MECHANICAL SUPPORT DEPARTMENT
ENGINEERING NOTE

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TITLE: Vacuum Window and Related Systems Design Note and Analysis

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KEY WORDS: KTeV Vacuum Window, Sailcloth Vacuum Window, Vacuum Window Failure Analysis, Vacuum Window Safety Barrier

ABSTRACT/SUMMARY:

This document is a compilation of several engineering design notes prepared by the various authors listed above. These engineering design notes were prepared during the design of the 71 inch diameter Kevlar sailcloth / mylar vacuum window for KTeV (E799II and E832) and the related components. This compilation has been assembled to succinctly demonstrate the design of the thin vacuum window conforms to the required standards, such as FESHM 5033, as stated in the KTeV Hazard Assessment Document dated 13 April 1994. Additionally, this compilation includes applicable sections of FESHM 5033 and the documents referenced therein so that all relevant information is contained herein.

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EXECUTIVE SUMMARY

The ES&H design criterion for vacuum windows is addressed in the Design Reference Data section of FESHM chapter 5033-TA which specifically references Mechanical Safety Subcommittee Guidelines for the Design of Thin Windows at Fermilab, TM-1380 (SCN 0121.585). This TM (latest revision dated March 1993) provides General Guidelines which state that the allowable stress shall be the most stringent of $S = 0.5\ F_u$ or $S = 0.9\ F_y$, where S= the allowable stress (psi), F_u = the ultimate tensile strength (psi), and F_y = the yield strength or stress to produce 5% elongation (psi). These ratios are calculated based on the theoretical material properties, not necessarily the actual material properties. No portions of FESHM 5033 other than TM 1380 explicitly address criterion for the design of vacuum windows.

To meet the criterion set forth in FESHM 5033, the following work has been performed:

1) Andrew Szymulanski and Ron Currier (retired) of RD/MSD/MSG designed the 71 inch vacuum window and the vacuum window test fixture. This effort drew upon existing designed used on previous 48 inch vacuum windows for E731, E773 and E799I. See the calculation in Section 1. Ang Lee of RD/MSD/EAG provided finite element analysis of the window material stresses and deflections in a report dated 13 August 1992. Refer to Section 2.

2) Results of these calculations were reviewed by the then existing KTeV Mechanical Safety Review Panel chaired by Joel Misek. A letter sent by Joel Misek to Roger Dixon on 19 January 1993 included the statement that "the window design appears to be consistant with existing window design and the upcoming test program should provide assurances to its safety and design limitations". See Appendix 4.

3) Fred Renken, a coop student in RD/MSD, performed tensile testing on Kevlar material samples from the roll destined to be used to fabricated KTeV vacuum window. These tests provided a qualitative measure of the clamping effectiveness and provided ultimate tensile strength values. See Section 3.

4) A window assembly procedure was drafted by Dave Erickson of RD/MSD/MSG and is in Section 5.0, Part A. (See Appendix 5 for the final version). Procedures for the test program were written by Andrew Szymulanski, and Ron Currier (retired) and reviewed by the KTeV ES&H/QA Review Panel chaired by Wes Smart (which superseded the KTeV Mechanical Safety Review Panel). A letter sent by Joel Misek and Wes Smart to Roger Dixon on 21 March 1994, (included in Section 6) concluded "sound engineering practices have been employed" and indicated that the Fermilab Pressure Testing Permit procedure would be followed. The letter indicated that one permit would suffice for all vacuum testing and a second permit would cover the hydrostatic tests. The letter concluded by recommending approval of the testing program. Roger Dixon approved it on 22 March 1994.

5) A puncture test was conducted on the first window on Thursday 21 April 1994 in MAB. See Section 5, part C. The local puncture propagated across the full diameter of the window. Ang Lee analyzed the failure in a

report dated 13 May 1994 to understand the failure mode. Conclusions reached in this analysis provide an understanding of the puncture process. See Section 4.

6) Four hydrostatic tests (with the appropriate Fermilab Pressure Testing Permits) were performed in MAB with failure occurring at ultimate pressures of 28, 24, 28, and 28 psig. The low value was recorded for a window fabricated by a different pair of technicians and is not considered to be representative of the actual ultimate pressure of the window. See Section 5.0 Part B

7) A long term creep test of a sixth window was initiated in TSB during November 1994. Results indicate that the initial increase in the deflection was approximately 0.1 inches during the first week with decreasing additional deflection increases measured thereafter. The test continued until May 1995 and additional measurements indicate that a creep related failure is not probable. Refer to Section 6.0

Results of the above described processes indicate the following:

1) The *calculated* ratio of design stress (F_d) to ultimate tensile strength is $F_d/F_u = 0.5$. Because of the very high Young's modulus for Kevlar, this value rather than the ratio of $F_d/F_y = 0.9$ sets the allowable stress. This result is in accordance with the FESHM criteria. However, the examples cited in TM-1380(SCN 0121.585) do not include a mylar / Kevlar composite design. Therefore, finite element techniques were used to determine the calculated design stress in the Kevlar material.

   This design is very close to the minimum allowable ultimate tensile strength. Approaching the minimum allowable ultimate tensile strength is appropriate because of the detrimental effects material in the window has on multiple scattering in the experiments and because window material thickness are not infinitely adjustable as metal sheet thickness are. It must be remembered that all of these fabrics are "step function" materials. The next thicker or stronger weave will have significantly more material because the factory does not offer material 5% stronger.

2) The *measured* ratio of the actual stress (F_a) to ultimate tensile strength (F_u), $F_a/F_u = 0.5$ for the three hydrostatically tested windows believed to be representative of the one to be used in the KTeV experimental hall. This result is in accordance with the FESHM criteria. The calculation is as follows: The stress is proportional to the applied pressure and the ratio F_a/F_u can be replaced by P_a/P_u where P_a is the actual pressure in service and P_u is the ultimate pressure at failure. Per the US. Standard Atmosphere Data, the atmospheric pressure at 734'-0 elevation is 14.29 psia. The average ultimate pressure at failure for the three representative windows was 28 psig. In both cases, these values also represent the differential pressure across the window. Substituting the pressure values and solving for the ratio gives $F_a/F_u = P_a/P_u = 14.3/28 = 0.5$.

   This value is based on a *measured* ratio, not a theoretical value. Potential errors due to differences from batch to batch variations in material ultimate strength, errors due to stress risers omitted from the analysis, and variations from the design conditions in the applied load have been implicitly included in determining this ratio.

The construction and test-to-failure of four vacuum windows exceed the requirements of FESHM 5033 and the documents referenced therein.

3) Long term creep tests, which far exceed any written FESHM requirement, have been performed. The results indicate that 50% of ultimate load measured on the window is sufficiently low to prevent creep related failure. Specifically, these test results show that the creep-time curve is safely within the second stage of the curve (secondary creep) for the loads and durations the window will encounter in service. From this curve the expected window lifetime will be extrapolated. It is this lifetime compared to the required lifetime of the window used in the experiment that is the most relevant way to quantify the safety factor of the window.

The creep test data presented here is based on an initial creep test of six months, a period equal to the scheduled duration of the first leg of the next fixed target run. It is possible to continue the creep test indefinitely (provided a second window flange set was purchased) to experimentally determine the window lifetime. It is also possible to simply replace the window in the experiment every six months. Either course will ensure that window used in the experiment is operated within its design lifetime.

There are three possibilities which would cause the window to fail; puncture, overpressure, and creep related failure. Puncture of the window will be prevented by a vacuum window safety barrier discussed in Section 7. A doubling of the absolute atmospheric pressure, which would cause failure, is very improbable. This leaves creep related failures as the sole remaining plausible possibility. For this reason, considerable effort was expended to perform the creep tests. Conducting this test exceeds any stated requirements in FESHM or documents referenced therein but is considered necessary by the engineers responsible for designing the window.

Hazard assessment:

Since there are inherent dangers with thin vacuum windows, the KTeV project has always recognized the need for a vacuum window safety barrier. This barrier was designed by engineers in RD/MSD/MSG with the following criteria:

1) The vacuum window safety barrier has been designed for remote operation. It will be closed (placed in front of the vacuum window) whenever it is necessary for people to enter the experimental hall while the vacuum system is under vacuum.

2) The vacuum window safety barrier will be interlocked to prevent putting beam on the target while the vacuum window safety barrier is in the closed position. This is necessary to prevent the accidental loss described in section 6.5.2.iii in the KTeV Beam Systems Design Report Version 1.1 dated June 1994.

3) The vacuum window safety barrier has been designed using ASME pressure vessel allowable stresses, to take the differential pressure resulting from full vacuum on one side and atmospheric pressure on the other side. It is designed to be forced against the vacuum window flange

in the event of a window failure. A clearance is provided to allow air into the vacuum vessels, repressurizing them.

4) The primary functions of the vacuum window safety barrier are to prevent access to the front of the vacuum window (thereby eliminating the potential for accidental window puncture) and to isolate people from the pressure wave which would occur during a window failure.

5) The vacuum window safety barrier is NOT designed to protect equipment in the experimental hall should the vacuum window fail while the shield is in the open position.

The vacuum window safety barrier described above is being provided to mitigate potential hazards to personnel.

In the event of a vacuum window sudden failure occurring while the vacuum window safety barrier is in the open position, the potential equipment damage is estimated to include:

1) Drift Chamber 1 (DC-1) is located immediately downstream of the vacuum window. Its close proximity to the vacuum window is necessitated to optimize the physics. In the event the window fails while the vacuum window safety barrier is in the open position, it is reasonable to assume that all of the drift chamber wires and windows would be destroyed. Additionally, one may assume that the entire drift chamber and chamber mounted electronics is damaged beyond repair. The value of DC-1 is estimated to be $50,000. This risk is acceptable to the KTeV collaboration.

2) Helium Bag 1 (HB-1) fills the space around the vacuum window and the upstream face of DC-1. It is reasonable to assume that HB-1 would be a total loss in the event of the window suddenly failing while the vacuum window safety barrier is in the open position. The value of HB-1 is estimated to be $3000. This risk is acceptable to the KTeV collaboration.

3) Spectrometer Anti 2 (SA2) is located 6 meters downstream of the vacuum window. The support for this detector is designed to accommodate a 3600 pound lateral force at the top of the detector due to lateral loads such as earthquake and window failure. Although the net area of the SA2 frame is approximately 200 square feet, sufficiently detailed analysis has been completed to quantify the loading on the detector frame resulting from a window failure while the vacuum window safety barrier is in the open position. It is calculated that the detector frame is capable of withstanding the loading imposed by the a vacuum window failure. That analysis was prepared by Zhijing Tang of RD/MSD/EAG. Please refer to Section 8.

4) Drift Chamber 2 (DC-2) is located immediately downstream of SA2 and would be exposed to the effects of a window failure since there is a large aperture in SA2. It is reasonable to assume that all wires and windows in DC-2 would need replacement after a window failure. Since the DC-2 electronics are behind SA2, it is not believed that they would be damaged. The cost of repairing DC-2 is estimated to be $35,000. This is acceptable to the KTeV collaboration.

5) Helium Bag 2 (HB-2) fills the space between the downstream face of DC-1 and the upstream face of DC-2. It is reasonable to assume that HB-2 would be a total loss in the event of the window suddenly failing while the

vacuum window safety barrier is in the open position. The value of HB-2 is known to be $3000. This risk is acceptable to the KTeV collaboration.

6) The sudden repressurization of the vacuum vessels would cause the building pressure to theoretically drop by 0.21 psi if the building was hermetically sealed. While this is a conservative assumption, the pressure reduction is equivalent to an external pressure of approximately 30 psf on the outside of the building. This building is designed to accommodate comparable loads due to wind (30 psf minimum) and snow (25 psf). During CDR and Title I building design, the building was volumetrically larger which reduced the building pressure drop in the event of a sudden repressurization. As the building was trimmed to control costs, the effect of a sudden repressurization of the vacuum vessels on the wall and roof loads increased but were not included in the design. A large frangible panel could be added to the north facade of the building to provide make up air and reduce the static pressure to within the design values. However, such modifications to the building may not be necessary due to the very conservative assumption of a hermetically sealed building.

7) The sudden unbalanced load due to the presence of atmospheric pressure on one end of the string of vacuum vessels and the absence of it on the other will place a net force on the vacuum vessels of approximately 57,000 pounds. The puncture testing of the first vacuum window resulted in the same force being applied to the much less massive vacuum window test table, but the video tape of this test does not show evidence of the table moving, although a slight displacement of the table was noted after the test ended. One can conclude that the force existed for such a short length of time that the vacuum window test table did not have an opportunity to accelerate before the pressure wave hit the bottom of the table, thereby restoring a balanced force condition. The ratio of volume to weight for the KTeV vacuum decay region is a factor of four greater than the same ratio for the vacuum window test table and sufficient calculations. Zhijing Tang has evaluated the effect on the KTeV Vacuum Vessels in the decay region and has determined that the existing vessel supports are sufficient to restrain vacuum vessel movement during a vacuum window failure. See Section 8.

8) The sudden repressurization of the vacuum vessels would cause the diffusion pump oil to be exposed to oxygen while hot. Although a silicone based oil is being used, this may degrade the oil. Replacing the pump oil will cost $5000. This is acceptable to the KTeV collaboration.

9) The sudden repressurization of the vacuum vessels would cause the ion gauge on the vacuum vessels to burn out. A replacement gauge will cost less than $200. This is acceptable to the KTeV collaboration.

In conclusion, the Research Division's Mechanical Support Department has methodically designed a large diameter vacuum window which exceeds the requirements of FESHM. This window is based on previously used 48" diameter windows for E731, E773, and E799I and a 36" diameter vacuum window used in Brookhaven which were constructed in the same manner with the same materials. The work has been performed with the full knowledge of both the KTeV ES&H/QA Review Committee and the Research Division Office. While this window does present some hazard, personnel hazards are mitigated by the vacuum window safety barrier. The identified hazards to equipment (which can not be mitigated through use of a vacuum window safety barrier) that have been analyzed are deemed acceptable. The window is categorized as a low hazard consistant with the KTeV Safety Assessment Document.

EAR # 43
Aug 13, 1992

Finite Element Analysis for KTev Thin Window with a Diameter of 1.7 m/1.8 m

Ang Lee

This note contains an finite element analysis for a large cloth-type thin window used for the Ktev experiment. It starts with an assumption that the window material will be the same as one used in design report written by S. Sobczynski " E731/E621 Cloth Vacuum Window"[1]. Since the thickness of window is so small, two type shell elements, membrane shell (STIF 41) and axisymmetric element (STIF 51), have been used to find the maximum stress and maximum deflection. The thickness of the shell is estimated based on the equivalent membrane stiffness concept. A poisson's ratio is modified into a small number to simulate an individual fiber behavior. A calculation result for these two elements is shown in Table-1. The difference is negligible since the bending stiffness in structure is so small compared with its membrane stiffness. This result justified that using equivalent membrane stiffness to estimate shell thickness is a reasonable approach.

Table -1 Comparison of two type element

Material	Thickness	Maximum deflection	Maximum stress	element type
Kevlar	t_o	4.88"	1.57E5 psi	STIF 41
Kevlar	t_o	4.89"	1.56E5 psi	STIF 51

In order to gain more confidence regarding this finite model, a same geometry from reference 1 is taken to calculate the window deflection, and compared it with the experiment data [1] for both E621(24") and E621 (48") window as shown in Fig 1 and Fig. 2. It shows that the shell element fits experiment data very well for 24" window but cable element . For the larger window, the shell element solution is still much closer to experiment data compared with cable element again. Since the experiment measurement is sometimes depend on the initial flatness of window or how good it claims to the fixture specially for the larger window size, a calculation is done for the increment pressure ΔP vs the increment deflection ΔD in order to eliminate these effects (Fig. 3). The agreement is improved. The conclusion is that the shell element might be a better approach to simulate the window behavior.

A similar calculation is done for the window size 1.7 m and 1.8 m as shown in Fig 4 and Fig 5 for a maximum deflection and safety factor as a function of the thickness ratio. The t_o is defined as a kevlar thickness, which is the same as for E731(48")[1]. The safety factor is defined as ratio of the tensile strength of Kevlar respected to the maximum calculated stress. Results show that the window deflection decreases as its thickness increases. For the same thickness the larger window gives a bigger deflection. Also, it can be seen that the thicker window has a larger safety factor. The analysis is done by ANSYS and thickness calculation is included in Appendix

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REFERENCE:

1) "E731/E621 Cloth Vacuum Window Design Report", Stan Sobczynski, RD/Mechanical Department, Jan. 28, 1985

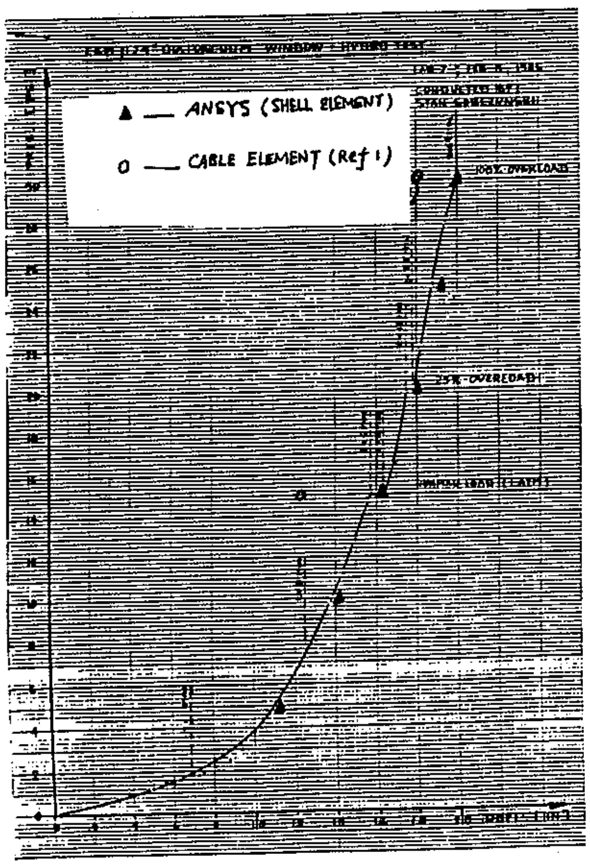


Fig. 1 Comparison for 24" window

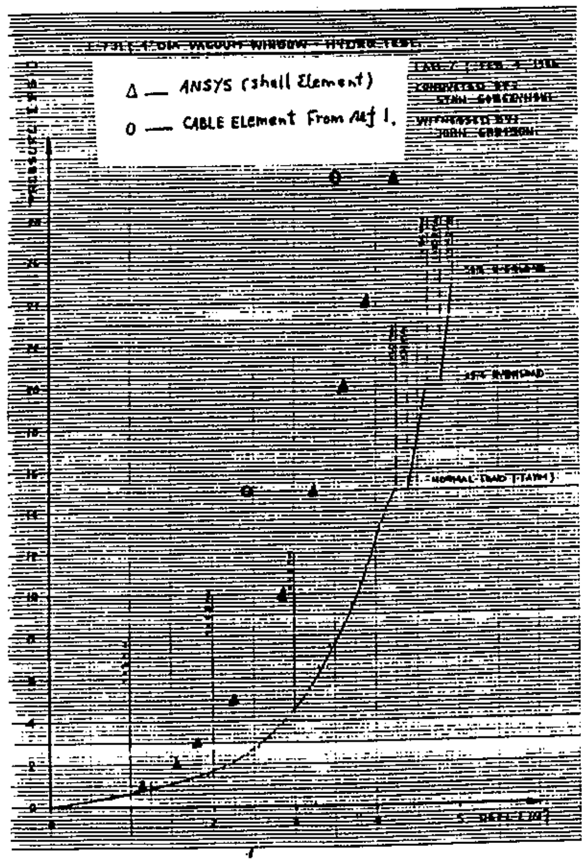


Fig. 2 Comparison for 48" window

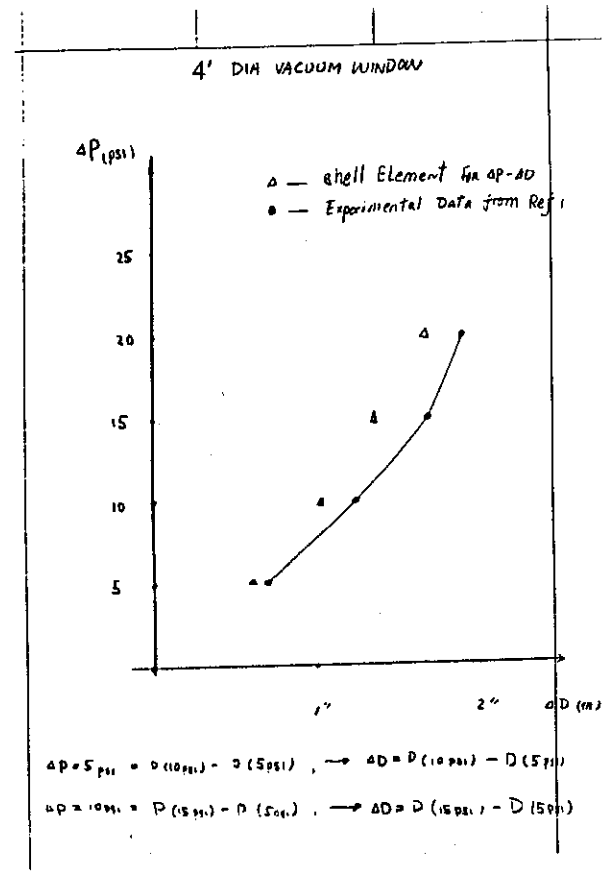


Fig. 3 Comparison for 48" window (ΔP vs ΔD)

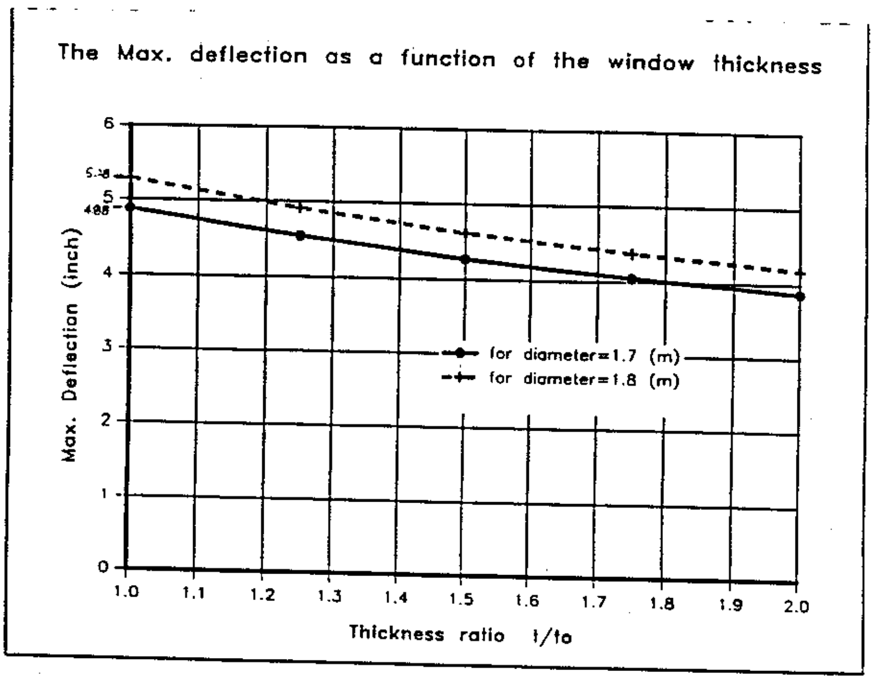


Fig. 4 The Max deflection as a function of the window thickness

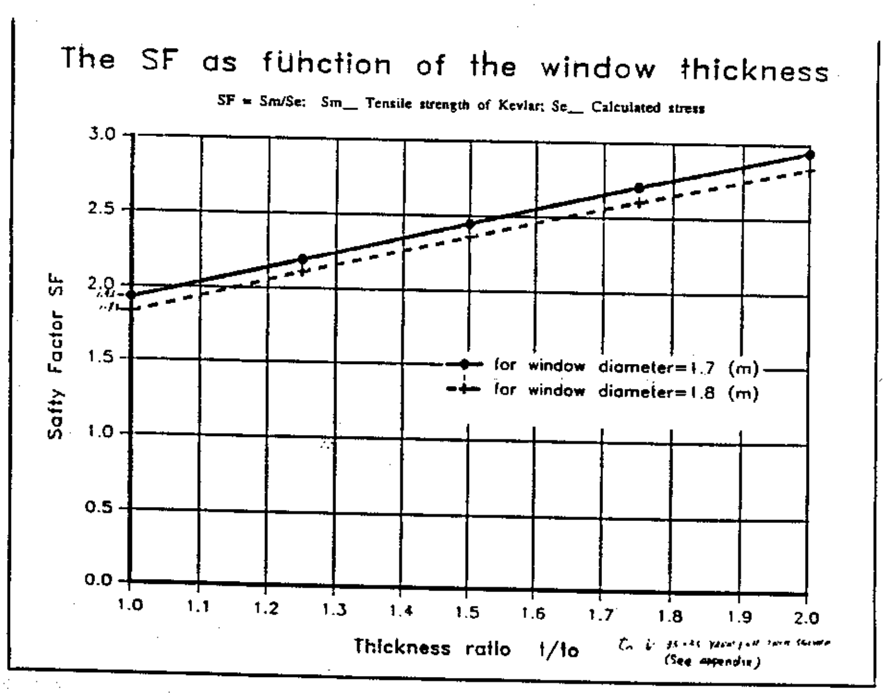
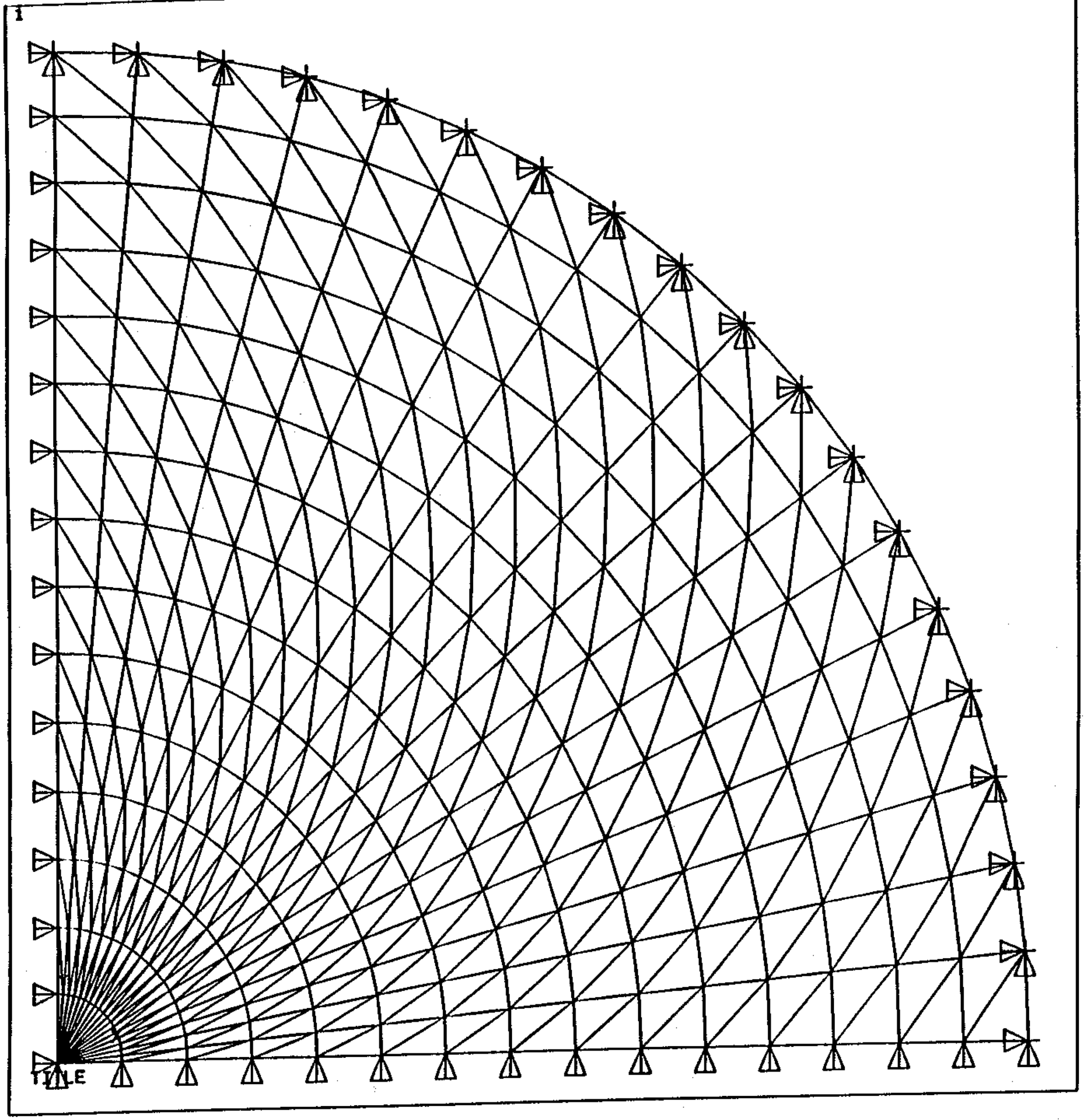


