Comparative analysis of 1DOF vs. 2DOF speed increasers for counter-rotating wind turbines

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Abstract. The paper presents a comparative study on the cinematic and static performance of a new type of planetary speed increaser for wind turbines. In this respect, the analytical modeling of the mechanical response for a counter-rotation wind system consisting of two identical counter-rotating wind turbines, an electric generator with fixed stator and a two-inputs one-output planetary speed increaser which can operate in two distinct situations: as a differential (2DOF) increaser (summing the input speeds) or as a monomobile (1DOF) increaser (summing the input moments) is obtained in the first part. For a numerical case study, the mechanical response of the wind system is simulated in the two stated functional situations, considering 1DOF vs. 2DOF speed increasers. The obtained results show a better energetic behavior of the 1DOF wind system as a result of a higher efficiency, but with a higher degree of load unevenness of the two wind turbines.

1. Introduction

One of the basic requirements in sustainable development is to increase the share of energy produced from renewable sources. Therefore, increasing the performance of wind energy conversion systems is a challenge for specialists, who have proposed in the literature a variety of solutions for increasing the electrical conversion efficiency. In this respect, new concepts of wind systems with counter-rotating rotors have been developed, which ensure the increase of the installed power by using a second rotor in parallel with the main one, which requires the integration of planetary speed increasers with the role of harmonizing the lower speed of wind turbines with generator higher speeds. These two-wind turbines are capable of generating more electricity compared to conventional systems with an input due to the sum of the input powers.

A single rotor wind system with a differential planetary unit is described in [1], where an input of the planetary speed increaser is connected to the wind rotor and the other input to a servomotor in order to maintain a constant speed at the input of the electric generator under the inherent fluctuation of wind speed. A differential planetary speed increaser for counter-rotating wind systems was cinematically and dynamically analyzed by Climescu et al. [2], based on a concept presented by Shin [3]. Also, the dynamic modeling of a counter-rotation wind system is presented in [4] on a case study of 1 kW turbine. In order to compare the energetic performances of counter-rotating wind systems with the classical solutions, the works [5, 6] present similar constructive variants of planetary differential speed increasers and their efficiency, obtained by modeling and simulation. Also, modeling of kinematics, efficiency and
power flow through complex planetary transmissions are addressed in various papers, using the theory of graphs for kinematic analysis [7,8] or efficiency [9], kinematic screw theory for mechanical power determination [10], where a calculation methodology for complex transmissions consisting of planetary units and variable transmissions is presented. Different methods for the analytical establishment of the efficiency of planetary units are systematized in [11]; one of these methods is developed in [12], for different operating situations of differential planetary transmissions. An approach to the problem of planetary transmissions with three inputs and outputs is presented in [13], where the kinematic and static analysis is performed for different combinations of external links. An in-depth discussion of this issue in terms of considering a two-inputs one-output planetary transmission as both speed reducer and increaser is presented in [14, 15]. A similar approach for a system with two counter-rotating inputs and one output and 1DOF speed increaser is presented in [16]. This type of transmission provides, in addition to input speed amplification, the sum of the two input moments and implicitly of the input powers.

The paper presents a new structure of a planetary speed increaser with three inputs and outputs, for implementation in wind turbines with counter-rotating rotors, able to operate in two main situations: as differential (summing inputs) or monomobile (summing of input moments) speed increaser. For the proposed speed increaser, the kinematic and static modeling is performed for each functional situation. Considering the two counter-rotating wind rotors with identical mechanical characteristics and the same mechanical characteristic of the electric generator, the wind system mechanical response obtained in the two functional situations is modeled for: a) a differential (2DOF) speed increaser, b) a monomobile (1DOF) speed increaser. Thus, a comparative study of the energy performance associated with the two functional situations is carried out based on a relevant numerical example, and implicitly, obtaining relevant information on the steady-state optimal operation of the compared wind systems. Finally, conclusions and recommendations for optimal design of planetary transmissions with two inputs and one output, including differential or monomobile speed amplifier, with possibilities to be implemented in renewable energy systems are drawn.

