Experimental Determination Of Losses In Planetary Gears By Means Of Static Loading

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Abstract: *The paper reviews experiments performed with a coupled two-carrier planetary gear train with four external shafts and two brakes. The brake works as a two-speed gearbox. In one case of operation the power flows only through one of the coupling gear trains while in the other it flows through both coupling gears with internal power circulation. The losses in the gear train are determined by means of static loading. A check is made on the validity of the relations deduced for determination of the gear train's efficiency as a function of the efficiencies of its coupling gear trains.*

Key words: *planetary gear, efficiency, static loading, losses*

1. Introduction

The problem of losses and the related heating is very important in the field of planetary gear trains [1], [5], [6], [8].

Static loading presents one way to determine the losses in gear trains [1]. This method could be used in quality control of batch-produced gear trains [7].

Experimental determination of the static efficiency has been performed in Indiana University-Purdue University Indianapolis [3] with a gear train with greased teeth (to avoid churn losses).

The static efficiency can be used for comparative analysis of the energy effectiveness of the various speeds in multi-speed planetary gear trains (planetary gearboxes).

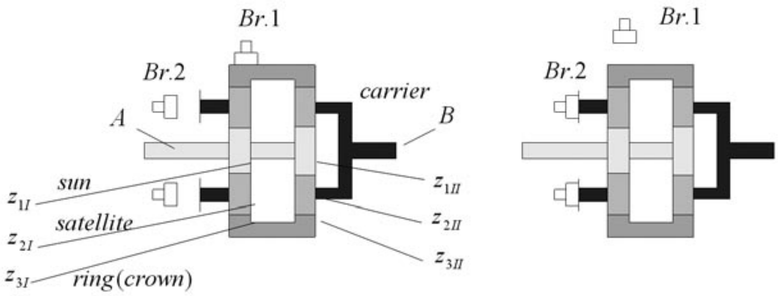
In some cases of machinery need arises for the use of two-speed transmissions with a definite ratio between the two speeds. The ability to switch over while in motion and loaded is a certain advantage and in some cases an inevitable necessity. One proper solution is the use of a coupled two-carrier planetary gear with four external shafts and two brakes. In [4] there is a review on the various possible ways of brake mounting and power flowing in the above mentioned gear when it consists of two simple gears of the most common type (consisting of a sun-gear, a ring-gear and one-rimmed satellites, situated on the carrier).

The aim of this paper is to present the results of a static experimental study of the two-carrier two-speed planetary gearbox and to verify the theoretical relations used in gear efficiency determination [4].

2. Experimental bench

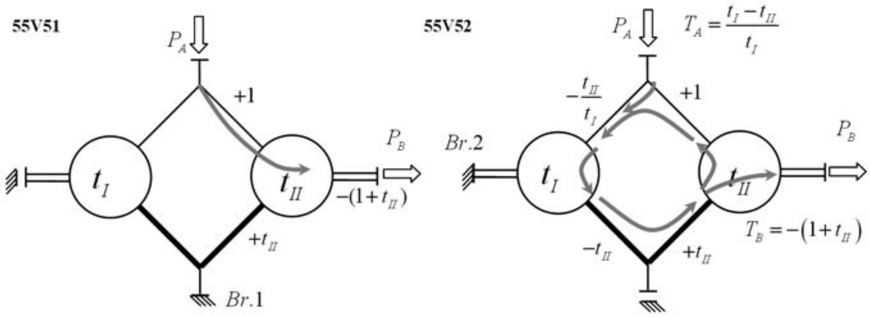
Experiments are performed to determine the static efficiency a two-carrier, two-speed planetary gear train (Fig.1) manufactured by scheme 55V5 in [4]. Fig. 2 depicts the structural scheme and the determined shaft torques with no regard to losses. The gear train is reversible with two speed ratios $i_{Br.1} = 5$ and $i_{Br.2} = -26,4267$. With the help of original additional elements any of the coupling gear trains could be blocked so that only the other one is operational [5].

A bench is developed (Fig. 3) and it allows static and dynamic studies on the gear train.



