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DEPARTMENT OF MECHANICAL ENGINEERING AND MECHANICS
COLLEGE OF ENGINEERING AND TECHNOLOGY
OLD DOMINION UNIVERSITY
NORFOLK, VIRGINIA 23508

Old Dominion University Research Foundation

ASSESSMENT OF THE COFSI/MAST I PROJECT

By

Meng-Sang Chew, Principal Investigator

Final Report
For the period ended September 15, 1987

Prepared for the
National Aeronautics and Space Administration
Langley Research Center
Hampton, Virginia 23665

Under
Master Contract Agreement NAS1-17993
Task Authorization No. 75
Melvin H. Lucy, Technical Monitor
SED-Mechanical Systems Section

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1. INTRODUCTION

The COFS (MAST I) deployer/retractor assembly (DRA) which haves a cluster of mechanisms that constitute the collapsible/extensible Mast, contains mechanisms/linkages that deploy and retract same.

The Mast is a flexible spatial (3D) linkage with hinges that lock into place during deployment to form a truss type structure. It is 60 meters (54 bays) long with repeating sections of two bays. Each bay has altenating diagonals. All joints are single-degree-of-freedom hinges, arranged such that the Mast does not rotate during deployment/restow and that deformation energy is minimized. Mispan hinges are incoporated in the diagonals and half of the batten members.

In the following sections, the contractors will analyze the various operational aspects and characteristics of the various mechanisms within the DRA.

2. ANALYSIS OF MECHANISMS ASSOCIATED WITH DEPLOYMENT/RESTOW AND LATCHING OF COFS I HARDWARE

There are two linkages mechanisms; one for deployment and another to assist in restow. The deployment linkage system consists primarily of a recirculating gear train and such a gear train has generally been inefficient. Furthermore, it is anticipated that the size and weight of the recirculating gear arrangement may be reduced by other approaches without sacrificing reliability.

Part of the restow mechanism is the Bell-Crank mechansim that actuates the diagonal fold-arms. This mechanism, on the other hand, should be designed very carefully because it is inherently a structure that works as a mechanism due to "special dimension." It is anticipated that the hostile environment of space will preclude such a set of "special dimensions" from maintaining its dimensional integrity.

3. ASSESSMENT OF EFFECTS OF DEPLOYMENT/RESTOW LOADS ON COFS I DEPLOYABLE BEAM STRESS DISTRIBUTION, ESPECIALLY IN JOINT/HINGE BOND REGIONS

During deployment, the process is rather slow so that the dynamic effects may be neglected. As the screw loads "draw" on the "A" corner bodies, the spring loads in the diagonal hinges actually causes these corner bodies to be pushed up against the screw leads, in the direction of deployment. This means that loading on the various joints and hinges is not caused by the deployment lead screws but by the high spring loadings within the diagonal hinges. In fact the lead screws actually provide that constrained deployment of the bays.

During the restow process, the diagonal fold arms act on the diagonal hinges. Again the effective loading on the entire beam is directly a function of the diagonal hinge design as well as the loading with the spring of the hinge. This points, all importantly, towards the necessity of determining accurately, the loadings within the diagonal hinge because these loads directly affect that of all of the other linkage mechanisms within the DRA.

4. VERIFICATION OF FORCES AND MOMENTS WITHIN THE DIAGONAL FOLD HINGE

A force and displacement analysis of the diagonal fold hinge is simulated using the Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software. Figure 1 below shows a model of the diagonal fold hinge.

