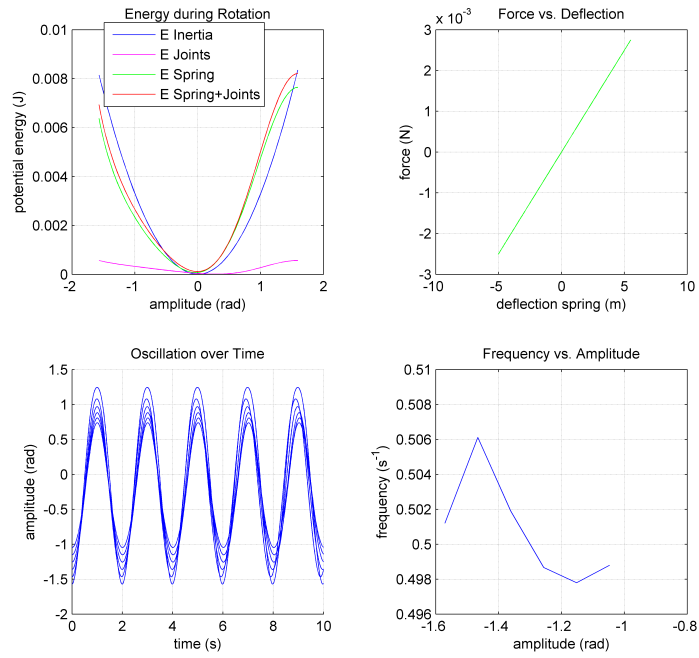


Figure 3.23: Performance of the optimized Partially Compliant Six Bar Linkage. The standard deviation of the frequency is $3.0716 \times 10^{-3} s^{-1}$.

4 Compliant Prismatic Joint selection

The potential energy for the compliant oscillator is partially stored in a compliant prismatic joint. Several successful designs have already been proposed in the Literature Report.

However, there are only two designs that exhibit a large stroke to size ratio which is desirable for this design project. The first option is the folded suspension, proposed by (Saggere et al., 1994) and is shown in figure 4.1a. The second option is the Xbob Joint, based on the Roberts' straight line mechanism. It was proposed by (Hubbard et al., 2004) and is shown in figure 4.1b.

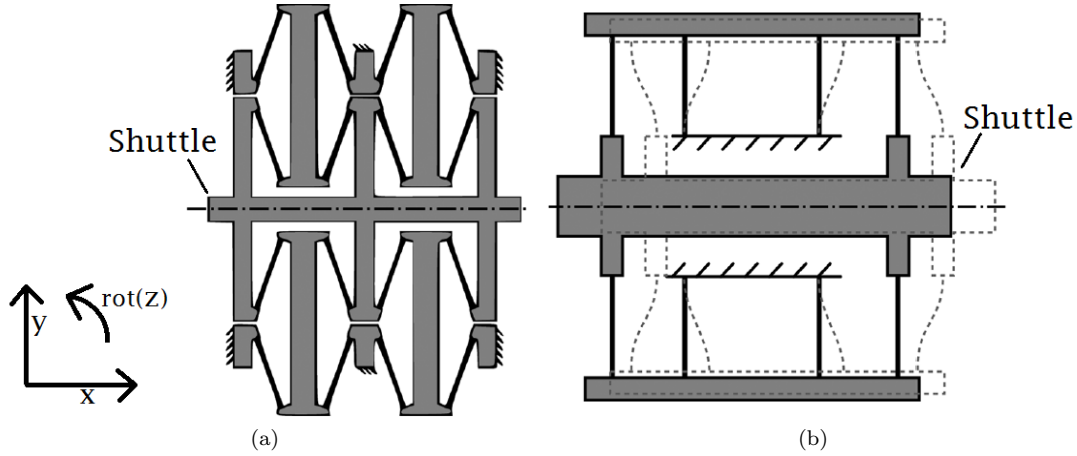


Figure 4.1: Two compliant prismatic joints: Folded Suspension (a) and Xbob (b). The shuttle is the part of the joint that moves relative to the ground.

4.1 Criteria

To make a choice between these two options, both joints were analyzed for their deflection properties. The first significant property is the linearity between force and deflection. In section 3 mechanisms were designed with a linear transmission. If the joint has linear force deflection properties ($Corr(U_x, F_x) = 1$), then the overall oscillation behaves as a linear oscillator. To measure the linearity, the Pearson product-moment correlation (PPMC) is used. This correlation shows if the relation between two variables is linear (PPCM = 1) or if there is no linear correlation at all (PPCM = 0).

Compliant prismatic joints but also compliant joints in general do not derive their degrees of freedom from kinematic constraints like rigid body joints do. For a compliant joint the degrees of freedom are established by a high ratio of stiffness of the undesired direction divided by the desired direction. If this ratio is high then the undesired direction is no longer considered a degree of freedom and it makes the joint function more robust against shocks.

Both the folded suspension and the Xbob are two dimensional designs. In two dimensions, there are three degrees of freedom, namely : x , y and rotation along the axis orthogonal to the plane ($rot(z)$). A prismatic joint should only have one translational degree of freedom, therefore the joint should show high $\frac{k_y}{k_x}$ and $\frac{k_{rot(z)}}{k_x}$ ratios. Although the ratio of $\frac{k_{rot(z)}}{k_x}$ is not dimensionless, it is valid to use this

metric to compare two joints with equal size. A summary of the criteria that were used to evaluate the joints is given in table 4.1

Table 4.1: criteria for the evaluation of the compliant prismatic joints

Criteria	Metric	Goal
Linear force deflection behavior e.g. constant stiffness	$PPMC(U_x, F_x)$	$= 1$
High ratio of offaxial to axial stiffness	$\frac{k_y}{k_x}$	$= High$
High ratio of rotational to axial stiffness	$\frac{k_{rot(z)}}{k_x}$	$= High$

4.2 Analysis

In order to chose the most suitable joint, both joints have been modeled in ANSYS (2011) (a commercial FEA program). The joints were scaled to have the same surface area. The undeformed and deformed model of both the Xbob and the Folded Suspension are presented in figure 4.2 respectively.

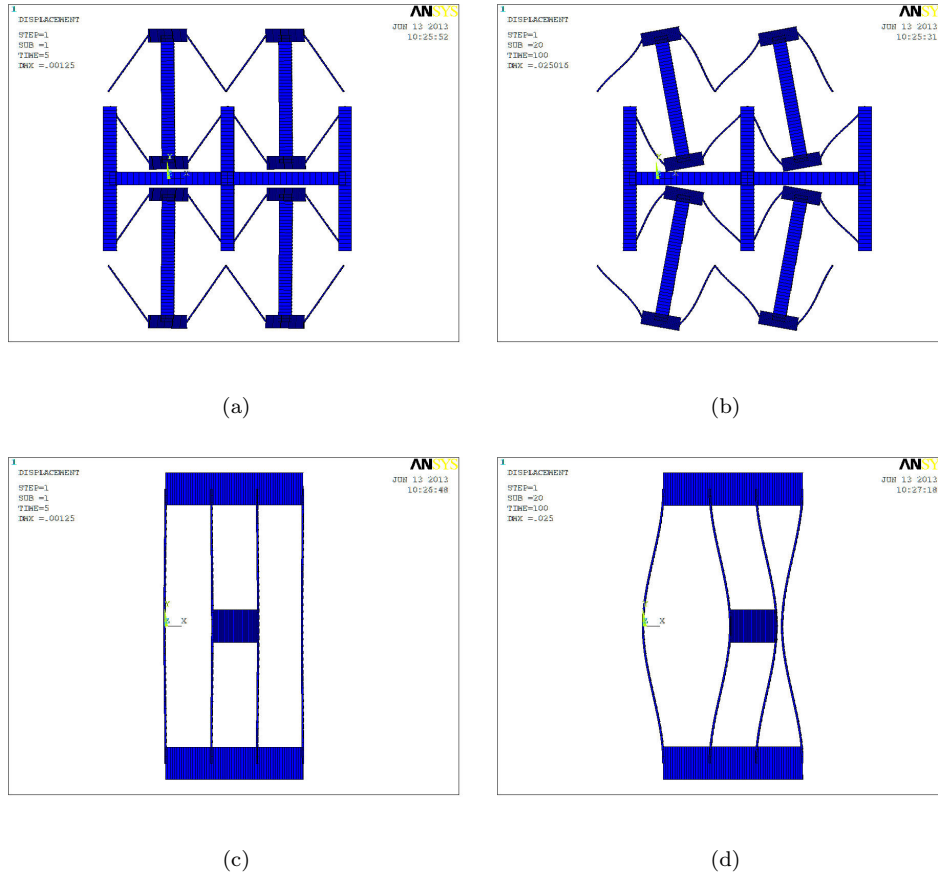


Figure 4.2: Ansys model of the Xbob (a,b) and the Folded Suspension (c,d)

The stiffness in the undesirable degrees of freedom is not constant over the entire stroke of the mechanism. For that reason, the off-axial and rotational stiffnesses were derived over the entire stroke of the mechanism. This was done by displacing the shuttle (indicated in figure 4.1) in an undesirable degree of freedom and measure the required force to maintain this displacement over the entire stroke of the shuttle. The results are shown in figure 4.3.

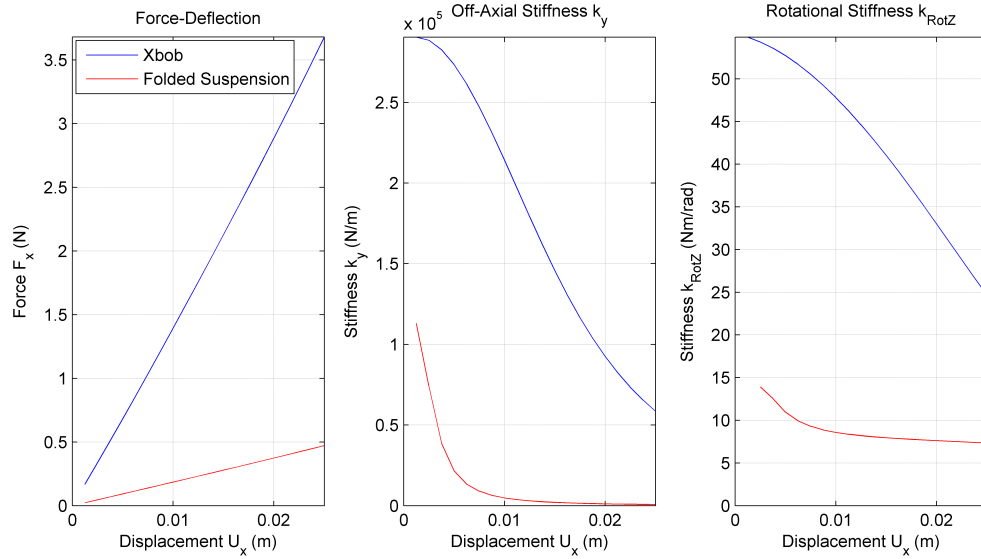


Figure 4.3: Results of the FEA of the Xbob and Folded Suspension joints

4.3 Evaluation

From figure 4.4 it can be seen that for both the joints, there is a strong linear correlation between the displacement U_x and F_x . The Pearson product-moment correlation is give in table 4.2.

From the results shown in figure 4.4, the metrics for evaluation from table 4.1 can be established. To acquire the stiffness ratios, at each horizontal position of the shuttle (U_x), the stiffness of the undesirable degree of freedom is divided by the stiffness of the desirable degree of freedom.

It can be seen that for both joints the relative stiffness $\frac{k_y}{k_x}$ drops rapidly as the displacement U_x increases. For that reason, the minimal value for the relative stiffness ratio gives the best indication of the robustness of the joint. The minimal values of the stiffness ratios for both joints are presented in table 4.2.

Table 4.2: Result for the evaluation metric for the prismatic joints

Metric	PPMC(U_x, F_x)	$\min(\frac{k_y}{k_x})$	$\min(\frac{k_{rot(z)}}{k_x})$
Xbob	0.999743	359.54	0.151
Folded Suspension	0.999953	36.56	0.371

From table 4.2 it can again be seen that there is a very strong linear relation between the force F_x and displacement U_x . The minimal off-axial stiffness ratio shows a significant difference between the two

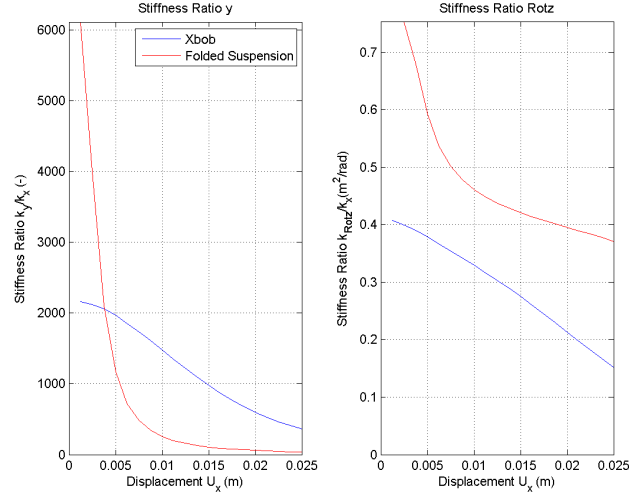


Figure 4.4: Stiffness ratios for the Xbob and Folded Suspension joints

joints of nearly a factor 10 in favor of the Xbob. For the minimal stiffness ratio, the Folded Suspension has an advantage of a factor 2.5.

This result does not depict a clear winner in terms of the metrics presented. However, because off-axial robustness is of high importance for this joint, both for the functional quality and the robustness of the joint and also because the relative rotational stiffness will increase when the design is scaled down, the Xbob is chosen for the final design of the oscillator.

5 Compliant Linkage

5.1 Method

The transmission linkage was chosen in section 3.6 and the prismatic compliant joint was selected in section 4. The resulting mechanisms are shown in figure 5.1.

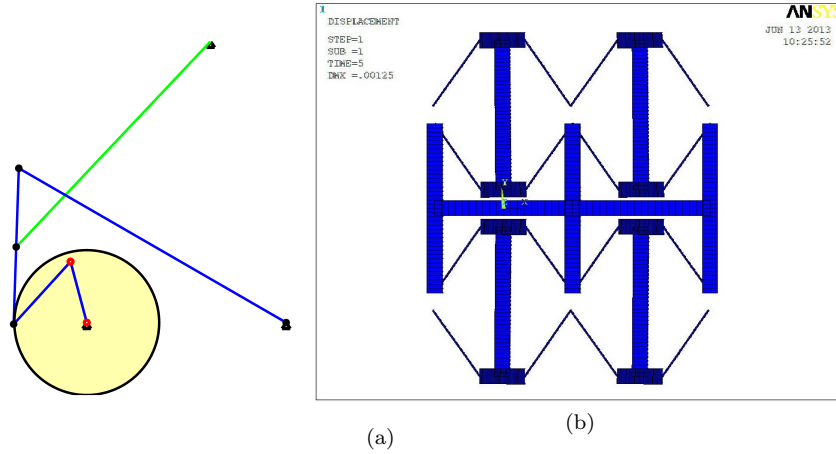


Figure 5.1: The resulting transmission linkage (a) and the prismatic compliant Xbob joint (b)

To create one compliant structure, the transmission linkage must be converted to a compliant mechanism. In figure 5.2 the process is shown for one revolute joint.

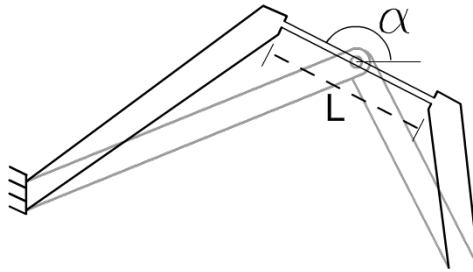


Figure 5.2: Conversion from linkage system to compliant mechanism

A thin straight elastic element is placed through each revolute joint that must be converted into a compliant joint. The ends of these beams are connected according to the kinematics of the initial linkage.

Both the length (l) of this element and the angle (ϕ) at which it is placed will be optimized. This optimization procedure uses interfacing between Matlab (which runs the optimization algorithm) and Ansys (which calculates the behavior of a candidate solution).

The virtual centers of rotation for each compliant joint will not deviate far from the kinematics of the linkage. For that reason, the angular deflection for each joint will not differ much from the values

calculated for this linkage. Thus it is no longer needed to regard the maximum angular deflection as a variable for the optimization.

To keep control of the maximum stress inside the mechanism during operation, the lower bound for the length of the elastic element is chosen to still allow the maximum angular deflection in the mechanism without surpassing the yield strength. This method was used because the time it takes to evaluate a compliant solution is much longer then for the rigid body linkages. By reducing the number of objective, the overall time needed for the optimization decreases to compensate.

