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PRASAD VINAYAK MARULKAR
Rajarambapu Institute of Technology, Sakharale, Islampur, India, prasad.marulkar@gmail.com

S.G. JOSHI
Rajarambapu Institute of Technology, Sakharale, Islampur, India, S.G.JOSHI@gmail.com

PARALLEL DISK CONTINUOUSLY VARIABLE TRANSMISSION (PDCVT):PROTOTYPE DEVELOPMENT AND PERFORMANCE TESTING

PRASAD VINAYAK MARULKAR[1] & S.G.JOSHI[2]

[1,2]Department of Mechanical Engineering, Rajarambapu Institute of Technology, Sakharale, Islampur, India.415 414.
E-mail: prasad.marulkar@gmail.com

Abstract- In this paper, a typical Parallel Disk Continuously Variable Transmission System (PDCVT) is developed in the spirit and approach of Kazerounian and Furu-Szekely. In situations where a speed ratio is required to be changed frequently, continuously variable transmission is one of the desirable solutions. The PDCVT system is one such solution which offers the advantages such as high power to weight ratio and reliability in operation. First of all, the development and manufacturing details of the developed PDCVT system are discussed and experimental and theoretical values of transmission ratios have been determined with different ball diameters and materials. Also, the theoretical and experimental evaluation of transmission torque has been carried out. Experiments have been carried out to determine the system parameters: the stiffness of the spring for preloading mechanism of the system and the area of contact between the balls and rotating disks to determine the coefficient of friction.

I. INTRODUCTION

A Continuously Variable Transmission or step less transmission is suitable for the situations where frequent speed ratio changes are necessary to obtain more output power and to decrease fuel consumption. The PDCVT system is one such solution which offers the advantages such as high power to weight ratio, reliability in operation and simple mechanism to control the speed ratios and the transmitted torque.

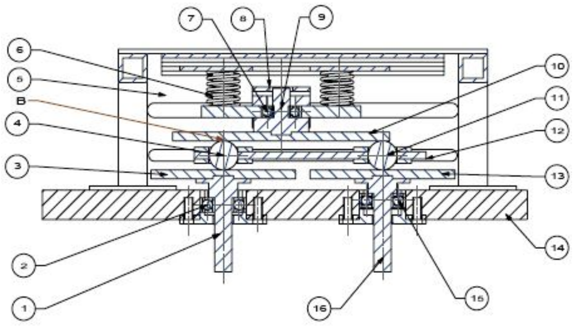
Kazerounian and Furu-Szekely [1] have presented the design and analysis of a newly developed parallel disk continuously variable transmission, focusing on its structure, operating principle and energy efficiency. Ang et al. [2] have discussed a novel application of an associative memory called the Modified Cerebellar Articulation Controller (MCMAC) in Continuously Variable Transmission (CVT). Akehurst et al. [3] have presented performance investigations on a 90 mm diameter protype variator which was sized for a maximum rated input power of 12kw. Carter et al. [4] have worked on use of a continuously variable transmission to optimize performance and efficiency of a two-wheeled Light Electric Vehicle (LEV). Chein et al. [5] have presented the simulation procedure for an automatic transmission for bicycles. Menster et al. [6] have discussed identification of a Toroidal Continuous Variable Transmission (TCVT) using continuous time system identification methods. Carbone et al. [7] have presented dynamics of CVTs- a comparison between theory and experiment. Srivastav and Haque [8] have reviewed the dynamics and control of belt and chain Continuously Variable Transmissions.

As such in this paper, a typical Parallel Disk Continuously Variable Transmission System (PDCVT) is developed in the spirit and approach of Kazerounian and Furu-Szekely [1]. First of all, the development and manufacturing details of the developed PDCVT system are discussed and experimental and theoretical values of transmission ratio have been determined with different ball diameters and materials. Also, the theoretical and experimental evaluation of transmission torque has been carried out. Experiments have been carried out to determine the system parameters: the stiffness of the spring for preloading mechanism of the system and the area of contact between the balls and rotating disks to determine the coefficient of friction.