$z_{1I} = 11;\ z_{3I} = 37;\ t_I = \dfrac{z_{3I}}{z_{1I}} = 3{,}364;\qquad z_{1II} = 9;\ z_{3II} = 36;\ t_{II} = \dfrac{z_{3II}}{z_{1II}} = 4$

Fig.1 Experimental gear train



$$i_{Br.1} \equiv i_{V51} = -\frac{T_B}{T_A} = +1 + t_{II} \qquad i_{Br.2} \equiv i_{V52} = -\frac{T_B}{T_A} = -\frac{-(1+t_{II})t_I}{t_I - t_{II}} = \frac{t_I + t_I.t_{II}}{t_I - t_{II}}$$

Reversing the rotational direction of the output shaft while switching between the brakes when $t_{II} > t_I$

$$T_A = \frac{t_I - t_{II}}{t_I} < 0 \qquad i_{Br.2} \equiv i_{V2} = \frac{t_I + t_I.t_{II}}{t_I - t_{II}} < 0$$

Fig. 2. Structural scheme of the experimental gear

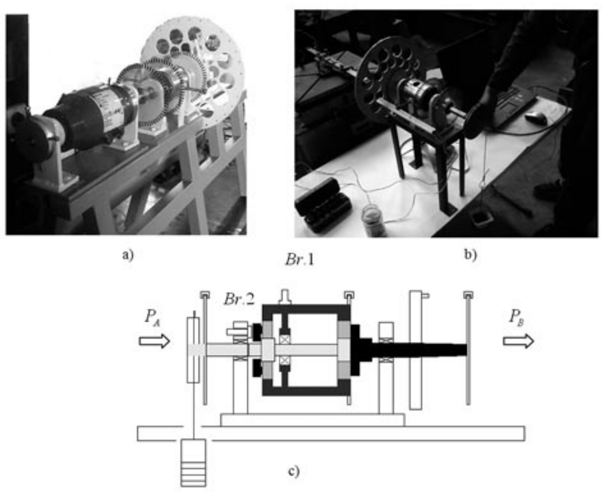
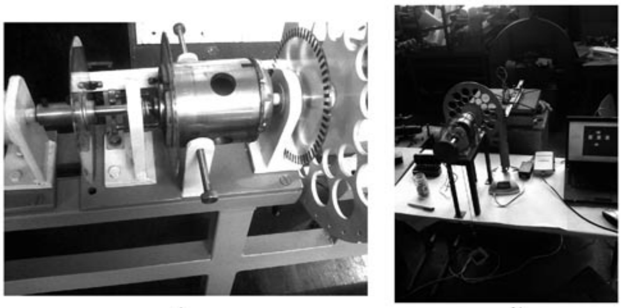


Fig. 3. Bench for testing planetary gear trains
a) for dynamic testing – general view
b) for determining the static efficiency – general view
c) for determining the static efficiency – sketch

Various units of the tested gear train are fixed by means of bolts (Fig. 4a). The situation of the bolts allows the inducement of different levels of eccentricity of the crown-gear (common for the two gear trains) and the carrier of the first train. Thus the impact of these inaccuracies on the efficiency could be examined. The raster discs fixed to the three external shafts of the train allow the measurement of the angle of rotation as well as the angular speed (with photo-sensors).

In the case of static testing (Fig. 3b,c) a torque T_A is applied to the input shaft by means of a disc and weights. The-output torque T_B is measured with a tenso-beam (Fig. 4b) and tensometric equipment and is registered on a PC.

The input shaft is loaded with a disc 46mm in radius and the output torque is determined with the help of disc 125mm in radius.



a) b)

Fig. 4. Bench for testing planetary gear trains - fragments
a) bolts fixing the units of the train
b) measuring the output torque

3. Theoretical basis of the experiment

The kinematic speed ratio of the gear train when working with one degree of freedom (one brake is active) is $i_k = \frac{\omega_A}{\omega_B}$, where ω_A and ω_B are the respective angular speeds of the input and the output torques.

The power of the input and output torques is respectively $P_A = T_A.\omega_A$ and $P_B = T_B.\omega_B$.

The efficiency is

$$\eta = \frac{P_B}{P_A} = \frac{T_B.\omega_B}{T_A.\omega_A} < 1 \qquad (1)$$

If the losses are disregarded (η = 1)

$$T_B = -T_A.i_k \ , \qquad (2)$$

and if the losses are taken into account

$$T_B = T_A.i_T \ , \qquad (3)$$

where i_k and i_T are the kinematic speed ratio and the ratio of the torques (input and output).

From (1), (2) and (3) it follows as a consequence that the efficiency could be presented using the two ratios

$$\eta = -\frac{i_T}{i_k} < 1 \qquad (4)$$

4. Aim of the experiment

The main aim of the experiment is to verify the analytical expressions for determination of the efficiency of a coupled gear train as a function of the torque ratios t_I and t_{II} and the internal efficiencies η_{0I} and η_{0II} of the coupling gear trains.