The only remaining objective for this optimization procedure is minimization of the variation of the frequency at different initial amplitudes. This is modeled by first calculating the force-deflection and torque-rotation behavior of the proposed compliant mechanism and then use matlab to run a dynamic analysis with this data to check the oscillation frequencies at different initial amplitudes. The process is depicted in figure 5.3.

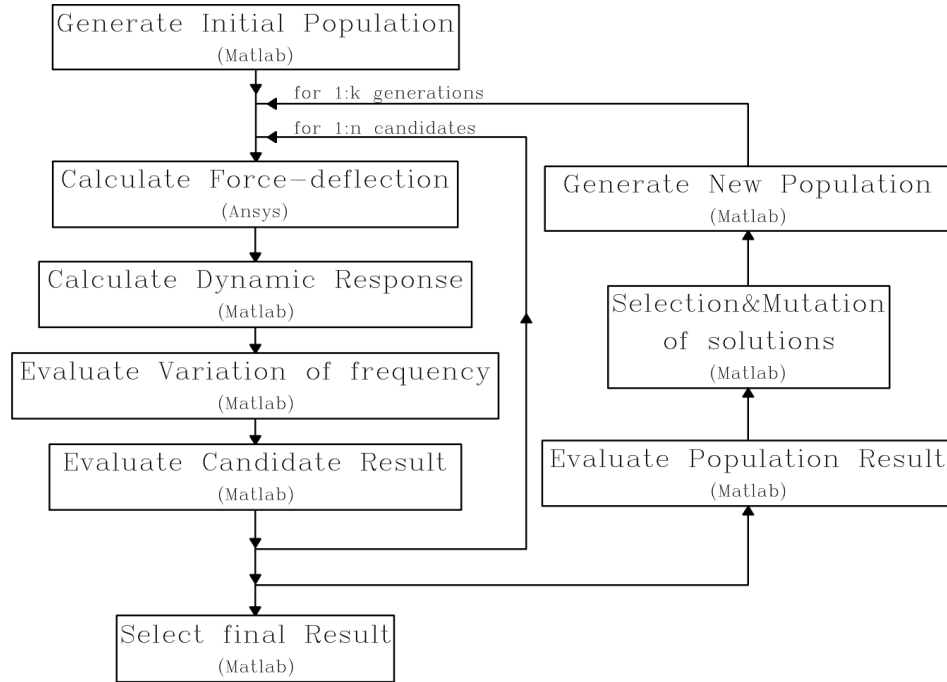


Figure 5.3: Optimization procedure for the Compliant Oscillator

5.2 Results

From all the solutions that were generated, a solution was selected. This selection was done based on the minimal length of the elastic elements present in each solution. For easy fabrication purposes a solution was chosen that has the longest length. The selected solution is shown in figure 5.4. The Xbob was modeled as a single section with a double dept, thus creating mimicking the original Xbob joint, but with requiring less nodes for the simulation.

The Behavior of this mechanism is shown in figure 5.5. In figure 5.5 the non-linearity of the Xbob Joint is visible. The performance has decreased from the previous oscillators.

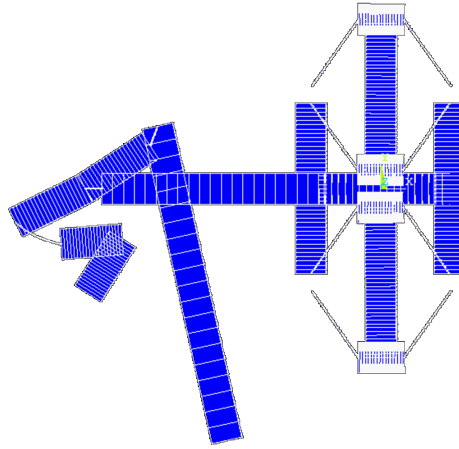


Figure 5.4: Optimized Compliant Oscillator

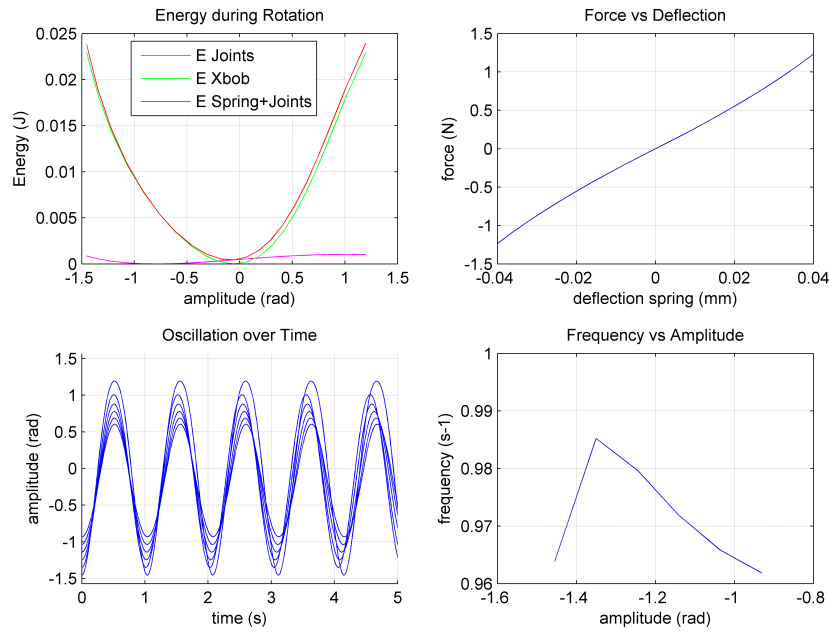


Figure 5.5: Performance of the optimized Compliant Oscillator. The standard deviation of the frequency is $9.635 \times 10^{-3} \text{ s}^{-1}$.

In figure 5.4 it can be seen that this mechanism has overlapping surfaces. This is resolved by rerouting some of the rigid parts of the mechanism. The final result for fabrication is shown in figure 5.6

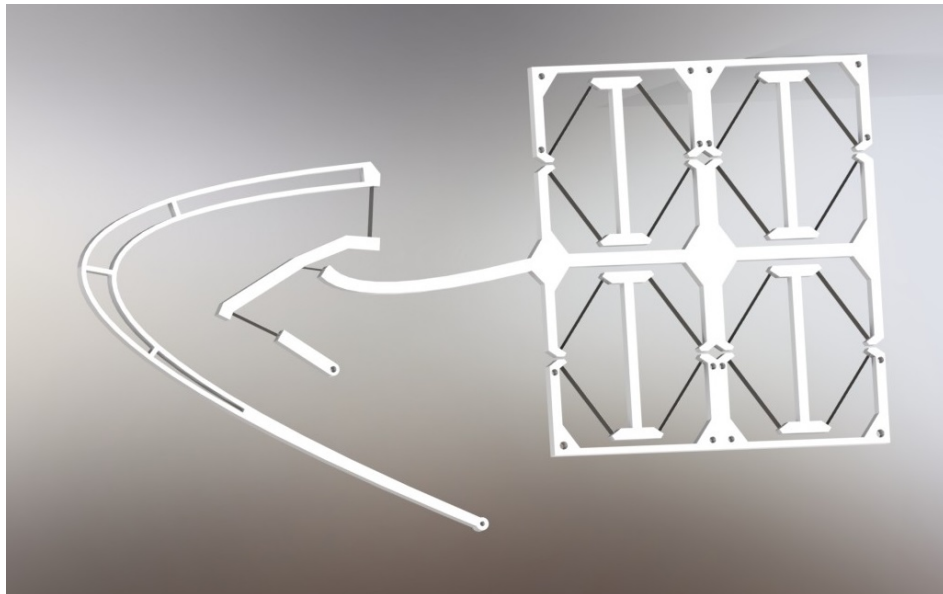


Figure 5.6: Final Design for fabrication of a Partially Compliant Oscillator

6 Fabrication and Testing

6.1 Fabrication

The design that is used for fabrication is discussed in section 5.2. For fabrication, several options exist in terms of material and method. The options that were considered are listed in table 6.1

Table 6.1: Materials and Fabrication methods that are considered for fabrication

Method	Material	Advantage	Disadvantage
Laser cutting	PolyPropylene	High Yield Strength vs. Young's Modulus ratio	Facility not available at TUDelft
	PolyEthylene	Medium Yield Strength vs. Young's Modulus ratio	High Hysteresis, cutting at TUDelft only up to 3mm thickness
	PolyCarbonate	High Yield Strength vs. Young's Modulus ratio	Facility not available at TUDelft
	Acrylic	Facilities available, Cheap	Low Yield Strength vs. Young's Modulus ratio
Stereo lithography	VeroBlue RGD840	Medium Yield vs. Young's Modulus ratio	High Young's Modulus, High Hysteresis
	DurusWhite RGD430	High Yield vs. Young's Modulus ratio	Material expensive and not available at TUDelft
CNC Milling	PolyPropylene	High Yield Strength vs. Young's Modulus ratio	Facility not readily available at TUDelft, clean corners cannot be milled
Glued assembly	Acrylic+ Spring Steel	High Yield Strength vs. Young's Modulus ratio, Cheap	More parts, Prone to assembly errors, Fragile

From table 6.1, the choice for a glued assembly was chosen. The product from this fabrication option will have more parts, an error due to assembly and be very fragile.

However, this model will only serve as a verification that the results from the optimization are correct. Also, using metal-like spring steel for the storage of elastic energy will result in lower energy losses in the oscillator due to hysteresis. However, it is expected that the friction in the bearings that remain in this design will trump the impact that hysteresis presents.

For future development, fabrication options with polypropylene seems to be a good choice, but as of yet there are no facilities available at the TUDelft to use this material.

The method to produce a glued assembly is done by firstly identifying all the elastic elements in the design. These elements are removed from the design and at their connection points, a small groove is cut into the rigid parts. The rigid parts are cut out of acrylic with a laser cutter

Spring steel of 0.1mm thick was cut a little longer than the lengths of the elastic elements. This extra length allows the spring steel to slot into the grooves of the rigid parts.

To assemble the spring steel and the acrylic, a 1:1 scale drawing of the top view of the assembled model is printed onto paper. The rigid parts are temporarily attached to the paper using glue. Then the spring steel parts are slotted into the grooves of the acrylic. glue is continually dropped into the

slots from the top. It flows in between the acrylic and the spring steel through the slot. This spreads the glue along surface of the connection between the spring steel and the acrylic.

After the glue in the slots dries, the paper on which the rigid parts were placed is cut away, and the assembled oscillator remains.

The finalized prototype is shown in figure 6.1

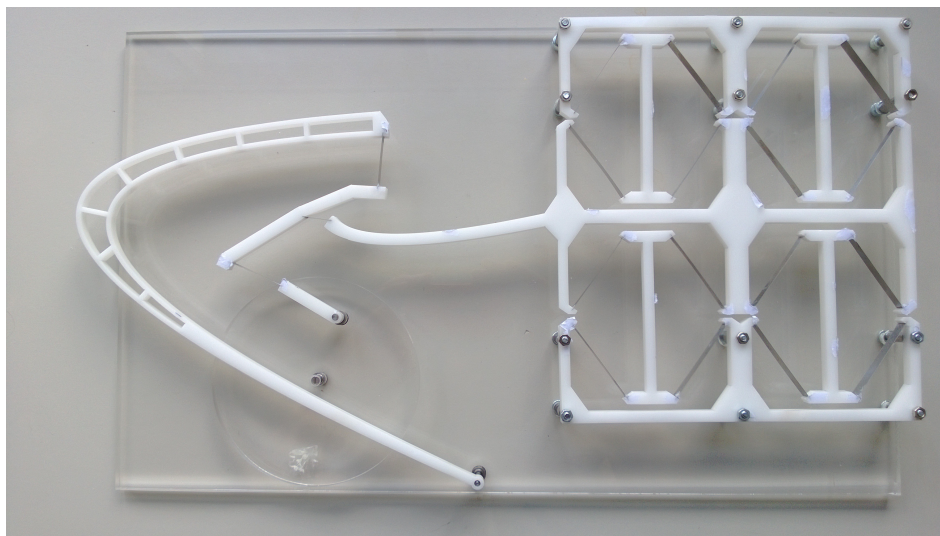


Figure 6.1: Prototype made from thin straps of spring steel and laser cut acrylic

6.2 Testing

The code that was used in the Matlab script to optimize the kinematics was validated in section 3.3.1. This prototype is tested to validate the results from the Ansys results. The validation is done in two steps. The first step is to validate the transmission that was predicted by the Ansys model. The second step is to validate the force-deflection. The measurement setup is shown in figure 6.2.

The shuttle of the Xbob joint is translated by a precision stage which is connected to a force-sensor. By displacing the shuttle of the Xbob and measuring the force and the angle of the inertia wheel, both of the validations have been completed. The results of the measurements are shown in figure 6.3

In figure 6.3 both the rotation of the inertia and the actuation force are shown as a function of the displacement of the shuttle of the Xbob joint. Notice that the range of the measurements is very limited. The testing was not performed over the full range that was calculated by Ansys because of plastic deformation that already occurred at the boundaries that were used for this measurement. Although this plastic deformation was not predicted to occur by Ansys, the use of glue at the connection points of the spring steel has influenced the stress distribution inside the structure. Also, the exact properties of the spring steel were not available.

The second thing to notice is the low level of hysteresis that exists in this system. When the measured force-deflection behavior of this mechanism is compared to the force-deflection behavior of the rigid body linkage that was tested in section 3.3.1 in figure 3.15, a decrease in hysteresis is clearly noticeable.

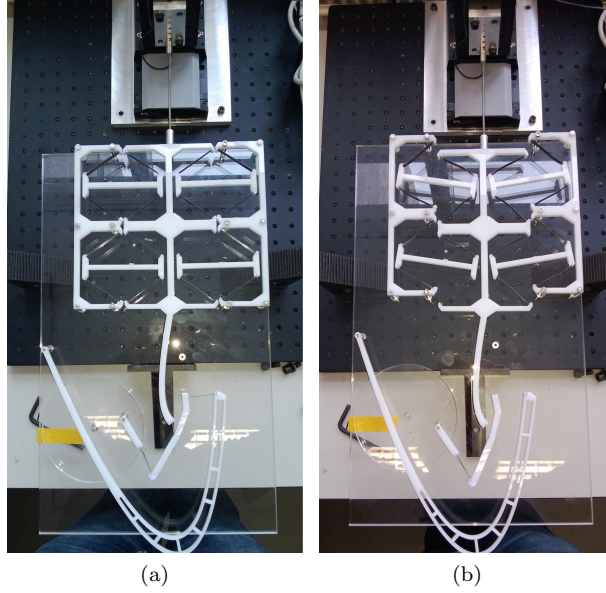


Figure 6.2: Testing of the Oscillator in (a) undeformed state (b) deformed state

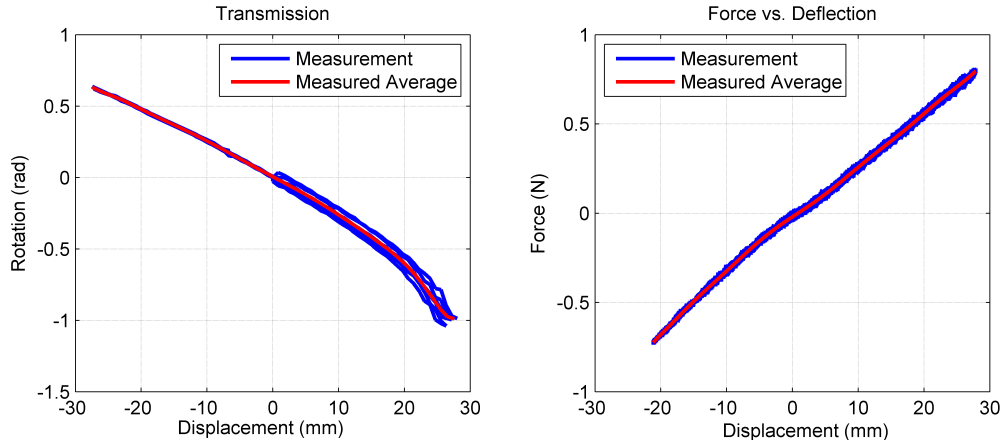


Figure 6.3: Results of the measurements: The transmission from translation to rotation, and the force-deflection behavior of the oscillator (a) undeformed state (b) deformed state

The data can be added together to attain the actuation force as a function of the rotation of the inertia. This result is plotted in figure 6.4, together with the predictions of the Ansys model from section 5.2.

From figure 6.4 it can be concluded that both the measurement results for the transmission (top left) and the force deflection behavior (top right) bear a strong resemblance to the predictions from the model.

