THE MODEL ENGINEER'S HANDBOOK

Third Edition

Tubal Cain

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AUTHOR'S PREFACE

Over the years I, like most engineers (amateur or professional) have collected a considerable number of handbooks, pocket-books, data sheets and reference tables. Even so, it is not unknown for a piece of wanted information to be found in none of them, and most unusual to find any information which is lodged in all. For the amateur engineer this is more than frustrating, for he seldom has access to the libraries and reference services open to the professional. This I know full well, for some years ago I 'retired' and left just such a source of knowledge – colleagues as well as books – and found the loss serious. To make matters worse, the Public Library, which can be relied upon to obtain almost any book within a few days, is a gallon of petrol away; no mean distance these days!

So this Handbook was born. It started as a number of entries in my private log-book (the 'little black book' which every engineer of my generation carried about with him) which now runs into three volumes. Later, some of the data was assembled onto charts to hang in the workshop. Finally it seemed a good idea to put most of this, as well as other more accessible data, into a book which would serve the modelling fraternity as a whole.

I must emphasise that the book has been written *for amateurs*. I have tried to provide simple explanations for some of the figures, so that those who use them may adjust them to suit their own needs; and if some of my former colleagues raise an eyebrow at what they may certainly consider to be over-simplification, I would remind them that butchers, bakers, and candlestick-makers could easily blind the average production engineer with science, and it is precisely *for* these 'professionals of other fields' that this book is written.

Much of the data presented have been transcribed 'from source' and some have been recalculated from first principles. Inevitably there will be a few errors – I am a very amateur typist. Notes of these, sent to me through the publishers, will be much appreciated. So will suggestions for the inclusion of matter not presently covered, within reason! Much that I would have liked to include had to be pruned to keep the first edition within bounds of space and cost. The interests of model engineers are as wide as are their occupations, and to cover all of these would demand an encyclopaedia. However, I hope there will be something within these pages to interest all tastes and to be of use to most.

The help and advice I have received from engineering firms have been considerable but I hope my friends in the industry will forgive me for not acknowledging them all by name. To mention a few would be invidious – to list them all would require a miniature Kelly's Directory. The debt we owe to our 'full-scale' friends is deep indeed; we use the machines they make to model the engines they build, but that is but part of the story. For we also depend on the information and knowledge which they provide, often obtained only after considerable time and expense spent on experiment and research in their works. I am deeply grateful for their interest and their help.

I must also acknowledge the help and encouragement I have had from Martin Evans, former Editor of the *Model Engineer*, and from Prof. Dennis Chaddock, whose critical observations on some of the sections when in draft form were most helpful. Above all, I owe much to my wife, whose toleration of swarf migrating from the workshop over the

years has had to be extended to accept the clatter of the typewriter, echoing round the house like a demented rock-drill. I may be named as 'The Author' but many others have contributed to this book, each in his own way. My thanks to them all!

Tubal Cain
Westmorland, 1980

PREFACE TO THE SECOND EDITION

With this reprint – the fourth since first publication – the opportunity has been taken to effect a number of revisions. Data have been brought up to date to take account of recent developments and a number of the sections have been rearranged to make for more convenience in use. I hope that these alterations, together with the several pages of new material which I have been able to include, will make the 'Handbook' even more useful.

Tubal Cain
April 1986

PREFACE TO THE THIRD EDITION

It is now sixteen years since this book was written. During that time the world of models has changed considerably. A visit to any club exhibition will show a marked increase in the variety of models on display and a much greater emphasis on the pursuit of 'authenticity'. There have been changes in workshop practice, both in the machine tools and in the measuring equipment we use. And, not least, a welcome increase in the field of 'experimentation'. The time seems to be more than ripe for a second, comprehensive, revision of this Handbook.

Not the least of the changes which have affected us, even if indirectly, is the worldwide replacement of both metric and imperial units by the 'International' system; universally in the field of science, and almost completely in engineering and technology. The first section, which formerly concentrated on conversion factors, has, therefore, been re-titled and completely re-written to deal with this subject.

The section which deals with workshop calculations, has been re-arranged, with several pages added to make it more complete. Those readers who wrote suggesting that 'Logs' be omitted in these days of calculators and computers may be surprised to find this retained (though revised) but the truth is that your numbers were equally balanced by those asking for more examples of use! Some have been provided.

The pages on tapers and collets now include both the International and R8 types used on modern machines, and those dealing with screw threads have been revised and rationalised by bringing together both the threads and their hexagon sizes. The section 'Workshop Practice' has been enlarged to include some notes on cutting oils.

Both 'Metal Joining' and 'Properties of Materials' have been brought up to date, and the latter has been extended to include data on the 'preferred' millimetric dimensions of wire, sheet and barstock which are now replacing the former gauge, inch, and metric standards.

The sections dealing with steam, the steam engine, boilers, and the gas laws, have been completely revised, with considerable additional matter, including consideration of engine indicator diagrams and valve events. Recent research on safety valves is covered as are the latest rules on boiler testing. The associated section on piston and gland seals now includes data on piston rings, and the tables of 'O' rings have been brought up to date.

Electrical notes have been checked against current manufacturers' data, and the final pages dealing with general matters have been revised and extended, with some topics transferred to earlier sections for greater convenience.

The format of the book has been changed, so that the page number now incorporates the section number. And you will find that blank spaces have been left at certain points. This will allow topics at present under discussion to be noted on by the reader as they develop. You will also find that 'squared' pages for your own sketches and notes have been provided following the index.

Such a major revision could not be carried out without help. I am most grateful to those readers who have made suggestions, as well as to the many firms which sent me information, often in great detail. My wife has frequently set aside her own work so that I could use her bench (the kitchen table!) when the paperwork overflowed from my desk. I could not have managed without her co-operation. Finally, all books must be processed into print. Lyn Corson and Beverly Laughlin have had to cope with hundreds of scribbled marginal notes in the original book as well as some 70 A4 pages of additional matter. Their patience and understanding is much appreciated!

Tubal Cain
Westmorland, 1996

SECTION ONE
SI UNITS AND METRICATION

SI UNITS

It is unfortunate that the introduction of 'SI' ('Systéme Internationale') units to this country was publicised as 'metrication', for the 'metric' countries have been affected by the change almost as much as those using the 'imperial' or 'foot/pound/second' system. Both the 'metre/kilogram/second' and the 'centimetre/gram/second' systems have been abandoned in favour of SI. The change from feet and inches to metres and millimetres is but a very small, though important, part of the introduction of a system which is both rational and consistent throughout the *whole* of science, engineering and technology.

This consistency has been achieved chiefly by the adoption of a new unit of *force*, which is quite independent of 'gravity'. This has not only removed the confusion between 'mass' and 'weight' ('lbf' and 'lb', or 'kgf' and 'kg', as well as the 'w/g' used in dynamics) but has also eliminated all the conversion factors needed when dealing with work and energy (e.g. the 'mechanical equivalent of heat' is unity, and 1 heat unit/sec = 1 watt of power).

The result has been a phenomenal simplification of calculations (and concepts) in all branches of science, from microbiology to astrophysics, and in engineering from micro-electronics to space travel. The SI system is now universally adopted in science, and is almost so in engineering; about the only country not to have made the formal change is the United States, although even there most areas of science and an appreciable section of engineering have adopted the system.

It may, perhaps, be wise to correct a misapprehension. Neither SI nor the ISO (International Standards Organisation) has anything to do with European union or harmonisation. The system was devised (in 1960) by the ISO (on which the British Standards Institution is strongly represented) following many years of deliberation. The British decision to adopt SI units (i.e. 'go metric') preceded our entry to the European Union by almost ten years.

Orders of magnitude

Instead of changing the basic *name* of the unit as we are used to (inch, foot, yard, furlong, mile, etc.) as the quantity changes, the name of the unit is preceded by a prefix which indicates the multiplier to be used. This is a very old practice – for 70 years or more structural engineers have used kilopounds and radio designers megohms and microfarads. One simply adds the necessary number of zeros after, or a decimal point and zeros before the unit, to correspond to the multiplier.

The 'preferred' prefixes are those which go up or down by 1000 at a time, this being about the largest order of increase that can conveniently be visualised, or marked on a scale. In length, therefore, the centimetre (1/100 metre) is *not* used in engineering practice and though it may be retained in domestic usage (e.g. in dress-making) it is best avoided even here. The only other 'non-preferred' magnitude likely to be met with is the hectare (10 000 sq. metres) for measurement of land sales.

The preferred prefixes of magnitude are given in the table overleaf.

Multiplying Factor		*Prefix*	*Symbol*
1 000 000 000 000	$=10^{12}$	Tera-	T
1 000 000 000	$=10^{9}$	Giga-	G
1 000 000	$=10^{6}$	Mega-	M
1 000	$=10^{3}$	kilo-	k
1	$=10^{0}$	none	none
0·001	$=10^{-3}$	milli-	m
0·000 001	$=10^{-6}$	micro-⁂	μ (Greek – 'mu')
0·000 000 001	$=10^{-9}$	nano-	n
0·000 000 000 001	$=10^{-12}$	pico-	p

⁂ To avoid confusion with the measuring instrument, the 'micro-metre' is called a 'micron' (=0·001 mm).

The *non*-preferred factors, relics of the old metric system and seldom used (apart from centimetres) even there, are:

× 100	hecto-	**These prefixes**
× 10	deca-	**should be**
$\frac{1}{10}$	deci-	**avoided by all**
$\frac{1}{100}$	centi-	**model engineers.**

UNITS OF THE ISO SYSTEM

(For a more detailed consideration of this matter, see *The I.S.O. System of Units*, T. D. Walshaw. Nexus Special Interests. ISBN 1-85486-063-1.)

The basic units

Length The *metre*, abbreviation 'm'.
This is an **absolute** standard, determined from the wavelength of radiation from *krypton-86*. (The yard is now similarly defined, so that one inch becomes **exactly 24·400 mm.**)

Mass The *kilogram*, abbreviation 'kg'.
This is an **arbitrary** standard, not derived from any physical quantity, so that reference must be made to the *prototype mass* held at the International Bureau of Standards. However, it is very close indeed to the mass of 0·001 cu. metre of pure water at its maximum density.

1 kg = 2·204 622 622 lb mass exactly.

Time The *second*, abbreviation 's' or 'sec'. This is an **absolute** standard, being derived from the frequency of radiation from *caesium-133*.

Temperature The *degree Kelvin*. This is identical to the *degree Celsius*. Abbreviations °K and °C. (The name Celsius replaces the former centigrade, changed because angles are measured in grades in a few countries.)
0°C = 273·15°K. This standard depends on a known physical phenomenon.

Angles *The radian* (abbreviation 'rad') is used in calculations, but degrees (°) are used for measurement. 1 radian is the angle subtended by an arc equal to the radius of the circle, so that 1 rad = 2Pi degrees (57·296°). There are 60 minutes of arc in 1°, and 60 seconds of arc in 1 minute.

Electric current The *ampere*, abbreviation 'A' or 'amp'. This is really a derived unit as it depends on the units of mass and length, but is regarded as basic as all other electrical units are derived from it.

Derived units

The majority are quite straightforward, but a few notes may help to avoid confusion.

Area The *sq. metre* and its multiples/sub-multiples are used, but note that mm^2 means square millimetres and *not* milli-square metre. The accepted unit for land measure is the *hectare* – 0·01 km^2 or 10 000 sq. metre.

Volume The *cu. metre* and its multiple/sub multiples are the rule, but the *litre* (abbreviation L) has been adopted for the measurement of liquids and gases. The litre is now **exactly 0·001 m^3**, and the term millilitre (mL) is now used in place of cm^3 or cc for fluid measure.

Force This is a new concept. Hitherto the unit of force, metric or imperial, involved the acceleration due to gravity; inconvenient in space! Although engineers were used to it there is no doubt that this was a nuisance, and often caused confusion. Equally confusing, the same *names* – kilogram or pound – were used for force and mass.

In the SI system a *new* unit is used, the *newton* (N) which is '*That force necessary to accelerate a mass of 1 kg at a rate of one metre/second/second*'. The advantage of this new concept is immediate. The old w/g found in so many engineering formulae disappears and dynamic calculations are vastly simplified.

To give some idea of scale, 1 newton is about 0·225 lbf, or 9·81 kgf. (The SI standard value of g is **9·806 650** m/sec^2.) This idea will take a little getting used to, but once mastered it makes life very much easier.

1 newton = 0·224 808 94 lb force exactly.

Weight Weight is really a force, being that exerted on any mass by gravitational acceleration. It ought, therefore, to be quoted in newtons. However, the British Weights & Measures Act *permits* the use of the kilogram as a weight *for commercial purposes only*. As all commercial and domestic weighing machines are really mass comparators there will be no error, but model engineers should realise that a **mass** of 100 kg will exert a **force** on the supports of 100 **kgf**, or 981 newton. It is advisable to *avoid* the use of 'weight' except when buying coal or grocer.es!

Pressure and stress Both of these are stated as 'force per unit area', and the SI unit is *newton/sq. metre* (N/m^2), which has been given the name *pascal* (Pa). However, this unit is really too small for engineering use (1 Pa = 0·000 14 lbf/in^2 approx) so that the megapascal (MPa) is often used instead. But 1 MPa = 1 newton/sq. mm (about 145 lbf/sq. in.) and it is now usual to state both pressure and stress in N/mm^2, KN/mm^2, etc.

A non SI unit is the bar, which is 0.1 N/mm^2. This is very old, and is now in common use for gas and steam pressures, but not for stress. 1 bar is very nearly atmospheric pressure. It is recommended that model engineers use N/mm^2 (which says what it is!) rather than MPa in all calculations, but bar is quite acceptable when *describing* a fluid pressure.

Energy This comprises mechanical, electrical, thermal and pressure energy, and all can, if need be, be expressed in the same single unit (which is very old) the *joule* (J) such that:

1 joule = 1 watt.second = 1 newton.metre = 1 pascal.metre^3.

Heating values are expressed in MJ/kg, MJ/m^3, MJ/l etc., and *specific heats* in kJ/kg.°K.

1 BTU = 1055·056044 J exactly.

Power

Note that energy and work are expressed in the same units, as is torque. Power is 'rate of doing work', and the SI unit is the *watt*. From the above it is seen that 1 N.m/sec = 1 J/sec = 1 watt and 1 kJ/sec = 1 kW. The *horsepower* is not an SI unit, but can still be used by those prepared to apply the conversion factor that 0·746 kW = 1 hp!

1 kW = 1000 J/sec exactly.

Speeds

In the SI system all speeds or rates are related to the *second* – metres/sec, rev/sec, rad/sec, etc. in *calculations*, but km/hr, m/min etc. are permitted when needed for convenience. Water supply, for example, is measured in ml (mega-litres)/*day*.

Electrical units

These remain unchanged both in name and magnitude, and as they are all derived from the basic SI units they share the same universality – and the improved precision arising from the current figures of 1 part in 10^9 for the metre and 1 part in 10^{12} for the second.

Conclusion

There is a common misunderstanding over the change to SI units, both amongst professional as well as model engineers. No-one claims that SI is more *scientific* than the imperial system – both are based on man-devised units. The object of the change has been to make our system of measurements more *rational* – and, of course, to change from a fractional arrangement to a decimal one. That the SI system is metric is almost accidental – it would have been very difficult to decimalise a system whose unit of length went up by multipliers of 12 (feet) 3 (yards) 5½ (rods) 4 (chains) 10 (furlongs) and finally 8 (miles) and whose units of mass had a similarly incongruous series of increments. It is, perhaps, understandable that there will be dismay at the abandonment of units which have served for hundreds of years, but it would be well to remember that within living memory (at the time of writing) there were no less than 28 different legal versions of the foot in Europe, ranging from just over 9 inches to almost twice as great. (To say nothing of the fact that the US pint is considerably smaller than the British – it is, in fact, ⅛ of the 16th century 'wine gallon'!) The change from that state of affairs, obviously essential as engineering developed, must have been even more difficult than our change to SI.

There are difficulties. Some units – notably the pascal, or newton/sq. metre – are ludicrously small. But most spring more from unfamiliarity rather than anything else; in which connection it is worth remembering that there are millions of youngsters who find it impossible to visualise an inch or a pint! They have been brought up from 5 years old on the SI system, and, more important, found it far easier to use than the old imperial, just because it *is* rational. Try converting inch.ounces/hour into hp – and then see that to obtain kW from newton.mm needs only a shift in the decimal point!

Model engineers are mainly concerned with linear dimensions, and the best advice that can be offered is:

(a) Wherever possible work in millimetres and decimals.
(b) If working in inches is unavoidable, abandon fractions and use decimals throughout, i.e. get used to a fully decimal system. (Use decimals of pounds, not ounces, too.)

(c) To get used to 'visualisation' remember that 100 mm is nearly 4 inches, that 1 mm is about 40 thou, and that 10 microns (0·01 mm) is about 'four tenths'.
(d) Finally, try to avoid converting SI units to inches; work to them, and get used to the system as quickly as you can.

CONVERSION FACTORS

In the tables which follow, the factors have been given to a far greater number of significant figures than is necessary for normal engineering work. This enables any desired degree of accuracy to be achieved in special cases, but for workshop use the factors should be rounded off. Two decimal places when working in millimetres and three when working in inches should suffice for most model work. However, when making calculations which involve the use of conversion factors it is good practice to use one more significant figure in the factors than is needed in the final answer, and to round off as needed at the end of the calculation. Similar principles should be applied when converting measurements other than those of length. (Those requiring more detailed tables should consult *The I.S.O. Systems and Units*, Nexus Special Interests, ISBN 1-85486-063-1.)

When comparing the factors given above for *volumes* with those given in earlier textbooks it should be borne in mind that the *litre*, formerly defined as the volume of one kilogram of water under specified conditions, is now defined as exactly 0·001 cu. metres. The difference (about 28 parts per million) is negligible in most calculations made by model engineers.

Note that in a few cases the *exact* value has been quoted (in bold type) for reference if needed.

(A) Decimal inch to mm

$\times\frac{1}{1000}$	$\times\frac{1}{100}$	$\times\frac{1}{10}$	Inches		+10″	+20″
			0		254·0	508·0
·0254	·254	2·54	1	**25·400**	279·4	533·4
·0508	·508	5·08	2	50·8	304·8	558·8
·0762	·762	7·62	3	76·2	330·2	584·2
·1016	1·016	10·16	4	101·6	355·6	609·6
·1270	1·270	12·70	5	127·0	381·0	635·0
·1524	1·524	15·24	6	152·4	406·4	660·4
·1778	1·778	17·78	7	177·8	431·8	685·8
·2032	2·032	20·32	8	203·2	457·2	711·2
·2286	2·286	22·86	9	228·6	482·6	736·6

In the timber trade the unit of length (of a plank) is 300 mm = 11·81 inches.
One foot = 304·8 mm. *One yard* = 914·4 mm. *One mile* = 1·6093 km.

INCH/MILLIMETRE CONVERSION TABLES

(B) Fractional inch to decimal and to millimetres (Basis – 1 inch = 25·400 millimetres)

Inches				*mm*	*Inches*				*mm*
		$\frac{1}{64}$	·01563″	0·3969			$\frac{33}{64}$	·51563″	13·0969
	$\frac{1}{32}$		·03125	0·7937		$\frac{17}{32}$		·53125	13·4937
		$\frac{3}{64}$	·04688	1·1906			$\frac{35}{64}$	·54688	13·8906
$\frac{1}{16}$			·06250	1·5875	$\frac{9}{16}$			·56250	14·2875
		$\frac{5}{64}$	·07813	1·9844			$\frac{37}{64}$	·57813	14·6844
	$\frac{3}{32}$		·09375	2·3812		$\frac{19}{32}$		·59375	15·0812
		$\frac{7}{64}$	·10938	2·7781			$\frac{39}{64}$	·60938	15·4781
$\frac{1}{8}$			·1250	3·1750	$\frac{5}{8}$			·6250	15·875
		$\frac{9}{64}$	·14063	3·5719			$\frac{41}{64}$	·64063	16·2719
	$\frac{5}{32}$		·15625	3·9687		$\frac{21}{32}$		·65625	16·6687
		$\frac{11}{64}$	·17188	4·3656			$\frac{43}{64}$	·67188	17·0656
$\frac{3}{16}$			·18750	4·7625	$\frac{11}{16}$			·68750	17·4625
		$\frac{13}{64}$	·20313	5·1594			$\frac{45}{64}$	·70313	17·8594
	$\frac{7}{32}$		·21875	5·5562		$\frac{23}{32}$		·71875	18·2562
		$\frac{15}{64}$	·23438	5·9531			$\frac{47}{64}$	·73438	18·6531
$\frac{1}{4}$			·250	6·3499	$\frac{3}{4}$			·750	19·0497
		$\frac{17}{64}$	·26563	6·7469			$\frac{49}{64}$	·76563	19·4469
	$\frac{9}{32}$		·28125	7·1437		$\frac{25}{32}$		·78125	19·8437
		$\frac{19}{64}$	·29688	7·5406			$\frac{51}{64}$	·79688	20·2406
$\frac{5}{16}$			·31250	7·9375	$\frac{13}{16}$			·81250	20·6375
		$\frac{21}{64}$	·32813	8·3344			$\frac{53}{64}$	·82813	21·0344
	$\frac{11}{32}$		·34375	8·7312		$\frac{27}{32}$		·84375	21·4312
		$\frac{23}{64}$	·35938	9·1281			$\frac{55}{64}$	·85938	21·8281
$\frac{3}{8}$			·3750	9·5250	$\frac{7}{8}$			·8750	22·225
		$\frac{25}{64}$	·39063	9·9219			$\frac{57}{64}$	·89063	22·6219
	$\frac{13}{32}$		·40625	10·3187		$\frac{29}{32}$		·90625	23·0187
		$\frac{27}{64}$	·42188	10·7156			$\frac{59}{64}$	·92188	23·4156
$\frac{7}{16}$			·43750	11·1125	$\frac{15}{16}$			·93750	23·8125
		$\frac{29}{64}$	·45313	11·5094			$\frac{61}{64}$	·95313	24·2094
	$\frac{15}{32}$		·46875	11·9062		$\frac{31}{32}$		·96875	24·6062
		$\frac{31}{64}$	·48438	12·3031			$\frac{63}{64}$	·98438	25·0031
$\frac{1}{2}$			·500	12·6999					

'Round off' mm column to 2 decimal places, for an accuracy to 0·0004″, e.g. $\frac{5}{16}$″ = 7·94 mm (= 0·31259″) for normal machining operations.

(C) Millimetres to inches

$\frac{1}{100}$ mm	$\frac{1}{10}$ mm	millimetre		+10 mm	+20 mm	+30 mm	+40 mm	+50 mm	+60 mm	+70 mm	+80 mm	+90 mm
		0		·39370	·78740	1·1811	1·5748	1·9685	2·3622	2·7559	3·1496	3·5433
·00039″	·00394″	1	·03937″	·44307	·82667	1·2205	1·6142	2·0079	2·4016	2·7953	3·1890	3·5827
·00079	·00787	2	·07874	·47244	·86614	1·2598	1·6535	2·0473	2·4410	2·8347	3·2284	3·6621
·00118	·01181	3	·11811	·51181	·90551	1·2992	1·6929	2·0866	2·4803	2·8740	3·2677	3·6614
·00158	·01595	4	·15748	·55118	·94488	1·3386	1·7323	2·1260	2·5197	2·9134	3·3071	3·7008
·00197	·01969	5	·19685	·59055	·98425	1·3780	1·7717	2·1654	2·5591	2·9528	3·3465	3·7402
·00236	·02362	6	·23622	·62992	1·0236	1·4173	1·8110	2·2047	2·5984	2·9921	3·3858	3·7795
·00276	·02756	7	·27559	·66929	1·0630	1·4567	1·8504	2·2441	2·6378	3·0315	3·4252	3·8189
·00315	·03150	8	·31496	·70866	1·1024	1·4961	1·8898	2·2835	2·6772	3·0709	3·4646	3·8583
·00354	·03543	9	·35433	·74803	1·1417	1·5354	1·9291	2·3228	2·7165	3·1102	3·5039	3·8976

100 mm = 3·93701 inches. 1000 mm = 1 metre = **39·3700797** inches **exactly**. 1 km = 0·6214 mile = 1093·61 yards. For normal machining operations, round off to the nearest 'thou' – e.g. 24 mm = 0·945″.

