

EXPERIMENTAL, THEORETICAL, AND NUMERICAL INVESTIGATION OF TEMPERATURE EFFECTS ON LOAD-DEFLECTION BEHAVIOR OF TAPE SPRINGS

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ABSTRACT: In this paper, folding of carbon steel tape springs under steady-state hot and cold temperature conditions is studied analytically, numerically, and experimentally. To consider the thermal load effects on deflection of tape springs, an existing model, namely Mansfield's theory is modified. An appropriate test apparatus is developed to test the tape at several temperature levels. Finite element modeling is implemented to do more validations on theoretical results. It is observed that rising in temperature causes a decrease in maximum moments stored in tape, and low temperature makes this value greater. The results also confirm that the extended theory successfully predicts buckling behavior of tape springs in different temperature conditions for the tapes with length to width ratios greater than ten.

KEY WORDS: tape spring, load-deflection, folding, high temperature, low temperature.

NOMENCLATURE

L_1, L_0	: (Final, initial) lengths	$\varepsilon_{x,0}, \varepsilon_{x,1}$	: Longitudinal strains due to stress-free state and applying moment
β	: $1 + \alpha_T(T_1 - T_0)$	α_T	: Thermal expansion coefficient
x, y	: Longitudinal and lateral coordinates, respectively	w_0, w_1	: Deflections of the strip
V_F, V_M	: Strain energies (flexural and middle surface)	w^*	: Chord wise distortion of the strip
R, r	: (Transversal, Longitudinal) curve radiuses	M	: Moment
L	: Length of strip	t	: Strip thickness
$\theta_+^{\max}$	: Angle relevant to the $M_+^{\max}$	a	: Width of the strip
$k_{x0}, k_{xy0}, k_{y0}, k_x, k_{xy}$	: Curvatures of the strip along with its index(es) coordinate(s) ('0' states initial curvature)	T_1, T_0	: (Final, initial) temperatures
		$M_+^{\max}$	: Maximum moment stored in tape

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1 INTRODUCTION

Tape springs are commonly employed in satellites to deploy and maintain some satellite subsystems such as solar arrays, antennas, and gravity gradient booms in a specific position [1–3]. In low earth orbit (LEO) and geosynchronous orbit, solar arrays pass through cycle temperature changes from -75°C to $+65^{\circ}\text{C}$ and -145°C to $+55^{\circ}\text{C}$ respectively [4]. In LEO, exposed systems, such as solar arrays and deployable mechanisms, are impressible to temperature variations [4]. Spacecrafts are subjected to radiant thermal energy, which these thermal energies may affect the mechanical parts such as tape spring [5]. Solar radiation is from a fixed direction. Therefore, spacecraft's outer surfaces gain high temperature from one side and lose these thermal energies from another side. This circle changes with the spinning speed; hence, temperature varies for external instruments, and values of temperature depend on spinning angles [6]. For example, the highest ascent temperatures measured on the inside of the payload fairing have ranged from 27°C for the Orbiter to 204°C for Delta and Atlas vehicles [7]. Change in temperature changes the physical properties of materials such as Young's modulus, thermal expansion coefficient, and thermal conductivity [8].

Young's modulus of metals decreases with an increase in temperature, which is due to variation in binding energy between neighboring atoms. Researchers [9–12] experimentally studied the behavior of carbon steels at elevated temperature conditions and obtained an analytical formula to predict elastic modulus according to temperature variation. Tape springs are defined as thin metallic strips with a curved cross section, which have the key property that they are continuous, i.e. they contain no mechanical hinges or other folding devices and yet they can still be folded elastically [13].