II. DEVELOPED PDCVT MECHANISM

Figure 1 shows a schematic of the developed PDCVT mechanism. Figure 2 shows the developed prototype of PDCVT. Input disk 3 with input shaft 1 and output disk 13 with output shaft 16 are rigidly supported in a base structure 14. Free rotation of input shaft 1 and output shaft 16 is possible with the help of sealed ball bearings 2 and 15 as shown in fig.3.The balls 4 and 11 are free to rotate in cage plate 12 and sandwiched between input disk 3, output disk 13 and intermediary disk 10. Intermediary disk 10 with intermediary shaft 9 is supported in support plate 8. Free rotation of intermediary shaft 9 in spring location plate 17 and support plate 8 assembly is possible with the help of sealed ball bearing 7. Figure 4 shows the cage plate 12 and support plate 8 with supporting bolts 19a and 19b, and 20a and 20b are free to move respectively in over slotted plates 14a and 14b during the transmission ratio change mechanism. Four helical compression springs 6 of equal dimensions and stiffnesses are preloaded by tightening nut and bolt arrangement located in sliding plate 21 and top plate 22. Figure 5 shows a D.C. motor 23, 0.37 kw and 1440 rpm. It drives the input shaft 1 of the system. It is mounted on a base structure 14 as shown in fig. 6. The distance between input shaft 1 and output shaft 16, size of ball 4 and 11 and normal force created by springs 6 are important system parameters. Due to tractive force, in first stage of transmission, input disk 1 rotates in clockwise direction which is powered by D.C. motor 23. Ball 4 rotates anticlockwise which forces intermediary disk 10 to rotate in clockwise direction. In the second stage of transmission, ball 11 rotates in anticlockwise direction forcing output disk 13 to rotate in clockwise direction.

It can be seen that the transmission ratio between input disk 3 and intermediary disk 10 depends upon the position of ball 4 with respect to centers of input disk 3 and intermediary disk 10. As shown in fig.1, contact point B is closer to center of intermediary disk 10 then intermediary disk 10 moves faster and intermediary disk 10 moves slower when contact point B is away from its center. The ratio of angular velocities between intermediary disk 10 and output disk 13 in changed similarly.



1-Input shaft, 2-Input bearing, 3-Input disk, 4 and 11-Balls, 5-Over-slotted plate, 6-Springs, 7-Intermediary bearing, 8-Support plate, 9-Intermediary shaft, 10-Intermediary disk, 12-Cage, 13-Output disk, 14-Base structure, 15-Output bearing, 16-Output shaft, B-Contact Point between Ball 4 and Intermediary disk 10.

Fig. 1 Schematic of developed PDCVT.

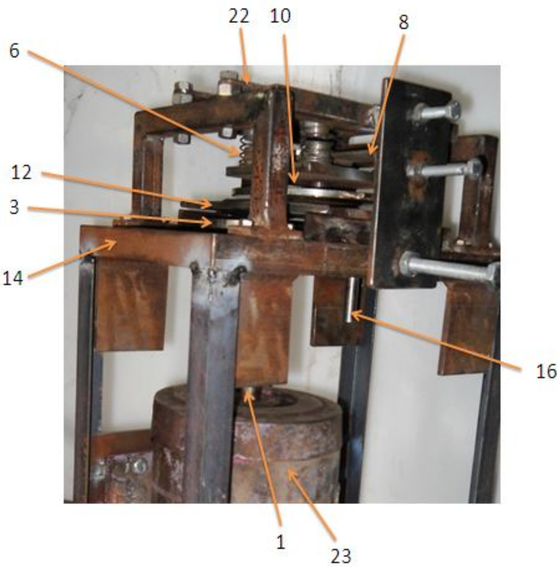


Fig. 2 PDCVT Assembly

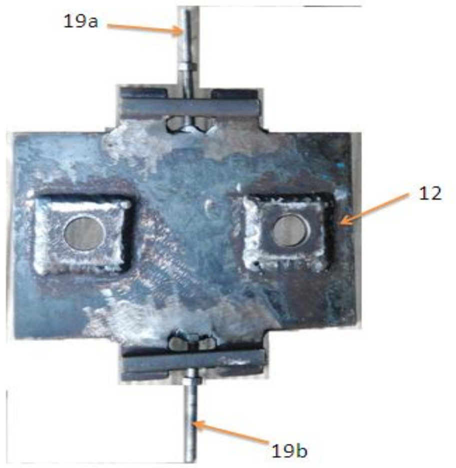


Fig.3 Cage Plate

Fig. 4 Top and Sliding Plate Assembly.

