TORQUE VALUE FOR BOLTS IN LBS FEET.

$T = \frac{F\,D\,K}{12}$ WHERE F = TENSION IN BOLT.

= 75% PROOF LOAD × STRESS AREA.

D = NOMINAL DIA. OF BOLT.

K = CONSTANT = ·2 .

STRESS AREA $= \frac{\pi\,(\text{PITCH DIA.} + \text{ROOT DIA.})^2}{16}$

75% PROOF LOAD R = 57200 S = 63800 T = 68900 V = 83200.

NOMINAL DIA. OF THREAD	U.N.F.					U.N.C.				
	STRESS AREA SQ. INS.	CODE SYMBOL				STRESS AREA SQ. INS.	CODE SYMBOL			
		R	S	T	V		R	S	T	V
$\frac{1}{4}$	·0362	8·6	9·6	10·4	12·5	·0317	7·5	8·4	9	11
$\frac{5}{16}$	·0579	17	19	20·5	25	·0522	15·5	17·2	18·7	22·5
$\frac{3}{8}$	·0876	31·4	35	38	46	·0773	27·5	30·5	33	40
$\frac{7}{16}$	·1185	50	56·5	60	73	·1060	45	50	54	65
$\frac{1}{2}$	·1597	75	84	90	109	·1416	67	75	81	97
$\frac{9}{16}$	·2026	108	120	128	157	·1816	98	109	118	142
$\frac{5}{8}$	·2555	152	170	183	220	·2256	134	148	161	195
$\frac{3}{4}$	·3724	266	297	320	387	·3340	236	263	285	343

FOR CADMIUM PLATED NUTS USE 75% OF TORQUE FOR STOVER NUTS USE 60% OF TORQUE.

NOMINAL DIA. OF THREAD	B.S.F.				B.S.W.			
	STRESS AREA SQ INS	CODE SYMBOL			STRESS AREA SQ INS	CODE SYMBOL		
		R	T	V		R	T	V
$\frac{1}{4}$	·0356	8·5	10·1	12·2	·0317	7·5	9	11
$\frac{5}{16}$	·0566	16·8	20·3	24·5	·526	15·7	18·9	22·8
$\frac{3}{8}$	·084	30	36	43·5	·780	28	33·5	40·5
$\frac{7}{16}$	·116	48·5	58	70	·103	43	52	63
$\frac{1}{2}$	·152	72	87	105	·138	66	80	96
$\frac{9}{16}$	·198	106	128	154	·182	98	118	142
$\frac{5}{8}$	·250	149	179	206	·225	134	161	195

GEARING.

1. SINGLE HELICAL GEAR CALCULATIONS.

(TAKEN FROM F. SHAWS MEMO 6.7.60)

WHEN IT IS REQUIRED TO PRODUCE NEW GEARS WITH EXISTING GUIDES AND CUTTERS THE FOLLOWING METHOD MUST BE ADOPTED :-

1. CALCULATE THE NEW P.C.D IN THE NORMAL WAY.
2. USE THE ADDENDUM AND DEDENDUM FROM THE EXISTING GEAR.
3. USE THE TOOTH CIRCULAR THICKNESS FROM THE EXISTING GEAR.

HELIX ANGLE.		HELIX ANGLE.
40° 14′ 11″		26° 31′ 55″
0·090″	ADDENDUM.	0·105″
0·226″	WHOLE DEPTH.	0·256″
0·137″	CIRC. THICK.	0·161″
11·1345	NORMAL PITCH.	9·5005

4. RE-CALCULATE THE CHORDAL HEIGHT AND CHORDAL THICKNESS CORRESPONDING TO THE NEW P.C.D.
5. CALCULATE THE LEAD FROM THE NEW P.C.D, AND THE EXISTING HELIX ANGLE.

ALL THE OTHER PARTICULARS WILL BE THE SAME AS THE EXISTING GEARS EXCEPT WHEN CORRECTION IS NECESSARY AS FOLLOWS :-

1. CHECK FOR CORRECTION IN THE NORMAL WAY.
2. IF CORRECTION IS REQUIRED ALTER THE ADDENDUM AND DEDENDUM WITH OPPOSITE SIGNS SO THAT THE WHOLE DEPTH IS THE SAME AS THE EXISTING GEAR. (THE LONG ADDENDUM IS ON THE PINION)
3. THIS WILL ALTER THE CIRCULAR THICKNESS THUS:-

TOOTH THICKNESS CHANGE = TWICE THE CORRECTION X TAN PRESSURE ANGLE.

TOOTH THICKNESS PINION = $\frac{P}{2}$ + TT CHANGE - BACKLASH-ALLOWANCE.

TOOTH THICKNESS GEAR = $\frac{P}{2}$ - TT CHANGE - BACKLASH ALLOWANCE.

N.B. TWICE THE CORRECTION MEANS THE CHANGE IN OUTSIDE DIAMETER.