Over the years, researchers have developed different mathematical models for tape springs. Firstly, Wust [14] studied the behavior of the tape spring bending and developed his theory based on opposite-sense bending. Afterward, Mansfield [15] developed another theory for deflection of tape springs. Seffen and Pellegrino [16] studied the deployment dynamics of a tape spring. Martin et al. [17] considered folding of tape spring by rod model with flexible cross-section and confirmed Seffen and Pellegrino's [16] estimation. Metals at high temperatures and elastomers at usual temperatures show viscoelastic behavior. The performance of components that are made from viscoelastic materials depends on working conditions such as temperature, frequency, and amplitude of applied loads [18, 19]. Kwok and Pellegrino [20] also examined the time and temperature-dependent behavior of homogeneous linear viscoelastic tape springs during folding stowage and deployment. In present work, the effects of varying temperature on carbon steel tape spring are investigated us-

ing theoretical, experimental, and finite elements methods. In Section 2, a previous theory on tape spring deflections under bending moments is extended in the case of applying the thermal expansion coefficient and defining physical properties as temperature dependence terms. In Section 3, an appropriate test apparatus for tests under high and low temperatures are presented. In Section 4, tape springs are modeled using the finite element method, and loads are applied under different steady-state thermal conditions. In simulations, the physical properties of tape springs are defined to change with the temperature.

2 PROBLEM STATEMENT AND ITS MECHANICAL MODELING

The investigated tapes in this paper have length to width ratio greater than 5. Therefore, they classify in the long-strips category [15]. Here, the theory of Mansfield for long strip tapes is extended by considering the thermal expansion relation and taking into account the physical characteristics of the tape as temperature dependence terms. A schematic of a tape spring and its geometric parameters are illustrated in Fig. 1. A tape spring has four geometrical properties, its transverse curve radius, “ R ”, subtended angle, “ α ”, its thickness, “ t ” and the strip length, “ L ”. Bending of a tape spring is in two senses, which are known as the same-sense and opposite-sense bending, Fig. 2. The behavior of tape springs under these two senses of bending was discussed by Seffen et al. [16]. The numeric values for physical properties of carbon steel in various temperatures extracted from imperial relation that has been presented by [8], are summarized in Table 1. The mechanical and geometrical specifications of model are given in Table 2.

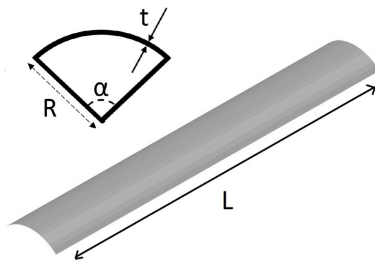


Fig. 1: Scheme of a typical tape spring.

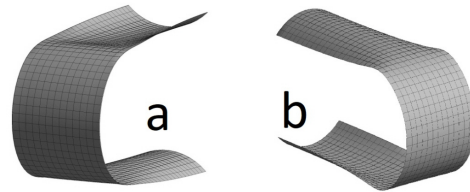


Fig. 2: (a): opposite sense bending (b): same sense bending.

Table 1: Physical properties of carbon-steel material in different temperatures [8]

T [°C]	E [GPa]	$\alpha_T (\times 10^{-6})$ [°C ⁻¹]	Thermal conductivity [W/(m °C)]
100	196.7	11.12	47.86
85	197.6	01.12	48.32
50	199.8	77.11	48.15
-50	206	04.11	47.25
-85	208.2	77.10	47.25
-100	209.1	65.10	47.25

Table 2: Physical and geometrical specifications of the specimens

t [mm]	R [mm]	α [rad]	ν	E [GPa]	α_T [°C ⁻¹ $\times 10^{-6}$]	ρ [kg/m ³]
0.18	16.14	1.55	0.29	201.5	57.11	7850

2.1 PROBLEM FORMULATION

The fractional change in length of an isotropic material due to thermal variation is described in Eq. (1) [10], and the thermal induced strain is expressed as Eq. (2)

$$(1) \quad L_1 - L_0 = L_0 \alpha_T (T_1 - T_0),$$

$$(2) \quad \varepsilon_T = \alpha_T (T_1 - T_0).$$

Due to changes in temperature, thermal expansion affects geometrical parameters such as longitudinal and transverse curvature radiuses of the tape. A curvature is defined as “ $1/r$ ” where “ r ” is the radius of an approximating circle passing through points on the curve. The curvature equation affected by thermal expansion is