Fig. 5 The SF as a function of the window thickness



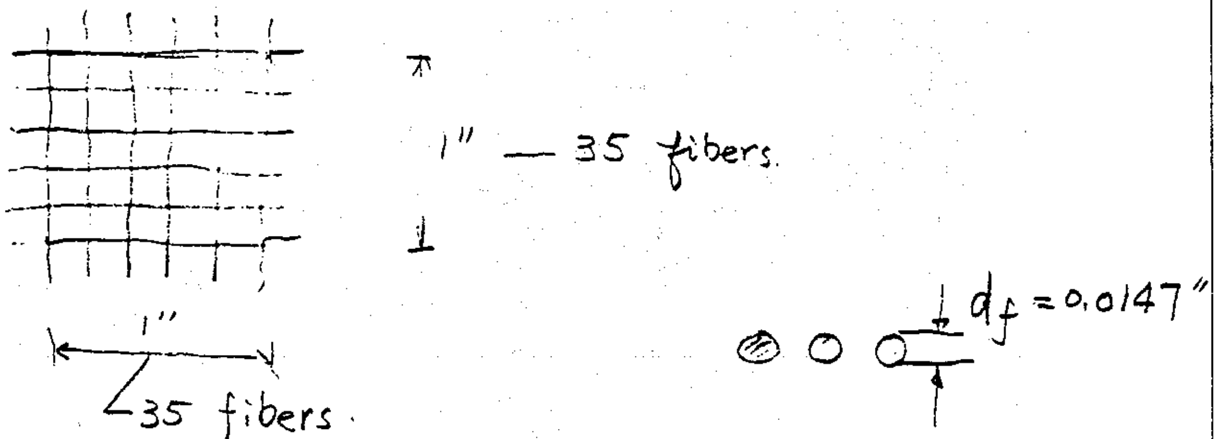
13:42:26
PLOT NO. 1
POST1 ELEMENTS
TYPE NUM
TDIS

ZV =1
DIST=0.4675
XF =0.425
YF =0.425

Fig. 6

Appendix

Shell Thickness Estimation



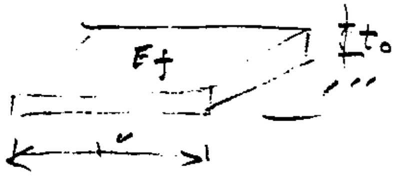
1) Cross-section area of fiber for 1'' width

$$A_f = 35 \times \frac{\pi}{4} \times (0.0147'')^2 = 5.94 \times 10^{-3} \text{ in}$$

∴ Membrane stiffness: will be

$$E_f \cdot A_f .$$

2): Assuming a plate with same material (E_f) and 1'' width, the membrane stiffness



$$E_f \cdot t_o \cdot 1$$

3): If: $E_f A_f = E_f \cdot t_o$. (Equivalent Membra

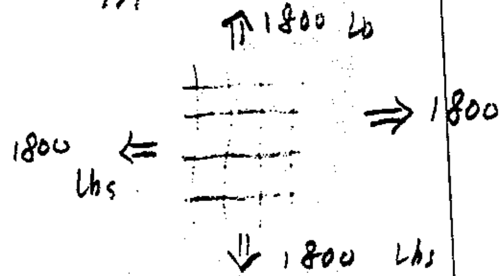
$$t_o = A_f = 5.94 \times 10^{-3}$$

Appendix

Safety Factor

1. Kevlar material has 1800 $\frac{Lbs}{in}$ × 1821 $\frac{Lbs}{in}$

Tensile strength.



∴ Max. stress of MATERIAL S_m is Define

$$S_m = \frac{1800\ Lb}{A_f} = \frac{1800\ lbs}{35 \times \frac{\pi}{4} \times 0.0147''} = 303,03$$

∴ Safety Factor (SF)

$$SF = \frac{S_m}{S_e}$$

where: S_e is maximum calculated Stres from ANSYS, $E_{kevlar} = 9.0 \times 10^9$ (PSI)

SAMPLE INPUT FILE

WIN-3W.dat

```
C**************this file for 1.7 (m) diameter window design 9/16/91
/PREP7
KAN,0
KAY,3,0
KAY,6,1
KAY,8,1
KAY,9,0
C*************kevlar 1 layer
ET,1,41,2,,,,1
EX,1,9.0e6
EY,1,9.0e6
C*************
NUXY,1,0.0001
ALPX,1,0.001
R,1,1.508e-4/25.4e-3
C*************mylar with 5 mils
ET,2,41,2,,,,1
EX,2,5.5e5
EY,2,5.5e5
ALPX,2,0.001
R,2,5*1e-3
C**************
CSYS,1
N,1
N,16,33.4645
FILL
NGEN,19,16,1,16,,,5
E,1,2,18,
EGEN,15,1,1
EGEN,18,16,1,15
E,2,19,18
EGEN,14,1,271
EGEN,18,16,271,284
C***************
TYPE,2
REAL,2
MAT,2
E,1,2,18
EGEN,15,1,523
EGEN,18,16,523,537
E,2,19,18
EGEN,14,1,793
EGEN,18,16,793,806
C**************
D,16,ALL,,, 304,16
SYMBC,,1
SYMBC,,2
CSYS,1
WSOR,Y
C**************
TREF,0
TUNIF,-1
ITER,1,1
LWRI
C*****************
EP,1,6,15,,522
ITER,-10,10
LWRI
C*****************
```

