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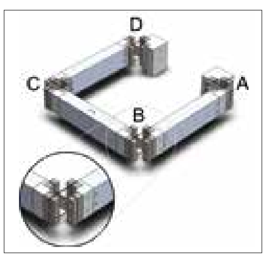

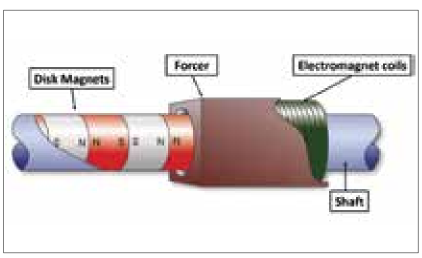

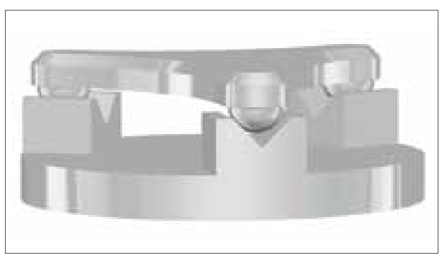

- KINETIC ART
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- HIGH-TECH SYSTEMS 2019 PREVIEW
- MISALIGNMENT AND SELF-ALIGNMENT

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The main cover photo (featuring beach animal 'Umerus silent beach') is courtesy of Theo Jansen (photo: Loek van der Klis). Read the article on page 12 ff.

IN THIS ISSUE

FEATURES

ROLE OF DEVELOPMENT & ENGINEERING IN THE MANUFACTURING INDUSTRY

After having worked in R&D at OEMs (Philips, FEI) for the biggest part of my working life and now since one year as MD for the Development & Engineering (D&E) Division of NTS-Group, I am starting to get a feel for the role that D&E should play in the manufacturing industry.

First things first, the goal should always be to delight and (positively) surprise the customer. D&E is a major contributor in reaching this goal and they need to play many roles. To start with the first one: ownership of the technology roadmap. To delight customers the technology roadmap should not only be driven by the customer's roadmap but also by the technology trends that we see in the world. This translates to proactively investing in technology and people development, which is a daunting task as questions need to be answered such as: what technologies to invest in and how to differentiate from others in the same industry?

On the customer roadmap: whatever we propose to a customer needs to be aligned with his needs. Customer intimacy (preferably going as far as the customer of the customer) is therefore key. D&E can play an important role here when properly linked to the customer's D&E department: they speak the same language and an early insight in customer's D&E activities will help to forecast what the relevant future technologies are and how they affect near-future direction in efficient modular machine development to enable phasing-in of these technologies.

The other role that D&E owns is joining forces with manufacturing to ensure that customer requests are properly addressed. For starters, D&E should be involved in assessing the initial customer request. In case the design is customer-owned, D&E should assess whether the design has been optimised for manufacturing (including the maturity of the design) and, if not, proactively propose alternatives to avoid future supply issues. Involvement of D&E in the requirements and design phase of the customer project is even better as we can insert the manufacturing knowledge upfront preventing additional costs in the later phases of the project. The additional advantage is that future maintenance and factory support is much easier due to the technical knowledge build-up in D&E in the design phase. The earlier a supplier is involved in the customer design process the better, for all parties.

Next to the initial involvement, D&E should also support the factories in the more mature phases of manufacturing. Manufacturing of high-tech equipment is never easy, there will always be issues where D&E can and should support: technical issues need to be fixed, new technical insights could lead to new opportunities for existing products, such as cost/performance improvement, but also proposals for a next-generation product. This will only work when there is a close cooperation between D&E and manufacturing, with no organisational, financial, physical and emotional barriers. To the latter point: we should strive to exchange D&E and manufacturing personnel whenever possible.

D&E plays an essential role in the manufacturing industry, but not in isolation: an intimate relation with manufacturing is essential to drive customer delight.

Hans Scholtz
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DETERMINING CRITICAL MISALIGNMENTS

Overconstrained compliant mechanisms are often avoided in precision machines since they are sensitive to misalignments, which deteriorate their behaviour, reflected for example by a lowered support stiffness or buckling. However, below a certain degree of misalignment the performance is unaffected. Obtaining this critical misalignment by simulations or experiments is often time-consuming and complicated. A more convenient method has been developed. It was applied to a three times overconstrained compliant four-bar mechanism and validated with simulations and an experiment.