CONVERSION FACTORS
(Figures in bold are EXACT)

(1) Length

1 yard = 0·914 400 metre; 1 inch = 25·400 mm exactly.

	mm	*metre*	*inch*	*foot*	*yard*
Millimetre	1·0	0·001	0·0394	0·0033	0·00109
Metre	1000	1·0	39·3701	3·2808	1·09361
Inch	25·4	0·0254	1·0	0·0833	0·02778
Foot	304·8	0·3408	12·0	1·0	0·33333
Yard	914·4	**0·91440**	36·0	3·0	1·0

One 'thou' (0·001″) is 0·0254 mm. $\frac{1}{10}$ mm = 0·00394″ (say 0·004″).

(2) Area

	sq. mm	*sq. metre*	*sq. in.*	*sq. ft*	*sq. yd*
sq. mm	1·0	0·000001	0·0015	0·000011	—
sq. metre	1 000 000	1·0	1550·0	10·7639	1·1960
sq. inch	645·16	0·00065	1·0	0·00694	0·0007716
sq. foot	92 903·0	0·0929	144·0	1·0	0·111111
sq. yard	—	0·83616	1296	9·0	1·0

		ha	*sq. km*	*acre*	*sq. mile*
1 ha	=	**1**	**0·01**	**2·471 053 8**	0·003 861
1 km^2	=	**100**	**1**	247·105	0·386 102
1 acre	=	0·404 686	0·004 047	**1**	**0·001 562 5**
1 sq. mile	=	258·999	2·589 99	**640**	**1**

1 rood = 1210 yd^2 = 0·25 acre = 1447·15 m^2.

(3) Volume

1 cubic yard = **0·764 554 875 984** m^3. 1 cu. metre = **35·314 666 88** cu. foot.
1 litre = **0·001** cubic metre exactly.

		ml. = cm^3	*litre*	*cu. metre*	*cu. inch*	*cu. foot*	*cu. yard*
1 ml.	=	**1**	**0·001**	**0·000 001**	**0·061 023 74**	—	—
1 litre	=	**1000**	**1**	**0·001**	**61·023 74**	0·035 315	0·001 130 8
1 metre^3	=	**10^6**	**1000**	**1**	**61023·74**	**35·314 670**	**1·307 950 7**
1 cu. inch	=	**16·387 064**	0·016 387	0·000 016	**1**	0·5787 × 10^{-3}	0·000 021 4
1 cu. foot	=	**28 316·84**	**28·316 84**	**0·028 316 846**	**1728**	**1**	0·037 037 04
1 cu. yard	=	764 555·2	764·555 2	0·764 555	**46 656**	**27**	**1**

		litre	*cu. metre*	*UK gall.*	*US gall.*
1 litre	=	**1**	**0·001**	**0·219 969**	0·264 174
1 cu. m.	=	**1000**	**1**	**219·969**	264·172
1 UK gall.	=	**4·546 095**	0·004 546	**1**	1·200 95
1 US gall.	=	3·785 41	0·003 785	**0·832 674**	**1**

(**1 US gall. = 231 cu. in.**, 1 UK gall = **277·420 032 cu. in.**)
1 fl. ounce (= 8 drachms) = 28·4131 mL. 1 gall. = 8 pint. 1 pint = 20 fl. oz. (liquid measure) 1 bushel = 4 peck. 1 peck = 2 gall. solid measure. 1 imp. pint = 34·675 cu. in.

(4) Mass

1 lb = 0·453 592 37 kg. 1 kg = 2·204 622 622 lb exactly.

	gram	*kg*	*lb*	*cwt*	*ton*	*tonne*
1 gram	1·0	0·001	0·0022	—	—	—
1 kg	1000	1·0	2·2046	0·0196	0·00098	0·001
1 pound	453·59	0·45359	1·0	0·0089	0·000446	0·00045
1 cwt	—	50·802	112·0	1·0	0·050	0·0508
1 ton	—	1016·05	2240·0	20·0	1·0	1·01605
1 tonne	—	1000·00	2204·6	19·684	0·9842	1·0

1 ounce troy = 1·0971 ounce avoirdupois. 12 oz troy = 1 lb troy.
1 lb avdp = 7000 grains; 1 lb troy = 5760 grains; the grain is common to troy, avoirdupois and apothecary's measures. 1 oz avdp = 28·349 gram.
1 US ('short') ton = 2000 lb = 0·89286 English ton = 0·907184 tonne.
1 '*shipping ton*' (in measuring bulky cargo) is in fact a VOLUME of 42 cu. ft.

(5) Density

	*gm/mL** (gm/cc)*	*gm/m³*	*lb/cu. ft*	*lb/imp. gall.*
1 gm/mL**	1·000	1 000 000	62·4276	10·0224
1 lb/cu. ft	0·016018	16018·0	1·000	0·16053
1 lb/gall.	0·099776	99 776·0	6·2288	1·000

** = kg/litre. 1 g/litre = 1 kg/m^3.

(6) Specific volume

1 cu. ft/lb = 0·062428 cu. metre/kilogram
1 cu. m/kg = 16·018 cu. ft/lb.

(7) Force

	newton	*kg force*	*poundal*	*lb force*	*ton force*
1 N =	**1**	**0·101 971 62**	7·233 01	**0·224 808 94**	$1·003\,61 \times 10^{-4}$
1 kgf =	**9·806 65**	**1**	70·931 444	**2·204 62**	$9·842\,07 \times 10^{-4}$
1 pdl =	**0·138 255 17**	0·014 098	**1**	**0·031 081 0**	$1·387\,54 \times 10^{-5}$
1 lbf =	**4·448 221 62**	**0·453 592 37**	**32·1740**	**1**	$4·464\,29 \times 10^{-4}$
1 tonf =	9964·02	1016·047	**72 069·76**	**2240·0**	**1**

1 dyne = 10^{-5} newton. **1 newton = 0·1 megadyne.**

Note: The dyne is that force required to accelerate 1 gram at 1 cm/sec^2
The poundal is that force required to accelerate 1 lb.mass at 1 ft/sec^2
The kg force is that force required to accelerate 1 kg at 'standard gravity', 9·80 665 m/sec^2
The lb force is that force required to accelerate 1 lb at 'standard gravity', 32·1740 ft/sec^2

(8) Pressure

	lbf/sq. in.	*kgf/sq. cm*	*bar*	*newton/ sq. mm*	*in. hg*	*mm hg*	*lbf/sq. ft*
1 lbf/sq. in.	1·0	0·0703	0·069	0·0069	2·0367	51·73	144·0
1 kgf/sq. cm	14·233	1·0	0·981	0·0981	28·958	750·098	2048
1 bar	14·504	1·0197	1·0	0·100	29·53	750·067	2088·5
1 newton/sq. mm	145·038	10·197	10·0	1·0	295·3	7500·67	20885·5
1 in. hg	0·4910	0·03452	0·0339	0·00339	1·0	25·4	70·71
1 mm hg	0·0193	0·00136	0·00133	0·00013	0·0394	1·0	2·784
1 lbf/sq. ft	0·00694	0·00049	0·00048	0·000048	0·0141	0·3592	1·0

1 Pascal = 1 N/m^2. 1 N/sq. mm = 1 000 000 pascal = 1 MPa.

1 'standard atmosphere' is 760 mm hg, 14·695 lbf/sq. in or 0·0988 N/mm^2.
One foot 'head of water' = 0·4335 lbf/sq. in; 1 m head = 0·0088 N/mm^2.

(9) Stress (see also page 7.10)

	tonf/sq. in.	*lbf/sq. in.*	*kg/sq. cm*	*newton/sq. mm*
1 tonf/sq. in.	1·0	2240	157·5	15·444
1 lbf/sq. in.	0·000446	1·0	0·0703	0·0069
1 kg/sq. cm	0·00635	14·233	1·0	0·0981
1 N/sq. mm	0·06475	145·038	10·197	1·0

The SI unit for both pressure and stress is the pascal = 1 newton/sq. metre, abbreviation Pa. This is very small, and in practice the megapascal (MPa) is required. It is much better to use newton/sq. mm or kN/sq. mm.

(10) Heat, work and energy

Definitions. *One Centigrade Heat Unit* (CHU) is the amount of heat required to raise one POUND mass of water through 1 degree Celsius. (Imperial)

One British Thermal Unit (BTU) is the amount of heat required to raise one POUND mass of water through 1 degree Fahrenheit. (Imperial)

One Kilogram Calorie (KgCal) is the amount of heat required to raise one KILOGRAM mass of water through 1 degree Celsius. (Former metric)

The THERM is 100 000 BTU = 29·31 kWh. (Imperial)

The JOULE is one newton metre, or one watt-second. (SI)

Note that heat, work and energy may be expressed in either thermal, mechanical or electrical units at will, whichever best suits the calculations being made.

	kg/cal (IT)	*CHU*	*BTU*	*ft/lbf*	*metre/kgf*	*joule, or newton metre*
1 kg/cal	1·0	2·2046	3·968 32	3087	426·8	4186·8
1 CHU	0·4536	1·0	1·800	1400·4	193·6	1899
1 BTU	0·252	0·55556	1·0	778	107·6	1055·056
1 ft/lbf	0·0003239	0·00071	0·00129	1·0	0·13825	1·355 818
1 m/kgf	0·002343	0·00517	0·009296	7·233	1·0	9·81
1 joule or newton metre	0·0002389	0·000527	0·000948	0·73756	0·10197	1·0

One horsepower = 33,000 ft lb/min = 550 ft lb/sec = 0·746 kW.
One kW = 1 kilo newton.metre/second = 1J/sec = 1·34 hp.
2545 BTU = 1414 CHU = 642 kg/cal = 2·685 megajoule will produce 1 hp for 1 hour.

(11) Temperature

The Fahrenheit and centigrade (Celsius) scales are quite arbitrary, and both scientists and engineers make use of the 'absolute' temperature, employing either Fahrenheit or Celsius degrees. Absolute zero of temperature is the point where, according to current atomic theory, all atomic and molecular motion ceases. This is taken to be at *minus* 459·69 Fahrenheit or *minus* 273·16 Celsius. The absolute temperature is thus the ordinary thermometer reading *plus* the above constants. Absolute Celsius temperature is quoted as 'degrees Kelvin' (°K) and Fahrenheit in 'degrees rankine' (°R). However, to avoid confusion between this and both the obsolete Réaumur scale and the Universal Gas Constant R, rankine degrees are better written as 'Fabs'. (= 'Fahrenheit absolute' abbreviated.)

From the above it will be seen that:
60°F becomes 60 + 459·59 = 519·59°Fabs
15°C becomes 15 + 273·09 = 288·09°K.

The decimal quantities are usually rounded off to give 519·6°Fabs and 288·1°K respectively.

To convert scales, Degrees K × 1·8 = Degrees Fabs
Degrees Fabs × $\frac{5}{9}$ = Degrees K.

The *interval* of 'one degree' of temperature change is identical on both the absolute and the conventional temperature scales – that is one degree C = one degree K and one degree F = one degree Fabs.

To convert on the conventional scales: t°F = $\frac{5}{9}$ (t – 32)°C
t°C = (32 + 1·8t)°F.

The Réaumur scale is now obsolete (water freezing at 0° and boiling at 80°) but may be met with in old documents emanating from Central Europe and Russia. One Réaumur degree = $1\cdot25°C$.

(12a) Heating or calorific values

1 BTU/lb = **2·326 kJ/kg** exactly = 0·5555 kg.cal/kg
1 BTU/cu. ft = 37·2589 kJ/cu m = 8·899 kg.cal/m^3
1 therm/imp. gall. = 23·208 MJ/litre

(12b) Specific heat

1 J/(kg°K) = $0\cdot238\,845\,9 \times 10^{-3}$ BThU (lb°F)
= 0·185 893 8 ft.lbf/(lb.°F).
1 BThU/(lb°F) = 4186·8 J/(kg°K)
1 BThU/(ft^3°F) = 67 066 J/(m^3°K) = 16·019 Cal_{IT}/(m^3°K)

(12c) Coefficient of heat transfer

The SI unit is the watt/(metre.°K) = W(/m.°K)
1 BThU(/ft.hr.°F) = 1·731 W/(m.°K)
1 k.cal(m.hr.°K) = 1·163 W/(m.°K)

Note: This coefficient is a complex derived unit, being heat units flowing in unit time over unit area for each unit temperature difference assuming unit thickness and formerly conveniently stated as BThU/sq. ft/hr/°F/ft.thickness, or KgCal/sq. metre/hr/°C/metre thickness. The *meaning* of the SI unit is identical.

(13) Compound conversion factors

Multiply	Pounds per foot*	by	1·488	= kg per metre
	Pound/sq. ft*	by	4·883	= kg/sq. metre
	Pound/cu. ft*	by	16·02	= kg/cu. metre
	Feet/second	by	0·68182	= miles/hr
	Feet/second	by	0·3048	= metres/second
	Miles/hr	by	1·6093	= km/hr
	Pounds/hp	by	0·6086	= kg/kW
	BTU/sq. ft/hr (heat loss)	by	2·7125	= kg/cal/sq. metre/hr
	Cu. ft/sec (cu. sec)	by	0·02832	= cu. metre/sec
	Cu. secs	by	28·32	= litres/sec
	Galls/minute	by	0·0757	= litres/sec
	lb/BHP/hr	by	453·6	= grams/BHP/hr
	Pints/BHP/hr	by	0·568	= litre/BHP/hr
	Miles/gall. (imp.)			= 282/(L/100 km)
	Litre/100 km			= 282/mpg

*'Pound mass' e.g. 'Density'.

(14) Former imperial measures

Many of these measures are found in early books on engineering practice. The following are extracted from *Fowler's Engineer's Pocket-book, 1856.*

Common length

12 inch	1 foot
3 feet	1 yard
$5\frac{1}{2}$ yard	1 rod, pole or perch
40 perch	1 furlong
8 furlong	1 mile

Cloth length

$2\frac{1}{4}$ inch	1 nail
4 nails	1 quarter
4 quarters	1 yard (= 16 nails)
5 quarters	1 ell (= 20 nails)

Particular length

1 link	7·92 inch
1 chain	100 link or 22 yard
1 fathom	6 feet
1 cable	120 fathom

6 points	1 line
12 lines	1 inch
4 inch	1 hand

Capacity, liquid

4 gill	1 pint
2 pint	1 quart
4 quart	1 gallon (imp.)**

(1 imp. gall. = 277·274 in^3)

Capacity, dry

2 gall. (imp.)**	1 peck
4 peck	1 bushel (1·284 ft^3)
8 bushel	1 quarter
5 quarters	1 load

**1 ale gallon = 1·16 imp. gallon; 1 wine gallon = 0·825 imp. gallon = 1 US gallon.

The 'hogshead', referred to in early descriptions of steam boilers, was 'three score and three' (63) wine gallons, equivalent to $52\frac{1}{2}$ imperial gallon.

Weight, avoirdupois

1 ounce	16 drachm
1 pound*	16 ounce
1 stone	14 pound
1 quarter	28 pound
1 hundredweight	4 quarter
1 ton	20 cwt

Weight, troy

1 ounce	20 pennyweight (dwt)
1 pound*	12 ounce

*1 lb troy = 5760 grains
*1 lb avdp = 7000 grains

Miscellaneous

1 load of timber: in the log – 40 cu. ft Sawn – 50 cu. ft
1 load of bricks – 500; of lime – 32 bushel; sand – 36 bushel.
1 bushel of wheat – 60 lb; of barley – 50 lb; of oats – 40 lb; of flour or salt – 56 lb.
One 'thousand' of nails – 10 'hundreds' each of 120 nails.
One coal chaldron = 35 bushels (= 180 cu. ft approx.); one 'sack' of coal = 224 lb (2 cwt).

SECTION TWO
WORKSHOP CALCULATIONS

USE OF LOGARITHMS

Despite the extended use of pocket calculators there is still a place for logarithms (logs) as indicated by the fact that scientific calculators can find the log of a number, though somewhat imperfectly.

Logarithms *can* be used without understanding how they work, but a little of the background does simplify some of the rules. The principles are not difficult. Look at it like this.

All numbers consist of figures, but the same figures can mean different numbers. Thus 123·4 and 1234 are different numbers. The figures give the weight of the number, and the position of the decimal point decides its power or characteristic. We can show these, weight and power, separately by writing the above numbers as:

$$1\cdot234 \times 10^2 \text{ and } 1\cdot234 \times 10^3$$

Both now have identical weights, and the powers are shown by the indexes of 10.

Without a calculator the multiplication of $1\cdot234 \times 1\cdot234$ is quite a job, but $10^2 \times 10^3$ is easy – we just *add* the indexes or powers, to get 10^5.

Over 300 years ago it was realised that if we knew the power of ten which made (in this case) 1·234 then these numbers, too, could be multiplied by adding the indexes. In fact, $10^{0\cdot0913} = 1\cdot234$. That is, 0·0913 is the *logarithm* (log) of 1·234. So, we can multiply the two numbers simply by adding up both the weight (called by the latin name the *mantissa*) *and* the index or power of the numbers. The sum of the mantissas comes to 0·1826, and 10 to this power works out at 1·523. The sum of the powers is 5. Hence $123\cdot4 \times 1234 = 1\cdot523 \times 10^5$.

Fortunately, Messrs. Napier and Briggs, who conceived the idea back in 1625, prepared tables of these powers (i.e. logarithms), and we don't have to work it all out. We just look up the logs, do the sums and then look up the anti-log to find the answer!

There is just one special drill which is not normal in arithmetic. 0·01234 is $1\cdot234 \times 10^{-2}$ (or $1\cdot234/10^2$, which is the same thing) and this means that the logarithm of 0·0123 is **−2** for the power and +0·0913 for the mantissa. This *could* be written as −1·9087, but over the centuries it has been found more convenient to write it as $\mathbf{\bar{2}\cdot0913}$, with the minus sign lying *above* the figure it refers to. This is *spoken* as '**bar**.2·0913', the word bar indicating that the minus applies only to the power. (The mantissa of a logarithm is *always* positive.)

Rules for the use of logarithms

(1) To find the logarithm of a number.

(a) Reduce the figures to a number between 1 and 9·999 together with a multiplier which is a power of 10 (remembering that $1 = 10^0$). This power or index forms the first part of the log, and if negative is written as bar, not minus.

(b) Look up the number (mantissa) in the table. The first two digits are found in the LH column, the third under one of the headings in the body of the table, and the last in the columns of 'mean differences'; this last is added to the number found in the body of the table. Then combine the index with the mantissa.

Example: Find the log of 3596. See page **2**.8.

$3596 = 3{\cdot}596 \times 10^3$. Hence the index is **3**.
From the table, look along the line for 35; under col. 9 see **0·5551**.
Under '6' in the mean difference col. read **7**. Add this to the above.
Hence log 3596 is **3·5558**.

Now try log 0·005683. The answer is $\bar{3}{\cdot}7545$.

(2) To find a number from its logarithm.

Disregard the index or power for the moment. Look up the remaining mantissa in the table of anti-logarithms on page **2**.10 (noting that these run from 0 to 0·9999) using exactly the same procedure. The figure found gives the digits of the required number.

Inspect the index to determine the decimal multiplier; apply this to the digits just found to give the required number.

Example: Find the number whose log is 2·0356.

From the table 0356 means 1·085. The index is 2, hence the required number is $1{\cdot}085 \times 10^2 = 108{\cdot}5$.

(3) To multiply and divide.

To multiply, add the logs; to divide, subtract the logs.

Example I: Convert 72" to metres. 1" = 25·4 mm = 0·0254 m.

	Number	*Index*	*Log*
	72	1	1·8573
	0·0254	−2	$\bar{2}{\cdot}4048$
Add to multiply (subtracting the negative index)			0·2621

From table of antilogs – answer is 1·828 metres.

Example II:

$$\text{Evaluate } \frac{9{\cdot}753 \times 10{\cdot}34 \times 0{\cdot}9252}{1{\cdot}453 \times 3{\cdot}142}$$

First work out the logs of the top and bottom decks of the fraction, subtract the logs of bottom from top, and then find the antilog.

Top Deck (Numerator)	*Number*	*Index*	*Log*
	9·753	0	0·9891
	10·34	1	1·0145
	0·9252	−1	$\bar{1}{\cdot}9962$
Add to multiply (subtracting the negative index)	Sum		1·9698

Bottom Deck (Denominator)	1·453	0	0·1623
	3·142	0	0·4872
Add to multiply	Sum		0·6595
Subtract log bottom deck from log top deck, to divide.			1·9698
			0·6595
		Difference	1·3103

From the table of antilogs, 3103 gives 2·043.
The index, (characteristic) is 1 so answer is $2{\cdot}043 \times 10^1 = \underline{20{\cdot}43}$.

(4) To find the power of a number.
First find its logarithm, multiply this by the power, and then find the antilog.

Example:
Find $1{\cdot}625^3$

Log 1·625 ....0·2109
multiply by 3 3
0·6327

From antilog table, answer = $\underline{4{\cdot}292}$.

(5) To find the 'n'th root of a number.
Find the logarithm, divide by the power of the root required, and then find the antilog. Important Note – a special procedure noted below is needed for roots of fractions.

Example I:
Find the cube root of 71.

Log 71 ....1·8513
Divide by 3 =0·6171

From antilog, answer = $\underline{4{\cdot}141}$.

Example II:
To show the special procedure for fractions. Find the cube root of 0·71.

Log 0·71 ....$\bar{1}{\cdot}8513$

We cannot divide 'bar 1' (−1) by 3 and get a whole number, so we must alter this till we can. Easy, for $-1 = (-3 + 2)$ so alter the characteristic of Log 0·71 thus:

$\bar{1}{\cdot}8513$ to . . $\bar{3}$ +2·8513
Divide by 3 = $\bar{1}$ +0·9504 or $\bar{1}{\cdot}9504$

From table of antilogs, 0·9504 means 8921 and $\bar{1}$ is 10^{-1},
hence $\underline{\sqrt[3]{0{\cdot}71} = 0{\cdot}8921}$.

(6) To find the fractional power of a number.
(This may occur e.g. when the compression ratio of an IC engine must be raised to a power like 1·3 to find the final pressure).

If the number is greater than 1, proceed as in (5) above.

Example I:

Find the value of $7 \cdot 5^{1 \cdot 3}$

Log $7 \cdot 5$	....	$0 \cdot 8751$
$\times 1 \cdot 3$		$= 1 \cdot 1376$

From antilogs, answer = $\underline{13 \cdot 71}$.

If less than 1, treat the index and the mantissa separately.

Example II:

Find the value of $0 \cdot 75^{2 \cdot 3}$

Log $0 \cdot 75$ ..... $\bar{1} \cdot 8751$

Index	$2 \cdot 3 \times \bar{1}$	$= -2 \cdot 3$
Mantissa	$2 \cdot 3 \times 0 \cdot 8751$	$= \underline{+2 \cdot 0127}$
	Add these two	$= -0 \cdot 2873$
		$= \bar{1} \cdot 7127$

From antilogs, answer = $\underline{0 \cdot 516}$.

(7) Negative powers.

These are found in some scientific writings, and in a few engineering papers as well. This may cause some problems, but need not do so if it is remembered that (e.g.) $1 \cdot 63^{-1 \cdot 2}$ means no more than ...... $\dfrac{1}{1 \cdot 63^{1 \cdot 2}}$. It is only necessary to evaluate the log of the bottom deck, subtract this from the log of 1 (= 0) and find the antilog.

Conclusion

It will be observed that quite a number of the above examples could not be done on many pocket calculators, and model engineers will come across such cases not infrequently. However, there is no reason at all why even the cheapest pocket calculators cannot be used to do the routine addition and multiplication or division of the logs themselves. This does save time, but remember to set the instrument to round off the decimal points to four figures.

The tables in this handbook are 'four-figure' logs. Tables can be bought for a small sum which run to five, giving the answers to one more place of decimals. *Seven* figure logs are very expensive and not easy to use – hardly necessary anyway, for model work.

THE SLIDE RULE

This is simply a logarithmic adding-stick. The scales are marked out in logarithmic progression, so that moving the slide and the cursor automatically takes the logs, adds or subtracts them, and presents the result in proper numbers – antilogs, in fact. Scale limitations reduce the accuracy to about half that of five-figure logs.

LOGARITHMIC GRAPHS

If instead of plotting figures in the ordinary way their *logs* are plotted on the two ordinates, awkward curves often appear as straight lines, making it easier to read off intermediate figures. For example, if P is plotted against log V for the compression curve of an IC engine, the result is a straight line, from which the value of 'n' in $PV^n = C$ can be deduced.

Fig. 1a opposite shows the calibration curve for a vee-notch water meter, giving Q in cu. ft/sec against H, the height of water over the base of the notch. Theory suggests that the 'law' will be in the form $Q = c \times H^n$. Taking logs this means that $\log Q = \log c + n \log H$. At Fig. 1b log Q is plotted against log H, and a straight line results. The value of log c is given by the intercept o–a, and of n by the slope – bc/ac (scaled to values).