4. NOW CALCULATE CHORDAL THICKNESS AND CHORDAL HEIGHT IN NORMAL WAY USING THE MODIFIED TOOTH CIRCULAR THICKNESSES.

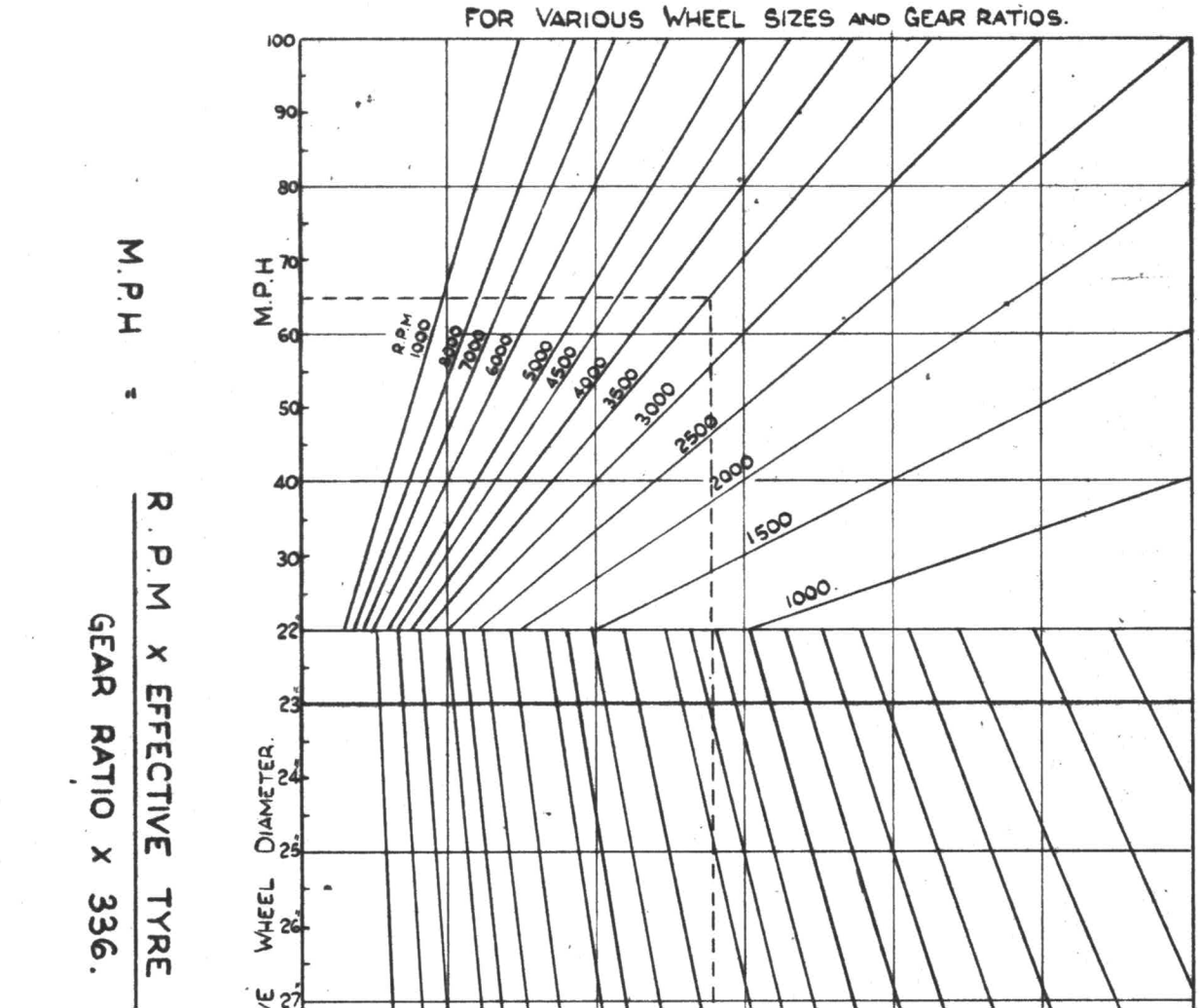
43. PERFORMANCE CHARTS R.P.M / M.P.H.

1. REVS. PER MINUTE (ENGINE) TO MILES PER HOUR AND VISA-VERSA.

THE FOLLOWING CHART CAN BE USED FOR THE RELATIONSHIP BETWEEN R.P.M OF ENGINE AND ROAD SPEEDS. IT CAN ALSO BE USED FOR THE ESTIMATION OF THE EFFECTS OF CHANGES IN TYRE SIZE AND GEAR RATIO ALTERATIONS.

TO USE THE CHART, FIRST LOCATE THE POINT ON THE EFFECTIVE TYRE DIAMETER SCALE AND FROM THIS PROJECT A LINE HORIZONTALLY TO MEET THE APPROPRIATE GEAR RATIO DIAGONAL. FROM THIS POINT PROJECT A LINE VERTICALLY TO CUT THE R.P.M LINES ON THE UPPER CHART. PROJECT THIS INTERSECTION POINT HORIZONTALLY TO M.P.H SCALE.

EXAMPLE :- WITH A 28" TYRE EFFECTIVE DIAMETER AND 4·5 TO 1 GEAR RATIO AND ENGINE SPEED OF 3,500 R.P.M, ROAD SPEED = 64·8 M.P.H.