When these measurements are combined, the actuation force as a function of the rotation of the inertia

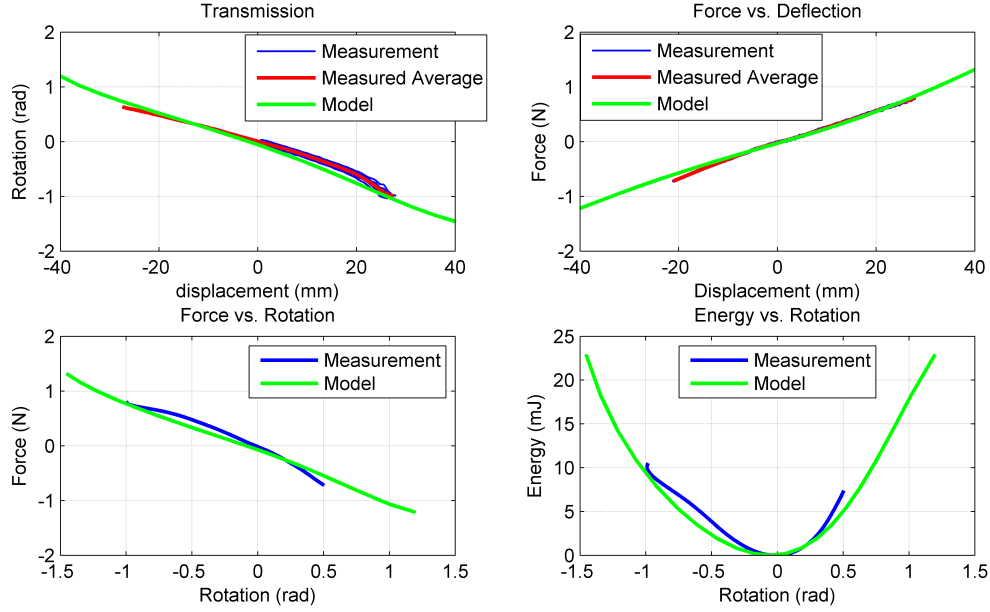


Figure 6.4: Results of the measurements: The transmission from translation to rotation, and the force-deflection behavior of the oscillator (a) undeformed state (b) deformed state

wheel can be plotted. The difference between the measurements and the model is now more evident than for the direct measurements of the transmission and the force-deflection behavior. The strong linear behavior that was predicted by the model, is no longer noticeable in the measurement results. However, the overall trend of the measurements does correspond with the model.

The storage of elastic energy inside the system can be calculated for the measurement setup using formula 6.1. Where F is the required actuation force and s is the traveled path of the actuation force.

$$E = \int_0^s F ds \quad (6.1)$$

The result is shown in the lower right plot of figure 6.4. In this plot the stored elastic energy is shown as a function of the rotation. In this plot it can be seen that the measured stored elastic energy is higher at every rotation compared to the model. This suggests that the prototype is stiffer than predicted.

The overall trend of an asymmetric oscillator is still visible. The energy level is not symmetric around zero radians which means that if this system oscillates, the inertia wheel would rotate further in one direction than in the other.

Dynamic testing was not possible due to the damping that still exists in the revolute joints. Also, with fabrication at a large scale, the mass of the shuttle and other parts of the model are not negligible with respect to the inertia wheel. This added mass will lower the frequency of the oscillator, but also influence the deviation of the frequency as a function of the amplitude.

It is also possible that other vibration modes exist in this large scale prototype that have a frequency very close to the overall vibration frequency that it was designed for. In the current prototype, these

vibration modes would be excited during operation and thus interfere with the overall energy that is used for the desired oscillation behavior.

The difference between the results of the measurements and the model are small enough to be attributed to fabrication and material data errors. The quasi-static testing validates the results of the model.

7 Conclusion

The method that is explained in this report shows the design of compliant oscillators with a oscillation frequency that is independent of the energy level inside the system for a certain range of initial amplitudes.

The design of the oscillator is done by optimization of rigid body linkages with one prismatic elastic joint. The aim of the optimization is to minimize the deviation of the oscillation frequency for a range of initial amplitudes. Unlike conventional harmonic oscillators, the solution space for this optimization is not restricted to symmetric oscillation behavior. Also, the optimization is only aimed at a small range of oscillation amplitudes. The basic kinematics of the transmission are a six bar and a four bar linkage.

With the help of modeling and optimization in Matlab, rigid body designs have been found that have a small deviation of the frequency for a range of initial amplitudes. The most precise results yielded asymmetric oscillators where the mass would have not oscillate symmetrically around its neutral position. This is achieved by the non-linear connection between the deflection of the elastic member and the movement of the inertia. A standard deviation of the frequency of $2.22 * 10^{-4} s^{-1}$ for an oscillator with a mean frequency of 0.500Hz was found. The frequency of a function of the initial amplitude is shown in figure 7.1a

To design a partially compliant oscillator with less revolute joints, a method called the pseudo rigid body model was used. Firstly a new optimization was done with rigid body joints. Every rigid body joint that would be replaced by a compliant member was modeled with positive constant stiffness and constrained to only allow small angular deflections. Optimization with Matlab was performed to minimize the deviation of the frequency. A standard deviation of the frequency of $1.032 * 10^{-3} s^{-1}$ for an oscillator with a mean frequency of 0.498Hz was found. The result is shown in figure 7.1b.

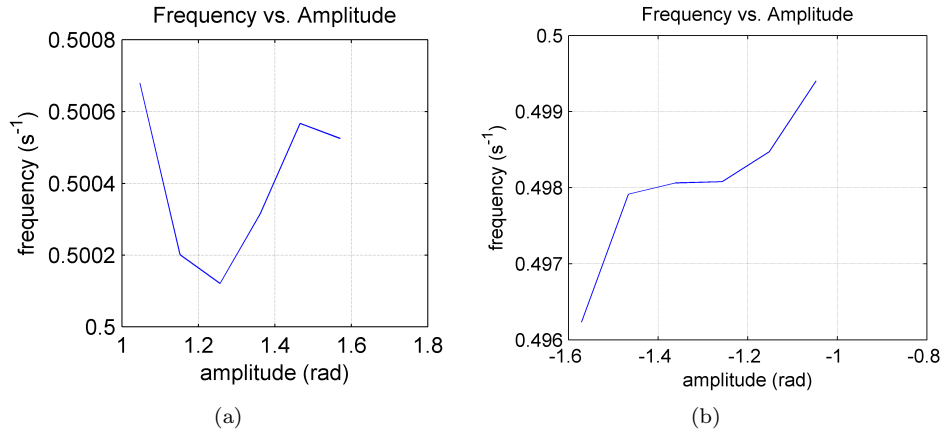


Figure 7.1: Frequency as a function of the initial amplitude for (a) Rigid body Oscillator
(b) Rigid body Oscillator with elastic joints for partially compliant design

The deviation of the frequency for the partially compliant oscillator was higher than for the rigid body oscillator. This was a predictable result from the added constraint that was imposed in the partial compliant design.

Another linkage with a standard deviation of $3.0716 * 10^{-3} s^{-1}$ and a maximum angular deflection

of 12 degrees for the compliant joints was converted to a partially compliant mechanism. After the conversion and further optimization, the standard deviation of the frequency rose to $9.635 * 10^{-3} s^{-1}$. The final compliant design was fabricated and tested. The tests validate the results found by the Ansys and Matlab models.

A Bibliography

References

- ANSYS (2011). Academic research. *Release 14.0*.
- Howell, L. L. and Midha, A. (1994). A method for the design of compliant mechanisms with small-length flexural pivots. *Journal of Mechanical Design*, 116(1):280–290.
- Hubbard, N. B., Wittwer, J. W., Kennedy, J. A., Wilcox, D. L., and Howell, L. L. (2004). A novel fully compliant planar linear-motion mechanism. ASME.
- Kilic, M., Yazicioglu, Y., and Kurtulus, D. F. (2012). Synthesis of a torsional spring mechanism with mechanically adjustable stiffness using wrapping cams. *Mechanism and Machine Theory*, 57:27–39.
- Saggere, L., Kota, S., and Crary, S. B. (1994). New design for suspension of linear microactuators. volume 55-2, pages 671–675.
- Shie, C.-F. and Huang, S.-C. (2010). Design and fabrication of a compliant mechanism for cell gripping. *Journal of Engineering Technology and Education*, 7(4):595–606.
- Tidwell, P. H., Bandukwala, N., Dhande, S. G., Reinholtz, C. F., and Webb, G. (1994). Synthesis of wrapping cams. *Journal of Mechanical Design, Transactions Of the ASME*, 116(2):634–638.

B Patent Search

B.1 Search Terms

The search terms used for the patent search in Escapenet are listed in table B.1. The combination of these search terms together with logical operators resulted in the search queries listed in table B.2.

Table B.1: search terms used

Aspect	Terms
1 escapement	swiss, detent, escapement, timing , ratchet, pacing, english lever
2 mechanical watch	timepiece, wound*, tick*, non-electric
3 locking	lock and release,
4 compliant	bendable, flexible, without/no hinges, low stiffness

Table B.2: search queries used

(english OR swiss OR detent OR watch) AND (escapement OR timing OR ratchet OR Pacing)
(english OR swiss OR detent OR watch) AND (compliant OR flexible OR bendable OR without hinges OR low stiffness)
(compliant OR flexible OR bendable OR without hinges OR low stiffness) AND (escapement OR timing OR ratchet OR Pacing)

B.2 Results

In Figure B.1, the IP map for compliant escapement mechanism is shown. The horizontal axis indicate the problem to be addresses in the escapement mechanism design. The Vertical axis indicates the solution to those problems. The definition of problems and solutions are described below:

Anchor:

- Efficiency - The highest energy loss for an escapement is typically found in the anchor. The efficiency of the anchor limits the time that a fully wound watch can run before it runs out;
- Robustness - The robustness of the anchor determines its ability to maintain operation while being subjected to shocks.

Oscillator:

- Accuracy - The accuracy of the oscillator is determined by a constant oscillation frequency regardless of being subject to shocks.
- Robustness - The robustness of the Oscillator determines its ability to maintain operation while being subjected to shocks.

Solution Approach:

- Compliant - Compliant design instead of rigid link mechanisms are used to resolve problems such as backlash, the need for lubrication, fabrication cost and energy efficiency

The chart displays the following data points (approximate values):

Design Strategy	Efficiency	Robustness	Accuracy	Unlabeled Metric
Compliant joints	2	1	-	-
Damping	-	1	-	-
Change Surface Escape Wheel	5	-	-	-
Add Blocking Parts	-	6	4	-
Switch Blocking with Guiding	2	-	-	-
Stiffness Adjustment of Anchor	5	4	-	-
Cam	1	3	-	-
Increase Frequency	-	-	1	-
Single Piece Manufacturing	-	-	5	-

- **Damping** - By mounting the escapement on a damped structure, energy from shocks to the system can be (partly) absorbed.
- **Change Surface Escape Wheel** - By changing the geometry of the faces of the escape wheel and the pallet stones, the efficiency of the sliding motion between the two can be improved.
- **Add Blocking Parts** - The escape wheel can be equipped safety blocks which prevent the possibility of a uncontrolled unwinding of the escape wheel.
- **Switch Blocking with Guiding** - The impact of the anchor onto the escape wheel creates a loss of kinetic energy. By gradually slowing down the anchor, the kinetic energy can be stored and reused.
- **Stiffness Adjustment of Anchor** - To keep the anchor in place a common practice is to lock the anchor against the escape wheel, known as draw. Unlocking the anchor costs energy that is then

lost. Alternatively, the Anchor can be held in place by creating a multibi-stable stiffness behavior on the rotation axis.

- Cam - The detent escapement uses a one-side actuated blade which is unreliable and inefficient. It is possible to replace this part with a cam-follower setup.
- Increase Frequency - By increasing the frequency of a typical oscillator, the average error decreases. In return however, the barrel spring is unwound faster reducing the run-time of the watch
- Single Piece Manufacturing - By using single piece manufacturing, assembly errors can be avoided. This may increase the accuracy of the watch.

B.3 Conclusion

From figure B.1 it can be seen that the focus area for our project Compliant Escapement Mechanism is among the most clear areas. *This indicates that the likelihood of infringement is low, but the effort to develop technology is high.* This also may promise an impact of innovation in this area due to prior art. Two threatening works have been found in this area; patent WO2011120180 and EP2037335. Initial evaluation of these IPs has shown open room for our project while interesting areas for bi-stability and employing compliant joint in the anchor might be blocked. *This should be evaluated in-depth by TAG legal department.*

B.4 Patent Search Results

In table B.3 an overview of the relevant patents is presented, categorized on the approach that was used by the inventor.

Table B.3: Categories of patents with subsequent patent numbers.

Compliant Joints	WO2011120180	EP2037335
Damping	EP1860511	
Change Surface Escape Wheel	US3628327 CH699268 US7731415	US20110149696 EP1892589
Add Blocking Parts	US2008259739 US2008279052 US7527424 US2012113765 CH704206A2	US2012307601 US8087819 US2006072376 CH703833
Switch Blocking with Guiding	EP1522001	US7927008
Stiffness adjustment of Anchor	CH700091 CH703814 US2012063273 CH703333	US3630018 US6942378 WO2011120180 EP2037335
Cam	US7192180 US2006221774 US7927008	US7458717 WO2011064682
Frequency	US2009168610	
Single Piece Manufacturing	US2008279052 WO2011120180 WO2004063822	US2012063274 EP2037335

Relevant information from every patent from table B.3 is listed below. Also the relevance and possible overlap is discussed.

B.4.1 US Patents

Name	Escapement
Link	US3538705A
Inventors	Perry, M.R.
Applicants	Perry, M.R.
Abstract	The present escapement includes a pallet and escape wheel manufactured to accept broad tolerances with the pallet and escape wheel having flat configurations. The locking and impulse surfaces on the pallet, as well as on the escape wheel teeth, are formed with corner radii. The present escapement provides for an effective deep lock of about 3°, a drop of about 4°, neutral draw, and high impulse energy near the zero beat position, to improve self-starting characteristics, minimize input torque motions sensitivity and substantially eliminate the possibility of setting or locking on an impulse surface.

Invention	An escapement for a timekeeping mechanism comprising a pallet mounted for oscillatory movement and having a pair of pallet arms, each of said arms having integral impulse and locking surfaces, means for oscillating said pallet, an escape wheel carrying a plurality of teeth, each of said teeth having impulse and locking surfaces cooperatively engageable with corresponding surfaces carried by said pallet arms, means for imparting a torque to said escape wheel, the impulse and locking surfaces of each of the pallet arms and escape wheel teeth being angularly related one to the other with a corner radius there between, said corresponding surfaces cooperating to provide unidirectional rotary motion of said escape wheel in response to oscillatory movement of said pallet.
Relevance	This design patent from 1970 describes a basic English escapement. The claim is very general implicating that this was a innovative design at the time. Notice that the lever is dynamically balanced against forces by distribution of weight. Additional claims are made on the mechanism of claim 1 with neutral draw.
Overlap	The first claim in this patent is very specific, which makes it easy to avoid. Little chance of overlap

Name	Clubtooth lever escapement
Link	US3628327
Inventors	Abe Kenji
Applicants	Suwa Seikosha KK
Abstract	A clubtooth lever escapement with high transmission efficiency having an escapement wheel and anchor, and satisfying at least one of $2.0 L_G/L_{At} > 1.0$ and $2.0 L_g/L_{ad} > 1.0$, where L_G is the length of the impulse face of said escape wheel tooth, L_{At} is the length of the impulse face of the entry pallet jewel of said anchor and L_{Ad} is the length of the impulse face of the exit pallet jewel of said anchor.
Invention	A clubtooth lever escapement with high transmission efficiency comprising an escape wheel having a plurality of teeth each of said theeth having an impulse face; and an anchor having an entry pallet jewel and an exit pallet jewel, each of said entry and exit pallet jewels being formed with an impulse face for cooperative engagement with said escape wheel teeth for the transmission of energy from said escape wheel to said anchor, and wherein, $2.0 > L_g/L_{at} > 1.0$, where L_g is the length of the impulse face of the escae wheel teeth, and L_{at} is the length of the ompulse face of the entry pallet jewel of the anchor.
Relevance	This patent describes an improvement on conventional escapement mechanisms only by changing the ratio of the faces of the pallet and escapement wheel. This conclusion of improvement of the efficiency is based solely on the mathematical calculations of the mechanical efficiency.
Overlap	This patent is only about the contact face ratio of the escape wheel and the anchor. These types of improvements are not the main focus of this research, but the claimed forms of transmission transfer can not be used.