Inspection shows that log c is 0·398, so that (from the antilog table) c = 2·50. Log Q at b is 2·878, and at c is 0·398; hence bc is 2·48, and as ac is 1 the value of n is 2·48. So, the law of the curve on Fig. 1a is $Q = 2{\cdot}5H^{2{\cdot}48}$. Note that if we had *known* that law we could plot 1b from *two* values only but for 1a we need ten or more.

The chart in Fig. 1b runs down only as far as H = 1, but the straight line can be extended downwards simply by carrying the two scales lower (e.g. log H from 1·0 to 0·1 and log Q from 0·1 down to 0·01) and lengthening the straight line. Logarithmic plotting *improves the sensitivity* at the lower values.

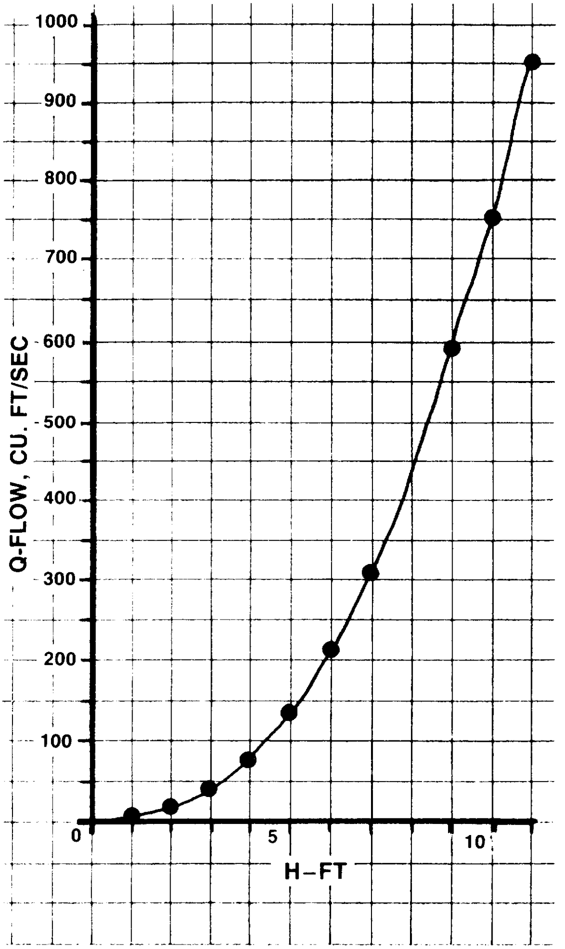


Fig. 1a

Logarithmic graph paper can be obtained, on which the lines are spread at logarithmic intervals and this has been used in example in Fig. 2. Fig. 2a is the PV diagram for a diesel engine, plotted in the normal way. In Fig. 2b the same diagram is shown on scales of log P and log V, though the scales are marked in pressure and % volume, not the *logs* of the values.

Fig. 2b gives much more data than Fig. 2a. From *a* to *b* we have compression following $PV^{1 \cdot 36}$. Combustion starts at *b*, and continues to *c*. From *c* to *d* is a slight curve, showing that heat is still being released (due to oxidation of dissociated products of combustion – the temperature is about 2 200°C) but from *d* to *e* expansion follows the law $PV^{1 \cdot 39}$. Finally, as the piston uncovers a larger area of cool cylinder wall, the curve steepens between *e* and *f*. A similar replotting of a steam engine cylinder diagram might be very revealing!

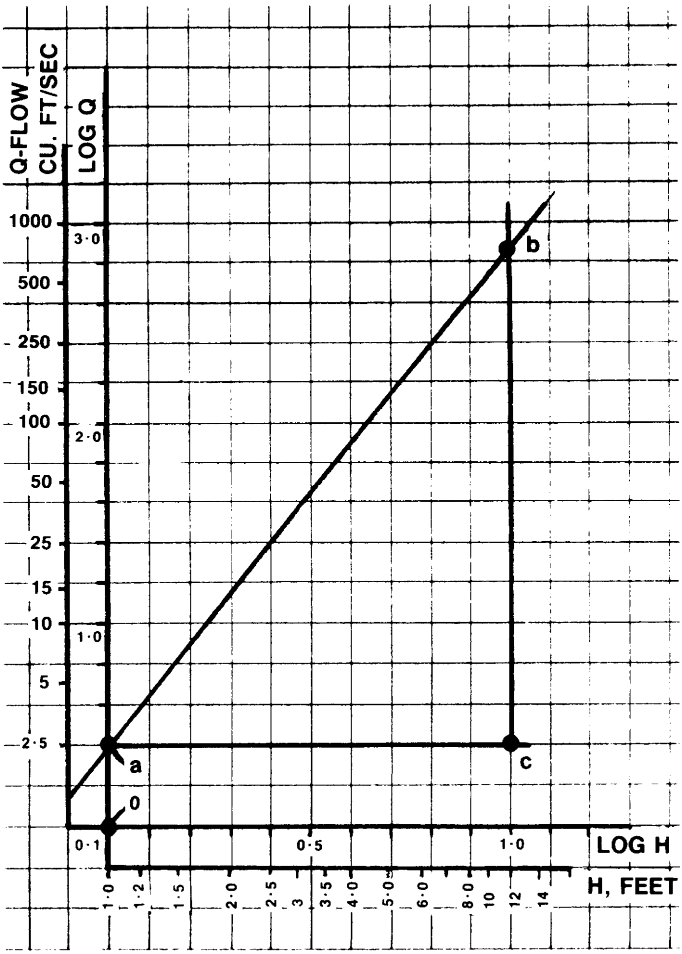


Fig. 1b

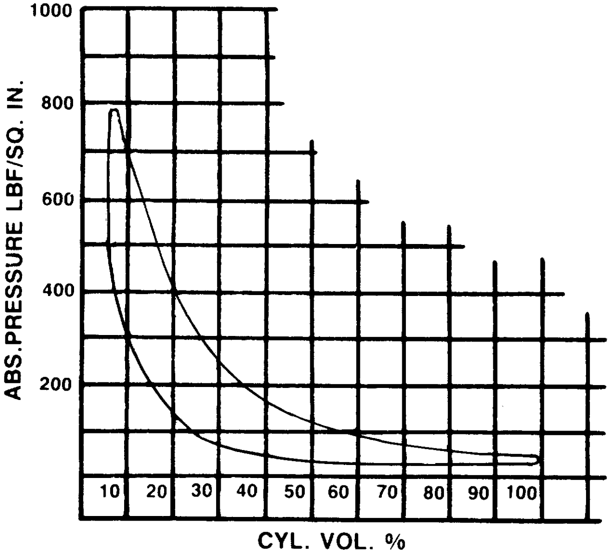

1000
800
600
400
200
ABS.PRESSURE LBF/SQ. IN.
10
20
30
40
50
60
70
80
90
100
CYL. VOL. %

Fig. 2a

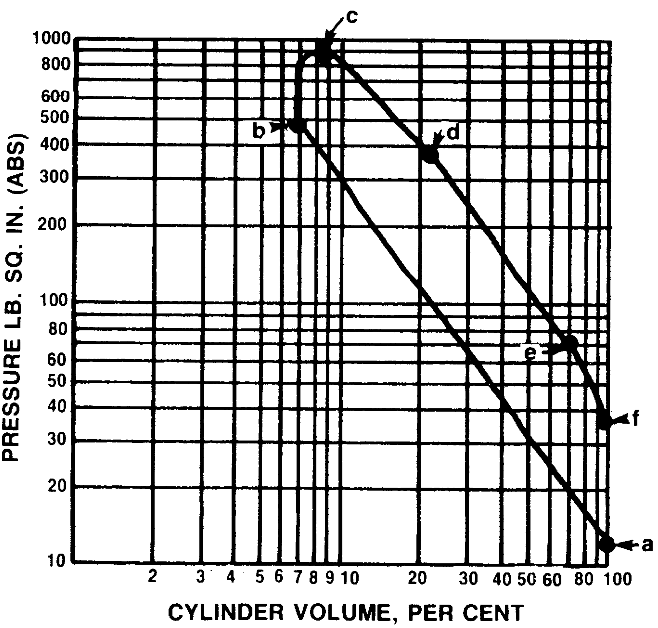

1000
800
600
500
400
300
200
100
80
60
50
40
30
20
10
PRESSURE LB. SQ. IN. (ABS)
a
b
c
d
e
f
2
3
4
5
6
7
8
9
10
20
30
40
50
60
80
100
CYLINDER VOLUME, PER CENT

Fig. 2b

LOGARITHMS, 1·0 TO 4·999

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
1·0	0000	0043	0086	0128	0170						5	9	13	17	21	26	30	34	38
						0212	0253	0294	0334	0374	4	8	12	16	20	24	28	32	36
1·1	0414	0453	0492	0531	0569						4	8	12	16	20	23	27	31	35
						0607	0645	0682	0719	0755	4	7	11	15	18	22	26	29	33
1·2	0792	0828	0864	0899	0934						3	7	11	14	18	21	25	28	32
						0969	1004	1038	1072	1106	3	7	10	14	17	20	24	27	31
1·3	1139	1173	1206	1239	1271						3	6	10	13	16	19	23	26	29
						1303	1335	1367	1399	1430	3	7	10	13	16	19	22	25	29
1·4	1461	1492	1523	1553	1584						3	6	9	12	15	19	22	25	28
						1614	1644	1673	1703	1732	3	6	9	12	14	17	20	23	26
1·5	1761	1790	1818	1847	1875						3	6	9	11	14	17	20	23	26
						1903	1931	1959	1987	2014	3	6	8	11	14	17	19	22	25
1·6	2041	2068	2095	2122	2148						3	6	8	11	14	16	19	22	24
						2175	2201	2227	2253	2279	3	5	8	10	13	16	18	21	23
1·7	2304	2330	2355	2380	2405						3	5	8	10	13	15	18	20	23
						2430	2455	2480	2504	2529	3	5	8	10	12	15	17	20	22
1·8	2553	2577	2601	2625	2648						2	5	7	9	12	14	17	19	21
						2672	2695	2718	2742	2765	2	4	7	9	11	14	16	18	21
1·9	2788	2810	2833	2856	2878						2	4	7	9	11	13	16	18	20
						2900	2923	2945	2967	2989	2	4	6	8	11	13	15	17	19
2·0	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
2·1	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
2·2	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
2·3	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
2·4	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
2·5	3979	3997	4114	4131	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
2·6	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
2·7	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
2·8	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
2·9	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
3·0	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
3·1	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
3·2	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
3·3	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
3·4	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
3·5	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
3·6	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
3·7	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
3·8	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
3·9	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
4·0	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
4·1	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
4·2	6232	6243	6253	6263	6274	6284	6294	6304	6214	6325	1	2	3	4	5	6	7	8	9
4·3	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
4·4	6435	6444	6454	6464	7474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
4·5	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
4·6	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
4·7	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
4·8	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
4·9	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8

LOGARITHMS 5·0 TO 9·999

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
5·0	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
5·1	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
5·2	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
5·3	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
5·4	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7
5·5	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
5·6	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
5·7	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
5·8	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	2	2	3	4	5	5	6	7
5·9	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
6·0	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
6·1	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
6·2	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
6·3	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
6·4	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
6·5	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
6·6	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
6·7	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
6·8	8325	8331	8338	8244	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
6·9	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
7·0	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
7·1	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
7·2	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
7·3	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
7·4	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
7·5	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
7·6	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
7·7	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
7·8	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
7·9	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
8·0	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
8·1	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
8·2	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
8·3	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
8·4	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
8·5	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
8·6	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
8·7	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
8·8	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
8·9	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
9·0	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
9·1	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
9·2	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
9·3	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
9·4	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
9·5	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
9·6	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
9·7	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
9·8	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
9·9	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

ANTILOGARITHMS

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
·15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
·19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
·20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
·21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
·25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
·30	1995	2000	2004	2009	2034	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
·31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
·41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
·43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	8122	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
·46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

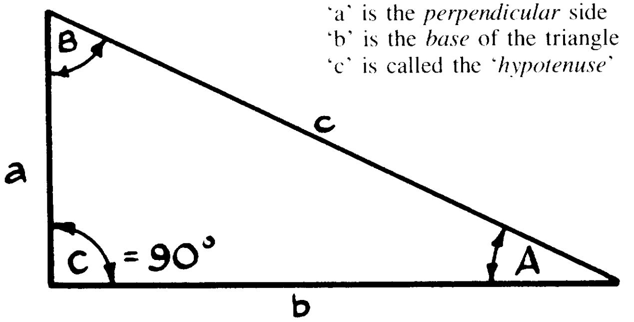
ANTILOGARITHMS

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	3	3	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	2219	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4918	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4667	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8091	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

FUNCTIONS OF ANGLES

These tables give values which enable angles to be described as tapers, the lengths and the sides of triangles to be determined, and such processes as angular marking out facilitated. These procedures are described in a later section.

The three basic functions are SINE, COSINE, and TANGENT, usually written as Sin, Cos, and Tan (the first being pronounced 'sine' to avoid confusion!). They are defined in terms of a right-angled triangle, as shown in the figure.



'a' is the *perpendicular* side
'b' is the *base* of the triangle
'c' is called the '*hypotenuse*'

Then: $\text{Sin A} = \dfrac{a}{c}$: $\text{Cos A} = \dfrac{b}{c}$: $\text{Tan A} = \dfrac{a}{b}$

and $\text{Sin B} = \dfrac{b}{c}$ $\quad \text{Cos B} = \dfrac{a}{c}$ $\quad \text{Tan B} = \dfrac{b}{a}$ if the triangle is turned over.

In words, Cos is 'base over hypotenuse'
Sin is 'perpendicular over hypotenuse'
Tan is 'perpendicular over base'

Note in this figure the convention that when letters are used to denote angles (instead of Greek symbols like θ) a capital letter means the angle, and the same letter but lower case means the opposite side.

Examination of the ratios of sides above shows that:

$$\text{Tan} = \frac{\text{Sin}}{\text{Cos}}$$

To use the tables, look up the angle in degrees in the LH column (pages **2**.14 to **2**.19).

The values are given in the body of the table at 6 minute (0·1°) intervals. At the right-hand side is a set of columns of *differences* at 1 minute intervals (one degree = 60 minutes). For Sine and Tan these differences are *added* to the value in the body of the table. For Cosines the differences are *subtracted*.

Example:
Find the Sine and Cosine of 35°44'.

Sin 35°42' is given as ·5835
difference for 2' = 5
Add ·5840 $\qquad$ Sin 35°44' = 0·5840

Cos $35°42'$ is given as $\cdot 8121$
difference for $2' =$ $\underline{3}$
Subtract $\underline{\cdot 8118}$ $\quad$ $\underline{\text{Cos } 35°44' = 0\cdot 8118}$

Note that for large angles (75° and above) the tangent of an angle is so large, and increasing so rapidly, that mean differences cannot be used. To find Tan to the nearest minute, divide the difference between adjacent columns by 6 to obtain the difference per minute of angle.

Example:
Find Tan 81°32'.

Under column $81°30'$ read $6\cdot 6912.$ (The units are suppressed in the table to save space)
Under column $81°36'$ read $6\cdot 7720.$
Difference for 6 minutes $= 0\cdot 0808.$
Difference per minute $= \underline{0\cdot 0135}$

Hence Tan $81°32' = 6\cdot 6912 + 2 \times 0\cdot 0135$
$= \underline{6\cdot 7182}$

There are other functions of angles, some being simply the reciprocals of the above,

thus: $1/\text{Sin}\theta = \text{Cosec}\theta$
$1/\text{Cos}\theta = \text{Sec}\theta$
$1/\text{Tan}\theta = \text{Cotan}\theta.$

Sin, Cos and Tan are by far the most useful.

If an angle is quoted in RADIANS, convert to degrees. There are 2π radians in a circle, so 1 radian = 57·30 degrees or 57°18'. (A radian is the angle subtended by an arc whose length is equal to the radius of the circle.) See p. **2**.20.

SINES

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
0	·0000	0017	0035	0052	0070	0087	0105	0122	0140	0157	3	6	9	12	15
1	·0175	0192	0209	0227	0244	0262	0279	0297	0314	0332	3	6	9	12	15
2	·0349	0366	0384	0401	0419	0436	0454	0471	0488	0506	3	6	9	12	15
3	·0523	0541	0558	0576	0593	0610	0628	0645	0663	0680	3	6	9	12	15
4	·0698	0715	0732	0750	0767	0785	0802	0819	0837	0854	3	6	9	12	15
5	·0872	0889	0906	0924	0941	0958	0976	0993	1011	1028	3	6	9	12	14
6	·1045	1063	1080	1097	1115	1132	1149	1167	1184	1201	3	6	9	12	14
7	·1219	1236	1253	1271	1288	1305	1323	1340	1357	1374	3	6	9	12	14
8	·1392	1409	1426	1444	1461	1478	1495	1513	1530	1547	3	6	9	12	14
9	·1564	1582	1599	1616	1633	1650	1668	1685	1702	1719	3	6	9	12	14
10	·1736	1754	1771	1788	1805	1822	1840	1857	1874	1891	3	6	9	12	14
11	·1908	1925	1942	1959	1977	1994	2011	2028	2045	2062	3	6	9	11	14
12	·2079	2096	2113	2130	2147	2164	2181	2198	2215	2232	3	6	9	11	14
13	·2250	2267	2284	2300	2317	2334	2351	2368	2385	2402	3	6	8	11	14
14	·2419	2436	2453	2470	2487	2504	2521	2538	2554	2571	3	6	8	11	14
15	·2588	2605	2622	2639	2656	2672	2689	2706	2723	2740	3	6	8	11	14
16	·2756	2773	2790	2807	2823	2840	2857	2874	2890	2907	3	6	8	11	14
17	·2924	2940	2957	2974	2990	3007	3024	3040	3057	3074	3	6	8	11	14
18	·3090	3107	3123	3140	3156	3173	3190	3206	3223	3239	3	6	8	11	14
19	·3256	3272	3289	3305	3322	3338	3355	3371	3387	3404	3	5	8	11	14
20	·3420	3437	3453	3469	3486	3502	3518	3535	3551	3567	3	5	8	11	14
21	·3584	3600	3616	3633	3649	3665	3681	3697	3714	3730	3	5	8	11	14
22	·3746	3762	3778	3795	3811	3827	3843	3859	3875	3891	3	5	8	11	14
23	·3907	3923	3939	3955	3971	3987	4003	4019	4035	4051	3	5	8	11	14
24	·4067	4083	4099	4115	4131	4147	4163	4179	4195	4210	3	5	8	11	13
25	·4226	4242	4258	4274	4289	4305	4321	4337	4352	4368	3	5	8	11	13
26	·4384	4399	4415	4431	4446	4462	4478	4493	4509	4524	3	5	8	10	13
27	·4540	4555	4571	4586	4602	4617	4633	4648	4664	4679	3	5	8	10	13
28	·4695	4710	4726	4741	4756	4772	4787	4802	4818	4833	3	5	8	10	13
29	·4848	4863	4879	4894	4909	4924	4939	4955	4970	4985	3	5	8	10	13
30	·5000	5015	5030	5045	5060	5075	5090	5105	5120	5135	3	5	8	10	13
31	·5150	5165	5180	5195	5210	5225	5240	5255	5270	5284	2	5	7	10	12
32	·5299	5314	5329	5344	5358	5373	5388	5402	5417	5432	2	5	7	10	12
33	·5446	5461	5476	5490	5505	5519	5534	5548	5563	5577	2	5	7	10	12
34	·5592	5606	5621	5635	5650	5664	5678	5693	5707	5721	2	5	7	10	12
35	·5736	4750	5764	5779	5793	5807	5821	5835	5850	5864	2	5	7	10	12
36	·5878	5892	5906	5920	5934	5948	5962	5976	5990	6004	2	5	7	9	12
37	·6018	6032	6046	6060	6074	6088	6101	6115	6129	6143	2	5	7	9	12
38	·6157	6170	6184	6198	6211	6225	6239	6252	6266	6280	2	5	7	9	11
39	·6293	6307	6320	6334	6347	6361	6374	6388	6401	6414	2	4	7	9	11
40	·6428	6441	6455	6468	6481	6494	6508	6521	6534	6547	2	4	7	9	11
41	·6561	6574	6587	6600	6613	6626	6639	6652	6665	6678	2	4	7	9	11
42	·6691	6704	6717	6730	6743	6756	6769	6782	6794	6807	2	4	6	9	11
43	·6820	6833	6845	6858	6871	6884	6896	6909	6921	6934	2	4	6	8	11
44	·6947	6959	6972	6984	6997	7009	7022	7034	7046	7059	2	4	6	8	10

SINES

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
45	·7071	7083	7096	7108	7120	7133	7145	7157	7169	7181	2	4	6	8	10
46	·7193	7206	7218	7230	7242	7254	7266	7278	7290	7302	2	4	6	8	10
47	·7314	7325	7337	7349	7361	7373	7385	7396	7408	7420	2	4	6	8	10
48	·7431	7443	7455	7466	7478	7490	7501	7513	7524	7536	2	4	6	8	10
49	·7547	7558	7570	7581	7593	7604	7615	7627	7638	7649	2	4	6	8	9
50	·7660	7672	7683	7694	7705	7716	7727	7738	7749	7760	2	4	6	7	9
51	·7771	7782	7793	7804	7815	7826	7837	7848	7859	7869	2	4	5	7	9
52	·7880	7891	7902	7912	7923	7934	7944	7955	7965	7976	2	4	5	7	9
53	·7986	7997	8007	8018	8028	8039	8049	8059	8070	8080	2	3	5	7	9
54	·8090	8100	8111	8121	8131	8141	8151	8161	8171	8181	2	3	5	7	8
55	·8192	8202	8211	8221	8231	8241	8251	8261	8271	8281	2	3	5	7	8
56	·8290	8300	8310	8320	8329	8339	8348	8358	8368	8377	2	3	5	6	8
57	·8387	8396	8406	8415	8425	8438	8443	8453	8462	8471	2	3	5	6	8
58	·8480	8490	8499	8508	8517	8526	8536	8545	8554	8563	2	3	5	6	8
59	·8572	8581	8590	8599	8607	8616	8625	8634	8643	8652	1	3	4	6	7
60	·8660	8669	8678	8686	8695	8704	8712	8721	8729	8738	1	3	4	6	7
61	·8746	8755	8763	8771	8780	8788	8796	8805	8813	8821	1	3	4	6	7
62	·8829	8838	8846	8854	8862	8870	8878	8886	8894	8902	1	3	4	5	7
63	·8910	8918	8926	8934	8942	8949	8957	8965	8973	8980	1	3	4	5	6
64	·8988	8996	9003	9011	9018	9026	9033	9041	9048	9056	1	3	4	5	6
65	·9063	9070	9078	9085	9092	9100	9107	9114	9121	9128	1	2	4	5	6
66	·9135	9143	9150	9157	9164	9171	9178	9184	9191	9198	1	2	3	5	6
67	·9205	9212	9219	9225	9232	9239	9245	9252	9259	9265	1	2	3	4	6
68	·9272	9278	9285	9291	9298	9304	9311	9317	9323	9330	1	2	3	4	5
69	·9336	9342	9348	9354	9361	9367	9373	9379	9385	9391	1	2	3	4	5
70	·9397	9403	9409	9415	9421	9426	9432	9438	9444	9449	1	2	3	4	5
71	·9455	9461	9466	9472	9478	9483	9489	9494	9500	9505	1	2	3	4	5
72	·9511	9516	9521	9527	9532	9537	9542	9548	9553	9558	1	2	3	3	4
73	·9563	9568	9573	9578	9583	9588	9593	9598	9603	9608	1	2	2	3	4
74	·9613	9617	9622	9627	9632	9636	9641	9646	9650	9655	1	2	2	3	4
75	·9659	9664	9668	9673	9677	9681	9686	9690	9694	9699	1	1	2	3	4
76	·9703	9707	9711	9715	9720	9724	9728	9732	9736	9740	1	1	2	3	3
77	·9744	9748	9751	9755	9759	9763	9767	9770	9774	9778	1	1	2	3	3
78	·9781	9785	9789	9792	9796	9799	9803	9806	9810	9813	1	1	2	2	3
79	·9816	9820	9823	9826	9829	9833	9836	9839	9842	9845	1	1	2	2	3
80	·9848	9851	9854	9857	9860	9863	9866	9869	9871	9874	0	1	1	2	2
81	·9877	9880	9882	9885	9888	9890	9893	9895	9898	9900	0	1	1	2	2
82	·9903	9905	9907	9910	9912	9914	9917	9919	9921	9923	0	1	1	2	2
83	·9925	9928	9930	9932	9934	9936	9938	9940	9942	9943	0	1	1	1	2
84	·9945	9947	9949	9951	9952	9954	9956	9957	9959	9960	0	1	1	1	2
85	·9962	9963	9965	9966	9968	9969	9971	9972	9973	9974	0	0	1	1	1
86	·9976	9977	9978	9979	9980	9981	9982	9983	9984	9985	0	0	1	1	1
87	·9986	9987	9988	9989	9990	9990	9991	9992	9993	9993	0	0	0	1	1
89	·9994	9995	9995	9996	9996	9997	9997	9997	9998	9998	0	0	0	0	0
89	·0009	9999	9999	9999	9999	1·000	1·000	1·000	1·000	1·000	0	0	0	0	0
90	1·000														

