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Landing Gear with Twin Tandem Wheel Units:

Cornering Characteristics as Determined by Model Tests

By

J. W. Blinkhorn, B.Sc., A.F.R.Ae.S.

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COMMUNICATED BY THE PRINCIPAL DIRECTOR OF SCIENTIFIC RESEARCH (AIR),
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Summary.—For twin tandem units the wheel loading conditions which arise when aircraft are turned on the ground may be critical for the landing gear. To estimate the magnitude of these loads, cornering tests were made on a small-scale model of the main undercarriage unit proposed for the *Brabazon I*, Mk. II. These tests showed that for zero turning radius, *i.e.*, turning about the central vertical axis of the model undercarriage, the wheel side loads were almost equal to the vertical load multiplied by the coefficient of sliding friction between the tyres and the ground. Tne side loads rapidly decreased as the turning radius increased, and with the turning radius equal to three times the wheel base, the wheel side loads were only about half of those at zero turning radius. The severity of the design loads for turning on the ground will therefore be considerably reduced if it can be ensured that the centre of the minimum turning circle of the aircraft is a short distance outboard of either main undercarriage unit.

1. *Introduction.*—On larger British aircraft of the near future, considerations of runway strength and undercarriage design will lead to the use of multi-wheels fitted to each unit, *e.g.* a twin tandem arrangement for main units. For the twin tandem arrangement in particular, this introduces special wheel ground loading conditions when the aircraft is turning on the ground. With this arrangement it is desirable to determine the side loads acting at the four tyres when the aircraft is turning in circles of various radii, in order to help in formulating appropriate design requirements.

This note covers a preliminary investigation on a small-scale twin tandem model and the results have been used to provide data for the tests to be made by the Air Ministry Works Dept. on a trolley giving a full-scale representation of the twin tandem units proposed for the *Brabazon I*, Mk. II.

2. *Range of Investigation.*—The torque on the unit and the equivalent side load at each tyre were determined on a small-scale twin tandem model constrained to move in turning circles of various radii. The equivalent side load at each tyre is defined as the total moment on the model divided by the product of half the wheel base and the number of tyres on the model, *i.e.*, by $5 \cdot 75 \times 4$ in.

The tests were made in the laboratory on a concrete floor having a coefficient of sliding friction with the tyres of about $0 \cdot 7$, and the tyres were of normal shape with smooth treads. Compared with the proposed twin tandem arrangement for the *Brabazon I*, Mk. II, the model scale was about $1/45$ for weight and $1/4 \cdot 8$ for linear dimensions.

* R.A.E. Tech. Note Mech. Eng. 18, received 19th October, 1948.

3. *Details of Tests*.—Particulars of the small-scale model and the proposed twin tandem arrangement for the *Brabazon I*, Mk. II aircraft are shown in the Table below.

	Small-scale Model	Proposed Brabazon I, Mk. II	Approximate Scale of Model
Static load (lb)	2,960	133,000 approx.	1/45
Wheel track (in.)	13·5	64·0	1/4·8
Wheel base (in.)	11·5	54·5	1/4·8
Type of tyre	Normal, smooth tread	Squat, smooth tread	
Tyre size (in.)	10 × 3 − 4	48 × 24·5 − 31	1/4·8
Tyre pressure (p.s.i.)	70	91	

In order to obtain some results as quickly as possible, the model was built around four redundant *Spitfire* tail wheels, which were readily available.

The general arrangement of the test apparatus is shown in the photographs of the model in Figs. 1, 2 and 3 and diagrammatically in Fig. 4. This arrangement ensured that the radius arms, and the forward towing load as measured by the spring balance, were maintained in the plane containing the wheel axles. For the radius arms, both tubes and cables were used in turn followed by a few tests with no radius arms. For these tests, tubes were found to be more practicable and convenient to use, but cables were also used because it was considered that they would be necessary for the full-scale *Brabazon I*, Mk. II trolley tests. Fig. 4 shows diagrammatically the loading system on the model, neglecting the moment exerted by the ground on each tyre. These individual moments could not be deduced from the readings taken. Suitable arrangements were made to vary the turning radius by special end fittings at the centre of rotation. The towing force was applied as slowly and smoothly as possible by means of block and tackle and the model towed until steady conditions were obtained, *i.e.*, fairly constant towing load and radius arm loads. The initial arm lengths OA and OB were adjusted initially so that they became equal to within 0·5 per cent when steady conditions were reached.