Name	Detent escapement for timepiece.
Link	US6942378B2
Inventors	Nicloas, H.G. Thierry, C.
Applicants	Andres, J.C. Montres Brequet SA

Abstract	The detent escapement includes a wheel 2 fitted with teeth 3 , a roller 4 fitted with an impulse pallet stone 5 , a blocking member in the form of a lever 6 hinged on a pin 8 . The first and second arms 9 and 10 of the lever respectively carry a locking pallet stone 7 and a first actuating finger 11 . An elastic member 12 is mounted on the roller 4 , said member carrying a second actuating finger 14 capable of driving the first finger 11 when the roller 4 rotates in a first direction a to actuate the blocking member 6 and to move around said first finger 11 without driving it, when the roller 4 rotates in a second direction b opposite to the first. The elastic member 12 is a spring of great length including a plurality of turns 15 wound about a centre 16.
Invention	A detent escapement for a timepiece including an escapement wheel fitted with teeth, a roller secured to a balance, said roller being fitted with an impulse pallet stone, a blocking member in the form of a lever hinged on a pin, the first and second arms of said lever respectively carrying a locking pallet stone and a first actuating finger, and an elastic member mounted on the roller and carrying a second actuating finger capable of driving the first finger when the roller rotates in a first direction to actuate the blocking member and to move around said first finger without driving it, when the roller rotates in a second direction opposite to the first, wherein the elastic member is a spring including a plurality of turns wound about a centre
Relevance	This design patent describes a variation of the detent escape mechanism. the one-sided actuated blade of the detent escapement is replaced by a structure of a one side actuated rotation spring with a pallet at the end, mounted on the balance. the second embodiment represents a compression spring mounted on the lever that functions as the one side actuated blade of a detent escapement. According to the patent, the new elastic structures have a much lower stiffness, requiring less actuation force and thus less energy waste during operation.l
Overlap	The embodiments (elastic part most of all) are very accurately described and there is no besides a lower stiffness of the elastic finger, there is no apparent difference with conventional detent mechanisms. Other ways or embodiments should be considered if a lower stiffness of the elastic finger is required.
Name	Detent escapement for timepiece.
Link	US7192180B2
Inventors	Nicloas, H.G. Thierry, C. Andres, J.C.
Applicants	Montres Brequet SA
Abstract	The escapement includes a large plate 4 carrying a first finger 14 and a blocking member 6 carrying a second finger 11 and a locking pallet-stone 7 . The first and second fingers 14 and 11 are shaped such that when the large plate 4 rotates in a first direction a, the first finger 14 drives the second 11 which moves around a first side 20 of said first finger to release the locking pallet-stone. Re-engagement occurs when the second finger 11 climbs over a vertical flank 25 of a notch 22 made in a small plate 23 . When the large plate 4 rotates in a second direction b, opposite to the first, the first finger 14 drives the second finger 11 which moves around a second side 21 , opposite to the first, of said first finger 14 to keep the locking pallet-stone 7 in the escapement wheel.

Invention	A detent escapement for a timepiece including an escapement wheel fitted with teeth, a balance, on the pin of which are fixed a large roller fitted with an impulse pallet stone and a first actuating finger and a small roller on the circular periphery of which there is made a notch, and a blocking member in the form of a lever hinged on a pin, said blocking member carrying a locking device of the escapement wheel and a second actuating finger, wherein said first and second fingers are shaped such that when the large and small rollers rotate in a first direction, the first finger drives the second finger which moves around a first side of said first finger to release the device locking the escapement wheel, the second finger being then driven by a rising flank of the notch of the smaller roller to re-engage the locking device in the escapement wheel, and such that when the large and small rollers rotate in a second direction opposite to the first, the first finger drives the second finger which moves around a second side, opposite to the first side of said first inger to keep the locking device engaged in the escapement wheel.
Relevance	This variation of the detent escapement uses a cam instead of elastic elements to get the desired motion of the lever. This makes the mechanism more robust against shocks, but increases friction and wear on the parts.
Overlap	This variation describes a cam in its claim. It is highly unlikely that the new design will have a cam mechanism placed on the balance wheel. Low chance of overlap
Name	Detent escapement for timepiece
Link	US7458717B2
Inventors	Baumberger, P Mojon, J
Applicants	Baumberger et al.
Abstract	An escapement for a timepiece movement includes a toothed escape wheel, a balance and a detent respectively pivoted on rotation axes. The detent supports a pallet lock that interacts with the first tooth of the escape wheel along a certain length of penetration in order to block it in an idle phase during which the balance executes a free arc of oscillation. An impulse pallet rotationally driven by the balance interacts with a second tooth when the detent is pivoted in order to release the escape wheel, this wheel turning by a forward pitch before being blocked again by the pallet lock by interaction with the second tooth. The escapement also includes a limiting plate, which is coaxial and rotationally secured to the balance, has a periphery including a substantially circular main portion and a cutout positioned facing the pallet lock in the release phase.

Invention	A detent escapement for timepiece comprising a toothed escape wheel, pivoted on a first axis (X1), a balance, pivoted on a second axis (X2) and where to an impulse pallet is coaxial and secured, a detent supporting a locking pallet designed to cooperate with a first tooth of said escape wheel, along a certain length of penetration, to block said first tooth in a locking phase, said detent being pivoted on a third axis (X3) to be capable of freeing said first tooth in an unlocking phase, said impulse pallet being designed to cooperate with a second tooth of said escape wheel to receive an impulse from the latter after said unlocking phase, said first and second teeth being adjacent and arranged on either side of a center straight line (L) joining said first (X1) and second (X2) axes in said locking phase, wherein it also comprises a limiting plate, coaxial and rotationally secured to said balance, having a periphery comprising a substantially circular main portion as well as a cutout arranged so as to be positioned across from said locking pallet in said unlocking phase, and wherein said limiting plate is dimensioned so as to define a security distance, separating said locking pallet from said circular main portion when they are positioned facing each other, less than said length of penetration.
Relevance	This design patent describes an improvement on the detent escapement mechanism. By placing the balance wheel close to the escape wheel, keeping teeth in locked state on either side of the center line between the escape wheel and balance wheel axes, the escapewheel is blocked not only by the pallet of the lever, but also by the balance wheel. This improves reliability for shock resistance.
Overlap	This patent presents an easy improvement for shock absorption for the detent escape mechanism. However, the claim is limited to the detent mechanism, thus the mechanism might be applicable to other mechanisms. Low chance of overlap.
Name	Anti-trip device for timepiece escapement
Link	US7527424B2
Inventors	Lechot, D. Zaugg, A Conus, T. Jurin, A.C.
Applicants	Lechot et al.
Abstract	The anti-trip device for a timepiece escapement essentially includes a sprung balance (1), this device preventing the angular extension of said balance beyond a normal angle of rotation. The device is characterized in that the arbour (2) fitted to the balance (1) is provided with a pinion (3) meshing with means (4) preventing said balance from rotating beyond said normal angle of rotation, whatever the direction of said rotation.

Invention	An anti-trip device associated with a timepiece escapement of a timepiece, wherein the timepiece escapement includes: (a) a balance disposed to oscillate on a support via an arbour; (b) a balance spring fixedly attached to the balance and to the support, wherein the anti-trip device is disposed to prevent angular extension of the balance beyond a normal angle of rotation, wherein the balance spring extends around the arbour, and wherein the arbour fitted to the balance is provided with a pinion meshing with the anti-trip device, wherein the anti-trip device includes means for preventing the balance from rotating beyond the normal angle of rotation whatever the direction of the rotation; (c) an escapement wheel provided with teeth; and (d) a pallet carrying locking pallet-stones, wherein said teeth rest in turn on the locking pallet-stones, wherein the means for preventing the balance from rotating beyond the normal angle of rotation comprises a multi-toothed wheel or a multi-toothed sector independent from the pallet, wherein the multi-toothed wheel or multi-toothed sector is pivoted in a fixed plate disposed in the timepiece.
Relevance	This design patent describes an anti-trip mechanism for any escapement mechanism. The mechanism consists of a gear connected to the balance which is blocked by direct contact (rotational guard pin).
Overlap	This patent presents a solution to the anti trip problem. However, this solutions demands a lot of space for such a simple problem. Also additional gears creates additional friction on the balance wheel, which interferes with its purpose. However, this solution is presented as a solution to any escapement mechanism, and thus needs to be avoided. Low chance of overlap
Name	Anti-tripping hairspring for detent type timepiece escapement, has blocking units for blocking consecutive spiral-turns when rotation amplitude of hairspring from balance position to extreme position reaches determined angle
Link	US7927008B2
Inventors	Colpo, F.
Applicants	Boulenguiez, B.
Abstract	Rolex SA
Abstract	This escapement comprises a balance wheel fastened to an impulse element (7a), an escape wheel (1), a detent swing-arm (2) having a stop element (2a) and a disengagement element (2b), a disengagement finger (11d), constrained to rotate with the balance wheel, to come into engagement with the disengagement element (2b) of the detent swing-arm (2) once per oscillation period of the balance wheel.; The disengagement finger (11d) is fastened to an inertial member (11) mounted to move freely between two extreme positions in one of which the trajectory of the disengagement finger (11d) passes through the disengagement element (2b) of the swing-arm (2) and in the other of which this trajectory does not pass through this disengagement element (11b), the passage of the inertial member (11) from one position to the other resulting from the inertia force caused by the variations of speed of the balance wheel in each alternation of oscillation of the balance wheel.

Invention	A detent escapement for a timepiece comprising a balance wheel fastened to an impulse element, an escape wheel the teeth whereof intersect the trajectory of the impulse element, a detent swing-arm having a stop element and a disengagement element, means for engaging the stop element in the trajectory of the teeth of the escape wheel, and a disengagement finger, constrained to rotate with the balance wheel, to come into engagement with the disengagement element of the detent swing-arm once per oscillation period of the balance wheel to release the stop element from the teeth of the escape wheel, wherein the disengagement finger is fastened to an inertial member mounted to move freely between two extreme positions in one of which the trajectory of the disengagement finger passes through the disengagement element of the swing-arm and in the other of which this trajectory does not pass through this disengagement element, the passage of the inertial member from one position to the other resulting from the inertia force acting on the inertial member caused by the variation of speed of the balance wheel in each half-cycle of oscillation of the balance wheel.
Relevance	This variation of the detent escapement uses has no one side actuated blade. Instead the pallet that grips the detent lever is put on a plate that slides under centrifugal forces. That way the pallet only slips outward in one swinging direction to hit the lever and misses the lever in the other direction.
Overlap	This variation uses sliding plates to remove the need for a one side actuated blade. Perhaps a compliant version of this mechanism can be built by replacing the sliding plate by a bistable elastic structure; a lever that can snap outwards during one swing and inwards for the other. The influence of the sliding plate on the consistency of the frequency of the balance is not specified in this patent, but should be a point of interest if a variation of this patent is to be used. Low chance of overlap.
Name	Direct-impulse escapement, especially of detent type, for a horological movement
Link	US8087819B2
Inventors	Chiuve, A. Colpo, F.
Applicants	Rolex S.A.
Abstract	This escapement comprises a balance wheel (3), an escape wheel (1), a detent rocker (4) having an arresting element (4a) and an elastic clearance element (4c), means for inserting the arresting element into the path of the teeth of the escape wheel (1), and a clearance pin (7) rotating integrally with the balance wheel (3) in order to engage with the elastic clearance element (4c) of the rocker (4) once per period of oscillation of the balance wheel. The means for inserting the arresting element (4a) into the path of the teeth of the escape wheel (1) comprise a sliding surface (4b) integral with the detent rocker (4) and arranged so as to move into the path of the teeth of the escape wheel (1) when the arresting element (4a) leaves it, this sliding surface being shaped so as to return the arresting element (4a) to the locking position.

Invention	Direct-impulse escapement, especially of detent type, for a horological movement, comprising: a balance wheel attached to an impulse element, an escape wheel whose teeth intersect the path of the impulse element, a detent rocker having an arresting element and a clearance element, means for intersecting the arresting element into the path of the teeth of the escape wheel, a clearance pin rotating integrally with the balance wheel, and means for engaging said clearance pin (7,11d) with the clearance element of the rocker once per period of oscillation of the rocker to clear the arresting element from the escape wheel tooth; said means for inserting the arresting element into the path of the teeth of the escape wheel comprising a sliding surface integral with the detent rocker and arranged so as to move into the path of the teeth of the escape wheel when arresting element leaves it, this sliding surface being shaped so that the force applied to it by a tooth of the escape wheel causes the arresting element of the detent rocker to move back into the path of the teeth of the escape wheel; the arresting element of the detent rocker comprising a safety surface situated outside of the path of the teeth of the escape wheel(1) and adjacent to this path when the detent rocker is in the unlocking position, in order to prevent the arresting element (4) from moving into the path of the teeth of the escape wheel while the latter is communicating a movement impulse to the balance wheel
Relevance	This design patent describes an improved detent escape mechanism. The improvement is that the return of the detent arm is now no longer dependent on a leaf or torsion spring at the rotation axis of the arm, but rather by a surface that is pushed back by the teeth of the escape wheel. This improves simplicity and reliability.
Overlap	The claim is very specific for a certain layout of the detent escape mechanism. This makes it easier to avoid this patent and still use this idea of having the escape wheel push the lever back to the original locking position. Low chance of overlap
Name	Antitripping device for watch-escapement
Link	US2006072376
Inventors	Gabus, R. Conus, T. Zaugg Jurin, C.
Applicants	Gabus et al.
Abstract	The anti-trip device is for a detent escapement mounted on a wristwatch. It includes a finger fixed to the arm of the balance, two columns between which the finger can pass, said columns being secured to the balance bridge, and a locking arm fixed to the outer coil of the balance spring, said locking arm being able to be inserted between said columns and said finger to prevent the balance from rotating beyond an angle exceeding its normal operating angle. The locking arm is a clamp hooked onto the outer coil of the balance spring.

Invention	: an anti-trip device for a timepiece escapement, said escapement including, amongst other elements, a balance spring made up of several coils and a balance provided by at least one arm, the balance being pivotably mounted between a plate and a bridge, said device including a finger fixed to the arm of the balance, at least one column by which the finger can pass when the balance is moving, said column being secured to the bridge of said balance, and a locking arm fixed to the outer coil of the balance spring, said locking arm being able to be inserted between said column and said finger to prevent the balance rotating beyond an angle exceeding its normal operating angle, wherein the locking arm is a clamp hooked onto the outer coil of the balance spring.
Relevance	a safety measure to prevent the balance wheel or any other part of rotating to far. although it is presented for the detent escapement mechanism, it is claimed for almost any timepiece mechanism (as it is described).
Overlap	This design patent claims a solution to a spin-through problem of the balance wheel. It is very specifically built for a conventional hairspring. If this problem arises in the new design, a solution needs to be found that works around this patent. Low chance of overlap.
Name	Detent escapement for a timepiece
Link	US2006221774
Inventors	Conus, T. Jurin, C.
Applicants	Conus et al.
Abstract	The escapement includes a large roller 4 carrying an impulse pallet stone 5 surmounted by a first finger-piece 14 and a small roller 23 in which a notch 22 is made. A blocking member carries, on the one hand, means 80 for locking the escape wheel 2 , and on the other hand, a second finger-piece 11 arranged for cooperating with the first finger-piece 14 , said blocking member also including a follower 20 ending in a beak 21 , said beak acting on the small roller and particularly with notch 22 , which is made therein. The first and second finger-pieces 14 and 11 are respectively rigidly secured to the table roller and the blocking member
Invention	A detent escapement for a timepiece including an escape wheel fitted with teeth, a balance onto whose staff there are secured rollers including a large roller provided with an impulse pallet stone and surmounted by a first actuating finger-piece, and a small roller in the circular periphery of which a notch is made, and a blocking member in the form of a lever hinged on a pin, said blocking member carrying means for locking the escape wheel, a second actuating finger-piece and a follower ending in a beak arranged for cooperating with a rising edge of the notch of the small roller wherein the first and second actuating finger-pieces are rigidly secured respectively to the large roller and to the blocking member and arranged for cooperating with each other such that when the rollers are rotating in a first direction, the first finger-piece drives the second finger-piece to release the locking means from the escape wheel, the beak of the follower being then driven by the rising edge of the notch to re-engage the locking means of the escape wheel, and such that when the rollers are rotating in a second direction opposite to the first, the first finger-piece drives the second finger-piece to keep the locking means engaged in the escape wheel.