2. Problem formulation

The wind systems with two counter-rotating rotors are a relatively recent solution promoted to increase the installed capacity by summing the input powers generated by the two wind rotors, as a result of the integration of speed amplifiers capable of summing the input speeds or input moments, figure 1. The problem addressed in this paper is to determine how the energy performance of the counter-rotating wind systems is influenced by the choice of the type of speed increaser: which of the two types of increaser, differential or monomobile, ensures the wind power system better energy performance? For this purpose, two equivalent counter-rotating wind systems are considered, where a wind system has a 2DOF speed increaser while the second one integrates a 1DOF speed increaser, for which the mechanical response is identified under identical operating conditions. As a result, the planetary transmission depicted in figure 2 is considered and the analytical modeling of the mechanical response of the two counter-rotating wind systems is approached in the following equivalence assumptions:

a) two identical wind rotors, with the same mechanical characteristic, are used in both wind systems, \( R_1 = R_2 \) (figure 1);

b) the same electric generator is used in both wind turbines;

c) use planetary amplifiers with two inputs and one output, characterized by a partially common structure, which can operate as a differential mechanism (summing input speeds), figure 2a, and as a monomobile mechanism (summation of input moments), figure 2b;

d) for both wind systems, the steady-state operating point is determined on the output shaft of the speed increaser by considering a wind velocity range, modeled through the parameters of the wind rotor mechanical characteristics.
Figure 1. Conceptual scheme of two-inputs one-output wind system.

Figure 2. Kinematic scheme of the planetary speed increaser with two-inputs (H and 3) and one-output (1): a) differential (2DOF), b) monomobile (1DOF).

As shown in figure 2, this planetary speed increaser is based on a differential planetary unit \((PU\ I = H\cdot3\cdot2\cdot1)\), connected in parallel with a second planetary unit \(PU\ II\ (H\cdot5\cdot4\equiv2\cdot1)\). In the differential transmission, figure 2a, the gear 5 is free and does not influence \(PU\ I\) functionality; by locking the gear 5 of the \(PU\ II\), the speed increaser becomes a 1DOF planetary transmission, figure 2b.

3. Modeling the differential speed increaser \((M = 2)\)

The planetary speed increaser from figure 2a allows summing the input speeds \((\omega_{R1} = \omega_H\) and \(\omega_{R2} = \omega_3)\) generated by the two counter-rotating wind rotors \(R1\) and \(R2\) (Fig. 1). For this operation case of the speed increaser, the gears 4 and 5 rotate freely and therefore they don’t participate in the transmission of the mechanical power.

3.1. Kinematic and static modeling

According to [13], the transmitting functions of the differential planetary speed increaser \((PU\ I)\) are expressed by (see figure 2a):

\[
\omega_1 = a_H \omega_H + a_3 \omega_3 = \omega_{13} + \omega_{1H} \tag{1}
\]
\[ T_H = A_H T_1 \]  
\[ T_3 = A_3 T_1 \] 

Taking into consideration the interior kinematic ratio \( i_{01} \), the kinematic and static coefficients \((a_H, a_3, A_H, A_3)\) can be determined by setting, one by one, the input speeds with zero.

For \( \omega_3 = 0 \):

\[ a_H = \frac{\omega_i^{(3)}}{\omega_H} = \frac{\omega_{i3}}{\omega_{H3}} = i_{iH} = 1 - i_{01} \]  
\[ T_H \omega_H i_{H1}^{(3)} + T_3 \omega_{i3} = 0 \]  
\[ T_H i_{H1}^{(3)} + T_1 = 0 ; \; i_{H1}^{(3)} = \frac{i_{i1}^{(3)}}{i_{i1}^{(3)}} ; \; T_1 = - \frac{T_H}{1 - i_{01}} \]  
\[ A_H = - \frac{i_{i1}^{(3)}}{\eta_{H1}^{(3)}} = \frac{i_{01}}{1 - i_{01}} \]  
\[ i_{01} = i_{i1}^{(3)} \eta_{01}^{(3)} = i_{01} \eta_{01}^{(3)} \]  
\[ x = \text{sgn} \frac{\omega_H T_1}{-\omega_3 T_1} = \text{sgn} \frac{i_{01}}{\omega_{3H} - \omega_{H}} = \text{sgn} \frac{i_{01}}{1 - i_{01}} = -1 \] 