WERNER VAN DE SANDE, RONALD AARTS AND DANNIS BROUWER

Introduction

Compliant mechanisms have attributes that make them useful in precision machines. Lack of play, backlash and friction, and low hysteresis make their behaviour predictable. Overconstrained designs jeopardise the predictable nature of compliant mechanisms. An over-constrained design constrains undesired motion in a particular direction more than once. As such, these mechanisms are sensitive to misalignments arising due to assembly, fabrication tolerances or heat sources, causing internal stress which will lead to undesired behaviour, such as buckling or lowered support stiffness. This is worrisome since in compliant mechanisms the difference between a degree of freedom and a constraint is a difference in stiffness by a couple of orders of magnitude.

Therefore, exactly constrained design is often applied to ensure the mechanism has the exact amount of constraints so none of these negative side-effects emerge. However, exactly constrained designs often suffer from, for example, a limited load carrying capability, limited support stiffness and complex design.

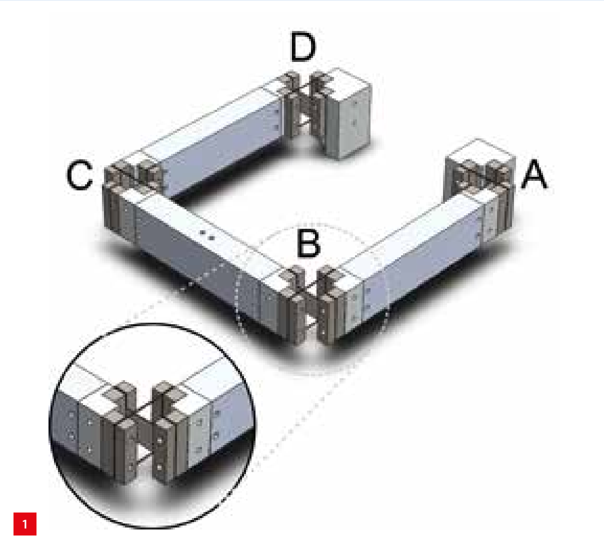
The effects of misalignments on a once overconstrained parallel leaf spring guidance have been investigated and this showed that a small misalignment in the overconstrained direction caused changes in the mechanism behaviour [1-2]. However, below this misalignment level the overconstraint has a negligible effect. This knowledge can be applied to make use of the benefits of overconstrained design while avoiding the drawbacks and create mechanisms with improved performance.

There can be more than one overconstraint in a mechanism. In each overconstrained direction there will be a certain amount of misalignment above which the behaviour of the mechanism is no longer acceptable. Modelling the entire mechanism and running simulations to analyse the behaviour is time-consuming.

A method has been developed to quickly identify the critical misalignments of a multiple overconstrained four-bar mechanism. Buckling of a flexure is taken as the point where the mechanism loses support stiffness and no longer performs as desired. This method compares the buckling loads of the flexures with the loads in the mechanism induced by the misalignments. This comparison yields a value of the critical misalignment in an overconstrained direction.

System topology, kinematics and constraints

Here, the effects of misalignments on an overconstrained compliant four-bar mechanism are investigated (Figure 1).



1

Overconstrained four-bar mechanism with exactly constrained cross-pivot flexures as hinges.

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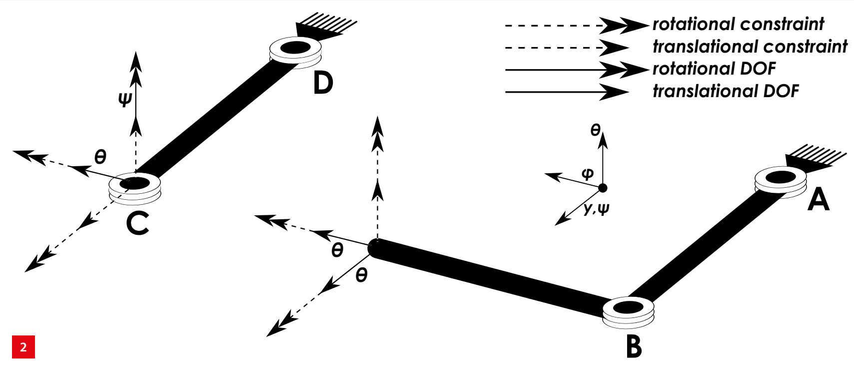


2 The freedom and constraints of a four-bar mechanism illustrated by opening the loop at hinge C.

The mechanism consists of three rigid bars connected by four compliant hinges to each other and the fixed world at hinge A and D. These four hinges are equal in dimension and topology, but different in orientation.
The hinges consist of a leaf spring and two parallel wire flexures that are placed perpendicular to the leaf spring. This creates an exactly constrained hinge. The four-bar mechanism itself has three overconstraints; these can be illustrated by opening the kinematic loop at one of the hinges (Figure 2). The left chain has two hinges and two links; the right chain has two hinges and a single link. Both chains have two degrees of freedom.

