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(54) POWER UNIT ARRANGEMENT IN TRACTIVE RAIL VEHICLES

(71) We, THYSSEN INDUSTRIE AKTIENGESELLSCHAFT, of Am Rheinstahlhaus 1, D-4300 Essen, Federal Republic of Germany, a Joint-Stock Company organised under the laws of the Federal Republic of Germany, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:-

This invention relates to a power unit arrangement in tractive rail vehicles having driving axles which are movably mounted in the vehicle frame or bogie frame and are provided with axle gearboxes and which are driven by way of articulated shafts without length compensation means by a drive unit mounted stationary in the vehicle or bogie frame.

A power unit arrangement is known in which distributor gearboxes are mounted stationary in the main frame and are connected by articulated shafts to the axle gearboxes of the wheel sets. In this arrangement the articulated shafts compensate for the relative movements between the wheel sets and the main frame. Since the distributor gearboxes are mounted in the main frame and the bogies are pivotable in relation to the latter, in this arrangement of the power unit the articulated shafts must be equipped with length compensation means (DT-AS 10 56 645). In this arrangement it is disadvantageous that the telescopic length compensation means of the articulated shafts is liable to wear and thus requires maintenance. It is considered an additional disadvantage that articulated shafts provided with length compensation means can be supplied only in minimum lengths which make it difficult to obtain short wheelbases.

A power unit arrangement is also known in which the articulated shafts between the drive unit and the axle gearboxes are made in one piece, while the output shafts of the drive unit is freely movable in the axial direction (DT-PS 17 75 238). In this power unit arrangement the disadvantages mentioned above in connection with articulated shafts having telescopic length compensation means are overcome, but it is disadvantageous that because of the necessary movability in the axial direction of the output shaft the output stage of the drive unit must be equipped with straight-toothed spur gears, which, particularly in high-speed tractive rail vehicles, do not provide the required quiet running because of the high peripheral speeds of the gears.

The problem underlying the invention is that of so designing the power unit arrangement in tractive rail vehicles of the foregoing kind that length compensation components subject to wear are not required for torque transmission between the drive unit and the axle gearboxes.

To this end, the present invention consists in a tractive rail vehicle having driving axles which are movably mounted in a vehicle frame or a bogie frame and are provided with axle gearboxes which are driven by way of articulated shafts without length compensation by a drive unit mounted stationary in the vehicle or bogie frame, characterised in that the drive axles are mounted resiliently in the longitudinal direction of the vehicle between fixed stops and are fastened by the axle gearbox input shafts which are mounted without axial movement, to a fixedly mounted output shaft of the drive unit by means of the articulated shafts which in addition to the torque can be loaded in the axial direction with tensile and compressive forces.

In order to keep at a low level the loading of the thrust bearing of the drive unit output shaft by the tensile and compressive forces originating from the propulsion force of the driving wheels and applied by way of the articulated shafts, it is advantageous for the drive unit to be in the form of a gearbox having a helical gear on the output shaft, the angle of inclination of the gear teeth being so directed that the axial thrust of the output shaft resulting from the tooth engagement of the gearbox output stage is directed oppositely to the axially directed propulsion forces in the articulated shafts.

In order to keep the loading of the thrust bearings of the input shaft in the axle gearboxes at a low level, it is advantageous for the angle of inclination of the teeth of the gear mounted on the input shaft of the axle gearboxes to be so directed that the axial thrust of the input shaft resulting from the tooth engagement of the axle gearbox input stage is directed oppositely to the axially directed propulsion forces in the articulated shafts.

The advantages achieved with the invention consist in that wear no longer occurs in length compensation components in the transmission of torque, that the thrust bearings on the gearbox shafts are not additionally loaded by the additional transmission of the propulsion forces, but on the contrary are even relieved of load, and that the cardan joints of the articulated shafts are loaded more heavily only to so slight an extent that reinforcement of the articulated shafts normally used is not necessary.

In order that the invention may be more readily understood, reference is made to the accompanying drawings which illustrate diagrammatically and by way of example, one embodiment thereof, and in which:-

Figure 1 is a plan view of a power unit arrangement of a two-axled running gear;

Figure 2 is a vertical section on the line A-A′ of Figure 1; and

Figure 3 is a vertical section on the line B-B′ of Figure 1.