3.1. *Tubes for Radius Arms*.—The tubes were 2 in. outside diameter, fittings at the centre of rotation being arranged so that various radii of turn could be obtained by sliding the tubes and relocking them in these fittings. The loads in the tubes were determined by the deflections of calibrated helical springs located on the ends of the tubes, and measured by vernier tapes, as shown in Figs. 1, 2 and 3. The spring deflections, of course, changed the length of the arms OA and OB, but the initial arm lengths for any one radius of curvature, were easily adjusted to obtain practically equal arm lengths when steady conditions were reached.

For various turning radii OG the towing force P_1 was slowly and smoothly increased until the spring deflections were approximately constant, the model having then been pulled through an appreciable arc. For each turning radius the steady values of P_1 and tube loads were thus determined as an average of several runs, the results of each run being compared to confirm that they were consistent.

Then, referring to Fig. 4, the moment about the central vertical axis G of the trolley, due to wheel loads, is equal to $(T + C)\,d$ where T and C are the tensile and compressive loads in tubes OA and OB respectively. The equivalent side load coefficient, equal to the side load at the tyre contact of each wheel divided by the vertical load at each wheel was determined for each turning radius by dividing the total moment by the product of the total vertical load (2,960 lb) and the semi-wheel base (5.75 in.). Table 1 gives the test results obtained in this way, and Fig. 5 presents the results graphically. The moment is practically all due to side loads at the tyre contacts with the ground—the wheel drag loads due to rolling resistance will be small and the moments arising from these loads, moreover, will tend to neutralise each other.

Fig. 5 shows that as the turning radius increases from zero, the equivalent side load coefficient rapidly decreases from a value rather less than the coefficient of sliding friction between the tyre and the ground (0·7) and the rate of decrease is progressively reduced. For large turning radii, a relatively considerable change in radius is required to produce an appreciable change in equivalent side load coefficient.

Fig. 6 shows the variation of the equivalent side-load coefficient on the complete model, with the cornering angle on the inner wheels, where the latter is defined as the angle between the axle and a line from the axle centre to the centre of rotation of the model, both the axle and the line lying in the same horizontal plane. The cornering angle on the outer wheels will always be a little smaller than that on the inner wheels.

3.2. *Cables for Radius Arms.*—The tubes and springs of the scheme described in Section 3.1 were replaced by thin cables, a spring-balance being inserted in each cable arm to measure the tension. Rollers on the floor supported the flat spring balances. The towing force P_2 was offset and inclined outwards to ensure that both cables were in tension. With a procedure similar to that of section 3.1, for steady conditions, the moment about the central vertical axis G of the model is

$$= (T_2 - T_1) \, . \, d + P_2 \, . \, y = (T_2 - T_1) \, d + P_2 \, (z \, . \cos \alpha - L \, . \sin \alpha)$$

(*see* Fig. 4), where T_2 and T_1 are the tensions in cables OA and OB respectively.

Table 1 gives the results of a few tests made in this way, and, as shown in Fig. 5, they agree quite well with the results obtained when using tubes for radius arms.

3.3. *No Radius Arms.*—It is apparent from Fig. 4 that for any offset b there will be a towing force P_3 which will cause the model to rotate at constant radius without constraint by radius arms. The moment about the central vertical axis G of the model is then equal to $P_3 b$. However, it was found very difficult to obtain accurate results by this method, which was most tedious. The model overrode the towing gear, although the towing load was applied as gradually and as smoothly as possible. An example of the results obtained is that with $P_2 = 90$ lb and $b = 48$ in., the turning radius was fairly constant at about 68 in. This moment of 4,320 lb in. compared with 3,800 lb in. from the curve in Fig. 5. This excess of moment is expected, because with no radius arms the model moved in slight jerks due to over-riding the towing gear, so that the measured turning radius was larger than that appropriate to the measured towing load.

4. *Discussion of Results.*—The method using tubes was more reliable and results were obtained more speedily than when cables were used. When neither tubes nor cables were used the method was almost completely impractical, and very tedious, apart from being inaccurate.

The tube and cable methods gave consistent results, and the results obtained by cables agreed quite closely with the more extensive series of results obtained with tubes.

