Design and Simulation of a New Flexible Constant Velocity Mechanism for Transmission of Power between Parallel Shafts

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Abstract: This paper presents a new mechanism for transmission of power between parallel shafts for adjusting height of farm or construction machineries in more gages. The mechanism consists of a drive shaft, 3 quad transmitter links, 8 connecting links and driven shaft. Advantage of this mechanism is that the velocity ratio between input and output shafts remains constant for all movements. Simulation results using software packages showed that this mechanism could transmit constant velocity ratio at all parallel movements between the shafts. Finally, tension analysis of the mechanism at 2000 rpm and input shaft torque of 8000 N.m. showed that the highest tension occurs in the shafts and the main components of mechanism have high load capacities in comparison with their dimensions. [Report and Opinion. 2010; 2(3):99-103]. (ISSN: 1553-9873).

Key words: Shaft, Power transmission, mechanisms, Constant velocity, Tension analysis, Simulation

1. Introduction

In many of agricultural machineries such as tractor certain mechanisms are necessary to transmit power between parallel shafts. Many researchers used the universal joints in series, gears or belts drive systems or available couplings to do this job. However, the drawback of the universal joint is its restricted intersecting angle and much occupied space [Erdman et al., 1991; Cantor, 2008]. The drawback of gears and belts drive is that they can not align the input and output shafts [Martin, 1969; Hojjati, 2000].

Some available couplings that are used between parallel shafts are Oldham, Wildhaber, Eccher and Schmitt couplings (Figure 1). They consist of slotted plates which slip on each other. However, they create much friction between the plates [Eccher, 1962; Haruo, 1982; Schmitt, 1982; Wildhaber, 1969]. Accordingly, the mentioned mechanisms are not suitable for farm or construction machineries, and therefore, most of heavy machineries are not equipped with a mechanism for leveling. Instead they use hydrostatic drives. But hydrostatic drives have high costs (Figure 2) [Shirkhorshidian, 2004; Makevet et al., 2002; Oberto et al., 2000; Reimpel et al., 2001].

The main goal of this study is proposing a new mechanism which has simple design, provide more flexibility and less cost.

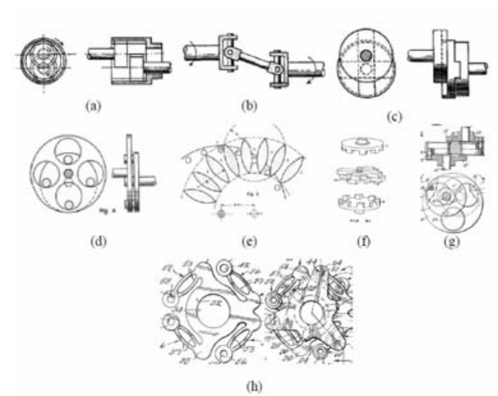


Figure 1. Different types of couplings currently in use. a, b) Gear and Universal couplings, respectively. c) Oldham coupling. d, e) Another couplings. f, g and h) Wildhaber, Eccher and Schmitt couplings, respectively.

Figure 2. Adjusting the height of a combine by a hydrostatic drive

2. Materials and methods

2.1. Design and theory

Figure 4 shows a mechanism with parallel links which transmits constant velocity. The constant velocity is maintained because of the parallel-link connections (Figure. 4) [Shirkhorshidian, 2004].

