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Corina Gruescu, Ioan Nicoara, Carmen Mesteroiu, Nicoleta Gillich

Modelling and Optimization of Multiple Closed Loop Stands for Testing Gearboxes

In order to increase efficiency of testing gearboxes there is the choice of multiple closed loop test rigs. The paper presents the design, mathematical modelling and analysis of a two closed loop test stand, working in reversible regime. Energetic optimization of the system is based on minimization of energetic flow paths, loading and stress variation at reversible motion. Energetic equilibrium of loops imposes a symmetrical scheme and a central position of the drive motor. Different loading laws are needed for each rotation way in order to equalize contact stress of gears.

Keywords*: test rig, gear transmission, multiple closed loops, energetic optimization, mathematical modelling, efficiency*

1. Introduction

Closed-loop test rigs base on different kinematical schemes [1], which influence the energetic and economic parameters of the testing process. The most usual scheme includes two transmissions symmetrically positioned along a closed chain. The efficiency of testing or running in of gearboxes can be achieved designing a more complex loop consisting of four transmissions or a double-loop circuit. The four transmissions scheme is a little bit unbalanced concerning loading on the gears' flanks in reversible regime, which is undesirable especially at high powers.

Multiple loops are more difficult to design and optimize, but ensure energetic balance and high efficiency of the process, as three transmissions are tested simultaneously.

2. Modeling of a two-loops mechanical test rig

Figure 1 presents a double-loop scheme, designed for testing three steps gearboxes. The scheme contains the transmissions MT0 (central position- return transmission) MT1 and MT2 (symmetrically placed).

The loading device (a quasi-differential mechanism) works for both loops as it is inserted within the central branch. Two pre-loading couplings PC are meant to compensate clearances and ensure equal indications at the measuring device attached to external branches, in the beginning of the test.

Mechanical closing of the loops is accomplished through six cylindrical gears (transmission ratio equal to 1).

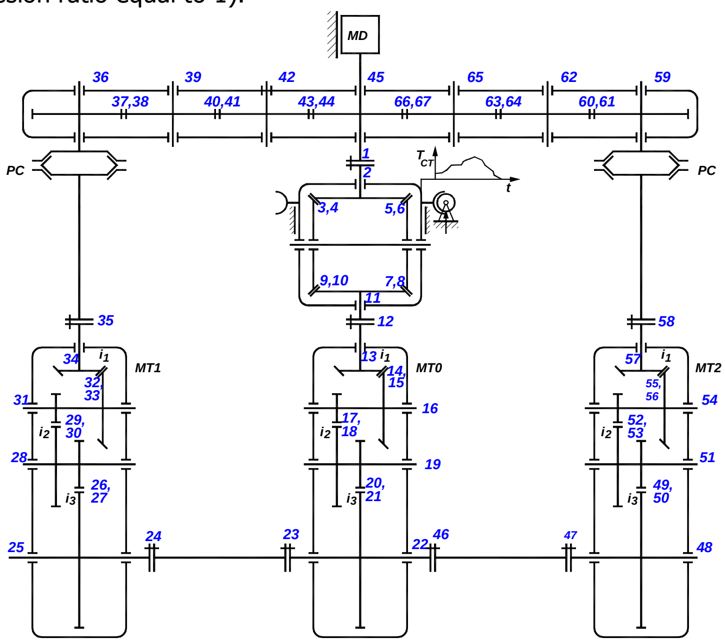

Figure. 1. Kinematic scheme of a two closed-loops stand

Figures 2 and 3 show the paths and directions of main and secondary energetic flows for the cases when angular velocity's sign and torque's sign coincide (case denoted A), respectively for the counter case (denoted B). In figures 2 and 3 the following notations are used: MT0, 1, 2 – mechanical transmissions; DM – driving motor, CT – closing transmission, CP – couplings to pre-load the two loops equally, T_{CT} – torque indicated at the loading device; ω (n) – angular velocity (rotation speed); Φ_{A1}, Φ_{B1} – main energetic flow along loop 1 at directions A and B; Φ_{A2}, Φ_{B2} – main energetic flow along loop 2 at directions A and B; Φ_{SA1}, Φ_{SB1} – secondary energetic flow along loop 1 at directions A and B; ΦSA2, ΦS_{B2} – secondary energetic flow along loop 2 at directions A and B.