For the reviewed coupled gear train (Fig. 1) the efficiency with fixed crown-gears (Fig. 1a) is [4]

$$\eta_{Br.1} = \frac{1+\eta_{0II}.t_{II}}{1+t_{II}} \tag{5}$$

and in the case of a fixed carrier of the first train (Fig. 2b) it is [4]

$$\eta_{Br.2} = \frac{\dfrac{\eta_{0I}.t_I + \dfrac{\eta_{0I}}{\eta_{0II}}.t_I.t_{II}}{\eta_{0I}.t_I - \dfrac{t_{II}}{\eta_{0II}}}}{\dfrac{t_I + t_I.t_{II}}{t_I - t_{II}}} \tag{6}$$

where $t_I = 3,3636$, and $t_{II} = 4,0$, and the internal efficiencies η_{0I} and η_{0II} of the coupling trains are determined empirically.
During the course of experiment the impact of the eccentricity of crown-gears and carrier of the first train on the efficiency of the coupled train is taken into account.

5. Method used in the experiment

The experimental determination of the static efficiency is performed in the following series:

- One of the train's units is fixed (e.g. the crown-gear with Br. 1);
- The input shaft is loaded with a torque T_A by means of a disc and weights;
- The output torque T_B is measured with a tenso-beam and the results are registered on a PC;
- The eccentricity of the fixed unit is changed

The experiment is performed for various positions (angles of rotation) of the input shaft within the kinematic cycle with loading torques of different magnitudes.
The same procedure is repeated with a fixed carrier for the coupled gear train (work with Br. 2) and the coupling trains so that the internal efficiencies η_{0I} and η_{0II} can be determined.

6. Results of the experiment

6.1. Efficiency determination for the coupled gear train $\eta_{Br.1}$ with active brake Br.1

The following summarized results (Table 1) are obtained after fixing the coupled shaft (the two crown- gears) and consecutively loading the input shaft with loads of 500, 550, 600 and 700 grams at a radius of 46mm.

Table 1. Results of the measurements on the coupled gear train with a fixed coupled shaft (the two crown-gears) – active brake is Br.1

Loading mass, g		500	550	600	700
Input torque, Nm		0,230	0,253	0,276	0,322
Output torque, Nm	min	0,825	0,900	0,987	1,138
	m	0,837	0,916	0,998	1,150
	max	0,860	0,927	1,010	1,162
Ratio of the torques i_T	min	3,587	3,557	3,578	3,534
	m	3,640	3,620	3,616	3,571
	max	3,740	3,664	3,659	3,609
Efficiency $\eta_{Br.1}$	min	0,717	0,711	0,716	0,707
	m	0,728	0,724	0,723	0,714
	max	0,748	0,733	0,731	0,722

For each stage of loading 7 consecutive values are registered while slowly lowering the weight. The table shows the minimum, average and maximum values.
Average value for the measured efficiency is

$\eta_{Br.1} = 0,722$

Dispersion of the experimental results is between – 2 % and + 3,6 %.

6.2. Efficiency determination for the coupled gear train $\eta_{Br.2}$ with active brake Br.2

The following summarized results (Table 2) are obtained after fixing the carrier shaft of the first gear train and consecutively loading the input shaft with loads of 500, 550, 600 and 700 grams at a radius of 46mm.

Table 2. Results of the measurements on the coupled gear train with a fixed single shaft (carrier of the first train) – operating brake is Br.2

Loading mass, g		500	550	600	700
Input torque, Nm		0,230	0,253	0,276	0,322
Output torque, Nm	min	1,164	1,294	1,425	1,670
	m	1,175	1,308	1,438	1,684
	max	1,186	1,325	1,450	1,698
Ratio of the torques i_T	min	5,061	5,115	5,163	5,168
	m	5,109	5,170	5,210	5,230
	max	5,156	5,237	5,254	5,273
Efficiency $h_{Br.2}$	min	0,1915	0,1935	0,1953	0,1962
	m	0,1933	0,1956	0,1971	0,1979
	max	0,1951	0,1982	0,1989	0,1995

For each stage of loading 7 consecutive values are registered while slowly lowering the weight. The table shows the minimum, average and maximum values.

Average value for the measured efficiency is

$\eta_{Br.1} = 0{,}196$

Dispersion of the experimental results is between – 2,3 % and + 1,8 %.

The experiment shows that when the eccentricity of the crown-gears and the shaft increases the average value of the efficiency decreases while its dispersion increases.

6.3. Efficiency determination for the first coupling train.

The internal efficiency of the first coupling train is determined with blocked second train – removed satellites and a pinion fixed to the carrier.

In this case, to avoid measurements of the crown-gear torque, the train efficiency is measured when transmitting motion from the pinion to the carrier and after that the internal efficiency is calculated.

$\eta_{0I} = 0{,}76$

6.4. Efficiency determination for the second coupling train.

The internal efficiency of the second coupling train is determined with blocked first train – removed satellites and a pinion fixed to the carrier.