Sample Input File

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/PREP7
 \N,0
KAY,3,0
KAY,6,1
KAY,8,1
KAY,9,0
C*************kevlar 1 layer
ET,1,51
EX,1,9.0e6
EY,1,9.0e6
C*************
NUXY,1,0.0001
ALPX,1,0.001
R,1,5.94e-3*1.25
C**************mylar with 5 mils
ET,2,51
EX,2,5.5e5
EY,2,5.5e5
ALPX,2,0.001
R,2,5*1e-3
C*************
N,1.0.001
N,31,35.43
FILL
e,1,2
egen,30,1,1
C****************
 YPE,2
 EAL,2
MAT,2
E,1,2
EGEN,30,1,31
C***************
D,31,ux,0,,,,uy,uz
C***************
TREF,0
TUNIF,-1
ITER,1,1
LWRI
C******************
esel,type,1
nele
EP,all,2,15
nall
eall
ITER,-20,10
LWRI
C******************
afwrite
fini
/input,27
fini
```

Extra Attachment from Ref. 1

2. TESTING OF FABRIC AND FABRIC CLAMPING

Tensile testing of fabric-mylar laminate was done in order to correlate the theoretical and actual properties of Kevlar 29, dacron and mylar, and to test various clamping techniques.

A clamping fixture was designed so as to match closely the behavior of the actual clamping flanges, their bolt pattern, surface finish, "o-ring" groove, etc. (See drawings MD-177035, and MC-177037 in the Appendix).

All initial tests were done with dacron, and without use of the aluminum wire. Mode of failure was the premature pull-out from the clamp. Installation of the aluminum wire solved this problem and shifted the mode of failure to the fabric. Dacron-mylar test sample was failing at approximately half the tensile strength listed for this fiber. The mylar failed also, just prior to the rupture of the fabric.

Due to unsatisfactory behavior of dacron (poor strength and excessive elasticity), Kevlar 29 became the focal point for further tests.

Based on preliminary analysis, the following fabric was purchased for evaluation:

- Style 735
- MIL-C-44050 (Military Spec.)
- 1500 Denier, Kevlar 29
- 2 x 2 Basket Weave
- 35 x 34 Count
- 1800 lb. x 1821 lb. Tensile Strength
- Vendor:

  Clark-Schwebel Fiber Glass Corp.
  5 Corporate Park Drive
  P. O. Box 851C
  White Plains, New York 10603
  J. E. McAdams - Sales

Test results are outlined in the following section.

Kevlar Window Tensile Tests

Fred Renken

12/23/92

FNAL RD/MSD Thermal Systems

Kevlar Tensile test

Kevlar tensile test results				
Test Number	Strain at	Maximum		
	Failure	Load		
	(inches)	lbs force		
9	0.52	7900		
10	0.47	8100		
12	0.47	11300		
13	0.47	10650		
14	0.54	9900		
15	0.5	9010		
16	0.49	8500		
17	0.41	10210		
18	0.42	11150		
Average of tests	0.47	9525		
Length of test samples	7.964			
Percent strain at failure				
for the average of tests			5.90155701	percent
Width of sample	9.858			
Load per inch of fabric			966.220329	Lbs/in

ABSTRACT:

For the new KTev 1.8 m window tests were necessary to complete safety requirements and assure the window design would meet the necessary standards. Three different configurations were tensile tested to find the torque values necessary to meet requirements based on calculations. It was found that the use of an aluminum gasket increased the clamping capability but also increased damage to the window materials. Without the gasket the clamping fixture slipped at lower loads but held higher maximum loads. Tests with spacers proved that such assemblies would be very difficult to use. Increasing torques will increase the capability of flanges to hold the fixture.

INTRODUCTION:

On the new fixed beam target experiment KTev, a 1.8 m diameter vacuum chamber window will be constructed for experimentation. This window, although very like others built in the past, is much larger and so extra safety precautions and documentation is needed. The window is constructed of a kevlar fabric sheet for strength between layers of mylar for protection and vacuum seal.

Three different clamping configurations were tested. The first clamp was the same as the fixture used on previous windows with a soft alloy aluminum gasket to achieve a tight seal and account for tolerances in bulk head construction. The second was to test without the gasket, holding the window between only the two bulk heads. Steel inserts were placed in the test samples to simulate a flat bulkhead. In order to use this configuration much tighter tolerances in the bulk head construction will be necessary in order to achieve a vacuum tight seal. The third test was conducted with metal spacers between the flanges inside the window materials to assure bulkhead strength. These tests required the use of the aluminum gasket.

Several aspects of the window were examined in each test. First, would the clamping assembly itself damage the window materials beyond usefulness by compressive forces cutting through the materials. Second, at what load will the first slip or indication of failure occur. At what clamping force or torque would the necessary load be upheld. Most importantly, what maximum load can be sustained in each configuration. Finally, the overall damages and performance of the test samples used.