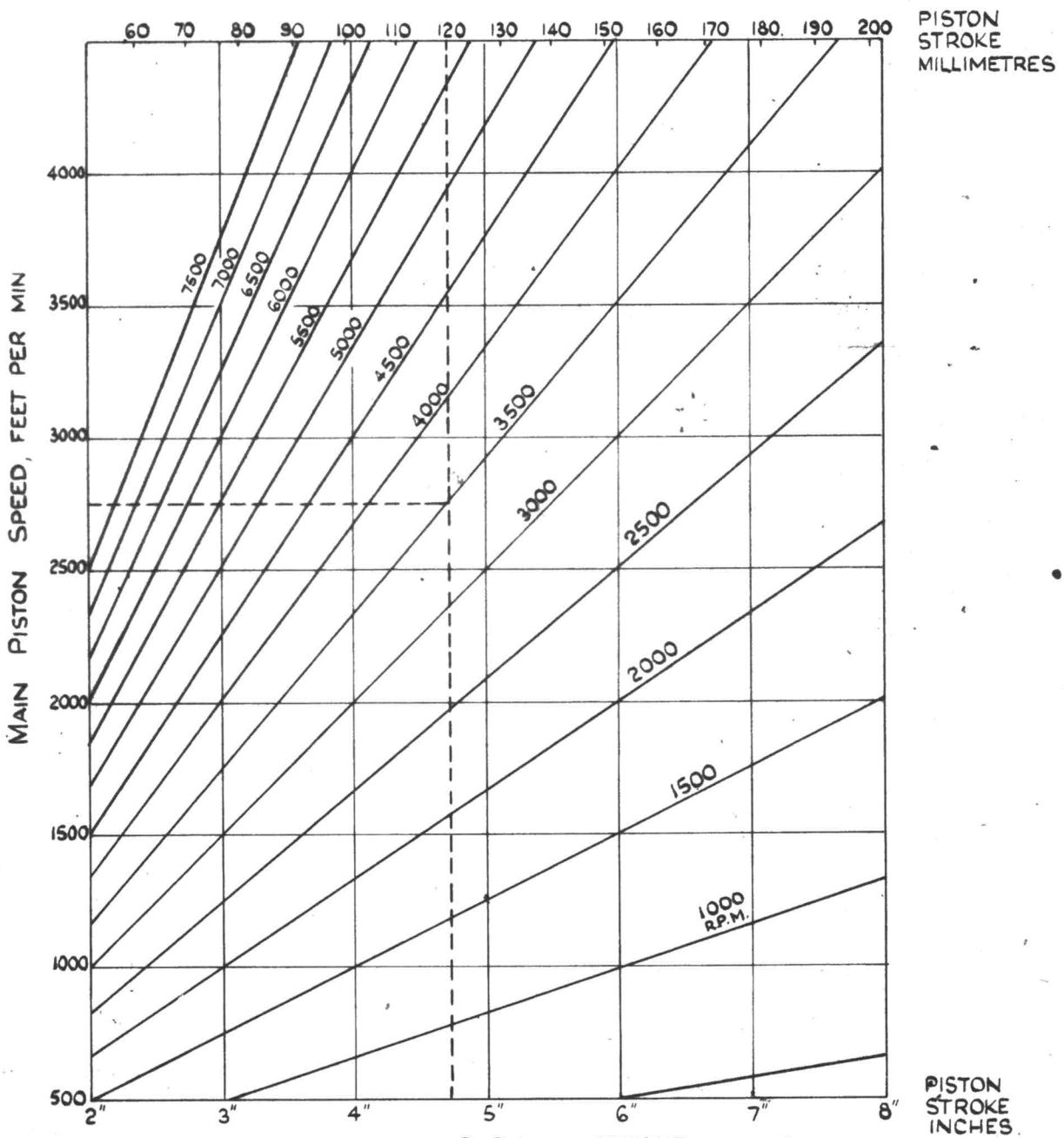


2. REVS. PER MINUTE (ENGINE) TO PISTON SPEED

THE FOLLOWING CHART ENABLES THE RELATIONSHIP BETWEEN PISTON SPEED AND R.P.M FOR ANY ENGINE TO BE OBTAINED WITHOUT CALCULATION.

TO USE THE CHART, FIRST LOCATE THE POINT ON THE PISTON STROKE SCALE AND PROJECT A LINE VERTICALLY TO MEET THE APPROPRIATE R.P.M DIAGONAL LINE. FROM THIS POINT PROJECT A LINE HORIZONTAL TO MEET THE MEAN PISTON SPEED SCALE.

EXAMPLE :- AN ENGINE WITH A STROKE OF 4·72 INS. (120 MMS.) AND R.P.M. OF 3,500 WILL GIVE A MEAN PISTON SPEED OF 2,750 FEET PER MINUTE.

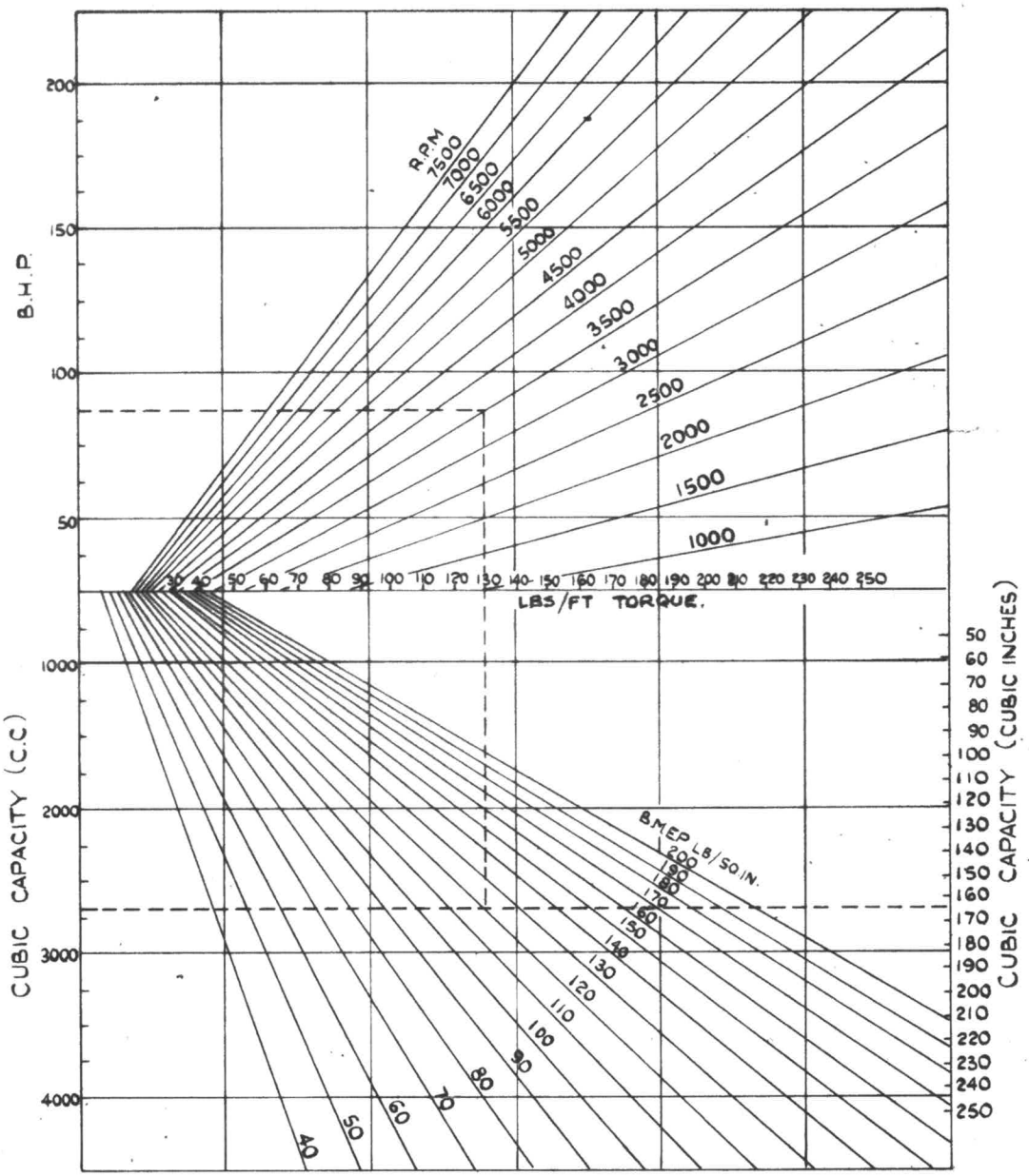


THE FOLLOWING CHART LINKS TOGETHER CUBIC CAPACITY, B.M.E.P, TORQUE, R.P.M. AND BRAKE HORSE - POWER.