$$(3) \quad k = \frac{1}{r} \left(\frac{1}{1 + \alpha_T (T_1 - T_0)} \right).$$

2.2 MANSFIELD’S MODEL

Mansfield’s theory discussed about torsion and bending of long strips with lenticular and constant thickness. In this theory, the effect of small initial longitudinal, twisting, and transverse curvatures was considered which were assumed to be constant. The length of the tape is assumed very larger than its width, which means tape’s length was expected infinity long. However, a comparison of the theory with the test and finite element simulation shows that ignoring end effect in the theory can considerably decrease the maximum moment stored in the tape [16]. In this paper, we extend

the theory of Mansfield by applying thermal expansion coefficient to predict the behavior of tape under high and low thermal conditions. Also effect of temperature on mechanical and thermal parameters of the tape is considered.

2.3 EXTENSION OF MANSFIELD'S MODEL

The equations were started with the chord-wise distortion relations. The relation for deformation of the strip in the stress-free state was written as [15]

$$(4) \quad w_0 = \frac{1}{2}k_{x0}x^2 + k_{xy0}xy + \frac{1}{2}k_{y0}y^2 .$$

For the condition that the strip is subjected to an external moment and torque, it was written as below:

$$(5) \quad w_1 = \frac{1}{2}k_x x^2 + k_{xy}xy + w^*(y) .$$

Total distortion of the strip is a combination of mechanical distortion and the impact of thermal expansion in distortion. When the strip is subjected to a moment, torque and steady-temperature condition, its deflection would be described as

$$(6) \quad w_1 = \left(\frac{1}{2}k_x x^2 + k_{xy}xy + w^*(y) \right) \beta ,$$

where β is equal to $1 + \alpha_T \Delta T$. The longitudinal strains relevant to introduced relations in Eq. (4) and Eq. (5) were driven in the form of:

$$(7) \quad \varepsilon_{x,0} = \frac{1}{2}y^2(k_{xy0}^2 - k_{xy0}k_{y0}) + A_0 + B_0y ,$$

$$(8) \quad \varepsilon_{x,1} = \frac{1}{2}y^2k_{xy}^2 - k_x w^* + A_1 + B_1y .$$

If initial thermal strain from Eq. (2) be added to the mechanical strains, the new form of Eq. (8) would be as Eq. (9) and the total strain can be written as Eq. (10)

$$(9) \quad \varepsilon_{x,1} = \frac{1}{2}y^2k_{xy}^2 - k_x w^* + A_1 + B_1y + (\beta - 1) ,$$

$$(10) \quad \varepsilon_x = \varepsilon_{x,1} - \varepsilon_{x,0} .$$

Therefore, total strain would be in form of Eq. (11), where A and B stand for $A_1 - A_0$ and $B_1 - B_0$ respectively and can be calculated from equilibrium equations in the plane of the strip by equations (12a) and (12b)

$$(11) \quad \varepsilon_x = \frac{1}{2}y^2(k_{x0}k_{y0} + k_{xy}^2 - k_{xy0}^2) - k_x w^* + By + A + \beta - 1 ,$$

$$(12a) \quad \int_{-a/2}^{a/2} Et\varepsilon_x dy = 0 ,$$

$$(12b) \quad \int_{-a/2}^{a/2} yEt\varepsilon_x dy = 0 .$$

The total strain energy of the strip, V , is a combination of the middle-surface strain energy V_M , and flexural strain energy V_F . The strain energy per unit of length caused by middle-surface stresses were defined as Eq. (13)

$$(13) \quad V_M = \frac{1}{2} \int_{-a/2}^{a/2} Et\varepsilon_x^2 dy = 0 .$$

Using Eq. (11), which takes into account the thermal strains, the Eq. (13) is modified as follows:

$$(14) \quad V_M = \int_{-a/2}^{a/2} -k_x Et \left(\frac{1}{2} y^2 (k_{x0} k_{y0} - k_{xy0}^2 + k_{xy}^2) - k_x w^* + A + By + \beta - 1 \right) dy .$$