The window bulkhead flanges were simulated by preparing samples for a tensile test. The clamps were designed to closely simulate the real flanges with both window assembly bolts and through bolts to the vessel. (See drawing numbers 9220.832.ME-285684 and 9220.832.ME-285674) All bolts would be torqued evenly. For this test sample a minimum load of 6845 lbf would be necessary to meet calculations done in an ANSYS analysis of the window. (Attached) The tensile test provides a unidirectional load rather than an even multi-directional force as in the actual window. The actual window should therefore perform better than these tests would indicate.

Test Data

Test #	Test Speed	Weave	Torque	First Slip	Max Load	Al dim: see fig		Notes:
	in/min	degrees	ft*lbs	lbf	lbf	X	Y	
1	0.05	90	56	1010	2478	—	—	No Mylar used
2	0.075	90	83	2010	2810	0.061	0.188	
3	0.075	90	111	2600	3700	0.084	0.184	
4	0.075	90	139	3260	3900	0.079	0.201	
5	0.075	45	83	1200	2005	0.063	0.181	
6	0.075	45	111	1440	2200	0.082	0.202	
7	0.075	45	55	590	1940	0.045	0.187	
8	0.075	90	195	6550	8300	0.143	0.157	Past Full Scale on Graph
9	0.05	90	195	5400	7900	0.132	0.156	
10	0.05	90	225	6700	9100	0.139	0.153	
11	0.05	90	167	3020	5700	0.119	0.166	
12	0.05	90	250	7500	11300	0.15	0.152	
13	0:05	90	280	6800	10650	0.156	0.152	
14	0.05	90	250	6300	9900	0.167	0.156	
15	0.05	90	250	5100	9010	0.134	0.154	Strange curve and values
16	0.05	90	250	5500	8500	0.148	0.153	
17	0.05	90	250	7400	10210	0.144	0.152	New Bolts used
18	0.05	90	250	7050	11150	0.136	0.154	
19	0.05	45	250	3990	4510	0.157	0.15	
20	0.05	45	250	3700	4005	0.147	0.149	Displays Yield Curve
21	0.05	90	250	5005	13600	N/A	N/A	No Aluminum Used
22	0.05	90	250	5010	12950	N/A	N/A	No Aluminum Used
23	0.05	90	250	7300	12440	N/A	N/A	No Aluminum Used
24	0.05	90	250	5200	12810	N/A	N/A	No Aluminum Used
25	0.05	90	250	7050	8000	0.146	0.151	Al Spacers
26	0.05	90	250	5100	5900	0.132	0.156	Al Spacers
27	0.05	90	250	5800	6810	0.143	0.157	Steel Spacers
28	0.05	90	250	5700	7005	0.158	0.153	Steel Spacers
29	0.05	90	250	5210	6100	—	—	Used Al Spacers

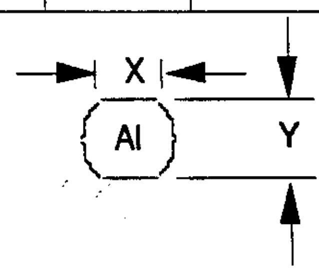


Fig 1: Aluminum Cross Section

Test Data

Test #	Test Speed	Weave	Torque	First Slip	Max Load	Al dim: see fig		Notes:
	in/min	degrees	ft*lbs	lbf	lbf	X	Y	
Al Gasket used: 250 FT*LBS Torque 4 samples								
1	0.05	90	250	7500	11300	0.15	0.152	
2	0.05	90	250	6300	9900	0.167	0.156	
3	0.05	90	250	7400	10210	0.144	0.152	New Bolts used
4	0.05	90	250	7050	11150	0.136	0.154	
Average:				7062.5	10640	0.149	0.154	
No Aluminum Gasket used.								
1	0.05	90	250	5005	13600	N/A	N/A	No Aluminum Used
2	0.05	90	250	5010	12950	N/A	N/A	No Aluminum Used
3	0.05	90	250	7300	12440	N/A	N/A	No Aluminum Used
4	0.05	90	250	5200	12810	N/A	N/A	No Aluminum Used
Average:				5628.75	12950			
Steel and Aluminum Spacers Used:								
1	0.05	90	250	7050	8000	0.146	0.151	Al Spacers
2	0.05	90	250	5800	6810	0.143	0.157	Steel Spacers
3	0.05	90	250	5700	7005	0.158	0.153	Steel Spacers
4	0.05	90	250	5210	6100	—	—	Used Al Spacers
Average:				5940	6978.75	0.149	0.154	

OBSERVATIONS:

Based on the best representative sample from each different clamping configuration.

Aluminum gasket 250 ft*lbf with 90° weave. (Test #12)

Kevlar: Failure along aluminum gasket location.
Failures due to fraying and slip from around bolt holes.

Mylar: Clearly indented along aluminum gasket. No holes or tears.

Data Curve: Smooth with only 2 slips before maximum load reached.

Aluminum gasket 250 ft*lbf with 45° weave. (Tests #19 and #20)

Kevlar: No failure by bolts or aluminum clamp.
All failure in center region 3.75 inches wide.

Mylar: Some indentation along gasket. One tear at frayed corner.

Data Curve: Apparent yield point exists present.
Very smooth line without slips or failures until maximum load reached.
The yield could not be calculated because the load takes a hyperbolic shape for which the cross sectional area is indeterminate.

No aluminum gasket 250 ft*lbf 90° weave. (test #21)

Kevlar: Failure outside clamped regions.
Failure areas lined up between bolt locations.
Fraying along edges very evident.

Mylar: Fused slightly to kevlar. Easily removed with very little damage.
No indentations except slight cloth weave pattern.

Data Curve: Multiple slips and failures before maximum load reached.

Spacers with aluminum gasket at 250 ft*lbf 90° weave. (test #25)

No significant difference in performance of aluminum spacers and steel spacers.

Kevlar: Severe failure along Al gasket. Bolt holes remain intact.
Little fraying except along edges.

Mylar: Both gasket and spacer indentations visible. No tearing, remained intact.

Spacers: Some seemingly untouched, others severely indented or bent. Aluminum spacers sustained more damage.

Data Curve: Smooth until several failures immediately before maximum load.

DISCUSSION OF DATA:

All tests had failures resulting from fraying along exposed edges. This would not happen if fabric pulled uniformly in all directions. Fraying would also be reduced with the use of epoxy as all previous windows were constructed. The actual window would be able to sustain higher loads than tensile samples.

45° weave tests formed hyperbolic shaped tension region resulting in higher fraying and indeterminable cross sectional area. Most 45° weave tests results are not helpful contributors to the data desired.

First eleven tests used to find minimal torque at which target loads of 13094 and 6845 lbf would be met or exceeded.