COSINES

[*Numbers in difference columns to be subtracted, not added.*]

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
0	1·000	1·000	1·000	1·000	1·000	1·000	·9999	9999	9999	9999	0	0	0	0	0
1	·9998	9998	9998	9997	9997	9997	9996	9996	9995	9995	0	0	0	0	0
2	·9994	9993	9993	9992	9991	9990	9990	9989	9988	9987	0	0	0	1	1
3	·9986	9985	9984	9983	9982	9981	9980	9979	9978	9977	0	0	1	1	1
4	·9976	9974	9973	9972	9971	9969	9968	9966	9965	9963	0	0	1	1	1
5	·9962	9960	9959	9957	9956	9954	9952	9951	9949	9947	0	1	1	1	2
6	·9945	9943	9942	9940	9938	9936	9934	9932	9930	9928	0	1	1	1	2
7	·9925	9923	9921	9919	9917	9914	9912	9910	9907	9905	0	1	1	2	2
8	·9903	9900	9898	9895	9893	9890	9888	9885	9882	9880	0	1	1	2	2
9	·9877	9874	9871	9869	9866	9863	9860	9857	9854	9851	0	1	1	2	2
10	·9848	9845	9842	9839	9836	9833	9829	9826	9823	9820	1	1	2	2	3
11	·9816	9813	9810	9806	9803	9799	9796	9792	9789	9785	1	1	2	2	3
12	·9781	9778	9774	9770	9767	9763	9759	9755	9751	9748	1	1	2	3	3
13	·9744	9740	9736	9732	9728	9724	9720	9715	9711	9707	1	1	2	3	3
14	·9703	9699	9694	9690	9686	9681	9677	9673	9668	9664	1	1	2	3	4
15	·9659	9655	9650	9646	9641	9636	9632	9627	9622	9617	1	2	2	3	4
16	·9613	9608	9603	9598	9593	9588	9583	9578	9573	9568	1	2	2	3	4
17	·9563	9558	9553	9548	9542	9537	9532	9527	9521	9516	1	2	3	3	4
18	·9511	9505	9500	9494	9489	9483	9478	9472	9466	9461	1	2	3	4	5
19	·9455	9449	9444	9438	9432	9426	9421	9415	9409	9403	1	2	3	4	5
20	·9397	9391	9385	9379	9373	9367	9361	9354	9348	9342	1	2	3	4	5
21	·9336	9330	9323	9317	9311	9304	9298	9291	9285	9278	1	2	3	4	5
22	·9272	9265	9259	9252	9245	9239	9232	9225	9219	9212	1	2	3	4	6
23	·9205	9198	9191	9184	9178	9171	9164	9157	9150	9143	1	2	3	5	6
24	·9135	9128	9121	9114	9107	9100	9092	9085	9078	9070	1	2	4	5	6
25	·9063	9056	9048	9041	9033	9026	9018	9011	9003	8996	1	3	4	5	6
26	·8988	8980	8973	8965	8957	8949	8942	8934	8926	8918	1	3	4	5	6
27	·8910	8902	8894	8886	8878	8870	8862	8854	8846	8838	1	3	4	5	7
28	·8829	8821	8813	8805	8796	8788	8780	8771	8763	8755	1	3	4	6	7
29	·8746	8738	8729	8721	8712	8704	8695	8686	8678	8669	1	3	4	6	7
30	·8660	8652	8643	8634	8625	8616	8607	8599	8590	8581	1	3	4	6	7
31	·8572	8563	8554	8545	8536	8526	8517	8508	8499	8490	2	3	5	6	8
32	·8480	8471	8462	8453	8443	8434	8425	8415	8406	8396	2	3	5	6	8
33	·8387	8277	8368	8358	8348	8339	8329	8320	8310	8300	2	3	5	6	8
34	·8290	8281	8271	8261	8251	8241	8231	8221	8211	8202	2	3	5	7	8
35	·8192	8181	8171	8161	8151	8141	8131	8121	8111	8100	2	3	5	7	8
36	·8090	8080	8070	8059	8049	8039	8028	8018	8007	7997	2	3	5	7	9
37	·7986	7976	7965	7955	7944	7934	7923	7912	7902	7891	2	4	5	9	9
38	·7880	7869	7859	7848	7837	7826	7815	7804	7793	7782	2	4	5	7	9
39	·7771	7760	7749	7738	7727	7716	7705	7694	7683	7672	2	4	6	7	9
40	·7660	7649	7638	7627	7615	7604	7593	7581	7570	7559	2	4	6	8	9
41	·7547	7536	7524	7513	7501	7490	7478	7466	7455	7443	2	4	6	8	10
42	·7431	7420	7408	7396	7385	7373	7361	7349	7337	7325	2	4	6	8	10
43	·7314	7302	7290	7278	7266	7254	7242	7230	7218	7206	2	4	6	8	10
44	·7193	7181	7169	7157	7145	7133	7120	7108	7096	7083	2	4	6	8	10

COSINES

[Numbers in difference columns to be subtracted, not added.]

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
45	·7071	7059	7046	7034	7022	7009	6997	6984	6972	6959	2	4	6	8	10
46	·6947	6934	6921	6909	6896	6884	6871	6858	6845	6833	2	4	6	8	11
47	·6820	6807	6794	6782	6769	6756	6743	6730	6717	6704	2	4	6	9	11
48	·6691	6678	6665	6652	6639	6626	6613	6600	6587	6574	2	4	7	9	11
49	·6561	6547	6534	6521	6508	6494	6481	6468	6455	6441	2	4	7	9	11
50	·6428	6414	6401	6388	6374	6361	6347	6334	6320	6307	2	4	7	9	11
51	·6293	6280	6266	6252	6239	6225	6211	6198	6184	6170	2	5	7	9	11
52	·6157	6143	6129	6115	6101	6088	6074	6060	6046	6032	2	5	7	9	12
53	·6018	6004	5990	5976	5962	5948	5934	5920	5906	5892	2	5	7	9	12
54	·5878	5864	5850	5835	5821	5807	5793	5779	5764	5750	2	5	7	9	12
55	·5736	5721	5707	5693	5678	5664	5650	5635	5621	5606	2	5	7	10	12
56	·5592	5577	5563	5548	5534	5519	5505	5490	5476	5461	2	5	7	10	12
57	·5446	5432	5417	5402	5388	5373	5358	5344	5329	5314	2	5	7	10	12
58	·5299	5284	5270	5255	5240	5225	5210	5195	5180	5165	2	5	7	10	12
59	·5150	5135	5120	5105	5090	5075	5060	5045	5030	5015	3	5	8	10	13
60	·5000	4985	4970	4955	4939	4924	4909	4894	4879	4863	3	5	8	10	13
61	·4848	4833	4818	4802	4787	4772	4756	4741	4726	4710	3	5	8	10	13
62	·4695	4679	4664	4648	4633	4617	4602	4586	4571	4555	3	5	8	10	13
63	·4540	4524	4509	4493	4478	4462	4446	4431	4415	4399	3	5	8	10	13
64	·4384	4368	4352	4337	4321	4305	4289	4274	4258	4242	3	5	8	11	13
65	·4226	4210	4195	4179	4163	4147	4131	4115	4099	4083	3	5	8	11	13
66	·4067	4051	4035	4019	4003	3987	3971	3955	3939	3923	3	5	8	11	14
67	·3907	3891	3875	3859	3843	3827	3811	3795	3778	3762	3	5	8	11	14
68	·3746	3730	3714	3697	3681	3665	3649	3633	3616	3600	3	5	8	11	14
69	·3584	3567	3551	3535	3518	3502	3486	3469	3453	3437	3	5	8	11	14
70	·3420	3404	3387	3371	3355	3338	3322	3305	3289	3272	3	5	8	11	14
71	·3256	3239	3223	3206	3190	3173	3156	3140	3123	3107	3	6	8	11	14
72	·3090	3074	3057	3040	3024	3007	2990	2974	2957	2940	3	6	8	11	14
73	·2924	2907	2890	2874	2857	2840	2823	2807	2790	2773	3	6	8	11	14
74	·2756	2740	2723	2706	2689	2672	2656	2639	2622	2605	3	6	8	11	14
75	·2588	2571	2554	2538	2521	2504	2487	2470	2453	2436	3	6	8	11	14
76	·2419	2402	2385	2368	2351	2334	2317	2300	2284	2267	3	6	8	11	14
77	·2270	2233	2215	2198	2181	2164	2147	2130	2113	2096	3	6	9	11	14
78	·2079	2062	2045	2028	2011	1994	1977	1959	1942	1925	3	6	9	11	14
79	·1908	1891	1874	1857	1840	1822	1805	1788	1771	1754	3	6	9	11	14
80	·1736	1719	1702	1685	1668	1650	1633	1616	1599	1582	3	6	9	12	14
81	·1564	1547	1530	1513	1495	1478	1461	1444	1426	1409	3	6	9	12	14
82	·1392	1374	1357	1340	1323	1305	1288	1271	1253	1236	3	6	9	12	14
83	·1219	1201	1184	1167	1149	1132	1115	1097	1080	1063	3	6	9	12	14
84	·1045	1028	1011	0993	0976	0958	0941	0924	0906	0889	3	6	9	12	14
85	·0872	0854	0837	0819	0802	0785	0767	0750	0732	0715	3	6	9	12	15
86	·0698	0680	0663	0645	0628	0610	0593	0576	0558	0541	3	6	9	12	15
87	·0523	0506	0488	0471	0454	0436	0419	0401	0384	0366	3	6	9	12	15
88	·0349	0332	0314	0297	0279	0262	0244	0227	0209	0192	3	6	9	12	15
89	·0175	0157	0140	0122	0105	0087	0070	0052	0035	0017	3	6	9	12	15
90	·0000														

TANGENTS

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
0	·0000	0017	0035	0052	0070	0087	0105	0122	0140	0157	3	6	9	12	15
1	·0175	0192	0209	0227	0244	0262	0279	0297	0314	0332	3	6	9	12	15
2	·0349	0367	0384	0402	0419	0437	0454	0472	0489	0507	3	6	9	12	15
3	·0524	0542	0559	0577	0594	0612	0629	0647	0664	0682	3	6	9	12	15
4	·0699	0717	0734	0752	0769	0787	0805	0822	0840	0857	3	6	9	12	15
5	·0875	0892	0910	0928	0945	0963	0981	0998	1016	1033	3	6	9	12	15
6	·1051	1069	1086	1104	1122	1139	1157	1175	1192	1210	3	6	9	12	15
7	·1228	1246	1263	1281	1299	1317	1334	1352	1370	1388	3	6	9	12	15
8	·1405	1423	1441	1459	1477	1495	1512	1530	1548	1566	3	6	9	12	15
9	·1584	1602	1620	1638	1655	1673	1691	1709	1727	1745	3	6	9	12	15
10	·1763	1781	1799	1817	1835	1853	1871	1890	1908	1926	3	6	9	12	15
11	·1944	1962	1980	1998	2016	2035	2053	2071	2089	2107	3	6	9	12	15
12	·2126	2144	2162	2180	2199	2217	2235	2254	2272	2290	3	6	9	12	15
13	·2309	2327	2345	2364	2382	2401	2419	2438	2456	2475	3	6	9	12	15
14	·2493	2512	2530	2549	2568	2586	2605	2623	2642	2662	3	6	9	12	16
15	·2679	2698	2717	2736	2754	2773	2792	2811	2830	2849	3	6	9	13	16
16	·2867	2886	2905	2924	2943	2962	2981	3000	3019	3038	3	6	9	13	16
17	·3057	3076	3096	3115	3134	3153	3172	3191	3211	3230	3	6	10	13	16
18	·3249	3269	3288	3307	3327	3346	3365	3385	3404	3424	3	6	10	13	16
19	·3443	3463	3482	3502	3522	3541	3561	3581	3600	3620	3	7	10	13	16
20	·3640	3659	3679	3699	3719	3739	3759	3779	3799	3819	3	7	10	13	17
21	·3839	3859	3879	3899	3919	3939	3959	3979	4000	4020	3	7	10	13	17
22	·4040	4061	4081	4101	4122	4142	4163	4183	4204	4224	3	7	10	14	17
23	·4245	4265	4286	4307	4327	4348	4369	4390	4411	4431	3	7	10	14	17
24	·4452	4473	4494	4515	4536	4557	4578	4599	4621	4642	4	7	11	14	18
25	·4663	4684	4706	4727	4748	4770	4791	4813	4834	4856	4	7	11	14	18
26	·4877	4899	4921	4942	4964	4986	5008	5029	5051	5073	4	7	11	15	18
27	·5095	5117	5139	5161	5184	5206	5228	5250	5272	5295	4	7	11	15	18
28	·5317	5340	5362	5384	5407	5430	5452	5475	5498	5520	4	8	11	15	19
29	·5543	5566	5589	5612	5635	5658	5681	5704	5727	5750	4	8	12	15	19
30	·5774	5797	5820	5844	5867	5890	5914	5938	5961	5985	4	8	12	16	20
31	·6009	6032	6056	6080	6104	6128	6152	6176	6200	6224	4	8	12	16	20
32	·6249	6273	6297	6322	6346	6371	6395	6420	6445	6469	4	8	12	16	20
33	·6494	6519	6544	6569	6594	6619	6644	6669	6694	6720	4	8	13	17	21
34	·6745	6771	6796	6822	6847	6873	6899	6924	6950	6976	4	9	13	17	21
35	·7002	7028	7054	7080	7107	7133	7159	7186	7212	7239	4	9	13	18	22
36	·7265	7292	7319	7346	7373	7400	7427	7454	7481	7508	5	9	14	18	23
37	·7536	7563	7590	7618	7646	7673	7701	7729	7757	7785	5	9	14	18	23
38	·7813	7841	7869	7898	7926	7954	7983	8012	8040	8069	5	9	14	19	24
39	·8098	8127	8156	8185	8214	8243	8273	8302	8332	8361	5	10	15	20	24
40	·8391	8421	8451	8481	8511	8541	8571	8601	8632	8662	5	10	15	20	25
41	·8693	8724	8754	8785	8816	8847	8878	8910	8941	8972	5	10	16	21	26
42	·9004	9036	9067	9099	9131	9163	9195	9228	9260	9293	5	11	16	21	27
43	·9325	9358	9391	9424	9457	9490	9523	9556	9590	9623	6	11	17	22	28
44	·9657	9691	9725	9759	9793	9827	9861	9896	9930	9965	6	11	17	23	29

TANGENTS

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
45	1·0000	0035	0070	0105	0141	0176	0212	0247	0283	0319	6	12	18	24	30
46	1·0355	0392	0428	0464	0501	0538	0575	0612	0649	0686	6	12	18	25	31
47	1·0724	0761	0799	0837	0875	0913	0951	0990	1028	1067	6	13	19	25	32
48	1·1106	1145	1184	1224	1263	1303	1343	1383	1423	1463	7	13	20	27	33
49	1·1504	1544	1585	1626	1667	1708	1750	1792	1833	1875	7	14	21	28	34
50	1·1918	1960	2002	2045	2088	2131	2174	2218	2261	2305	7	14	22	29	36
51	1·2349	2393	2437	2482	2527	2572	2617	2662	2708	2753	8	15	23	30	38
52	1·2799	2846	2892	2938	2985	3032	3079	3127	3175	3222	8	16	24	31	39
53	1·3270	3319	3367	3416	3465	3514	3564	3613	3663	3713	8	16	25	33	41
54	1·3764	3814	3865	3916	3968	4019	4071	4124	4176	4229	9	17	26	34	43
55	1·4281	4335	4388	4442	4496	4550	4605	4659	4715	4770	9	18	27	36	45
56	1·4826	4882	4938	4994	5051	5108	5166	5224	5282	5340	10	19	29	38	48
57	1·5399	5458	5517	5577	5637	5697	5757	5818	5880	5941	10	20	30	40	50
58	1·6003	6066	6128	6191	6255	6319	6383	6447	6512	6577	11	21	32	43	53
59	1·6643	6709	6775	6842	6909	6977	7045	7113	7182	7251	11	23	34	45	56
60	1·7321	7391	7461	7532	7603	7675	7747	7820	7893	7966	12	24	36	48	60
61	1·8040	8115	8190	8265	8341	8418	8495	8572	8650	8728	13	26	38	51	64
62	1·8807	8887	8967	9047	9128	9210	9292	9375	9458	9542	14	27	41	55	68
63	1·9626	9711	9797	9883	9970	2·0057	2·0145	2·0233	2·0323	2·0413	15	29	44	58	73
64	2·0503	0594	0686	0778	0872	0965	1060	1155	1251	1348	16	31	47	63	78
65	2·1445	1543	1642	1742	1842	1943	2045	2148	2251	2355	17	34	51	68	85
66	2·2460	2566	2673	2781	2889	2998	3109	3220	3332	3445	18	37	55	73	92
67	2·3559	3673	3789	3906	4023	4142	4262	4383	4504	4627	20	40	60	79	99
68	2·4751	4876	5002	5129	5257	5386	5517	5649	5782	5916	22	43	65	87	108
69	2·6051	6187	6325	6464	6605	6746	6889	7034	7179	7326	24	47	71	95	119
70	2·7475	7625	7776	7929	8083	8239	8397	8556	8716	8878	26	52	78	104	131
71	2·9042	9208	9375	9544	9714	9887	3·0061	3·0237	3·0415	3·0595	29	58	87	116	145
72	3·0777	0961	1146	1334	1524	1716	1910	2106	2305	2500	32	64	96	129	161
73	3·2709	2914	3122	3332	3544	3759	3977	4197	4420	4646	36	72	108	144	180
74	3·4874	5105	5339	5576	5816	6059	6305	6554	6806	7062	41	81	122	163	204
75	3·7321	7583	7848	8118	8391	8667	8947	9232	9520	9812	46	93	139	186	232
76	4·0108	0408	0713	1022	1335	1653	1976	2303	2635	2972	53	107	160	213	267
77	4·3315	3662	4015	4374	4737	5107	5483	5864	6252	6646					
78	4·7046	7453	7867	8288	8716	9152	9594	5·0045	5·0504	5·0970	Mean differences cease to be sufficiently accurate.				
79	5·1446	1929	2422	2924	3435	3955	4486	5026	5578	6140					
80	5·6713	7297	7894	8502	9124	9758	6·0405	6·1066	6·1742	6·2432					
81	6·3138	3859	4596	5350	6122	6912	7720	8548	9395	7·0264					
82	7·1154	2066	3002	3962	4947	5958	6996	8062	9158	8·0285					
83	8·1443	2636	3863	5126	6427	7769	9152	9·0579	9·2052	9·3572					
84	9·5144	9·677	9·845	10·02	10·20	10·39	10·58	10·78	10·99	11·20					
85	11·43	11·66	11·91	12·16	12·43	12·71	13·00	13·30	13·62	13·95					
86	14·30	14·67	15·06	15·46	15·89	16·35	16·83	17·34	17·89	18·46					
87	19·08	19·74	20·45	21·20	22·02	22·90	23·86	24·90	26·03	27·27					
88	28·64	30·14	31·82	33·69	35·80	38·19	40·92	44·07	47·74	52·08					
89	57·29	63·66	71·62	81·85	95·49	114·6	143·2	191·0	286·5	573·0					
90															

Note: Units digits are printed only in the first column or at the point of change.

DEGREES TO RADIANS

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
0	·0000	0017	0035	0052	0070	0087	0105	0122	0140	0157	3	6	9	12	15
1	·0175	0192	0209	0227	0244	0262	0279	0297	0314	0332	3	6	9	12	15
2	·0349	0367	0384	0401	0419	0436	0454	0471	0489	0506	3	6	9	12	15
3	·0524	0541	0559	0576	0593	0611	0628	0646	0663	0681	3	6	9	12	15
4	·0698	0716	0733	0750	0768	0785	0803	0820	0838	0855	3	6	9	12	15
5	·0873	0890	0908	0925	0942	0960	0977	0995	1012	1030	3	6	9	12	15
6	·1047	1065	1082	1100	1117	1134	1152	1169	1187	1204	3	6	9	12	15
7	·1222	1239	1257	1274	1292	1309	1326	1344	1361	1379	3	6	9	12	15
8	·1396	1414	1431	1449	1466	1484	1501	1518	1536	1553	3	6	9	12	15
9	·1571	1588	1606	1623	1641	1658	1676	1693	1710	1728	3	6	9	12	15
10	·1745	1763	1780	1798	1815	1833	1850	1868	1885	1902	3	6	9	12	15
11	·1920	1937	1955	1972	1990	2007	2025	2042	2060	2077	3	6	9	12	15
12	·2094	2112	2129	2147	2164	2182	2199	2217	2234	2251	3	6	9	12	15
13	·2269	2286	2304	2321	2339	2356	2374	2391	2409	2426	3	6	9	12	15
14	·2443	2461	2478	2496	2513	2531	2548	2566	2583	2601	3	6	9	12	15
15	·2618	2635	2653	2670	2688	2705	2723	2740	2758	2775	3	6	9	12	15
16	·2793	2810	2827	2845	2862	2880	2897	2915	2932	2950	3	6	9	12	15
17	·2967	2985	3002	3019	3037	3054	3072	3089	3107	3124	3	6	9	12	15
18	·3142	3159	3176	3194	3211	3229	3246	3264	3281	3299	3	6	9	12	15
19	·3316	3334	3351	3368	3386	3403	3421	3438	3456	3473	3	6	9	12	15
20	·3491	3508	3526	3543	3560	3578	3595	3613	3630	3648	3	6	9	12	15
21	·3665	3683	3700	3718	3735	3752	3770	3787	3805	3822	3	6	9	12	15
22	·3840	3857	3875	3892	3910	3927	3944	3962	3979	3997	3	6	9	12	15
23	·4014	4032	4049	4067	4084	4102	4119	4136	4154	4171	3	6	9	12	15
24	·4189	4206	4224	4241	4259	4276	4294	4311	4328	4346	3	6	9	12	15
25	·4363	4381	4398	4416	4433	4451	4468	4485	4503	4520	3	6	9	12	15
26	·4538	4555	4573	4590	4608	4625	4643	4660	4677	4695	3	6	9	12	15
27	·4712	4730	4747	4765	4782	4800	4817	4835	4852	4869	3	6	9	12	15
28	·4887	4904	4922	4939	4957	4974	4992	5009	5027	5044	3	6	9	12	15
29	·5061	5079	5096	5114	5131	5149	5166	5184	5201	5219	3	6	9	12	15
30	·5236	5253	5271	5288	5306	5323	5341	5358	5376	5393	3	6	9	12	15
31	·5411	5428	5445	5463	5480	5498	5515	5533	5550	5568	3	6	9	12	15
32	·5585	5603	5620	5637	5655	5672	5690	5707	5725	5742	3	6	9	12	15
33	·5760	5777	5794	5812	5829	5847	5864	5882	5899	5917	3	6	9	12	15
34	·5934	5952	5969	5986	6004	6021	6039	6056	6074	6091	3	6	9	12	15
35	·6109	6126	6144	6161	6178	6196	6213	6231	6248	6266	3	6	9	12	15
36	·6283	6301	6318	6336	6353	6370	6388	6405	6423	6440	3	6	9	12	15
37	·6458	6475	6493	6510	6528	6545	6562	6580	6597	6615	3	6	9	12	15
38	·6632	6650	6667	6685	6702	6720	6737	6754	6772	6789	3	6	9	12	15
39	·6807	6824	6842	6859	6877	6894	6912	6929	6946	6964	3	6	9	12	15
40	·6981	6999	7016	7034	7051	7069	7086	7103	7121	7138	3	6	9	12	15
41	·7156	7173	7191	7208	7226	7243	7261	7278	7295	7313	3	6	9	12	15
42	·7330	7348	7365	7383	7400	7418	7435	7453	7470	7487	3	6	9	12	15
43	·7505	7522	7540	7557	7575	7592	7610	7627	7645	7662	3	6	9	12	15
44	·7679	7697	7714	7732	7749	7767	7784	7802	7819	7837	3	6	9	12	15

