

An approach to the dynamics and control of a planar tensegrity structure with application in locomotion systems

Valter Böhm · Igor Zeidis · Klaus Zimmermann

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Abstract The use of mechanically compliant tensegrity structures in vibration-driven mobile robots is an attractive research topic, due to the principal possibility to adjust their dynamic properties reversibly during locomotion. In this paper vibration driven planar locomotion of mobile robots, based on a simple tensegrity structure, consisting of two rigid disconnected compressed members connected to a continuous net of four prestressed tensional members with pronounced elasticity, is discussed. The dynamic behaviour of the considered system is nonlinear, due to large vibration amplitudes and friction between robot and environment, and is mainly influenced by the magnitude of prestress. Therefore, the movement performance of the robot can be essentially influenced by the actuation parameters, e.g. by modifying the frequency or the magnitude of actuation the locomotion direction of the system varies. To study the system behaviour, the nonlinear equations of motion are derived and transient dynamic analyses are performed, including the consideration of chaotic system behaviour near to the primary and secondary eigenfrequencies. The dependency of the movement behaviour on the actuation parameters and on the prestress are discussed focused on single-actuated systems with minimal control effort.

Keywords Tensegrity structure · Vibration · Locomotion system · Chaos

V. Böhm (✉) · I. Zeidis · K. Zimmermann
Ilmenau University of Technology, Faculty of Mechanical Engineering
Technical Mechanics Group, 98684 Ilmenau, Germany
e-mail: valter.boehm@tu-ilmenau.de

I. Zeidis
e-mail: igor.zeidis@tu-ilmenau.de

K. Zimmermann
e-mail: klaus.zimmermann@tu-ilmenau.de

1 Introduction

To improve the effectiveness of future mobile robots, new non-conventional locomotion principles are required. By the inspection of complex unstructured environments, the use of shape-variable locomotion systems with pronounced mechanical compliance is necessary or associated with several advantages [1,2]. Mechanically prestressed compliant structures allow the required with the use of few actuators by a simple system design. One specific class of these structures builds tensegrity structures. Compliant tensegrity structures, consisting of a set of rigid disconnected compressed members connected to a continuous net of prestressed tensional members with pronounced elasticity, represent one particular type of these structures. The resulting shape of these structures is defined by the balance between the tensile and compressive forces of the elements. A spatially limited, local impact on the tensegrity structure yields a global change of their shape, independent from the relative position of the actuator [3]. Several additional advantageous properties of these structures make their use in robotics attractive. Robots based on these structures are deployable, lightweight, and have very high strength to weight ratio, and shock absorbing capabilities [3–5]. An overview of actual developments and development directions can be found in [6,7].

The application of tensegrity structures in context of locomotion systems is a recently discussed topic. The first works to mobile robots, based on tensegrity structures have been published in [8–10]. The movement of these robots is primarily based on the pronounced shape change ability of the structures, by using non-conventional locomotion techniques. By using different approaches for actuation and system configurations, this property is implemented in most known prototypes [5,11–13]. An another class of mobile tensegrity robots exploit during locomotion primarily the mechanical

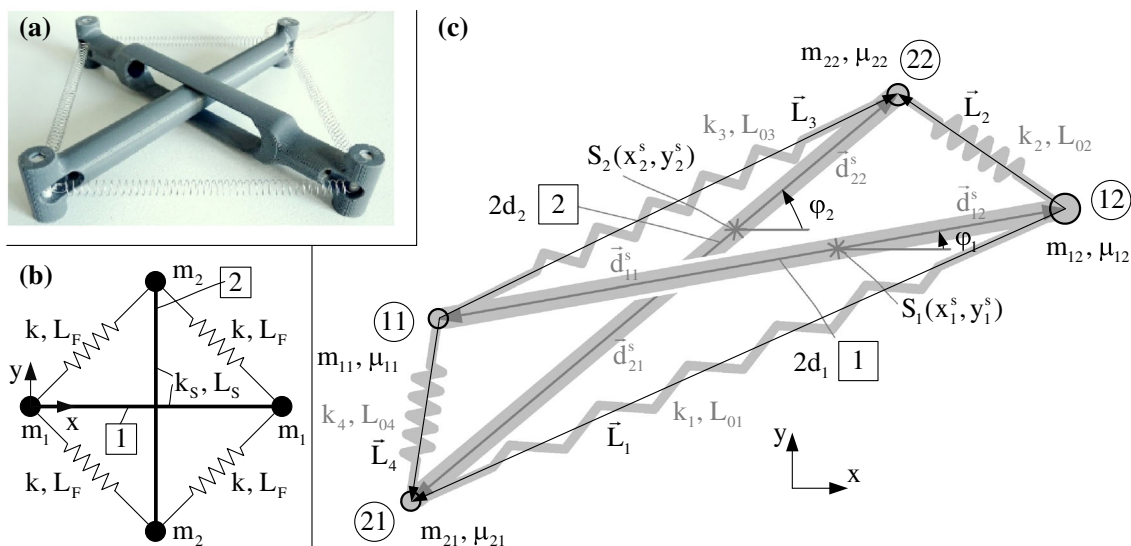


Fig. 1 The considered planar tensegrity structure: physical model (a), mechanical model in the symmetric configuration (b) and in the general case (c)