Just as in 6.3. the train efficiency is measured when transmitting motion from the pinion to the carrier and after that the internal efficiency is calculated.

$\eta_{0II} = 0{,}71$

7. Results processing

7.1. Internal efficiency of the second coupling gear train.

During operation of the coupled gear train with active brake Br.1 (fixed coupled shaft – the two crown gears) power flows through the second coupling train only.
After substituting the empirically obtained value for $\eta_{Br.1}$ in (5) the internal efficiency of the second train η_{0II} becomes

$$\eta_{0II} = \frac{(1+t_{II})\eta_{Br.1} - 1}{t_{II}} = \frac{(1+4)0{,}722 - 1}{4} = 0{,}6525$$

which is lower than the one determined when only the second coupling train is active. It is logical, having in mind that in the process of determining $\eta_{Br.1}$ the first train floats. In quality coupling trains with higher internal efficiency this difference will be smaller.

7.2. Efficiency of the coupled gear train with active brake Br.2 (fixed carrier of the first coupling train).

In this case the gear train works with power circulation.
After substituting the experimentally obtained in 6.3 and 6.4 values for the internal efficiencies of the coupling trains η_{0I} and η_{0II} in (6), the efficiency of the coupled gear train becomes

$$\eta_{Br.2}=\frac{\dfrac{\eta_{0I}.t_I+\dfrac{\eta_{0I}}{\eta_{0II}}.t_I.t_{II}}{\eta_{0I}.t_I-\dfrac{t_{II}}{\eta_{0II}}}}{\dfrac{t_I+t_I.t_{II}}{t_I-t_{II}}}=\frac{\dfrac{0,76.3,3636+\dfrac{0,76}{0,71}.3,3636.4}{0,76.3,3636-\dfrac{4}{0,0.71}}}{\dfrac{3,3636+3,3636.4}{3,3636-4}}=0,20355$$

This result is close enough to the empirically obtained in 6.2 and so the theoretical relation is assumed proven.

The experimentally obtained results for the efficiency of the coupled gear train (without eccentricity) are close to the ones calculated by (5) and (6). This applies to average values as well as to dissipation.

When eccentricity is introduced in the crown-gears and the carrier the experimental results significantly differ from the theoretical.

8. Conclusion

The results of the experiment verified the validity of the analytical expressions (5) and (6) [4] for determination of the efficiency of the coupled gear train as a function of the torque ratios t_I and t_{II} and the internal efficiencies η_{0I} and η_{0II} of the coupling trains.

It was established that when calculating the efficiency in the case of power transmission through one coupling train only (variant 11V51 in Fig. 2) one must reduce the result in order to take into account the floating of the other train.

The experimental results show that the method suggested in [7] for determination of the static efficiency could be applied to quality control of batch produced gear trains.

References

1. Александров, И. К. К определению потерь в механических передачах. // Вестник машиностроения, 1998, № 6, с. 12 – 14.
2. Arnaudov, K. Experimental determination of the efficiency of planetary gears.// Proceedings of the AUSTRIB′94 “Frontiers in tribology”, Perth, Australia, 1994.
3. Epicyclic Gear Train. Epicyclic Gear Train Experimental Lab, ME-372 Mechanical Design II, Page 1 of 5, S2007, GR.
4. Karaivanov, D., Troha, S. Examining the possibilities for using coupled two-carrier planetary gears in two-speed mechanical transmissions. // “Machinebuilding and electrical engineering”, Nr. 5 – 6, 2006, p. 124 – 127.
5. Karaivanov, D., Popov, R. Experimental study on the clearances of the two-stage planetary gear. // Journal of the University of Chemical Technology and Metallurgy, XXXVIII, 4 (2003), p.1331 – 1338.
6. Predki, W., Jarhov, F., Kettler, J., Calculation method for the determination of the oil sump temperature of industrial planetary gears. // International Conference on Gears, V. 1, 13-15 March 2002, Munich, VDI-Berichte 1665, p. 507 – 522.
7. Troha, S., Karaivanov, D. Experimental comparative analysis of the manufacturing quality of planetary gear trains. // Proceedings of the 25th Int. Conf. MTF’2007, v. II, Sozopol, Bulgaria, 14 -16 September 2007, p. 259 – 264.
8. Živković, P., Ognjanović, M. Experimental determination of losses and planetary gear set efficiency coefficient. // Journal of Mechanical Engineering Design, No 1, 2000, p. 21 – 28.

Acknowledgements

The authors would like to thank Vinko Sušanj, whose deft hands and invaluable advice have greatly contributed to the construction of the bench and the experimental gear train.
The authors would like to thank Iordan Pitkov, active participant in experiments.