The mobility of the end-effector mechanism can be determined by taking the degrees of freedom both chains share. In this case both chains share the translation in the x-direction. The overconstraints can be determined in a similar manner; the constraints that both chains share are in effect doubly constrained, i.e. overconstrained. The out-of-plane directions are constrained by both chains; the mechanism is overconstrained in the rotation around the x- and y-directions, i.e. the φ- and ψ-directions, and overconstrained in the translation along the z-direction.

Determining the critical misalignments

The four-bar mechanism has lumped compliance; all significant compliance is in the flexures. As such, only the flexures are prone to buckling. In this compliant hinge there are two buckling modes of interest: axial buckling of a wire flexure and lateral buckling of the leaf-spring flexure. Lateral buckling of a leaf spring can occur due to a force in the direction of the height of the flexure and a moment in the plane of the flexure. Axial buckling of a wire flexure could occur when an axial force on one of the wire flexures becomes too large. This force can also arise from a moment in the plane of the wire flexures. The buckling loads in the neutral position can be determined with the classical equations described by Timoshenko [3].
In this parallel configuration of the cross-pivot flexure, the stiffness of the flexure is the sum of the stiffness contributions of the leaf spring and the wire flexures.

The flexure is designed such that the centres of compliance of the leaf spring and wire flexures overlap. The centre of compliance is the point in certain flexures where a force or moment in a certain direction will only cause a displacement in that same direction [4]. Consequently, this will yield a diagonal compliance matrix.
The misalignments are applied to the manipulator at the centre of compliance of hinge D (see Figure 3). This can be seen as setting the location and orientation of the fixed world at hinge D.
The relation between a misalignment and the resultant loads at hinge D is expressed by the stiffness of the mechanism at hinge D. The four-bar mechanism is connected to the world at hinge A. Therefore, the mechanism can be seen as a serial chain of links and hinges towards hinge D.
The compliance of each hinge is expressed in its local reference frame. As such, the compliance of each hinge must be transformed to the mechanism reference frame. This can be achieved by a 6x6 transformation matrix:

$$H_{i,m} = \begin{bmatrix} R_{i,m} & T_{i,m}R_{i,m} \\ \mathbf{0} & R_{i,m} \end{bmatrix}$$

The transformation matrix, H, describes a transformation from the local coordinate system, i, to the mechanism coordinate system, m. It is made up of the rotation matrix, $\mathcal{R}$, which rotates the coordinate systems, and a translation matrix T, which translates the origin of the coordinate system. The translation matrix is the so-called cross-product matrix of the translation vector between the two origins of the coordinate systems [5].

The transformed compliances of all hinges are added together to obtain the compliance at hinge D:

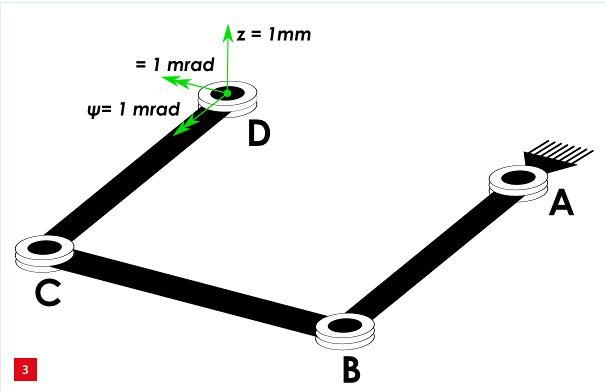
$$C_{tot}^{m} = \sum_{i=1}^{n} H_{i,m} C_i H_{i.m}^{T}$$

This compliance matrix is inverted to obtain the stiffness matrix, which enables the calculation of the loads in the overconstrained directions at hinge D as a result of chosen misalignments of the fixed world. These are chosen to be 1 mm in the z-direction and 1 mrad in the other overconstrained directions.
The load at hinge D, f, is determined using the applied misalignment, δ, in one of the overconstrained directions and the stiffness of the mechanism at hinge D, K. The load is still in the global coordinate system:

$$f_{D,j}^{m} = K_{tot}^{m} \delta_j$$

The resultant load vectors at the hinges are determined using a transformation matrix from hinge D to the hinge in question:

$$f_{i,j}^{m} = H_{D,i}^{-T} f_{D,j}^{m}$$



Unit misalignments of the fixed world at hinge D, all other directions are kept zero.