In the power unit arrangement shown in Figure 1 the drive unit mounted stationary in the vehicle frame 10 consists of a distributor gearbox whose output shaft 2 is taken out on both sides and connected by means of articulated shafts 3,3′ not provided with length compensation means to the input shafts 4,4′ of two axle gearboxes 5,5′. The axle gearboxes 5,5′ are mounted in known manner on the axle shafts 6,6′ which in turn carry the driving wheels 7,7′ and are mounted in axle bearings 8,8′. As can be seen in Figure 3, these axle bearings 8,8′ are supported against the vehicle frame 10 by rubber springs 9 mounted above the axle bearings. The rubber springs 9 are so shaped that they not only resiliently support the axle bearings 8,8′ in the vertical direction, but also have movability horizontally in the X and Y directions, this movability being limited by corresponding fixed stops 11 on the frame 10. The axle gearboxes 5,5′ are respectively provided with torque supports 12,12′. The torque supports 12,12′ are fastened on the vehicle frame 10 by means of forks 13 and spring elements 14.

The articulated shafts 3,3′ each have two cardan joints 15,15′. Thus the wheel sets connected to the distributor gearbox by the articulated shafts 3,3′ can move freely in all directions as permitted by the springs 9 and stops 11. The propulsion forces V acting between the driving wheel 7,7′ and the rail are at the same time transmitted through these articulated shafts to the drive unit 1 mounted stationary in the vehicle frame 10.

With uniform loading of the four driving wheels 7,7′, the axial force in each articulated shaft amounts to twice the propulsion force per driving wheel. In the power unit arrangement shown in Figures 1 and 2 the entire tractive force $Z = 4 \times V$ is thus transmitted to the vehicle frame 10 by way of the drive unit 1.

A thrust bearing 17 of the input shaft 4 is loaded with an axial force $P_K = 2\ V - A_K$, wherein A_K is the axial force from the teeth of the bevel pinion 16.

An axial bearing 19 of the output shaft 2 is loaded with an axial force $P_S = 4\ V - A_S$, wherein A_S is the axial force from the teeth of the spur gear 18.

If the bevel pinion 16 and the spur gear 18 are given suitable angles of inclination, the axial forces A_K and A_S can be so determined in respect of magnitude and direction that the axial bearings 17 and 19 can be loaded in the best possible manner.

With the aid of the following example, calculation of these forces will be explained more fully. The following data are selected:

Wheel load G = 8000 kg,

Driving wheel diameter D = 1 metre,

Coefficient of friction between wheel and rail $\mu = 0.3$, Propulsion force per wheel $V = G \times \mu kp$,

$= 8000 \times 0.3 = 2400$ kp.

Torque in the axle shaft $= M_A = 2 \times V \times \frac{D}{2}$ mkp,

$= 2 \times 2400 \times ½ = 2400$ mkp.

Mean diameter of bevel gear on axle shaft $d_m = 0.4$ metre.
Peripheral force on bevel gear on axle shaft $U = \frac{M_A}{\left(\frac{d_m}{2}\right)}$.

$$= \frac{2400}{\left(\frac{0.4}{2}\right)} = 12000 \text{ kp.}$$

Bevel gearing: The calculation is based on a transmission ratio $i = 2$,

$$i = \frac{Z_2}{Z_1} = \frac{\text{Number of teeth of bevel gear}}{\text{Number of teeth of bevel pinion}}$$

For the bevel gearing spiral gears are chosen having a mean spiral angle $\beta_r = 28°$ and a pressure angle $\alpha = 20°$
For the foregoing values of i, β_r and α, the relation of axial force to peripheral force can be read off from a calculation chart of Gleason Works, Rochester 3, N.Y., U.S.A.:
If the direction of rotation of the bevel pinion coincides with its spiral direction, the relation of

$$\frac{\text{Axial force}}{\text{Peripheral force}} = \frac{A_{K1}}{U} = +0.67$$

Thus axial force $A_{K1} = +0.67 \times U$
$= +0.67 \times 12000 = +8040$ kp.
If the direction of rotation of the bevel pinion does not coincide with its spiral direction, the relation of

$$\frac{\text{Axial force}}{\text{Peripheral force}} = \frac{A_{K2}}{U} = -0.29$$

Thus axial force $A_{K1} = -0.29 \times U$
$= -0.29 \times 12000 = -3480$ kp.

In connection with the direction arrows in Fig. 2, the sign "+" means here that the axial force A_{K1} is directed towards the left "←", the sign "-" means that the axial force is directed towards the right "→".

The thrust bearing 17 of the bevel gear shaft 4 is loaded, under the influence of the propulsion forces and torque transmission, with an axial force P_K.

For the direction of travel 1, in the representation according to Fig. 2 directed towards the left, the thrust bearing 17 is loaded with an axial force
$P_{K1} = 2V + A_{K1}$ kp
$= -2 \times 2400 + 8040 = +3240$ kp.