Relevance	This patent describes an detent escapement mechanism that uses teeth and a cam follower on a balance wheel to get the return action of the lever. This replaces the one side actuated blade. Also pallet of the lever is on the opposite side of the axis of the lever compared to the normal detent escapement.
Overlap	This variation describes a cam in its claim. It is highly unlikely that the new design will have a cam mechanism placed on the balance wheel. Low chance of overlap
Name	Lever escapement for a timepiece
Link	US2008259739
Inventors	Jurin, A.C. Conus, T.
Applicants	Jurin et al.
Abstract	The lever escapement includes at least one escape wheel (2). The pallet assembly (5) is fitted with locking pallet stones (6,7) and a first impulse pallet stone (8 cooperating with the teeth of the wheel (2). The pallet assembly (5) drives an auxiliary wheel set (9) carrying a second impulse pallet stone (10) cooperating with the teeth of the wheel (2).
Invention	A lever escapement for a timepiece including an escape wheel set fitted with at least one wheel, a balance roller carrying an impulse pin and a pallet assembly arranged for cooperating with the impulse pin, said pallet assembly being fitted with first and second locking pallet stones and an impulse pallet stone, said pallet stones being arranged for cooperating with the teeth of the scape wheel, wherein the pallet assembly drives an auxiliary wheel set carrying a second impulse pallet stone arranged for cooperating with the teeth of said escape wheel
Relevance	This design patent describes an adjustment to a swiss/English escapement mechanism. The adjustment includes a auxiliary wheel set that gives an extra impulse to the pallet. With this extra impulse, the mechanism is less likely to get to a standstill. Although this design is likely to get to a standstill, it is not self-starting
Overlap	This design partly resolves a problem of the swiss/English escapement mechanism, but it is still not self-starting. This patent limits the solution space for self-starting mechanisms in a way that auxiliary wheel sets driven by the pallet fork are prohibited. Low chance of overlap
Name	Watch escapement
Link	US2008279052
Inventors	Daniel, R Elio, M.
Applicants	Rolex SA

Abstract	The invention concerns an escapement for a mechanical watch comprising: an escape-wheel (10) kinematically connected to a power source, a roller (14) mounted on a balance (16) including, on two different levels, a pin (23) and an impulse pallet (24) co-operating with said wheel (10); a limiting member (26) performing a periodic movement during which it co-operates with said pin (23) to limit the travel of said roller (14); a control member (30) provided with a rest lift (32) performing, from a stable position wherein said wheel (10) is pressed on said rest lift (23), a periodic movement during which said wheel is released and provides an impulse to said impulse pallet (24).; The invention is characterized in that the periods of the limiting member (26) and of the control member (30) correspond, respectively, to an alternation and an oscillation of said balance
Invention	An escapement for a mechanical watch, comprising: an escape wheel kinematically connected to a power source, a roller mounted on a balance wheel including, on two different levels, a pin and an impulse pallet cooperating with said wheel, a pivoting member performing a periodic movement in the course of which the cooperates with said pin (23) and the periodic movement of which is limited by two stops, a control member provided with a locking pallet pin performing, from a stable position in which a tooth of said wheel is bearing upon said locking pallet pin, a periodic movement in the course of which said wheel is freed and provides an impetus to said impulse pallet (24), wherein the said pivoting member and the control member are connected together by a jumper, the end of which is shaped so as provide two stable bearing surfaces being applied alternately against the pallet via a spring cooperating with the jumper, and in that the control member includes a return part intended to cooperate with the teeth of the escape wheel in order to limit the pivoting of the control member, while the pivoting member continues its movement driven by said pin, making the pallet pass from the first bearing surface to the second bearing surface during the first half of the cycle of said periodic movement of the balance wheel, said pin bringing the pallet back in to contact with the first bearing surface during the second part of the cycle of said periodic movement of the balance wheel.
Relevance	This design patent describes an crossover between a detent and a swiss escapement mechanism. The connection between the pallet and balance wheel is set setup with a swiss fork. However, the connection between the pallet fork and escapement wheel is of the detent sort. the interaction between the swiss and detent part of this mechanism is done through a rough dwelling mechanism which is contact based but uses compliant components.
Overlap	This design patent shows a nice alternative dwelling mechanism. The dwell mechanism is described by " a jumper, the end of which is shaped so as provide two stable bearing surfaces being applied alternately against the pallet via a spring cooperating with the jumper, and in that the control member includes a return part intended to cooperate with the teeth of the escape wheel in order to limit the pivoting of the control member". This may restrict a design of a new dwelling mechanism.
Name	Horoligcal movement comprising a high oscillation frequency regulating device
Link	US2009168610
Inventors	Gigandet, C. Dias, F. Behrend, A.
Applicants	Rolex SA

Abstract	The horological movement comprises a regulating device having an oscillation frequency f and an escapement (11, 12) for maintaining the oscillations of the regulating device. The escapement comprises an escape wheel (11) having N teeth. The frequency f is at least equal to about 5 Hz, and the ratio between the number N of teeth and the frequency f is substantially equal to 5.
Invention	Horological movement comprising a regulating device having an oscillation frequency f and an escapement for maintaining the oscillations of the regulating device, the escapement comprising an escape wheel having N teeth, wherein the frequency f is at least equal to about 5 Hz and the ratio between the number N of teeth and the frequency f is substantially equal to 5.
Relevance	This patent corrolates the number of teeth of the escape wheel to the frequency of the escapement mechanism.
Overlap	The claim is set for any 'regulating device' and thus applies to every escapement mechanism known or unknown. Care must be taken to ensure that either frequencies of the escapement do not lie in the interval specified in the patent OR make sure that the ration between teeth and frequency is unequal to 5.

Name	One-piece regulating member and method of manufacturing the same
Link	US20110103197A1
Inventors	Bhler, P. Verardo, M. Conus, T. Thibaud, J. Peters, J. Cusin, P.
Applicants	Nivarox-Far S.A.
Abstract	The invention relates to a one-piece regulating member including a balance cooperating with a hairspring made in a layer of silicon-based material and including a balance spring coaxially mounted on a collet. According to the invention, the collet includes one extending part that projects from the balance spring and which is made in a second layer of silicon-based material and is secured to the balance.
Invention	An one-piece regulating member including a balance made in a first layer of silicon-based material and cooperating with a hairspring, said hairspring being made in a second layer of silicon-based material and including a balance spring coaxially mounted on a collet, wherein it comprises a part forming a spacer which is made in a third layer of silicon based material and is secured between said collet and the balance in order to form said regulating member in a one piece manner
Relevance	This patent describes is mainly about the production steps to produce a balance wheel together with a hairspring in order to come up with a one-piece oscillator.
Overlap	In this claim the production procedure is most important, but the term 'one piece manner' could refer to a compliant solution to an oscillator. However, the single piece oscillator is only protected under this patent if it consists of both a hairspring and balance and the production process is more or less the same as described in claims 26-47. So even though this patent describes a single piece oscillator, it can be avoided by choosing a different design and/or manufacturing method

Name | Swiss lever escapement

Link	US20110149696
Inventors	Cettour-Baron, R. Chiuv, A.
Applicants	Cettour-Baron et al.
Abstract	This Swiss lever escapement comprises an escape wheel having teeth, and a lever having on the one hand an entry pallet and an exit pallet that engage alternately with the teeth of the escape wheel, and on the other hand a fork that engages periodically with an impulse pin on a roller mounted on the staff of a regulator balance wheel. The relative width L_{pl} of each of said pallets, expressed as a percentage of the sum of the lengths of arc of one of said teeth and one of said pallets, measured at the circumference of the escape wheel is: $L_{pl} = \frac{L_s}{L_s+d}$ or $\frac{L_e}{L_e+d} \leq 60\%$ where L_s and L_e are the lengths of arc of the exit pallet and entry pallet, respectively, and d is the length of arc of one escape wheel tooth.
Invention	A swiss lever escapement comprising: an escape wheel having teeth, and a lever, wherein the lever comprises (i) an entry pallet and an exit pallet that engage alternately with the teeth of the scape wheel, and (ii) a fork that engages periodically with an impulse pin on a roller mounted on the staff of a regulator balance wheel, wherein the relative width L_{pl} of each of said pallets, expressed as a percentage of the sym of the lengths of arc of one of said teeth and one of said pallets measured at the circumference of the wheel is; $L_{pl} = \frac{L_s}{L_s+d}$ or $\frac{L_e}{L_e+d} \leq 60\%$ where L_s and L_e are the lengths of arc of the exit pallet and entry pallet, respectively and d is the length of arc of one escape wheel tooth.
Relevance	This patent is about the ratio of the faces of the teeth of the escape wheel and the impulse pallets. It also refers to patents EP1892589 and US3628327. The result of practical tests is different from the aforementioned patents and an increase in efficiency is achieved by using the formulation stated in the abstract/invention
Overlap	Much like in patents EP1892589 and US3628327. The ratio of faces of the impulse pallets and the theeth of the escape wheel may not comply with the formulations stated above.
Name	Detent escapement for timepiece and mechanical timepiece
Link	US2012063273A1
Inventors	Masayuki, K. Hiroki, U. Matsuo, K. Miei, S. Takashi, N Maier, F. Musy, J.
Applicants	-

Abstract	A detent escapement for a timepiece capable of decreasing energy loss with respect to a free oscillation of a balance and improving a timekeeping accuracy is provided. In addition, a detent escapement of a timepiece capable of realizing miniaturization and suppressing variations in the accuracy of a finished product due to assembly errors is provided. A one-side actuating spring of a detent 7 is formed so that a maximum stress portion, which is generated at the time of operating due to the contact of an unlocking stone 4 when a balance 5 is return-rotated, is present to be perpendicular to a first straight line L1 which connects the center of the balance staff 9 and a fulcrum 23a of the blade 23, and to be the side opposite to the balance by a second straight line L2 which passes through the fulcrum 23a.
Invention	A detent escapement for a timepiece comprising: an escape wheel; a balance that includes an impulse jewel which can contact a wheel tooth of the escape wheel and an unlocking stone, and that freely oscillates about a balance staff; a blade that includes a locking stone which can contact the wheel tooth of the escape wheel, and that is supported being capable of approaching to and separating from the escape wheel; and a one-side actuating spring that can contact the unlocking stone and be elastically deformed along the approaching and separating direction with respect to the blade, wherein the one-side actuating spring is formed so that a maximum stress portion, which is generated at the time of operating due to the contact of the unlocking stone when the balance is return-rotated, is presented to be the side opposite to the balance by the second straight line which is perpendicular to a first straight line which connects the center of the balance staff and fulcrum of the blade and passes through the fulcrum.
Relevance	This design patent describes a detente escapement where the maximum stress of the one side actuated blade lies under the rotational axis of the blade (thus on the opposite site of the contact point with the balance wheel)
Overlap	This design patent is definitely something to watch out for in the design of a new detent escapement. Not much is needed for this patent to hold besides the placement of the maximum stress of the structure. One possible way to avoid this patent is build a similar structure but create a flaw in the design making the maximum stress be somewhere else in the structure without affecting the overall functionality.
Name	Detent escapement for timepiece and mechanical timepiece
Link	US2012063274
Inventors	Uchiyama, H. Kishi, M. Niwa, T. Koda, M Sato, M
Applicants	Uchiyama et al.
Abstract	A detent escapement for a timepiece has an escape wheel and a balance that freely oscillates about a balance staff and includes an impulse jewel that contacts a wheel tooth of the escape wheel and an unlocking stone. A blade includes a locking stone that contacts the wheel tooth of the escape wheel and is supported so as to be capable of approaching to and separating from the escape wheel. A one-side actuating spring contacts the unlocking stone and is elastically deformed along the approaching and separating direction with respect to the blade. The blade is constituted of a blade main body and a blade adjustment portion that is separated from the blade main body.

Invention	A detent escapement for a timepiece comprising: an escape wheel; a balance that includes an impulse jewel which can contact a wheel tooth of the escape wheel and an unlocking stone, and that freely oscillates about a balance staff; a blade that includes a locking stone which can contact the wheel tooth of the scape wheel, and that is supported being capable of approaching to and separating from the escape wheel; a one-side actuating spring that can contact the unlocking stone and be elastically deformed along the approaching and separating direction with respect to the blade; and an adjustment mechanism in which the relative position to the unlocking stone of at least one of a blade tip portion of the blade and a spring tip portion of the one-side actuating spring can be adjusted
Relevance	This design patent describes an improvement to two types of the detent escape mechanism (spring and pivoted). The improvement is an adjustable mechanism which relates the position of the blade relative to the one-side actuated spring.
Overlap	If a detent escape mechanism is used and an adjustment mechanism is needed. This mechanism can't adjust the distance between the blade and the one-side actuated spring.
Name	Anti-trip device for an escape mechanism
Link	US2012113765
Inventors	Queval, A.
Applicants	Queval
Abstract	Anti-trip device (1) for a balance (2) pivoting about an axis (D1) whose position is fixed relative to a plate (3). It includes a banking pin (5) on said plate (3), a bistable assembly (8) including: a rotor (9) which is synchronous with said balance (2) and a lever (11) pivoting about another axis (D2) of said rotor (9) between two extreme positions of indexing means (12) memorizing the position of said balance (2), the trajectory of said lever (11) partially interfering with said banking pin (5) when said balance (2) is pivoting; means (15) for limiting amplitude in the event of a shock, which includes stop means (16) between said lever (11) and said banking pin (5), forming a pivot stop during a normal arc of the balance and which, when pressed, generates a change of position in said indexing means (12) and a stop for said balance (2) in the event of knocking.
Invention	An anti-trip device for an escape mechanism, arranged to cooperate with a balance, which pivots about a first pivot axis, whose position is fixed relative to a plate, wherein said device includes a banking pin arranged to be fixed to said plate, and at least one movable bistable assembly which includes, on the one hand, at least one rotor arranged to be fixed to said balance and to pivot synchronously therewith, and on the other hand, a movable bistable lever that pivots relative to said rotor about a second pivot axis, parallel to said first pivot axis, over a limited angular sector between two indexing position that can be occupied by indexing means comprised in said bistable assembly for memorizing the position of said balance, wherein at least one part of the trajectory of said bistable lever interferes with said banking pin when said balance pivots, and said bistable assembly further includes amplitude limiting means for limiting the amplitude of angular pivoting of said balance in the event of a shock.
Relevance	This design patent has the sole purpose of preventing an run-through or trip of the balance wheel. It is applicable to any escapement mechanism. it uses a contact-aided (compliant) bistable mechanism to prevent a trip.

Overlap	This patent has to be avoided if an anti-trip device is needed to prevent run-through of a balance.
Name	Detent escapement and manufacturing method thereof
Link	US2012300596
Inventors	Koda, M Niwa, T
Applicants	Koda et al.
Abstract	In a detent escapement 100, a blade 130 includes a plurality of blade components that includes a one side actuating spring 140 which includes a portion capable of contacting the unlocking stone 124, and a one side actuating spring support arm 133 which determines a position of an unlocking stone contact portion 140G which is positioned in a tip of the one side actuating spring 140. At least two of the blade components are formed of the same material as each other, and each thickness is the same as each other.
Invention	A detent escapement (100) for a timepiece comprising: an escape wheel (110); a balance (120) which has an impulse pallet (122) which can contact a wheel tooth of the escape wheel (110) and an unlucking stone (124); and a blade (130) which has a locking stone (132) which can contact the wheel tooth of the escape wheel (110), wherein the blade (130) includes a plurality of blade components that includes a one side actuating spring (140) which includes a portion capable of contacting the unlocking stone (124), and a one side actuating spring support arm (133) which determines a position of an unlocking stone contact portion (140G) which is positioned in a tip of the one side actuating spring 140), and at least two of the blade components are formed of the same material as each other, and each thickness is the same as each other.
Relevance	This design patent describes an improvement on the detent lever or pivoted escapement mechanism. The improvement is the reduction in parts by stating that 'at least two of the blade components are formed of the same material as each other and each thickness is the same as each other'.
Overlap	This patent will present a lot of conflict if a compliant version of the detent mechanism is constructed. The claim is limited to a particular setup of the detent mechanism, care must be taken to avoid this description.
Name	Impact-proof timepiece escapement
Link	US2012307601
Inventors	Kruttli, A
Applicants	Kruttli, A
Abstract	An escapement includes an escapement wheel (1) and an anchor (2). In the place of the traditional limiting walls or pins, the escapement includes, on the anchor (2) and/or the escapement wheel (1), elements (17, 18) for limiting the oscillations of the anchor (2) during normal operation of the escapement. In order to prevent contact between the impulse beak (10) of the pallets (6, 7) and the escapement wheel (1) upon impacts, the escapement wheel (1) has protrusions (20, 21, 22) at its periphery.

Invention	A timepiece escapement having an escapement wheel (1) and an anchor (2), the anchor (2) comprising an entry pallet (6) and an axit pallet (7) cooperating with teeth (8) on the escapement wheel (1), each of the entry and axit pallets (6,7) having a back side (9), an impulse beak (10), an impulse face (11) and a lock face (12), the anchor (2) and/or the escapement wheel (1) comprising means (17,18) for limiting th oscillations of the anchor (2) during normal operation of the escapement to a range of displacement defined by an entry lock position where the entryu pallet (6) blocks the escapement wheel (1) and by an exit lock position where the exit pallet (7) blocks the escapement wheel (1), wherein the escapement wheel (1) comprises, at a periphery, protrusions (20,21,22) arranged so that: upon an impact having the effect of causing the anchor(2) to leave the said range of displacement in a first direction (F3), the impulse face (11) and the back side (9) of the entry pallet (6) can come into abutment on two (20,21) of the protrusions (20,21,22) respectively and thus stop the anchor (2), without contact between the impulse beak (10) of the entry pallet (6) and the escapement wheel (1)
Relevance	This design patent describes adjustments to the shape of the surface of the escape wheel. By adding adding protrusions to the surface, impacts causing the anchor to leave its normal range of motion, can be absorbed and the anchor be maintained in a somewhat correct position.
Overlap	This design patent avoids the use of guards to keep the anchor in the desired range of motion. Thus, by using guards, this patent can easily be avoided.