compliance of the structures, including only moderate shape changes. In [3, 14, 15], two locomotion systems, based on a 3-prism tensegrity structure and on a 4-prism tensegrity structure, were considered. The control of the robots was realized by periodically changing of the effective rest lengths of selected elastic tensional segments. By considering the dynamic behaviour of the systems, possible effective control algorithms were developed using genetic algorithms to achieve locomotion. In these works, and also in [4, 16–20], have been shown, that the trajectory of the movement of the robots with multiple actuators varies significantly under varying the actuation speeds. Due to the complex system dynamics, the direction of locomotion can be changed by modifying the driving frequency, keeping the actuation scheme constant.

The use of compliant tensegrity structures in vibration driven locomotion systems seems to be advantageous mainly for the following reasons:

- due to the pronounced mechanical compliance of the structures, complex modes of vibrations can be induced with a small number of actuators. This property is also characteristic for simple structures, consisting of only a small number of segments
- the dynamic properties of the system are tunable by actively adjusting the initial prestress of the structure, which can be realized by simultaneously changing the length or compliance of selected segments.

Based on these properties, simple locomotion systems with variable movement performance can be realized. The movement speed and direction can be controlled and adapted to different environmental conditions with the frequency and magnitude of actuation and if the initial prestress of the struc-

ture is tunable also with the magnitude of prestress. In the simplest case the initial prestress is not varied during locomotion and the system is driven by only one actuator. To realize variable movement performance in this case, the frequency and/or the magnitude of actuation must be reversibly variable. By using only a single actuator, the system design and also their control could be kept simple. However, detailed theoretical investigations on the system dynamics are necessary, to determine the system parameters, which enable locomotion in different directions in dependence of the actuation frequency or magnitude. In this paper we focus on the dynamics of a single-actuated locomotion system based on a simple planar compliant tensegrity structure. The actuation of the structure is realized by periodically changing the effective rest length of selected compliant tensioned segments. The paper begins by providing the mechanical model of the system in Sect. 2. In this section also the basic dynamic properties of the structure are considered, followed by numerical results for harmonic excitation of a fixed structure and for uniaxial and planar movement of a mobile robot based on the structure in Sect. 3. Finally, in Sect. 4 conclusions and further research directions are considered.

2 Mechanical model and equations of motion

We consider the simple planar tensegrity structure, as depicted in Fig. 1. The structure consists of six segments: two rigid struts (index: $j = 1, 2$, length: $2d_j$), which are indirectly connected by four linear elastic spring elements (with index: $f = 1, 2, 3, 4$, spring constant k_f , rest length L_{0f}). The spring elements are tensioned in the initial equilibrium configuration. The mass of the springs and the inertial properties