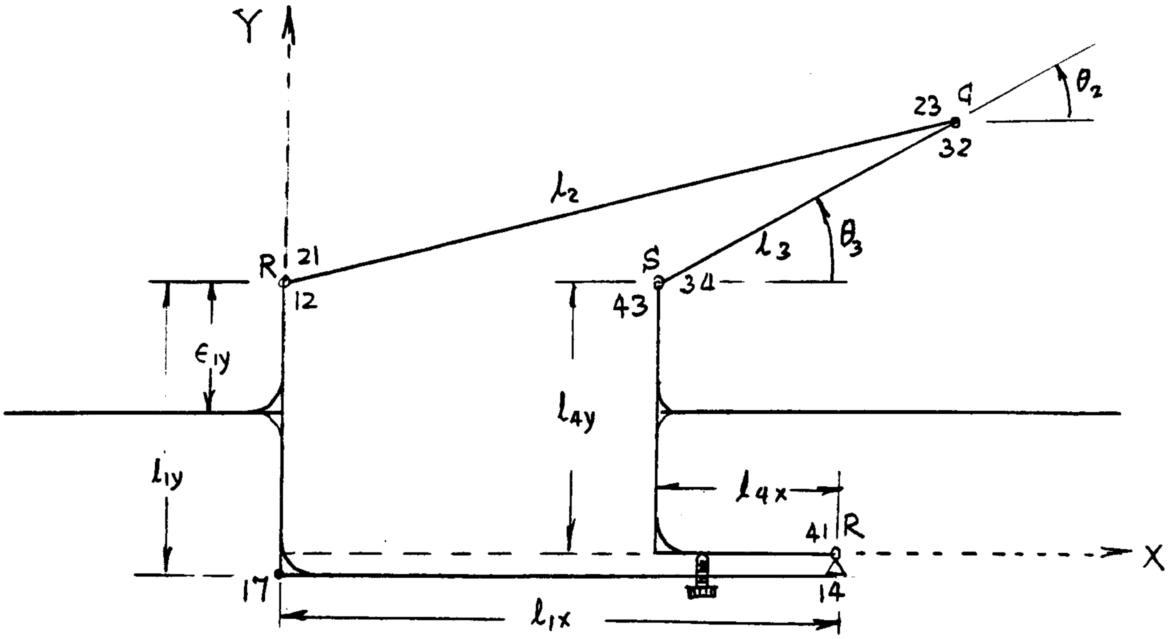

Figure 1

Based on the data obtained from blue prints of the dimensions and characteristics of the torsion spring of the diagonal fold hinge, the results of the simulation are illustrated by graphs shown in Figures 2a through 2d.

Figure 2a relates the link orientation of links #2 and #3 to the rotation of the diagonal fold hinge. The interesting point to note is that the orientation of link #2 increases to a maximum and then decreases. Since the torsion spring is connected to this same link, the spring torque on the diagonal fold hinge should be proportional to the angular displacement of link #2 plus a constant (the preload). This is verified by the graph in Figure 2b which shows the spring torque as a function of hinge rotation. Note the exact similarity of the link orientation curve for link #2 and the spring torque curve. Note also the exact similarity of where the maximum link orientation of link #2 to the where maximum spring torque occurs.

Figure 2c shows the torque requirement needed to keep the diagonal fold hinge at a given angle of rotation. Note the tremendously high initial torque needed to start the rotation of the diagonal fold hinge.

In view of the high torque needed to start the hinge rotation, it is necessary to determine the loadings in the joints of the hinge. This is shown in Figure 2d where the initial loadings of 157.2 lb occur within hinge #23 and hinge #34. Therefore, it is crucial to design the joints of the hinge to be able to take such high loadings.

It was subsequently discovered that Mr. L. Adams of ASTRO Corporation had used a slightly different set of numbers. Based on that set of L. Adams' data, a simulation of the torque requirement is obtained. This is shown plotted in Figure 3. Note in this case the torque requirement becomes negative and this somewhat corresponds to the result obtained by Mr. L. Adams. The main difference is due to the inclusion of joint friction in Mr. L. Adams' model. A copy of his results are shown as Figure 4.

To reduce the initial high torque needed to rotate the diagonal fold hinge, a cam is incorporated within the hinge. When one segment of the cam is pushed upon, another segment of the cam acts on link #3 to assist in reducing the torque needed to rotate the hinge. A simulation (using the ADAMS software) on the torque needed to rotate the cam is shown in Figure 5. It is fascinating to note that a small initial torque of less than 4.0 lb-in is sufficient to over come the high hinge torque.

It turns out that a statical force determination of the recirculating gear train as well as the bell-crank linkage is impossible because of redundancy in the linkage structure. That is to say, there two linkage mechanisms are statically indeterminate. Because of this, no simulation is needed for these two mechanisms until they are redesigned.

5. IDENTIFICATION OF DESIGN IMPROVEMENTS

In view of the disadvantages of statical in determinancy as well as the inefficiencies inherent in recirculating gear trains, it is recommended that the bevel gear trains and the bell-crank mechanisms be redesigned.