Strip's flexural strain energy was written as Eq. (15)

$$(15) \quad V_F = \int_{-a/2}^{a/2} \frac{1}{2} D \left(\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial xy} + \frac{\partial^2 w}{\partial y^2} \right)^2 - 2(1-v) \left(\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} - \frac{\partial^2 w}{\partial xy} \right) \right) dy ,$$

where

$$(16) \quad w = w_1 - w_0 ,$$

$$(17) \quad D = \frac{Et^3}{12(1-v^2)} .$$

Using Eqs. (4)–(6) will give Eq. (15) in form of

$$(18) \quad V_F = \int_{-a/2}^{a/2} \frac{1}{2} D \left(\left(k_x \beta + k_{xy} \beta - k_{x0} - k_{xy0} + \left(\frac{\partial^2 w^*}{\partial y^2} \right) \beta - k_{y0} \right)^2 - 2(1-v) \left((\beta k_x - k_{x0}) \left(\frac{\partial^2 w^*}{\partial y^2} \right) \beta - k_{y0} \right) - (\beta k_{xy} - k_{xy0})^2 \right) dy.$$

The condition that the total strain energy is minimum needs w^* to satisfy Eq. (19)

$$(19) \quad \frac{\partial V}{\partial w^*} = 0,$$

which yields Eq. (20) and substituting A and B in Eq. (20), we obtain Eq. (22)

$$(20) \quad D\beta^2 \left(\frac{\partial^4 w^*}{\partial y^4} \right) - k_x Et \left(\frac{1}{2} y^2 (k_{x0} k_{y0} - k_{xy0}^2 + k_{xy}^2) - k_x w^* + A + By + \beta - 1 \right) = 0,$$

where

$$(21) \quad A = -\frac{1}{24} (k_{x0} k_{y0} - k_{xy0}^2 + k_{xy}^2) a^2 + \beta - 1, \quad B = 0,$$

$$(22) \quad \frac{1}{4\lambda^4} \frac{\partial^4 w^*(y)}{\partial y^4} + w^*(y) - \eta \left(y^2 - \frac{a^2}{12} \right) = 0$$

where

$$(23) \quad \lambda = \sqrt[4]{\frac{Et k_x^2}{4D\beta^2}}, \quad \eta = \frac{1}{2k_x} (k_{x0} k_{y0} - k_{xy0}^2 + k_{xy}^2).$$

The side edges of the strip are free; Therefore, the four boundary conditions for the governing Eq. (22) are defined as Eq. (24), using BCs will give w^* as Eq. (26)

$$(24) \quad D\beta^2 \left(\frac{\partial^3 w^*(y)}{\partial y^3} \right) \Big|_{y=\pm \frac{a}{2}} = 0, \quad D\beta^2 \left(\frac{\partial^2 w^*(y)}{\partial y^2} + P \right) \Big|_{y=\pm \frac{a}{2}} = 0,$$

in which

$$(25) \quad P = v(k_x - k_{x0}) + k_{xy} - k_{xy0} - k_{y0}$$

$$(26) \quad w^* = h_3 \eta \lambda^2 (a^2 - 12y^2) + h_2 12 \left(\eta + \frac{P}{2} \right) (\zeta_1 - \zeta_2) + h_1 \zeta_1 (P + 2\eta),$$

where

$$(27) \quad \left\{ \begin{array}{l} h_1 = \frac{-\cos(\Phi) \sinh(\Phi)}{\lambda^2 (\cos(\Phi) \cdot \sin(\Phi) + \cosh(\Phi) \sinh(\Phi))} \\ h_2 = \frac{\cos(\Phi) \sinh(\Phi) + \sin(\Phi) \cosh(\Phi)}{12\lambda^2 (\cos(\Phi) \sin(\Phi) + \cosh(\Phi) \sinh(\Phi))} \\ h_3 = \frac{-\cos(\Phi) \sin(\Phi) - \cosh(\Phi) \sinh(\Phi)}{12\lambda^2 (\cos(\Phi) \sin(\Phi) + \cosh(\Phi) \sinh(\Phi))} \\ \zeta_1 = \cos(\lambda y) \cosh(\lambda y), \quad \zeta_2 = \sin(\lambda y) \sinh(\lambda y), \quad \Phi = \frac{\lambda a}{2}. \end{array} \right.$$