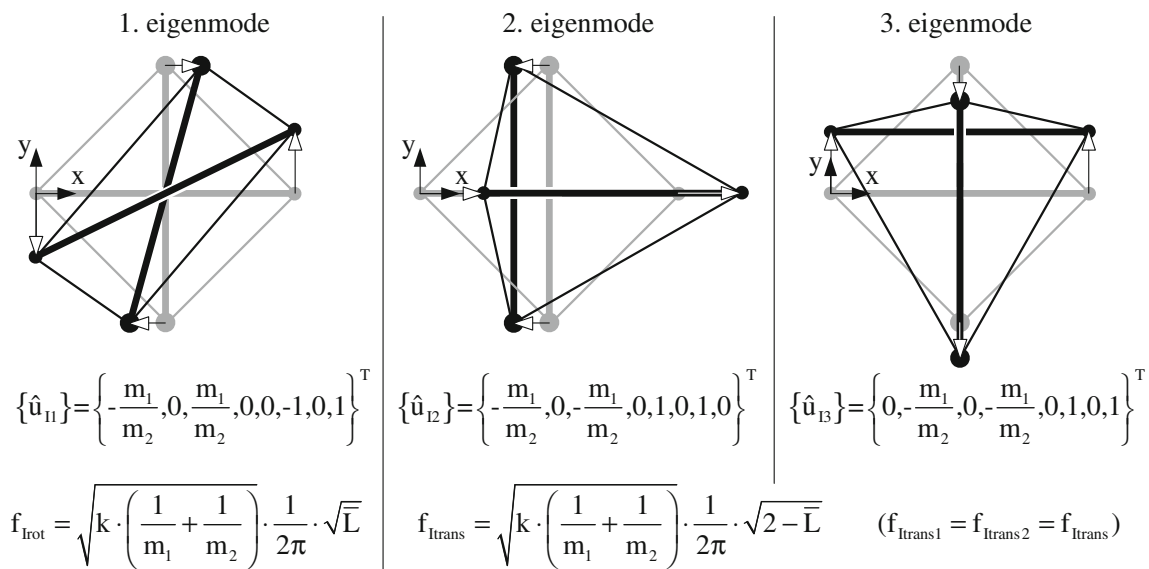


Fig. 2 Relevant eigenmodes and the corresponding eigenfrequencies of the planar tensegrity structure, without any fixation on the x–y plane ($\bar{L} = \sqrt{2}L_F/L_S$, $0 < \bar{L} < 1$, $m_1 \neq 0$, $m_2 \neq 0$, coordinates of the eigenvectors: $\{\hat{u}\} = \{\hat{u}_{21x}, \hat{u}_{21y}, \hat{u}_{22x}, \hat{u}_{22y}, \hat{u}_{11x}, \hat{u}_{11y}, \hat{u}_{12x}, \hat{u}_{12y}\}^T$)

of the struts are neglected. Masses are considered as discrete mass points, placed on the end points of the struts (m_{ji} , with strut endpoint index: $i = 1, 2$). Contacts between the struts or between struts and spring elements are not considered.

2.1 Modal analysis

The basic dynamic properties of tensegrity structures are essentially dependent on the magnitude of their initial prestress. The initial prestress is defined by the segment stiffnesses and segment rest lengths. For the considered planar structure in the symmetric case (see Fig. 1b), if $d_1 = d_2(2d_j = L_S)$, $k_1 = k_2 = k_3 = k_4$ ($k_f = k$), $L_{01} = L_{02} = L_{03} = L_{04}$ ($L_{0f} = L_F$), with linear elastic strut elements (stiffness: k_s), where the strut endpoints $m_{21}, m_{22}, m_{11}, m_{12}$ correspond to nodes 1, 2, 3, 4 respectively, and where each segment of the structure is regarded as one linear spring element, the eigenfrequencies of the structure can be find by solving the eigenvalue problem: $\det(\lambda^2[M] + [K]) = 0$, where $[M] = \text{diag}(m_{21}, m_{21}, m_{22}, m_{22}, m_{11}, m_{11}, m_{12}, m_{12})$ is the mass matrix and $[K]$ the tangent stiffness matrix:

$$[K] = \begin{bmatrix} k & 0 & p & 0 & a & b & a & -b \\ c & 0 & -k_s & b & a & -b & a & a \\ k & 0 & a & -b & a & b & a & b \\ & c & -b & a & b & a & a & a \\ & & c & 0 & -k_s & 0 & & \\ & & & k & 0 & p & & \\ \text{sym.} & & & & c & 0 & & \\ & & & & & & k & \end{bmatrix}, \quad (1)$$

where $p = k(1 - (\sqrt{2}L_F(k_S + k))(\sqrt{2}kL_F + k_S L_S)^{-1})$, $2a = -k - p$, $2b = k - p$, $c = k + k_s + p$. If the masses connected to strut 1, and also to strut 2 are equal ($m_{11} = m_{12} = m_1$, $m_{21} = m_{22} = m_2$), the eigenfrequencies can be determined symbolically. For this case, excluding the three rigid body modes and the two modes corresponding to the longitudinal vibrations of the struts, the three relevant eigenfrequencies and corresponding eigenmodes are depicted in Fig. 2. In the case that strut No. 1 is fixed, the two relevant eigenfrequencies can be solved analogous, using the reduced tangent stiffness and mass matrices ($K(1:4, 1:4)$, $M(1:4, 1:4)$) with following formulas:

$$\begin{aligned} f_{1rot} &= \sqrt{k \cdot \frac{1}{m_2} \cdot \frac{1}{2\pi} \cdot \sqrt{\bar{L}}} \\ f_{1trans} &= \sqrt{k \cdot \frac{1}{m_2} \cdot \frac{1}{2\pi} \cdot \sqrt{2 - \bar{L}}} \end{aligned} \quad (2)$$

The first eigenmode corresponds to a rotational oscillation. The second and third eigenmodes correspond to translational oscillations, with identical associated eigenfrequencies, due to symmetric system parameters. The ratio between the translational and rotational eigenfrequency is independent of the type of bearing and is defined solely by the prestress of the structure: $f_{rot}/f_{trans} = \sqrt{\bar{L}/(2 - \bar{L})}$. Accordingly, if the spring elements are tensioned in the considered initial configuration ($0 < \bar{L} < 1$): $f_{rot} < f_{trans}$, and if the spring elements are not tensioned ($\bar{L} = 1$): $f_{rot} = f_{trans}$. Moreover, it is apparent that with increasing the prestress the rotational eigenfrequency will decrease, and the translational eigenfrequency will increase.