Run A

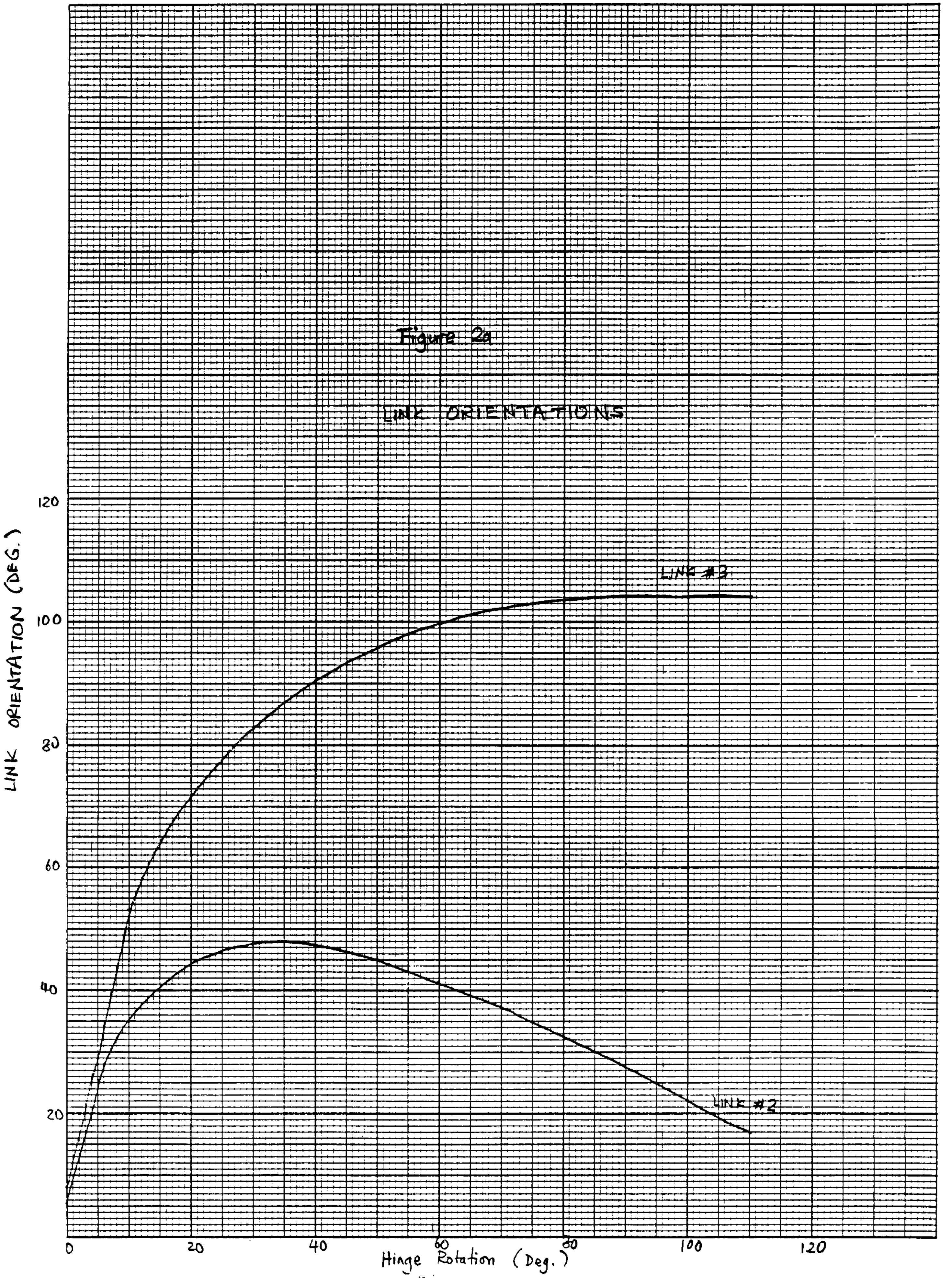

Figure 2a
LINK ORIENTATIONS
LINK #3
LINK #2
LINK ORIENTATION (DEG.)
120
100
80
60
40
20
0
20
40
60
80
100
120
Hinge Rotation (Deg.)