Widespread values of maximum loads indicate tests not completely valid. Experience and design theory should not be blatantly overridden by these results.

Data for the first slip and maximum load included to provide information on which to base safety factors. Fraying edges often the cause for first slip. At this value vacuum may be lost, in the window application, but no severe endangering failure would occur.

Tests in which spacers were used suspect because of the difficulty in assembly of test samples. The difficulty and failure to assemble good samples clearly shown by damage to the mylar and spacers. Better assembly needed than could be done with this test apparatus.

NOTE: Scales change from test to test on graphs so read test curves carefully.
NOTE: The machine could test to a maximum load of 13000 lbf so tests at higher torques were not done.

CONCLUSIONS:

Tears in mylar pieces was primarily due to slips after the maximum load failure, not from the assembly process. The damage incurred because test was taken to failure. The mylar did become permanently indented.

At 250 ft*lbf a total bolt load of 54395 lbf is exerted on the fixture. The resulting average X dimension on the aluminum gasket of 0.149 yields a compressive force of 37032 psi. This compressive force is higher than mylar's yield and ultimate strengths but failure did not occur since mylar is extremely elastic. This force is below the strength of kevlar so no damage to the cloth was induced by the assembly.

Significantly less damage to the mylar occurred on the tests without the aluminum gasket.

250 ft*lbf torque was necessary with the aluminum gasket to obtain a first slip above the desired 6845 lbf. The first slip for samples without a gasket occurred at a much lower value than either with the gasket or the desired load. Therefore the aluminum gasket significantly aided a secure hold.

The aluminum gasket was not deformed to flush with the clamping fixture and did supply the primary compressive force. This is clearly shown by an average Y of 0.154 which is greater than the maximum 0.145 depth of the slot.

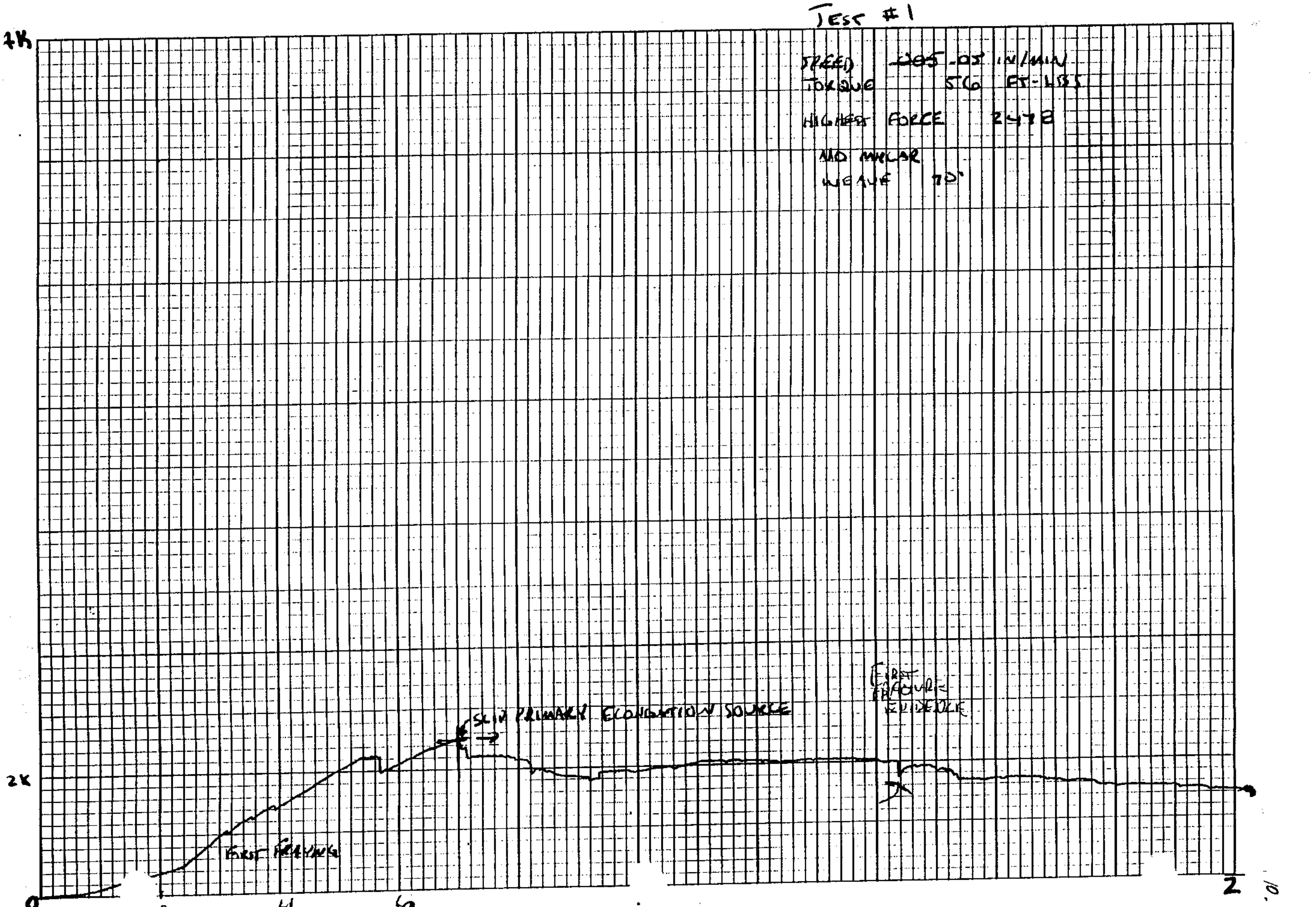
The highest maximum load at 250 ft*lbf torque was attained without the gasket. this would show that the gasket did contribute to failure at maximum loads.

Spacers yielded unacceptable results for both first slip and maximum load. To align all spacers correctly is very difficult and was never done successfully in these tests.

Increased torques would improve the performance of all fixtures and would be reasonable based on past window construction and performance.

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



~~45 psi~~ A.52.

~~ANSYS:~~ $T_x = 296280$ lbs

T_x PER LINEAR INCH OF THE CIRCUMFERENCE

$$T_{x_i} = \frac{296280}{\pi(71)} = 1328.2 \text{ lbs}$$

FOR 9.858 WIDE FIXTURE —

$$T_{x_9} = \underline{13,094} \quad [\text{lbs}]$$

14.7 psi ANSYS:

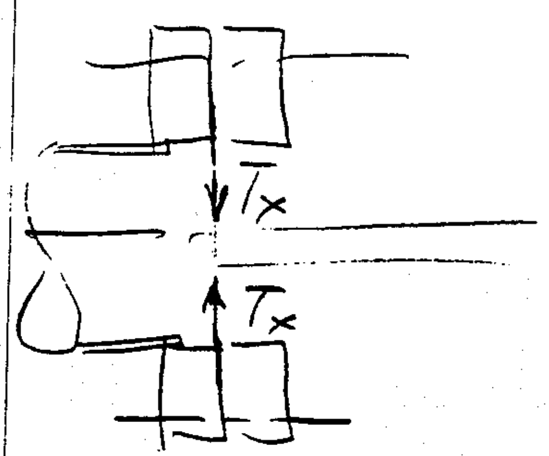
$T_x = 154880$ lbs

T_x PER LINEAR INCH OF THE CIRCUMFERENCE

$$T_{x_i} = \frac{154880}{\pi(71)} = 694.36$$

FOR 9.858 WIDE FIXTURE

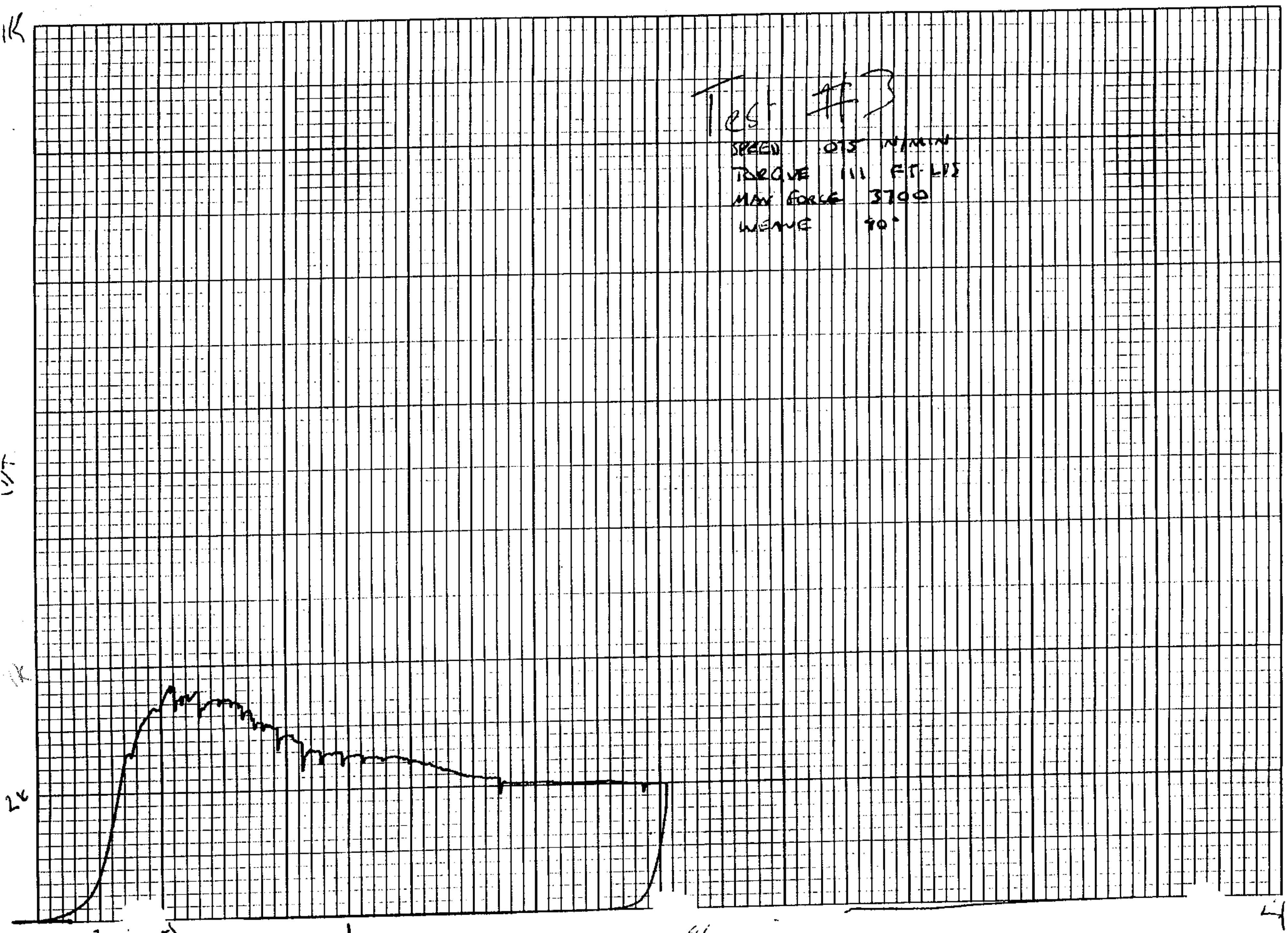
$$T_{x_9} = \underline{6845} \quad [\text{lbs}]$$

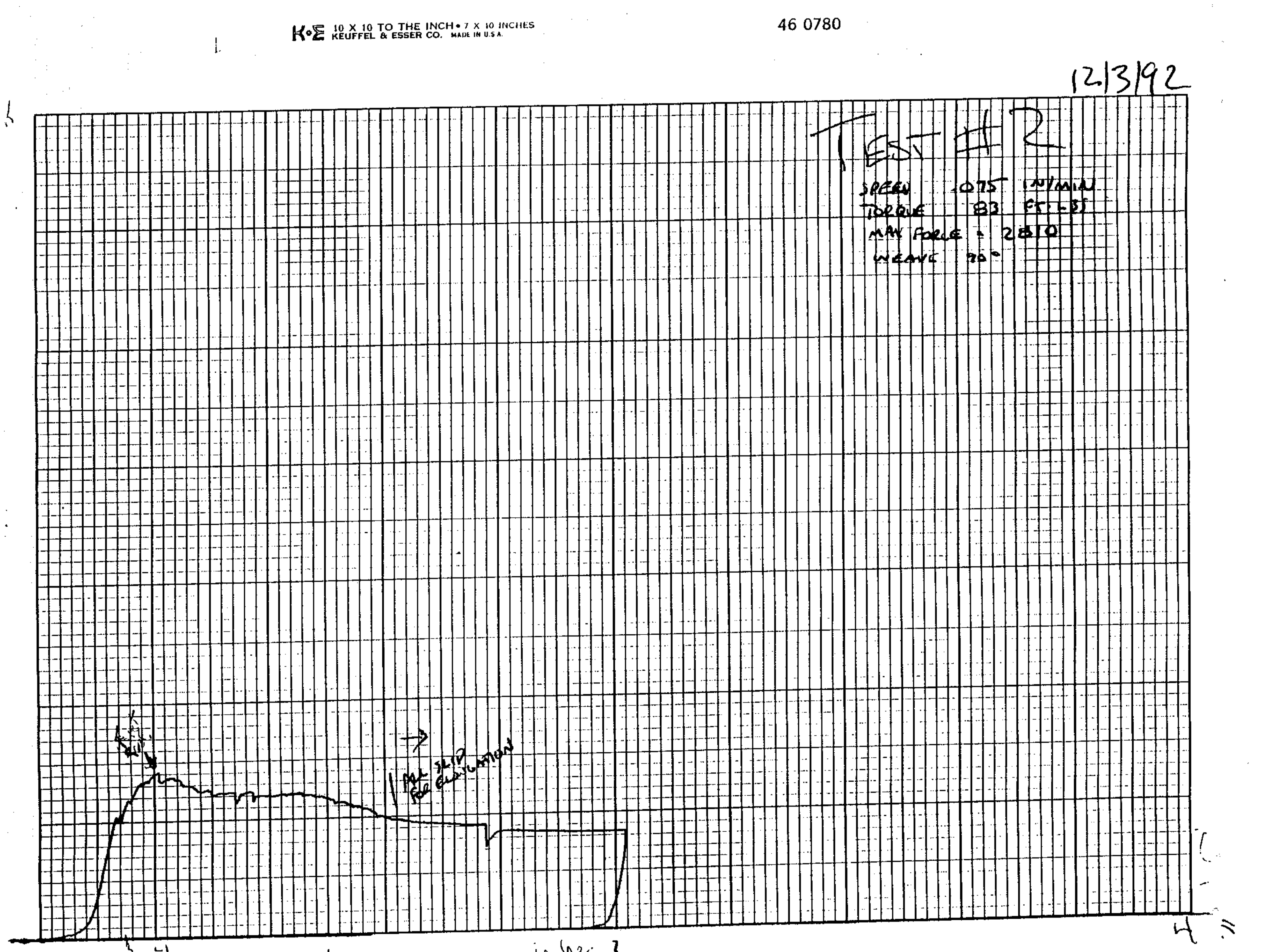


KEVLAR

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

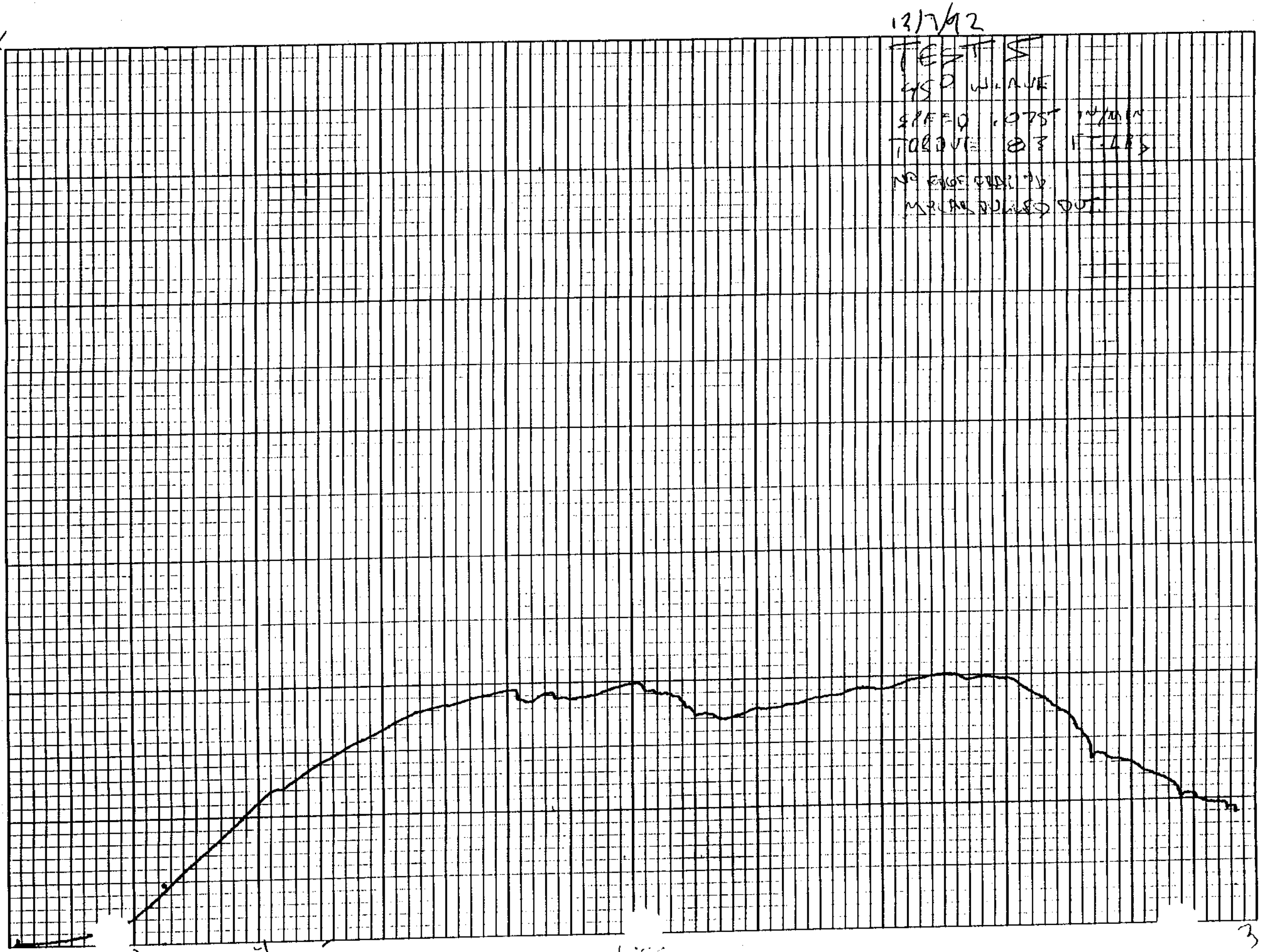
46 0780




K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.
46 0780
12/3/92
TEST #2
SPEED .075 IN/MIN
TORQUE 83 FT-LBS
MAX FORCE = 2810
WEAVE 90°
ALL SLIP FOR ELONGATION

K+E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

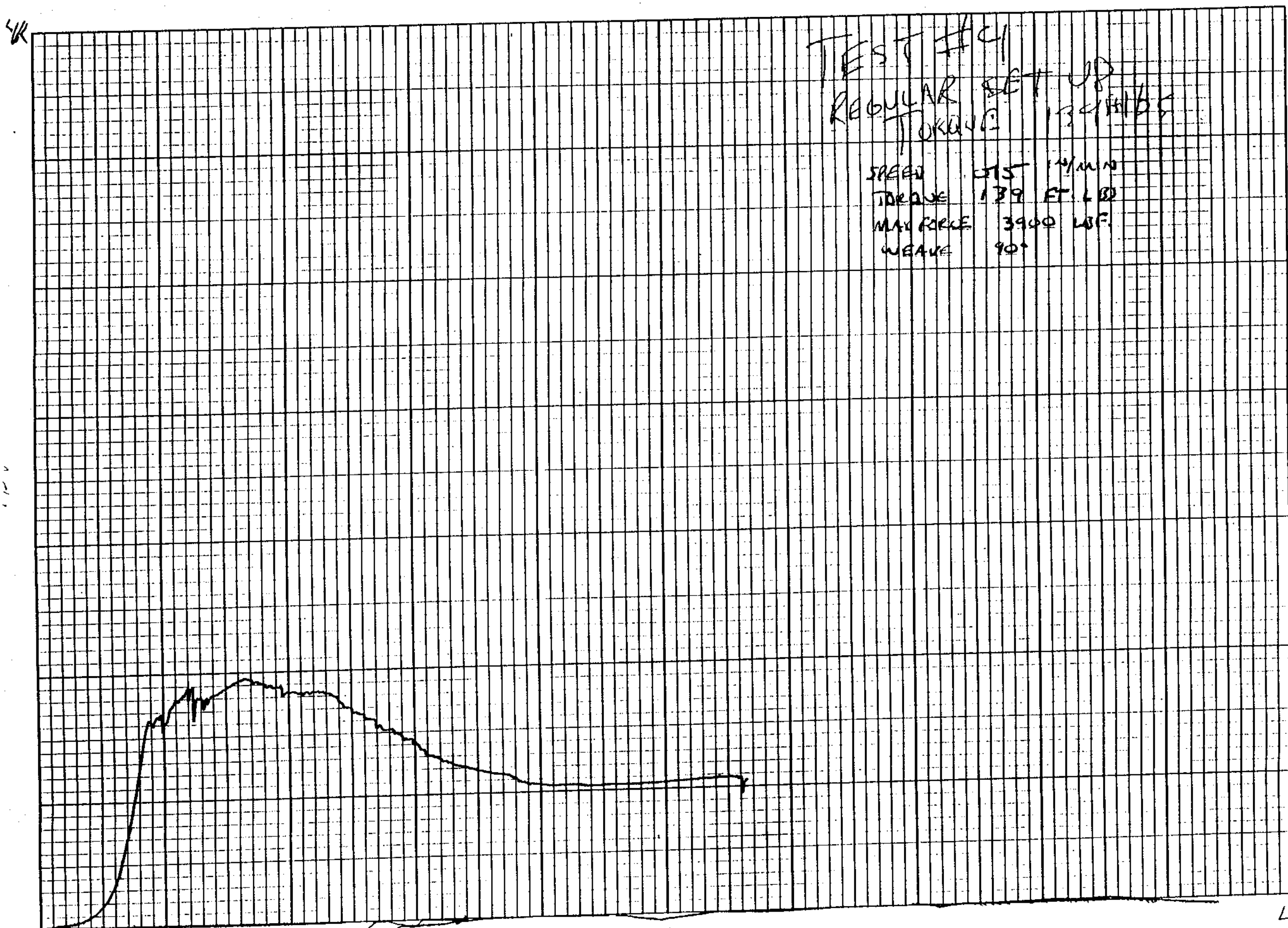
46 0780



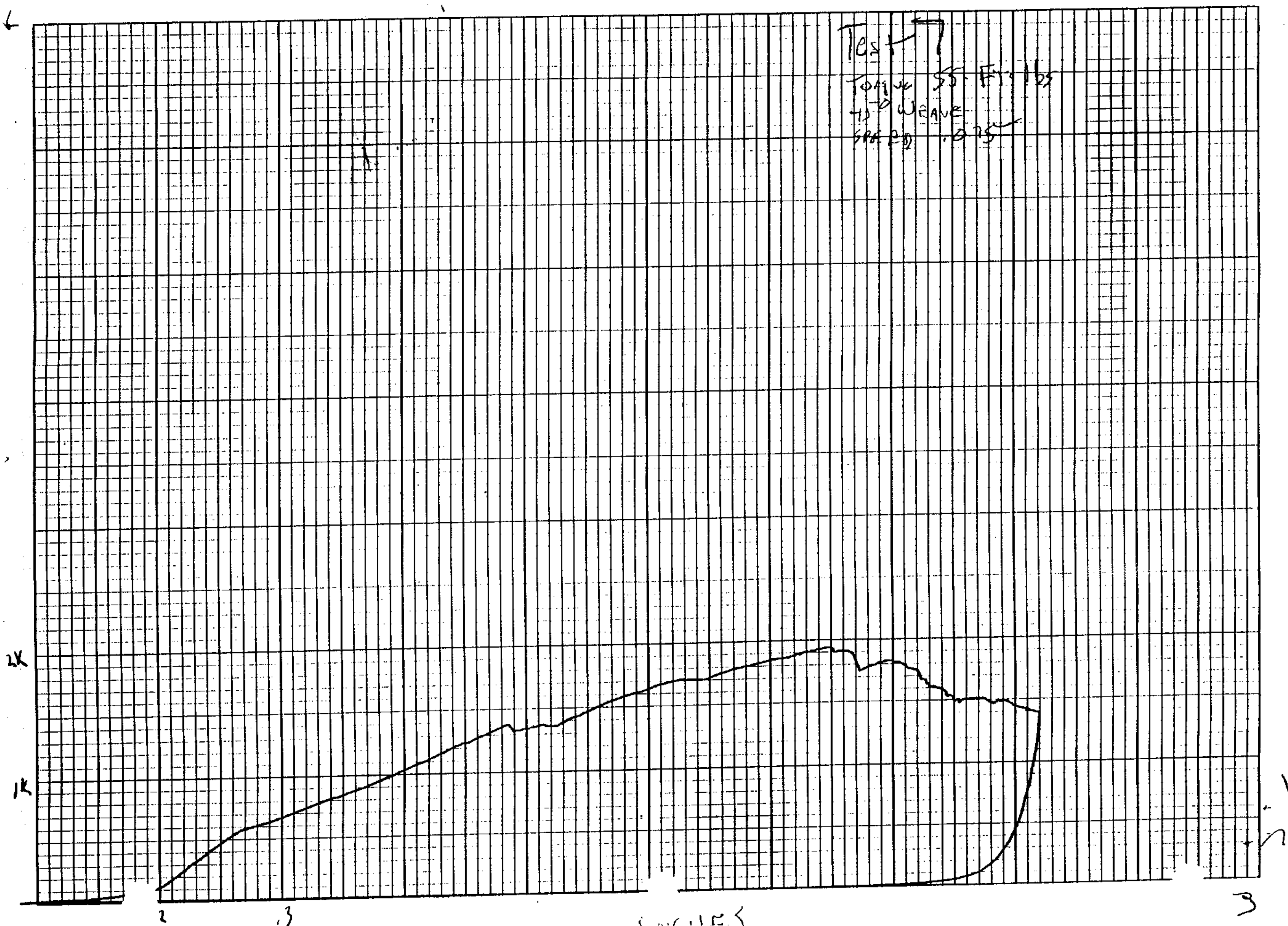
14.

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



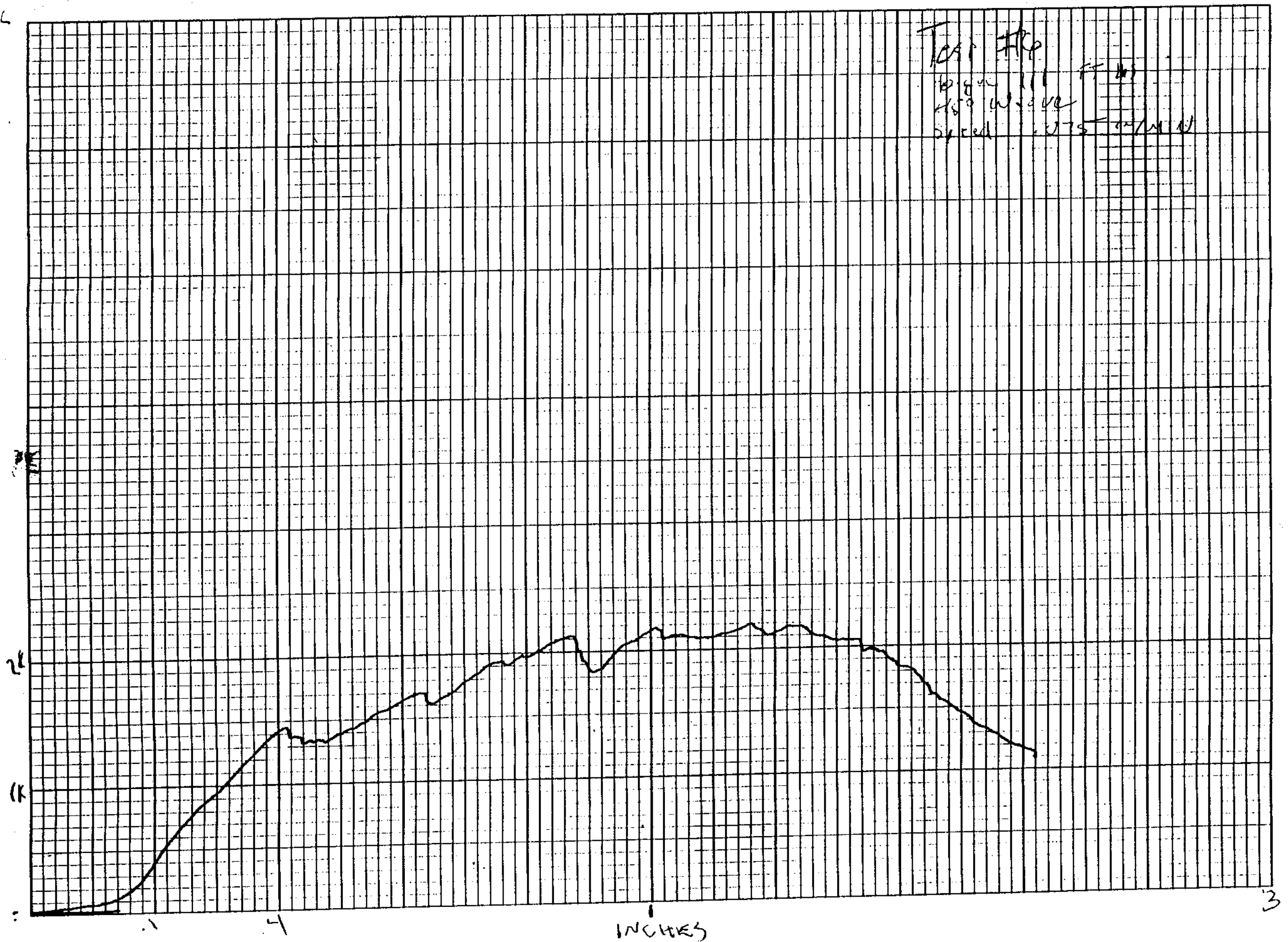
46 0780



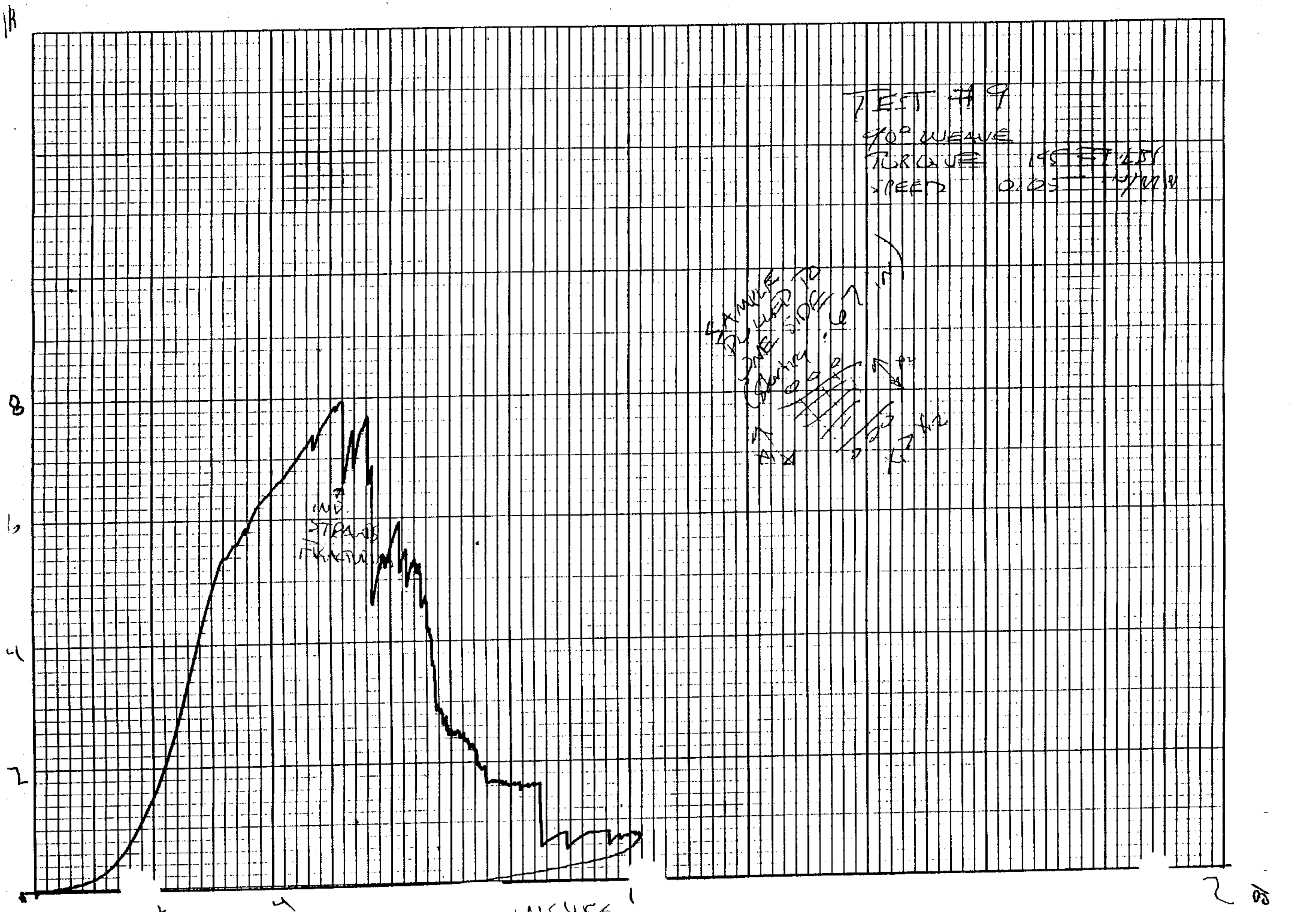
16.

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780

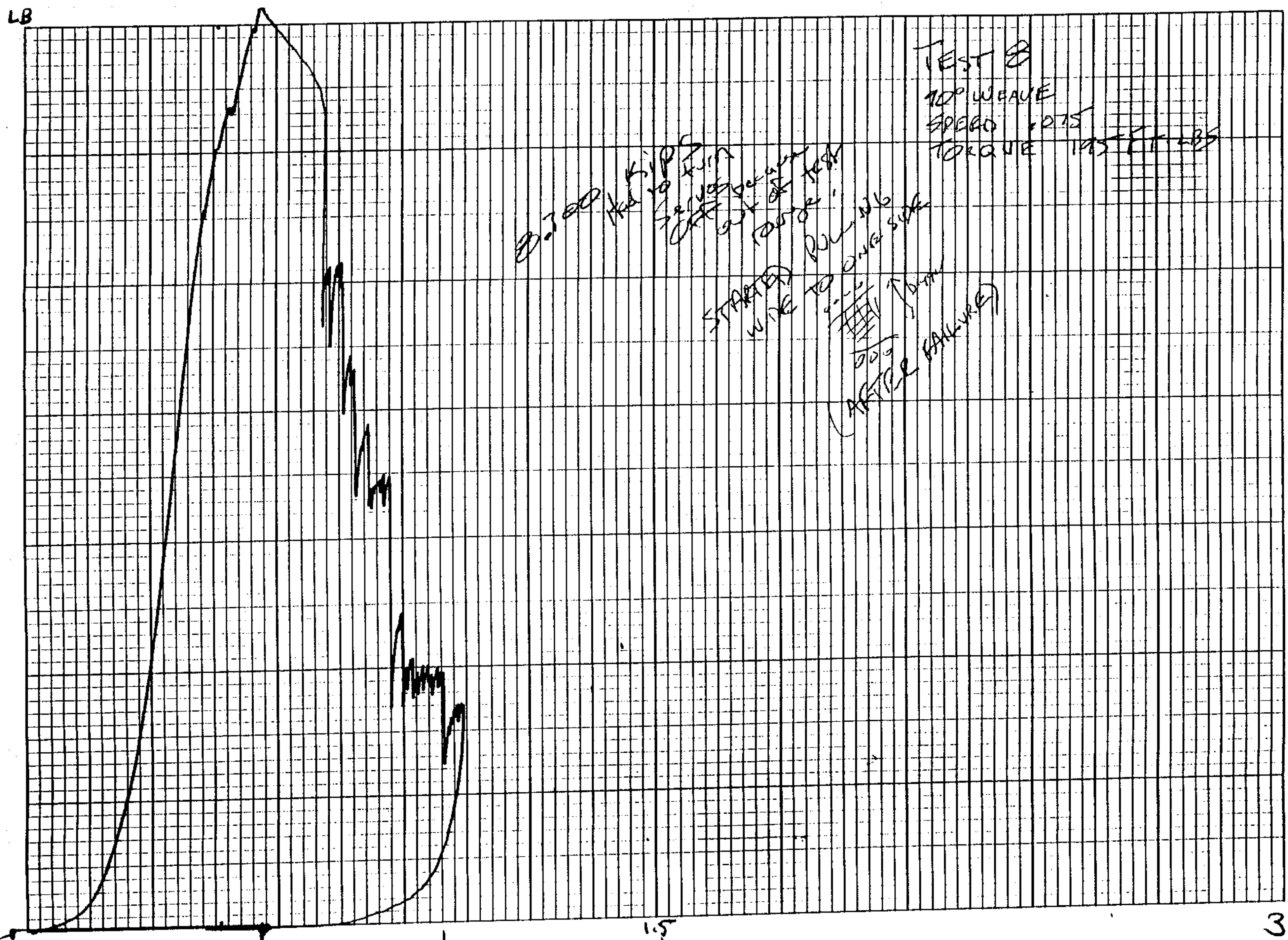


46 0780


TEST #9
40° WEAVE
TORQUE
SPEED 0.05
SAMPLE PULLED TO ONE SIDE
8
6
4
2
1
2

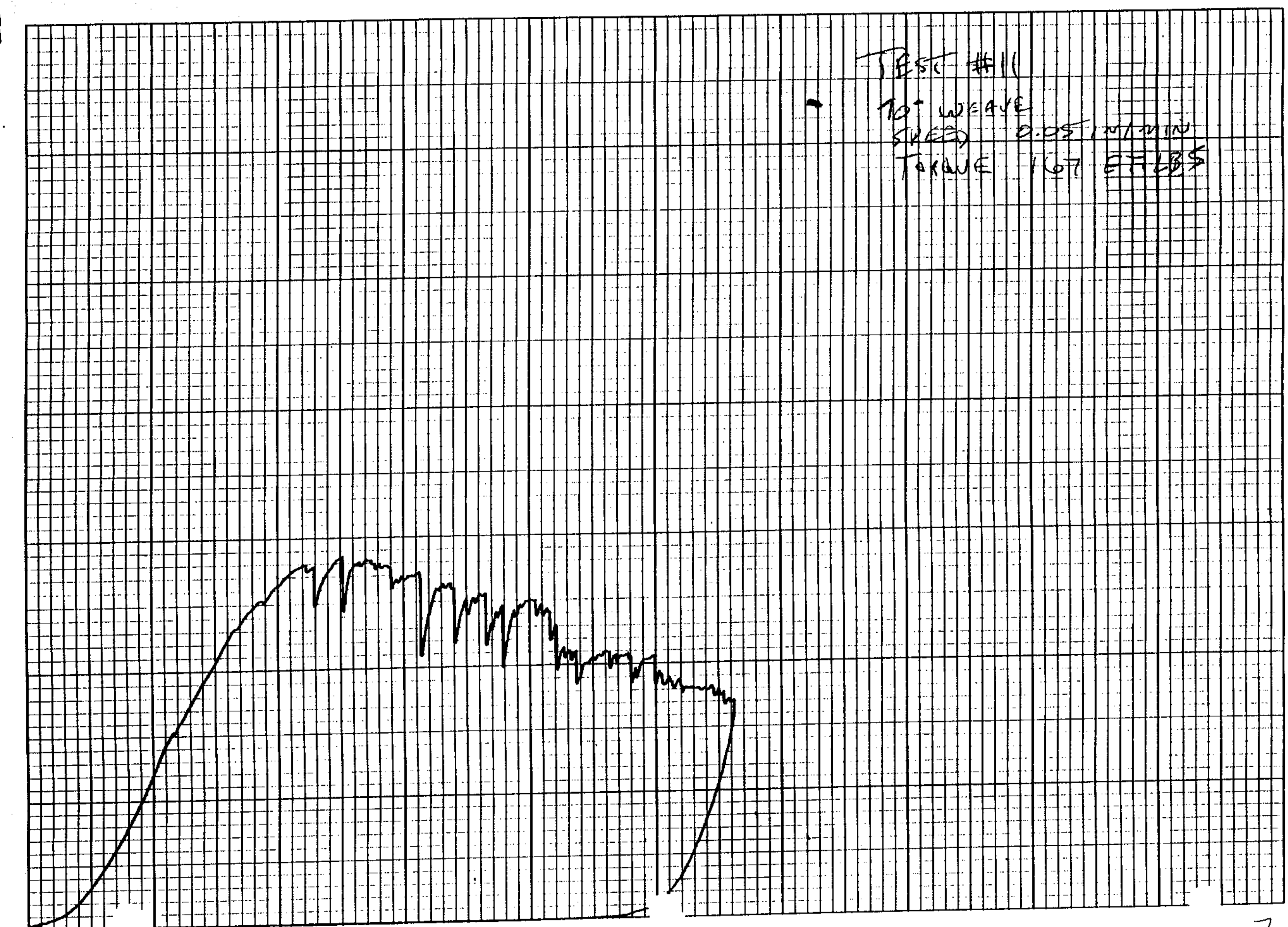
K+E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

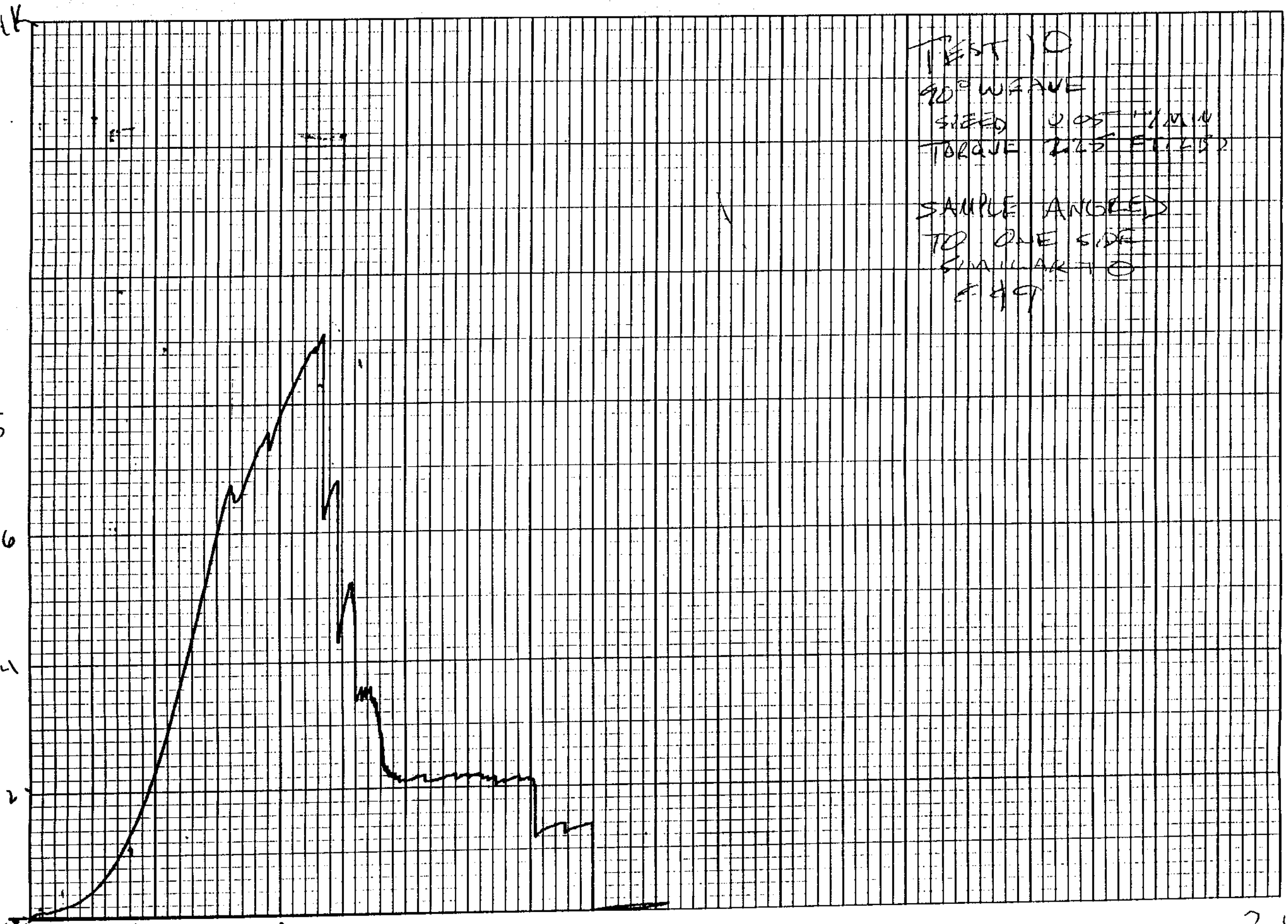
46 0780



K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

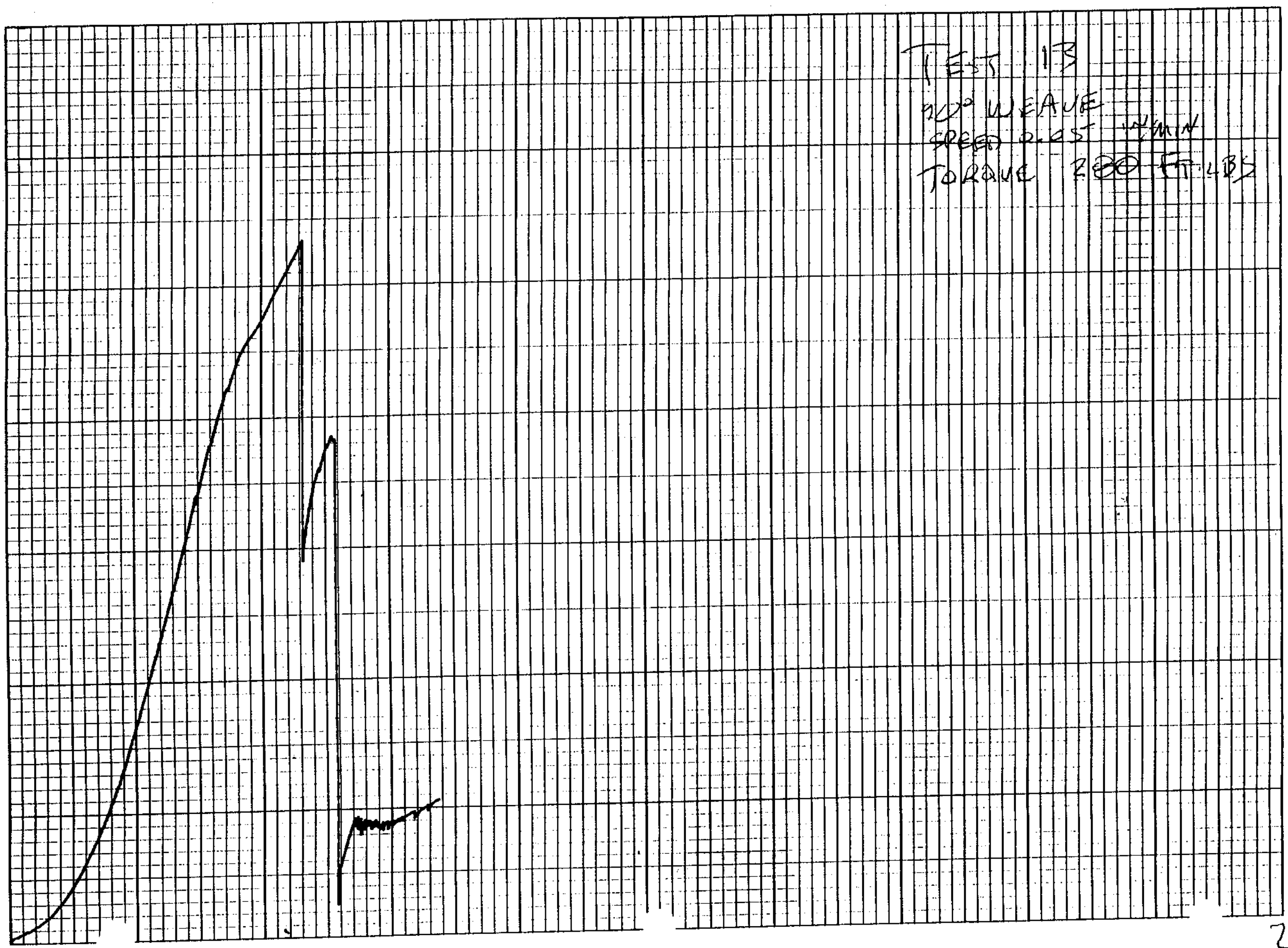
46 0780

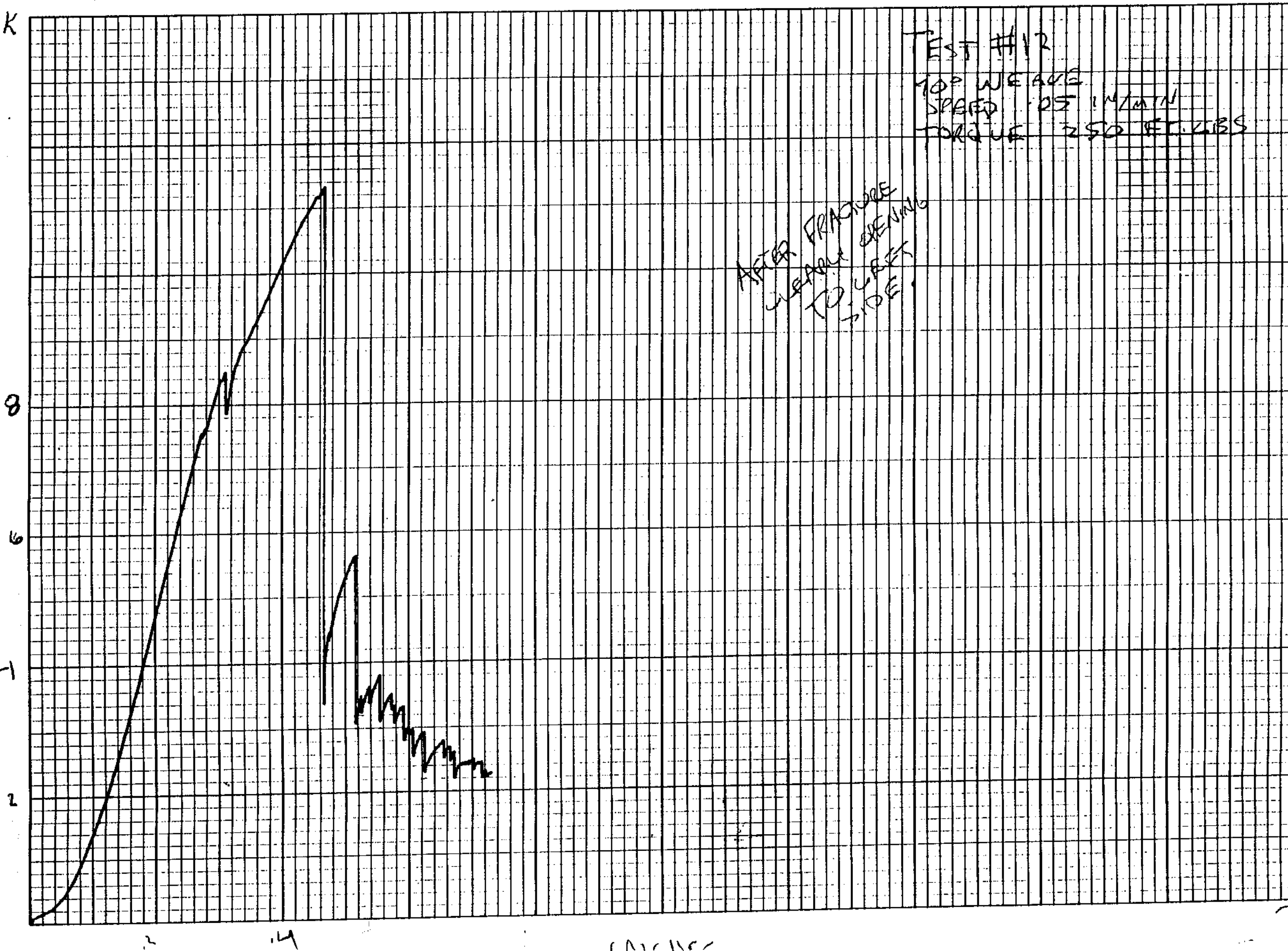



TEST 10
90° WEAVE
SPEED .05"/MIN
TORQUE 225 FT-LBS
SAMPLE ANGLED
TO ONE SIDE
SIMILAR TO
#49

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780

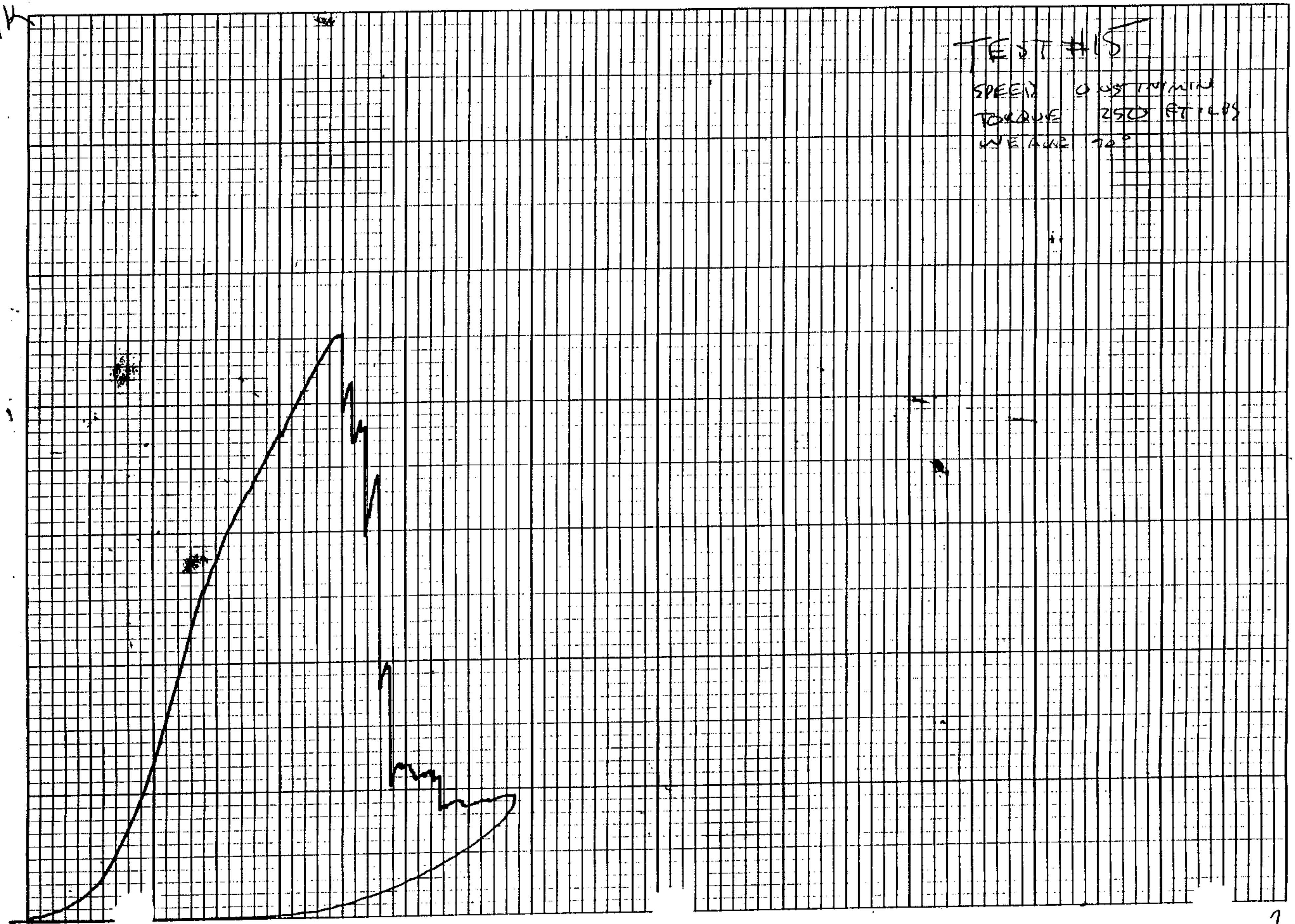



K
8
6
4
2
.2
.4
TEST #12
10° WEAVE
SPEED .05 IN/MIN
TORQUE 250 FT.LBS
AFTER FRACTURE
CLEARLY OPENING
TO LEFT
SIDE.

21.

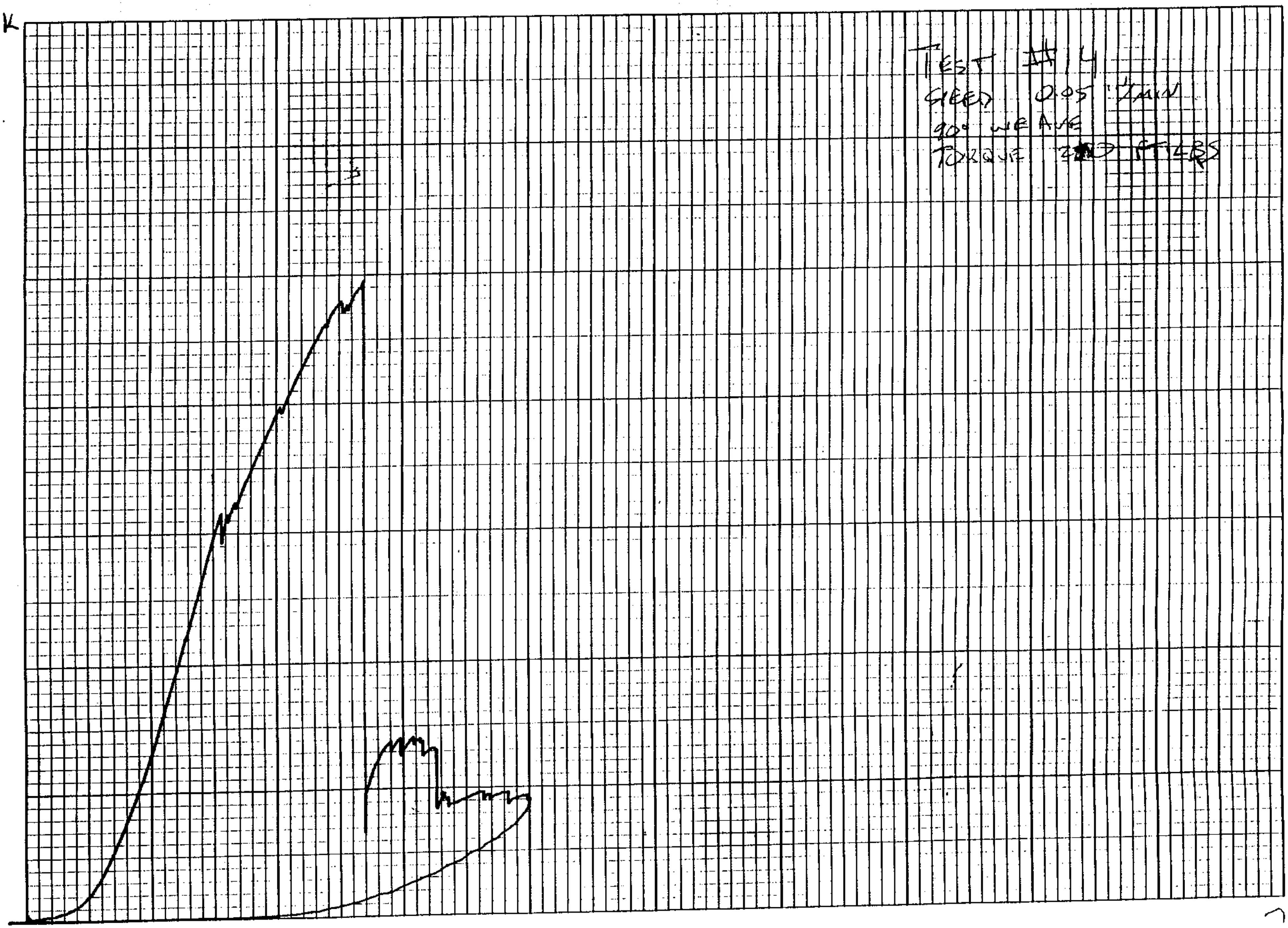
K+E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

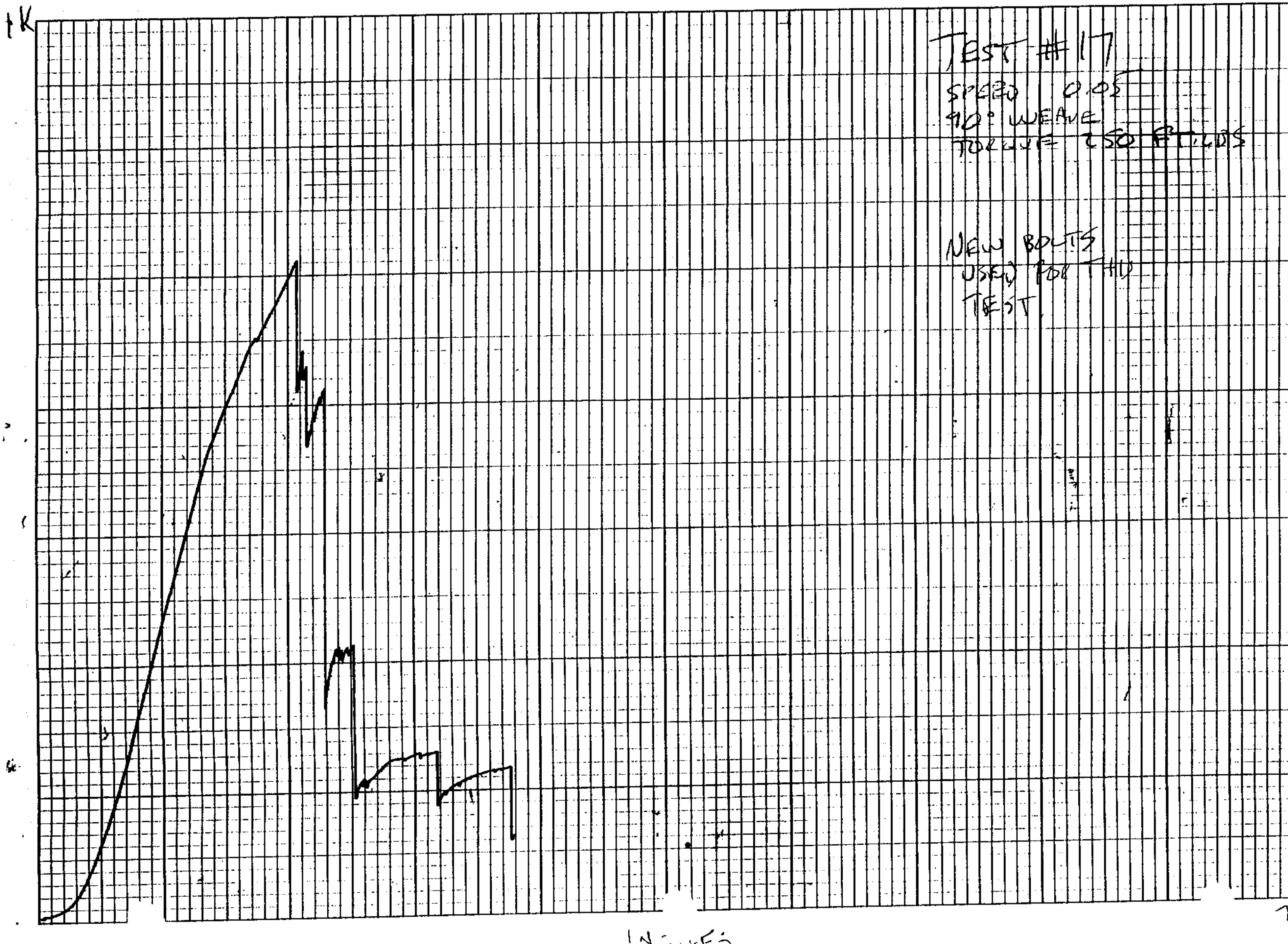
46 0780

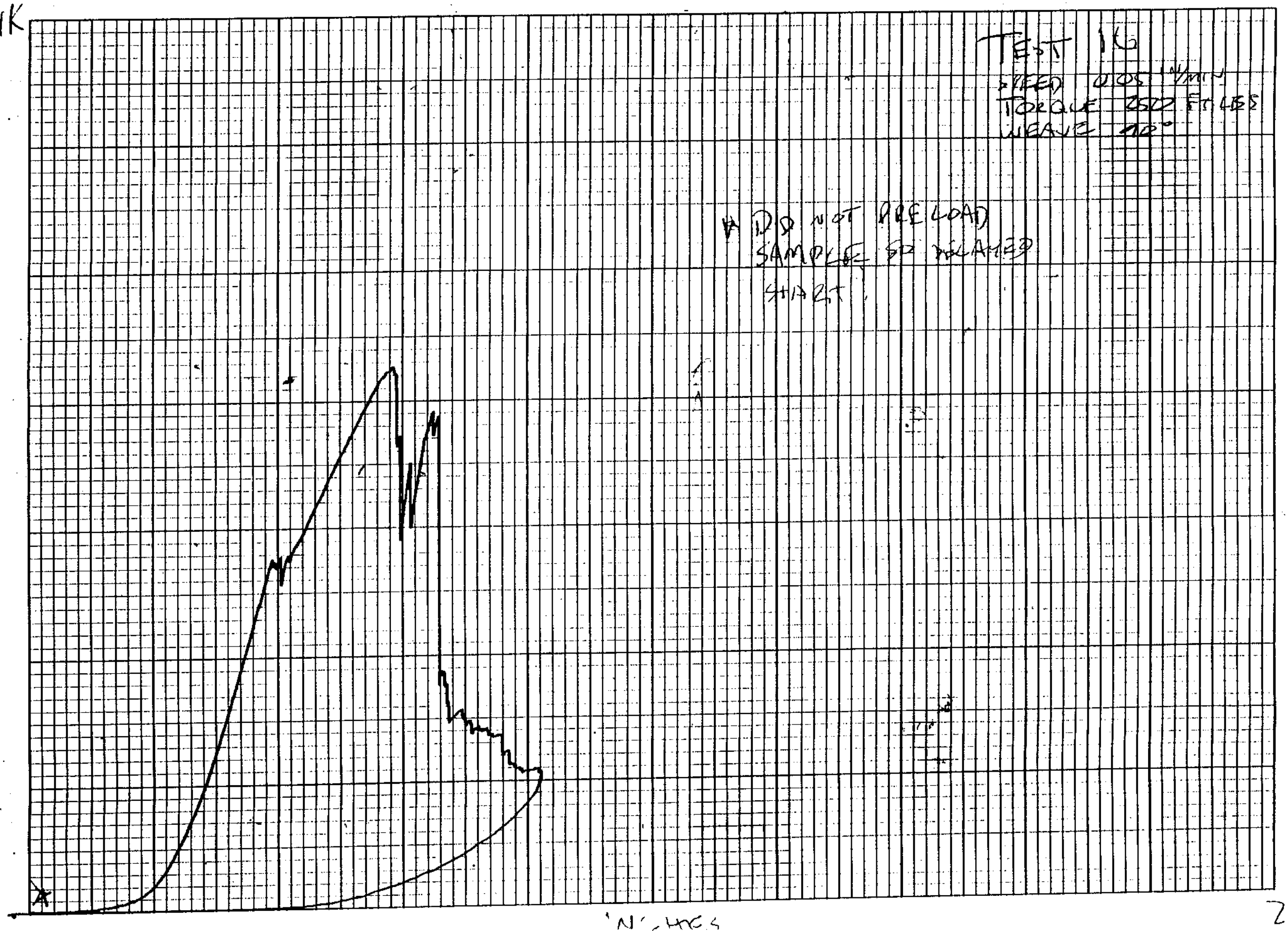


K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



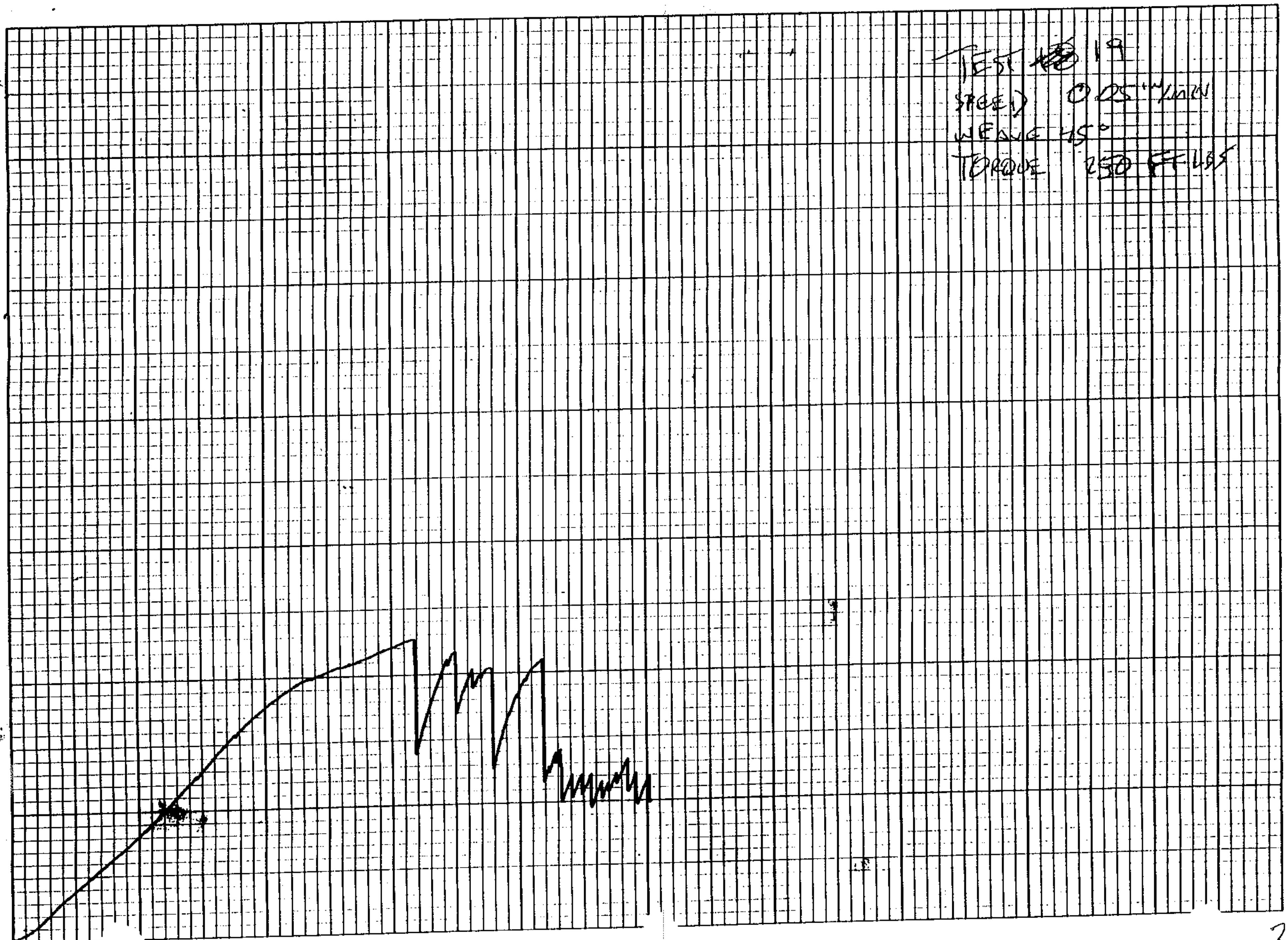

1K
TEST #17
SPEED 0.05
90° WEAVE
TORQUE 250 FT.LBS
NEW BOLTS
USED FOR THIS
TEST
2
INCHES


1K
TEST 16
SPEED 0.05 IN/MIN
TORQUE 2500 FT LBS
WEAVE
DID NOT PRELOAD
SAMPLE
SHAFT
2
INCHES