Fig. 5 Intermediary Disk Assembly.

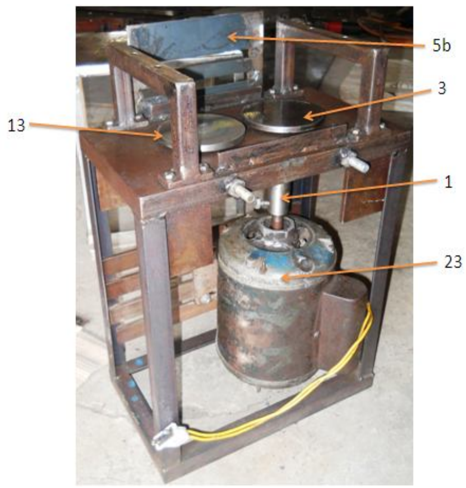


Fig. 6 Base Structure

III. THEORETICAL AND EXPERIMENTAL ANALYSIS

A. Specifications of Developed PDCVT:

Diameters of Input and Output disk: 100mm, Diameters of Input/output and intermediary shaft: 12mm, Diameter of Intermediary disk: 150mm, Diameters of steel balls: 20mm, 25mm and 30mm, Diameter of Allumina ceramic ball: 20mm, D.C. motor 0.3kw 1440rpm.

TABLE I. POSITION OF LEVER, OFFSET DISTANCE AND R1, R2, RI, RO.

Posi-tion	Offset distance (d) (mm)	r1 (mm)	r2 (mm)	ri (mm)	ro (mm)
I.	6	49	61	15	15
II.	-3	58	52	16	16
III.	-12	67	43	38	38

B. Determination of Transmission Ratio Theoretical Values

The steel balls are rotated between steel disks and by changing the position of balls and intermediary disk different speed ratios are obtained. Transmission ratio TR is varied by changing the offset distance d. TR is given as [1]

$$TR = \left(\left(r2 - \frac{d}{2}\right) \times \left(ri + \frac{d}{2}\right)\right) / \left(\left(ro - \frac{d}{2}\right) \times \left(r1 + \frac{d}{2}\right)\right) \quad (1)$$

Where, r1 and r2 are the distances of ball 7 and 8 respectively from the center of intermediary disk 10, ri is the distance between ball 7 and center of input disk and ro is the distance between ball 8 and center of output disk, d is offset distance and taken as positive if it is measured towards right.

For the case of position I with offset distance d=6mm of Table I and with the values of r1, r2, ri and ro obtained for position I and substituting the values in equation (1), TR is obtained as, 1.24. Similarly, the values of TR for position II (d=-3mm) and position III (d=-12mm) are obtained respectively as 0.9 and 0.62, the theoretical values of transmission ratio are given in Table II.

Experimental values

Figure 7 shows the PDCVT which is powered by D.C. motor of 0.3 kw and 1440 rpm. The transmission ratios were measured for three different offset positions of Table I with the steel ball diameter 20mm and corresponding positions of intermediary disk. The speeds of input and output shafts were measured by using non-contact type tachometer. The corresponding experimental values of TR are given in Table II, column 4. Using a similar procedure the values of TR with alumina ceramic ball diameter 20mm, and steel ball diameters 25mm and 30mm, with the offset values given in Table I, were measured and are given in column 5,6 and 7 of the Table II, respectively. The results of analysis are given in figs. 8 to 11. The differences between theoretical and experimental values are observed because of some slippage between balls and disks.

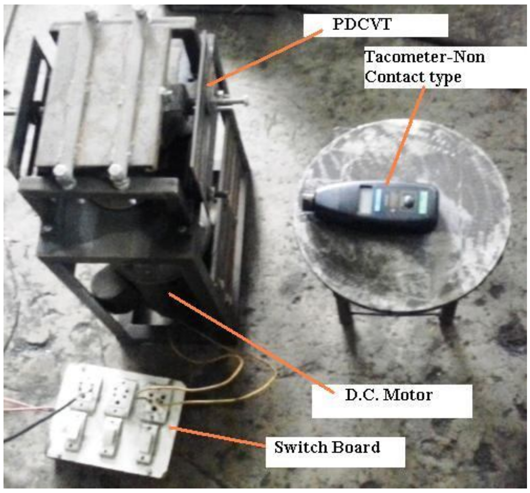
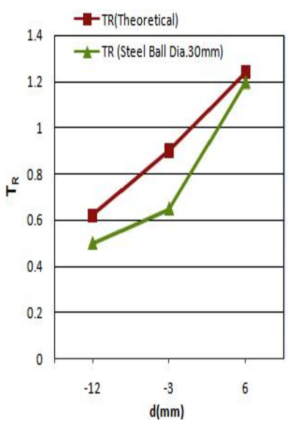