Run 4

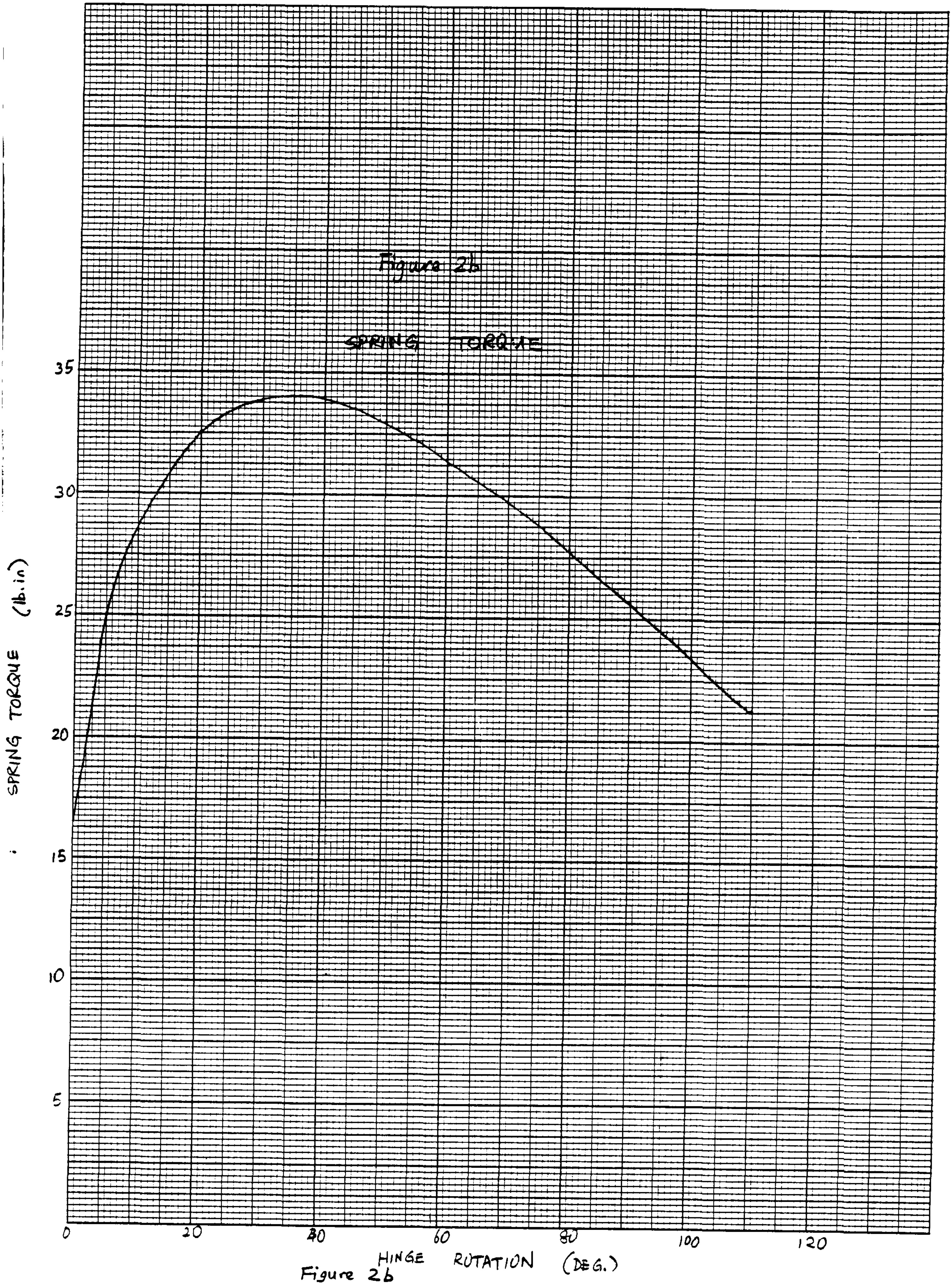

Figure 2b

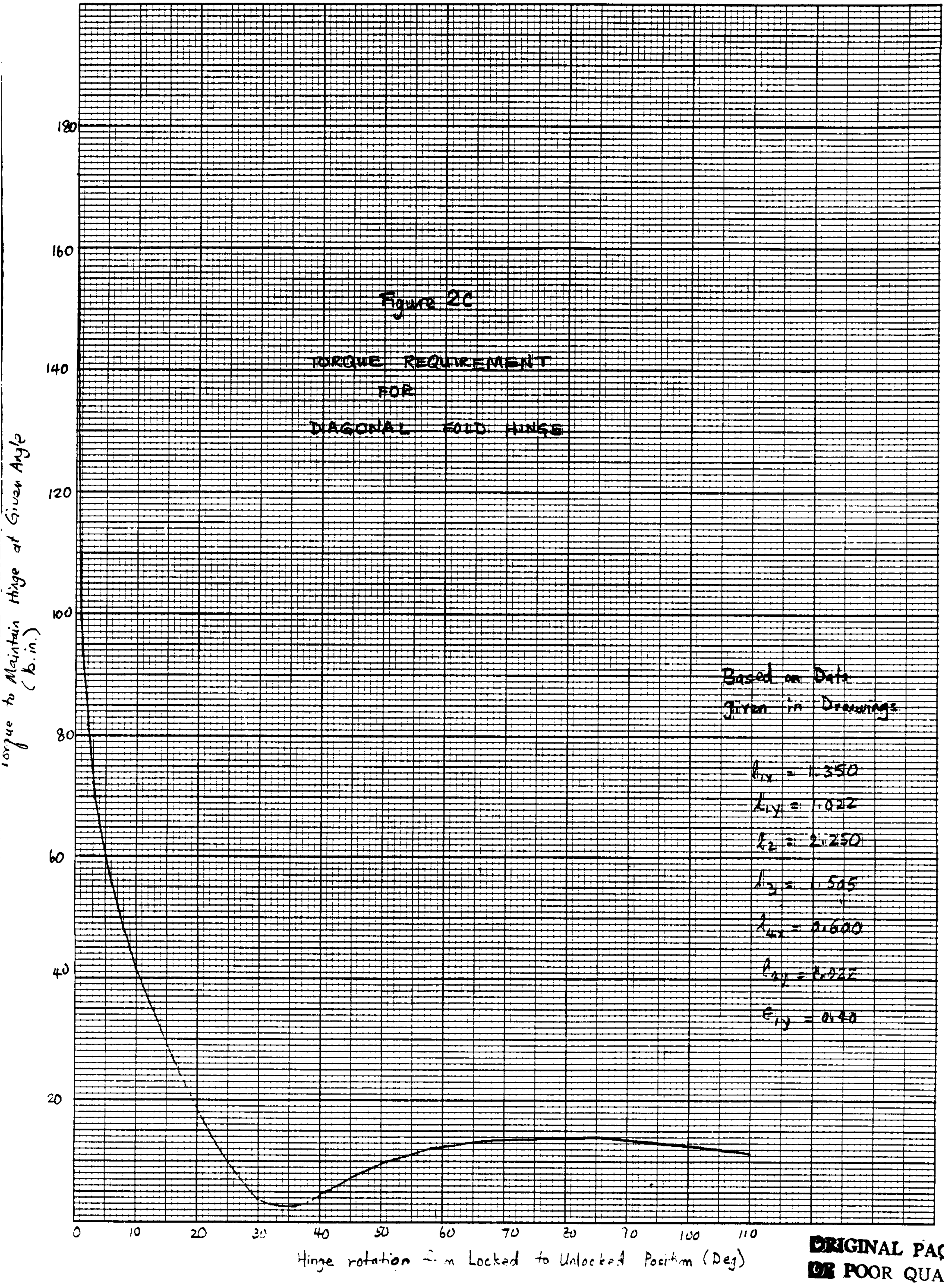
Figure 2C
TORQUE REQUIREMENT
FOR
DIAGONAL FOLD HINGE
Based on Data
Given in Drawings
l_{1x} = 1.350
l_{1y} = 1.022
l_2 = 2.250
l_3 = 1.505
l_{4x} = 0.600
l_{4y} = 1.022
e_{1y} = 0.40
Torque to Maintain Hinge at Given Angle (lb.in.)
180
160
140
120
100
80
60
40
20
0
10
20
30
40
50
60
70
80
90
100
110
Hinge rotation from Locked to Unlocked Position (Deg)