2.2 Equations of motion

In the following we consider movement of the structure for the case, that all strut endpoints lie on a plane ground surface (x - y plane). The gravity vector is perpendicular to the plane ($\vec{g} = -g \cdot \vec{e}_z$, $g = 9.81 \times 10^3 \text{ mm/s}^2$). The isotropic Coulomb friction model is implemented between a supporting plane and strut endpoints. The positions and orientations of the struts are characterized with the location vectors of the centre of masses of the struts $\vec{r}_j^s = x_j^s \vec{e}_x + y_j^s \vec{e}_y$ and with the angles φ_j (Fig. 1c). The location vectors of the strut endpoints can be expressed with:

$$\begin{aligned} \vec{r}_{ji} = \vec{r}_j^s + \vec{d}_{ji}^s = & \left(x_j^s + (2i - 3) \frac{2d_j m_{jk} \cos \varphi_j}{m_{j1} + m_{j2}} \right) \vec{e}_x \\ & + \left(y_j^s + (2i - 3) \frac{2d_j m_{jk} \sin \varphi_j}{m_{j1} + m_{j2}} \right) \vec{e}_y, \end{aligned} \quad (3)$$

where:

$$\begin{aligned} \vec{r}_j^s &= \frac{\sum_i m_{ji} \vec{r}_{ji}}{\sum_i m_{ji}}, \\ \vec{d}_{ji}^s &= \left| \vec{d}_{ji}^s \right| (\cos \varphi_j \vec{e}_x + \sin \varphi_j \vec{e}_y) \cdot (2i - 3) \\ \text{with } \left| \vec{d}_{ji}^s \right| &= d_{ji}^s = 2d_j \frac{m_{jk}}{m_{j1} + m_{j2}}; \quad k = 2/i \quad (i, j = 1, 2). \end{aligned} \quad (4)$$

The moment of inertia of strut j , with respect to the centre of mass is $J_{zz,j} = J_j = \sum_i m_{ji} (d_{ji}^s)^2$. Using the Lagrangian

$$\begin{aligned} L = T - U = & \frac{1}{2} \sum_j \left(\left(\sum_i m_{ji} \right) (\dot{\vec{r}}_j^s)^2 + J_j \dot{\varphi}_j^2 \right) \\ & - \frac{1}{2} \sum_f k_f (L_f - L_{0f})^2, \end{aligned} \quad (5)$$

with $L_f = \left| \vec{L}_f \right|$ —actual length of spring f ($L_1 = |\vec{r}_{21} - \vec{r}_{12}|$, $L_2 = |\vec{r}_{22} - \vec{r}_{12}|$, $L_3 = |\vec{r}_{22} - \vec{r}_{11}|$, $L_4 = |\vec{r}_{21} - \vec{r}_{11}|$), the equations of motion can be written as

$$\begin{aligned} \left(\sum_i m_{ji} \right) \ddot{\vec{r}}_j^s &= \sum_i \left(\vec{A}_{ji} + \vec{F}_{ji} + \vec{F}_{ji}^R \right), \\ \left(4d_j^2 \frac{m_{j1} m_{j2}}{m_{j1} + m_{j2}} \right) \ddot{\varphi}_j &= \left(\sum_i \left(\vec{d}_{ji}^s \times \left(\vec{A}_{ji} + \vec{F}_{ji} + \vec{F}_{ji}^R \right) \right) \right) \cdot \vec{e}_z, \end{aligned} \quad (6)$$

where \vec{F}_{ji} , \vec{F}_{ji}^R and \vec{A}_{ji} are the external force (vector in the x - y plane), the frictional force, and the sum of the spring forces at strut endpoint ji . The last quantities can be written as

$$\vec{F}_{ji}^R = \begin{cases} -\mu_i m_{ji} g \frac{\dot{\vec{r}}_{ji}}{\left| \dot{\vec{r}}_{ji} \right|}, & \left| \dot{\vec{r}}_{ji} \right| > 0 \\ -\min \left\{ \mu_i m_{ji} g, \left| \vec{A}_{ji} + \vec{F}_{ji} \right| \right\} \frac{\vec{A}_{ji} + \vec{F}_{ji}}{\left| \vec{A}_{ji} + \vec{F}_{ji} \right|}, & \left| \dot{\vec{r}}_{ji} \right| = 0 \end{cases}, \quad (7)$$

$$\sum_i \vec{A}_{ji} = (3 - 2j) \cdot \sum_f \left(k_f \left(\left| \vec{L}_f \right| - L_{0f} \right) \cdot \frac{\vec{L}_f}{\left| \vec{L}_f \right|} \right); \quad \text{with } \left| \vec{L}_f \right| \neq 0.$$

Several methods of actuation are known to generate movement of tensegrity structures (see e.g. [3, 11]). In our model the method introduced in [3] is implemented: the movement of the structure is induced by forces, due to periodically changing the effective rest length ($L_{0f}(t)$) of selected spring elements:

$$L_{0f}(t) = L_{0f, \text{init}} \cdot (1 - L_{f, \text{fak}} \cdot \sin(2\pi f \cdot t)) \quad (8)$$

or

$$L_{0f}(t) = L_{0f, \text{init}} \cdot (\min \{1, 1 - L_{f, \text{fak}} \cdot \sin(2\pi f \cdot t)\}) \quad (9)$$

with $L_{f, \text{fak}}$ —magnitude parameter and $L_{0f, \text{init}}$ —initial rest length of spring Nr. F , and f —driving frequency.