For \( \omega_H = 0 \):

\[ a_3 = \frac{\omega_i^{(3)}}{\omega_3} = \frac{\omega_{i3}}{\omega_{33}} = i_{i3} = i_{01} \]  
\[ T_3 \omega_3 i_{31}^{(3)} + T_i \omega_{i3} = 0 ; \; \frac{T_3}{T_i} = A_3 \]  
\[ T_3 i_{13}^{(3)} \eta_{31}^{(3)} + T_1 = 0 ; \; \eta_{13}^{(3)} = \eta_{01} ; \; T_1 = - \frac{T_3 \eta_{01}}{i_{01}} \]  
\[ A_3 = - \frac{i_{01}}{\eta_{01}} \] 

where \( i_{01} = i_{13}^{(3)} = \frac{z_3}{z_3} \) and \( \eta_{01} = \eta_{13}^{(3)} = \eta_{12} \eta_{23} \).

The efficiency \( \eta \) of the differential speed increaser is determined starting from the power equation of the transmission with friction, equation (14):

\[ (\omega_H T_H + \omega_3 T_3) \eta_H + T_i \omega_i = 0 \] 

\[ \eta = \frac{a_3 a_R (A_3 b_G - b_{R2}) + a_H a_R (A_H b_G - b_{R1})}{(a_G a_H b_{R1} - a_R b_G) \cdot A_2 + (a_G a_3 b_{R2} - a_R b_G) \cdot A_3 - (a_G a_H b_{R2} + a_G a_3 b_{R1}) A_3 A_H + a_R b_{R2} A_3 + a_R b_{R1} A_H} \]
where \(a_R, b_R, a_G, b_G\) are the coefficients of the mechanical characteristics of wind rotors (R1 and R2) and electric generator (G) respectively, equations (16)–(18).

3.2. Determination of steady-state operation point

The operating point of the wind turbine of type two counter-rotating rotors – 2DOF planetary speed increaser – electric generator is determined based on the transmitting functions of the speed increaser, equations (1) ... (3), and considering the mechanical characteristics of the wind rotors, equations (16) and (18), and of the electric generator, equation (19), respectively the relations corresponding to their connection to the considered speed increaser (see figure 1):

\[
T_{R1} = -a_R \omega_R + b_R, \quad \omega_R = \omega_H, \quad T_{R1} - T_H = 0
\]

(16)

\[
T_{R2} = -a_R \omega_R + b_R, \quad \omega_R = \omega_2, \quad T_{R2} - T_3 = 0
\]

(17)

\[
T_G = -a_G \omega_G + b_G, \quad \omega_G = \omega_1, \quad T_G - T_1 = 0
\]

(18)

By solving the system of equations (1) ...(3), combined with equations (16)...(18), the angular velocity \(\omega_{1F}\), the moment \(T_{1F}\), and the power \(P_{1F}\) respectively, reduced to the output shaft of the speed increaser, corresponding the system operating point in steady-state are:

\[
\omega_{1F} = \omega_1 = \frac{a_3 a_R (b_R - A_t b_G) + a_H a_R (b_R - A_t b_G)}{a_R a_R - a_3 A_t a_G a_R - a_H A_H a_G a_R}
\]

(19)

\[
T_{1F} = T_1 = \frac{a_3 a_2 a_R (b_R - A_t b_G) + a_H a_2 a_R (b_R - A_t b_G)}{a_R a_R - a_3 A_t a_G a_R - a_H A_H a_G a_R}
\]

(20)

\[
P_{1F} = T_{1F} \omega_{1F} = \frac{-a_R a_2 (a_3 A_t a_R + a_H A_H a_R) b_G^2 + (a_R a_2 (a_3 A_t b_R + a_H a_R) b_R) + a_G (a_3 A_t a_R) b_G + (a_3 a_2 a_R (a_3 A_t b_R + a_H a_R) b_G - a_G (a_3 a_R b_R + a_H a_R b_R)) b_G}{(a_R a_R - a_3 A_t a_G a_R - a_H A_H a_G a_R)^2}
\]

(21)

4. Modeling the monomobile speed increaser (M=1)

Fixing the gear 5 to the base (figure 2b) allows the differential speed increaser from figure 2a to turn into a monomobile (1DOF) transmission by entering into load the second planetary unit \(PU II (H-5-4-2-1)\). Such transmission allows both increasing the speed and summing the input moments \(T_{R1}\) and \(T_{R2}\) generated by the two counter-rotating wind rotors R1 and R2.