Run A

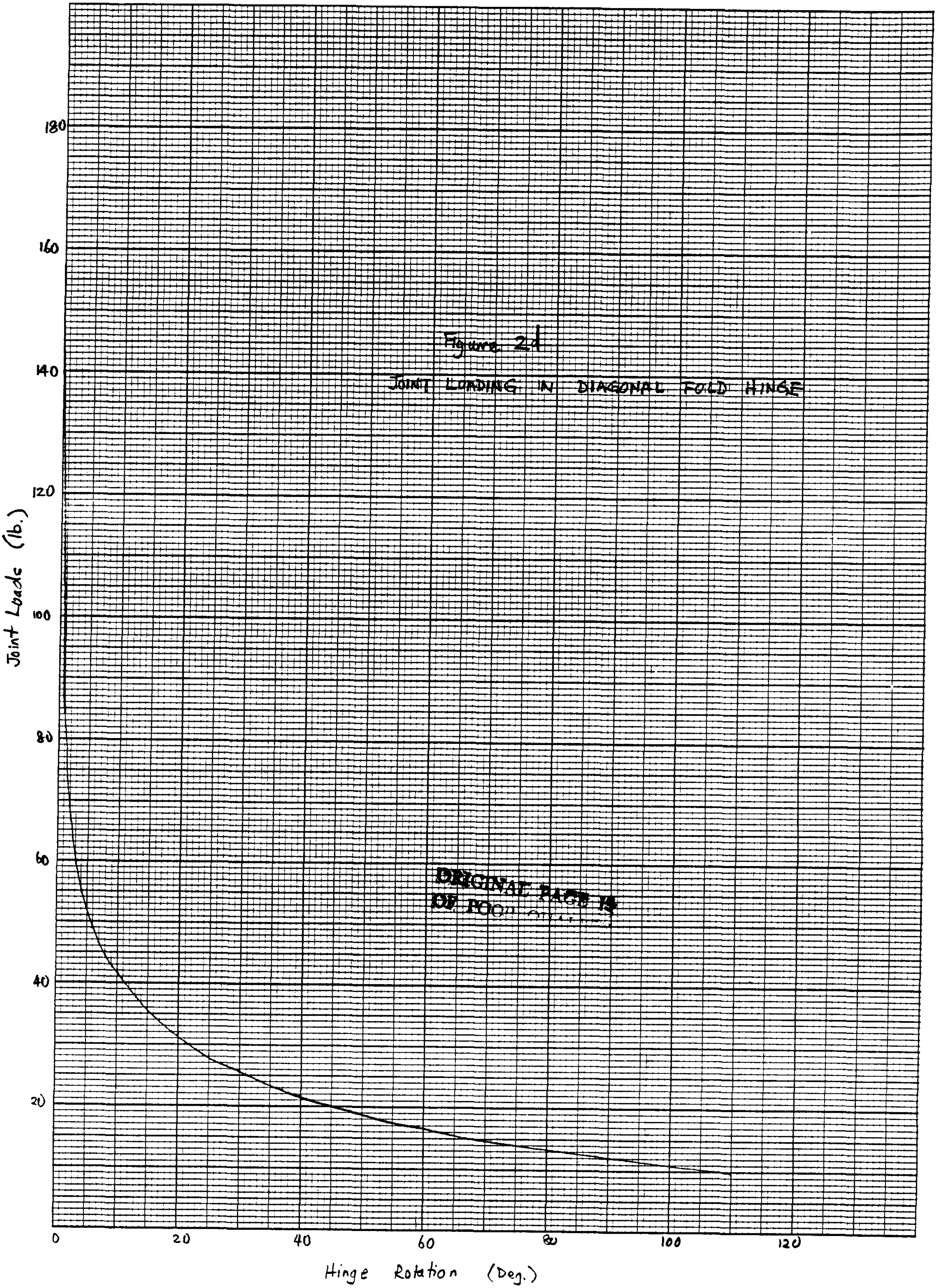

Figure 2d
JOINT LOADING IN DIAGONAL FOLD HINGE
Joint Loads (lb.)
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
Hinge Rotation (Deg.)
ORIGINAL PAGE IS
OF POOR QUALITY

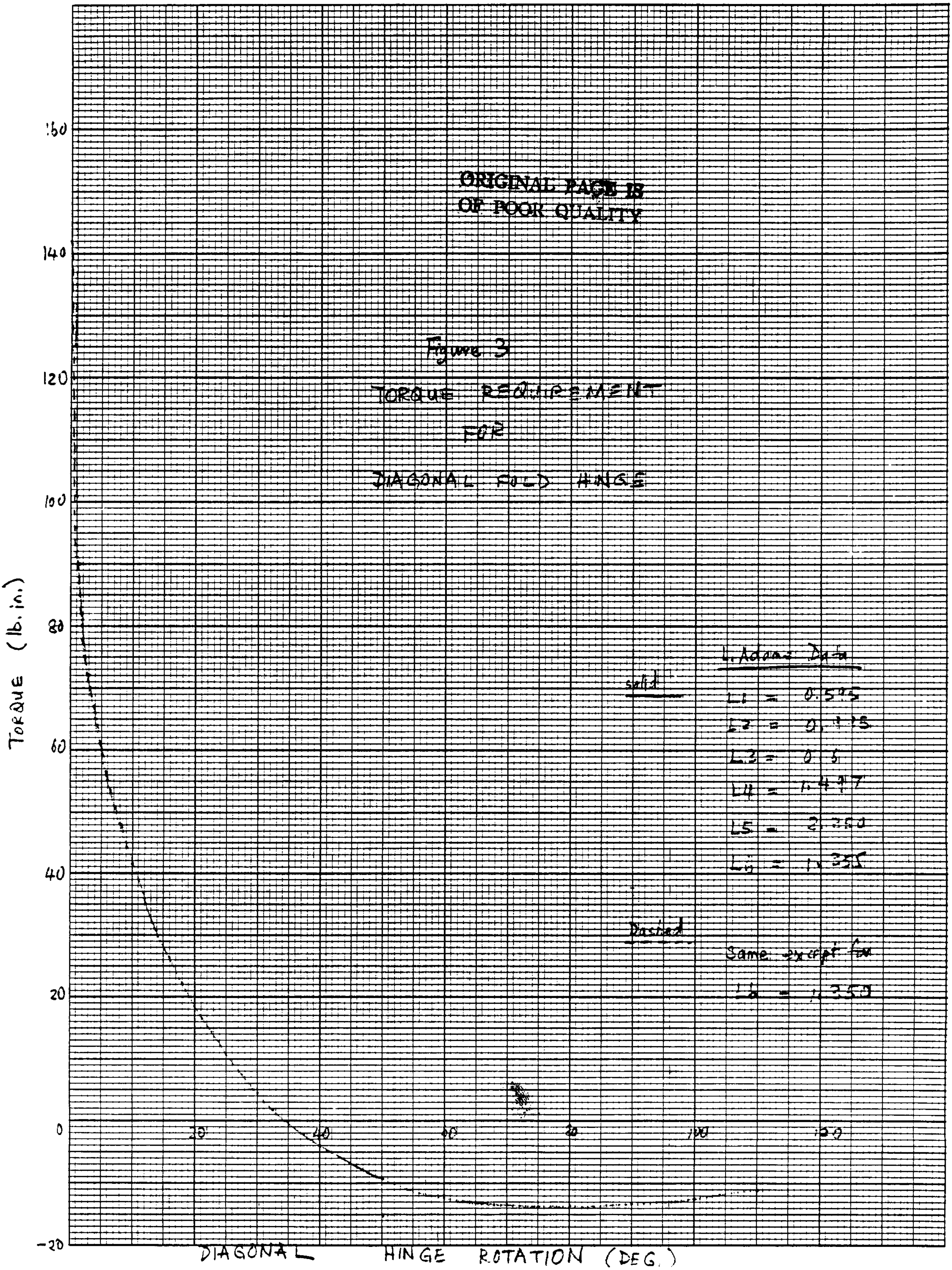

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Figure 3
TORQUE REQUIREMENT
FOR
DIAGONAL FOLD HINGE
L. Adams Data
solid
L3 = 0.5
L4 = 1.447
Dashed
Same except for
160
140
120
100
80
60
40
20
0
-20
20
40
60
80
100
120
TORQUE (lb. in.)
DIAGONAL HINGE ROTATION (DEG.)