3 Numerical simulation

The numerical simulations are obtained in Matlab®R2009b, using the fourth-order Runge-Kutta integration algorithm. The oscillation behaviour of the structure is considered for two cases. In the first case strut No. 1 is fixed. This case enables the simplified consideration of characteristic influences on the vibration behaviour of the structure. In the second case the structure is free movable on the x - y plane.

3.1 Oscillation of the system in the fixed case

Following the results of the modal analysis complex vibration modes are to be expected during movement of the structure. The movement is induced by general movements of the two struts. Depending on the driving frequency rotational or translational vibrations of the struts are dominant: the vibration modes vary qualitatively with change of the driving frequency. This effect can be easily shown by considering exemplarily a simplified case, by which strut No. 1 is fixed ($\varphi_1 = 0$, $x_1^s = d_1$, $y_1^s = 0$). The motion of strut No. 2 is induced by harmonic change of the effective rest length of spring element No. 3 (see Eq. 8). In the simulations the driving frequency ($f = 2$ –40 Hz), the initial prestress of the structure ($L_{0f} = 10$ –70 mm) and the magnitude of actuation ($L_{3, \text{fak}} = 0.1$ –0.5) were varied and following fixed parameter values are used: $k_f = 0.25 \text{ N/mm}$, $m_{21} = m_{22} = 10 \text{ g}$, $d_1 = d_2 = 50 \text{ mm}$ $\mu_{21} = \mu_{22} = 0.1$.

Selected steady state results, that show the assumed behaviour of the system, are shown in Fig. 3 for different driving