Table 2
The derived and simulated buckling multipliers of the mechanism without gravity.

	z (mm)	*φ* (mrad)	*ψ* (mrad)
Simulation	2.10	8.48	8.48
New Method	2.64	8.79	8.79

This results in 12 unique load cases: one for each of the four hinges due to one of the three possible misalignments. Each of the 12 unique load cases is compared to each of the three buckling scenarios to obtain the buckling multipliers. These ratios link the load required for buckling to the load due to a unit misalignment.
One of the buckling scenarios will occur first: the scenario with the lowest buckling multiplier. The critical misalignment is the one associated with the lowest buckling multiplier found in the entire mechanism.

Validation

The buckling loads of the cross-pivot hinge were obtained using the program SPACAR [6]. The buckling multipliers could then be determined, see Table 1. The lowest buckling multiplier for each hinge is listed in bold. In the case of a *z*-misalignment there is no preference. In the other overconstrained directions this is hinge D and C. The orientation of the hinges ensures that in every case the wire flexures buckle first.

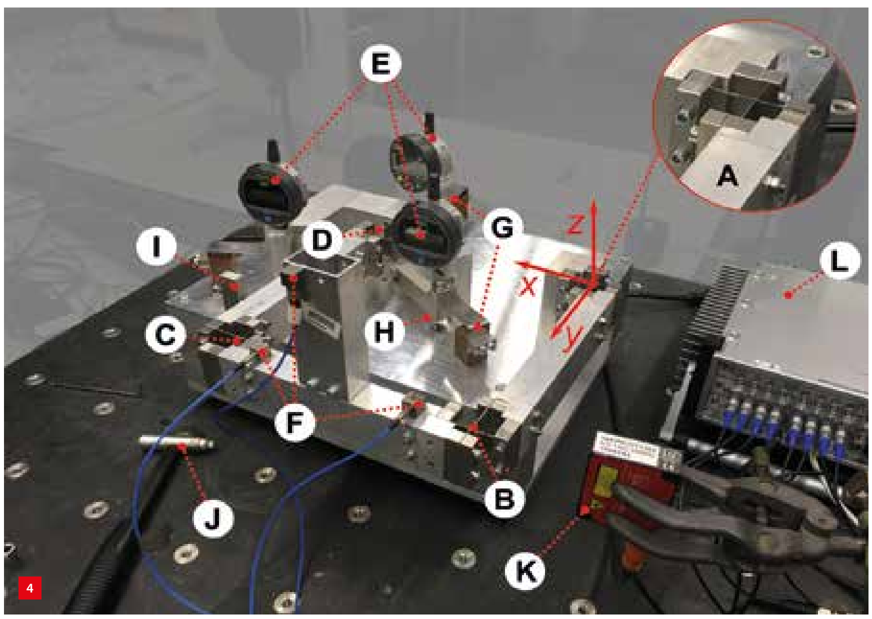
These results have been checked against a full multi-body simulation, also performed using the SPACAR software. Gravity was not considered here, but constrained warping was. The results can be seen in Table 2. The mechanism simulation shows overall lower values for the critical buckling multipliers when compared to the values obtained with the new method. The hinge that buckles first in the simulation is the same as obtained with the new method.

Table 1
The buckling multipliers of the hinges; the buckling scenario is listed (w) for wire flexures and (l) for leaf-spring flexures. The lowest value for each hinge is shown in bold.

	z (mm)	*φ* (mrad)	*ψ* (mrad)
Hinge A	**2.64 (w)**	10.87 (l)	26.2 (w)
Hinge B	**2.64 (w)**	26.2 (w)	26.43 (w)
Hinge C	**2.64 (w)**	26.43 (w)	**8.79 (w)**
Hinge D	**2.64 (w)**	**8.79 (w)**	10.87 (l)

Experiment

An experiment was set up to check the results of the method and the multi-body simulation, see Figure 4. Another multi-body simulation, this time with gravity included, was performed; this simulation better matches the conditions of the experiment. The simulations and the experiment also yield modal data.
Three rigid-body modes of the end-effector have been compared for all three types of misalignments (Figure 5). The misalignments were varied using the manipulator and increased until buckling was observed, see Figure 6.
The results of the simulations and the experiment can be seen in Figure 7 for the *z*-, *φ*- and *ψ*-misalignments, respectively.