The reaction moment is related to the curvature by Eq. (28) [15]. This equation is still valid in extended theory, because thermal expansion has effects on the geometrical parameters of the tape

$$(28) \quad M = \frac{dV}{dk_x}.$$

2.4 SOLUTION

Mathematical software “Maple” is used to substitute w^* into the equations and obtain the values of the moment for each longitudinal curvature quantity. The diagrams of moment reaction versus curvature for different values of $T_2 = (-100, +100, 22)^\circ\text{C}$ are illustrated in Fig. 3. It is observed from this figure that at the same rotation angles, as much as β approaches the value of unity ($\Delta T = 0^\circ\text{C}$), the diagram obtained from the extended theory coincides with its counterpart from the Mansfield’s

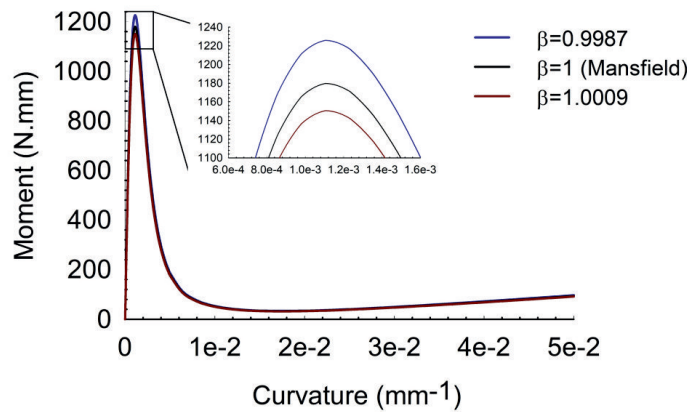


Fig. 3: Moment curvature graph for different β values.

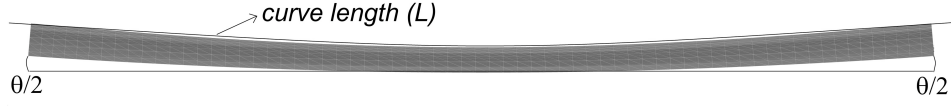


Fig. 4: Longitudinal side-view of tape just before buckling.

theory. The moment reaction results fall into lower values for elevated temperatures ($\beta > 1$), and the moment value becomes lesser, where $\beta < 1$. In small rotation angles, the middle surface curvature, and longitudinal curvature are equal approximately (see Fig. 4); therefore, angle of bending can be calculated theoretically using Eq. (29):

$$(29) \quad \theta = L(1/r) = Lk_x.$$

Numerical values for the temperature variation effects based on Eq. (28) are given in Table 3. The numerical outcome of the $M_+^{\max}$, in theory, proves that elevated and reduced temperatures have been influenced on maximum moment stored in tape. With using this extended theory, one can predict the behavior of tape spring in different temperatures while its solution for room temperature ($\beta = 1$) is equal with Mansfield's theory.

Table 3: Numerical values obtained from extended theory, Eq. (28)

T [°C]	$M_+^{\max}$ [N mm]	$\theta_+^{\max}$ [degree]	
		for $L = 150$ mm	for $L = 270$ mm
22	1179.7201	9.497	17.096
50	1169.3654	9.497	17.096
85	1155.9744	9.497	17.096
100	1150.4836	9.497	17.096
-50	1207.0645	9.497	17.096
-85	1220.4089	9.497	17.096
-100	1225.8715	9.497	17.096

3 EXPERIMENTAL TESTS

To test the linear elastic, buckling, and post-buckling deformation behavior of the long tape springs in high and low steady-state thermal conditions, an appropriate test setup is constructed. A view of the main structure and Components of the developed test apparatus is shown in Fig. 5.