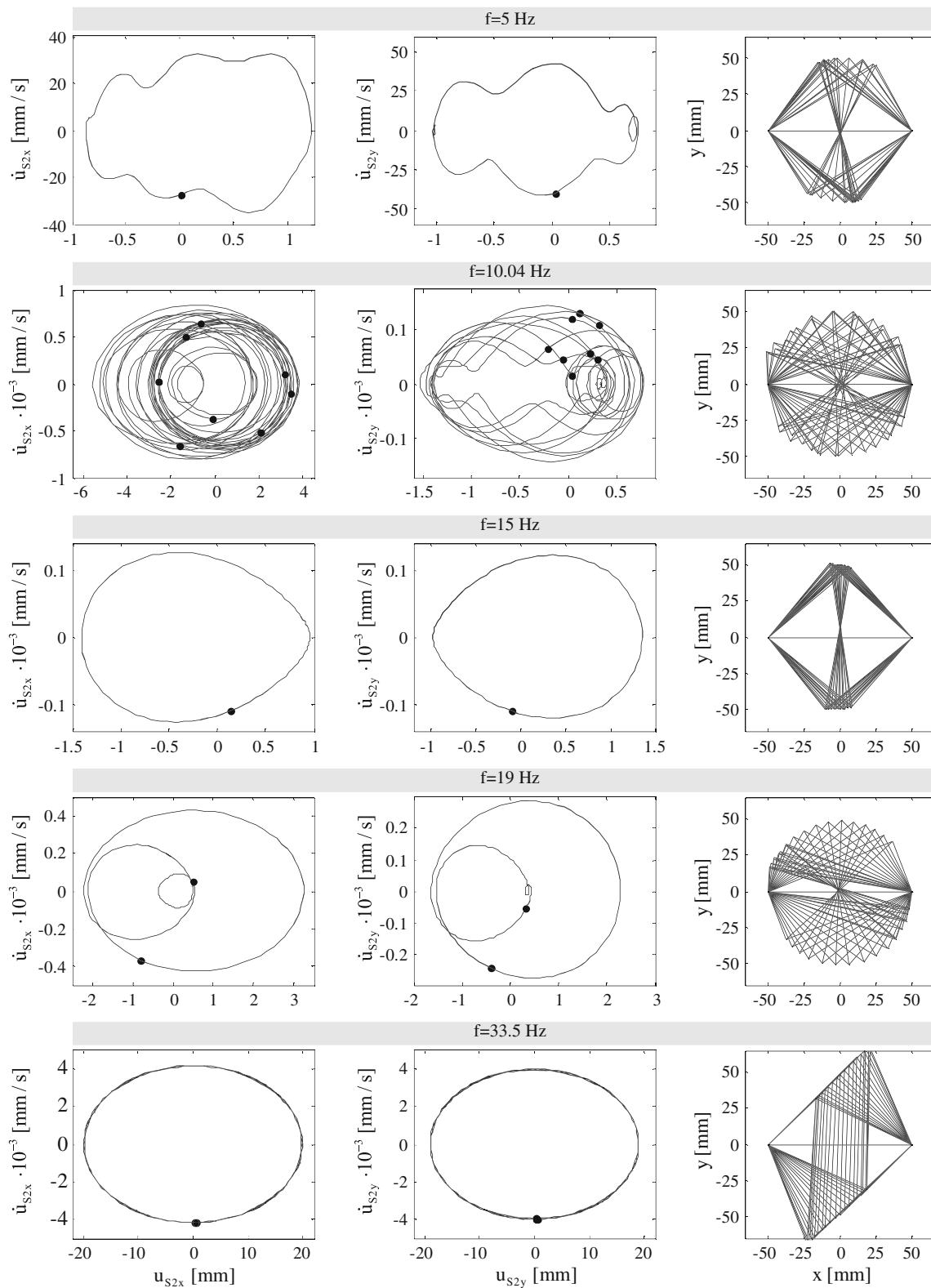
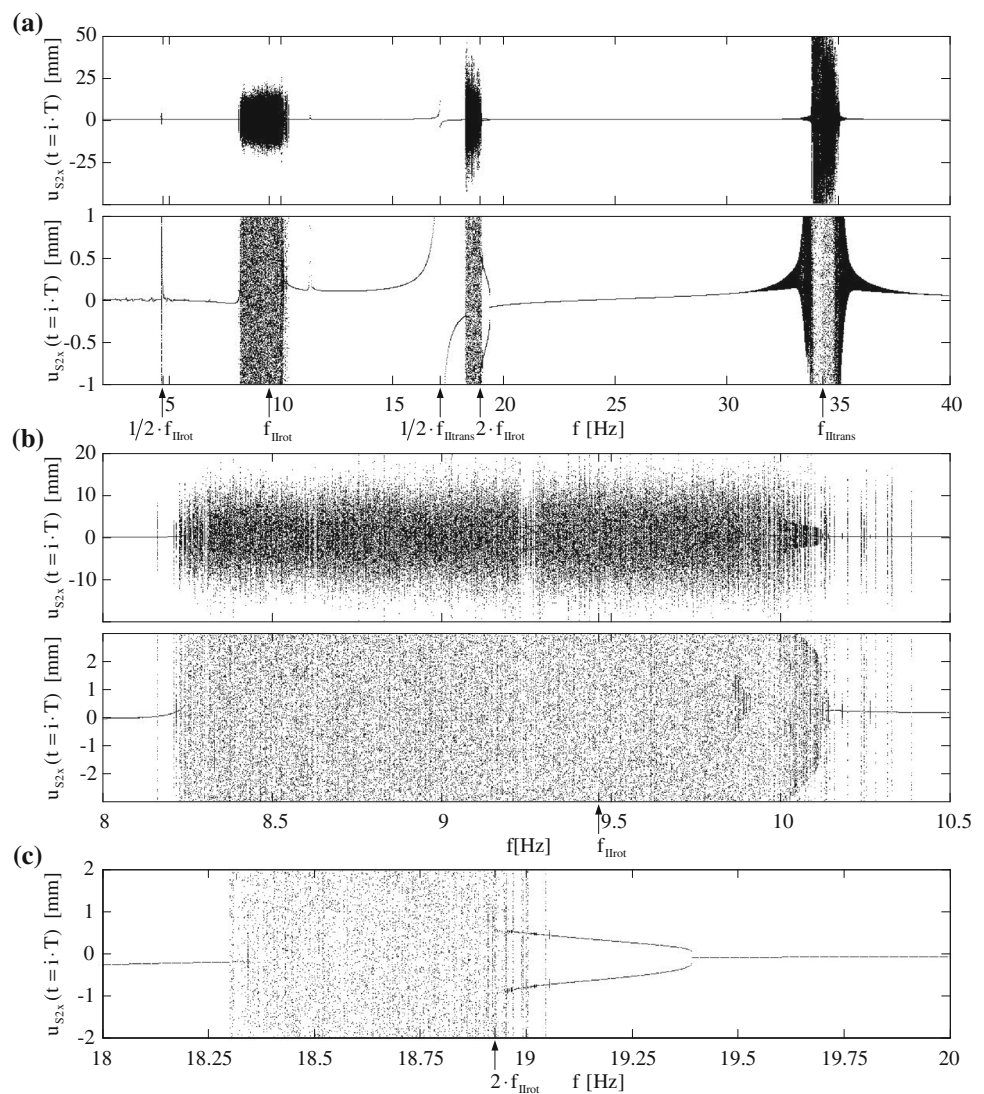


Fig. 3 Phase portraits and oscillation modes of the structure for different driving frequencies f in the case that strut No. 1 is fixed. *Black dots* mark the Poincaré points. u_{s2x} and u_{s2y} are the displacements of the centre of mass of strut No. 2 in the x - and y -directions

Fig. 4 Bifurcation diagram for $f = 2\text{--}40$ Hz (a) and detailed views (b, c). ($i = 1,900\text{--}2,000$; $i \in \mathbb{N}$, $T = 1/f$, frequency resolution: $0.01/0.002$ Hz in the frequency ranges at the primary and secondary eigenfrequencies)



frequencies f . The primary eigenfrequencies for this system with $L_{of} = 10$ mm and $L_{3,fak} = 0.5$ are: $f_{Irot} = 9.4634$ Hz, $f_{Itrans} = 34.3068$ Hz. The bifurcation diagram for the system with the same parameter values is depicted in Fig. 4. Here for each f , 100 cycles, i.e. Poincaré points of the displacement of the center of gravity of strut No. 2 in the x -direction have been calculated after an initial number of 1,900 cycles.