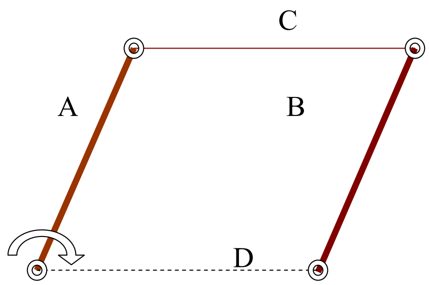


Figure 3. Parallel links mechanism

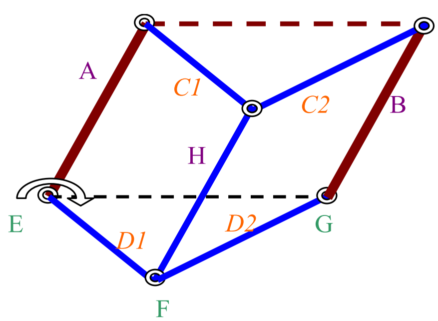


Figure 4. Preliminary design of proposed mechanism with adjustable links

If this mechanism to be designed so that the link C is always parallel to link D and its length could be also adjustable, then the mechanism can transmit constant velocity ratio in any position whenever the input and output shafts are parallel. Figure 4 shows the new mechanism with adjustable and parallel links C and D. As shown in Figure 4, the links C and D are divided into C_1, C_2 and D_1, D_2, respectively. They are connected to each other by H. Figure 5 shows a 3-D diagram of the mechanism. Since the center of gravity of each link in this mechanism is not on its own axis of rotating, this mechanism is not dynamically balanced. Thus, new mechanism with symmetrical links as shown in Figure 6 was finally designed (one transmitter links was added to each transmitter link in Figure 4).

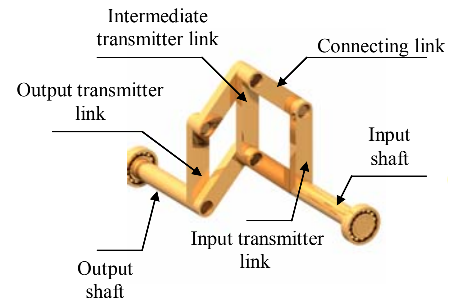


Figure 5. A 3D diagram of preliminary design of mechanism

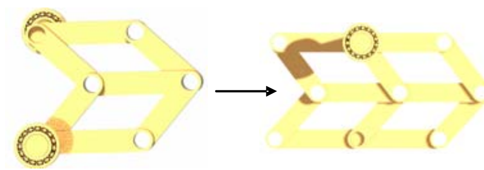


Figure 6. Side view of the mechanism before and after balancing

Based on links positions, two dead -centers occur when links be aligned (Figure 7). To solve this problem, two perpendicular transmitter links shown in Figure 8 were added to each transmitter link in Figure 7.

Figure 7. Dead point of balanced mechanism

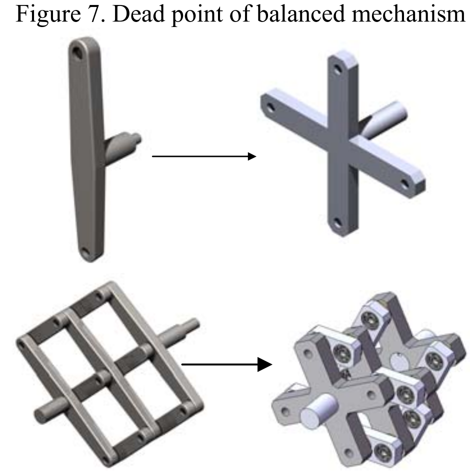


Figure 8. Improved design of the mechanism

2.2. Kinematics analysis

To show the mechanism has constant velocity ratio the components method was used [Oberto et al., 2000].

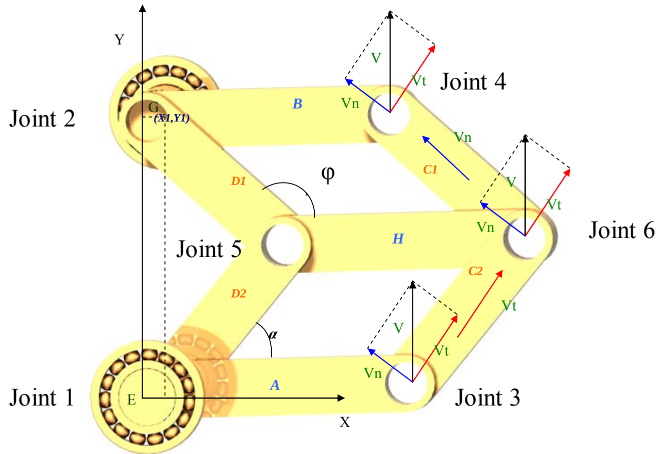


Figure 9. Simplified kinematics diagram of the mechanism

Figure 9 illustrates the kinematics diagram of the mechanism. The joint 2 (output shaft joint) is fixed and has coordinate (X_1, Y_1) with respect to origin in joint 1. Joint 3 has linear velocity, V, from input shaft (input shaft and input transmitter link are connected together).