THE METHOD OF USING THIS CHART IS BEST SEEN BY STUDY OF THE PLOTTED EXAMPLE :- AN ENGINE OF 2,700 cc DISPLACEMENT WITH A B.M.E.P OF 120 LB/SQ.IN. WILL GIVE A TORQUE OF 130 LB/FT. AT A SPEED OF 3,500 R.P.M, THIS TORQUE GIVES AN OUTPUT OF 87 B.H.P

THE ORDER OF PROCEDURE USED IN THE EXAMPLE MAY BE VARIED ACCORDING TO THE INFORMATION AVAILABLE OR REQUIRED.

GIVEN TORQUE OR B.M.E.P AND CUBIC CAPACITY.



$$\frac{\text{B.H.P}}{} = \frac{\text{R.P.M} \times \text{TORQUE LB/FT}}{5,250.}$$

$$\frac{\text{TORQUE}}{} = \frac{\text{B.M.E.P} \times \text{SWEPT VOLUME C.C}}{2.473}$$

44. IMPACT LOADING.

'K' FACTOR FOR IMPACT LOADING.

THIS ALLOWANCE FOR LOSS OF ENERGY ON IMPACT GIVES MORE ACCURATE VALUES FOR STRESS AND DEFORMATION.

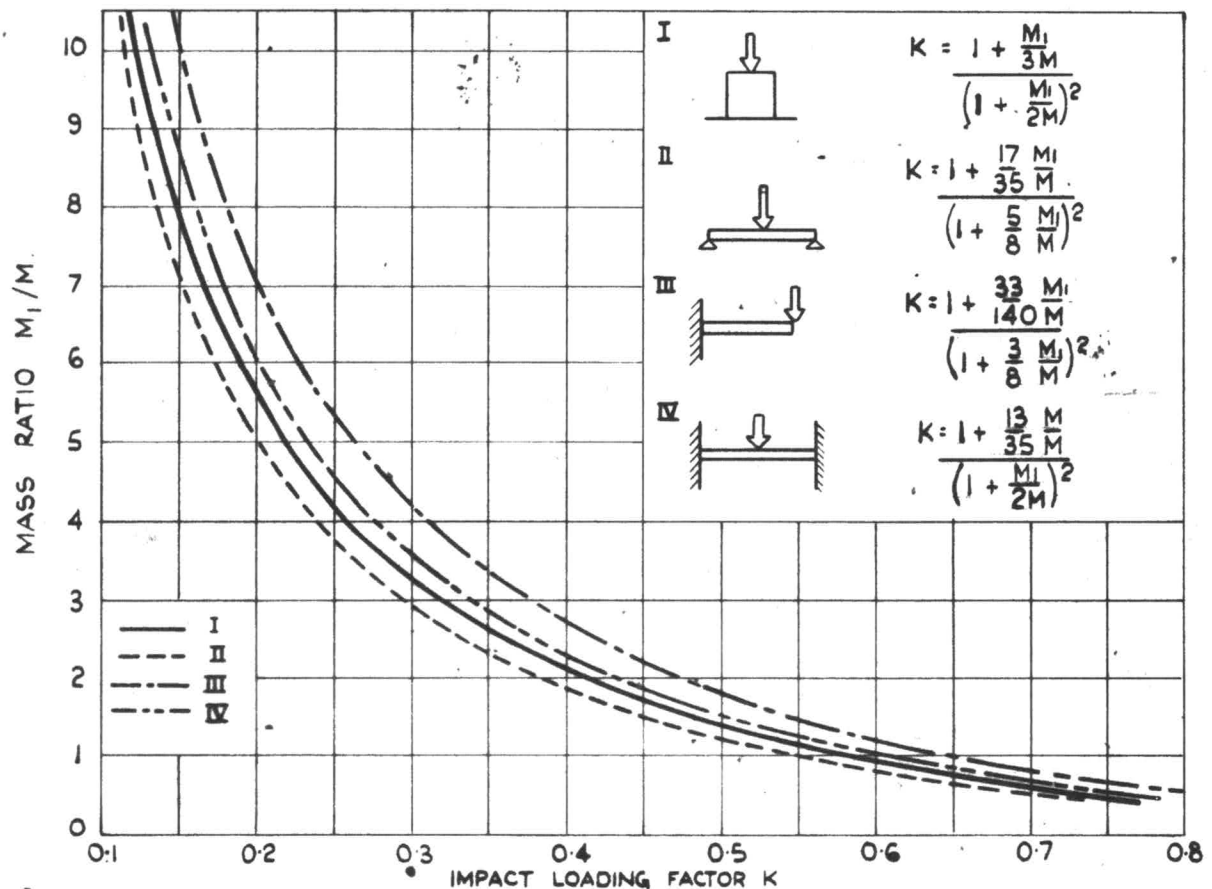
USUAL FORMULAS FOR IMPACT LOADING ASSUME COMPLETE RECOVERY OF THE KINETIC ENERGY OF THE MOVING BODY. THIS IS NOT SO AS SOME KINETIC ENERGY IS DISSIPATED. THE K FACTOR IN FOLLOWING EQUATIONS IS A MEASURE OF THIS ENERGY LOSS.

THE CORRECTED FORMULA FOR DEFORMATION AND STRESS IS :-

WITH VERTICAL IMPACT $$\frac{d_1}{d} = \frac{S_1}{S} = 1 + \sqrt{1 + 2K\frac{h}{d}}$$

WITH HORIZONTAL IMPACT

$$\frac{d_1}{d} = \frac{S_1}{S} = \sqrt{\frac{Kv^2}{384 \cdot 6\, d}}$$



d = DEFORMATION FOR STATIC LOADING INS.

d_1 = DEFORMATION FOR IMPACT LOADING INS.