The experimental set-up, with four cross-pivot flexures (A through D). The manipulator at hinge D allows for applying misalignment in the three overconstrained directions. The manipulator is connected to three folded sheet flexures (G); these constrain the in-plane directions. The height of each of the three arms can be set by a screw (H); the arms are preloaded against the screws with three helical springs. The height of the arms is measured by three dial gauges (E). The vibrations of the end-effector are measured using three accelerometers (F). The end-effector is excited using a modal hammer (J). The eigenfrequency of the degree of freedom is measured using a laser displacement sensor (K); to avoid nonlinear effects the compliant mode is positioned against a stop (I).

THE BEAUTY OF MOTION

Kinetic art, with its moving elements, uses technology more than any other art in the realisation of its expression. Its creators, however, do have an entirely different way of working than the engineer. This article aims to introduce kinetic art and present some Dutch and foreign artists. Maybe they can inspire engineers to take a more creative approach in order to find new solutions to design challenges. And kinetic art is also just fun to look at!

RINI ZWIKKER

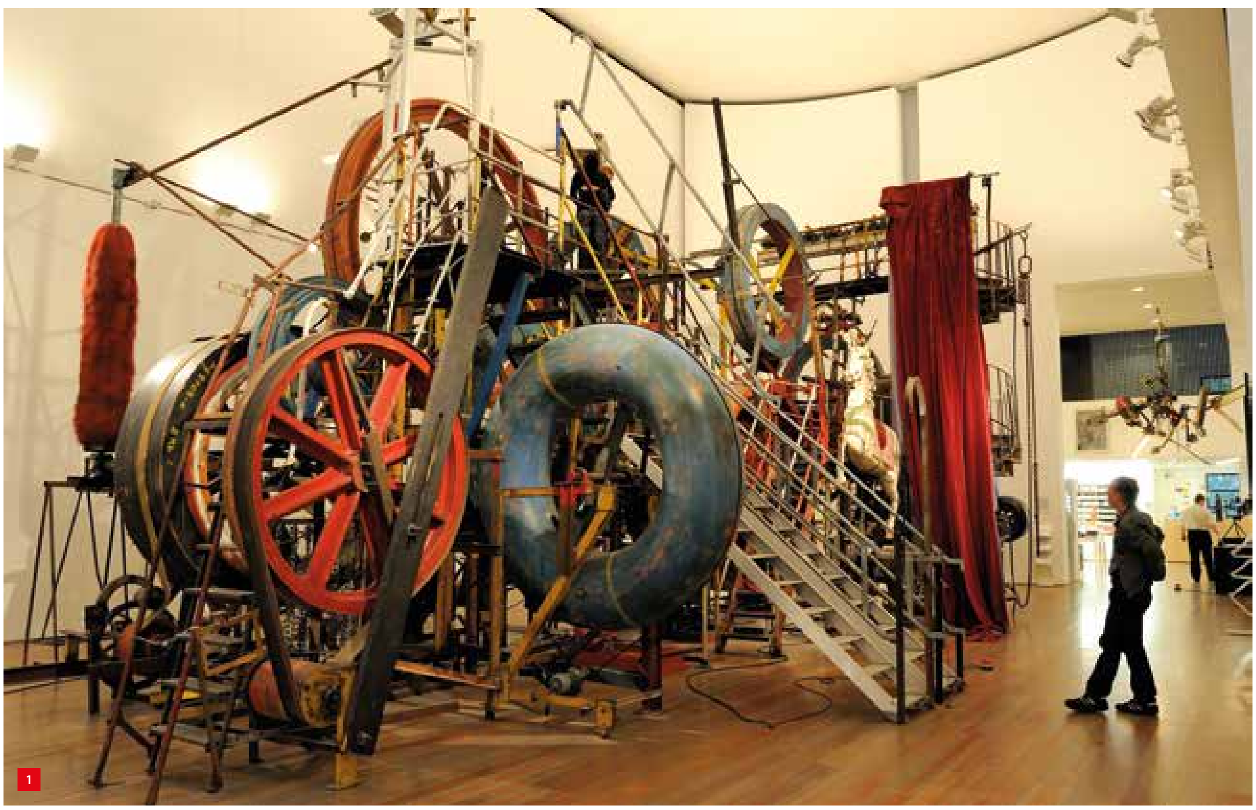
What is art?