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

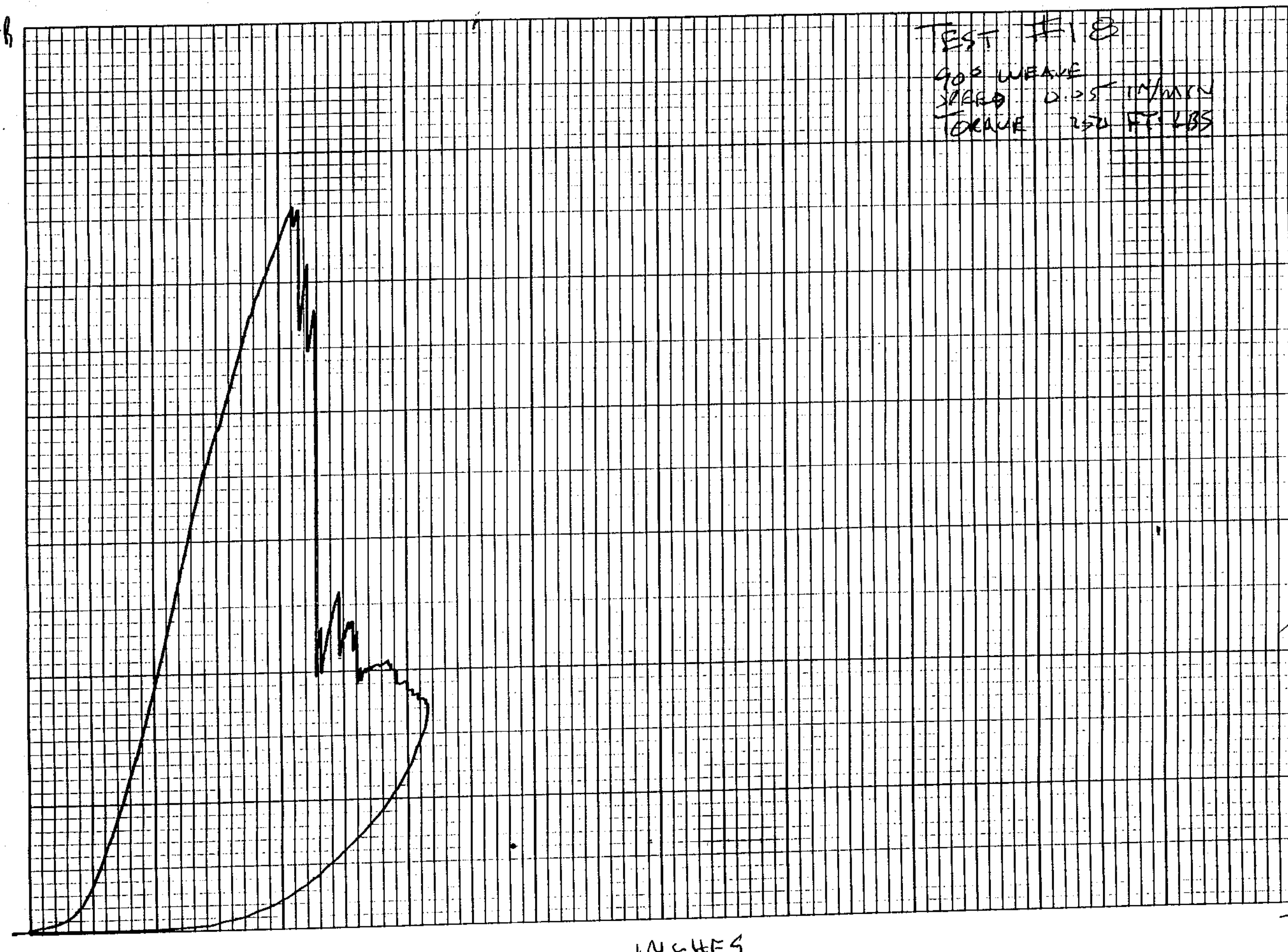
46 0780

K



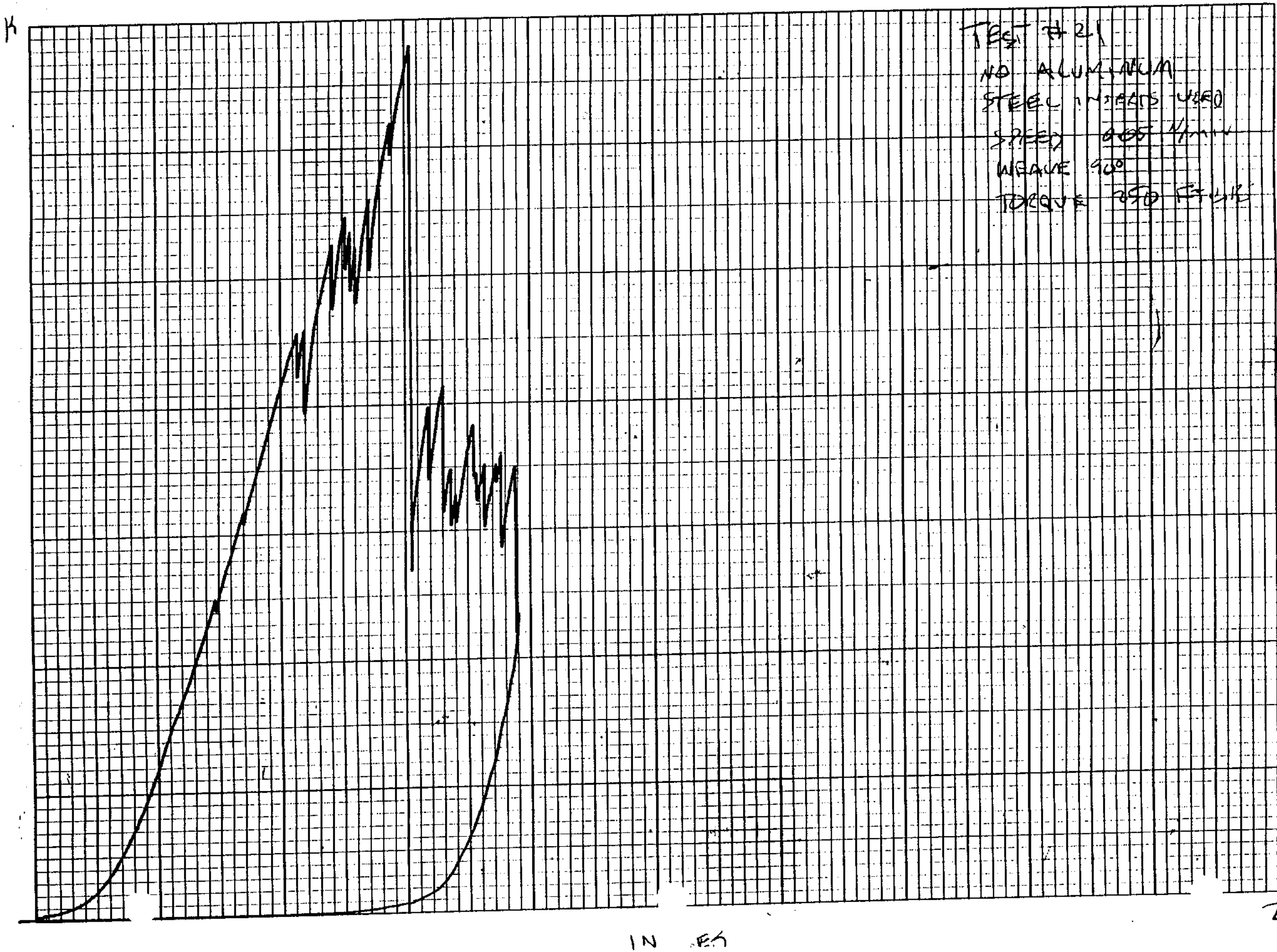
INCHES

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A. 46 0780

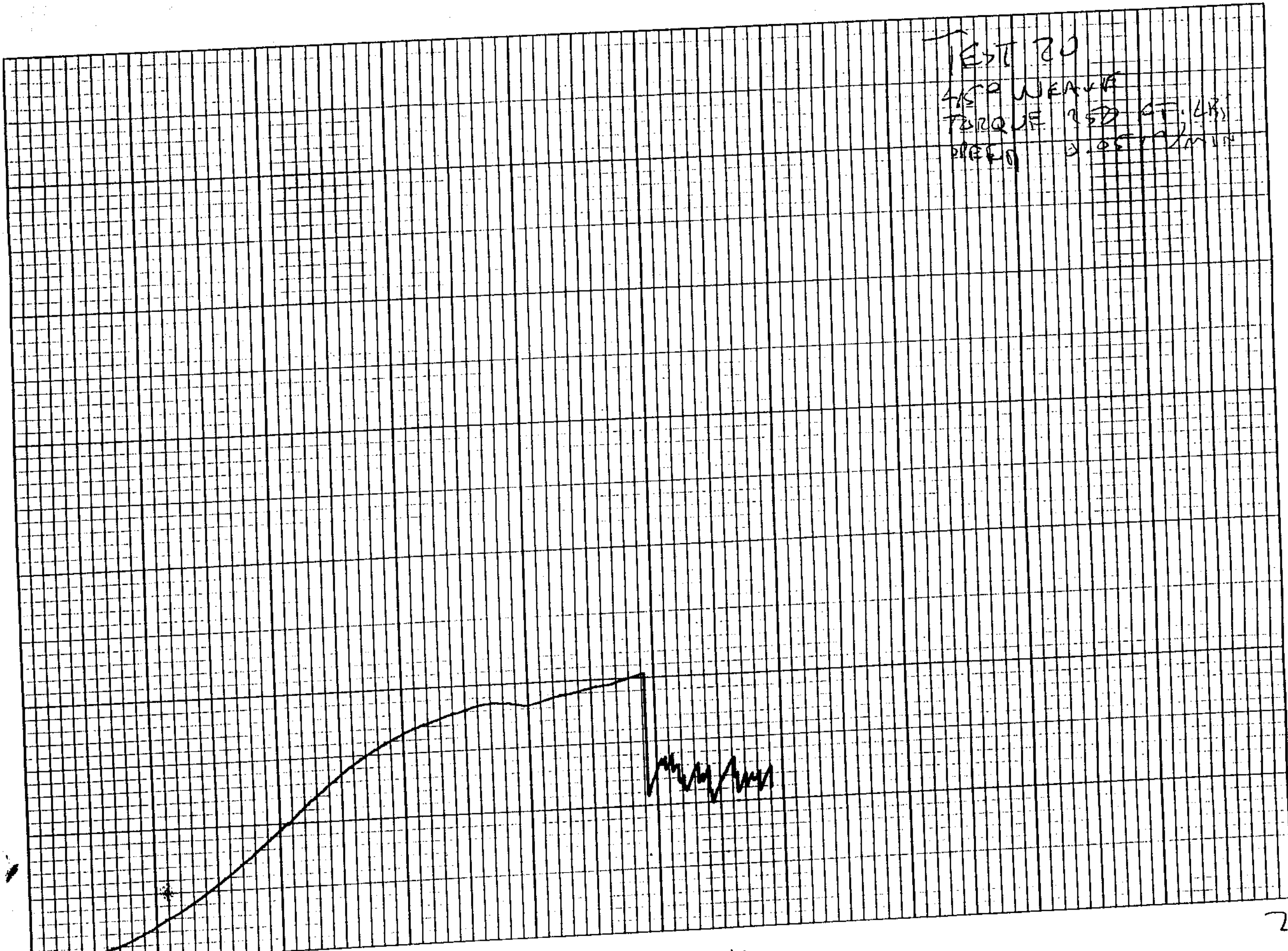


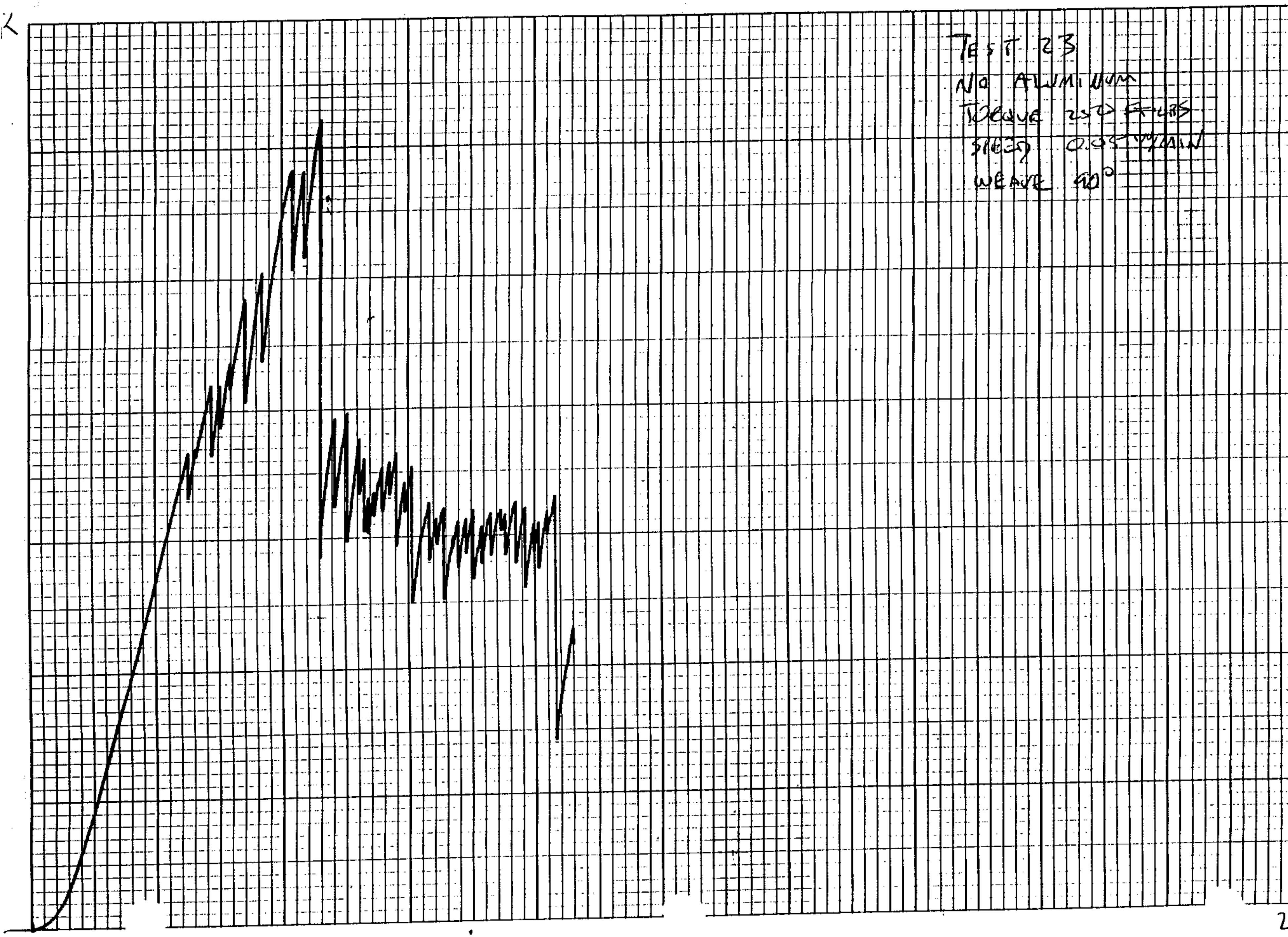
K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



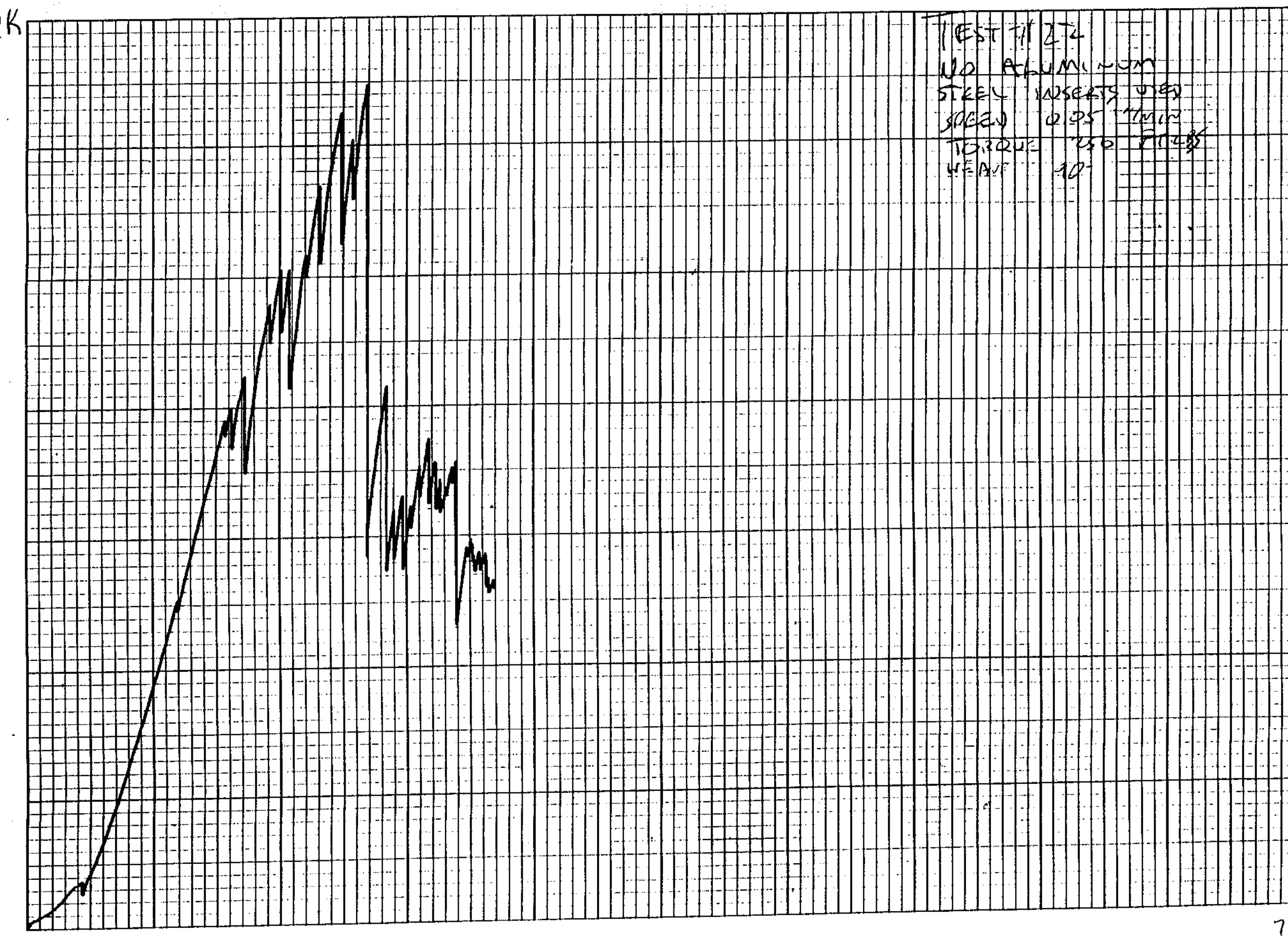
30.

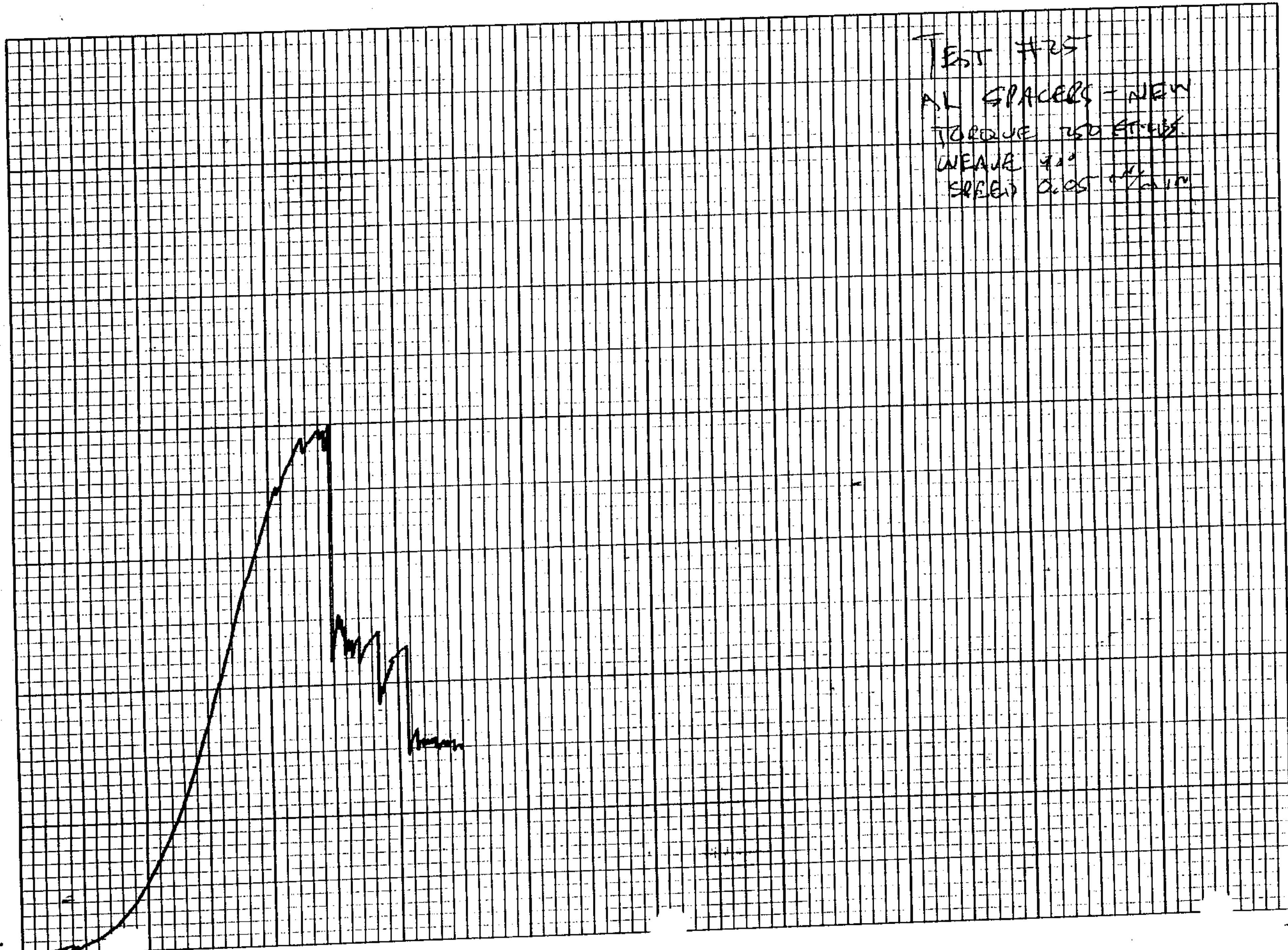

TEST 20
45° WEAVE
TORQUE 250 FT-LB
SPEED 0.05 IN/MIN


TEST 23
NO ALUMINUM
TORQUE 250 FT-LBS
SPEED 0.05"/MIN
WEAVE 90°

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

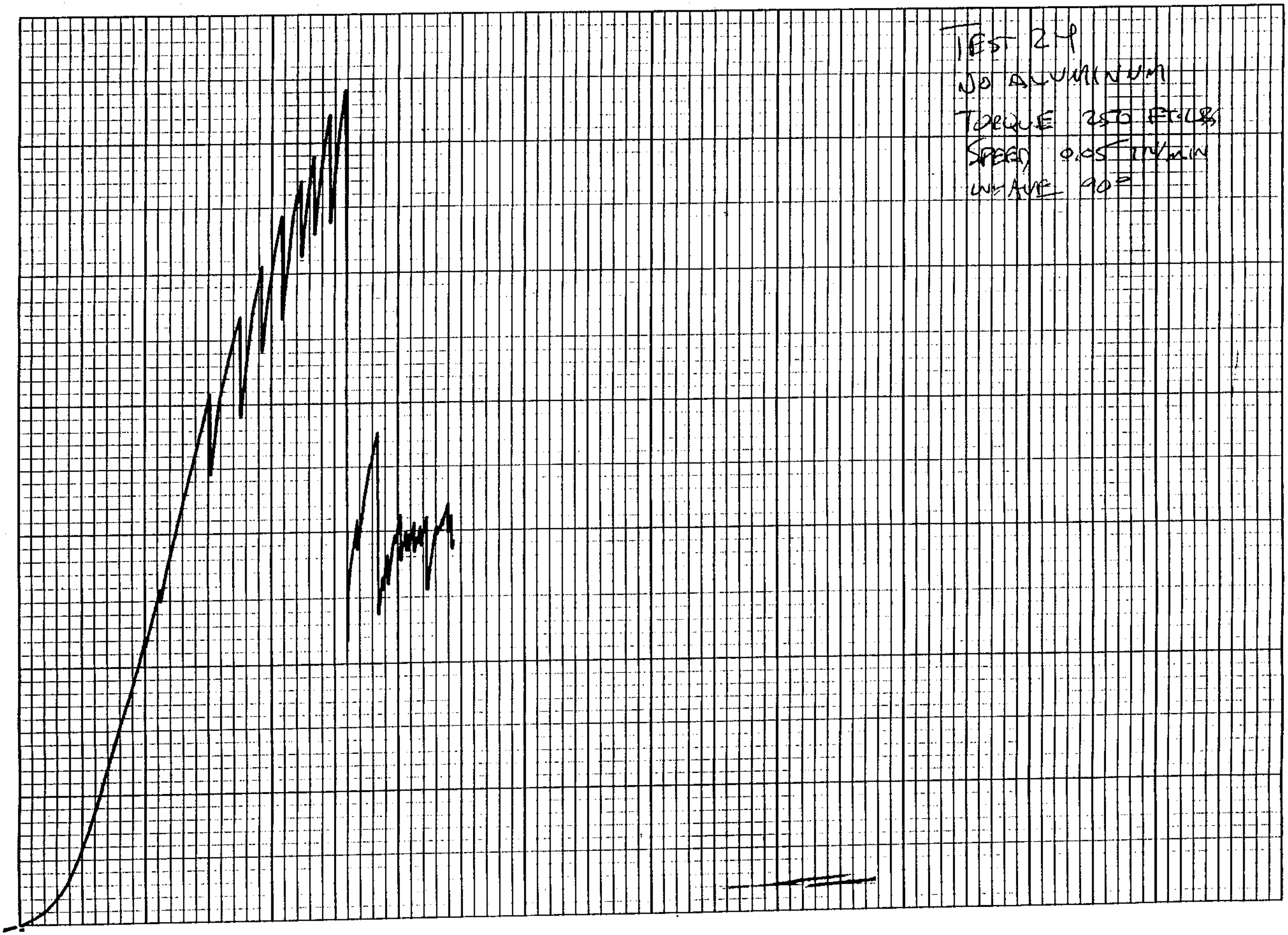
46 0780




TEST #25
AL SPACERS - NEW
TORQUE 250 FT-LBS
WEAVE 4"
SPEED 0.05 in/min

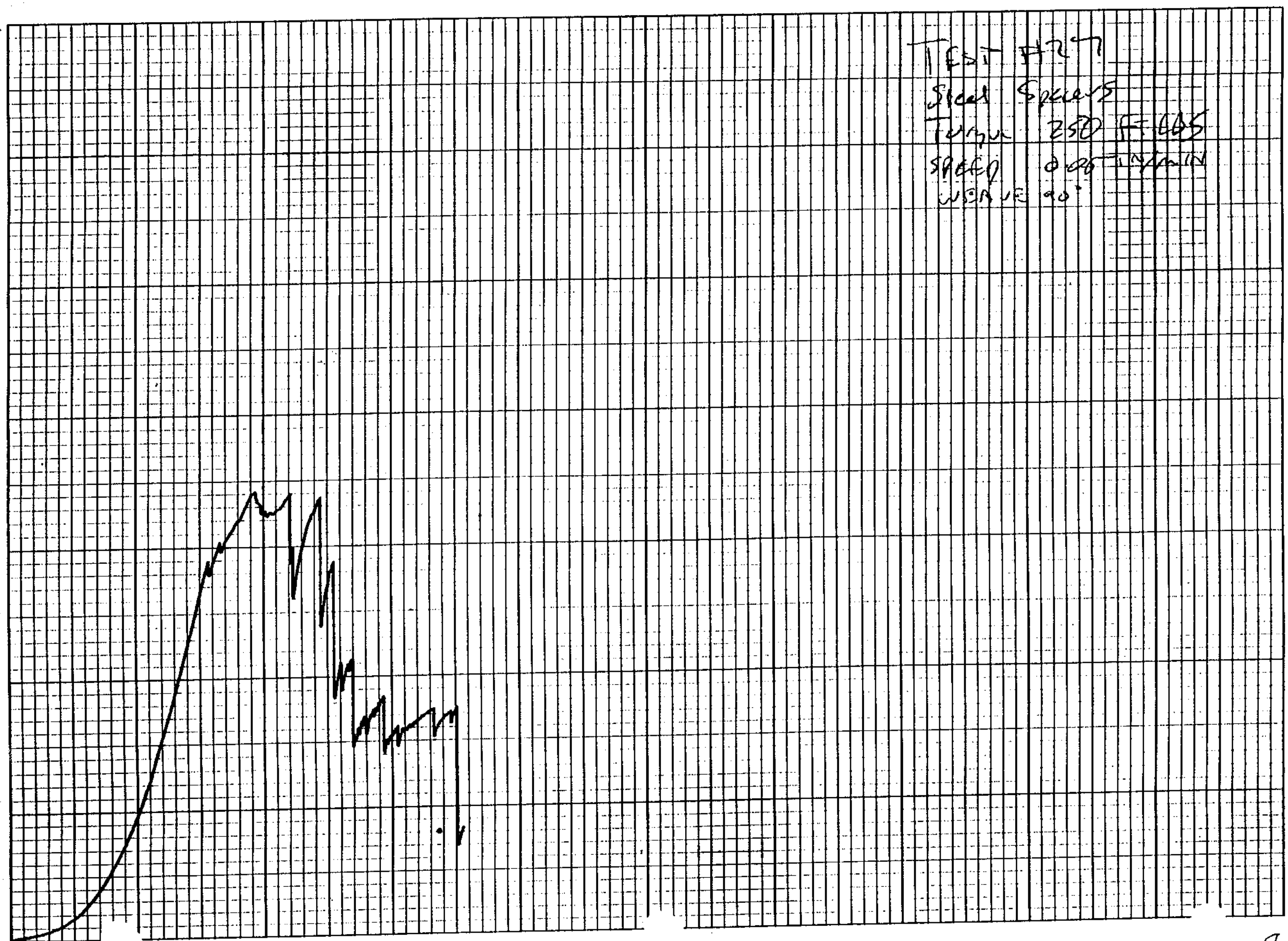
K·E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



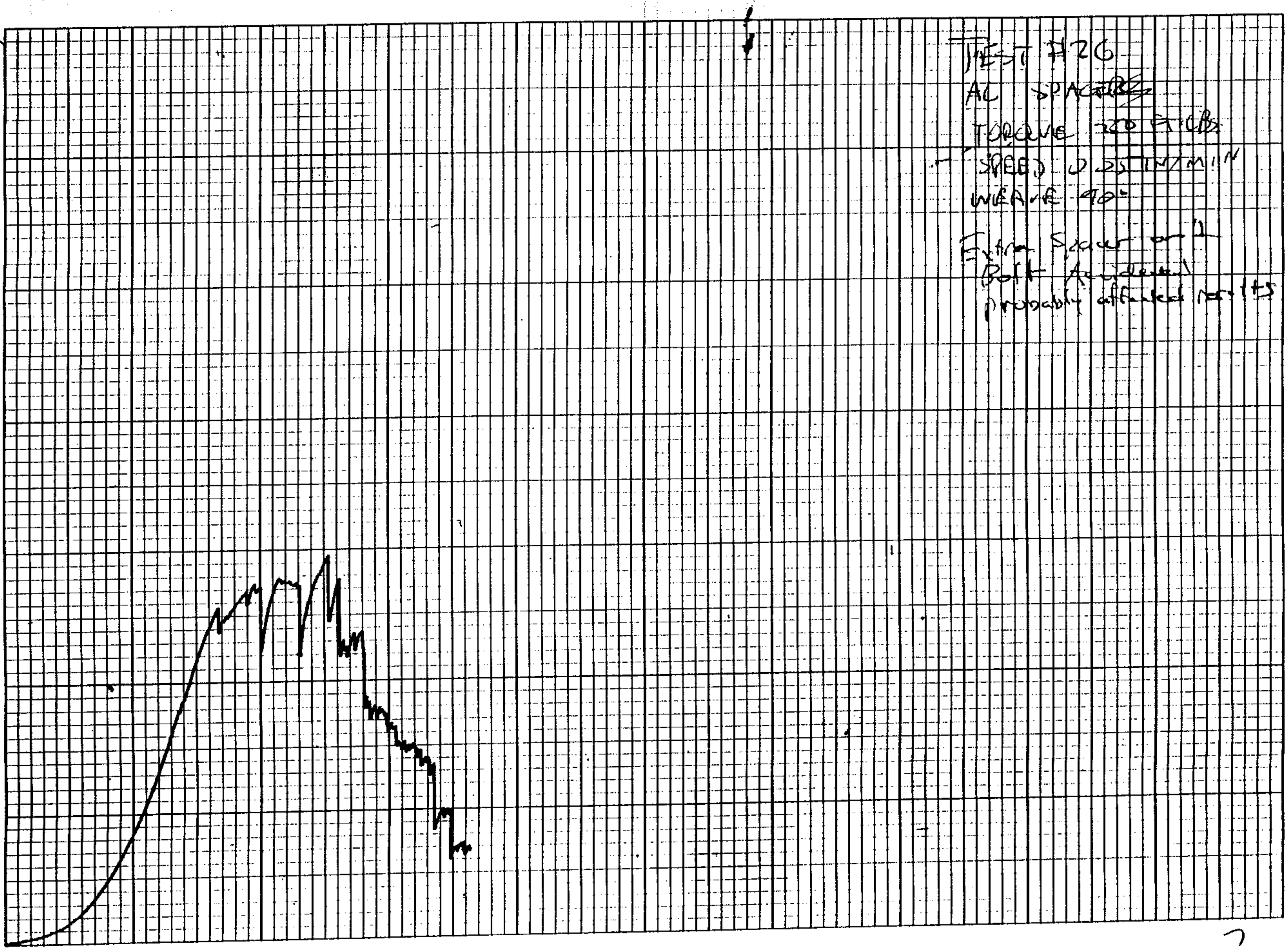
K•E 10 X 10 TO THE INCH • 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.

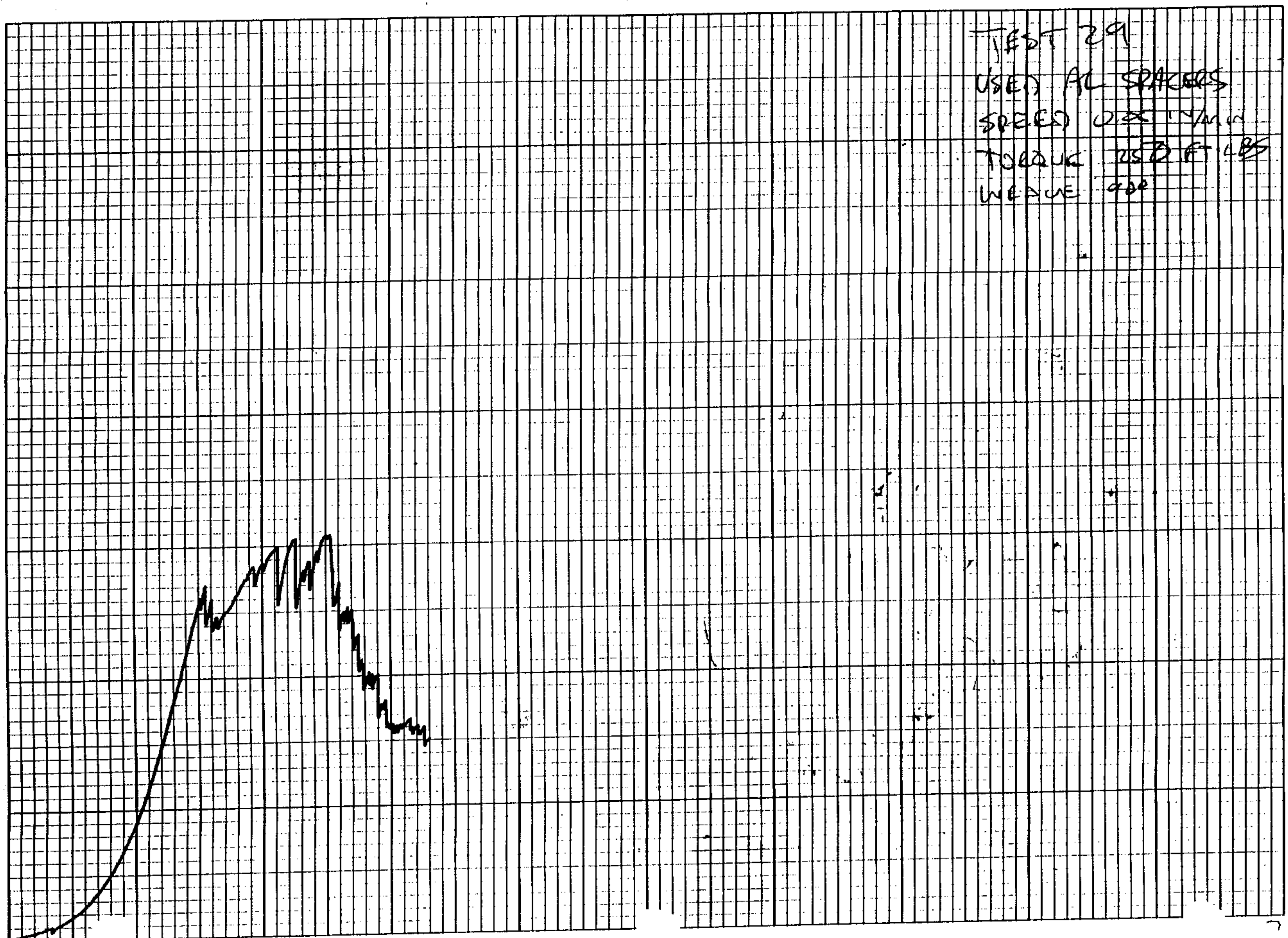
46 0780



K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

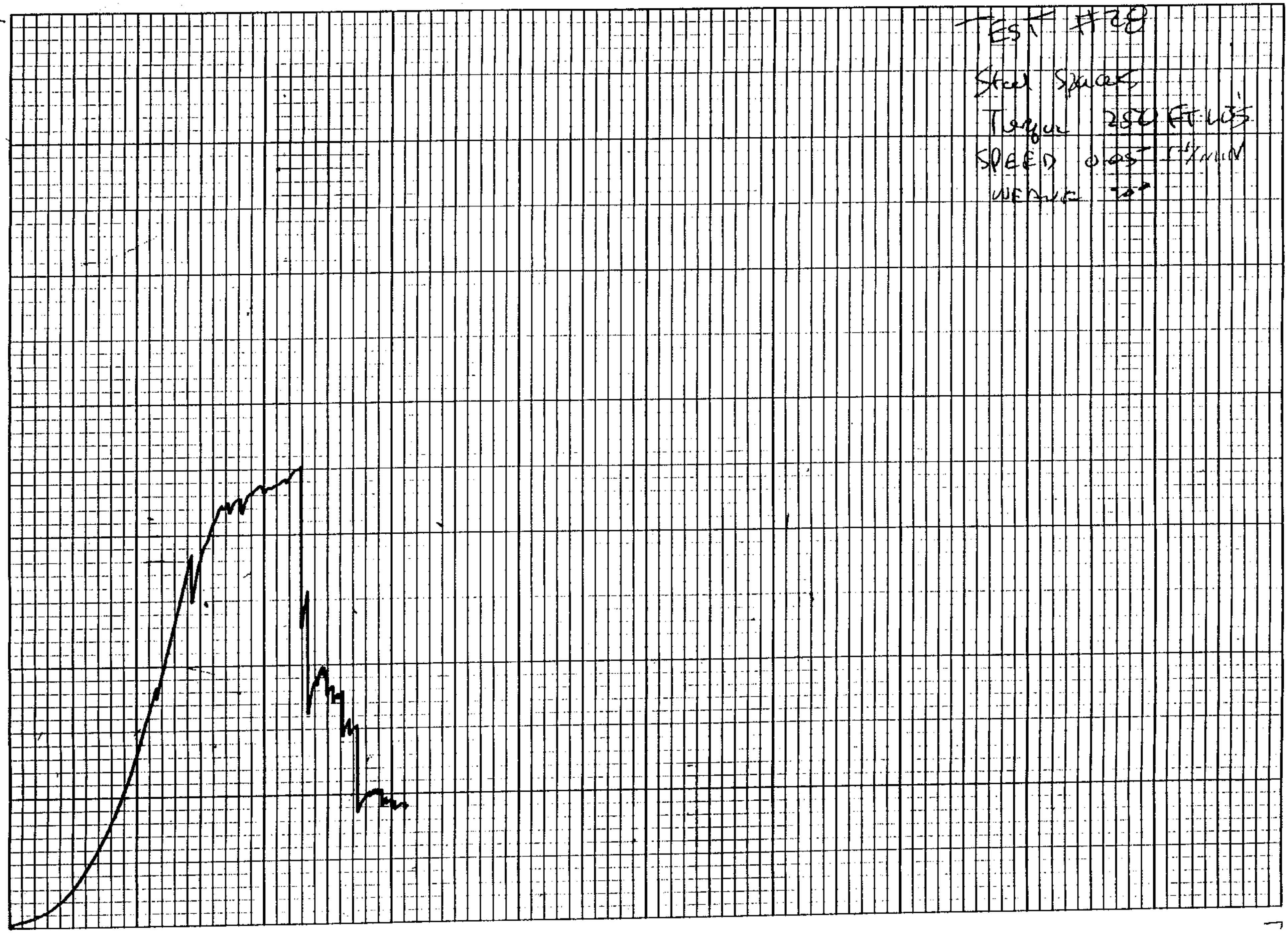
46 0780




TEST 29
USED AL SPACERS
SPEED 0.2" 1/min
TORQUE 250 FT LBS
WEAVE 90°

K•E 10 X 10 TO THE INCH • 7 X 10 INCHES
KEUFFEL & ESSER CO. MADE IN U.S.A.

46 0780



ABSTRACT:

A second series of tests was performed for the KTev 1.8m kevlar reinforced vacuum window. These tests were done to verify the results of previous tests (December 1992) and to find the effects of added epoxy and a different kevlar fabric manufacturer. The tests indicate that the bulkhead design and construction is sufficient to handle three times the operating pressure exerted during operating vacuum conditions. The addition of the epoxy dramatically improved the bulkhead performance and no significant differences in kevlar fabric could be noted. Some concerns remain over the validity of these tests but the window should perform at least as well or better than the tests indicate.

Kevlar Window Tests Part II
KTev 1.8m Window

Frederick Renken
7/23/93
FNAL RD/MSD Mechanical Support Group

A:

r Test Data
Renken

ample	Orientation (vertical threads/inch)	First Slip (kips)	Max Load (kips)	Notes
	35	12.8	13.4	
	35	12.9	13.05	Test Curve Lost/Computer Failure
	35	11.9	11.99	Unusual Fail Pattern
	35	12.5	13.33	
	35	11.9	11.96	
	Average:	12.4	12.746	
	Deviation:	0.480	0.716	
	34	14.5	14.51	Almost Complete Fracture
	34	13.1	13.1	Sample made with "Tension"
	34	15.25	15.25	
	34	13.9	13.96	
	Average:	14.188	14.205	
	Deviation:	0.911	0.907	
s:	Average:	13.194	13.394	Desired is 13.094
	Deviation:	1.146	1.075	

ES:

.tation signifies the number of threads per inch sustaining the load or perpendicular to the ›ing fixtures. When one is looking at the sample, during the test, this is the number of ls running vertically. The kevlar fabric is supplied with 35 threads per inch in one direction n through 34 threads perpendicular.

lesired load to sustain was calculated from an ANSYS analysis of the window [SZYM]. These calculations determined the load along a section of the bulkhead the same length · test sample for both vacuum conditions and for a safety factor of three. (Attached)

·sts were performed at a crosshead speed of 0.05 inches per minute complying with ASME e testing standards.

INTRODUCTION:

KTev, the new fixed target experiment at Fermilab, uses a large kevlar fabric reinforced mylar vacuum window. To verify the window meets safety and performance requirements a series of tests were performed using a clamping fixture prepared for a tensile test. The fixture was designed to simulate the bulkhead for a tension load. The results of these tests provided a bolt torque at which the bulkhead would be able to maintain a vacuum seal without damaging the mylar window material. The tests also verified which bulkhead construction and configuration would perform best. [RENK 1992] [SZYM 1993]

After the completion of tests in December 1992 it became necessary to perform another series of tests. These tests differed from the previous tests by the addition of epoxy to the test samples and a change in kevlar fabric manufacturer. The epoxy, used in all vacuum windows on site, was not used in previous tests because the cure requires 24 hours. This amount of time was neither available nor practical for the earlier tests. The second tests investigated the effects of the epoxy on both the load at first slip and the load at failure. A new fabric manufacturer was necessary because the old one could not supply a single sheet to cover the 1.8m window. Testing was performed in two orientations because the new fabric did not contain the same number of threads per inch in both weave directions. Finally, a new part was required for the Instron testing machine to safely test to the higher loads expected. Everything else, including bolt torque of 250 ft lbs and the clamping fixtures using aluminum o-rings, was the same for the second tests.

CONCLUSIONS:

The 1.8m window should be able to sustain a 45 psig load. This is a safety factor of three over the operating conditions of the window (14.7 psia on vacuum). As attached calculations show, the force exerted in the x direction at 45 psig is 13,094 lbf which is below both the average first slip and failure values of 13194 lbf and 13394 lbf respectively.