Two specimens from carbon steel material with lengths 150 mm and 270 mm are chosen for the test. The mechanical and physical properties of the two investigated

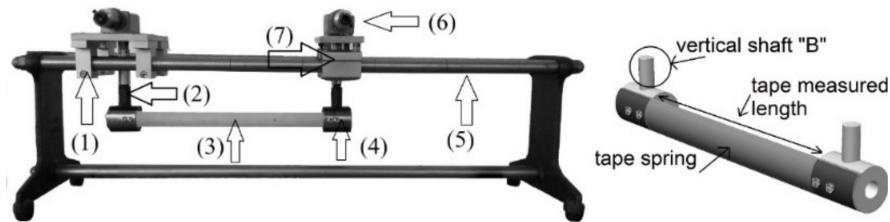


Fig. 5: Tape spring's bending apparatus: (1) shaft holder; (2) vertical shaft; (3) tape spring; (4) clamp; (5) guide-rail; (6) gear-box; (7) linear bearing.

tapes are introduced in Tables 1 and 2. Clamps zone lengths are equalled to 40 mm and measured length (150 mm and 270 mm) are for unrestraint lengths. The diameter of the clamps is 32.2 mm, which means the ends of the tape are attached to the hinges without any distortion. Cylinders are connected to the gearboxes through vertical shafts. Each gearbox is rotated manually through a scaled knob, and this scale measures the rotations. To specify reaction moment in tape's ends, two strain gauges are bonded to the vertical shaft's surfaces. Turning one of the knobs starts the deformation. Bending angle and the strain are recorded, and this process is repeated for another knob as long as the strain indicator shows the same values for the shafts.

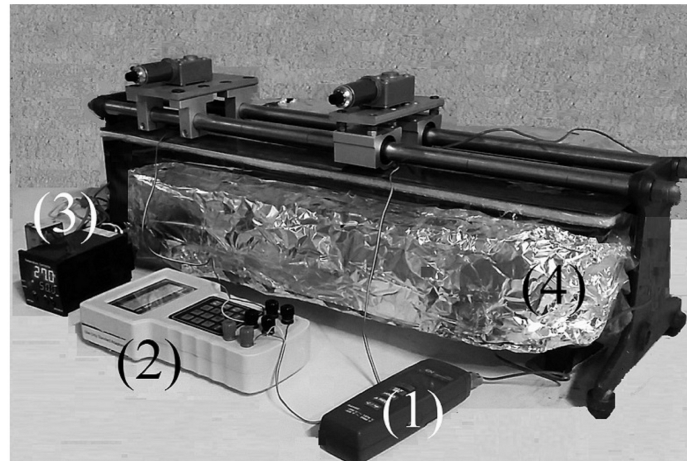


Fig. 6: Temperature controller box constructed to test under steady-state thermal conditions: (1) thermometer; (2) strain indicator; (3) temperature controller; (4) insulated cover.

3.1 THERMAL CONDITION TESTS DESCRIPTION

A test chamber with electric heating elements, which from now is called “thermal-box” is constructed to do tests under high-temperature conditions, a view of the thermal-box is shown in Fig. 6.

A heating element is fixed inside of the box to supply high temperatures, and a temperature controller is prepared to maintain tape spring’s temperature in a specific value. The folding process of the tape is visible during thermal tests through a glass door on top of the thermal-box. Strain gauges are located out of the box and their temperatures were controlled during the test. Many techniques explained to make and maintain low temperatures. Among these techniques, a nitrogen-organic solvent slush bath is most prevalent [21]. In the present work, to prepare a cooling bath, a mixture of liquid nitrogen, and methanol are used. The mixture is poured into the insulated box. Tape’s temperature is controlled to ensure the accuracy of the tests.

4 FINITE ELEMENT ANALYSIS

According to the experimental method explained in Section 3, two tape springs are modeled in Ansys software according to the properties in Tables 1 and 2. Tape springs are modeled with four-node shell elements namely the shell 181 in which each node has 6 DOF. A mesh with a 2 mm element size is used. Two additional nodes on each middle-end sections of the tape are created, as shown in Fig. 7. Folding of the hinges is a nonlinear problem for large angles; therefore, deformation is broken into steps to converge the simulation. For nonlinear control during the solution, a damping factor in stabilization option with the value of 110^{-6} is used; this value was used before by Soykasap [22] and was given consistent results. The boundary conditions of the tape are defined based on experiment tests, which was used earlier by Soykasap [22]. Static-structural analysis is conducted with considering steady-state thermal conditions and mechanical loads, which are applied concurrently.