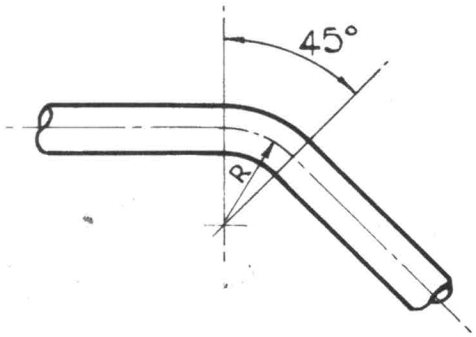
h = INITIAL HEIGHT OF STRIKING BODY ABOVE STRUCK BODY INS.

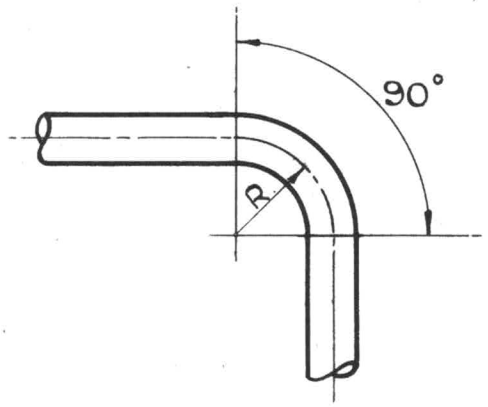
K = IMPACT LOADING FACTOR.

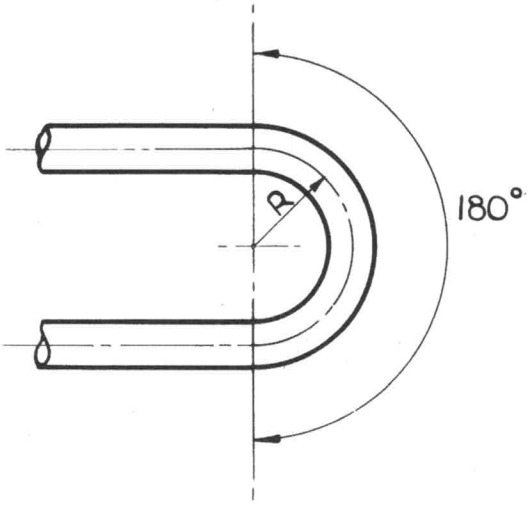
45.

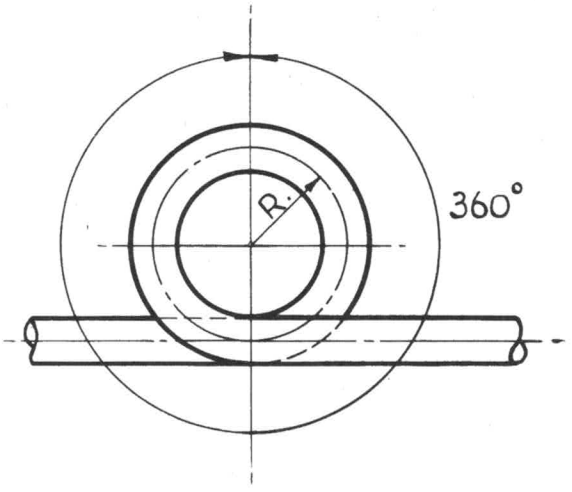
BUNDY TUBING.

MINIMUM RADII FOR BENDS.