B.4.2 WO Patents

Name	Hgih Efficiency escapement
Link	WO2011064682
Inventors	Ferrara, C.
Applicants	Ferrara, C.
Abstract	A mechanical device for horology, comprising angular reduction means (24, 31, 33) apt to decrease the width of the oscillation of a lever (2) pacing the motion of a gear (1).
Invention	a mechanical device (5; 105) for pacing a rotary motion of a rotating toothed member (1; 101), particularly suitable for applications in the horology field and the like, comprising: a balance wheel (4;104) apt to move with an oscillatory motion; a lever (2;102) tilting about a first axis of rotation (23;123) and comprising engagement means (21,22;121,122) apt to cooperate with said rotating member (1;101); a rod (3; 103) apt to oscillate about a second axis of rotation (33; 133) when moved by said balance wheel (4; 104) and articulation means (24, 31, 33; 121, 131, 133) apt to connect said rod (3, 103) to said lever (2;102) so as to cause corresponding oscillations thereof, performing an angular reduction ratio between the oscillations of said rod (3; 103) and the oscillations of said lever (2; 102); wherein said articulation means (24, 31, 33, 121, 131, 133) comprises a mechanical joint with a rotary articulation apt to allow relative rotation between said lever (2; 102) and said rod (3; 103), said rotary articulation being carried out through mutual motion between a yoke (24; 124) located at one end of said lever (2; 102) and a cam (31; 131) eccentric with respect to said second axis of rotation (33; 133), integral to said rod (3; 103) and cooperating with said yoke (24; 124).
Relevance	This design patent describes a crossover between the English, and detent escapement. There is only one impulse to the balance wheel per oscillation (detent). The escape wheel does advance twice per oscillation (English). The anchor is consists of two parts. The first part holds the pallet stones and the second is the connection between the anchor and the balance wheel. These two parts are connected through an 'angular reduction' mechanism. Because the impulse is no longer transfered through the pallets on the anchor but directly from the escape wheel, the pallets only need to move a minimal distance. In the present invention the pulse between the escape wheel and the balance wheel is significant since they are in opposing motion at the time of impact, no clarification is given.
Overlap	This design patent presents a nice principle that the angular rotation of the anchor (movement of pallets) can be very minimal if the pulse is give directly from the escape wheel to the balance wheel. This idea may be used in another way than an intermittent rod, thus resolving overlap with this patent.
Name	Immobilizing device for a toothed wheel
Link	WO2011120180
Inventors	Colpo, F Henein, S
Applicants	Rolex SA Colpo, F Henein, S

Abstract	<p>The invention relates to an immobilizing device for a toothed wheel suitable for the field of horology where it can be part of a direct- or indirect-impulse escapement, in particular in a wristwatch. This immobilizing device (1, 2, 3, 4, 5, 6) comprises: - a base (7); - an immobilizer (9) comprising two arms (10, 11) each provided with a pallet (14, 15) intended to come into contact with a tooth of the toothed wheel (40); - a first and a second elastic element (12, 13) each having an end connected to the immobilizer (9) and another end connected to the base (7); - a third elastic element (16) connected to the immobilizer (9), and it has the particular feature that it is in one piece or in one piece apart from at least one of the pallets (14, 15). The invention also relates to a timepiece and to a method for assembling such a timepiece.</p>
Invention(fr)	<ol style="list-style-type: none"> 1. dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) comprenant: - un chssis (7); - un bloqueur (9) comprenant deux bras (10,11) munis chacun d'une palette (14,15) destine 's venir en contact avec une dent de la roue dente (40); - un premier et un deuxime lments lastiques (12,13) ayant chacun une extrmit relie au bloquer (9) et une autre extrmit relie au chssis (7); - un troisieme lment lastique (16) reli au bloqueur (9), ce dispotif de blocage tant caractris en ce qu'il est d'un seul tenant ou d'un seul tenant l'exception de l'une au moins des palettes (14,15). 2. Dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon la revendication 1, dans lequel les premier et deuxime lments lastiques (12,13) sont chacun relis un bras (10,11) 3. dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon la revendication 1 ou 2, dans lequel le troisieme lment lastique (16) est reli l'un des bras (10,11) un endroit diffrent de la zone de jonction de ces bras (10,11) 4. dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon la revendication 3, dans lequel le troisieme lment lastique est reli l'extrmit de l'un des bras (10,11). 5. Dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon l'une des revendications 1 4, dans lequel les premier et deuxime lments lastiques (12,13) forment entre eux un angle obtus. 6. Dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon l'une des revendications 1 5, dans lequel les premier et deuxime lments lastiques (12,13) sont des premre et deuxime lames flexibles (12,13). 7. dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon l'une des revendications 1 6, dans lequel le troisieme lment lastique (16) est un bloc rigide (17) comportant, sur deux ctes oppos, une troisieme et une quatrieme lames flexibles (18,19). 8. Dispositif de blocage (2,4,5,6) pour roue dente (40) selon l'une des revendications 1 7, dans lequel le troisieme lment lastique (16) est galement reli au chssis (7). 9. Dispositif de blocage (3,4,5,6) pour roue dente (40) selon la revendication 7, dans lequel la quatrieme lame flexible (19) est relie un bloc supplmentaire (240,30), celui-ci tant, le cas chaant, reli au chssis (7). 10. Dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon l'une des revendications 1 9, comportant en outre un systme de prcontrainte exerçant une force sur le troisieme lment lastique. 11. Dispositif de blocage (3,4,5,6) pour roue dente (40) selon la revendication 10, dans lequel le systme de prcontrainte est apte faire varier la force exerce sur le troisieme lment lastique (16). 12. dispositif de blocage (3,4,5,6) pour roue dente (40) selon la revendication 11, dans lequel le systme de prcontrainte variable comprend une vis excentrique (29) ou micromtrique (38).

13. Dispositif de blocage (4,5,6) pour roue dente (40) selon la revendication 11, dans lequel le systme de prcontrainte variable comprend un bloc supplmentaire (30) reli au chssis (7) par des cinquieme et sixime lames flexibles (31, 32) ou au moyen d'un bloc intermdiaire (33) lui-mme reli au chssis (7) par des septime et huitime lames (34,35).

14. Dispositif de blocage (4,5,6) pour roue dente (40) selon la revendication 13, dans lequel les septime et huitime lames (34,35) sont disposes de telle sorte que lors du dplacement des quatre lames (31,32,34,35), leurs raccourcissements s'annulent entre eux, de manire viter tout mouvement parasite du bloc (30) lors du rglage de la prcontrainte.

15. Dispositif de blocage (5,6) pour roue dente (40) selon la revendication 13 ou 14, dans lequel le bloc intermdiaire (33) comprend un plot (36) et le chssis (7) comprend un videment (37) apte recevoir le plot (36) et dlimiter ses mouvements.

16. pice d'horlogerie comprenant un dispositif de blocage (1,2,3,4,5,6) pour roue dente (40) selon l'une des revendications 1 15.

17. pice d'horlogerie selon la revendication 14, le dispositif de blocage (1,2,3,4,5,6) faisant partie d'un chappement et la roue dente (40) tant une roue d'chappement (12).

18. procd de montage d'une pice d'horlogerie compenant les tapes suivantes: -on fixe sur la platine un dispositif de blocage (3) selon la revendication 12; et - on tourne la vis excentrique (29) jusqu' l'obtention d'un systme bistable.

19. proc'd'e de montage d'une pice d'horlogerie comprenant les 'etapes suivantes: -on fixe sur la platine du mouvement un dispositif de blocage (4) selon la revendication 13 ou 14; -on fixe un vis micromtrique ou excentrique de facon ce qu'elle soit en contact avec le block supplmentaire (30); et - on tourne la premiere vis micromtrique ou excentrique jusqu' l'obtention d'un systme bistable.

20 procd de montage d'une pice d'horlogerie comprenant les tapes suivantes: -on fixe sur la platine du mouvement un dispositif de blocage (5,6) selon l'une des revendications 13 15; -on fixe une vis microm'etrique (38) de facon ce qu'elle soit en contact avec le bloc intermdiaire (33); et -on tourne la vis micromtrique (38) jusqu' l'obtention d'un systme bistable.

21 procd de montage d'une pice d'horlogerie selon la revendication 20, comprenant en outre l'etape suivante : -avant de tourner la vis micromtrique (38) pour obtenir le systme bistable, on introduit une cale (39) entre le chssis (7) et le bloc intermdiaire (33).

Invention(en)

1. locking means (1,2,3,4,5,6) for gear (40) comprising: - a frame (7), - a hold circuit (9) comprising two arms (10,11) each provided with a pallet (14,15) intended to come into contact with a tooth of the toothed wheel (40), - first and second elastic members (12,13) each having one end connected to the block (9) and another end connected the frame (7), - a third elastic member (16) connected to the stopper (9), locking dispotif being characterized in that it is in one piece or integrally with the exception of at least one of the pallets (14,15).

2. A locking device (1,2,3,4,5,6) to gear (40) of claim 1, wherein the first and second elastic members (12,13) are each connected to an arm (10,11)

3. locking means (1,2,3,4,5,6) for gear (40) according to claim 1 or 2, wherein the third elastic member (16) is connected to one of the arms (10,11) at a different location of the junction area of said arms (10,11)

4. locking means (1,2,3,4,5,6) for gear (40) of claim 3, wherein the third resilient member is connected to the end of one of the arms (10,11).

5. A locking device (1,2,3,4,5,6) to gear (40) according to one of claims 1 to 4, wherein the first and second elastic members (12,13) form between them an angle obtuse.

6. A locking device (1,2,3,4,5,6) to gear (40) according to one of claims 1 to 5, wherein the first and second elastic members (12,13) are first and second flexible blades (12,13).
 7. locking means (1,2,3,4,5,6) for gear (40) according to one of claims 1 to 6, wherein the third elastic member (16) is a rigid block (17) comprising, on two opposite sides, a third and fourth flexible blades (18,19).
 8. A locking device (2,4,5,6) for gear (40) according to one of claims 1 to 7, wherein the third elastic member (16) is also connected to the frame (7).
 9. A locking device (3,4,5,6) to gear (40) of claim 7, wherein the fourth flexible blade (19) is connected to an additional block (240,30), the latter being the if necessary, connected to the frame (7).
 10. A locking device (1,2,3,4,5,6) to gear (40) according to one of claims 1 to 9, further comprising a tensioning system exerting a force on the third elastic member.
 11. A locking device (3,4,5,6) to gear (40) of claim 10, wherein the prestressing system is able to vary the force exerted on the third spring element (16).
 12. locking means (3,4,5,6) for gear (40) of claim 11, wherein the prestressing system comprises a variable eccentric screw (29) or micrometer (38).
 13. A locking device (4,5,6) for gear (40) of claim 11, wherein the prestressing system comprises a variable additional block (30) connected to the frame (7) by the fifth and sixth leaf springs (31 , 32) or by means of an intermediate block (33) itself connected to the chassis (7) by the seventh and eighth plates (34,35).
 14. A locking device (4,5,6) for gear (40) of claim 13, wherein the seventh and eighth plates (34,35) are arranged such that upon movement of the four blades (31,32, 34,35), their shortening cancel each other, so as to prevent movement of the parasite block (30) when adjusting the prestress.
 15. A locking device (5,6) for toothed wheel (40) according to claim 13 or 14, wherein the intermediate block (33) comprises a stud (36) and the chassis (7) comprises a recess (37) adapted to receive the stud (36) and to define its movements.
 16. timepiece comprising a clamping device (1,2,3,4,5,6) to gear (40) according to one of claims 1 to 15.
 17. Timepiece according to claim 14, the locking device (1,2,3,4,5,6) forming part of a gear wheel and the exhaust (40) being an escape wheel (12).
 18. method of mounting a timepiece comprising the following steps: on the plate is fixed a locking device (3) according to claim 12, and - turning the eccentric screw (29) until a bistable system is obtained.
 19. method of mounting a timepiece comprising the steps of: one fixed plate of the movement of the locking device (4) according to claim 13 or 14; -fixing a micrometer screw or eccentric way that it is in contact with the additional block (30) and - turning the first micrometer screw or eccentric until a bistable system is obtained.
 20. method of mounting a timepiece comprising the steps of: one fixed plate of the movement of the locking device (5,6) according to one of claims 13 to 15; be fixed a micrometer screw (38) so as to contact with the intermediate block (33), and turns-on the micrometer screw (38) until obtaining a bistable system.
 21. method of mounting a timepiece according to claim 20, further comprising the step of: before-turning the micrometer screw (38) to obtain the bistable system is introduced a wedge (39) between the frame (7) and the intermediate block (33).
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Relevance	This design patent describes a compliant (bistable) anchor for an escapement mechanism. Of this patent, claims 1,2,6,8,10-12,16-18 are rejected because of patent EP2037335 (also included in this report). The anchor is mounted by two elastic members. The list of claims that were not rejected are mostly about a pre-stressing structure that makes the anchor have bistable behavior (claim 19). Also a system to apply and adjust the pre-stress is claimed (claim 13,14,15,19,20 and 21).
Overlap	This design patent is definitely something to watch out for. It is interesting to see that claims 10-12 were rejected, these claims apply to the general idea of applying pre-stress to the system by a 'third element'. However, a specific system (claim 13,14) that can be used to apply pre-stress is no problem. There are already some patents that claim bistability of the anchor in different manners, but this is the first claim for compliant bistability of the anchor. Care should be taken to avoid the designs of claim 13 and 14 to introduce pre-stress to the system if bistability is needed, otherwise claim 19 will likely create a problem. Chance of overlap is high.

B.4.3 CH Patents

Name	Swiss anchor escapement for wristwatch, has output pallet whose pulse face has angular length measured from center of escape wheel and in position at which rest tip of tooth of escape wheel is in contact with rest tip of output pallet
Link	CH699268B1
Inventors	Kruettli, A. Maier, F. Musy, J.
Applicants	Patek Philippe SA
Abstract	The escapement has an anchor (3) provided with an input pallet (4) and an output pallet (5), where an angular length (Ls) of a pulse face (10) of the output pallet is higher than 7.5 degrees. The angular length is measured from a center (C1) of an escape wheel (1) and in a position at which a rest tip of a tooth (2) of the escape wheel is in contact with a rest tip of the output pallet. An angular length of a pulse face of teeth of the escape wheel is higher than 2 degrees and measured from the center of the escape wheel.
Invention(fr)	1. Echappement ancre suisse comprenant une roue d'échappement (1) munie de dents (2) et une ancre (3) munie d'une palette d'entre (4) et d'une palette de sortie (5), caractérisé en ce que la longueur angulaire (Ls) de la face d'impulsion (10) de la palette de sortie (5) est supérieure 6.5 degrés, ladite longueur angulaire (Ls) tant mesurée depuis le centre (C1) de la roue d'échappement (1) et dans une position où le bec de repos (17) d'une dent (2) est en contact avec le bec de repos (13) de la palette de sortie (5).
Invention(en)	1. Swiss escapement comprising an escape wheel (1) provided with teeth (2) and an anchor (3) provided with an entry pallet (4) and a blade outlet (5), characterized in that the angular length (Ls) of the impulse face (10) of the exit pallet (5) is greater than 6.5 degrees, said angular length (Ls) being measured from the center (C1) of the escapement wheel (1) and in a position of rest the spout (17) of a tooth (2) is in contact with the beak of break (13) of the exit pallet (5).
Relevance	This design patent lists several claims on the angular lengths Ls, Le and d (i.e. faces of entry pallet, exit pallet, escape wheel). It is claimed that by using the specified angular lengths the yield of the mechanism can be increased.
Overlap	This design patent is very specific for the swiss escapement mechanism and research should show whether or not the claimed increase in yield is justified. These types of improvements are not the main focus of this research, but this patent is granted B status and these angular lengths combinations should be avoided to avoid problems.
Name	Echappement force constante de mouvement d'horlogerie.
Link	CH700091A1
Inventors	Nature Connecting Schiesser, A.
Applicants	Claret, C.