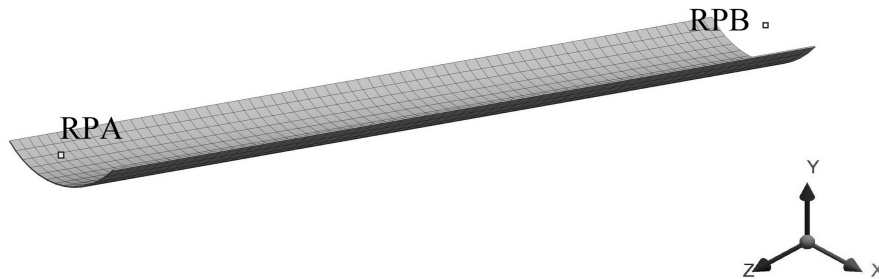


Fig. 7: Remote-points positions in the finite element model.

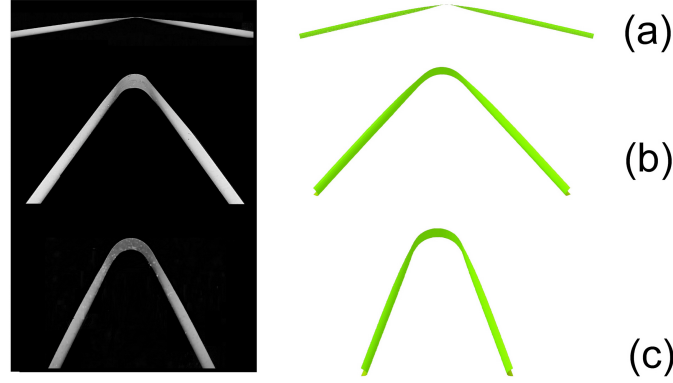


Fig. 8: Tape folding sequence: (a) the tape passed linear-elastic and its middle surface became flat; (b) and (c) in post-buckling, the bending angle became larger, but the moment reaction has not changed considerably.

The moment reaction is read from remote points. The deformation sequences from test and finite element are illustrated at left and right sides in Fig. 8 respectively.

A comparison with these figures shows the folding sequences in tests and simulations are alike. The nonlinear behavior of the tape spring in tests and the finite element analysis with length 270 mm under different temperatures is presented in Fig. 9. In the finite element analysis, it is observed that the variation in temperature partially impressed in the angle maximum reaction moment $\theta_+^{\max}$.

The finite element investigation also shows that the value of $M_+^{\max}$ is impressed due to temperature variations; hence, its quantity is diminished in the elevated temperatures, and it is higher in low temperatures. The results obtained during buckling

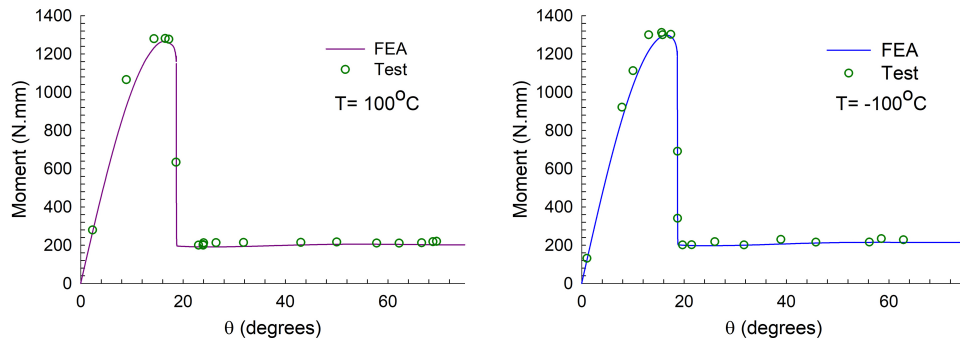


Fig. 9: Comparison of tests and finite element analysis results for $L = 270$ mm.