DEGREES TO RADIANS

Degrees	0′ 0°·0	6′ 0°·1	12′ 0°·2	18′ 0°·3	24′ 0°·4	30′ 0°·5	36′ 0°·6	42′ 0°·7	48′ 0°·8	54′ 0°·9	Mean Differences 1	2	3	4	5
45	·7854	7871	7889	7906	7924	7941	7959	7976	7994	8011	3	6	9	12	15
46	·8029	8046	8063	8081	8098	8116	8133	8151	8168	8186	3	6	9	12	15
47	·8203	8221	8238	8255	8273	8290	8308	8325	8343	8360	3	6	9	12	15
48	·8378	8395	8412	8430	8447	8465	8482	8500	8517	8535	3	6	9	12	15
49	·8552	8570	8587	8604	8622	8639	8657	8674	8692	8709	3	6	9	12	15
50	·8727	8744	8762	8779	8796	8814	8831	8849	8866	8884	3	6	9	12	15
51	·8901	8919	8936	8954	8971	8988	9006	9023	9041	9058	3	6	9	12	15
52	·9076	9093	9111	9128	9146	9163	9180	9198	9215	9233	3	6	9	12	15
53	·9250	9268	9285	9303	9320	9338	9355	9372	9390	9407	3	6	9	12	15
54	·9425	9442	9460	9477	9495	9512	9529	9547	9564	9582	3	6	9	12	15
55	·9599	9617	9634	9652	9669	9687	9704	9721	9739	9756	3	6	9	12	15
56	·9774	9791	9809	9826	9844	9861	9879	9896	9913	9931	3	6	9	12	15
57	·9948	9966	9983	1·0001	1·0018	1·0036	1·0053	1·0071	1·0088	1·0105	3	6	9	12	15
58	1·0123	0140	0158	0175	0193	0210	0228	0245	0263	0280	3	6	9	12	15
59	1·0297	0315	0332	0350	0367	0385	0402	0420	0437	0455	3	6	9	12	15
60	1·0472	0489	0507	0524	0542	0559	0577	0594	0612	0629	3	6	9	12	15
61	1·0647	0664	0681	0699	0716	0734	0751	0769	0786	0804	3	6	9	12	15
62	1·0821	0838	0856	0873	0891	0908	0926	0943	0961	0978	3	6	9	12	15
63	1·0996	1013	1030	1048	1065	1083	1100	1118	1135	1153	3	6	9	12	15
64	1·1170	1188	1205	1222	1240	1257	1275	1292	1310	1327	3	6	9	12	15
65	1·1345	1362	1380	1397	1414	1432	1449	1467	1484	1502	3	6	9	12	15
66	1·1519	1537	1554	1572	1589	1606	1624	1641	1659	1676	3	6	9	12	15
67	1·1694	1711	1729	1746	1764	1781	1798	1816	1833	1851	3	6	9	12	15
68	1·1868	1886	1903	1921	1938	1956	1973	1990	2008	2025	3	6	9	12	15
69	1·2043	2060	2078	2095	2113	2130	2147	2165	2182	2200	3	6	9	12	15
70	1·2217	2235	2252	2270	2287	2305	2322	2339	2357	2374	3	6	9	12	15
71	1·2392	2409	2427	2444	2462	2479	2497	2514	2531	2549	3	6	9	12	15
72	1·2566	2584	2601	2619	2636	2654	2671	2689	2706	2723	3	6	9	12	15
73	1·2741	2758	2776	2793	2811	2828	2846	2863	2881	2898	3	6	9	12	15
74	1·2915	2933	2950	2968	2985	3003	3020	3038	3055	3073	3	6	9	12	15
75	1·3090	3107	3125	3142	3160	3177	3195	3212	3230	3247	3	6	9	12	15
76	1·3265	3282	3299	3317	3334	3352	3369	3387	3404	3422	3	6	9	12	15
77	1·3439	3456	3474	3491	3509	3526	3544	3561	3579	3596	3	6	9	12	15
78	1·3614	3631	3648	3666	3683	3701	3718	3736	3753	3771	3	6	9	12	15
79	1·3788	3806	3823	3840	3858	3875	3893	3910	3928	3945	3	6	9	12	15
80	1·3963	3980	3998	4015	4032	4050	4067	4085	4102	4120	3	6	9	12	15
81	1·4137	4155	4172	4190	4207	4224	4242	4259	4277	4294	3	6	9	12	15
82	1·4312	4329	4347	4364	4382	4399	4416	4434	4451	4469	3	6	9	12	15
83	1·4486	4504	4521	4539	4556	4573	4591	4608	4626	4643	3	6	9	12	15
84	1·4661	4678	4696	4713	4731	4748	4765	4783	4800	4818	3	6	9	12	15
85	1·4835	4853	4870	4888	4905	4923	4940	4957	4975	4992	3	6	9	12	15
86	1·5010	5027	5045	5062	5080	5097	5115	5132	5149	5167	3	6	9	12	15
87	1·5184	5202	5219	5237	5254	5272	5289	5307	5324	5341	3	6	9	12	15
88	1·5359	5376	5394	5411	5429	5446	5464	5481	5499	5516	3	6	9	12	15
89	1·5533	5551	5568	5586	5603	5621	5638	5656	5673	5691	3	6	9	12	15

POWERS, ROOTS AND RECIPROCALS

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\sqrt{10n}$	$\sqrt[3]{10n}$	$\sqrt[3]{100n}$	$\frac{1}{n}$
1	1	1	1	1	3·162	2·154	4·642	1
2	4	8	1·414	1·260	4·472	2·714	5·848	·5000
3	9	27	1·732	1·442	5·477	3·107	6·694	·3333
4	16	64	2	1·587	6·325	3·420	7·368	·2500
5	25	125	2·236	1·710	7·071	3·684	7·937	·2000
6	36	216	2·449	1·817	7·746	3·915	8·434	·1667
7	49	343	2·646	1·913	8·367	4·121	8·879	·1429
8	64	512	2·828	2·000	8·944	4·309	9·283	·1250
9	81	729	3·000	2·080	9·487	4·481	9·655	·1111
10	100	1000	3·162	2·154	10·0	4·642	10·000	·1000
11	121	1331	3·317	2·224	10·488	4·791	10·323	·09091
12	144	1728	3·464	2·289	10·954	4·932	10·627	·08333
13	169	2197	3·606	2·351	11·402	5·066	10·914	·07692
14	196	2744	3·742	2·410	11·832	5·192	11·187	·07143
15	225	3375	3·873	2·466	12·247	5·313	11·447	·06667
16	256	4096	4·000	2·520	12·649	5·429	11·696	·06250
17	289	4913	4·123	2·571	13·038	5·540	11·935	·05882
18	324	5832	4·243	2·621	13·416	5·646	12·164	·05556
19	361	6859	4·359	2·668	13·784	5·749	12·386	·05263
20	400	8000	4·472	2·714	14·142	5·848	12·599	·0500
21	441	9261	4·583	2·759	14·491	5·944	12·806	·04762
22	484	10648	4·690	2·802	14·832	6·037	13·006	·04545
23	529	12167	4·796	2·844	15·166	6·127	13·200	·04348
24	576	13824	4·899	2·884	15·492	6·214	13·389	·04167
25	625	15625	5·000	2·924	15·811	6·300	13·572	·0400
26	676	17576	5·099	2·962	16·125	6·383	13·751	·03846
27	729	19683	5·196	3·000	16·432	6·463	13·925	·03704
28	784	21952	5·292	3·037	16·733	6·542	14·095	·03571
29	841	24389	5·385	3·072	17·029	6·619	14·260	·03448
30	900	27000	5·477	3·107	17·321	6·694	14·422	·03333
31	961	29791	5·568	3·141	17·607	6·768	14·581	·03226
32	1024	32768	5·657	3·175	17·889	6·840	14·736	·03125
33	1089	35937	5·745	3·208	18·166	6·910	14·888	·03030
34	1156	39304	5·831	3·240	18·439	6·980	15·037	·02941
35	1225	42875	5·916	3·271	18·708	7·047	15·183	·02857
36	1296	46656	6·000	3·302	18·974	7·114	15·326	·02778
37	1369	50653	6·083	3·332	19·235	7·179	15·467	·02703
38	1444	54872	6·164	3·362	19·494	7·243	15·605	·02632
39	1521	59319	6·245	3·391	19·748	7·306	15·741	·02564
40	1600	64000	6·325	3·420	20·00	7·368	15·874	·0250
41	1681	68921	6·403	3·448	20·248	7·429	16·005	·02439
42	1764	74088	6·481	3·476	20·494	7·489	16·134	·02381
43	1849	79507	6·557	3·503	20·736	7·548	16·261	·02326
44	1936	85184	6·633	3·530	20·976	7·606	16·386	·02273
45	2025	91125	6·708	3·557	21·213	7·663	16·510	·02222
46	2116	97336	6·782	3·583	21·448	7·719	16·631	·02174
47	2209	103823	6·856	3·609	21·679	7·775	16·751	·02128
48	2304	110592	6·928	3·634	21·909	7·830	16·869	·02083
49	2401	117649	7·000	3·659	22·136	7·884	16·985	·02041
50	2500	125000	7·071	3·684	22·361	7·937	17·100	·020

POWERS, ROOTS AND RECIPROCALS

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\sqrt{10n}$	$\sqrt[3]{10n}$	$\sqrt[3]{100n}$	$\frac{1}{n}$
51	2601	132651	7·141	3·708	22·583	7·990	17·213	·01961
52	2704	140608	7·211	3·733	22·804	8·041	17·325	·01923
53	2809	148877	7·280	3·756	23·022	8·093	17·435	·01887
54	2916	157464	7·348	3·780	23·238	8·143	17·544	·01852
55	3025	166375	7·416	3·803	23·452	8·193	17·652	·01818
56	3136	175616	7·483	3·826	23·664	8·243	17·758	·01786
57	3249	185193	7·550	3·849	23·875	8·291	17·863	·01754
58	3364	195112	7·616	3·871	24·083	8·340	17·967	·01724
59	3481	205379	7·681	3·893	24·290	8·387	18·070	·01695
60	3600	216000	7·746	3·915	24·495	8·434	18·171	·01667
61	3721	226981	7·810	3·936	24·698	8·481	18·272	·01639
62	3844	238328	7·874	3·958	24·900	8·527	18·371	·01613
63	3969	250047	7·937	3·979	25·100	8·573	18·469	·01587
64	4096	262144	8·000	4·000	25·298	8·618	18·566	·01562
65	4225	274625	8·062	4·021	25·495	8·662	18·663	·01538
66	4356	287496	8·124	4·041	25·690	8·707	18·758	·01515
67	4489	300763	8·185	4·062	25·884	8·750	18·852	·01493
68	4624	314432	8·246	4·082	26·077	8·794	18·945	·01471
69	4761	328509	8·307	4·102	26·268	8·837	19·038	·01449
70	4900	343000	8·367	4·121	26·458	8·879	19·129	·01429
71	5041	357911	8·426	4·141	26·646	8·921	19·220	·01408
72	5184	373248	8·485	4·160	26·833	8·963	19·310	·01389
73	5329	389017	8·544	4·179	27·019	9·004	19·399	·01370
74	5476	405224	8·602	4·198	27·203	9·045	19·487	·01351
75	5625	421875	8·660	4·217	27·386	9·086	19·574	·01333
76	5776	438976	8·718	4·236	27·568	9·126	19·661	·01316
77	5929	456533	8·775	4·254	27·749	9·166	19·747	·01299
78	6084	474552	8·832	4·273	27·928	9·205	19·832	·01282
79	6241	493039	8·888	4·291	28·107	9·244	19·916	·01266
80	6400	512000	8·944	4·309	28·284	9·283	20·000	·01250
81	6561	531441	9·000	4·327	28·460	9·322	20·083	·01235
82	6724	551368	9·055	4·344	28·636	9·360	20·165	·01220
83	6889	571787	9·110	4·362	28·810	9·398	2·0247	·01205
84	7056	592704	9·165	4·380	28·983	9·435	20·328	·01190
85	7225	614125	9·220	4·397	29·155	9·473	20·408	·01176
86	7396	636056	9·274	4·414	29·326	9·510	20·488	·01163
87	7569	658503	9·327	4·431	29·496	9·546	20·567	·01149
88	7744	681472	9·381	4·448	29·665	9·583	20·646	·01136
89	7921	704969	9·434	4·465	29·833	9·619	20·724	·01124
90	8100	729000	9·487	4·481	30·000	9·655	20·801	·01111
91	8281	753571	9·539	4·498	30·166	9·691	20·878	·01099
92	8464	778688	9·592	4·514	30·332	9·726	20·954	·01087
93	8649	804357	9·644	4·531	30·496	9·761	21·029	·01075
94	8836	830584	9·695	4·547	30·659	9·796	21·105	·01064
95	9025	857375	9·747	4·563	30·822	9·830	21·179	·01053
96	9216	884736	9·798	4·579	30·984	9·865	21·253	·01042
97	9409	912673	9·849	4·595	31·145	9·899	21·327	·01031
98	9604	941192	9·899	4·610	31·305	9·933	21·400	·01020
99	9801	970299	9·950	4·626	31·464	9·967	21·472	·01010
100	10000	1000000	10·000	4·642	31·623	10·000	21·544	·0100

AREAS OF CIRCLES

Diameter.	0·0	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
0	0·0	0·0078	0·0314	0·0706	0·1256	0·1963	0·2827	0·3848	0·5027	0·6362
1	0·7854	0·9503	1·131	1·327	1·539	1·767	2·011	2·270	2·545	2·835
2	3·142	3·464	3·801	4·155	4·524	4·909	5·309	5·725	6·157	6·605
3	7·069	7·548	8·042	8·553	9·079	9·621	10·18	10·75	11·34	11·95
4	12·57	13·20	13·85	14·52	15·20	15·90	16·62	17·35	18·10	18·86
5	19·63	20·43	21·24	22·06	22·90	23·76	24·63	25·52	26·42	27·34
6	28·27	29·22	30·19	31·17	32·17	33·18	34·21	35·26	36·32	37·39
7	38·48	39·59	40·71	41·85	43·01	44·18	45·36	46·57	47·78	49·02
8	50·27	51·53	52·81	54·11	55·42	56·74	58·09	59·45	60·82	62·21
9	63·62	65·04	66·48	67·93	69·40	70·88	72·38	73·90	75·43	76·98
10	78·54	80·12	81·71	83·32	84·95	86·59	88·25	89·92	91·61	93·31
11	95·03	96·77	98·52	100·3	102·1	103·9	105·7	107·5	109·4	111·2
12	113·1	115·0	116·9	118·8	120·8	122·7	124·7	126·7	128·7	130·7
13	132·7	134·8	136·8	138·9	141·0	143·1	145·3	147·4	149·6	151·7
14	153·9	156·1	158·4	160·6	162·9	165·1	167·4	169·7	172·0	174·4
15	176·7	179·1	181·5	183·8	186·3	188·7	191·1	193·6	196·1	198·6
16	201·1	203·6	206·1	208·7	211·2	213·8	216·4	219·0	221·7	224·3
17	227·0	229·7	232·3	235·1	237·8	240·5	243·3	246·1	248·8	251·6
18	254·5	257·3	260·2	263·0	265·9	268·8	271·7	274·6	277·6	280·6
19	283·5	286·5	289·5	292·6	295·6	298·6	301·7	304·8	307·9	311·0
20	314·2	317·3	320·5	323·7	326·8	330·1	333·3	336·5	339·8	343·1
21	346·4	349·7	353·0	356·3	359·7	363·0	366·4	369·8	373·2	376·7
22	380·1	383·6	387·1	390·6	394·1	397·6	401·1	404·7	408·3	411·9
23	415·5	419·1	422·7	426·4	430·0	433·7	437·4	441·1	444·9	448·6
24	452·4	456·2	460·0	463·8	467·6	471·4	475·3	479·2	483·1	487·0
25	490·9	494·8	498·8	502·7	506·7	510·7	514·7	518·7	522·8	526·8
26	530·9	535·0	539·1	543·2	547·4	551·5	555·7	559·9	564·1	568·3
27	572·6	576·8	581·1	585·3	589·6	594·0	598·3	602·6	607·0	611·4
28	615·7	620·2	624·6	629·0	633·5	637·9	642·4	646·9	651·4	656·0
29	660·5	665·1	669·7	674·3	678·9	683·5	688·1	692·8	697·5	702·1
30	706·9	711·6	716·3	721·1	725·8	730·6	735·4	740·2	745·1	749·9
31	754·8	759·6	764·5	769·4	774·4	779·3	784·3	789·2	794·2	799·2
32	804·2	809·3	814·3	819·4	824·5	829·6	834·7	839·8	845·0	850·1
33	855·3	860·5	865·7	870·9	876·2	881·4	886·7	892·0	897·3	902·6
34	907·9	913·3	918·6	924·0	929·4	934·8	940·2	945·7	951·1	956·6
35	962·1	967·6	973·1	978·7	984·2	989·8	995·4	1000·9	1006·6	1012·2
36	1017·9	1023·5	1029·2	1034·9	1040·6	1046·3	1052·1	1057·8	1063·6	1069·4
37	1075·2	1081·0	1086·9	1092·7	1098·6	1104·5	1110·4	1116·3	1122·2	1128·1
38	1134·1	1140·1	1146·1	1152·1	1158·1	1164·2	1170·2	1176·3	1182·4	1188·5
39	1194·6	1200·7	1206·9	1213·0	1219·2	1225·4	1231·6	1237·9	1244·1	1250·4
40	1256·6	1262·9	1269·2	1275·6	1281·9	1288·2	1294·6	1310·0	1307·4	1313·8
41	1320·3	1326·7	1333·2	1339·6	1346·1	1352·6	1359·2	1365·7	1372·3	1378·8
42	1385·4	1392·0	1398·7	1405·3	1412·0	1418·6	1425·3	1432·0	1438·7	1445·4
43	1452·2	1459·0	1465·7	1472·5	1479·3	1486·2	1493·0	1499·9	1506·7	1513·6
44	1520·5	1527·4	1534·4	1541·3	1548·3	1555·3	1562·3	1569·3	1576·3	1583·4
45	1590·4	1597·5	1604·6	1611·7	1618·8	1626·0	1633·1	1640·3	1647·5	1654·7
46	1661·9	1669·1	1676·4	1683·6	1690·9	1698·2	1705·5	1712·9	1720·2	1727·6
47	1734·9	1742·3	1749·7	1757·2	1764·6	1772·0	1779·5	1787·0	1794·5	1802·0
48	1809·6	1817·1	1824·7	1832·2	1839·8	1847·4	1855·1	1862·7	1870·4	1878·0
49	1885·7	1893·4	1901·2	1908·9	1916·6	1924·4	1932·2	1940·0	1947·8	1955·6
50	1963·5	1971·4	1979·2	1987·1	1995·0	2003·0	2010·9	2018·9	2026·8	2034·8

NOTE: If diameter is divided by 10, Area is divided by 100.
If diameter is divided by 100, Area is divided by 10,000.
Example. Area of circle 0·875″ dia. is 0·60132 sq in.

AREAS OF CIRCLES

Diameter.	0·0	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
51	2042·8	2050·8	2058·9	2066·9	2075·0	2083·1	2091·2	2099·3	2107·4	2115·6
52	2123·7	2131·9	2140·1	2148·3	2156·5	2164·7	2173·0	2181·3	2189·6	2197·9
53	2206·2	2214·5	2222·9	2231·2	2239·6	2248·0	2256·4	2264·8	2273·3	2281·7
54	2290·2	2298·7	2307·2	2315·7	2324·3	2332·8	2341·4	2350·0	2358·6	2367·2
55	2375·8	2384·5	2393·1	2401·8	2410·5	2419·2	2428·0	2436·7	2445·4	2454·2
56	2463·0	2471·8	2480·6	2489·5	2498·3	2507·2	2516·1	2525·0	2533·9	2542·8
57	2551·8	2560·7	2569·7	2578·7	2587·7	2596·7	2605·8	2614·8	2623·9	2633·0
58	2642·1	2651·2	2660·3	2669·5	2678·6	2687·8	2697·0	2706·2	2715·5	2724·7
59	2734·0	2743·2	2752·5	2761·8	2771·2	2780·5	2789·9	2799·2	2808·6	2818·0
60	2827·4	2836·9	2846·3	2855·8	2865·3	2874·7	2884·3	2893·8	2903·3	2912·9
61	2922·5	2932·1	2941·7	2951·3	2960·9	2970·6	2980·2	2989·9	2999·6	3009·3
62	3019·1	3028·8	3038·6	3048·4	3058·1	3068·0	3077·8	3087·6	3097·5	3107·4
63	3117·2	3127·1	3137·1	3147·0	3157·0	3166·9	3176·9	3186·9	3196·9	3206·9
64	3217·0	3227·1	3237·1	3247·2	3257·3	3267·4	3277·6	3287·7	3297·9	3308·1
65	3318·3	3328·5	3338·8	3349·0	3359·3	3369·6	3379·8	3390·2	3400·5	3410·8
66	3421·2	3431·6	3442·0	3452·4	3462·8	3473·2	3483·7	3494·1	3504·6	3515·1
67	3525·7	3536·2	3546·7	3557·3	3567·9	3578·5	3589·1	3599·7	3610·3	3621·0
68	3631·7	3642·4	3653·1	3663·8	3674·5	3685·3	3696·1	3706·8	3717·6	3728·4
69	3739·3	3750·1	3761·0	3771·9	3782·8	3793·7	3804·6	3815·5	3826·5	3837·5
70	3848·5	3859·4	3870·5	3881·5	3892·6	3903·6	3914·7	3925·8	3936·9	3948·0
71	3959·2	3970·3	3981·5	3992·7	4003·9	4015·1	4026·4	4037·6	4048·9	4060·2
72	4071·5	4082·8	4094·2	4105·5	4116·9	4128·2	4139·6	4151·1	4162·5	4173·9
73	4185·4	4196·9	4208·3	4219·9	4231·4	4242·9	4254·5	4266·0	4277·6	4289·2
74	4300·8	4312·5	4324·1	4335·8	4347·5	4359·2	4370·9	4382·6	4394·3	4406·1
75	4417·9	4429·6	4441·5	4453·3	4465·1	4477·0	4488·8	4500·7	4512·6	4524·5
76	4536·5	4548·4	4560·4	4572·3	4584·3	4596·3	4608·4	4620·4	4632·5	4644·5
77	4656·6	4668·7	4680·8	4693·0	4705·1	4717·3	4729·5	4741·7	4753·9	4766·1
78	4778·4	4790·6	4802·9	4815·2	4827·5	4839·8	4852·2	4864·5	4876·9	4889·3
79	4901·7	4914·1	4926·5	4939·0	4951·4	4963·9	4976·4	4988·9	5001·4	5014·0
80	5026·6	5039·1	5051·7	5064·3	5076·9	5089·6	5102·2	5114·9	5127·6	5140·3
81	5153·0	5165·7	5178·5	5191·2	5204·0	5216·8	5229·6	5242·4	5255·3	5268·1
82	5281·0	5293·9	5306·8	5319·7	5332·7	5345·6	5358·6	5371·6	5384·6	5397·6
83	5410·6	5423·6	5436·7	5449·8	5462·9	5476·0	5489·1	5502·3	5515·4	5528·6
84	5541·8	5555·0	5568·2	5581·4	5594·7	5607·9	5621·2	5634·5	5647·8	5661·2
85	5674·5	5687·9	5701·2	5714·6	5728·0	5741·5	5754·9	5768·3	5781·8	5795·3
86	5808·8	5822·3	5835·8	5849·4	5863·0	5876·6	5890·1	5903·7	5917·4	5931·0
87	5944·7	5958·3	5972·0	5985·7	5999·5	6013·2	6027·0	6040·7	6054·5	6068·3
88	6082·1	6096·0	6109·8	6123·7	6137·5	6151·4	6165·3	6179·3	6193·2	6207·2
89	6221·1	6235·1	6249·1	6263·1	6277·2	6291·2	6305·3	6319·4	6333·5	6347·6
90	6361·7	6375·9	6390·0	6404·2	6418·4	6432·6	6446·8	6461·1	6475·3	6489·6
91	6503·9	6518·2	6532·5	6546·8	6561·2	6575·6	6589·9	6604·3	6618·7	6633·2
92	6647·6	6662·1	6676·5	6691·0	6705·5	6720·1	6774·6	6749·1	6763·7	6778·3
93	6792·9	6807·5	6822·2	6836·8	6851·5	6866·1	6880·8	6895·6	6910·3	6925·0
94	6939·8	6954·6	6969·3	6984·1	6999·0	7013·8	7028·7	7043·5	7058·4	7073·3
95	7088·2	7103·1	7118·1	7133·1	7148·0	7163·0	7178·0	7193·1	7208·1	7223·2
96	7238·2	7253·3	7268·4	7283·5	7298·7	7313·8	7329·0	7344·2	7359·4	7374·6
97	7389·8	7405·1	7420·3	7435·6	7450·9	7466·2	7481·5	7496·9	7512·2	7527·6
98	7543·0	7558·4	7573·8	7589·2	7604·7	7620·1	7635·6	7651·1	7666·6	7682·1
99	7697·7	7713·2	7728·8	7744·4	7760·0	7775·6	7791·3	7806·9	7822·6	7838·3
100	7854·0	7869·7	7885·4	7901·2	7916·9	7932·7	7948·5	7964·3	7980·1	7996·0

NOTE: If diameter is divided by 10, Area is divided by 100.
If diameter is divided by 100, Area is divided by 10,000.
Example. Area of circle 0·875″ dia. is 0·60132 sq in.