The side loads on the tyres, and hence the moment about the vertical axis of the model, rapidly decreased as the turning radius increased from zero. The rate of decrease became relatively small for large turning radii. For zero radius of turn the equivalent side-load coefficient estimated by extrapolating the curve shown in Fig. 5 was slightly less than 0·7, which was the coefficient of sliding friction between the tyre and the ground, as determined by pulling the model sideways, *i.e.*, with no wheel rotation. This was as expected, because at no radius of turn is the cornering angle 90 deg on all four tyres, *i.e.*, there are always at least two tandem wheels rotating about their axles.

TABLE 1

Variation of Ground Moment M and Equivalent Side-Load Coefficient S with Radius of Turn

(a) *Tubes for Radius Arms*

Radius of Turn (in.)	Tensile Load in Tube OA (T lb)	Compressive Load in Tube OB (C lb)	Moment Arm (d in.)	Moment on Unit due to Horizontal Wheel Loads ($M = (T + C)\, d$ lb in.)	Equivalent Side-Load Coefficient $\left(S = \dfrac{M}{5 \cdot 75 \times 2{,}960}\right)$
21·7	260	185	17·0	7,560	0·44
27·7	275	108	17·6	6,740	0·40
44·0	254	43	17·0	5,050	0·30
62·5	176	57	16·5	3,847	0·23
82·8	161	49	16·0	3,360	0·20
111·0	158	23	15·4	2,788	0·16
154·0	43	88	15·2	1,990	0·12
201·8	28	80	14·7	1,586	0·093

(b) *Cables for Radius Arms*

Radius of Turn (in.)	Tension in Cable OA (T_1 lb)	Tension in Cable OB (T_2 lb)	Towing Force (P_2 lb)	α (deg)	z (in.)	L (in.)	d (in.)	Moment $M = (T_2 - T_1)\, d + P_2 (z \cos \alpha - L \sin \alpha)$ (lb in.)	Equivalent Side-Load Coefficient $\left(S = \dfrac{M}{5 \cdot 75 \times 2{,}960}\right)$
37·5	34	174	145	31·05	30	11·75	17·5	5,340	0·31
61·5	28	147	123	38·85	30	11·75	16·7	3,960	0·23
110·0	23	106	90	35·25	30	11·75	15·5	2,880	0·17
130	20	68	50	39·5	35	11·75	15·0	1,690	0·10

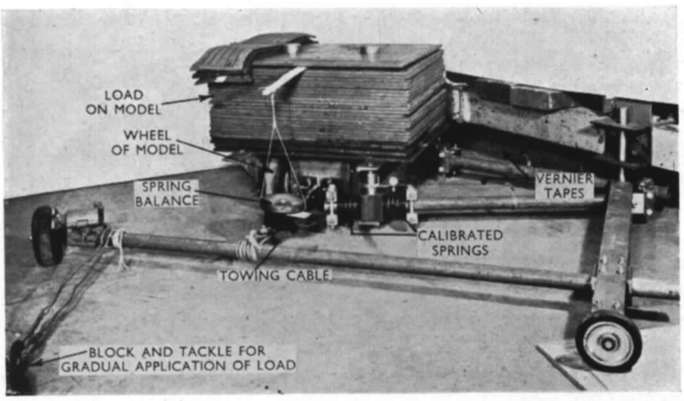

FIG. 1. Front view of model.

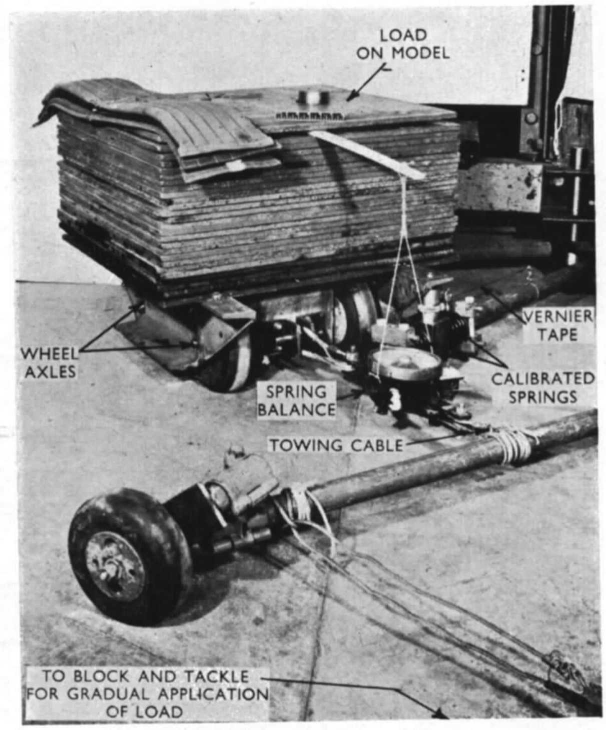

FIG. 2. Three-quarter view of model.