Fig.7 Experimental Setup for Measurement of Transmission Ratio

TABLE II. TRANSMISSION RATIOS FOR THREE DIFFERENT POSITIONS.

Position	Offset Distance, d (mm)	Material	Steel	Al-C er.	Steel	Steel
		Theoretical	Experimental			
		TR	TR dB=20	TR dB=20	TR dB=25	TR dB=30
I.	6	1.24	1.02	1.02	1	1.2
II.	-3	0.9	0.8	0.76	0.68	0.65
III.	-12	0.62	0.64	0.63	0.48	0.5



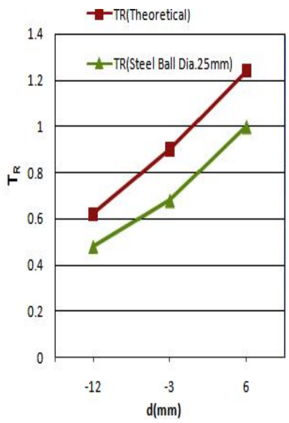


Fig.8 TR vs d (dB=30mm) **Fig.9 TR vs d (dB=25mm)**

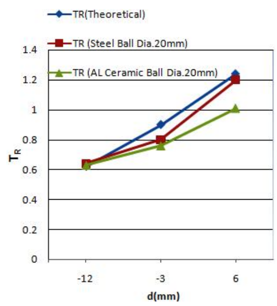


Fig.10 TR vs d (dB=20mm)

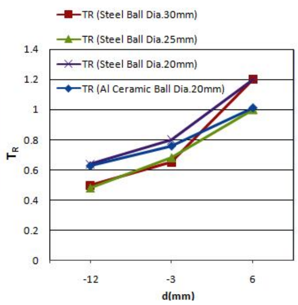


Fig.11 TR vs d (Experimental)

C. Determination of torque transmitted

The transmitted torque is given as [1].

$$T = Nf \times \mu s \times (ro - d) \qquad (2)$$

Where, Nf is normal force between ball and intermediary disk.

To determine the normal force 'Nf' between ball and the intermediary disk it is necessary to determine the contact radius 'a' between ball and intermediary disk. For this purpose, the experimental procedure as shown in fig. 13 is adopted. Figure 13 shows the loading arrangement and the ball is fixed in holder plate and placed on the steel disk of which the upper face is polished by blue ink. The pan and weight arrangement of fig. 13 applies the force on each ball.

System parameters

Different experiments have been carried out to find out system parameters like stiffness of the spring, area of contact and coefficient of friction between the balls and steel disk. The stiffness of the helical compression spring in preloading mechanism has been determined using the usual procedure by plotting load vs deflection graph the slope of this graph is 65N/mm which is the stiffness of the spring as shown in fig.12.

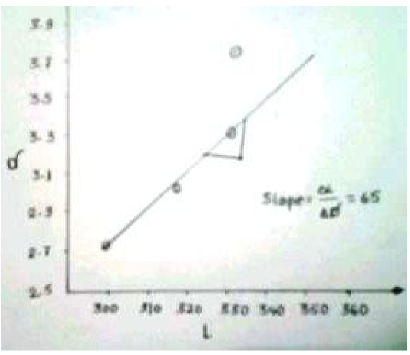


Fig. 12 Load (L) vs Deflection (δ)

The total load acting at the center of the balls is the sum of the loads created by the deflection of four springs, weight of intermediary disk assembly and weight of four springs. Total load is equally distributed on each ball and deforms them elastically and creates its area of contact on the steel disks. Fig.13 shows the experimental set up to determine the area of contact of the ball on the steel disk. The total load with pan and weight arrangement is acting at the center of the ball fixed in holder plate and placed on the steel disk of which upper face is polished by blue ink. The impression of ink on the ball is taken on a graph paper which gives the area of contact of the ball under the total load 25N. Table III shows the areas of contact for alumina ceramic balls with diameter 20mm, steel balls with diameters of 20mm, 25mm and 30mm as 0.6mm^2, 0.5mm^2, 0.75mm^2 and 1mm^2 respectively.