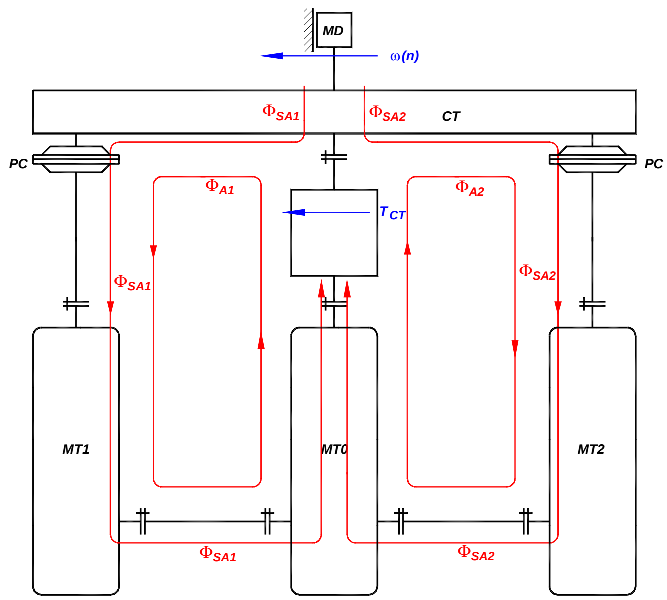

Figure. 2. Path of energy for direction A

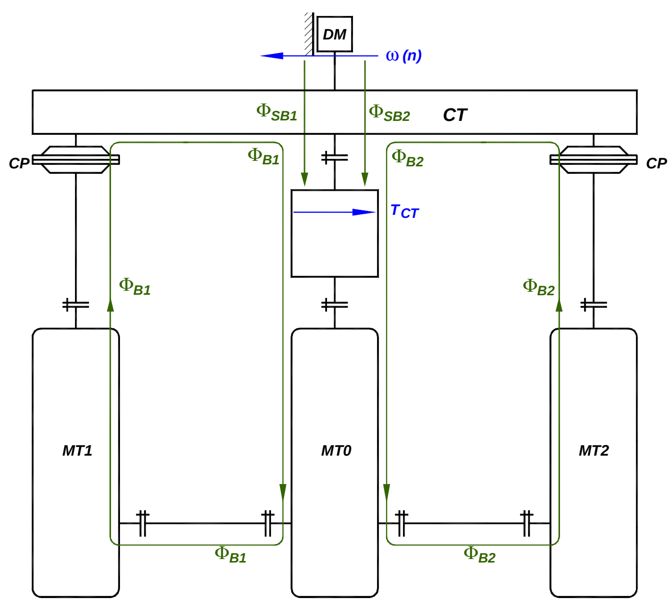

Figure. 3. Path of energy for direction B

The main energetic flow is associated to the energy effectively recycled around the loop. The secondary energetic flow describes the energy needed to compensate mechanical losses along the loop. The lengths of secondary flows are very different in reversible regime.

Geometrical and loading symmetry imposes the placement of the motor on the central branch of the scheme. Motor's position is not subject of optimization. The important issue is energetic and loading balance at reversible regime. Thus, optimization aims to correlate the loading laws for directions A and B.

For the numerical modelling of the scheme the following entrance data are used:

- the transmissions are identical and achieve a total transmission ratio $i_{total} = 28.8 = 2.5*3.2*3.6$
- there are 67 points where energy loss occurs or kinematic parameters modify, as bellow:
  - bearings in points 2, 11, 13, 16, 19, 20, 22, 25, 28, 31, 34, 36, 39, 42, 45, 48, 51, 54, 57, 59, 62, 65
  - couplings in points 1, 12, 23, 24, 35, 46, 47, 58
  - flanks of gears in contact in points 3, 5, 7, 9, 14, 17, 20, 26, 29, 32, 37, 40, 43, 49, 52, 55, 60, 63, 66
  - friction of gears with oil in points 4, 6, 8, 10, 15, 18, 21, 27, 33, 38, 41, 44, 50, 53, 56, 61, 64, 67.
- efficiency of ball-bearings: 0.99
- efficiency of couplings: 0.98
- efficiency of bevel gears: 0.99
- efficiency of cylindrical gears: 0.99
- losses due to friction between gears and oil: 0.02.

After calculating the loading $T_{A,B}/T_{CT}$ in all 67 points, [2], and considering that the indication T_{CT} at the loading device is constant, the following important parameters are taken into account:

$\Sigma(T_A/T_{CT})=366.9185$, $\Sigma(T_B/T_{CT})=686.3229$, $\Sigma[(T_A-T_B)/T_{CT}]= - 319.4047$,

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The coefficient 0.534615 was applied to parameters referring to direction B. Energetic characteristics of the circuit in all points for the optimized scheme are shown in table 1.