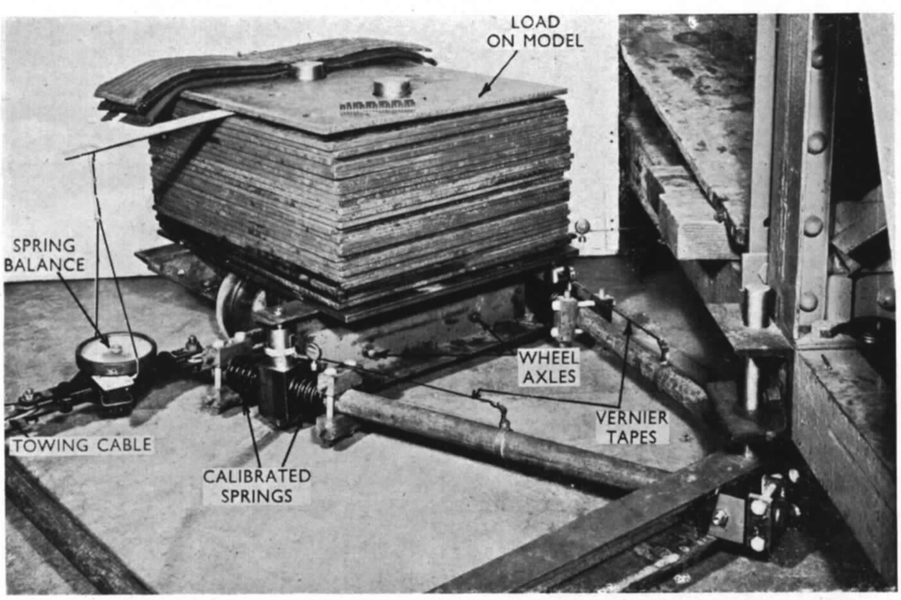

FIG. 3. Side view of model.