The fabric appears to be stronger in the direction with fewer threads per inch. This is probably due to fraying which was more evident in the first tests performed with 35 threads per inch sustaining the load. Without fraying the fabric is most likely the same in any direction. The actual window, being circular and enclosed, will not have any fraying and so will be able to sustain higher loads than these tests indicate. Otherwise no differences in kevlar manufacturers can be determined from this test.

Failure patterns indicate uneven load distribution on the sample or the presence of weaker possibly damaged points. It was impossible to test distribution with the apparatus despite attempts to create an even load throughout a preloaded test sample (kev7). To construct the actual window, a plywood frame will be used to apply tension to the kevlar cloth to evenly and efficiently distribute it across the bulkhead. This was also done on the four foot window previously constructed.

The use of epoxy on the test samples dramatically increased the load at which first slip occurred. Without the epoxy, the average first slip occurred at 7062 lbf. The final failure value was also increased by the use of the epoxy. This is shown by the increase from 10670 lbf, without the epoxy, to 13394 with the epoxy. Additional epoxy placed around the bolt holes, as was done on previous windows, may continue to improve load values but not significantly.

DISCUSSION OF DATA AND OBSERVATIONS:

Fraying was clearly evident on all samples. It was more prevalent in samples that fractured at lower values, especially around the bolt holes and epoxy region, . The test samples would fray along their edges during the entire test until failure. Fraying was also more significant on the samples with more threads in the pull direction. Fraying will not occur on the actual window because there are no free edges from which it can begin.

Effects of handling the samples during transportation and loading are indeterminate. The samples were prepared at Meson Assembly Building and then transported to the testing machine in the village. Since the testing clamps were somewhat heavy, loading samples into the machine was awkward and could have introduced failure points. Attempts to verify or control any effects were unsuccessful. Clearly the actual window will suffer no ill effects from handling and excessive transportation.

The data range is very high as indicated by the standard deviation. A high deviation leads to concerns over the validity of the tests. Engineering design and experience with existing windows should be considered along with these results. Furthermore, tests of the actual window will be the most accurate indicator of window performance.

The aluminum gaskets sustained similar deformation to previous tests. [RENK 1992]

Elongation of test samples approximately the same as previous tests. [RENK 1992]

Mylar remained undamaged, but reflected indentations from the aluminum o-ring.

LOAD CALCULATIONS FOR TEST CLAMPING FIXTURE:

A. Szymulanski 12/18/93

ANSYS: At 45 psig (Safety factor of 3)

T_X =296280 lbf: see Figure 1.

T_X per linear inch of the circumference:

$$T_{x_1} = \frac{296280}{\Pi(71)} = 1328.2 \quad \text{lb}$$

For a 9.858 inch wide fixture:

T_X = 13094

ANSYS: At 14.7 psia (vacuum operating conditions)

T_X =154880 lbf.

T_X per linear inch of the circumference:

$$T_{x_2} = \frac{154880}{\Pi(71)} = 694.36$$

For 9.858 wide fixture:

T_X = 6845 lbf

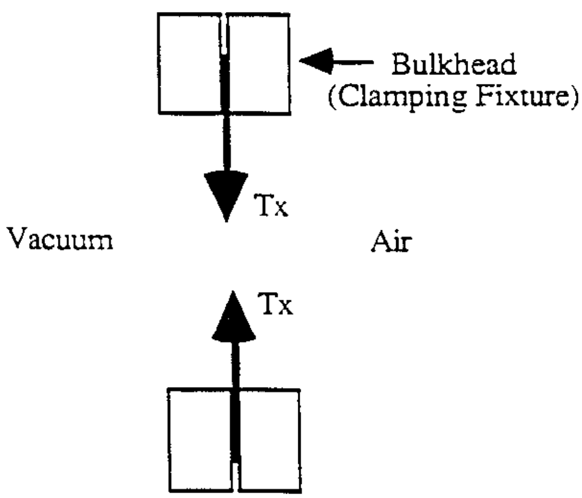


Figure 1: Cross Section of Bulkhead with force Tx exerted by fabric.

REFERENCES:

[RENK 1992] Renken, Frederick. "Kevlar Window Tests" 12/23/93 unpublished, submitted to A. Szymulanski and J. Kilmer.

[SZYM 1993] Szymulanski, Andrew. "The 48 Inch Vacuum Window Component Inspection and its Impact on 1.8m Window Analysis." unpublished, a memo to J. Misek

Kev2.7/2/93

Fred, Renken

2-Jul-93

Kevlar Test 2

Parallel count: 34

Perpendicular count: 35

Maximum Load 13.05 Kips

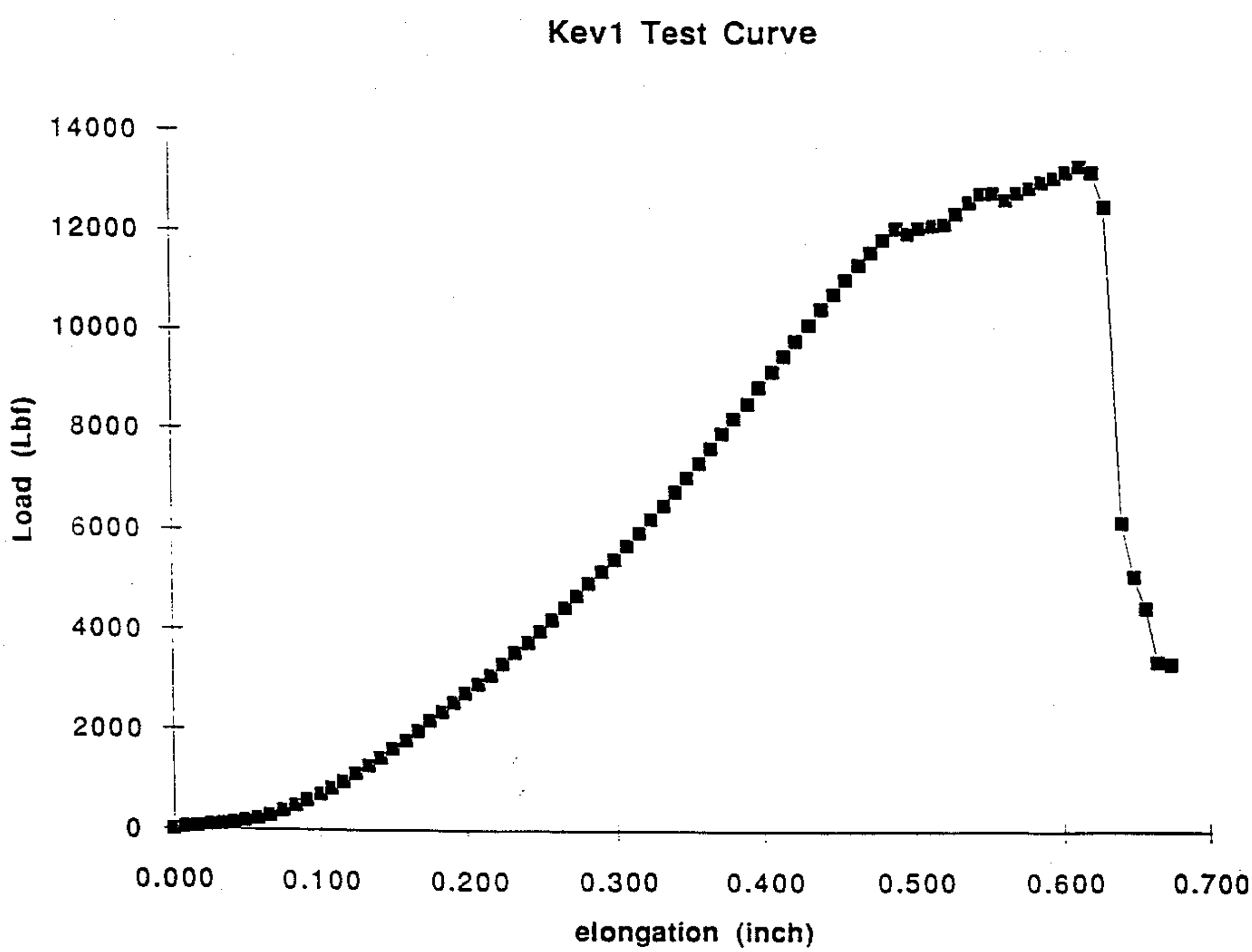
First Slip 12.9 kips

Standard Const—250 Ft•Lbs Torque

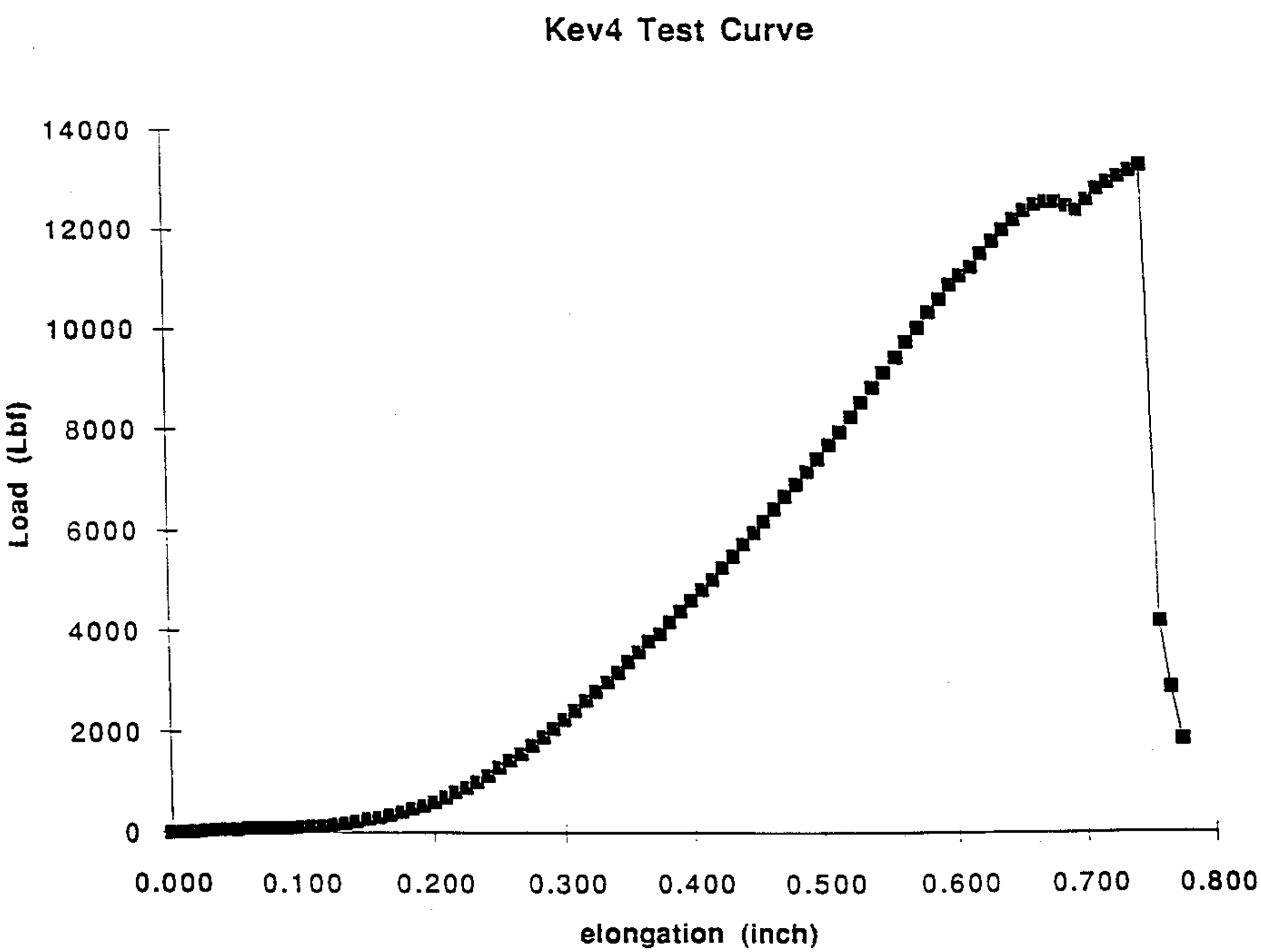
Computer failure resulted in loss of data curve.

Kev1.6/30/93

	Fred Renken				
	30-Jun-93				
	Kevlar Test 1				
	Parallel count: 34				
	Perpendicular count: 35				
	Maximum Load 13.4 Kips				
	First Slip 12.8 kips				
	Standard Const—250 Ft•Lbs Torque				

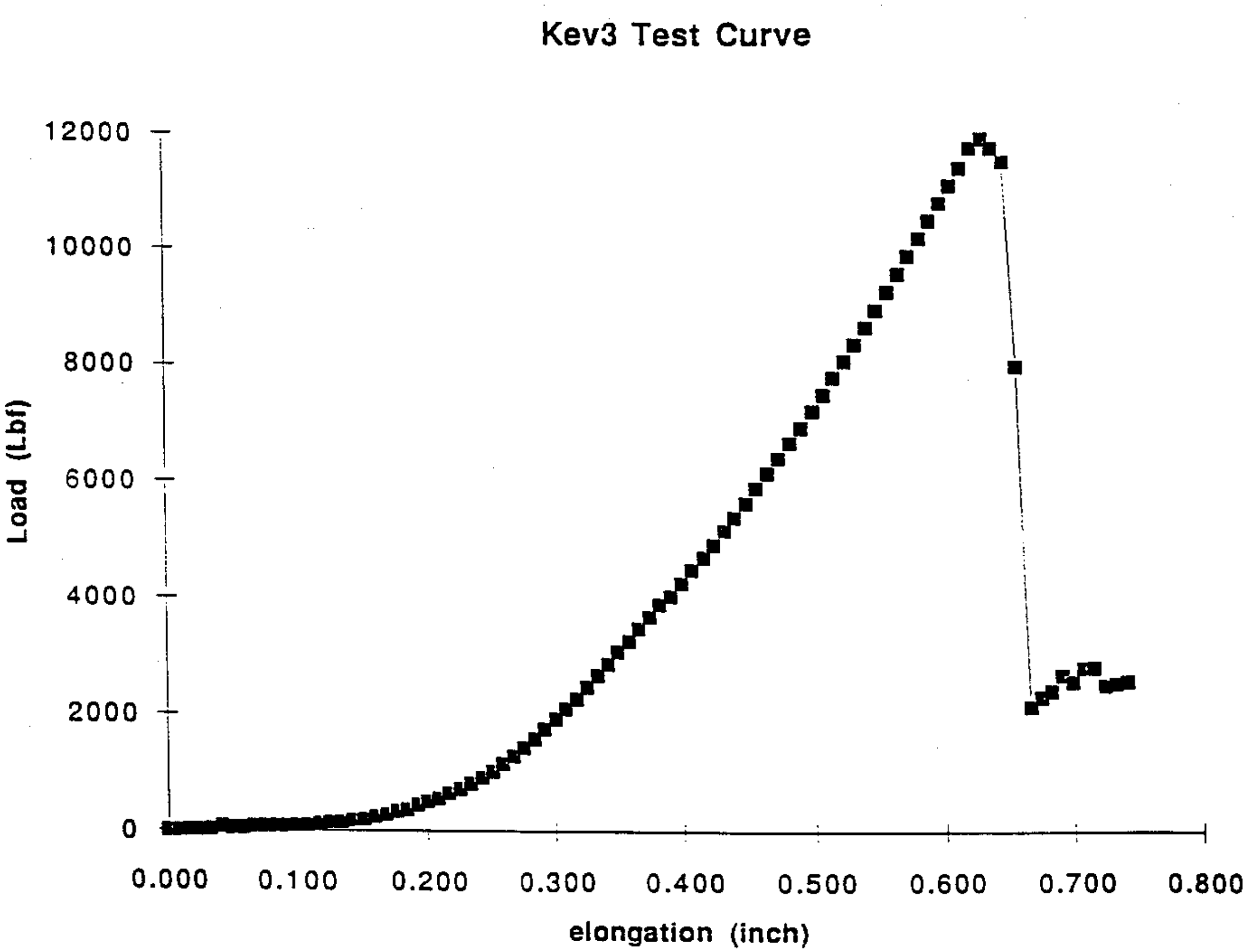


ion column						
	Fred Renken					
	9-Jul-93					
	Kevlar Test 4					
	Parallel count: 34					
	Perpendicular count: 35					
	Maximum Load 13.33 Kips					
	First Slip 12.5 kips					
	Standard Const—250 Ft•Lbs Torque					



Kev3.7/6/93

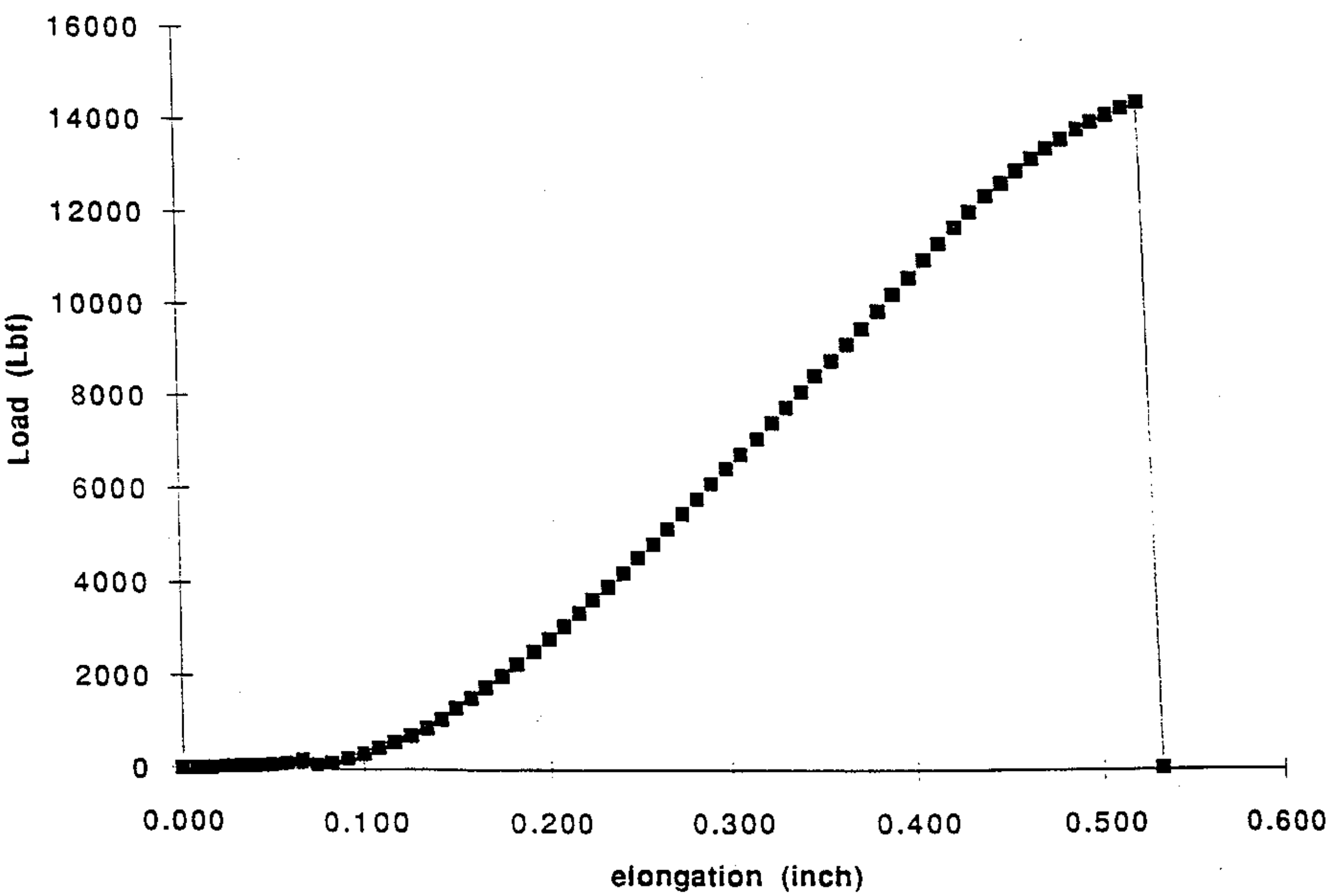
Fred Renken				
6-Jul-93				
Kevlar Test 3				
Parallel count: 34	Orientation to test fixture			
Perpendicular count: 35				
Maximum Load 11.99 Kips				
First Slip 11.9 kips				
Standard Const—250 Ft•Lbs Torque				



Kev6.7/14/93

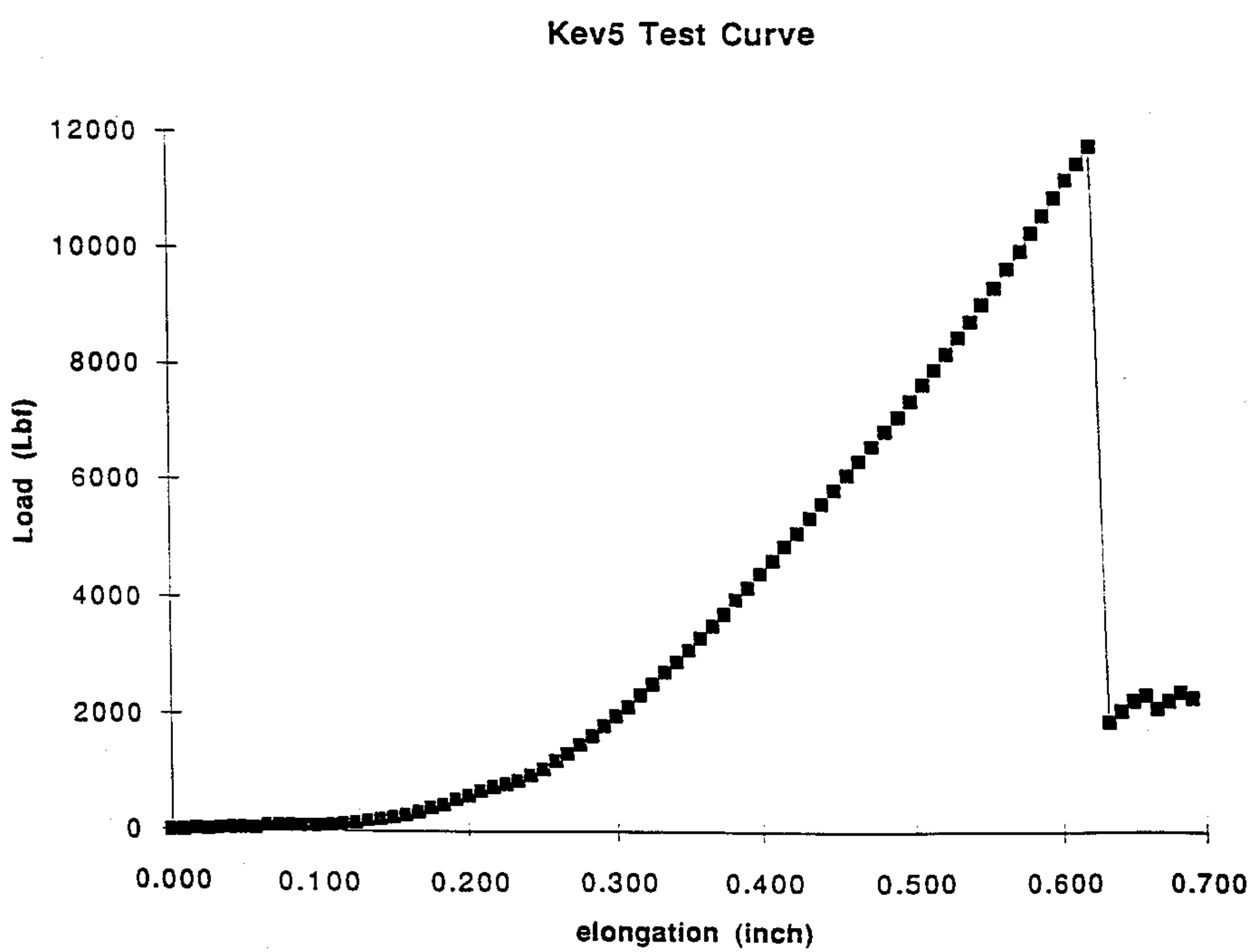
	Fred Renken				