Table 1. Optimal energetic parameters of the two-loop circuit

ID point	T_A/T_{CT}	$T_{B\,opt}/T_{CT}$	$(T_A-T_{Bopt})/T_{CT}$	T_A/T_{Bopt}
0	**1**	**2**	**3**	**4**
1	2.0614	1.0480	1.0135	1.9671
2	1.9800	1.0800	0.9000	1.8333
3	1.9602	1.0909	0.8693	1.7968
0	**1**	**2**	**3**	**4**
4	1.9210	1.1132	0.8078	1.7256

5	1.9018	1.1244	0.7773	1.6913
6	1.8638	1.1474	0.7164	1.6243
...				
17	4.0057	3.3366	0.6691	1.2005
18	12.5619	10.8950	1.6669	1.1530
19	12.4363	11.0050	1.4312	1.1301
20	12.3119	11.1162	1.1957	1.1076
21	43.4364	40.8350	2.6014	1.0637
22	43.0020	41.2475	1.7545	1.0425
23	21.0710	21.0446	0.0264	1.0013
24	20.6496	21.4741	-0.8245	0.9616
25	20.4431	21.6910	-1.2479	0.9425
26	20.2386	21.9101	-1.6715	0.9237
27	5.5094	6.2103	-0.7009	0.8871
28	5.4543	6.2731	-0.8188	0.8695
29	5.3998	6.3364	-0.9367	0.8522
30	1.6537	2.0205	-0.3669	0.8184
...				
41	0.5687	0.9401	-0.3714	0.6049
42	0.5630	0.9496	-0.3866	0.5929
43	0.5574	0.9592	-0.4018	0.5811
44	0.5462	0.9788	-0.4326	0.5581
45	1.0815	1.9773	-0.8958	0.5469
45	2.0408	1.0585	0.9823	1.9280
46	21.0710	21.0446	0.0264	1.0013
47	20.6496	21.4741	-0.8245	0.9616
48	20.4431	21.6910	-1.2479	0.9425
49	5.6218	6.0861	-0.4643	0.9237
50	5.5094	6.2103	-0.7009	0.8871
...				
61	0.5921	0.9030	-0.3109	0.6557
62	0.5861	0.9121	-0.3260	0.6426
63	0.5803	0.9213	-0.3410	0.6298
64	0.5687	0.9401	-0.3714	0.6049
65	0.5630	0.9496	-0.3866	0.5929
66	0.5574	0.9592	-0.4018	0.5811
67	0.5462	0.9788	-0.4326	0.5581
sum			0.0000	
mean				0.9872

3. Discussion and conclusions

The following notices are to be brought to attention:

- in reversible regime, the balance of loadings is, practically, ensured. The ratio of reversible loading along the most stressed sections is [1.06…0.92] between points 21…26, on loop 1 and [1.00…0.92] between points 46…49 on loop 2. In table 1 these points are emphasized with red characters
- the mean ratio of reversible loading turned from 0.5278 for the circuit before optimization to 0.9872 for the optimized scheme
- the energetic disequilibrium between directions A and B was substantially reduced. From

$$\left|(T_A - T_B)/T_{CT}\right| \approx 35$$

for the initial scheme, the optimization ensured

$$\left|(T_A - T_B)/T_{CT}\right| \approx 2.6 .$$

The results above lead to the conclusion that the scheme is adequately designed for testing or running in of gear transmissions for best energetic conditions.

References

[1] Nicoara. I. , *Incercarea angrenajelor*. Editura Orizonturi Universitare, Timisoara, 2001.

[2] Eftimie, M. et al., *Optimization Criteria For Closed-Loop Transmission Test Stands*. The 2nd International Conference, Power Transmissions, Novi Sad, Serbia & Montenegro 2006: (L127) 309-316.

- *Conf. Dr. Eng. Corina Gruescu, "Politehnica" University of Timişoara, Mechatronics Department, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara,* corina.gruescu@mec.upt.ro
- *Prof. Dr. Eng. Ioan Nicoară, "Politehnica" University of Timişoara, Mechatronics Department, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara,* ioan.nicoara@mec.upt.ro
- *Dr. Eng. Carmen Meşteroiu, "Politehnica" University of Timişoara, Mechatronics Department, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara,* mesteroiu_carmen@yahoo.com
- *Prof. Dr. Eng. Nicoleta Gillich, "Eftimie Murgu" University of Reşiţa, Piaţa Traian Vuia, nr. 1-4, 320085, Reşiţa,* n.gillich@uem.ro