Based on the simulations also under varying the initial prestress and magnitude of actuation following important results are found:

- the system show chaotic behaviour near to the primary and secondary eigenfrequencies
- with increasing the initial prestress of the system the width of frequency ranges with chaotic behaviour will be smaller (see Fig. 5a–c)

- the oscillation of the movable strut is a coupled rotational and translational oscillation in frequency ranges near the primary and secondary eigenfrequencies, and is not dependent on the magnitude of the initial prestress and on the ratio between the primary eigenfrequencies.

With respect to the planned application of this structure in a vibration driven locomotion system are all these properties important. An effective locomotion of the system can be realized as expected if the driving frequency is near to the resonance. However, the driving frequency must be carefully selected: chaotic system behaviour during locomotion is to be avoided. Additionally, by changing the prestress magnitude the vibration and therefore the locomotion properties (direction, speed) can be influenced.

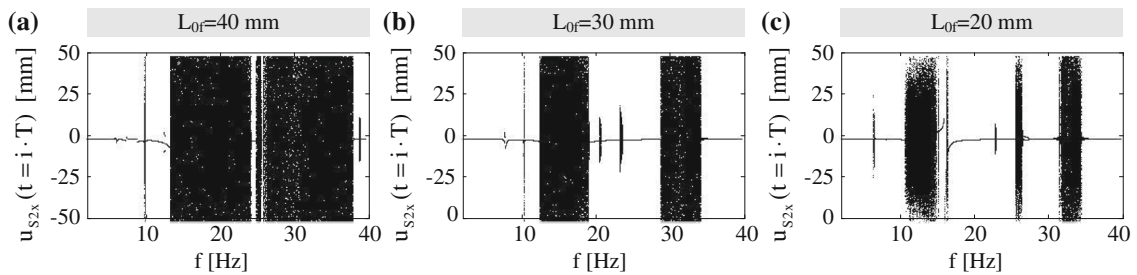


Fig. 5 Bifurcation diagrams for different magnitudes of the initial prestress, realized by using springs of different initial rest length ($L_{3,fak} = 0.5$, freq. res.: 0.01, $i = 1,900-2,000$)

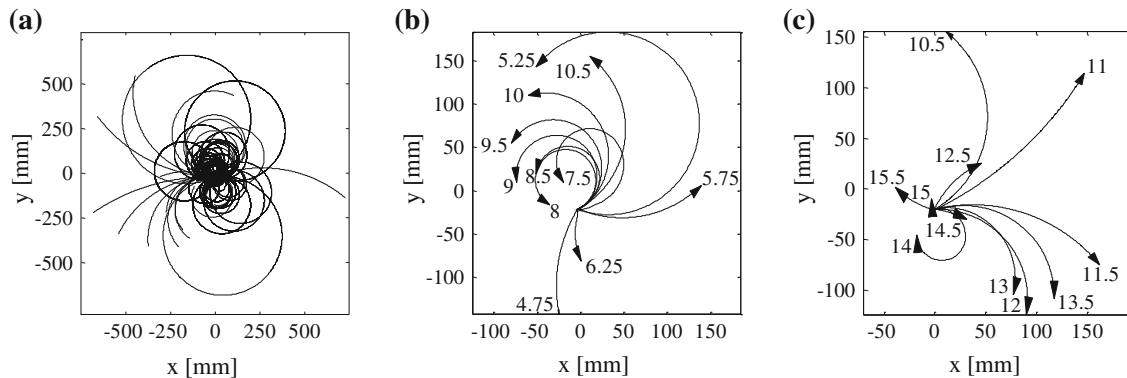


Fig. 6 Trajectories, showing the travel of the centre of mass of the planar tensegrity structure with asymmetrical mass distribution for different driving frequencies ($f = 2-32$ Hz) for 5,000 actuation periods

(a) and for the first 300 actuation periods (b, c the values besides the trajectories mark the corresponding driving frequencies in Hz)

3.2 Vibration induced locomotion

General possibilities for the uniaxial and planar locomotion of systems based on the considered structure are discussed in [21]. In the following we focus on locomotion of the system in the x–y plane. The type of locomotion (uniaxial/planar) depends on the geometric configuration, the mass distribution and friction properties. Planar movement of the system can be realized if an asymmetry exists in these system parameters. With single actuation, realized by periodic altering the initial rest length of only one spring element, one of the following conditions must be fulfilled: (i) the mass of one mass point or (ii) the friction coefficient at one contact point or (iii) the initial rest length of one non actuated spring element is different to the others or (iv) the strut lengths are non equal. As example therefore, selected numerical results for the case of asymmetrical mass distribution [$m_{11} = m_{12} = m_{22} = 25$ g, $m_{21} = 100$ g, $k_f = 0.25$ N/mm, $L_{0f} = 10$ mm, $d_1 = d_2 = 50$ mm, $\mu_{11} = \mu_{12} = \mu_{21} = \mu_{22} = 0.1$, actuated spring: No. 3 with $L_{3,fak} = 0.5$, see Eq. (9)] are depicted in Figs. 6 and 7. Figure 6 shows the trajectory of the center of mass of the structure during locomotion for different driving frequencies. The system moves after the transient phase on a circular path, where path radius and movement direction depend on the driving frequency. By changing the driving