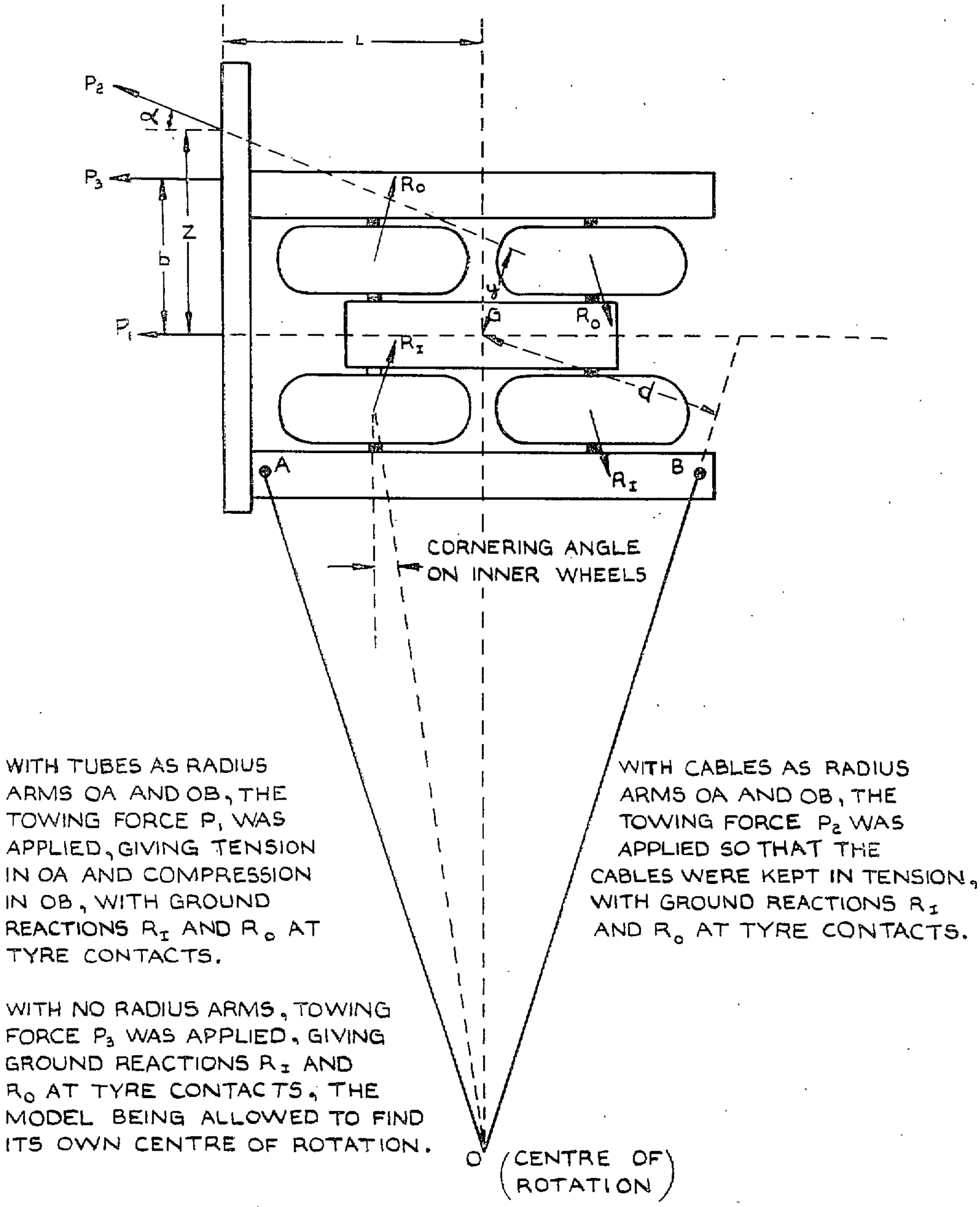

FIG. 4. Horizontal loading system on model.

(51889) Wt. 15/680 K.5 7/52 H.W.