4.1. Kinematic and static modeling

The monomobile transmission from figure 2b is associated with the block diagram of figure 3, which is the basis of the following analytical modeling.
The kinematic and static relationships based on the scheme of figure 3, considering known the interior kinematic ratios \((i_{01} \text{ and } i_{02})\) and the interior efficiencies \((\eta_{01} \text{ and } \eta_{02})\) for the two component planetary units \(UP I \text{ and } UP II\), are further on derived. As a result, the kinematic and static parameters \((a_3^*, a_H^*, A_3^*, A_H^*)\) can be determined as follows:

\[
\begin{align*}
\omega_H &= a_H^* \omega_1 \\
\omega_3 &= a_3^* \omega_h \\
T_1 &= A_H^* T_H + A_3^* T_3 \\
a_H^* &= \frac{1}{1-i_{02}} \\
\omega_3 &= a_3^* \omega_h = \omega_h \left( \frac{1}{i_{01}} \right) + \omega_H \left( \frac{i_{01} - 1}{i_{01}} \right) = \omega_h \left( \frac{1}{i_{01}} + \frac{i_{01} - 1}{(1-i_{02})i_{01}} \right) \\
a_3^* &= \frac{1}{i_{01}} \frac{1-i_{01}}{(1-i_{02})i_{01}} = \frac{i_{01} - i_{02}}{i_{01}(1-i_{02})}
\end{align*}
\]

The moments on the shafts \(H\) and \(1\) are computed with equations (28) and (29):

\[
\begin{align*}
T_H &= T_H^* + T_H^- \\
T_1 &= T_1^* + T_1^-
\end{align*}
\]

The input and output moments corresponding to the first planetary unit \(UP I\), considered isolated, are derived from the balance energy equation of the associated fixed axes mechanism \((H = 0)\).

\[
T_1 i_{01} \eta_{01}^{i1} + T_3 = 0; \quad T_1 = -\frac{T_3}{i_{01} \eta_{01}^{i1}} = -\frac{T_3}{i_{01}}; \quad i_{01} = i_{01} \eta_{01}^{i1}
\]

\[
\chi_1 = \text{sgn} \left( \frac{\omega_H T_1}{-\omega_1 T_1} \right) = \text{sgn} \left( \frac{\omega_H - \omega_1}{\omega_1} \right) = \text{sgn}(\chi_{H1}) = \text{sgn} \left( \frac{1}{1-i_{02}} - 1 \right) = \text{sgn} \left( \frac{i_{02}}{1-i_{02}} \right) = -1
\]
T_r + T_H + T_z = 0; T_H' = T_{3} - T_1 = T_3 \frac{1}{l_{01}} - 1 = T_3 \frac{1 - \bar{i}_{01}}{l_{01}} \quad (32)

Using the previous relationships and the balance energy equation for the second planetary unit \( PU II \), the static parameters of the monomobile speed increaser result:

\[ T_H * \omega_H n^{(s)} + T_r * \omega_1 = 0; T_H' n^{(s)} + T_r = 0 \]  \quad (33)

\[ T_r = -T_H' \frac{1}{1 - l_{02}}; \bar{i}_{02} = i_{02} n_{02}^2; \quad x2 = x1 = -1 \]  \quad (34)

\[ T_H' = T_H - T_H' = T_H - T_3 \frac{1 - \bar{i}_{01}}{l_{01}} \]  \quad (35)

\[ T_r = \frac{T_{H'}}{1 - l_{02}} = T_3 \frac{1 - \bar{i}_{01}}{l_{01}(1 - l_{02})} \]  \quad (36)

\[ T_1 = T_1 + T_r = -T_3 \frac{1}{l_{01}} - \frac{1 - \bar{i}_{01}}{l_{01}(1 - l_{02})} \]  \quad (37)

\[ A_H^* = -\frac{1}{1 - l_{02}}; \quad A_r^* = \frac{i_{02} - \bar{i}_{01}}{l_{01}(1 - l_{02})}. \]  \quad (38)

where \( i_{01} = i_{01}^{(h)} = -\frac{z_3}{z_1}, i_{02} = i_{02}^{(h)} = \omega_H n_H = -\frac{z_5}{z_2}, i_{02}^{(h)} = \frac{z_3}{z_1} \) and \( n_{01} = n_{01}^{(h)} = n_{12}^{(h)} \), \( n_{02} = n_{02}^{(h)} = n_{12}^{(h)} \).