	14-Jul-93				
	Kevlar Test 6				
	Parallel count: 35				
	Perpendicular count: 34				
	Maximum Load 14.51 Kips				
	First Slip 14.5 kips				
	Standard Const—250 Ft•Lbs Torque				

Kev6 Test Curve



Kev5.7/12/93

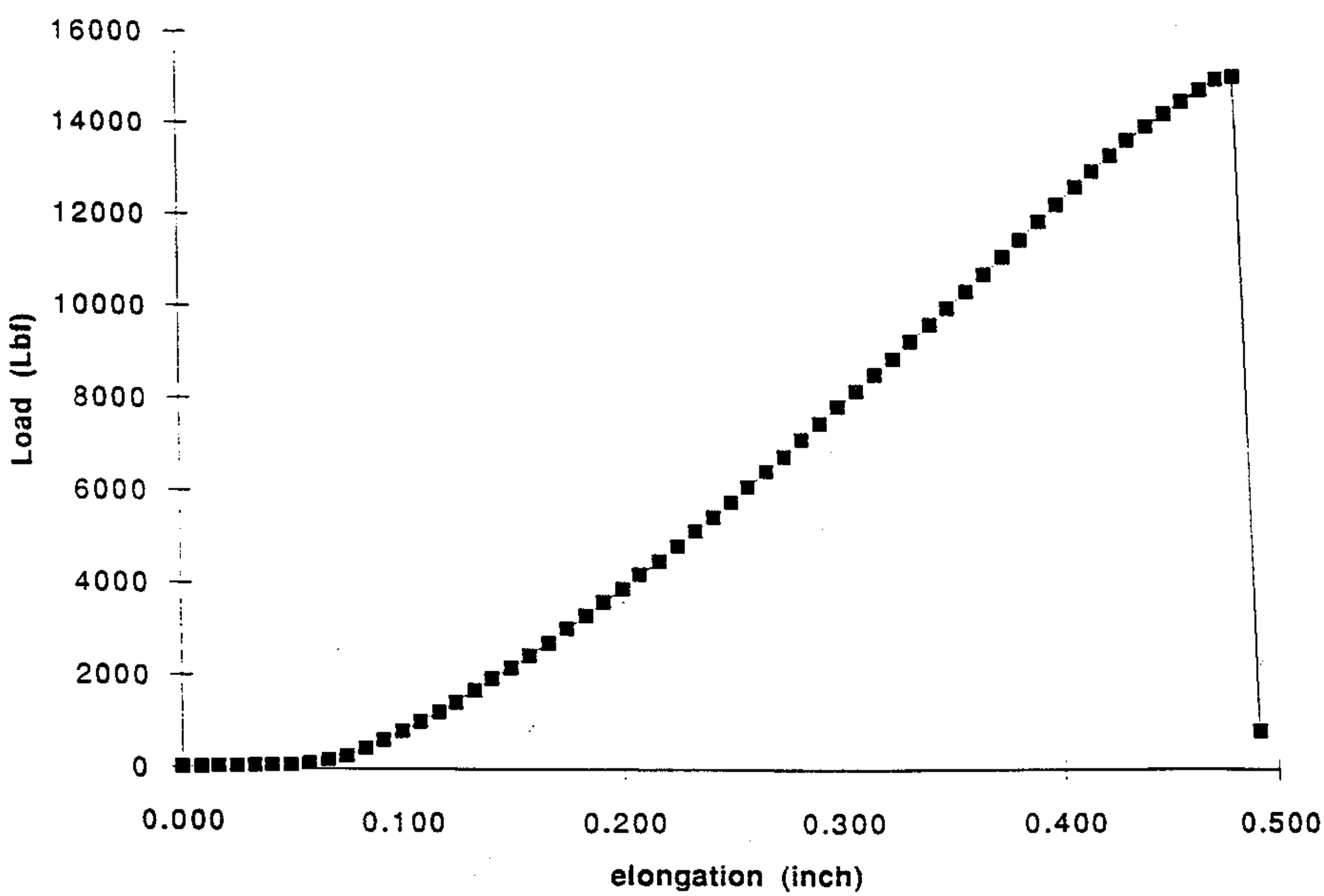
	Fred Renken				
	12-Jul-93				
	Kevlar Test 5				
	Parallel count: 34				
	Perpendicular count: 35				
	Maximum Load 11.96 Kips				
	First Slip 11.9 kips				
	Standard Const—250 Ft•Lbs Torque				



Kev8.7/19/93

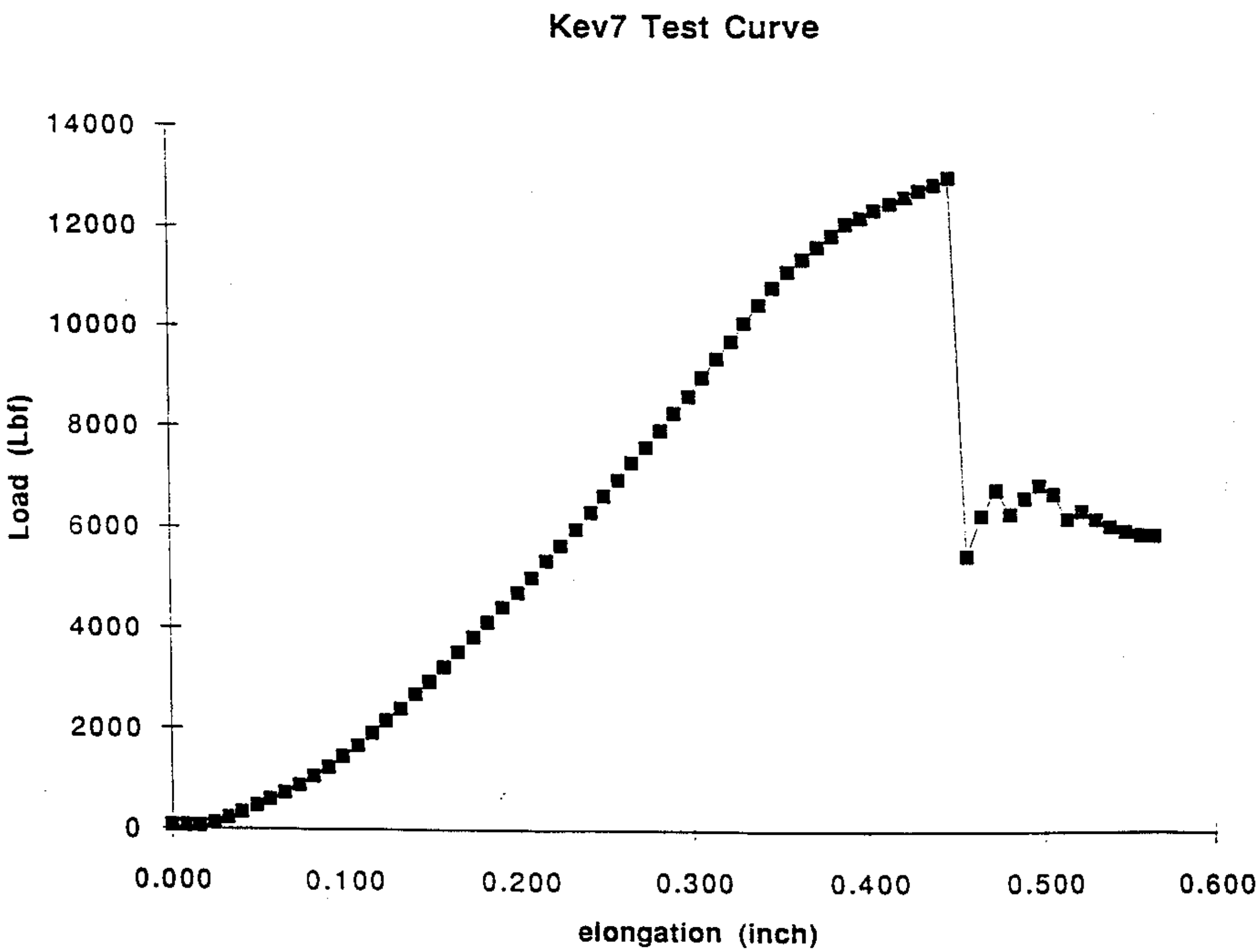
	Fred Renken				
	19-Jul-93				
	Kevlar Test 8				
	Parallel count: 35				
	Perpendicular count: 34				
	Maximum Load 15.25 Kips				
	First Slip 15.25 kips				
	Standard Const—250 Ft•Lbs Torque				

Kev8 Test Curve



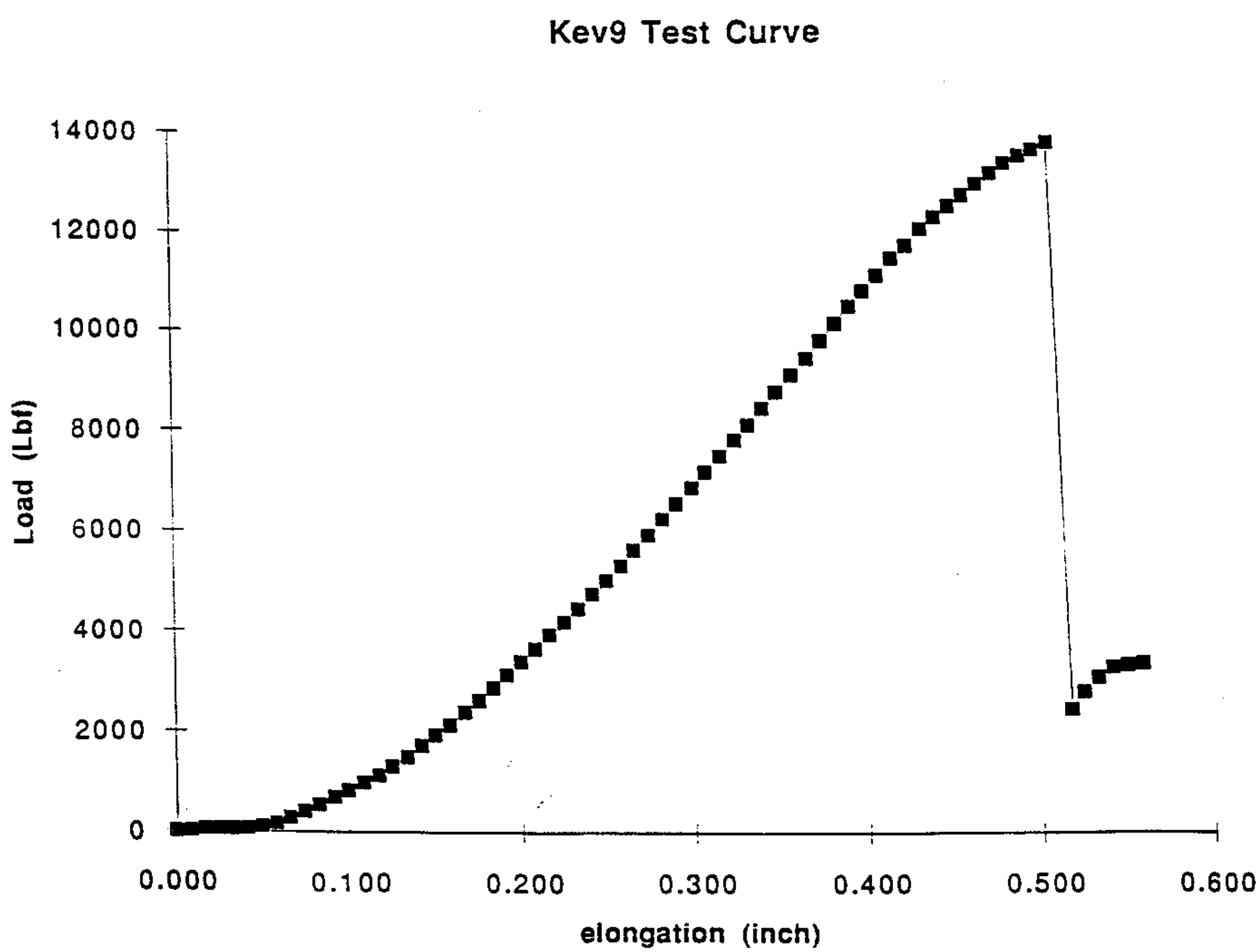
Kev7.7/16/93

	Fred Renken
	16-Jul-93
	Kevlar Test 7
	Parallel count: 35
	Perpendicular count: 34
	Maximum Load 13.1 Kips
	First Slip 13.1 kips
	Standard Const—250 Ft•Lbs Torque



Kev9.7/21/93

	Fred Renken					
	21-Jul-93					
	Kevlar Test 9					
	Parallel count: 35					
	Perpendicular count: 34					
	Maximum Load 13.96 Kips					
	First Slip 13.9 kips					
	Standard Const—250 Ft•Lbs Torque					



ANALYSIS OF TENSILE TEST DATA
Jim Kilmer
May 25, 1995

This papers presents some analysis and comment on the December, 1992 tensile tests of the Kevlar fabric used in making the 71" vacuum windows for KTEV. The main goal of the test was to determine the clamping force needed to hold the window by comparing tensile tests of samples of the material in grippers modeled like the actual design flanges. A secondary goal would have been to look for the maximum tensile strength of the material. Samples were tested with the fibers on a 45 degree angle to the axis of the force to understand the strength of the weave better. The Instron testing machine in the Materials Development Lab was used for these tests.

Calculations showed that the minimum force the clamping has to support is 6845 pounds force. The tests showed that 250 ft-lbs of torque on the flange bolts would be able to support the needed force. Nine tests were done at 250 ft-lbs of torque with the flanges exactly as the final design and all passed the clamping test. See the attached spreadsheet. The percent elongation at failure of the fabric sample was found to be 50% higher than the published data from DuPont for Kevlar fibers alone. This discrepancy can be explained by the fact that for our tests the fabric does not strain at the same rate as single fibers. The extra length of fibers because of the weave makes the simple measurement of sample length only an approximation which could be in error by as much as 50%.

The spreadsheet also calculates the load per inch of fabric to compare with the published data of the fabric manufacturer. The manufacturer uses ASTM D-1682 to test for breaking loads and elongation of fabrics. We have no elongation figures for the fabric from the manufacturer but they do publish the strength. Their number for this fabric is 1800 pounds force per inch of fabric. If our data set is analyzed the measured number is 966 pounds per inch. The difference is in the samples and tests. In ASTM D-1682 a one inch wide sample is put between grippers that typically grip harder as the tensile force rises and does not require bolts to supply clamping force. The clamping system in our tests does not supply the same level of grip on the samples. In our sample testing scheme what happens is that all of the fibers in the areas of the bolts (50% of the total sample) are not gripped hard enough to test to failure. The aluminum ring is meant to distribute the load of clamping to the areas between the bolts but that force is limited to the amount provided by a specific level of bolt torque. At some level of testing force the fibers can still slip past the aluminum ring. The best example we have of that is test number 21 which didn't use the aluminum o-ring but used the entire flange area for clamping. In that sample it is clear that only 1/2 of the fibers are participating in the test. If the number of LOADED fibers is compared then our tests give a load per inch of 1932 pounds per inch which is comparable to the published data for the fabric. We have kept many of the samples for later visual inspection.

The success of these tests has been indicated by the fact that in none of the later pressure or creep tests has any window shown a tendency to pull out from between the flanges. That was the principal reason for the tests. The secondary reason of measuring the strength of the fabric has show results in agreement with the published values of the manufacturer.

The conclusions above for the 1992 tests are also supported by the July, 1993 testing. The major difference in the two rounds of tests is that in the second round of tests epoxy was used to better bind the fabric into a more continuous sample, and minimize the effects of the short fibers in the bolt areas. The epoxy was successful in that as seen by the universally higher failure loads in the second set of samples. All of the windows in later pressure testing have been made with the epoxy bond on the circumference.