PREFERRED NUMBERS

Traditionally, the sizes of materials or tools rose by uniform increments e.g. drills from $\frac{1}{16}$" to $\frac{1}{2}$" × $\frac{1}{16}$", or electrical resistors from 100 to 1000 × 100. This had serious disadvantages. For example, the next size up from $\frac{1}{8}$" drill was 50% larger, but from $\frac{7}{16}$" to $\frac{1}{2}$" only 14%. Further, many sizes were seldom required. Similar conditions applied to bar-stock; most needed machining to size, and it was common to find that the next size up had either inadequate or excessive allowance.

Initiated in the electronics industry for component values (resistors and capacitors) a succession of number series was devised in which sizes rose by *geometric* (or logarithmic) intervals. These intervals were chosen to suit the tolerance on the components, so that between repetitive figures (e.g. 100 and 1000) the minimum number of sizes was required. For example:

The R3 series ran	100–220–470–1000;	increments of ×2·2
R6	100–150–220–330–470–etc.	increments of ×1·5
R12	100–120–150–180–220–etc.	increments of ×1·2

It will be seen that all numbers in a lower series appear in those higher, and that the 'number of numbers' between repetitions corresponds to the R number. The above series, based on three, go up to R96. The numbers are, of course, rounded to avoid anomalies.

In mechanical engineering the series are normally decimal based – R10, R20, R40, etc. The sequence is shown in the table below. R10 may be regarded as *first preference*, R20 as second, and R40 as third, depending on the application e.g. the R20 number "36" is derived from "3548". It may be found as 0·35, 3·55, 360 in different stock dimensions.

R10	R20	R40	·020	·040	·080	·160	·315	·63	1·25	2·5	5·0	10·0	20·0
		R40	·021	·042	·085	·170	·335	·67	1·32	2·65	5·3	10·6	21·2
	R20	R40	·022	·045	·090	·180	·355	·71	1·40	2·8	5·6	11·2	22·4
		R40	·024	·048	·095	·190	·375	·75	1·50	3·0	6·0	11·8	23·6
R10	R20	R40	·025	·050	·100	·200	·40	·80	1·60	3·15	6·3	12·5	25·0
		R40	·026	·053	·106	·212	·425	·85	1·70	3·35	6·7	13·2	
	R20	R40	·028	·056	·112	·224	·45	·90	1·80	3·55	7·1	14·0	
		R40	·030	·060	·118	·236	·475	·95	1·90	3·75	7·5	15·0	
R10	R20	R40	·032	·063	·125	·250	·50	1·00	2·00	4·0	8·0	16·0	
		R40	·034	·067	·132	·265	·53	1·06	2·12	4·25	8·5	17·0	
	R20	R40	·036	·071	·140	·28	·56	1·12	2·24	4·5	9·0	18·0	
		R40	·038	·075	·150	·30	·60	1·18	2·36	4·75	9·5	19·0	

Note that the 'succession' of numbers run *down* the columns e.g. R10 runs: 0·02, 0·025, 0·032, 0·040, 0·050 etc.

Although sets of drills etc. are widely marketed in the former uniform increments (e.g. 1 mm–6 mm × 0·1 mm) readers who buy drills one at a time, as needed, might care to consider spacing them according to these series! Suitably rounded, of course, to match those available.

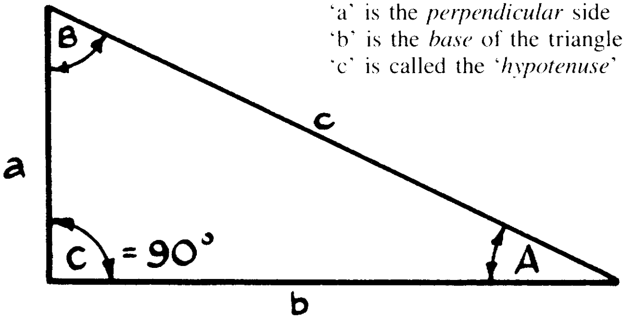
The industrial first preference is R10, but this is too coarse for model engineering. It is suggested that for scaled or stressed components R20 as first and R40 as second preference is more appropriate. This means using (e.g.) 6·3 mm or 6·7 mm instead of 6½ mm on models. When scaling *metric* prototypes scales of 1/5, 1/10, 1/20 etc. will, almost automatically, result in the retention of the R series figures.

TRIGONOMETRY

This is the art or science which enables angles to be used to measure distances and vice versa, and is based on the properties of triangles. It is invaluable to model engineers in marking-out and in many machining operations.

The basic trig. ratios are described on page 2.12 and the subsequent tables give values.

The basic triangle is repeated in the figure below.



'a' is the *perpendicular* side
'b' is the *base* of the triangle
'c' is called the '*hypotenuse*'

(a) Using the basic definitions, if one angle and one side of any right-angled triangle is known, we can find any other side or angle. Thus:

As C = 90°, A = (90 − B) and B = (90 − A).
a = b Tan A = c Sin A.
c = a/Sin A = b/Cos A.
b = c Cos A = a/Tan A.

Example:
Find the setover for the topslide to turn a taper on the diameter of one inch to the foot.

The setover will be half the included angle on the diameter, or $\frac{1}{2}$ inch per foot = 1/24 = 0·0417. This is the ratio a/b in the diagram = the Tangent of the angle.

Look up the table of tangents, to find that this angle is 2°23′, to which the topslide should be set. If set at 2·4°, using a 'top-slide vernier' (*Model Engineer* Vol 140, 1974, p. 803) the error would be less than 0·00025″/inch.

(b) For triangles other than right-angled, the following relationship applies:

a/SinA = b/Sin B = c/Sin C; and A + B + C = 180°

Hence, if two sides and one angle are known:

$$\text{Angle A} = 180 - (B + C); \quad b = \frac{a \text{ Sin B}}{\text{Sin A}}; \quad c = \frac{a \text{ Sin C}}{\text{Sin A}};$$

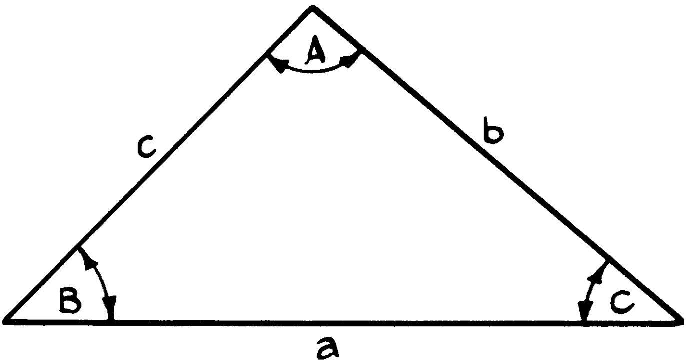
This can be applied whichever angles or sides are known.

(c) If the lengths of the three sides are known, but no angles, then another relationship can be used. This is:

$$\text{Cos A} = \frac{b^2 + c^2 - a^2}{2bc}$$

$$\text{Cos B} = \frac{a^2 + c^2 - b^2}{2ac}$$

Angle C is then found from $C = 180° - (A + B)$



(d) If two sides and the included angle are known, (say a, b and C).
Then:

$$\tfrac{1}{2}(A + B) = 90° - \tfrac{1}{2}C$$

$$\text{and } \tfrac{1}{2}(A - B) = \frac{a - b}{a + b} \times \frac{1}{\text{Tan } \frac{1}{2}C}$$

Add $\frac{1}{2}(A + B)$ and $\frac{1}{2}(A - B)$ to get A.

Subtract $\frac{1}{2}(A - B)$ from $\frac{1}{2}(A + B)$ to get B.

To find the side c, use the procedure of (a) or (b).

(e) If s is the 'semi-perimeter' [i.e. $s = \frac{1}{2}(a + b + c)$] the following relationships apply:
Area of the triangle $= \sqrt{s(s - a)(s - b)(s - c)}$
Radius of the inscribed circle, just touching all three sides is given by:

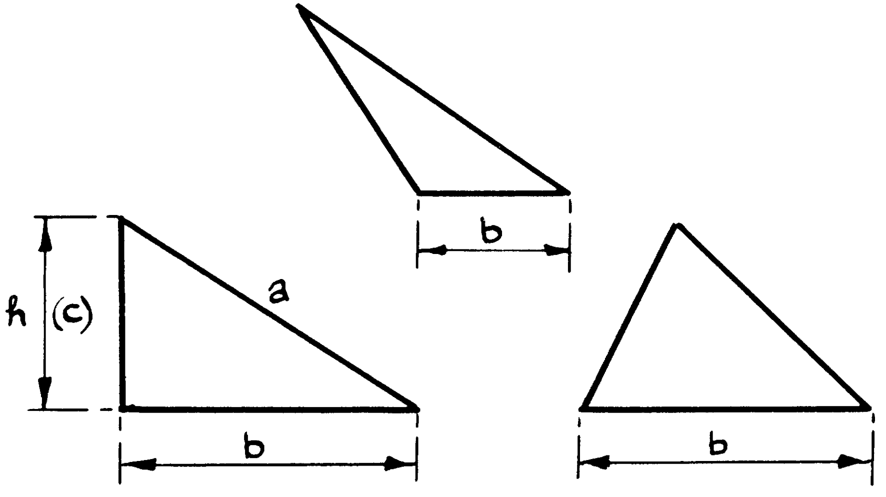
$$r = (s - a)\text{Tan}\tfrac{1}{2}A$$

and $r = \text{Area of Triangle}/s$

(f) A few useful figures to remember.

Sin 30 = Cos 60 = 0·5000 $\qquad$ Tan 30 $= \frac{1}{\sqrt{3}}$
Sin 60 = Cos 30 $= \frac{1}{2}\sqrt{3}$ $\qquad$ Tan 45 = 1.
Sin 45 = Cos 45 $= \frac{1}{2}\sqrt{2}$ $\qquad$ Tan 60 $= \sqrt{3}$

(g) Triangle.

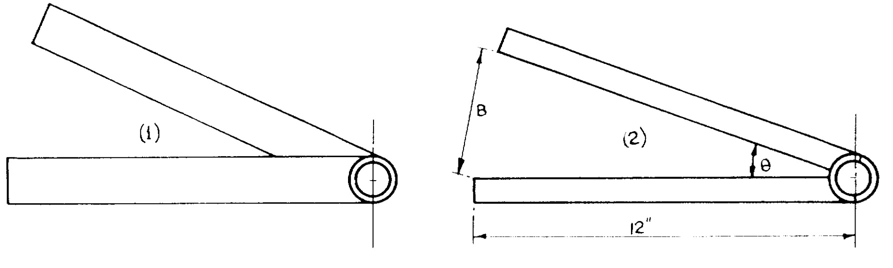


The area of any triangle is given by $A = \frac{1}{2}b \times h$.

For a right-angled triangle only (left-hand fig. above) $a^2 = b^2 + h^2$

(h) To measure an angle using a folding 2-ft rule.
θ = angle of opening, in degrees.
B = opening at rule end in inches.
Then $\text{Sin}\frac{1}{2}\theta = B/24$.

If the rule is other than 24″ long (opened) substitute the opened length for the figure 24 above.



This equation holds only for rules which butt when folded as at (2).
It does NOT apply to rules which overlap when folded as in (1).

VOLUME OF SPHERES AND BALLS

$$V = \pi \frac{D^3}{6} \qquad A = \pi D^2$$

Diameter	*Volume*	*Diameter*	*Volume*	*Diameter*	*Volume*
$\frac{1}{32}$	·000016	$\frac{23}{32}$	·19442	4	33·510
$\frac{1}{16}$	·000128	$\frac{3}{4}$	·22089	5	65·450
$\frac{3}{32}$	·000431	$\frac{25}{32}$	·24967	6	113·10
$\frac{1}{8}$	·001022	$\frac{13}{16}$	·28084	7	179·59
$\frac{5}{32}$	·002016	$\frac{27}{32}$	·31451	8	268·08
$\frac{3}{16}$	·00345	$\frac{7}{8}$	·35077	9	381·70
$\frac{7}{32}$	·00548	$\frac{29}{32}$	·38971	10	523·60
$\frac{1}{4}$	·00818	$\frac{15}{16}$	·43142	11	696·91
$\frac{9}{32}$	·01165	$\frac{31}{32}$	·47603	12	904·78
$\frac{5}{16}$	·01598	1	·52360	13	1150·3
$\frac{11}{32}$	·02127	$1\frac{1}{8}$	·74551	14	1436·8
$\frac{3}{8}$	·02761	$1\frac{1}{4}$	1·0227	15	1767·2
$\frac{13}{32}$	·03511	$1\frac{3}{8}$	1·3611	16	2144·7
$\frac{7}{16}$	·04385	$1\frac{1}{2}$	1·7671	17	2572·4
$\frac{15}{32}$	·05393	$1\frac{5}{8}$	2·2468	18	3053·6
$\frac{1}{2}$	·06545	$1\frac{3}{4}$	2·8062	19	3591·4
$\frac{17}{32}$	·07850	$1\frac{7}{8}$	3·4514	20	4188·8
$\frac{9}{16}$	·09319	2	4·1888	21	4851·0
$\frac{19}{32}$	·10960	$2\frac{1}{4}$	5·9641	22	5577·5
$\frac{5}{8}$	·12783	$2\frac{1}{2}$	8·1813	23	6373·2
$\frac{21}{32}$	·14798	$2\frac{3}{4}$	10·889	24	7241·1
$\frac{11}{16}$	·17014	3	14·137	25	8181·3

For decimal diameters. If D is divided by 10 the volume is divided by 1000.
If D is multiplied by ten the volume is multiplied by 1000.
Examples Volume of 25 dia = 8181·3
Volume of 2·5 dia = 8·1813
Volume of 30 dia = 14137.

Sizes not given in the table can be calculated using the table of cubes of numbers on pages **2**.22 and **2**.23.

To find the **area of the surface** of a sphere, divide the above figures by $\frac{D}{6}$

USEFUL CONSTANTS

Pi

$\pi = 22/7$ or $3 \cdot 142$ or $3 \cdot 14159$ or $355/113$ in ascending order of accuracy. $\frac{1}{\pi} = 0 \cdot 3183$. $\pi^2 = 9 \cdot 870$. Log $\pi = 0 \cdot 4971$.

Gravity

'Standard' acceleration due to gravity, 'g' = 32·174 ft/sec/sec = 9·8066 m/sec/sec. For practical purposes 32·2 ft/sec^2 or 9·81 m/sec^2 may be used with little error.

$\sqrt{2g} = 8 \cdot 025$ ft units, or 4·429 metre units.

Exponential constant

Naperian constant e = 2·71828. Log e to base 10 = 0·43429.

Radian

One radian = $360/2\pi$degrees = 57·296°. 1 degree = 0·01745 radians. See p. **2**.20.

Air

1 standard atmosphere = 14·696 psi = 29·92 in. mercury = 1·013 Bar
= 0·1013 newton/sq. mm.
1 lb of air occupies 13·123 cu. ft at 60°F and 29·92″hg (1 atmos)
1 kg of air occupies 773·51 litre at 0°C and 760 mm mercury.

Composition of dry air

	By Volume	*By Weight*
Oxygen	21%	23·2%
Nitrogen	78·9%*	76·7%*
CO_2	0·1%	0·1%

*'Nitrogen' includes the very small quantities of rare gases. CO_2 can vary with location between 0·03% and 0·3% by volume.

Coefficient of expansion of air at constant pressure
1/273·1 times the volume at 0°C for each degree C temperature rise.
1/491·6 times the volume at 32°F for each degree F temperature rise.
(This coefficient of expansion applies to all true gases, but **not** steam.)

Mean specific heat of dry air at temperatures between atmos and 300°C
At constant pressure – Cp = 0·0194 BTU/cu. ft/°F = 1301·1 J (m^3/°C).
By weight = 0·228 BTU/lb/°F or 954·59 J (kg/°C).

Barometric correction for altitude
The pressure is reduced by approximately 0·11 inch mercury for each 100 ft above sea level = 3·73 millibar. This effect falls with altitude, being about 0·1 inch/100 ft at 3000 ft and 0·09″ at 6000 ft.

Water

1 gallon weighs 10 lbf. 1 cu. ft weighs 62·42 lbf. 1 cu. in. weighs 0·036 lbf.
1 litre contains a mass of 1 kg.

Specific heat of water at 15°C = 1·000 $\left.\right\}$ kg.cal/kg.°C or BTU/16°F
at 100°C = 1·0063 $\left.\right\}$ (× 4186·8 for J/kg°C)

One foot head of water exerts a pressure of 0·432 lbf/sq. in.
One metre head of water exerts a pressure of 0·0978 bar = 0·0098 N/sq. mm.
A cylinder 12″ dia × 12″ long holds 4·896 galls.
One inch of rain deposits approximately 100 tons of water to the acre.

Velocities

100 ft/min = 0·508 metres/sec.
30 mph = 48·3 km/hr = 13·4 m/sec.
100 mph = 161 km/hr = 44·7 m/sec.
Sonic velocity in air at sea level (Mach 1) = 762 mph = 341 m/sec.
Velocity of light = 186 000 miles/sec = 299 000 km/sec.
'Mach No' is the ratio actual speed/speed of sound; e.g. at sea level 'Mach 2' = 2 × 762 = 1524 mph.

USEFUL FORMULAE

Quadratic equations

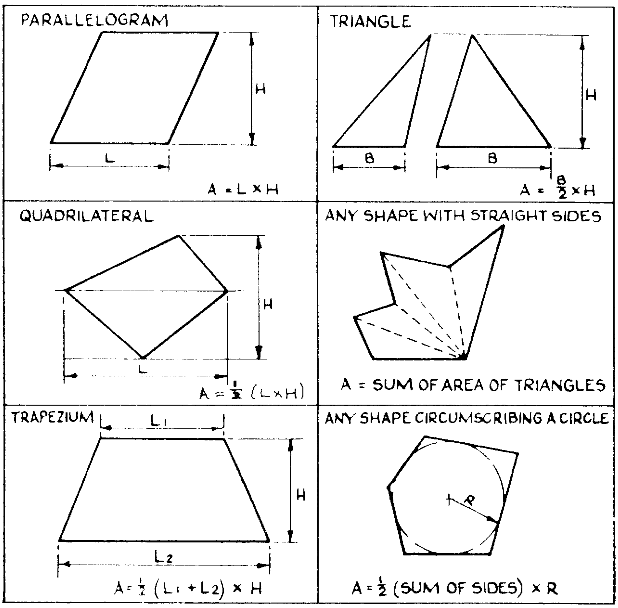
If $ax^2 + bx + c = 0$

Then $x = \dfrac{-b \pm \sqrt{b^2 - 4ac}}{2a}$

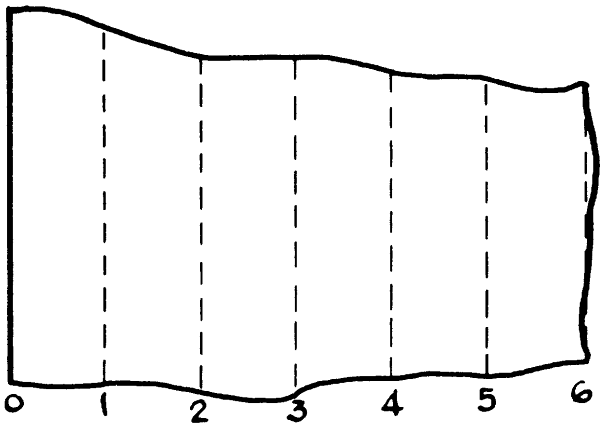
This gives two solutions for x. The illogical or absurd value is rejected, e.g. if x is the diameter of a circle and one value is negative.

Areas and volumes of various shapes

(a) Areas of figures.



(b) Area of irregular shape ('Simpson's Rule').



Divide the figure into an *even* number of equal slices, as shown in the figure. There will then be an *odd* number of ordinates 0, 1, 2, 3, etc. Measure the length of each ordinate, including the two end ones. (0 and 6 in the figure.)

Then:

Add the length of the first ordinate (0) to the last = A

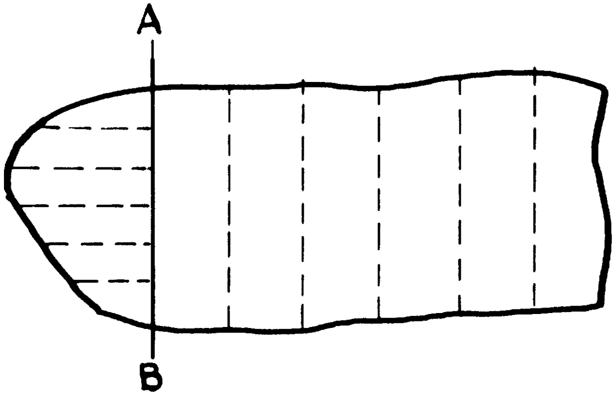
Add together the length of all *odd* ordinates and multiply by four, i.e. $(1 + 3 + 5) \times 4 = B$

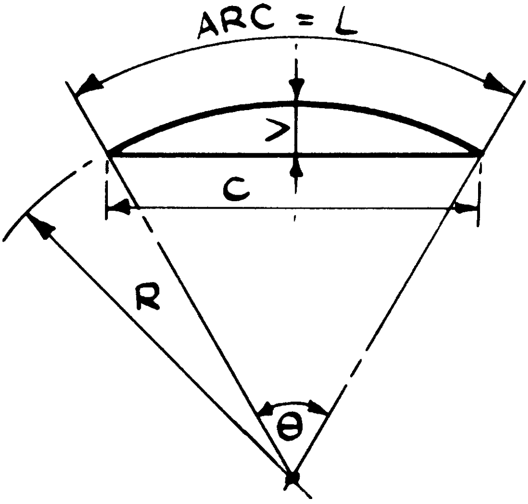
Add together *even* ordinates except the last one and multiply by two = C

Measure the width of the slices = W

The area is given by Area $= \frac{1}{3} \times W \times (A + B + C)$.

Clearly the narrower the slices, the more accurate. But if the figure has a 'bulge' where the slices are nearly parallel to the curve, accuracy will be improved if it is divided into parts as shown in the sketch below, and the area of that on the left of AB is treated separately from the right, with slices at right-angles.





(c) Circle.

Area $= (\frac{\pi}{4})\ D^2 = 0{\cdot}7854D^2$. Diameter $D = 1{\cdot}128\,A^{\frac{1}{2}}$

Circumference $= \pi D = 3{\cdot}1416D$. $D = 0{\cdot}3183 \times$ Circumference

Arc $L = \theta \times 0{\cdot}0175R$ (θ in degrees)

$V = \text{Versin} = (R - \frac{1}{2}\sqrt{4R^2 - C^2})$

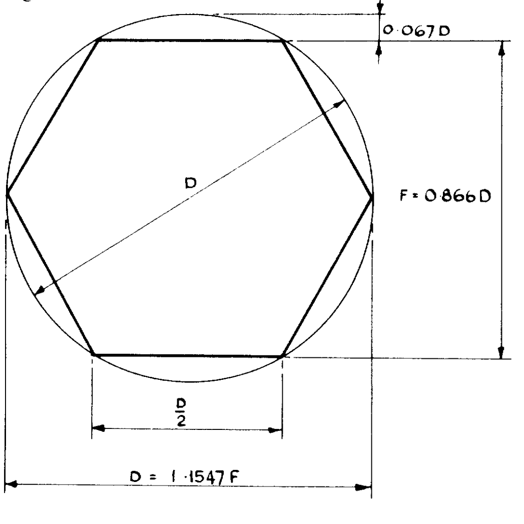
$$D = \text{Dia} = \frac{V^2 + \frac{1}{4}C^2}{V}$$

$C = \text{Chord} = \sqrt{V(2R - V)}$

$A = \text{Area} \ = \frac{1}{2}\,[RL - C(R - V)]$

$D = 2R$

(d) Circle and hexagon.