This question has probably already been discussed since the first man made a painting on the wall of a cave. It is certainly more than just an aesthetically pleasing decorative object. The jury of the 'Volkskrant Beeldende Kunstprijs' (Volkskrant Visual Arts Prize) 2018 listed a number of useful criteria. The most complete artist:

- is admired for his/her attention to detail;
- surprises by the craziness of his/her art;
- shows good technical craftsmanship;
- celebrates beauty without setting it above everything;
- (includes current themes in a casual manner).

The author considers the second criterion important to distinguish art from decoration: art must surprise, excite (not shock), captivate, touch the viewer, and may even change his view on the world. And also, the third criterion is essential for real art: the craftsmanship with which it has been made.

Art is usually non-functional but wants to express an emotion and capture the senses. It includes a wide range of objects and performances, from paintings and sculptures to poetry, theatre and film. But there is no sharp distinction



1

Jean Tinguely, Grosse Meta-Maxi-Maxi-Utopia (1987), Tinguely Museum, Basel. (Copyright: Hemis / Alamy Stock Photo)

AUTHOR'S NOTE

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between art and design. Designer objects usually do have a user function, with a good design giving added value in use and appreciation. Design aims at mass production, and art is unique or has a limited print. But are the buildings from Gaudi, the furniture from Rietveld or the 3D-printed dresses from Iris van Herpen art or design?

Early history

Art is a mirror of the time in which it is made. This applies both to the subjects and the technology used. The invention of the paint tube around 1840 enabled painters to work outdoors. Photography and film were invented and used for artworks. So, when mechanical moving elements became available these were soon included as well. This is what we now call kinetic art, where the motion is essential for the experience of the artwork.

Usually considered to be the earliest example is the bicycle wheel with fork mounted on a stool by Marcel Duchamp in 1913, one of his provocative 'ready-made' artworks. But the real start is considered to be the sculptor Alexander Calder in 1930, with his hanging mobiles, set in motion by air flow. In the 1930s a few artists, like László Moholy-Nagy, included electric motors in their moving sculptures, using the effects of mechanical noise and light and shadow as well. Around 1950 more artists followed, like Nicolas Schöffer and the best known early kinetic artist, the Swiss Jean Tinguely (1925-1991).

Tinguely created complex mechanisms with multiple motions using rods, wheels and belts. The author uses a picture of the Gigantic Kinetic Toy (see Figure 1 and video [V1]) in his workshop on systems engineering to show what the result may be of an unstructured incremental engineering process. His way of working is a really good example of how engineers should not work.

Tinguely taught himself to weld, and mostly used waste material for his creations. These were (intentionally?) badly constructed, with mechanical play making the motion non-deterministic, and the rattles and squeaks adding to the experience. The type of mechanisms he used were rather limited: most were based on the ordinary crank mechanism. Just like early professional automated machinery, the devices were capable of only one prescribed and repeated motion, with no 'intelligence' from sensors or processing. He used old washing machine motors to drive the mechanisms, which now poses a problem to museum curators, as they have to find similar replacements when these fail. Some of the machines he made were intentionally self-destructing, but most failed after some time anyhow. The website of the Tinguely Museum contains short videos of a number of his creations [1].

Tinguely was the first kinetic artist to use natural elements like tree branches and animal skulls in his work. In this way he wanted to investigate and show the connections or conflicts between the technical and the natural worlds. Most kinetic artists since include natural elements in their work in some form. It may make the viewer to identify himself with the machine and to repulse it at the same time.

The Netherlands

There are quite a number of artists making kinetic art in the Netherlands. A short presentation of six of them may serve to show the differences in their aims and ways of working. It also demonstrates the development from mechanics to mechatronics.

Theo Jansen

This artist designs and builds *strandbeesten* (beach animals), which are fully mechanical driven by external power from the wind. These are multi-legged creatures mainly made from yellow plastic tube connected by cable ties and nylon cord. They come to life on the beach when driven by the wind in sails attached to some outriggers. Their structure resembles pre-historic skeletons.

Jansen compares the plastic tube to proteins in nature: widely available and cheap. His way of working is by means of evolution. The creatures experience a one-year evolution cycle: designed and built in winter, born in spring, tested and improved in summer, declared extinct in autumn. Over several decades the species has made an evolution to improve strength, speed and stability. This evolution leads to increased complexity, with sometimes reduced reliability.

2

Theo Jansen, beach animal Animaris Umerus (2009), Scheveningen. (Copyright: Loek van der Klis)