Figure 4

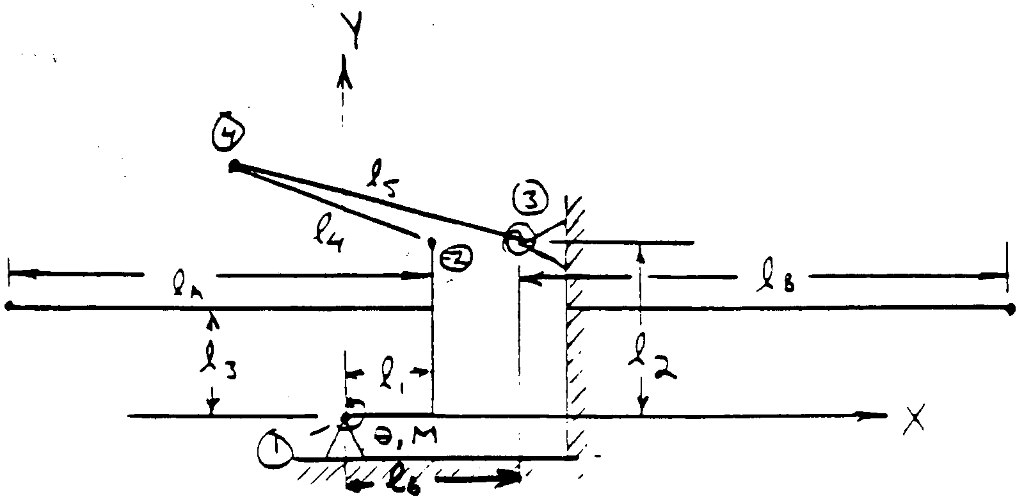
COEFFICIENTS OF FRICTION: .100 .100 .100 .100 .100
SPRING CONSTANT(in-lb/rad) = 26.000
PIN RADII(in) = .12500 .09400 .09400 .09400 .12500

TABLE 1. DIAGONAL DEPLOYMENT MOMENT
2 SPRINGS
NOMINAL PIN FRICTION

PRELOAD ANGLE (rad) = 0.6109

LA = 30.00 LB = 30.00 L1 = 0.600 L2 = 1.900 L3 = 0.600 L4 = 1.505 L5 = 2.250 L6 = 1.350

THETA(deg)	ALPHA(deg)	DELTA L(in)	TENSION (lbf)	MOMENT(in-lb)	R1(lbf)	R2(lbf)	SPRING TORQUE(in-lbf)
0.00	5.13	0.000	211.36	130.77	91.2	123.9	15.9
1.49	14.30	0.010	318.43	77.70	256.5	58.6	20.0
2.98	19.81	0.011	-326.49	65.68	376.7	57.3	22.5
4.47	24.07	0.001	-100.77	58.24	144.3	51.6	24.5
5.96	27.59	-0.020	-55.75	52.37	95.9	47.7	26.1
7.45	30.57	-0.050	-35.82	47.30	73.3	44.9	27.4
8.94	33.14	-0.091	-25.11	42.72	60.6	42.5	28.6
10.43	35.36	-0.142	-18.50	38.51	52.5	40.6	29.6
11.92	37.30	-0.203	-14.05	34.60	46.9	38.9	30.5
13.41	39.00	-0.275	-10.89	30.96	42.8	37.4	31.2
14.90	40.47	-0.357	-8.55	27.55	39.6	36.0	31.9
16.39	41.76	-0.448	-6.77	24.36	37.2	34.9	32.5
17.88	42.89	-0.551	-5.38	21.38	35.2	33.6	33.0
19.37	43.86	-0.663	-4.27	18.60	33.5	32.5	33.5
20.86	44.69	-0.785	-3.38	16.00	32.1	31.5	33.8
22.35	45.40	-0.918	-2.66	13.57	30.9	30.6	34.2
23.84	46.00	-1.060	-2.06	11.31	29.9	29.7	34.4
25.32	46.50	-1.213	-1.57	9.19	28.9	28.9	34.7
26.81	46.90	-1.376	-1.16	7.23	28.1	28.1	34.8
28.30	47.22	-1.548	-0.82	5.40	27.3	27.4	35.0
29.79	47.46	-1.731	-0.53	3.69	26.6	26.7	35.1
31.28	47.62	-1.924	-0.29	2.11	25.9	26.0	35.2
32.77	47.72	-2.127	-0.08	0.64	25.3	25.3	35.2
34.26	47.76	-2.339	0.09	-0.73	24.7	24.7	35.2
35.75	47.74	-2.561	0.24	-2.00	24.2	24.1	35.2
37.24	47.66	-2.794	0.36	-3.21	24.0	23.9	35.2
38.73	47.54	-3.036	0.47	-4.32	23.5	23.3	35.1
40.22	47.37	-3.287	0.56	-5.33	23.0	22.8	35.1
41.71	47.16	-3.549	0.63	-6.27	22.5	22.2	35.0
43.20	46.90	-3.820	0.70	-7.14	22.0	21.8	34.8