Using components method, velocity V of joint 1 is divided into velocity components (V_t and V_n) as follows:

$$V_t = V\sin\alpha \qquad V_n = V\cos\alpha \qquad (1)$$

Since joint 2 (point G) is at fixed point (X_1, Y_1), then D_1, D_2 are also fixed. Thus velocity V_t is translated to link H since the link H has only a rotating movement around joint 5, then V_h can be obtained as:

$$V_h = \frac{V_t}{\sin\alpha} = \frac{V\sin\alpha}{\sin\alpha} = V \qquad (2)$$

As shown in Eq. 2, V_h is equal to V. As described above, velocity V_h in joint 6 is divided to two components; velocity V_n and V_t, and because the joint 2 is fixed, then velocity in joint 4 (V_f) will be perpendicular to link B and is obtained as follows:

$$\begin{cases} V_t = V\sin\alpha \\ V_f = \dfrac{V_t}{\sin\alpha} = \dfrac{V\sin\alpha}{\sin\alpha} = V \end{cases} \qquad (3)$$

Thus, the linear velocity of the input and output shaft is equal and does not depend on the two angles φ and α (which depend on coordinates X_1 and Y_1). Accordingly, the output velocity is constant in any coordinates X and Y.

Where: *A*= Input transmitter link
B= Output transmitter link
C= Connecting link
C_1, C_2= Dividing the link C to two links
D= Ground link
D_1, D_2= Dividing the link D to two links
E= Rotating center of input shaft
F= Rotating center of connecting link
G= Rotating center of output shaft
H= Intermediate transmitter link
Joint 1, 2, 3, 4, 5, 6= Joint of links
V(m/s)= Linear velocity of joint 3
V_h *(m/s)*= Linear velocity of joint 6
V_t , V_n *(m/s)*= V components in longitudinal and transverse directions of links C_1 and C_2
α *(rad)*= Angle between D_2 and E
β*(rad)*= $\alpha + \gamma$
γ *(rad)*= Angle between C and A links
φ = Angle between D and H_1

2.3. Eccentricity of shafts

As shown in Figure. 10, the output shaft can rotate around the input shaft. Therefore, it has boundary limitation with a radius of 2c. Thus the eccentricity of offset of the shafts is twice the length of the links.

Where: *c(m)* = Length of connecting link

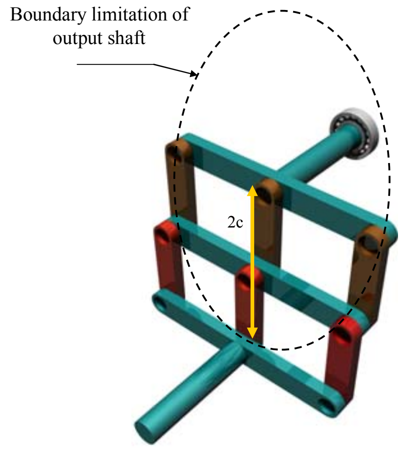
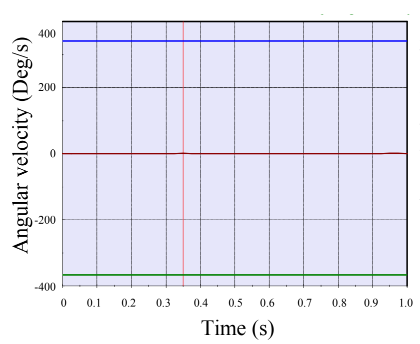


Figure 10. The boundary space of output shaft as revolving around input shaft

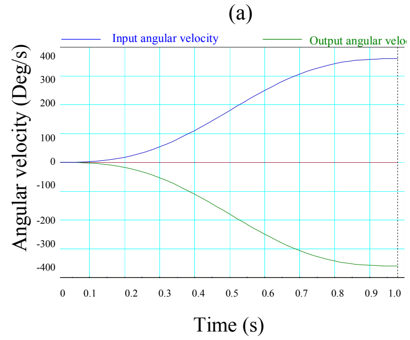
3. Results and discussions

3.1. Simulation results

The greatest advantage of the proposed mechanism is its constant velocity ratio during rotation. Simulations were carried out using powerful software packages such as COSMOS Motion, Visual Nastran, and Autodesk Inventor also confirmed this claim and the results are shown in Figure 11. The simulations shown in Figure 11 are (a) with a constant velocity of 360 Deg/s and in Figure 11 (b) and (c) at variable velocities of *f(x)* and *g(x)*. The output velocity is negative because of the positions of the input and output axes in the Cartesian coordinates system which are reversed in simulation.