The **across flats** dimension of the largest hexagon that can be contained in a circle of diameter D is 0·866 D, i.e. if a 1″ dia round bar is milled to an exact hexagon it will be 0·866″ A–F.

The **across corners** of a true hexagon is 1·1547 × across flats.

The **side** of a hexagon is half the across corners dimension, and equal to the radius of the diameter D above.

The **area** of a hexagon = $0{\cdot}866F^2$, where F is the across flats dimension.

Squares and hexagons from round bar

To machine a true hexagon from a round bar of diameter D, machine off 0·067 times D to form each face. To form a square, machine off 0·147 × D.

(e) Circle spacing table.

If it is necessary to set out a number (N) of circles of diameter 'd' which just touch each other round a PCD of 'D', the table below may be used, where c = d/D.

N	*c*	*N*	*c*	*N*	*c*	*N*	*c*
3	0·8660	15	0·2079	27	0·11609	39	0·08046
4	0·7071	16	0·1951	28	0·11197	40	0·07846
5	0·5878	17	0·1837	29	0·10812	41	0·07654
6	0·5000	18	0·1736	30	0·10453	42	0·07473
7	0·4339	19	0·1646	31	0·10117	43	0·07299
8	0·3827	20	0·1564	32	0·09801	44	0·07133
9	0·3420	21	0·1490	33	0·09505	45	0·06975
10	0·3090	22	0·1423	34	0·09226	46	0·06824
11	0·2817	23	0·1362	35	0·08964	47	0·06679
12	0·2588	24	0·1305	36	0·08715	48	0·06540
13	0·2393	25	0·1253	37	0·08480	49	0·06407
14	0·2225	26	0·1205	38	0·08258	50	0·06275
60	0·0523	120	0·0262	180	0·01745	360	0·00873

The general rule is that c = Sin (180/N).

(f) Solids. *Sphere*, dia D.

Area = πD^2 Volume = $\pi D^3/6$.

Cone, height H, dia of base D.

Area = $\frac{1}{2}\sqrt{(D^2/4 + H^2)}$ (Conical surface only)

Volume = $\pi D^2 H/12$.

Pyramid, Height H, area of base A.

Volume = $\frac{1}{3}AH$.

Distance, velocity, acceleration

S = distance travelled. T = time taken. U = initial velocity. V = final velocity.
F = Acceleration. (All to be in consistent units.)

Then: $S = UT + \frac{1}{2}FT^2$

$V = U + FT$

$V^2 = U^2 + 2FS$

True velocity of an engine piston at any point in the stroke.
R = crank radius, feet or metres
L = length of connecting rod, feet or metres.
θ = crank angle from inner dead centre, degrees.
ω = angular speed of crank in radians/sec ($= 2\pi\,.\,\text{rpm}/60$).

Then:

$$V = \omega R(\text{Sin.}\,\theta + \tfrac{1}{2}\frac{R}{L}\,\text{Sin}\,2\,\theta)\ \text{ft/sec or M/sec.}$$

Acceleration of the piston at the same point is given by:

$$F = \omega^2 R(\text{Cos.}\,\theta + \frac{R}{L}\,\text{Cos}\,2\,\theta)\ \text{ft/sec}^2\ \text{or M/sec}^2.$$

Example: Find the maximum force on the gudgeon pin of an IC engine due to piston inertia if the piston weighs $1\frac{1}{2}$ ounces (43 g), the engine makes 2400 rpm, the stroke being 0·72″ (18·25 mm) and the connecting rod/crank ratio = 4.

(a) *In ft.lb.sec. units*

$1\frac{1}{2}$ oz = 0·094 lb. Crank radius = $1/12 \times \dfrac{0\cdot72}{2}$ = 0·03 ft. 2400 rpm = 251·3 rads/sec.

Force is '*mass* × acceleration' and is maximum at top dead centre, when $\theta = 0$ and Cos. $\theta = 1$; 2θ also = 0, so Cos. $2\theta = 1$.

From above, P = W/g × (Piston acceleration)
= $0\cdot094/32\cdot2 \times 251\cdot3^2 \times 0\cdot03(1 + 1/4)$
= 6·91 lbf.

(b) *In SI Units*

R must be in metres (0·0091), and the *mass* in kg (0·043).

The values of ω^2, $\dfrac{R}{L}$ Cos. θ and Cos. 2θ remain unchanged.

Hence P = M × piston acceleration
= $0\cdot043 \times 251\cdot3^2 \times 0\cdot0091 \times 1\cdot25$
= 30.9 newton

Force and acceleration

If a body of mass M kg is accelerated at F metres/sec^2, the force needed to produce that acceleration is:

P = M × F.newtons. (M = $\dfrac{W}{g}$ if W is the 'weight' in lb, when P is in lbf.)

Centrifugal force

A body moving in a circle experiences a constant acceleration towards the centre of the circle. Hence a force must be exerted in (e.g.) the spokes of the wheel, or whatever restraining member may be used.

This force is given by:
$P = M.\omega^2 R$, working in kg, R in metres and P in newtons.
$P = (W/g)\,\omega^2 R$, working in pounds weight, feet and P in lbf.
ω = angular velocity rads/sec = 2π.rpm/60.
R = radius, in feet or metres as appropriate.

SECTION THREE
STANDARD TAPERS AND COLLETS

TAPERS AND ANGLES

Table shows taper on diameter and included angles

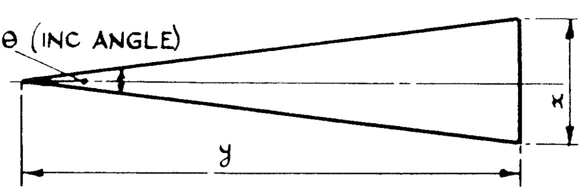
Inch/ft	*Degrees & minutes*	*Inch/in.*	*Taper*	*Degrees & minutes*	*Degrees*	*Inch/in.*	*1 in. –*
$\frac{1}{16}$	0–18	·0052	1 in 2	28–05	1	·01746	57·295
$\frac{1}{8}$	0–36	·01042	3	18–55	2	·03492	28·645
$\frac{3}{16}$	0–54	·015625	4	14–15	3	·05238	19·094
$\frac{1}{4}$	1–12	·02083	5	11–26	4	·06984	14·318
$\frac{5}{16}$	1–30	·02604	6	9–32	5	·08732	11·432
$\frac{3}{8}$	1–47	·03125	7	8–10	6	·10482	9·541
$\frac{7}{16}$	2–05	·03646	8	7–09	7	·12232	8·1749
$\frac{1}{2}$	2–23	·041667	9	6–21	8	·13986	7·1503
$\frac{5}{8}$	2–59	·052084	10	5–44	9	·15740	6·3531
$\frac{3}{4}$	3–35	·06250	11	5–12	10	·17498	5·7150
$\frac{7}{8}$	4–11	·072917	12	4–46	12	·2102	4·7572
1	4–46	·08333	13	4–24	14	·24556	4·0722
$1\frac{1}{8}$	5–22	·09375	14	4–06	15	·26330	3·79787
$1\frac{1}{4}$	5–58	·10416	15	3–49	16	·28108	3·5577
$1\frac{3}{8}$	6–33	·11460	16	3–35	18	·31676	3·1569
$1\frac{1}{2}$	7–09	·1250	17	3–22	20	·35266	2·8356
$1\frac{5}{8}$	7–43	·1354	18	3–10	25	·44338	2·2554
$1\frac{3}{4}$	8–20	·145833	19	3–00	30	·53590	1·866
$1\frac{7}{8}$	8–56	·15625	20	2–52	40	·72794	1·37374
2	9–32	·16667	24	2–23	45	·82842	1·20710
			30	1–54			

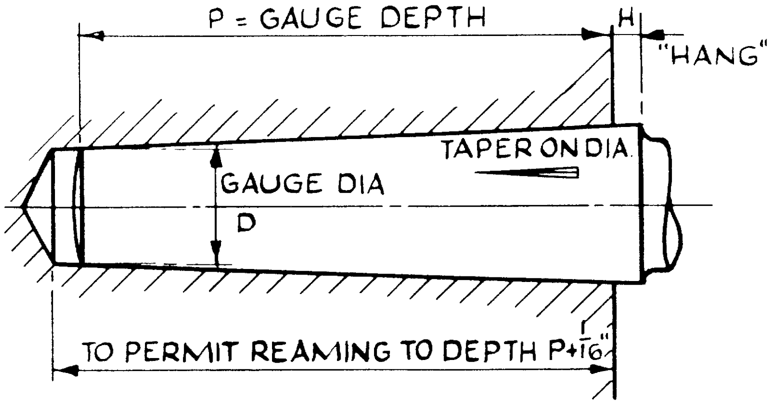
The British Standard Shaft Key taper is $\frac{1}{100}$ = very nearly $\frac{1}{8}''$ per foot.

Note:
Taper/inch = TAN $\frac{1}{2}\theta$

θ = inc. angle

Example
$\frac{1}{4}$ inch/foot
$x = \frac{1}{4}''$, $y = 12''$, $\theta = 1°–12'$





TAPER SHANKS
Morse and Brown & Sharp

MORSE TAPER				
No. of taper (in.)	*Plug dia. D (in.)*	*Depth P (in.)*	*Hang H (in.)*	*Taper per inch on dia. (in.)*
0	·252	2	$\frac{1}{8}$	·05205
1	·369	$2\frac{1}{8}$	$\frac{1}{8}$	·04988
2	·572	$2\frac{9}{16}$	$\frac{3}{16}$	·04995
3	·778	$3\frac{3}{16}$	$\frac{3}{16}$	·05020
4	1·020	$4\frac{1}{16}$	$\frac{1}{4}$	·05194
5	1·475	$5\frac{3}{16}$	$\frac{1}{4}$	·05263
6	2·116	$7\frac{1}{4}$	$\frac{5}{16}$	·05214
7	2·750	10	$\frac{3}{8}$	·05200

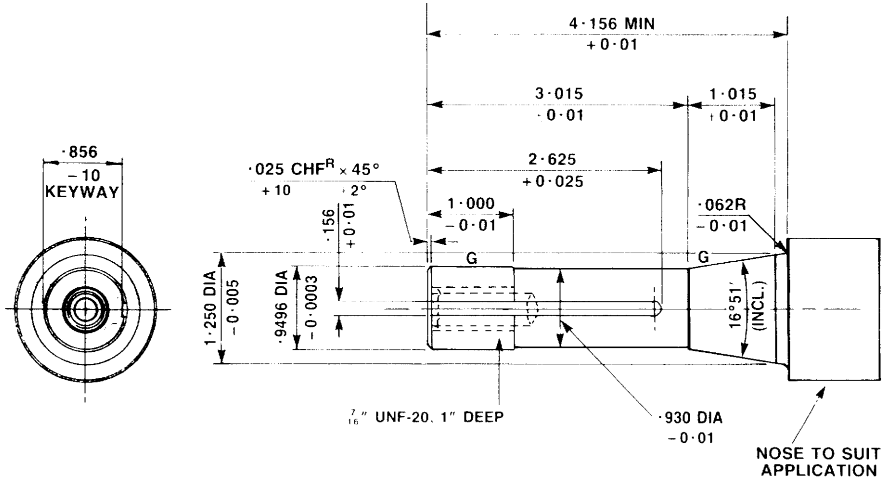
BROWN & SHARP ENDMILL TAPERS				
No. of taper (in.)	*Dia. D (in.)*	*Depth P (in.)*	*Hang H (in.)*	*Taper inch/foot on dia.*
4*	·350	$1\frac{1}{4}$	$\frac{3}{32}$	0·500
5*	·450	$1\frac{3}{4}$	$\frac{3}{32}$	0·500
7*	·600	3	$\frac{3}{32}$	0·500
9*	·900	4	$\frac{1}{8}$	0·500
10*	1·0446	5	$\frac{1}{8}$	0·5161
11*	1·250	$5\frac{15}{16}$	$\frac{1}{8}$	0·500
12*	1·500	$7\frac{1}{8}$	$\frac{1}{8}$	0·500

***The table includes only those tapers which are recognised British Standards for Endmill shanks etc. See notes opposite.

Notes

(a) The dimensions given are those for the Standard PLUG-GAUGE. The actual taper shank can be made to a length to suit the application, as can the socket. The diameter at the large end will be D + (Depth or length) × (taper per inch).
(b) Allowances in holes must be made to accept any tang or shank.
(c) No. 0 Morse taper centres on small lathes may be as short as $\frac{3}{4}''$; nevertheless diameter D applies at the small end of the centre.

DIMENSIONS OF BRIDGEPORT R8 COLLET

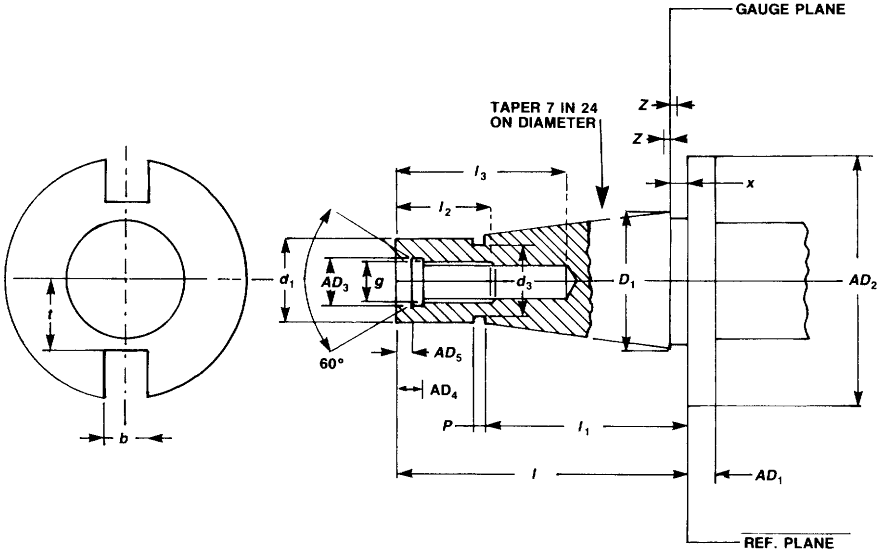


ALL DIMENSIONS IN INCHES

DIMENSIONS OF 'INTERNATIONAL' TAPER SHANKS

Most industrial milling and similar machines use these tapers on the mandrel nose. They are not *driving* tapers, but are designed to be self-releasing; the toolholder is driven from lugs on the mandrel which engage with the slots shown on the sketch. These slots are integral with most milling cutter chuck bodies having these tapers. The taper is 7 in 24 ($3\frac{1}{2}$ in./ft) on the diameter, and the tolerance "z" (0·4 mm or 0·0156″) refers to the axial position of the gauge diameter "G" relative to the end of the taper. (See over.)

The dimensions are given to the current ISO and British standards, but both the taper and the leading dimensions are directly interchangeable with the former imperial standards. The drawbar thread "G" may vary between makes.

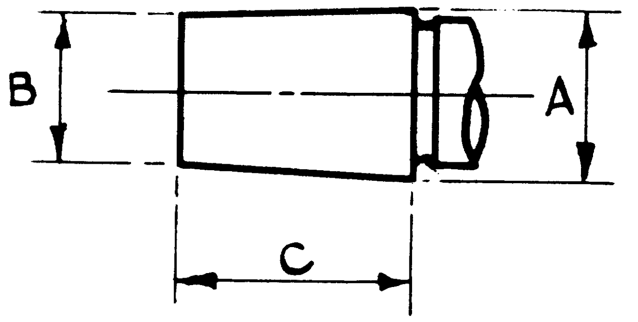


Dimensions in mm

IT Designation No.	*30*	*40*	*45*	*50*	*60*
D_1	31·750* (1·250″)	44·450 (1·750″)	57·150 (2·250″)	69·850 (2·750″)	107·950 (4·250″)
d_1 (−0·03/−0·04)	17·4*	25·3	32·4	39·6	60·2
d_3	16·5	24	30	38	58
l (maximum)	70	95	110	130	210
l_1	50	67	86	105	165
AD_1	8	10	10	12	16
AD_2	50	63	80	100	160
AD_3	13	17	21	26	32
AD_4	6	8	10	11	14
AD_5	4·5	5	5	6·5	8
Thread, g	M12	M16	M20	M24	M30
l_2	24	30	38	45	56
l_3 (minimum)	50	70	70	90	110
t (maximum)	16·2*	22·5	29	35·3	60
b (+0/+0·2)	16·1*	16·1	19·3	25·7	25·7
x	1·6	1·6	3·2	3·2	3·2
z	0·4	0·4	0·4	0·4	0·4

Note: Critical dimensions (*) in mm are exact within tolerances to those of imperial dimensions. Others are the nearest preferred metric number to the imperial equivalents. Arbors are virtually interchangeable apart from the drawbar thread.

JACOBS CHUCK ARBOR TAPERS

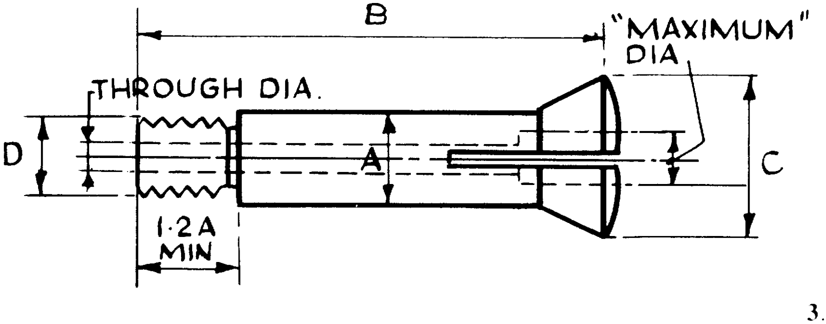


Series no.	*A (in.)*	*B (in.)*	*C (in.)*	*Taper (in./ft)*
0	·2500	·22844	·4375	·59145
1	·3840	·33341	·65625	·92508
2	·5590	·48764	·87500	·97861
2A	·5488	·49764	·75000	·97861
3	·8110	·74610	1·21875	·63898
4	1·1240	1·0372	1·6563	·62886
5	1·4130	1·3161	1·8750	·62010
6	0·6760	·6241	1·0000	·62292
33	0·6240	·5605	1·0000	·76194

The correct arbor is usually stated on the box containing the chuck (and is repeated on the arbor packing) but as a guide the following may help: 0–$\frac{1}{2}''$ chucks, nos 6 or 33; 0–$\frac{5}{32}''$, no. 0; 0–$\frac{1}{4}''$, no. 1; 0–$\frac{3}{8}''$, no. 2; taper no. 2A may be found on 0–$\frac{1}{4}''$ and 0–$\frac{5}{16}''$ chucks. Large size chucks with nos 3, 4, 5 tapers should always be checked as there is a wide variety of chucks from $\frac{17}{32}''$ to $1''$ capacity.

SPLIT COLLETS

The 'bore' of a collet is usually stated in units of 0·1 mm, i.e. a no. 43 will grip stock 4·3 mm dia. (Inch sizes are given as the actual bore.)



(a) General

Overall dimensions of collets, letter series, as used by George Adams.

Type	*L*	*A*	*B*	*C*	*C (long)*	*D*
Dia. A, mm	6	8	10	15	15	$12\frac{1}{2}$
B inches	$\frac{31}{32}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{13}{16}$	$3\frac{3}{8}$
D, mm	5	6·87	10	13	13	$11\frac{1}{2}$
Thread	0.7 mm	40 tpi	0·85 mm	1 mm	1 mm	1 mm
Max bore right through, mm	2·75	4·5	6	9·5	9·5	7·9

The above may be used to select a suitable size of collet for a new application, but the table below gives data for specific machines.

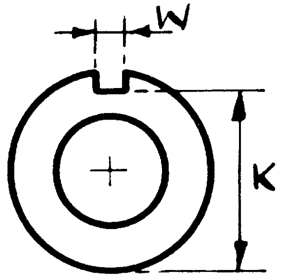
(b) Particular

Make	*A (mm)*	*B (mm)*	*C (mm)*	*D (in.)*	*Pitch or tpi*	*Capacity (mm) Thro.*	*Max.*
Adams	8	34·5	10·7	0·268	0·625 mm	43	70
Boley	8	32	12·2	0·268	0·625 mm	43	70
Boley/Leinen	8	34	12·8	0·268	40 tpi	43	70
Coronet	8	32	12·2	0·268	0·625 mm	43	70
Derbyshire	8	34	12·8	0·268	40 tpi	43	70
Derbyshire	10	41·5	14	0·392	1 mm**	59	90
I.M.E.	8	34	12·5	0·275	0·625 mm	43	70
Lorch	8	34	12·5	0·275	0·625	43	70
Lorch	10	73	14	0·362	1·0 mm	60	100
Pultra	8	32	12·2	0·268	0·625 mm	43	70
'WW'[1]	8	32	12·2	0·268	0·625 mm	43	70
Wolf Jahn	8	34·5	13·2	0·270	40 tpi	43	70
Most 6 mm lathes	6	30	10·7	0·199	0·7 mm	30	50

**Buttress thread. [1]Webster-Whitcomb.

Important note

The cone angle should always be checked. Though most are nominally 40° included there may be small variations. The Lorch 10 mm collet is about $38\frac{1}{2}$° and some makers use 30°. *Interchangeability*. The difference in thread sizes for the 8 mm range is so small that an 'easy' fit draw-tube will accept most makes of collets. The obstacle to such use is usually the *keyway*. There are substantial differences in size of key or peg used, but a keyway 2·0 mm wide × 0·7 mm deep will fit most common lathes. Actual keyway sizes for four machines are given below, all using 8 mm collets. See p. **3**.7.



Make	*W (mm)*	*K (mm)*
Boley & Leinen	2·0	7·5
Geo. Adams	1·75	7·3
Lorch	1·75	7·35
Wolf-Jahn	1·875	7·30

Body clearance
The hole in the mandrel is the dimension 'A' and the body of the collet has a small clearance. This ranges from 0·0005″ to 0·0007″ on examples of 8 mm collets measured.

DIMENSIONS OF CLARKSON SCREWED SHANK ENDMILLS

To enable model engineers to make arbors to carry (e.g.) slitting saws in an 'Autolock' milling chuck Messrs Clarkson have kindly provided the following details. The arbor should be turned between centres, and a $\frac{3}{16}$″ centre drill (BS 985 S2) used at the screwed end to fit the bearing cone. In all cases the thread is 20 tpi Whitworth form, cut 0·003″ to 0·006″ undersize on the effective diameter, and the tolerance on the parallel shank is +0·000″ to −0·001″.

Nominal shank diameter	$\frac{1}{4}$ in. (6 mm)	$\frac{3}{8}$ in. (10 mm)	$\frac{1}{2}$ in. (12 mm)	$\frac{5}{8}$ in. (16 mm)	1 in. (25 mm)	$1\frac{1}{4}$ in. (32 mm)
Minimum thread length	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{9}{16}$″	$\frac{9}{16}$″
Minimum shank length (including thread)	$1\frac{15}{32}$″	$1\frac{1}{2}$″	$1\frac{1}{2}$″	$1\frac{17}{32}$″	$2\frac{1}{16}$″	$2\frac{1}{8}$″

THREADED DRILL CHUCKS

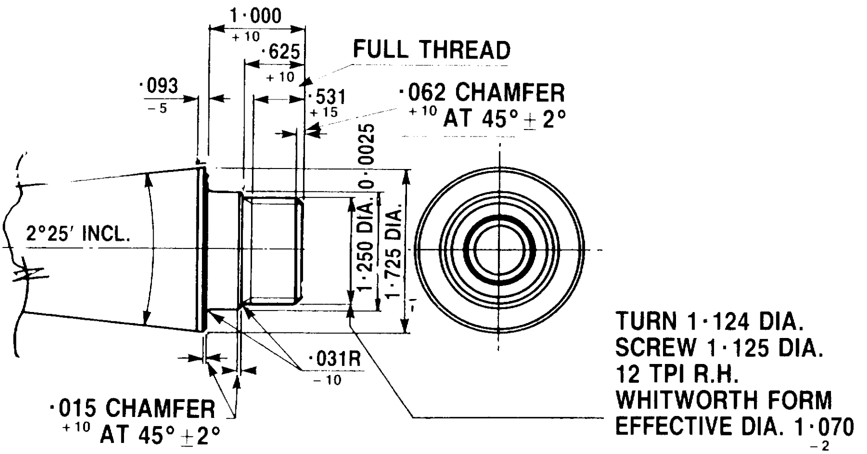
Screwed socket or shank chucks found on power tools should always be checked; there is a wide variety of threads, and they may be to US standards.