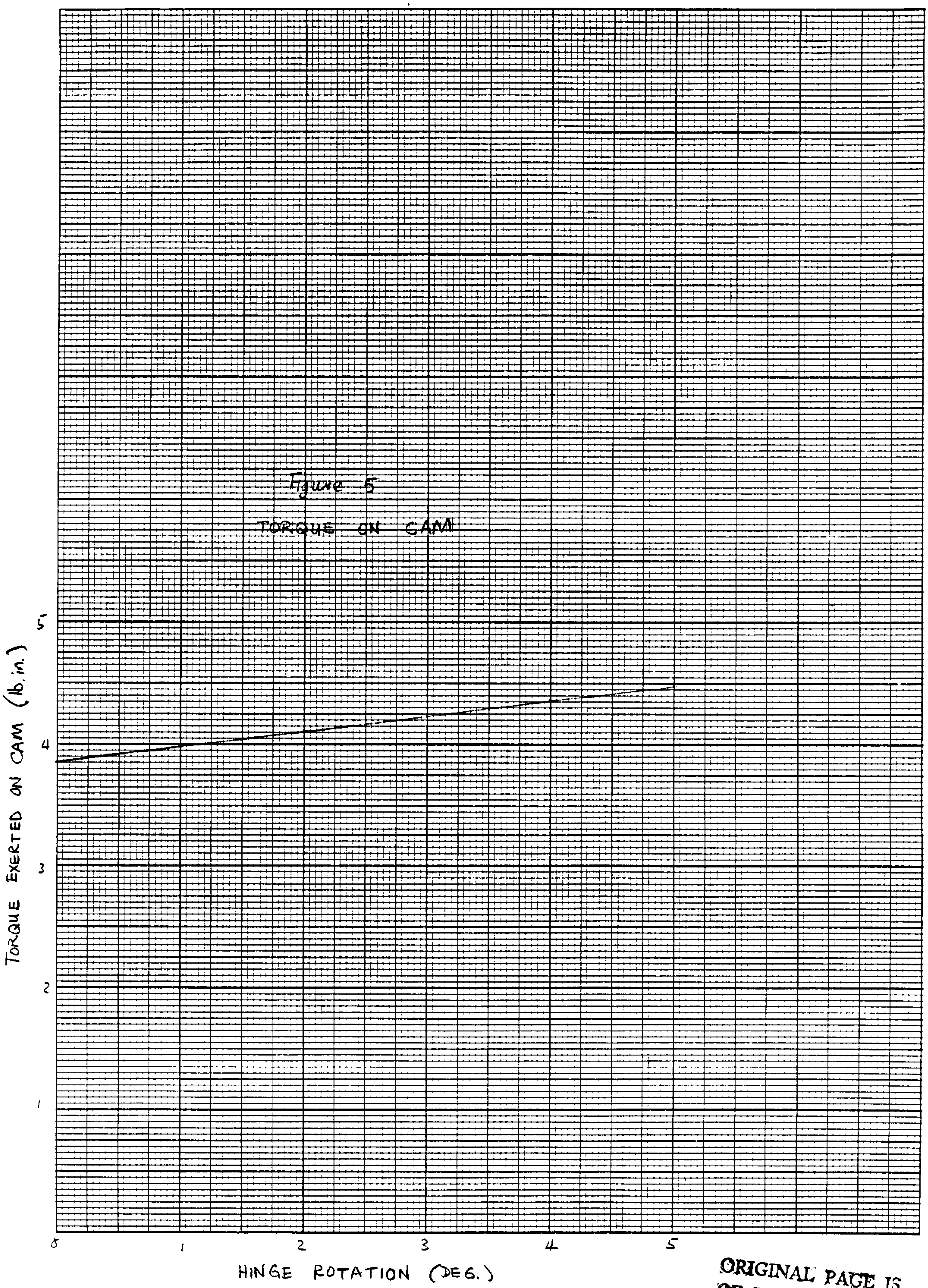


Figure 5
TORQUE ON CAM
TORQUE EXERTED ON CAM (lb.in.)
1
2
3
4
5
0
1
2
3
4
5
HINGE ROTATION (DEG.)