TABLE III. AREA OF CONTACT FOR VARIOUS BALL DIAMETERS

Ball diameter (mm)	Material	Area of contact (mm2)
20	Steel	0.5
20	Al-Cer.	0.6
25	Steel	0.75
30	Steel	1

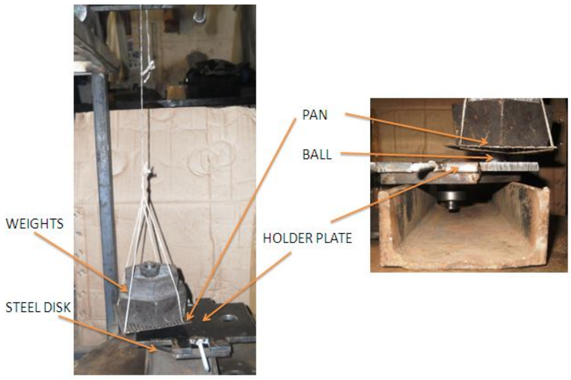


Fig. 13 Arrangement for measurement of contact area

To measure the coefficient of friction the ball was blocked in cage plate and an arm of known length is attached to the input shaft. Pan is attached to another end of arm. Small weights are gradually added pan until the disks just starts to slip. The coefficient of friction μs is given as [1].

$$\mu s = \frac{F \times l}{Nf \times r} \qquad (3)$$

Where, F is load hanged to arm, l is length of arm, r is radius of ball and Nf is normal force acting by the ball on the disk.

Maximum permitted contact pressure P max. is given as [1].

$$Pmax. = 1.67 \times Y \qquad (4)$$

$= 1.67 \times 353.1$

$= 589.67\ N/mm^2$

Where, Y=353.1N/mm^2 is the yield strength of weaker material i.e. the disk material. Average contact pressure P avg. is given as [1].

$$P_{avg.} = \frac{2}{3} \times P_{max.} \qquad (5)$$

= 2/3 x 589.67

= 393.1 N/mm^2

Also,

$$P_{avg.} = \frac{Nf}{\pi \times a^2} \qquad (6)$$

= 393.1 N/mm^2

Where ‘a’ is the radius of contact area. Using the values of radii of contact area for steel 20mm, alumina ceramic 20, steel 25mm and 30mm ball diameters as 0.5mm^2, 0.6mm^{2,} 0.75mm^2 and 1mm^2 respectively the normal forces ’Nf’ are calculated as 197N,236N,295N and 393N respectively.

By using equation (3), the values for coefficient of friction for balls with diameters and material 20mm-steel, 20mm- alumina ceramic, 25mm-steel and 30mm-steel are calculated as 0.26, 0.21, 0.06 and 0.15, when the values of the Nf are taken respectively as 197N, 236N, 295N and 393N.

Torque measurement

Figure 14 shows the setup for torque measurement of PDCVT. The ball with diameter 20mm is set to position I.

The input shaft is connected to the D C motor with coupling. An arm of length 0.25m is fixed to output shaft with bolts as shown in fig.14. An empty pan with a string is connected to the end of the arm making sure that the arm is perpendicular to output shaft. Weights are gradually added in pan until the output shaft just starts to turns without the rotation of input shaft.

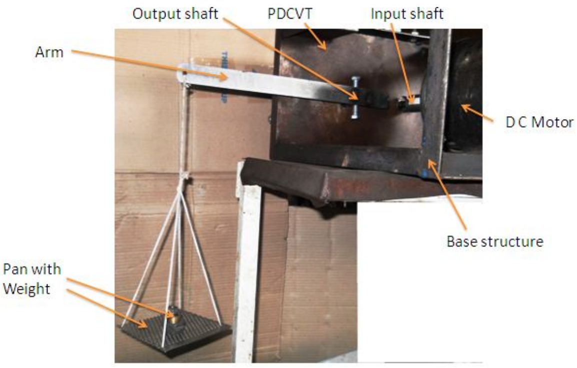


Fig. 14 Arrangement for torque Measurement

The torque,

T=Weights attached (N) X Length of arm (m) (7)

= 1.93X0.25

= 0.47 N-m.