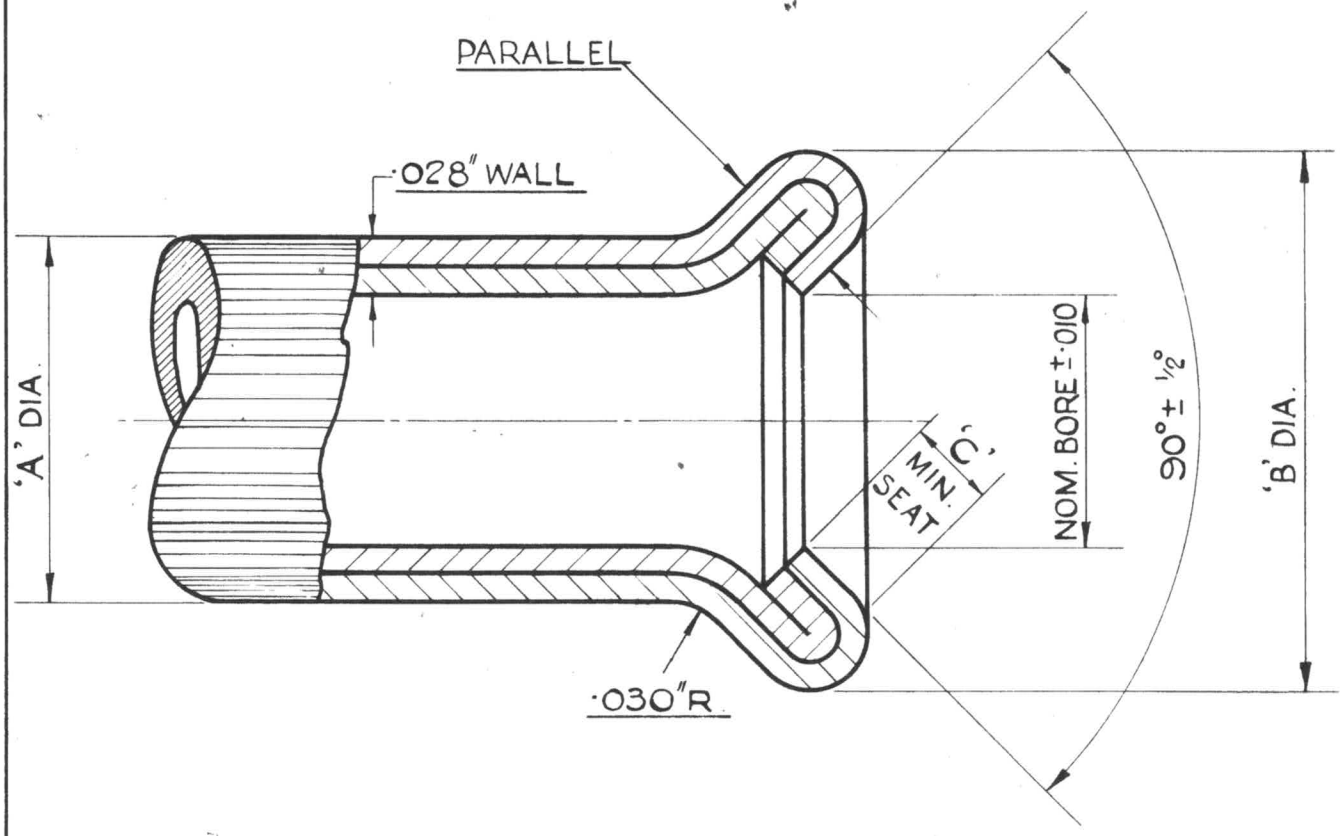




WALL THICK-NESS INS.	TUBE O/D INS.	MINIMUM RADII (R)							
		UP TO 45°		46° — 90°		91° — 180°		181° — 360°	
		HAND BEND INS.	DIE BEND INS.	HAND BEND INS.	DIE BEND INS.	HAND BEND INS.	DIE BEND INS.	HAND BEND INS.	DIE BEND INS.
·028"	1/8"	1/4"	1/8"	3/8"	1/4"	1/2"	1/4"	1/2"	1/4"
·028"	3/16"	3/8"	3/16"	3/4"	3/8"	7/8"	7/16"	7/8"	7/16"
·028"	1/4"	1/2"	3/8"	1 1/8"	1/2"	1 3/8"	5/8"	1 3/8"	5/8"
·028"	5/16"	3/4"	1/2"	1 5/8"	3/4"	1 7/8"	7/8"	1 7/8"	7/8"
·028"	3/8"	1 1/4"	5/8"	2 3/8"	15/16"	2 5/8"	1 1/8"	2 5/8"	1 1/8"
·028"	1/2"	4 3/4"	1 1/2"	4 3/4"	1 1/2"	5 1/4"	2"	5 1/4"	2"

BUNDY TUBING

FLARE ON TUBING 45°



TUBE SIZE	'A' DIA	'B' DIA	'C'
1/8"	·122" ·127"	·203" ·213"	·040"
3/16"	·184" ·189"	·270" ·280"	·040"
1/4"	·247" ·252"	·350" ·360"	·040"
5/16"	·309" ·314"	·415" ·425"	·062"
3/8"	·372" ·377"	·490" ·500"	·062"
	·497"	·630"	

A surface is the boundary of an object and is represented on a drawing by a line. In practice this surface does not possess the perfect smoothness shown by this line. Actually it is composed of "Peaks and Valleys" of varying heights and depths, which controls the degree of roughness.

These are measured by a Tally-surf machine, and the unit of this measurement is one millionth of an inch, .000001", or one micro inch. This machine records the arithmetical average, from a number of readings taken from the mean surface, to the formulae $\frac{Ya + Yb + Yc + Yd + Ye + Yn}{n}$

where Y = height of the peak, or depth of the valley. This is abbreviated to, C.L.A. (centre line average), and is the method adopted by the Rover Co.

When calling for a surface finish on the drawing, the symbol ✓ is used, the point being on the surface, and the maximum micro inch requirement is placed on the left hand of the large stroke as shown in Fig. 1.
Fig. 2 shows the approximate dimensions of the symbol.

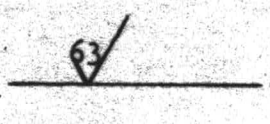


Fig. 1.