Starting from the balance energy equation of the considered planetary transmission, equation (14), and the static balance equation (24), the 1DOF transmission efficiency has the expression:

\[ \eta_0 = \frac{a_G (A_h^* b_{R2} + A_r^* b_{R1}) - b_G (a_G a_{R1} A_h^* + a_G a_{R2} A_r^*)}{(A_h^* b_{R1} - b_G) a_{R2} A_r^* + (A_h^* b_{R2} - b_G) a_{R1} A_r^* - (a_G a_{R1} A_h^* + a_G a_{R2} A_r^*) a_G + (a_G a_{R2} A_r^*) a_G - (a_G a_{R1} A_h^*) a_G} \]  \quad (39)

4.2. Determination of steady-state operation point

The steady-state operation point can be established by a similar approach as presented in chapter 3.2.

\[ \omega_{r,F} = \omega_r = - \frac{A_h^* b_{R2} + A_r^* b_{R1} - b_G}{a_G a_{R1} A_h^* + a_G a_{R2} A_r^* - a_G} \]  \quad (40)

\[ T_{r,F} = T_1 = \frac{a_G (A_h^* b_{R2} + A_r^* b_{R1}) - b_G (a_G a_{R1} A_h^* + a_G a_{R2} A_r^*)}{a_G a_{R1} A_h^* + a_G a_{R2} A_r^* - a_G} \]  \quad (41)

\[ P_{r,F} = T_{r,F} \omega_{r,F} = \frac{- (a_G a_{R1} A_h^* + a_G a_{R2} A_r^*) b_G^2 + (a_G a_{R1} A_h^* + a_G a_{R2} A_r^*)^2 + a_G (A_h^* b_{R2} + A_r^* b_{R1}) b_G - a_G (A_h^* b_{R2} + A_r^* b_{R1})^2}{(a_G a_{R1} A_h^* + a_G a_{R2} A_r^* - a_G)^2} \]  \quad (42)

In the case of the 1DOF speed amplifier, the interior transmission ratio \( i_{02} \) of \( PU II \) significantly influences the energy behavior of the system. The optimal value of the \( i_{02} \) ratio is determined by the
condition of maximizing output power $P_{1F}$, knowing that the parameters $a^*_1$, $a^*_H$, $A^*_3$, $A^*_H$ are functions of $i_{02}$:

$$P_{1F} = \max \Leftrightarrow \frac{\partial P_{1F}}{\partial i_{02}} = 0 = \frac{\partial P_{1F}}{\partial a^*_1} \frac{\partial a^*_1}{\partial i_{02}} + \frac{\partial P_{1F}}{\partial A^*_3} \frac{\partial A^*_3}{\partial i_{02}} + \frac{\partial P_{1F}}{\partial a^*_H} \frac{\partial a^*_H}{\partial i_{02}} + \frac{\partial P_{1F}}{\partial A^*_H} \frac{\partial A^*_H}{\partial i_{02}} = 0$$ (43)

By solving equation (43) the optimal value of the ratio $i_{02}$, which does not depend on the wind speed (modeled by the parameter value $b_R = b_{R1} = b_{R2}$), is shown in figure 4. In the numerical simulations of the 1DOF planetary speed increaser, the optimal value of the cinematic ratio $i_{02}(opt) = -5.629$.

![Figure 4. Determination of the optimum value of the interior kinematic ratio $i_{02}$ and non-functional wind turbine areas with 1DOF speed increaser.](image)

**5. Comparative analysis. Discussions**

The two wind systems considered in this study differ only in the functional situation of the planetary speed increaser (differential or monomobile), consisting of identical wind turbines and identical electric generators. Under the same external operating conditions (same wind speed), the operating points of the two wind systems may be different as a result of the different way of transmitting the power flow from the inputs to the outputs. A wind system is even more energy efficient as the output operating point is at a higher speed $\omega_{1F}$ (or higher moment $T_{1F}$), i.e. it provides more mechanical power to drive the electric generator and, implicitly, the electrical power is bigger. From a mechanical point of view, the wind speed influences the mechanical characteristic of the wind rotors, mainly by the parameter $b_{R1,2}$, the slope $a_{R1,2}$ of the mechanical characteristic remaining relatively constant on the operating zone [17].