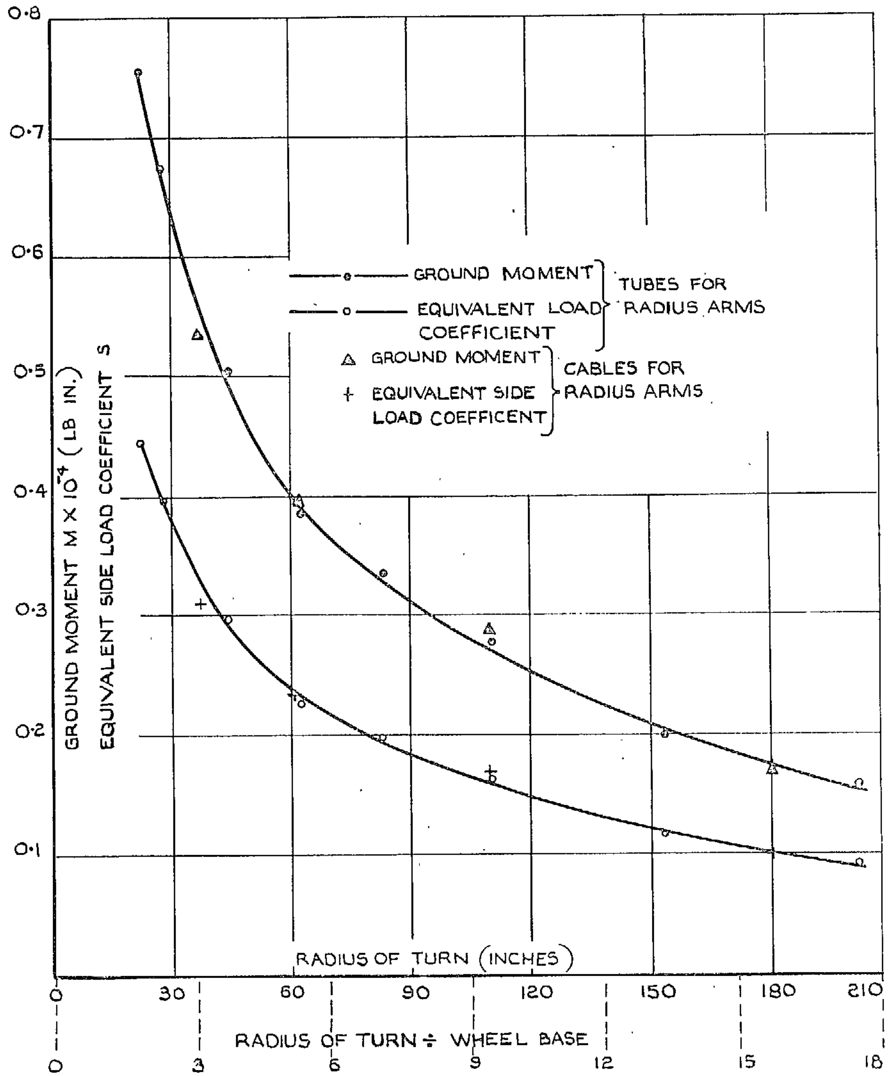

FIG. 5. Variation of ground moment and equivalent side-load coefficient with radius of turn.

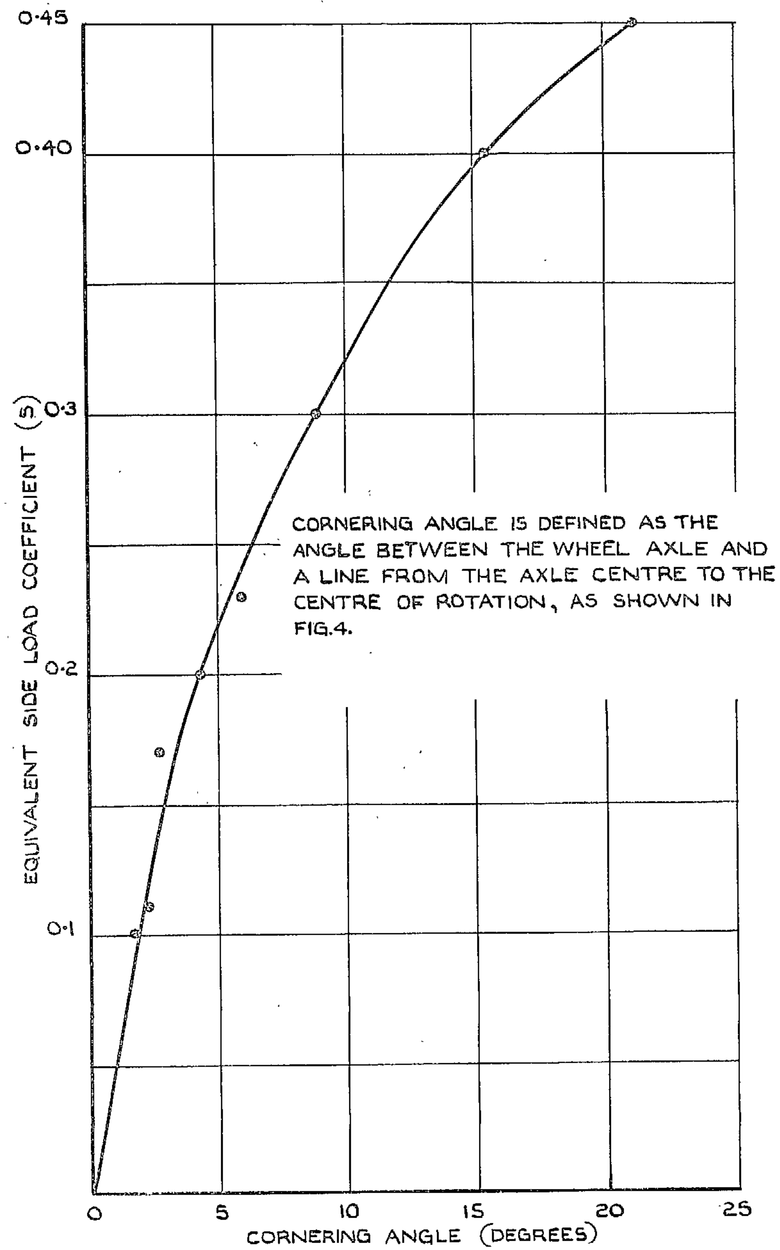

FIG. 6. Variation of equivalent side-load coefficient with cornering angle on inner wheels.

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