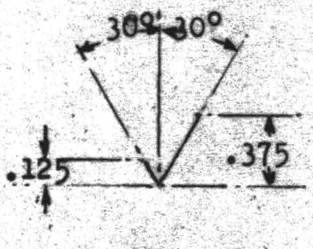
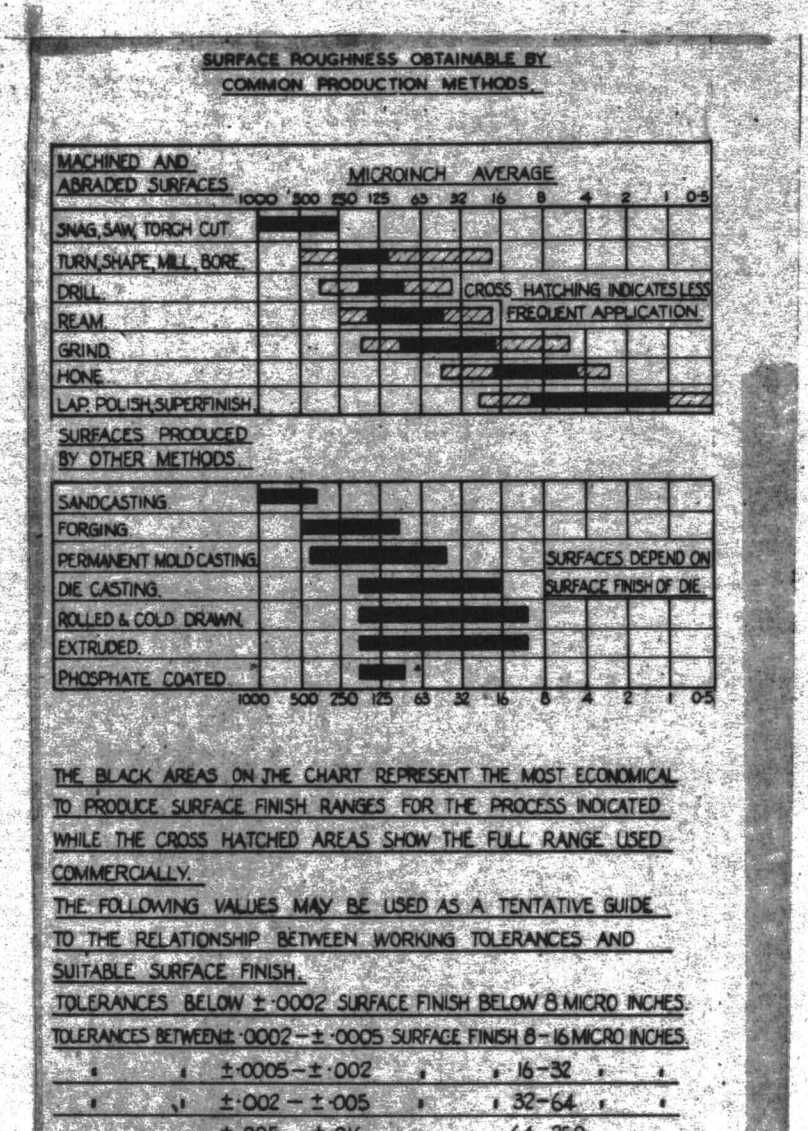
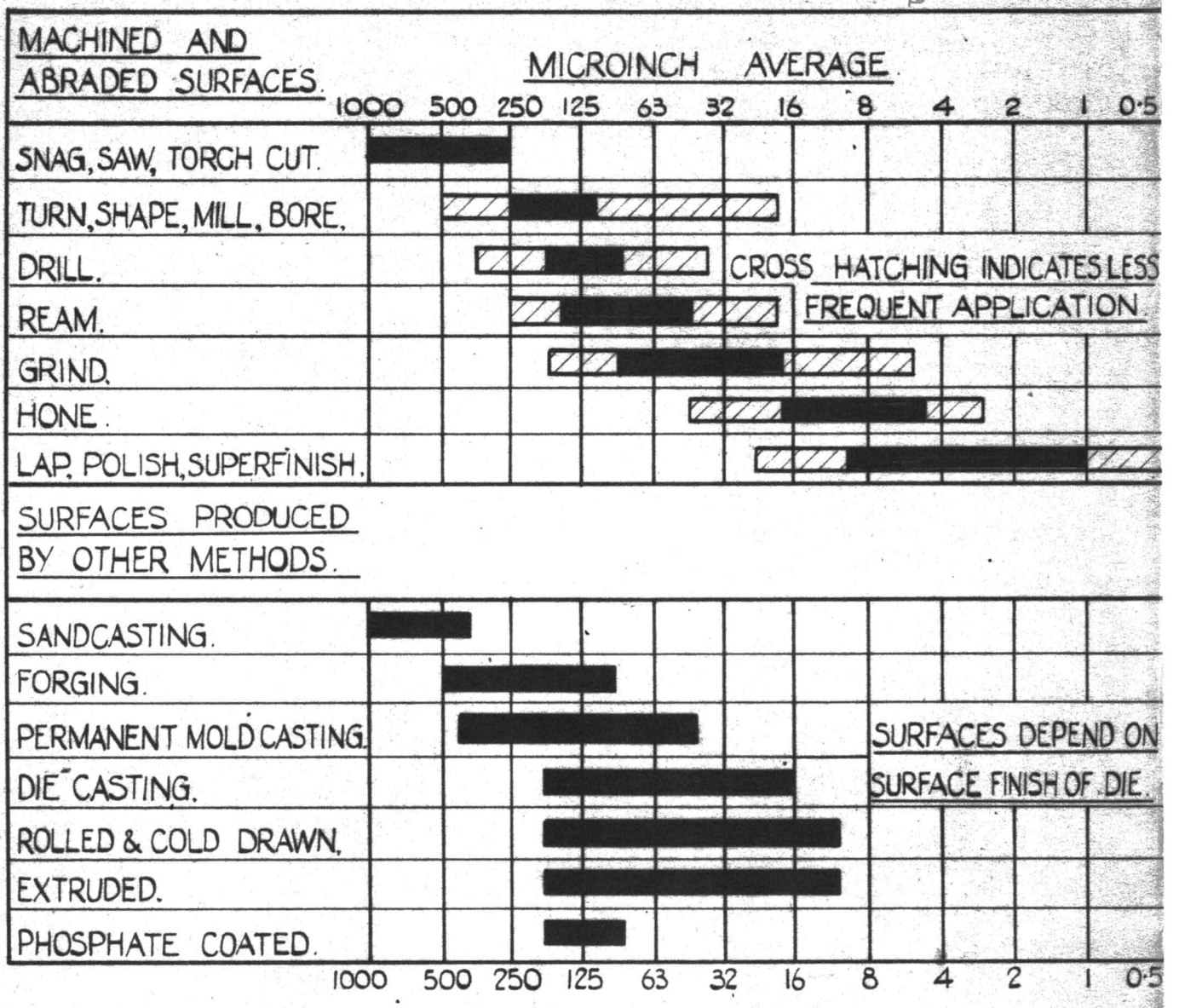


Fig. 2.

When a finish has a minimum as well as a maximum requirement then it is called for :- $\frac{63}{32}$ ✓.



COMMON PRODUCTION METHODS.



THE BLACK AREAS ON THE CHART REPRESENT THE MOST ECONOMICAL TO PRODUCE SURFACE FINISH RANGES FOR THE PROCESS INDICATED WHILE THE CROSS HATCHED AREAS SHOW THE FULL RANGE USED COMMERCIALLY.

THE FOLLOWING VALUES MAY BE USED AS A TENTATIVE GUIDE TO THE RELATIONSHIP BETWEEN WORKING TOLERANCES AND SUITABLE SURFACE FINISH.

TOLERANCES BELOW ± ·0002 SURFACE FINISH BELOW 8 MICRO INCHES.

TOLERANCES BETWEEN ± ·0002 – ± ·0005 SURFACE FINISH 8 – 16 MICRO INCHES.