Therefore, the variance of the parameter $b_{R1} = b_{R2} = b_R$, for both operating conditions of the planetary speed increaser ($M = 2$ and $M = 1$), between the minimum value ($b_{Rmin} = 580$ Nm) and maximum ($b_{Rmax} = 1000$ Nm) respectively, corresponding on the power curve of a wind system to the cut-in and cut-off speeds. Thus for values $b_R < 580$ Nm, one of the two wind rotors becomes a brake (power consumer), figure 4, and for $b_R > 1000$ Nm the generated electric power would exceed the production capacity of the system, which causes its operating off.

The values of the coefficients for the mechanical characteristics of the wind rotors and of the electric generator, considering the two extremes values of the parameter $b_R$, are centralized in table 1.
Table 1. The values of the mechanical characteristics coefficients.

| \(a_{R1}\) [Nms] | \(b_{R1}\) [Nm] | \(a_{R2}\) [Nms] | \(b_{R2}\) [Nm] | \(a_{G}\) [Nms] | \(b_{G}\) [Nm] |
|-----------------|----------------|-----------------|----------------|----------------|----------------|
| 35.22           | 580            | 35.22           | -580           | 1              | 100            |
| 1000            | -1000          |                 |                |                |                |

The operating range of the two wind rotors, considering the two limit values of the \(b_R\) parameter, is specified in figure 5, and the mechanical characteristic of the electric generator is represented in figure 6; the electric generator comes into the production mode at rotational speed \(\omega_G > 100\text{ rad/s}\).

**Figure 5.** The mechanical characteristics of the two wind rotors with their operating ranges.

**Figure 6.** Mechanical characteristic of the electric generator.
The parameters of mechanical characteristics of wind rotors (table 1) are required to determine the mechanical characteristic of the moto-increaser, reduced on the transmission output shaft. This characteristic is intersected with the mechanical characteristic of the electric generator to obtain the system steady-state operating point for the two extreme values of $b_R$ (figure 7).

![Figure 7](image_url)

**Figure 7.** Operating points of the two wind systems (with differential speed increaser $M = 2$ and monomobil speed increaser $M = 1$) for the two extreme values of the parameter $b_R$.

In order to highlight the performance of the two wind systems, figures 8 and 9 depict the N and Q details in figure 7.

![Figure 8](image_url)

**Figure 8.** The detail N of the operating points from Fig. 7, in case $b_R = 580$ Nm.
From the comparative analysis of the data represented in figures 7-9 the following aspects are highlighted:

- the wind system with monomobile planetary speed increaser has an efficiency ($\eta_a = 91.59\%$) higher than the bimobile wind system ($\eta_a = \text{max} 91.52\%$, for $|b_R| = 580 \text{ Nm}$);
- the efficiency of the monomobile wind system is not influenced by the operating mode (wind variation); instead, the efficiency of the bimobile wind system decreases slightly ($\eta_a = \text{min.} 91.49\%$) with the increase of the generated power (for $|b_R| = 1000 \text{ Nm}$);
- the mechanical output power generated at maximum wind speed by the monomobile wind system ($P_1 = -8809 \text{ W}$) is approx. 1.7% higher than the power ($P_1 = -8663 \text{ W}$) delivered by the bimobile wind system;
- during maximum load (at $b_{R_{\text{max}}}$), the input shafts of the bimobile speed increaser have a more harmonized load (TR1 = 240 Nm and TR2 = -185 Nm) compared to those of the monomobile wind system (TR1 = 169 Nm and TR2 = -272 Nm).

6. Conclusions

This paper presents a comparative study of the energy performance of two equivalent wind systems, which integrates two identical counter-rotating wind rotors, the same electric generator and a bimobile vs. monomobile planetary speed increaser respectively. The numerical simulation of the mechanical response in steady-state conditions, for different values of the wind speed (i.e. of the $b_R$ parameter of the mechanical characteristics of the wind rotors), allowed to highlight the following main conclusions:

- The monomobile wind systems are able to achieve higher output power than bimobile wind systems, under the same operating conditions;
- The bimobile wind systems provide a more harmonized loading of the transmission shafts and, therefore, more balanced mechanical stress on the system elements.

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