frequency during locomotion the movement direction can be changed (Fig. 7). Thus, any movement of the system on a plane can be realized with only one actuator by modifying only of the driving frequency. By the design of systems with the considered asymmetrical mass distribution it must be taken into account, that the eigenfrequencies corresponding to translational oscillations are non equal. Therefore, the number of critical frequency regimes in the regions of the primary and secondary eigenfrequencies is larger, at which chaotic movement behaviour of the system occurs. Thus, the number of usable driving frequency regimes, which cause regular oscillations during locomotion, decreases.

4 Conclusions

This paper presents a concept for vibration-driven locomotion systems based on tensegrity structures as a basis for terrestrial mobile robots. It was demonstrated that vibration-driven locomotion with compliant tensegrity structures can be realized using only of a single actuator. It was shown, that with asymmetric tensegrity structures vibration-driven locomotion in the plane is possible by periodic dynamic excitation of only a single actuator. The locomotion direction is defined only by the driving frequency. As a further result it

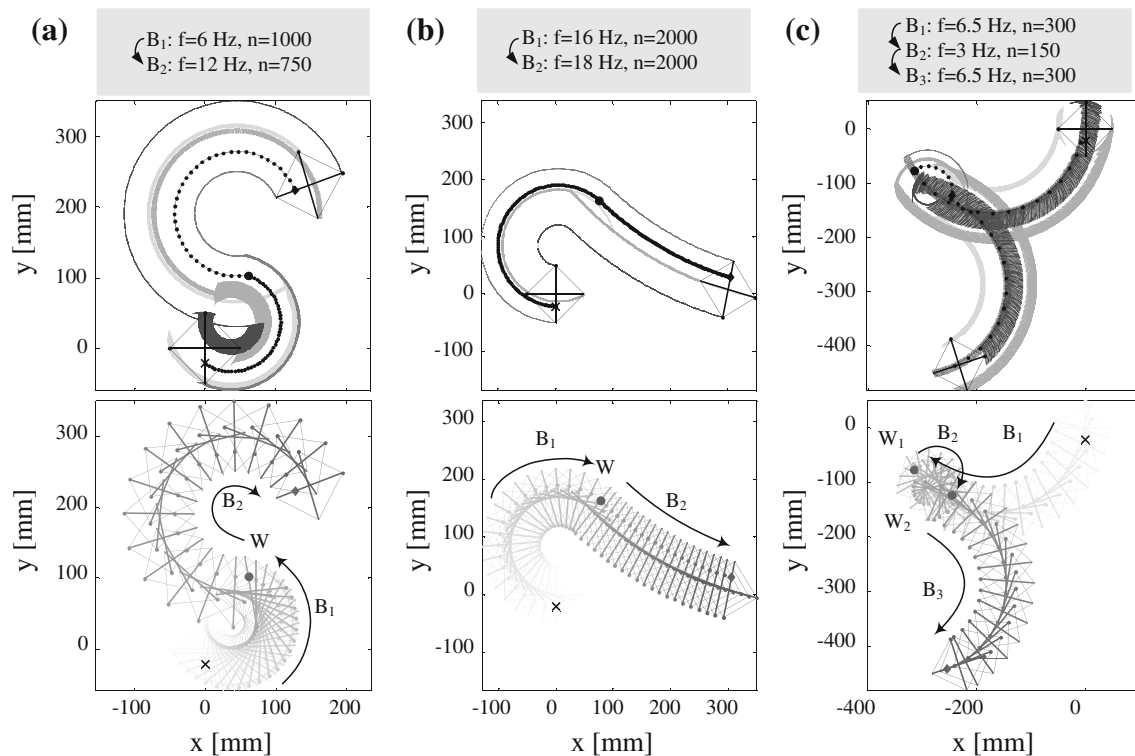


Fig. 7 Vibration induced locomotion of the planar tensegrity structure with asymmetrical mass distribution. In the three depicted cases the change of the driving frequency causes the change of the locomotion direction (**a** 6→12 Hz; **b** 16→20 Hz; **c** 6.5→3→6.5 Hz). *Top* trajec-

jectories of the points 11, 12, 21, 22 and the centre of mass; *bottom* the system during locomotion (*arrows* motion direction, *n* number of actuation periods, *x* start position, *open diamond* end position, *W* system position by the change of the driving frequency)

was shown, that chaotic system behaviour near to the primary and secondary eigenfrequencies can occur, so that a careful selection of driving frequencies is needed during locomotion.

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