"	"	± ·0005 – ± ·002	"	" 16 – 32	" "
"	"	± ·002 – ± ·005	"	" 32 – 64	" "
		± ·005 – ± ·016	"	" 64 – 250	" "

DEFLECTIONS OF BEAMS DUE TO COMBINED BENDING AND SHEAR FORCES

BEAM DEFLECTIONS DUE TO SHEARING STRESSES ARE GENERALLY SO SMALL WHEN COMPARED WITH THE BENDING DEFLECTION THAT THEY ARE USUALLY NEGLECTED IN PRACTICE. IN SHORT BEAMS WHERE THE RATIO OF SECTION DEPTH TO BEAM LENGTH IS GREATER THAN 1/10 THE SHEAR DEFLECTION BECOMES SIGNIFICANT.
THE FOLLOWING FORMULA MAY BE USED TO OBTAIN THE TOTAL DEFLECTION DUE TO COMBINED BENDING AND SHEARING FORCES.

$$y_T = \frac{WL}{K_s AG}\left[1 + \frac{1}{C}\left(\frac{L}{d}\right)^2\right]$$

THE TABLE PAGE No. 110 GIVES VALUES FOR K_s AND C. ALSO, FOR COMPARISON, THE FORMULA FOR y_s IS INCLUDED.
EXAMPLE: FIND THE MAXIMUM DEFLECTION FOR A SIMPLY SUPPORTED STEEL I-BEAM 12 FT. LONG. THE BEAM IS 36 IN. DEEP AND IS SUBJECTED TO A LOAD OF 2,000 lb PER FT. THE CROSS-SECTIONAL AREA OF THE BEAM IS 31 IN.^{2} SHEAR MODULUS IS 12.0×10^6 PSI.
SOLUTION: FROM THE TABLE (CASE 6) $K_s = 8$ AND $C = 3.84$.
THEN FROM THE FORMULA FOR TOTAL DEFLECTION.

$$y_t = \frac{2000 \times 12 \times 144}{8 \times 31 \times 12 \times 10^6}\left[1 + \frac{1}{3.84}\,(144/36)^2\right]$$

$$= 0.006 \text{ IN.}$$

THE DEFLECTION DUE TO SHEARING ALONE WOULD BE:—

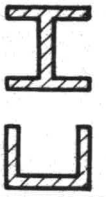
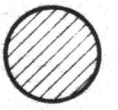
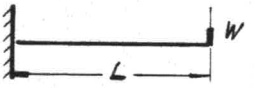
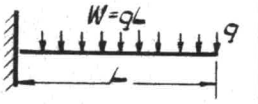
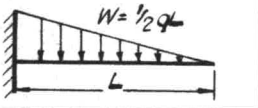
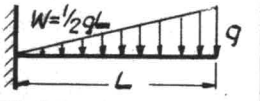
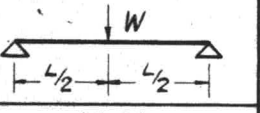
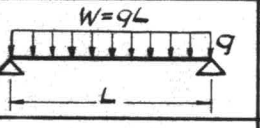
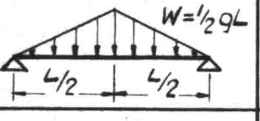
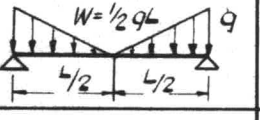
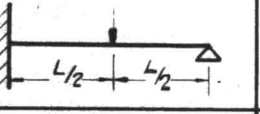
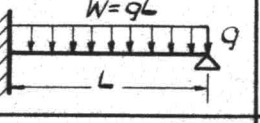
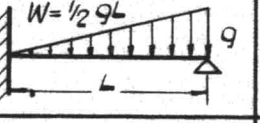
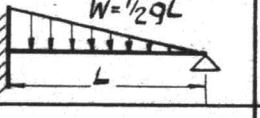
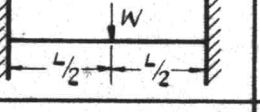
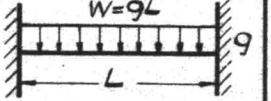
$$y_s = \frac{WL}{8AG} = \frac{2000 \times 12 \times 144}{8 \times 31 \times 12 \times 10^6}$$

$$= 0.00117 \text{ IN.}$$

THIS IS ABOUT 20% OF THE TOTAL DEFLECTION AND WOULD REPRESENT A SIZABLE ERROR IF NOT INCLUDED IN THE TOTAL DEFLECTION.

NOTATION:

A = CROSS-SECTIONAL AREA, IN.^{2}
C = CONSTANT (SEE TABLE)
d = DEPTH OF BEAM, IN.
G = MODULUS OF ELASTICITY FOR SHEAR. PSI
L = SPAN OF BEAM, IN.
q = UNIT LOAD, lb/IN.
W = TOTAL APPLIED LOAD, lb.
y_s = MAXIMUM SHEARING DEFLECTION. IN

LOADING STYLE AND SUPPORT POSITION		SECTION SHAPE				
		ALL SECTIONS				
		Y_s	K_s	C, FACTOR		
1. CANTILEVER, END LOAD.		$\frac{WL}{AG}$	1	1.20	0.75	0.46
2. CANTILEVER, UNIFORM LOAD.		$\frac{WL}{2AG}$	2	1.60	1.00	0.62
3. CANTILEVER, TRIANGULAR LOAD.		$\frac{WL}{3AG}$	3	2.00	1.25	0.77
4. CANTILEVER TRIANGULAR LOAD.		$\frac{WL}{1.5AG}$	1.5	1.56	0.91	0.56
5. END SUPPORTS, CENTER LOAD.		$\frac{WL}{4AG}$	4	4.80	3.00	1.86
6. END SUPPORTS, UNIFORM LOAD.		$\frac{WL}{8AG}$	8	3.84	2.40	1.49
7. END SUPPORTS, TRIANGULAR LOAD.		$\frac{WL}{6AG}$	6	4.00	2.50	1.55
8. END SUPPORTS TRIANGULAR LOAD.		$\frac{WL}{12AG}$	12	3.56	2.22	1.38
9. ONE END FIXED, ONE END SUPPORTED, CENTER LOAD		$\frac{WL}{6.4AG}$	6.4	6.71	4.19	2.60
10. ONE END FIXED, ONE END SUPPORTED, UNIFORM LOAD.		$\frac{WL}{14.2AG}$	14.2	5.21	3.25	2.02
11. ONE END FIXED, ONE END SUPPORTED, TRIANGULAR LOAD		$\frac{WL}{11.8AG}$	11.8	5.56	3.47	2.15
12. ONE END FIXED, ONE END SUPPORTED, TRIANGULAR LOAD		$\frac{WL}{16.7AG}$	16.7	5.03	3.14	1.95
13. BOTH ENDS FIXED, CENTER LOAD.		$\frac{WL}{8AG}$	8	9.60	6.00	3.72
14. BOTH ENDS FIXED UNIFORM LOAD.		$\frac{WL}{24AG}$	24	6.4	4.00	2.48