Pablo López García:

- PAPER "DYNAMIC MODELLING AND TORQUE RIPPLE MINIMIZATION OF A LIGHTWEIGHT ULTRA-HIGH TRANSMISSION RATIO HARMONIC DRIVE" FROM E. TSOLAKIS ET AL., P. 68 OF THE PROCEEDINGS

I found the paper nicely written and very interesting: designing a new gearbox based on a complex system such as a Strain Wave gear is a process full of different challenges, particularly using thermoplastic materials. Many thanks for your interesting contribution! Based on my own struggles to build plastic gearboxes capable of resisting high loads, I have two questions on the practical feasibility of the proposed design:

Q1: SYSTEM EFFICIENCY: On p.70 and 71 the authors mention that a "standard" efficiency for plastic reducers (75%) was used in the calculations... According to the Harmonic Drive (HD) catalogue, a lubricated steel HD has a peak efficiency around 82% for this size and gains around 1:50, which would result in a total peak system efficiency of 82%*82% = 67% for a complete system of two stages, built in steel and ideally lubricated. Could you provide some additional background information on the selection/assumption for this 75% standard system efficiency?

A1: Concerning the comment "standard efficiency for plastic reducers (75%)", it was indeed attributed to a typo-error. In the final selection of the stepper on the frame of the project, it was considered 75% per stage, thus 56.25% in total.

Q2: TEETH LOADING: A conventional (case-hardened steel) Harmonic Drive with 200mm outer diameter needs around than 100mm width to handle repeated peak torques of 1700Nm (CSG/CSF catalogue of HD). This raises some questions in terms of how a PPS-thermoplastic second stage in your gearbox can handle these torques with a slightly larger width: have you somehow been able to validate your FEM load calculations with some practical or experimental data?

A2: This can be explained due to the intrinsic deformability of selected thermoplastics which facilitate multiple tooth contact (more than 7 pairs in engagement) and also ameliorate peak Hertzian stresses at roller contact. Experimental work is planned for the future steps of the presented work, along with integration of research outcomes from our on-going research activity on plastic and plastic reinforced gears.

With best regards,
Efstratios Tsolakis

- PAPER "A WOLFROM TRANSMISSION WITHOUT CARRIER" FROM PROF. HÖHN ET AL., P. 87 OF THE PROCEEDINGS

I found this paper and the proposed approach to obtain a Wolfrom PGT design avoiding the need for a rigid, conventional carrier highly interesting. I particularly appreciated the idea of using two differentiated axial bearings and some small plates to solve the rotation of the planets both around their own pin axis and around the axis of the central shaft, and its combination with the proposed low-loss macrogeometry: Many thanks for this interesting contribution! I have just two questions related to your introduction statement about the efficiency limitations of conventional Wolfrom PGTs, and to the constructive arrangement of the gearbox:

Q1: EFFICIENCY INCONSISTENCIES & EXPANSIONS: In the Introduction a certain "inconsistency in the measured efficiency" is evaluated as "disadvantageous", and that the "range of efficiency is too expanded" therefore limiting their practical uses. Could you explain further in which way the measured efficiency of these devices is inconsistent, and/or expanded?

A1: I can only refer to the sentences after the word "disadvantageous" . That is an information which I have got oral by two producers of wolfrom-transmissions, There is nothing published form these firms an no one will be citated and give the allowance to name them. But nevertheless it could be the true. They produce the wolfrom-transmission with the same tolerances and their own quality requirements, but they sometimes measure very different efficiencies for the same transmissions. Though I make my own thinking to this phenomen and that what I can suppose today-The text behind "disadvantageous" should be read so: "All companies thatproduce such transmission systems try to manufacture all parts with low tolerances- often with unsatisfactory results. So far, these transmission systems are only rarely produced and used in practical applications as the range of efficiency is too expanded. Details on the design and the properties of developed systems are usually not published as no one wants to acknowledge the problems."

Q2: CARRIERLESS ARRANGEMENT: I found Figs. 15 and 16 a bit confusing: these plates 36 and 40, are they (i) two small discs on each side of each planet (therefore 4x2 = 8 plates) and coaxial to the planets, as it seems to be represented in Fig. 15, or are they just (ii) two large flat rings coaxial to the central axis of the gearbox, and one on each side of the gearbox, as it seems to correspond more to Fig. 16...? I tried to find myself the response on reference [9], but it is apparently not public yet...

A2: The two plates no 36 and 40 are coaxial to the input or central shaft, the four planets have the axial bearings No. 38 and 42 (4 axial bearings equivalent to the number of planets), and they are running against the two plates 36 and 40, sorry for the small mistake, you are right, I have not drawn a line for these plates in fig.15. with this line it would be correct from the drawing. Sorry for this mistake. The fig are a copy of the patent application, it is now so in the paper. In the ppt-Version I would correct this mistake.

With kind fregards
Bernd-Robert Höhn