(a)



(b)

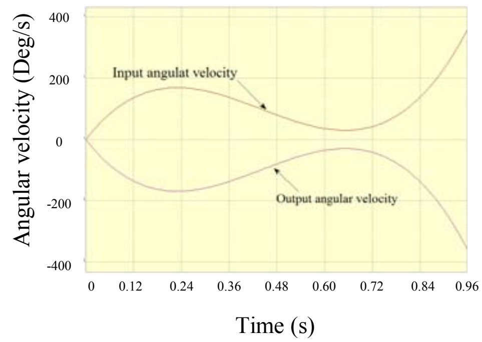
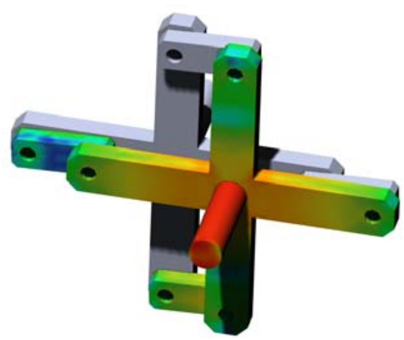


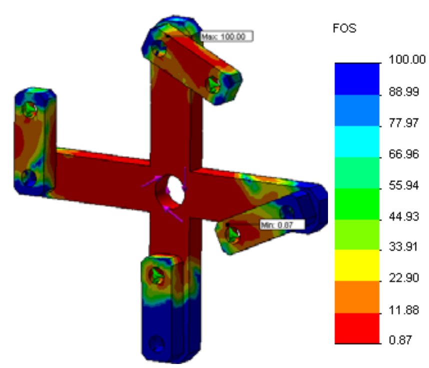
Figure 11. Comparisons between input- and output velocities of the mechanism:
a) Simulation by COSMOS Motion software.
b) Simulation by Visual Nastran software.
c) Simulation by Autodesk Inventor software.
[Note: Blue line: Input velocity, Green line: Output velocity]

3.2. Tension Analysis

The mechanism was tension analyzed using COSMOS Works and Visual Nastran software. The simulation results are shown in Figure12.



(a)



(b)

Figure 12. Safety factor Simulation in whole mechanism: a): By Visual Nastran. b) By COSMOS Works.

The analysis of tension in the mechanism's components was carried out at the velocity of

2000 rpm and the shaft torque of 8000 N/m. This mechanism is made of alloy steel and has a shaft diameter of 30 mm, components thickness each of 10 mm. The simulation in COSMOS Works was carried statically (without velocity) and in Visual Nastran was carried dynamically. The safety factor of 0.87 was obtained and it revealed that the maximum tension occurs in the shafts (Figure 12 b).

Mechanical properties	Values
Elastic modulus	2.1E11(Nm^{-2})
Poisson's ratio	0.28
Shear modulus	7.89E10 (Nm^{-2})
Tensile strength	723825618 (Nm^{-2})
Yield strength	620421997.8 (Nm^{-2})

Table 1. Mechanical properties of adjusting height mechanism material

4. Conclusion

The proposed mechanism has a constant velocity ratio and transmits power between the two parallel shafts at any movement. Changes in radial displacement do not affect the constant-velocity relationship between the input and output shafts further more they do not affect the initial radial reaction forces that might cause imbalances in the system. This mechanism has much flexibility than the existing ones and can be used for height adjustment in heavy machineries. Using commercial software packages, the maximum tension was found to occur in the shaft. The mechanism transmits the power without vibration. By altering the dimensions of its components, this mechanism can transmit much more power.

This mechanism can be used on automobiles for adjusting height with more gages marine, machine-tool and rolling-mill machinery or wherever power must be translated to offset shafts (Figure12).

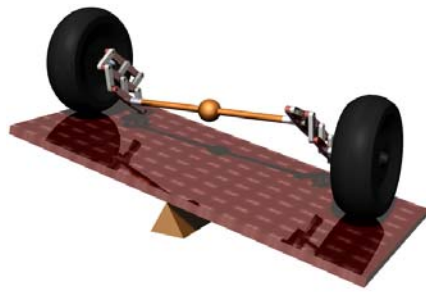


Figure 12. An application of proposed mechanism on a tractor axle for adjusting height on slope land

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