For the direction of travel 2, in the representation according to Fig. 2 directed towards the right, the thrust bearing 17 is loaded with an axial force
$P_{K2} = 2V + A_{K2}$ kp
$= +2 \times 2400 - 3480 = +1320$ kp.

If travelling time is equally divided between forward and reverse travel, the mean bearing loading will be (assuming a varying loading and constant number of revolutions - according to an equation given in the German Katalog 4100, Edition 1965 of FAG Kugelfischer Georg Schafer & Co., Schweinfurt, page 32):

$$P_m = \sqrt[3]{P_1^3 \times \frac{q_1}{100} + P_2^3 \times \frac{q_2}{100}} \quad kp$$

where, $q_1 = 50\%$ of the portion of travelling time for forward travel,
$q_2 = 50\%$ of the portion of travelling time for reverse travel.

Thus, the mean loading bearing 17 in the present example is:-

$$P_m = \sqrt[3]{0.5 \times 3240^3 + 0.5 \times 1320^3} = 2628 \ kp$$

If the same bearing were loaded only by the torque transmission, as is the case with a conventional articulated shaft provided with length compensation means, the mean bearing load would be as follows:-

$$P_{mx} = \sqrt[3]{0.5 \times 8040^3 + 0.5 \times 3480^3} = 6549 \ kp$$

$$\frac{P_{mx}}{P_m} = \frac{6549}{2628} = 2.49$$

Thus, in the case of pure torque transmission the loading of the thrust bearing 17 would be approximately $2\frac{1}{2}$ times as great as in the power unit arrangement of the invention, in which the articulated shafts not provided with length compensation means additionally transmit the propulsion forces as axial forces.

Calculation of additional loading of the cardan joints of the articulated shafts by the propulsion forces to be transmitted

Torque in axle shaft $M_A = 2400$ mkp

Torque in articulated shaft: $M_G = \frac{M_A}{i} = \frac{2400}{2}$ mkp

Articulated shaft selected: GWB, joint size 190.55 (from Data sheet "Mittelschwere Gelenkwellen") ("Medium-heavy Articulated Shafts) of Gelenkwellenbau GmbH, Essen). Mean radius for roller mounting on the two pins of the pin cross $r \approx 95$ mm $= 0.095$ metre.

Peripheral force to be transmitted per pin $P_U = \frac{M_G}{2r} =$

$$\frac{1200}{2 \times 0.095} = 6300 \text{ kp}$$

Propulsion force to be transmitted as axial force per articulated shaft

$P_A = 2 \times V = 2 \times 2400$ kp $= 4800$ kp

Axial force to be transmitted per pin

$$P_{AZ} = \frac{P_A}{2} = \frac{4800}{2} = 2400 \text{ kp}$$

The forces P_U and P_A to be transmitted per pin are at right angles to one another. Thus the resultant load per pin is:

$$P_{res} = \sqrt{P_U^2 + P_{AZ}^2} = \sqrt{6300^2 + 2400^2} = 6750 \ kp$$

The load per pin of the pin cross of the cardan joints (P_{res} = 6750 kp) is only 450 kp, that is to say about 7%, higher than the peripheral force (P_U = 6300 kp).

This additional loading of the cardan joints of the articulated shaft amounting to about 7% can be transmitted without reinforcement by the articulated shafts normally used.

WHAT WE CLAIM IS:-

1. A tractive rail vehicle having driving axles which are movably mounted in a vehicle frame or a bogie frame and are provided with axle gearboxes which are driven by way of articulated shafts without length compensation by a drive unit mounted stationary in the vehicle or bogie frame, characterised in that the drive axles are mounted resiliently in the longitudinal direction of the vehicle between fixed stops and are fastened by the axle gearbox input shafts which are mounted without axial movement, to a fixedly mounted output shaft of the drive unit by means of the articulated shafts which in addition to the torque can be loaded in the axial direction with tensile and compressive forces.

2. A rail vehicle as claimed in claim 1, wherein the drive unit is in the form of a gearbox having a helical gear on the output shaft, the angle of inclination of the gear teeth being so directed that the axial thrust of the output shaft resulting from the tooth engagement of the gearbox output stage is directed oppositely to the axially directed propulsion forces in the articulated shafts.

3. A rail vehicle as claimed in claim 1, wherein the angle of inclination of the teeth of a gear mounted on the input shaft of each axle gear box is so directed that the axial thrust of the input shaft resulting from the tooth engagement of the axle gearbox input stage is directed oppositely to the axially directed propulsion forces in the articulated shafts.