Table 4: Comparison of the finite element and experiments numerical results for the tape with lengths, $L = 150$ mm and 270 mm

T [°C]	$M_+^{\max}$ [N mm]				$\theta_+^{\max}$ [degree]			
	FEA		Test		FEA		Test	
	$L = 150$	$L = 270$	$L = 150$	$L = 270$	$L = 150$	$L = 270$	$L = 150$	$L = 270$
100	1398.4	1265	1425.9	1281	10.008	16.11	9.9	16.5
85	1406.1	1274.6	1427.1	1284.5	10.008	16.48	9.8	16.3
50	1420.2	1287.1	1429.1	1294.7	10.008	16.48	9.8	15.8
22	1432.1	1298.2	1444.7	1305	10.008	16.48	9.8	15.58
-50	1463.8	1326.4	1468.8	1336.2	10.005	16.48	9.5	16.3
-85	1479.3	1340.4	1485.6	1347.4	10.005	16.48	9.4	16.66
-100	1485.6	1346.1	1493.9	1350.1	10.005	16.5	9.4	16.23

of the two tapes (lengths: $L = 150$ mm and 270 mm) from finite element analysis, and experimental tests are given in Table 4. This table shows that the tests and finite element simulation results are in good agreement. According to the Table 3 and Table 4 the theory could not estimate the value of the $M_+^{\max}$ correctly with limited lengths of tape springs even for $\beta = 1$. The difference between theory and finite element simulation is due to neglecting boundaries effect in theory, which comes from supposing an infinite length.

The deviation of theory from finite element modeling for various length to width ratios is shown in Fig. 10.

The difference between $M_+^{\max}$ in theory and finite element method is up to 45% for small length to width ratios (L/a) < 5 and this variation becomes less than 10% for (L/a) > 10.

5 CONCLUSIONS

In this paper, load-deflection characteristics of tape springs under mechanical, and thermal loads were investigated, analytically, numerically, and experimentally. The existing theory was extended to consider the effects of steady-state thermal loads. An appropriate experimental test setup was developed to study the deflection mechanism and behavior of tape springs in different hot and cold situations. Finite element modeling was conducted to assure the correctness and accuracy of developed experimental and theoretical techniques and also doing more surveys. Investigations showed that the behavior of the tape spring changes with the temperature variation. In theoretical study, relations between loads and consequent deflections in Mans-

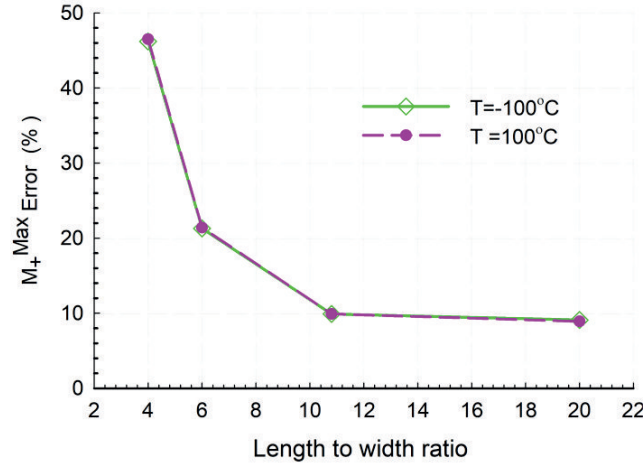


Fig. 10: $M_+^{\max}$ errors between the extended theory and finite element analysis for length to width ratio variations in two different temperatures ($T = -100$ and 100°C).

field's theory were extended using thermal expansion and thermal strain relations. Also, the physical properties of the tapes, such as Young's modulus, thermal expansion coefficient were defined as temperature-dependent terms. The results from this investigation showed a variation in $M_+^{\max}$ due to temperature change. Experimental results also approved the influence of the temperature variations effect on the tape spring's behavior. Experiments and finite element analysis are in good agreement with considering thermal-effect in all length of the investigated tapes. Also, considering length to width ratios greater than 10 for tapes, the extended theory complies with experiment and finite element methods.

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