Abstract	The invention relates to a detent escapement comprising: a detent (12) carrying a locking pallet stone (20) and pivotally mounted to perform a step of release, a first spring leaf (22) intended to cooperate with the trigger (12), a rocker having a first discharging pallet (18) arranged to cooperate with the first spring leaf (22) and actuating the trigger (12) when the first discharging pallet (18) intersects the first blade spring (22) in a first direction and to cooperate only with the first blade spring (18) without actuating the trigger (12) when the first discharging pallet (18) intersects the first blade spring (22) in a second direction. According to the invention, the exhaust has a cam (26) for changing between at least a first state in which it cooperates with the trigger (12) so as to restrict its movements, and a second state wherein the trigger (12) is free to perform its release. In addition, the pendulum carries a second discharging pallet (32) arranged to pass the cam (26) in its second state in coordination with the release of the trigger (12).
Invention(fr)	1. Echappement dtente comprenant: -une dtente (12) portant une palette de repos (20) et monte pivotante pour effectuer une tape de dgagement, -une premire lame ressort (22) destine cooprer avec la dtente (12). -un balancier muni dne premire palette de dgagement (18) agence pour cooprer avec la premire lame ressort (22) et actionner la dtente (12) lorsque la premire palette de dgagement (18) croise la premire lame resoort (22) dans une premire direction et pour cooprer seulement avec la premire lame ressort (22) dans une deuxime direction, caractris en ce qu'il comporte une came (26) destine voluer entre au moins un premier tat dans lequel elle coopre avec la dtente (12) de manire limiter ses mouvements, et un deuxime tat dans lequel la dtente (12) est libre d'effectuer son dgagement, et en ce que le balancier porte une deuxime palette de dgagement (32) agence de maniere faire passer la came (26) dans son deuxime tat de manire coordonne avec le d'egagement de la d'etente (12).
invention(en)	1.Detent escapement comprising:-a detent (12) carrying a locking pallet stone (20) and pivotally mounted to perform a clearing step, a first-leaf spring (22) for cooperating with the trigger (12). -one rocker having a first discharging pallet (18) arranged to cooperate with the first spring leaf (22) and actuating the trigger (12) when the first discharging pallet (18) intersects the first blade spring (22) in a first direction and to cooperate only with the first spring leaf (22) in a second direction, characterized in that it comprises a cam (26) for changing between at least a first state in which it cooperates with the trigger (12) so as to restrict its movements, and a second state wherein the trigger (12) is free to perform its release, and in that the rocker carries a second discharging pallet (32) arranged to pass the cam (26) in its second state in coordination with the release of the trigger (12).
Relevance	This design patent describes an improvement on the standard detent design. in order to increase its resilience to shocks, a spring actuated mechanism is put into place at the end of the detent lever to hold in in place when the lever is not actuated.
Overlap	This design is very specific for the design of the detent escapement, there is little to no chance of overlap.
Name	Pallet assembly for Swiss escapement of timepiece, has pallet-stones whose respective rest faces are oriented to form null drawing angle, and return spring acting as bistable spring in intermediate position of pallet
Link	CH703333A2
Inventors	Fragniere, B.
Applicants	Fragniere, B.

Abstract	The assembly has a return spring (5) fixed on a pallet (3) and a frame. Positions of fixation points of ends of the spring are chosen such that the fixation points are located on a straight line passing via a swivel pin (3e) of the pallet for enabling the spring to act as a bistable spring in an intermediate position of the pallet. An input pallet-stone (3a) and an output pallet-stone (3b) respectively comprise rest faces that are oriented to form a null drawing angle.
Invention(fr)	1. Ancre d'échappement pour pice d'horlogerie prsentant une leve d'entre (3a) et une lev'ee de sortie (3b) comportant chacune une face de repos, caractrise en ce qu'un ressort de rappel (5) est fix, d'une part sur l'ancre (3), d'autre part sur le bti, les positions des deux points de fixation des extrmits du ressort de rappel tant choisies pour que, dans une position intermdiaire de l'ancre (3), les points de fixation dudit resoor (5) se situent sur une droite passant par l'axe de pivotement (3e) de l'ancre, de manire que le ressort de rappel (5) se comporte comme un ressort bistable
Invention(en)	1.Anchor escapement for a timepiece having a closed inlet (3a) and a closed outlet (3b) each having a face of rest, characterized in that a return spring (5) is fixed to one part of the anchor (3), on the other hand on the frame, the positions of the two fixing points of the ends of the spring being selected such that, in an intermediate position of the anchor (3), points fixing said spring (5) lie on a straight line passing through the pivot axis (3e) of the anchor, so that the return spring (5) acts as a bistable spring
Relevance	This design patent makes the first notion of a bistable anchor. In the patent it is argued that a bistable mechanism can replace the need for a draw angle on the faces of the pallets and escape wheel.
Overlap	The claims in this patent for the bistability is something to avoid. Bistability is not claimed as a concept for the anchor, but its embodiment with a single spring should be avoided.
Name	Lever i.e. Swiss lever, for use in escapement mechanism in watch, has pallets whose resting surfaces are portion of prism of ellipsoidal section, where symmetry axis of ellipsoidal section coincides with pivoting axis of lever.
Link	CH703814A2
Inventors	Calabrese, V.
Applicants	Blancpain SA
Abstract	The lever has a fork (13) connected to input arms (11A) and output arms (11B) by a fork rod (14). The fork comprises input claws (17A) and output claws (17B) arranged to limit movement of an impulse pin of a balance-wheel of a regulating element. The fork comprises a guard pin arranged to cooperate with a notch of the balance-wheel. Resting surfaces (21A, 21B) of pallets (10A, 10B) are a portion of a prism of an ellipsoidal section, where symmetry axis of the ellipsoidal section coincides with a pivoting axis of the lever. Independent claims are also included for the following: (1) an escapement mechanism comprising an elastic return or repulsion unit constituted by a spring (2) a method for fabrication or transformation of a lever or escapement mechanism

Invention(fr)	1. Ancre amliore (1) pour mouvement d'horlogerie, notamment une ancre suisse amliore, comportant:- de part et d'autre d'une tige d'ancre (12) dfinissant un axe de pivotement de ladite ancre (1), - une premire extrmit, au moins deux palettes, d'entre (10A) et de sortie (10B) tant termine par une surface d'impulsion (20A;20B) agence pour recevoir une impulsion communique par une dent (3) d'une roue d'chappement (2), et chaque dite palette (10A;10B) comportant, connexe ladite surface d'impulsion (20A;20B), une surface de repos (21A;21B) agence pour recevoir en appui de repos une dent (3) d'une telle roue d'chappement (2);- une deuxime extrmit, une fourchette (13) relie auxdits bras d'entre (11A) et de sortie (11B) par une baguette de fourchette (14) agence pour pivoter entre des butes de limitation, respectivement d'entre (15A) et de sortie (15B) du ct desdits bras respectifs; -ladite fourchette (13) comportant, de part et d'autre d'un dgagement d'entre de fourchette (16), deux cornes d'entre (17A) et de sortie (17B), agences pour limiter le dplacement d'une cheville de plateau (31) d'un balancier (30) d'un organe rgulateur; -ladite fourchette (13) comportant encore un dard (19), agenc pour cooprer avec une encoche (34) d'un tel balancier (30); et caractrise en ce que ladite surface de repos (21A;21B) de chaque dite palette (10A;10B) est une portion d'un prisme de section ellipsodale dont l'axe de symtrie est confondu avec ledit axe de pivotement de ladite ancre (1).
Invention(en)	1. Improved anchor (1) for a timepiece movement, including improved Swiss lever, comprising: - on either side of a rod anchor (12) defining a pivot axis of said anchor (1), - in a first end, at least two pallets, first (10A) and output (10B) being terminated by a pulse surface (20a, 20b) arranged to receive a pulse provided by a tooth (3) of a wheel exhaust valves (2), and each said blade (10A; 10B) comprising, connected to said pulse surface (20a, 20b), a bearing surface (21A, 21B) arranged to receive and support a tooth of rest (3) of such an escapement wheel (2),-at a second end, a fork (13) connected to said input arm (11A) and output (11B) by a fork rod (14) arranged to pivot between limiting stops, respectively inlet (15A) and output (15B) of said side arm respective-said fork (13) comprising, on either side of a release input range (16), two horns input (17A) and output (17B), arranged to limit the movement of a roller pin (31) of a rocker (30) of a regulating member, said fork-(13) further including a stinger (19) arranged to cooperate with a notch (34) of such a beam (30), and characterized in that said rest surface (21A, 21B) of each said pallet (10a, 10b) is a portion of an ellipsoidal section prism whose axis of symmetry coincides with said pivot axis of said anchor (1).
Relevance	This design patent describes an adjustment to the swiss escapement. Claim 1 is only about the positioning of the pallets on the anchor relative to the pivot point of said anchor. More interesting is claim 7. This claim is about a repulsion system (elastic,magnetic or electric) that provides a push back from the extreme locking positions that the anchor has.
Overlap	Claim 7 describes a method that decreases the initial impact that the balance wheel has to make on the anchor to initiate rotation of the anchor. This is the only part of the patent that should be taken into consideration.
Name	Anti-tripping hairspring for detent type timepiece escapement, has blocking units for blocking consecutive spiral-turns when rotation amplitude of hairspring from balance position to extreme position reaches determined angle
Link	CH703833A2
Inventors	Zaugg, A.

Applicants	Montres Brequet SA
Abstract	The hairspring (1) has blocking units for blocking consecutive spiral-turns (13) when rotation amplitude of the hairspring from a balance position to an extreme position reaches a determined angle, where the hairspring is made of silicon or metal. The blocking units have transversal segments (15) fixed to the consecutive spiral-turns. The segments are angularly offset in the balance position so as to abut against each other when the amplitude of the hairspring from the balance position to the extreme position reaches the determined angle. An independent claim is also included for a timepiece escapement.
Invention(fr)	Spiral anti-galop (1) pour chappement d'horlogerie, destiné à osciller entre deux positions extrêmes, en passant par une position d'équilibre, et comportant une pluralité de spires (13), caractérisé en ce qu'il comprend, en outre, des moyens (15,) pour bloquer au moins deux spires (13) consécutives lorsque son amplitude de rotation depuis la position d'équilibre jusqu'au moins l'une des positions extrêmes, atteint un angle déterminé ψ
Invention(en)	Anti-tripping spring (1) for escapement clock, for oscillating between two extreme positions, through an equilibrium position, and having a plurality of turns (13), characterized in that it further comprises, means (15,) for locking at least two turns (13) when the amplitude of consecutive rotation from the equilibrium position to at least one of the extreme positions reached determines an angle of
Relevance	This design patent describes an anti trip mechanism, the design is integrated into the hairspring making it lock on itself after a certain amplitude
Overlap	Unless we use a hairspring, there is no chance of overlap because of "having a plurality of turns".
Name	Single-piece lever for Swiss lever escapement of mechanical timepiece, has stop pallets arranged in projection with respect to arms in common plane and comprising ends having convex portions defining points of contact with escapement wheel.
Link	CH704206A2
Inventors	Behrend, A. Dias, F. Gigandet, C.
Applicants	Chopard Technologies SA
Abstract	The lever (10) has a bar (12) whose free end is equipped with a fork (14) cooperating with a balance (17), where the lever is made of single crystal silicon material. The bar is prolonged, from a side opposite the fork, by two arms (20). Each arm comprises a stop pallet (26) that is distinct from a rest and impulse pallet (22) of the arm and is arranged in projection with respect to the arm in a common plane defined by the bar. One end of the stop pallet is equipped with a convex portion defining point of contact with an escapement wheel (24). An independent claim is also included for an escapement comprising a lever.
Invention(fr)	1. Ancre (10) d'chappement ancre suisse, réalisée en une pièce monolithique, comportant une baguette (12) définissant un plan général dans lequel s'inscrit l'ancre (10) et munie, son extrémité libre d'une fourchette (14) pour coopérer avec un balancier (17) et se prolongeant, du côté opposé la fourchette (14), par deux bras (20), chaque bras (20) comportant: -une palette d'impulsion et de repos (22), -une palette d'arrêt (26), distincte de la première, et disposée en saillie par rapport au bras (20), dans ledit plan général, ladite palette d'arrêt (26) présentant son extrémité une portion convexe définissant un point de contact avec une roue d'chappement (24).

Invention(en)	1. Anchor (10) escape Swiss lever, made of a monolithic piece, comprising a strip (12) defining a general plane of within which the anchor (10) and provided at its free end with a fork (14) to cooperate with a lever (17) and extending on the opposite side to the fork (14), by two arms (20), each arm (20) comprising: an impulse pallet-and rest (22), a pallet-stop (26), distinct from the first, and arranged with respect to the projecting arm (20) in said general plane, said blade stop (26) having at its end a convex portion defining a point of contact with an escape wheel (24).
Relevance	This design patent is an improvement to the swiss escapement. It describes guards that protect the anchor from rotating its pallets to far into the escape wheel. These guards are fixed to the anchor and contact the faces of teeth of the escape wheel.
Overlap	It is not quite clear what the improvement is of adding these guards to the anchor whilst still keeping the old guards that are fixed to the housing of the watch. I don't see us using this solution any time soon.
Name	Single-piece lever for Swiss lever escapement of mechanical timepiece, has stop pallets arranged in projection with respect to arms in common plane and comprising ends having convex portions defining points of contact with escapement wheel.
Link	CH704206A2
Inventors	Behrend, A. Dias, F. Gigandet, C.
Applicants	Chopard Technologies SA
Abstract	The lever (10) has a bar (12) whose free end is equipped with a fork (14) cooperating with a balance (17), where the lever is made of single crystal silicon material. The bar is prolonged, from a side opposite the fork, by two arms (20). Each arm comprises a stop pallet (26) that is distinct from a rest and impulse pallet (22) of the arm and is arranged in projection with respect to the arm in a common plane defined by the bar. One end of the stop pallet is equipped with a convex portion defining point of contact with an escapement wheel (24). An independent claim is also included for an escapement comprising a lever.
Invention(fr)	1. Ancre (10) d'chappement ancre suisse, ralise en une pice monolithique, comportant une baguette (12) d'efinissant un plan gnral dans lequel s'inscrit l'ancre (10) et munie, son extrmit libre d'une fourchette (14) pour cooprer avec un balancier (17) et se prolongeant, du cote oppos la fourchette (14), par deux bras (20), chaque bras (20) comportant: -une palette d'impulsion et de repos (22), -une palette d'arret (26), distincte de la premire, et dispose en saillie par rapport au bras (20), dans ledit plan gnral, ladite palette d'arret (26) prsentant son extrmit une portion convexe dfinissant un point de contact avec une roue d'chappement (24).
Invention(en)	1. Anchor (10) escape Swiss lever, made of a monolithic piece, comprising a strip (12) defining a general plane of within which the anchor (10) and provided at its free end with a fork (14) to cooperate with a lever (17) and extending on the opposite side to the fork (14), by two arms (20), each arm (20) comprising: an impulse pallet-and rest (22), a pallet-stop (26), distinct from the first, and arranged with respect to the projecting arm (20) in said general plane, said blade stop (26) having at its end a convex portion defining a point of contact with an escape wheel (24).
Relevance	This design patent is an improvement to the swiss escapement. It describes guards that protect the anchor from rotating its pallets to far into the escape wheel. These guards are fixed to the anchor and contact the faces of teeth of the escape wheel.

Overlap	It is not quite clear what the improvement is of adding these guards to the anchor whilst still keeping the old guards that are fixed to the housing of the watch. I don't see us using this solution any time soon.
Name	Anchor escapement mechanism e.g. Swiss anchor escapement mechanism, for clockwork of watch, has anchor arranged around anchor axis, where distance between wheel and anchor axes is less than distance between anchor axis and pin location
Link	CH704536A2
Inventors	Witteveen Bontko
Applicants	Mock Elmar
Abstract	Creaholic SA
Abstract	The mechanism has a balance wheel rotatably arranged around a balance wheel axis (3), and a lever pin (5) moving around spring axis. The lever pin is intermittently engaged in an anchor fork (9) of an anchor (8). The anchor is rotatably arranged around an anchor axis (13). Distance between the balance wheel axis and the anchor axis is less than distance between the anchor axis and intervention location of the lever pin on the anchor fork. The anchor and an anchor wheel are supported under a bearing element. Another balance wheel is supported under the bearing element.
Invention(de)	1. Ankerhemmung (1) fr ein Uhrwerk, aufweisend eine Unruh (2), welche um eine Unruhachse (3) drehbar angeordnet ist, und einem um die Unruhachse (3) bewegten hebelstift (5), wobei der hebelstift (5) zum intermittierenden Eingriff in einer Anker-gabel (9) eines Ankers (8) angeordnet ist, wobei der Anker (8) um eine Ankerachse (13) drehbar angeordnet ist, dadurch gekennzeichnet, dass der Abstand zwischen Unruhachse (3) und Ankerachse (13) kleiner ist als der Abstand zwischen Ankerachse (13) und der Eingriffsstelle des Hebelstiftes (5) an der Ankergabel (9)
Invention(en)	Escapement (1) for a timepiece, comprising a balance (2), which is a balance shaft (3) that is rotatably arranged, around a balance shaft (3) and a lever pin (5) moving around the shaft, wherein the lever pin (5) intermittently engages a pallet fork (9) of an armature (8), wherein the armature (8) is rotably arranged around an armature shaft (13), characterized in that the distance between the balance shaft (3) and armature shaft (13) is smaller than the distance between the armature shaft (13) and the point of engagement of the lever pin (5) on the pallet fork (9)
Relevance	This design patent is an improvement on the swiss escapement. The improvement is achieved by letting the crown of the anchor contact the balance wheel on the opposite side of its center of rotation. With this adjustment the path of the contacts of the balance wheel and anchor are curved in the same direction. This reduces friction and thus improves the energy efficiency. This design patent proposes an interesting hypothesis of an improved efficiency. However, no numbers are given and research should be done to test this setup
Overlap	This patent describes an interesting and simple improvement. Because of the formulation of claim 1 which includes the notion of shafts, this patent would not hold for any compliant design.