4. A tractive rail vehicle substantially as hereindescribed with reference to and as shown in the accompanying drawings.

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1568606 COMPLETE SPECIFICATION

3 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

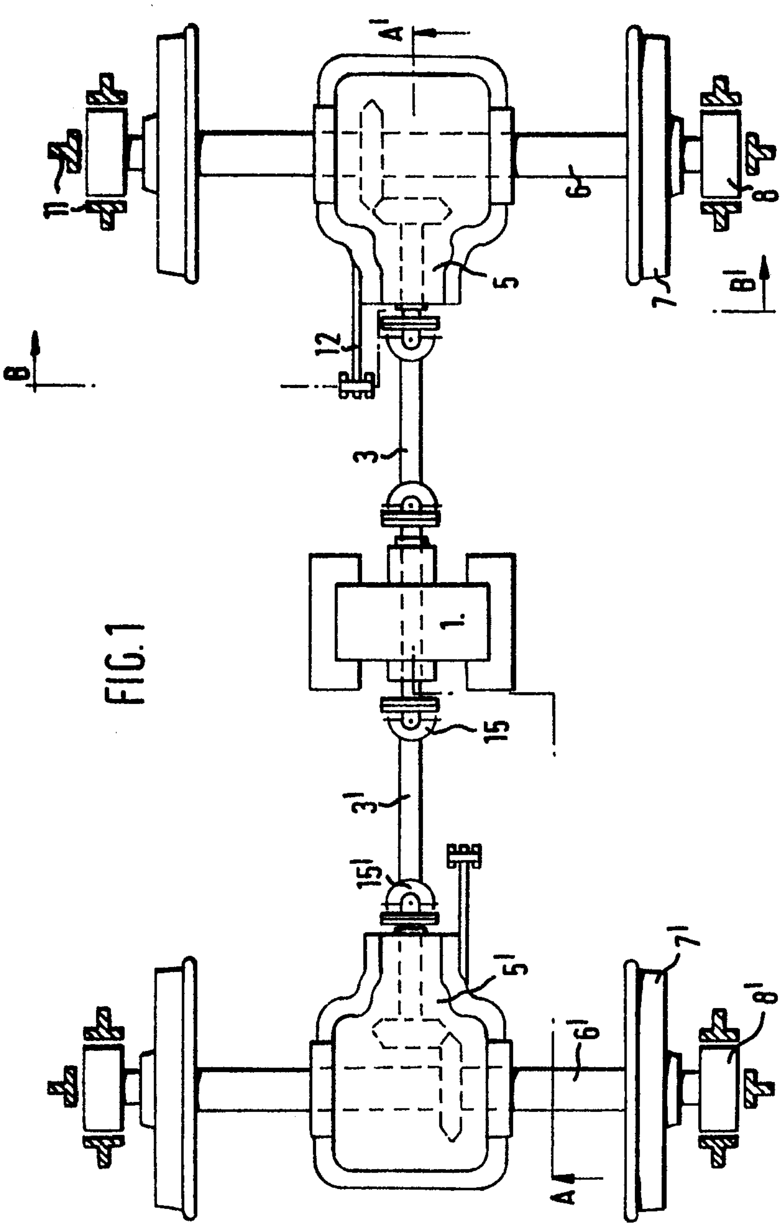

FIG.1

1568606 COMPLETE SPECIFICATION

3 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 2

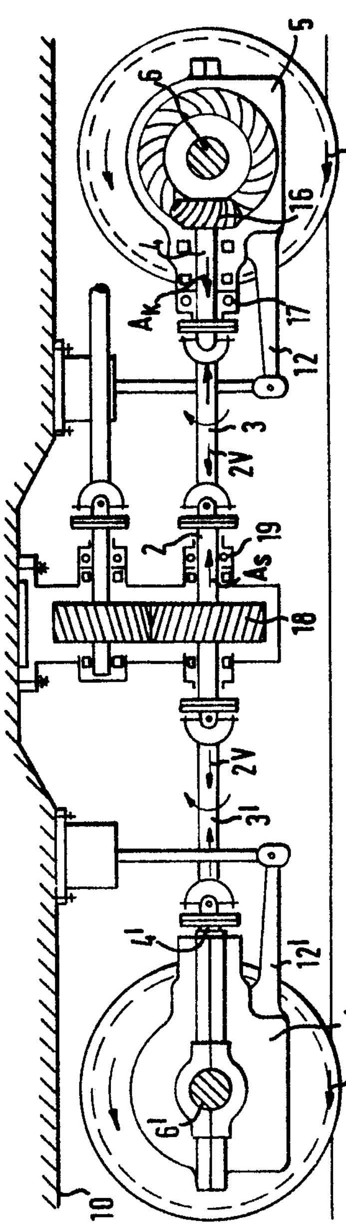

FIG. 2

FIG.3