The same procedure was repeated for the ball with 20mm diameter for position II and position III (Table I) and the values of torque ‘T’ for these positions are obtained respectively as 0.595N-m and 0.547N-m.Table IV shows these values in column 4 along with corresponding theoretical values in column 3. Theoretical and experimental values of torque for systems with alumina ceramic ball diameter of 20mm and steel ball diameters of 25mm and 30mm have been determined by using equation (2) and equation (7) and are given in columns 5,6,7,8,9,10 of Table IV, along with figs.15, 16, 17,18 and 19.

TABLE IV. THEORETICAL AND EXPERIMENTAL VALUES OF TORQUE

Positi-ons	d (mm)	dB=Ball diameter in mm.							
		20-Steel		20-Alumina-cermic		25-Steel		30-Steel	
		T Th.	T Exp.	T Th.	T Exp.	T Th.	T Exp.	T Th.	T Exp.
I.	6	0.5	0.47	0.51	0.49	0.25	0.2	0.85	0.73
II.	-3	1	0.5	0.95	0.51	0.4	0.22	1.09	0.86
III.	-12	2.5	0.55	2.47	0.54	1	0.29	2.8	1.03

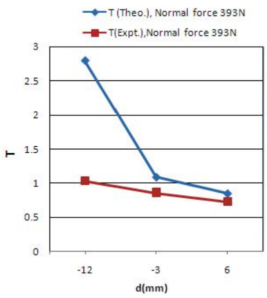


Fig.15 T vs d (dB=30mm) Steel

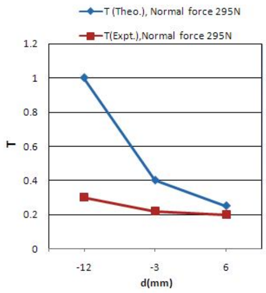


Fig.16 T vs d (dB=25mm) Steel

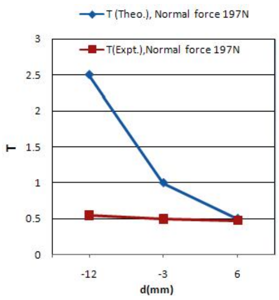


Fig.17 T vs d (dB=20mm) Steel

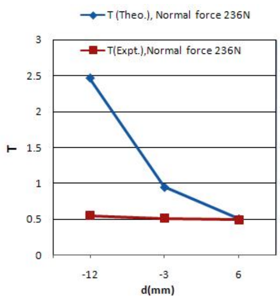


Fig.18 T vs d (dB=20mm) Alumina Ceramic

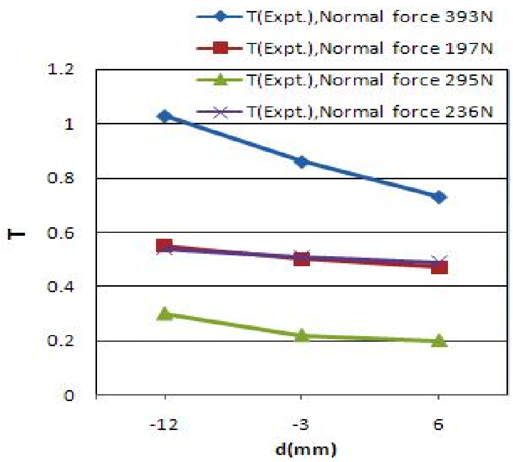


Fig.19 T vs d (Experimental)

IV. CONCLUSIONS

In this paper, a typical Parallel Disk Continuously Variable Transmission System (PDCVT) is developed in the spirit and approach of Kazerounian and Furu-Szekely. Different experiments have been carried out to find out system parameters like stiffness of the spring, area of contact and coefficient of friction between the balls and steel disk.The theoretical and experimental values of transmission ratios for different offset values of intermediary disk show a fairly good agreement except for the case of ball diameter of 30mm.

The theoretical and experimental values of the torque transmitted, however, do not agree very well for the offset distance d= -12mm. The developed PDCVT will be useful in applications where a speed ratio is required to be changed frequently.

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