B.4.4 EP Patents

Name	Timepiece movement comprising a mobile bridge
Link	EP1860511B1
Inventors	Claret, C.
Applicants	Claret, C. SA
Abstract	The movement has a bridge (28) elastically pivoted with respect to a plate (26) around an axis. The bridge supports a regulator unit (30), an escapement and a part of a finishing gear train. Another part of the gear train is mounted fixed with respect to the plate. A finishing gear train wheel has a pivoting axis that is merged with the axis, for providing a connection between the parts of the gear train. The bridge is integrated around a pivot element on which the wheel pivots. The bridge has a damping system comprising an elastic unit placed at an end of a shaft of the system.
Invention	A mechanical clockwork movement comprising a plate (26), an energy source, a regulator organ (30), an escapement mechanism, and a going train formed from a train of several wheels kinematically connecting the energy source and the escapement mechanism, characterized in that said movement comprises a bridge (28) elastically pivoted relative to the plate (26) around a first axis, said bridge supporting said regulator organ (30), the escapement and a first part of the going train, a second part of the going train being mounted fixedly relative to the plate, a going train whereof the pivot axis is combined with said first axis, ensuring the connection between the two parts.
Relevance	This design patent describes an improvement for shock resilience for any type of escapement mechanism. The entire escapement is elastically pivoted and damped to reduce the impact that shocks have on the escapement
Overlap	This patent describes a very invasive improvement which increases the size of the watch quite a bit. These types of solutions may be applicable to a smaller elements inside the escapement, but for our research, this types of solutions fall outside of the range of possibilities, no overlap is expected.
Name	Swiss anchor escapement
Link	EP1892589A1
Inventors	Kruettli, A. Maier, F. Musy, J.P.
Applicants	Patek Phillipe SA
Abstract	The escapement has an escape wheel (1) equipped with 16 teeth (2), and a pallet (3) equipped with input and output pallet forks (4, 5). The input and the output pallet forks have pulse surfaces (10) whose angular lengths are greater than 7.5 degrees respectively. The angular length (Ls) of the surface of the output pallet fork is measured from a center (C1) of the escape wheel and in a position where a rest tip of the teeth is in contact with a rest tip of the output pallet fork.

Invention	1. Swiss lever escapement comprising an escapement wheel (1) provided with teeth (2) and an anchor (3) provided with an entry pallet (4) and an exit pallet (5), the angular length (Ls) of the impulse face (10) of the exit pallet (5) being greater than 6.5 degrees, the said angular length (Ls) being measured from the centre (C1) of the escapement wheel (1) and in a position where the locking beak (17) of a tooth (2) is in contact with the locking beak (13) of the exit pallet (5), characterized in that the ratio between the angular length (Ls) of the impulse face (10) of the exit pallet (5) and the angular length (d), measured from the center (C1) of the escapement wheel (1), of the impulse face (15) of the teeth (2) of the escapement wheel (1) is greater than 2.5.
Relevance	This design patent lists several claims on the angular lengths Ls, Le and d (i.e. faces of entry pallet, exit pallet, escape wheel). It is claimed that by using the specified angular lengths the yield of the mechanism can be increased.
Overlap	This design patent is very specific for the swiss escapement mechanism and research should show whether or not the claimed increase in yield is justified. This patent is already granted B status and thus these angular lengths combinations should be avoided to avoid problems.

Name	Anchor for a timepiece escapement
Link	EP2037335A2
Inventors	Enzler, A.
Applicants	Enzler, A.
Abstract	The anchor (1) has two anchor arms (2) in each of which a pallet (3) is held. A fork (4) and a fork clamp are provided at its end, which acts on a balance wheel. The exit pallets carrying anchor arms and fork together with two fastening arms (7) are integrally manufactured. The fastening arms, which run in the same plane as the other anchor units, are elastically shaped with a certain length. An independent claim is included for a method for manufacturing an anchor.
Invention(fr)	<p>1. Anker (1) für eine Uhrenhemmung einer mechanischen Uhr mit Hemmungsrad, umfassend zwei Ankerarme (2) in der je eine Palette (3) gehalten ist und einer Gabel (4) an der endseitig ein Gabelhorn vorhanden ist welches auf die Unruh wirkt, dadurch gekennzeichnet, dass die die Ausgangspaletten tragenden Ankerarme (2) und die Gabel (4) zusammen mit zwei Befestigungsarmen (7) einstückig gefertigt sind, wobei die Befestigungsarme (7), die in derselben Ebene verlaufen wie die übrigen Ankerteile, mindestens mit einer Teilstrecke so biegeelastisch gestaltet sind, dass der Anker in der Ebene um eine virtuelle Achse unter Einwirkung der vom Hemmungsrad auf sie übertragene Energie zu schwingen vermag, wobei die Mittelachsen der beiden Befestigungsarme (7) sich in der virtuellen Achse schneiden.</p> <p>2. Anker nach Anspruch 1, dadurch gekennzeichnet, dass die virtuelle Achse im Verbindungsbereich (6) von Ankerarmen (2) und Ankergabel (4) liegt.</p> <p>3. Anker nach Anspruch 1, dadurch gekennzeichnet, dass die virtuelle Achse ausserhalb des Ankers (1) liegt.</p> <p>4. Anker nach Anspruch 1, dadurch gekennzeichnet, dass die beiden Befestigungsarme (7) symmetrisch zur Ankergabel (4) angeordnet sind, so dass die Ankergabel auf den Winkelhalbierenden des von den beiden Befestigungsarmen (7) beziehungsweise deren beiden elastischen Teilstrecken eingeschlossenen Winkels liegt.</p> <p>5. Anker nach Anspruch 1, dadurch gekennzeichnet, dass an den freien Enden der beiden Befestigungsarme (7) Befestigungselemente (8) angeformt sind.</p>

6. Anker nach Anspruch 5, dadurch gekennzeichnet, dass sich die beiden Befestigungsarme (7) von dem Befestigungselement (8) bis zu dem Bereich der Verbindung (6) von Ankergabel (4) und Ankerarmen (2) erstrecken und auf der gesamten Länge elastisch gestaltet sind.
 7. Anker nach Anspruch 5, dadurch gekennzeichnet, dass die Befestigungselemente (8) so angeordnet sind, dass deren periphere Bewegungsbegrenzungsanschlüsse bilden, die eine maximale Auslenkung der Ankergabel definieren.
 8. Anker nach Anspruch 5, dadurch gekennzeichnet, dass die Befestigungselemente (8) eine Form aufweisen mit zu der Ankergabel hin gerichteten Ausformungen, die Bewegungsbegrenzungsanschlüsse (9) bilden, die eine maximale Auslenkung der Ankergabel (4) definieren.
 9. Anker nach Anspruch 1, dadurch gekennzeichnet, dass beide Befestigungsarme (7) auf derselben Seite zwischen der Ankergabel und einem Ankerarm (2) liegen.
 10. Anker nach Anspruch 9, dadurch gekennzeichnet, dass auf der Winkelhalbierenden zwischen den beiden Befestigungsarmen (7) ein zugelastisches Federelement (10) einstückig angeformt ist, welches mit einer Befestigungsspatte (11) zur Befestigung versehen ist, um die Drehsteifigkeit des Ankers (1) zu vermindern.
 11. Anker nach Anspruch 10, dadurch gekennzeichnet, dass die Befestigungsplatte (11) ein Langloch (12) aufweist, zur Befestigung des Federelementes (1) mit einstellbarer Vorspannung um die Drehsteifigkeit des Ankers (1) zu justieren.
 12. Anker nach Anspruch 1, dadurch gekennzeichnet, dass die Befestigungsarme (7) je zwei elastische Teilstrecken (70,71) aufweisen, die parallel gegenüberliegend angeordnet sind.
 13. Anker nach Anspruch 12, dadurch gekennzeichnet, dass die beiden Teilstrecken (70,71) haarnadelartig gestaltet sind und über einer verdickten Verbindungsstelle (72) die beiden Teilstrecken (70,71) miteinander verbunden sind.
 14. Anker nach Anspruch 1, dadurch gekennzeichnet, dass die Befestigungsarme (7) an einem unbeweglichen Teil des Uhrwerkes befestigt sind.
 15. Anker nach den Ansprüchen 5 und 14, dadurch gekennzeichnet, dass die Befestigungselemente (8) mittels Schrauben am Uhrwerk befestigt sind.
 16. Anker nach den Ansprüchen 5 und 14, dadurch gekennzeichnet, dass die Befestigungselemente (8) unlösbar mittels Schweißen, Lötens oder Klebens am Uhrwerk befestigt sind.
 17. Anker nach Anspruch 12, dadurch gekennzeichnet, dass dieser monolithisch aus einem Material gefertigt ist, insbesondere aus der Auswahl von - synthetischen Edelstein, insbesondere Diamant -synthetische Halbleiter -Silizium oder Siliziumverbindung
 18. Anker nach Anspruch 17, dadurch gekennzeichnet, dass der Anker aus Silizium-Wafer hergestellt ist, dessen Oberfläche nitriert oder oxidiert ist.
 19. Verfahren zur Herstellung eines Ankers nach Anspruch 1m dadurch gekennzeichnet, dass dieser aus Silizium nach dem DRIE-Verfahren (Deep Reactive-Ion Etching), insbesondere nach dem cryo-DRIE-Verfahren hergestellt wird.
 20. Verfahren zur Herstellung eines Ankers, dadurch gekennzeichnet, dass dieser nach dem lithographisch galvanischen Abformverfahren (KUGA-Verfahren) hergestellt wird.
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Invention(en)	<ol style="list-style-type: none"> 1. Anchor (1) for a timepiece escapement a mechanical clock with Escapement comprising two anchor arms (2) in each case one pallet is held (3) and of a fork (4) present at the end a fork horn which acts on the balance, characterized in that carries the output pallet anchor arms (2) and the fork (4), together with two fastening arms (7) made in one piece, whereby the fastening arms (7) extending in the same plane as therefore the other anchor part, with at least one leg bent elastically designed, are that the anchor in the plane is able to swing around a virtual axis by the action of front Escapement transmitted energy to them, wherein the central axes of the two mounting arms cut (7) is in the virtual axis. 2. Anchor according to claim 1, characterized in that the virtual axis lies in the connecting region (6) of the armature arms (2) and the pallet fork (4). 3. Anchor according to claim 1, characterized in that the virtual axis lies outside of the armature (1). 4. Anchor according to claim 1, characterized in that the two fastening arms (7) are arranged symmetrically to the pallet fork (4), so that the pallet fork is on the bisector of the two fastening arms (7) and its two elastic legs included angle. 5. Anchor according to claim 1, characterized in that at the free ends of the two fastening arms (7) fastening element (8) are formed. 6. Anchor according to claim 5, characterized in that the two fastening arms (7) of the fastening element (8) is designed extending up to the region of the connection (6) of the pallet fork (4) and anchor arms (2) and elastic along its entire length is. 7. Anchor according to claim 5, characterized in that the fastening elements (8) are arranged so that their peripheral motion limit stops form, which define a maximum deflection of the pallet fork. 8. Anchor according to claim 5, characterized in that the fastening element (8) having a shape directed towards the anchor with fork back formations forming movement limit stops (9), the maximum deflection of the pallet fork is defined (4) 9. Anchor according to claim 1, characterized in that the two fastening arms (7) are on the same side, between the armature and an armature arm fork (2). 10. Anchor according to claim 9, characterized in that on the angle bisector between the two fastening arms (7) one tension spring element (10) is integrally formed, which is provided with a fastening flap (11) for fastening to reduce the torsional stiffness of the anchor (1) . 11. Anchor according to claim 10, characterized in that the fastening plate (11) an elongated hole (12) for fixing the spring element (1) with an adjustable prestress in order to adjust the rotational stiffness of the anchor (1). 12. Anchor according to claim 1, characterized in that the fastening arms (7) each have two elastic legs (70,71) which are arranged in parallel in opposite directions. 13. Anchor according to claim 12, characterized in that the two legs (70,71) designed hairpin-like manner and a thickened junction (72) the two legs (70,71) are connected together. 14. Anchor according to claim 1, characterized in that the fastening arms (7) are fixed to a stationary part of the movement. 15. An anchor according to claims 5 and 14, characterized in that the fastening elements (8) are secured by screws to the clockwork. 16. An anchor according to claims 5 and 14, characterized in that the fastening elements (8) are non-detachably fastened by means of welding, soldering or gluing on the movement.
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	<p>17. Anchor according to claim 12, characterized in that it is manufactured monolithically from brittle material, in particular the selection of - synthetic gemstone, particularly diamond, synthetical semiprecious stone, silicon or silicon compound</p> <p>18. Anchor according to claim 17, characterized in that the anchor is prepared on a silicon wafer , a special surface is nitrided or oxidized.</p> <p>19. A process for producing an armature according to claim 1 characterized in that it is made of silicon using the DRIE processes (Deep Reactive Ion Etching), in particular after cryo-DRIE process.</p> <p>20. A method for manufacturing an armature, characterized in that that it is prepared according to the lithographically galvanic molding process (KUGA method).</p>
Relevance	This design patent describes a elastically mounted anchor for an escapement mechanism. The claims covers a rigid body replacement for the anchor pivot. It consists of two elastic blades mounted at an angle of each other. Claim 10 describes a method of pre-stressing the beams to reduce the anchor stiffness. Claim 11 describes a way to adjust the pre-stress. Claims 14, 15 and 16 describe ways of attaching the mechanism to the clockworks.
Overlap	This design patent is something to watch out for. Besides avoiding the geometry of the compliant anchor pivot (claim 1), there are claims 10,11 (joint stiffness) and 14,15 and 16 (attachment technique) which require more attention. During the design process these claims are likely to interfere with our work. Chance of overlap is very high.
Name	Elastically-mounted escapement anchor
Link	EP2372473A2
Inventors	Kruettli, A.
Applicants	Patek Phillipe SA
Abstract	The assembly (1) has an elastic locking part (14) to fix the assembly on a staff (15), where the assembly is made of silicon, silicon carbide, glass, diamond, crystallized aluminum oxide, or silicon, silicon carbide, glass, diamond or crystallized aluminum oxide based material. The locking part defines a face or a face part (19) arranged directly opposite an escapement wheel (11). Rigid arms (20, 21) are respectively arranged on sides of the locking part to rigidly connect a pallet to a fork (7). The face or face part is an outer face of one of three elastic arms (16-18) of the locking part.
Invention(fr)	1. Ancre d’chappement (1;1a;1b) pour mouvement d’horlogerie, destine cooprer avec une roue d’chappement (11;11a;11b) et comprenant une partie de serrage lastique (14;14a;14b) pour le montage de l’ancre sur un axe (15;15a;15b), caracterise en ce que la partie de serrage lastique (14;14a;14b) dfinit une face ou une partie de face (19;19a;19b) de l’ancre destine tre directement en regard de la roue d’chappement (11;11a;11b).
Invention(en)	1. Escapement anchor (1, 1a, 1b) for a timepiece movement, intended to cooperate with an escape wheel (11, 11a, 11b) and comprising a elastic clamping portion (14, 14a, 14b) for mounting of the anchor on an axis (15; 15a, 15b), characterized in that the elastic clamping portion (14, 14a, 14b) defines a face or a face portion (19; 19a; 19b) of the anchor intended to be directly opposite the escapement wheel (11, 11a, 11b).
Relevance	This design patent only makes claims on mounting an escapement anchor on an axis. It has no influence on the workings of the anchor
Overlap	The title is misleading from an compliant engineers point of view. The claim is only for attaching the anchor to the axis. No chance of overlap to be expected.