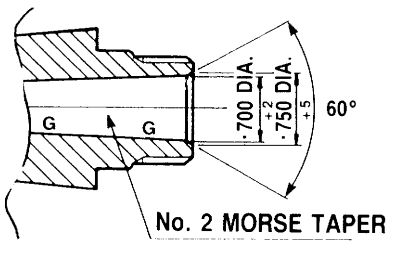
The $\frac{1}{2}$″ screwed arbor found on the hand-tightened chucks of breast drills is usually 24 tpi.

CONED SHAFT ENDS

Shaft ends which are coned for the fitting of flywheels, etc., should have a taper of 1 in 10 on the diameter, but some automobile tapers to SAE standards will be found to be 1 in 8 on the diameter. Tapers for small marine propeller fittings are often finer, at 1 in 16. In all cases the smaller diameter is treated as the 'ruling' dimension. Keyways should be cut parallel to the slope of the taper.

DIMENSIONS OF MYFORD SERIES 7 MANDREL NOSE



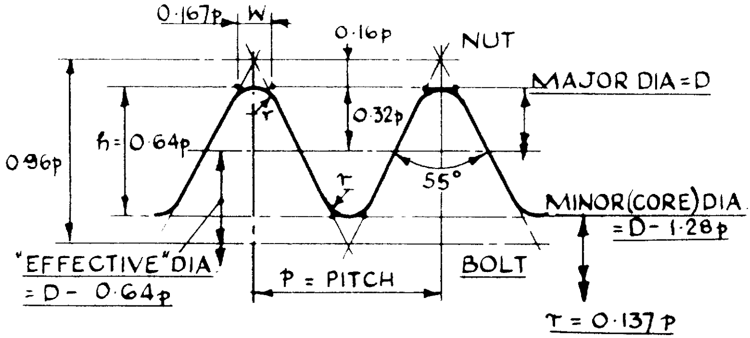


Note.
Mandrel bearing shown is for the SUPER 7, but the nose profile is the same for all types.

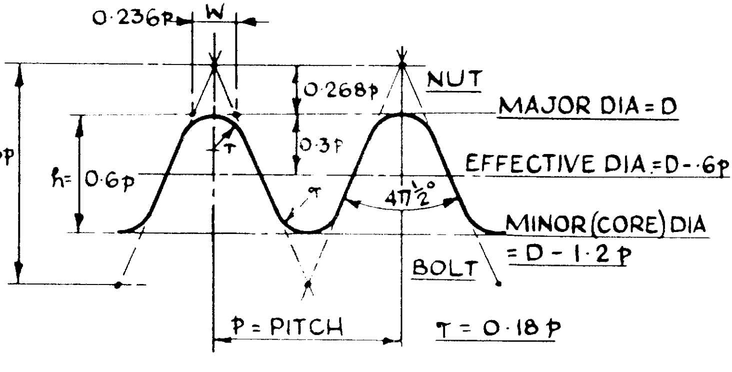
SECTION FOUR
SCREW THREADS
WITH TABLES OF TAPPING DRILLS AND HEXAGON DIMENSIONS

SCREW THREADS AND SCREWCUTTING

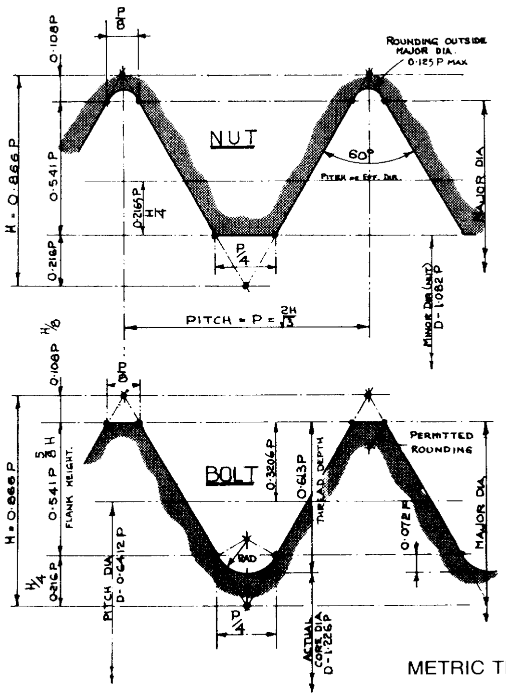
The following figures show the profiles of thread forms likely to be of use to the model engineer. (The Whitworth form applies to BSF, ME, BSP, and the 26 and 32 tpi series as well as BSW.) In each case the female (nut) thread is the upper profile and the male thread the lower. The 'major' diameter is the nominal or top diameter of the bolt, and the 'minor' the diameter at the root of the bolt thread, sometimes called the 'core' diameter. The dimension 'h' is the height of the bolt thread if cut to perfect shape with an unworn tool.



WHITWORTH FORM



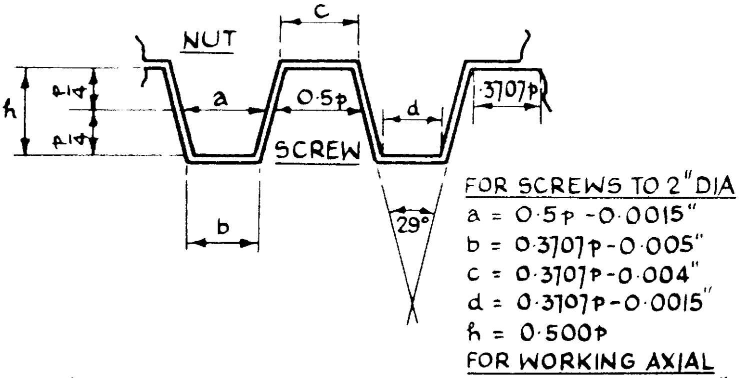
BRITISH ASSOCIATION (B.A) FORM



METRIC THREAD to BS 3643

Both ISO metric and unified threads have flats to one or both of the root and crest of the thread. However, these are usually rounded a trifle to avoid sharp corners, the condition being that any rounding on the CREST must be *within* and tangential to the flat, and rounding of the roots must lie wholly *outside* the flat, to avoid any risk of interference.

Details are given in the tables which follow which will cover needs both for design and construction of models. The core area is that used for calculating the strength of the bolt,



STANDARD ACME FORM

based on the selected stress conditions. Although model engineers are unlikely to use unified threads, these have been included as many commercial accessories – chucks especially – may include stud-holes to this profile. This thread has in some fields replaced the standard Whitworth and BSF threads, and completely replaced the old American threads. After a relatively short life it is, in turn being replaced by the ISO metric system in Great Britain, but not at present in the USA. The relevant British Standard Specification for British metric threads is BS 3643, p. 58. Metric model engineer threads are covered by BS PD6507–1982.

Tapping drills

The action of a perfect tap in a perfect drilled hole would be to cut a clean thread, the crest of which had a degree of flattening depending on the size of the drill; if the drill were exactly at 'core diameter' then the thread would be of perfect form. In practice all taps produce some degree of extrusion of metal due to the sideways component of the cutting forces. For this reason tapping drills are always larger than the core diameter of the thread. How much larger is a matter for judgement, and much of the trouble experienced with tap breakages is due to the use of too small a drill.

Published tables of tap drills tend to work on figures between 70% and 85%, and in small sizes especially the 'nearest drill' size may increase this figure. The 'percentage thread' *ignores* any extrusion effect, and indicates the proportion of the thread height 'h' which would remain in the nut after cutting if no extrusion took place. This figure, however, in no way indicates either that the thread is that much weaker, or that it will be that much slacker a fit. The actual bearing area on a Whitworth form thread, for example, is only 78% of the thread height in a **perfect** thread – the nut and bolt should bear only on the straight flanks – and if a '70%' tapping drill is used the loss of bearing area is only 25% (for a BA thread this loss is less still and is negligible on an ISO thread).

The strength of a perfect thread in bearing – that is, the resistance of the thread to failure by crushing of the metal – is at least four times as great as the resistance to failure in tension (i.e. to snapping off at the root) so that even a loss of bearing area of 50% would be acceptable. The other type of failure – shear across the thread, or stripping – is unaffected by any reduction of thread engagement until at least 50% of the thread has been removed; even then the screw would still be stronger in shear than in tension.

One proviso must be made, however – that is that the bolt must be within reasonable tolerance of the correct top diameter. This is easier to achieve than a 'correct nut', as the bolt can be measured, even down to 16 BA, with some accuracy. This is recognised by the British Standards Institution which, for commercial bolts and nuts, allows three to four times the tolerance on the minor diameter of a nut compared with that for the top diameter of the bolt.

In determining the size of tapping drill another factor is involved – the torque required to drive the tap; excessive torque will cause breakage especially in small sizes. Some work done on the unified thread shows that if the torque to drive a tap into a 60% hole is 100 units, that for a 72% hole is 200 units, and for an 80% hole as much as 300 units. (One drill manufacturer's table for BA threads shows drills averaging 85% between 4 BA and 10 BA!)

The majority of published tapping drill tables are intended for use with tapping machines or adaptors. These have slipping clutches and/or automatic reversal to cope with overload or bottoming of the tap. The tables are designed to accept the higher thread engagement found in 'production' workshops and are not suitable for hand tapping.

In view of the above, the tapping drills in the following tables have been calculated on a size to give at least 65% thread height for threads below $\frac{3}{8}$ inch, and up to 75% for larger sizes. It is, however, recommended that for all holes of $\frac{1}{4}''$ dia. (6 mm) and below users should set aside drills *specially for use with taps*, rather than employing the general workshop stock, and that these drills should be replaced when worn rather than be resharpened. This will reduce the risk of drilling oversize holes.

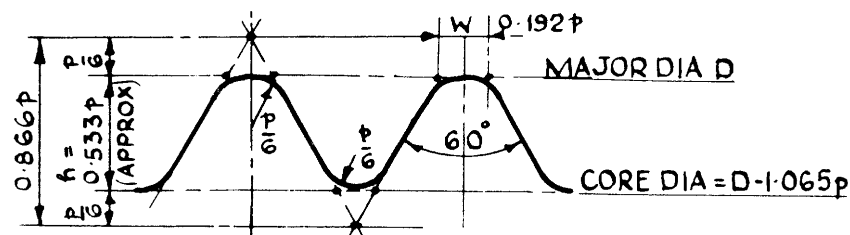
These tapping sizes will serve for most materials, but the data and method of calculating drill size for other desired depths of engagement are shown below each table. Note, however, that it is common practice to use engagements of 85% or higher when tapping into sheet metal when the thickness is less than the core diameter of the thread.

The following figures have been recommended by one authority for a range of materials:

Material	*% Thread*
Mild- and un-heat treated steels	60–65
High-chrome high-carbon steel	50
High speed steel (soft)	55
Stainless steel	50
Stainless steel, free-cutting	60
Cast iron	70–75
Wrought aluminium	65
Cast aluminium	75
Wrought copper	60
Free-cutting yellow brass	70
Drawn brass	65
Manganese bronze	55
Monel metal	55–60
German silver (nickel silver)	50–60

It will be seen that these differ little from the tabulated values, which may be used for most metals. (But use no more than 50% for uranium!)

See also *Drills, Taps & Dies* by the same author, which deals with tapping sizes in more detail (published by Nexus Special Interests).



BRITISH CYCLE THREAD

(a) Table of dimensions and tapping drills for BSW, BSF, 'Model Engineer', and Unified Fine Threads.

	WHITWORTH					*B.S. FINE*					*MODEL ENGINEER*					*UNIFIED FINE (UNF)*			
Dia. in.	*TPI*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill no.*	*TPI*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill no.*	*TPI*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill no.*	*TPI*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*
$\frac{1}{16}$	60	·0413	·0013	1·05	59														
$\frac{3}{32}$	48	·067	·0035	1·95	48														
$\frac{1}{8}$	40	·093	·0068	2·65	37						As BSW								
$\frac{5}{32}$	32	·116	·0106	3·3	30						40	·125	·0123	3·4	29				
$\frac{3}{16}$	24	·134	·0141	3·9	24	32	·148	·0171	4·1	20	40	·155	·0189	4·2	19				
$\frac{7}{32}$	24	·154	·0186	4·7	14	28	·173	·0235	4·8	12	40	·187	·0275	5·0	9				
$\frac{1}{4}$	20	·186	·0272	5·3	5	26	·201	·0317	5·5	3	40	·218	·0373	5·8	1	28	·211	·0349	5·6
$\frac{5}{16}$	18	·241	·0457	6·7	H	22	·254	·0508	7·0	I	32	·272	·0581	7·2	K	24	·267	·0560	7·0
$\frac{3}{8}$	16	·295	·0683	8·2	P	20	·311	·0760	8·4	Q	32	·335	·0882	8·8	S	24	·330	·0856	8·6
$\frac{7}{16}$	14	·346	·0941	9·5	$\frac{3}{8}$	18	·366	·1054	10·0	W	32	·397	·1238	10·4	Z	20	·383	·1153	10·0
$\frac{1}{2}$	12	·393	·1214	10·7	$\frac{27}{64}$	16	·420	·1384	11·3	$\frac{7}{16}$	32	·460	·1663	12·0		20	·446	·1563	11·5
$\frac{9}{16}$	12	·456	·1631	12·4		16	·483	·1828	12·9	$\frac{1}{2}$"	*See also below*					18	·502	·1980	13·0
$\frac{5}{8}$	11	·509	·2032	13·8	—	14	·534	·2236	14·25	—	$\frac{5}{16}\times 40$	·282	·063	7·4	L	18	·565	·2508	14·5
$\frac{3}{4}$	10	·622	·3039	16·75		12	·643	·3249	17·0		$\frac{3}{8}\times 40$	·343	·0924	9·0	T	16	·682	·3655	17·5
$\frac{7}{8}$	9	·733	·4218	19·50		11	·759	·4520	20·0		$\frac{7}{16}\times 40$	·405	·1289	10·5	Z	14	·798	·5003	20·5
1	8	·840	·5542	22·5		10	·872	·5972	23		$\frac{1}{2}\times 40$	·468	·1721	12·1		12	·898	·6336	23·5

Tapping drills quoted provide 65% thread engagement up to $\frac{3}{8}$" dia.; thereafter from 70 to 75% in larger sizes.
The metric drill sizes shown are the *preferred* drill; Morse No. & Letter sizes approximate, usually slightly larger thread engagement (these drill sizes are now obsolete). BSP tapping sizes are for 70–75% engagement.

Thread heights For Whitworth and BSF screws, thread height $= \dfrac{0\cdot 64}{\text{tpi}}$, and core diameter $= D - 1\cdot 28P$ ($P = 1/\text{tpi}$).
For 40 tpi M.E. series, thread height = 0·016", core dia. = D − 0·032"
For 32 tpi M.E. series, thread height = 0·020", core dia. = D − 0·040"
For UNF threads, bolt thread height = 0·613P, and core dia. = D − 1·226P, but note that the crest of the UNF standard nut is truncated by approximately 0·072P.

To find a tapping drill to give any desired depth of engagement E%

$$\text{Drill diameter} = \text{Bolt diameter} - \left(2 \times \text{thread height} \times \frac{E}{100}\right).$$

(b) Table of dimensions and tapping drills for 26 tpi, 32 tpi, and British Standard Pipe Threads (Parallel) of Whitworth form.

	26 TPI SERIES				*32 TPI SERIES*				*BSP (PARALLEL)*					
Dia. in.	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill no.*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill no.*	*O. dia. in.*	*TPI*	*Core dia. in.*	*Core area sq. in.*	*Tap drill mm*	*Tap drill in.*
$\frac{1}{8}$									0·383	28	·337	·0892	8·8	$\frac{23}{64}$
$\frac{3}{16}$					·147	·0170	4·1	20						
$\frac{1}{4}$	·200	·0314	5·4	3	·210	·0346	5·7	2	·0518	19	·451	·1598	11·9	$\frac{15}{32}$
$\frac{5}{16}$	·263	·0543	7·2	K	·272	·0581	7·2	$\frac{9}{32}$						
$\frac{3}{8}$	·326	·0835	8·7	S	·335	·0882	8·8	S	0·656	19	·589	·2726	15·25	$\frac{39}{64}$
$\frac{7}{16}$	·388	·1183	10·3	Y	·397	·1283	10·4	Z						
$\frac{1}{2}$	·451	·1598	11·8		·460	·1663	12		0·825	14	·734	·4231	19·25	$\frac{49}{64}$
$\frac{3}{4}$	·652	·3339	18·0						1·041	14	·950	·7091	24·75	$\frac{63}{64}$

(c) Table of dimensions and tapping drills for British Association (BA) threads.

BA no.	*Top dia. in.*	*Top dia. mm*	*TPI (approx.)*	*Pitch mm*	*Core dia. mm*	*Core area sq. mm*	*Tap drill mm*	*Tap drill Number (approx.)*	*Scale bolt at 1 in. = 1 ft inches*
0	·2362	6·000	25·4	1·0	4·8	18·1	5·2	6	$2\frac{7}{8}$
1	·2087	5·3	28·2	0·9	4·22	14·0	4·6	15	$2\frac{1}{2}$
2	·1850	4·7	31·4	0·81	3·73	10·9	4·1	21	$2\frac{1}{4}$
3	·161	4·1	34·8	0·73	3·22	8·15	3·5	29	2
4	·1417	3·60	38·5	0·66	2·81	6·20	3·1	31	$1\frac{3}{4}$
5	·1260	3·2	43·1	0·59	2·49	4·87	2·75	36	$1\frac{1}{2}$
6	·1102	2·80	47·9	0·53	2·16	3·67	2·35	42	$1\frac{5}{16}$
7	·0984	2·50	52·9	0·48	1·92	2·90	2·1	45	$1\frac{3}{16}$
8	·0866	2·20	59·1	0·43	1·68	2·22	1·85	50	1
9	·0748	1·90	65·1	0·39	1·43	1·61	1·6	52	$\frac{7}{8}$
10	·0669	1·70	72·6	0·35	1·28	1·29	1·45	54	$\frac{3}{4}$
11	·0590	1·50	81·9	0·31	1·13	1·00	1·25	56	—
12	·0512	1·3	90·7	0·28	0·960	0·724	1·10	58	$\frac{5}{8}$
14	·0394	1·00	110	0·23	0·720	0·407	0·82	66	$\frac{1}{2}$
16	·0311	0·79	134	0·19	0·560	0·246	0·65	71	$\frac{3}{8}$
18	·024	0·62	169	0·15	0·440	0·150	0·5	76	$\frac{9}{32}$

The BA tapping sizes are all calculated for a thread engagement of 65%. For this series, thread height = 0·6 × pitch; core dia. = D − 1·2P.

Commercial nuts made to British Standard limits may have a minor (tap drill) diameter such that there is a flat on the crest of the nut thread. Such nuts offer about 65% thread engagement.

The *pitch* of each BA thread given by $P = 0{\cdot}9^{N}$, where N is the BA number, rounded off to the nearest 0·01 mm. The *diameter* of a BA screw is given by $D = 6P^{1{\cdot}2}$.

BA sizes from no. 17 to no. 25 are generally unobtainable, being replaced by the standard 'watch' or horological threads.

General note on metric threads for model engineers

In May 1980 Messrs Model & Allied Publications Ltd set up a working party to study the implications of a future change from BA and ME to metric threads on the practice of model engineering, and to recommend suitable diameter-pitch combinations both for fasteners (nuts and bolts) and attachment or union threads. The working party also considered the problem of hexagon sizes, the standard ISO hexagons being the wrong proportions for accurate model work.

The report was finalised in April 1981 following a conference at Wembley in January of that year. It was later considered by the British Standards Institution and, whilst it cannot be issued as a 'British Standard', it has been published by BSI under the title *Guidance on Metric Screw Threads and Fasteners for Use by Model Engineers* as PD6507-1982. The tables which follow contain, with the permission of BSI, extracts from that document as well as from BS3643 and 4827 *ISO Metric Screw Threads*.

It should be noted that apart from the hexagon sizes all the threads in Table (d) are the normal (coarse) pitch threads to BS3643 (except that for 0·8 mm, which is from BS4827) and all those shown in Table (e) except 12 × 0·75 are from the BS standard 'constant pitch' series. Taps and dies are readily available. No '1st or 2nd preferences' have been indicated, as these will depend on the needs of model engineers when the system comes into common usage.

One practical point. The *OD of a metric tap will not correspond to the nominal diameter of the screw* owing to the rounding of the major diameter of the female thread.

(d) Metric fastener threads for model engineers to PD6507-1982, with tapping drills for 65% flank engagement. The table includes sizes from 6 mm to 16 mm (to BS3643, but not included in PD6507), which the working party recommended be used above 5·0 mm dia.

Nom. dia. mm	*Pitch mm*	*Minor dia. mm*	*Tensile stress area mm^2*	*Tapping drill mm*	*Hexagon, A–F ISO mm*	*Hexagon, A–F MME mm*	*Nearest BA or imperial size*
0·8	0·2	0·608	0·31	0·68	—	1·3	16 BA
1·0	0·25	0·675	0·46	0·82	2·5	1·5	14 BA
1·2	0·25	0·875	0·73	1·0	3·0	2·0	13 BA
1·4	0·30	1·014	0·98	1·2	3·0	2·5	11–12 BA
1·6	0·35	1·151	1·27	1·35	3·2	3·0	10–11 BA
1·8	0·35	1·351	1·70	1·55	**	3·2	9–10 BA
2·0	0·40	1·490	2·1	1·70	4·0	3·5	9 BA
2·2	0·45	1·628	2·5	1·90	**	4·0	8 BA
2·5	0·45	1·928	3·4	2·20	5·0	4·5	7 BA
3·0	0·5	2·367	5·0	2·65	5·5	5·0	5–6 BA
3·5	0·6	2·743	6·8	3·10	**	6·0	4–5 BA
4·0	0·7	3·120	8·8	3·50	7·0	7·0	3–4 BA

Continued on following page.

Nom. dia. mm	*Pitch mm*	*Minor dia. mm*	*Tensile stress area mm^2*	*Tapping drill mm*	*Hexagon, A–F ISO mm*	*MME mm*	*Nearest BA or imperial size*
4·5	0·75	3·558	11·5	4·0	**	8·0	2–3 BA
5·0	0·8	3·995	14·2	4·50	8·0	9·0	1–2 BA
6·0	1·0	4·747	20·1	5·3	10·0	—	0 BA–$\frac{1}{4}$″ BSF
8·0	1·25	6·438	36·6	7·1	13·0	—	$\frac{5}{16}$″ BSF
10·0	1·50	8·128	58·0	8·8	17·0	—	$\frac{3}{8}$″ BSF
12·0	1·75	9·819	84·3	10·70	19·0	—	$\frac{1}{2}$″ BSF
16·0	2·00	13·51	157·0	14·5	24·0	—	$\frac{5}{8}$″ BSF

**There is no standard 'machined' nut quoted in BS for these sizes, but commercial (usually pressed) nuts may be available.

(e) Metric attachment threads (eg for unions and fittings) for model engineers, PD6507-1982.

Nom. dia. mm	*Pitch mm*	*Minor (core) dia. (1) mm*	*Nominal core area mm^2*	*Tapping drill 75% mm*	*Tapping drill 65% mm*	*(4) Hexagon A–F mm*
3·0 (2)	0·5	2·38	4·45	2·60	2·65	4·5
4·0	0·5	3·39	9·02	3·60	3·65	5·5
4·5	0·5	3·9	11·95	4·10	4·15	6·0
5·0	0·5	4·39	15·1	4·60	4·65	7·0
5·5	0·5	4·9	18·9	5·10	5·15	7·0
6·0	0·5	5·39	22·8	6·60	6·65	8·0
4·5 (2)	0·75	3·58	10·1	3·9	4·0	6·0
6·0	0·75	5·08	20·2	5·4	5·5	8·0
7·0	0·75	6·08	29·0	6·4	6·5	9·0
8·0	0·75	7·08	39·4	7·4	7·5	10·0
10·0	0·75	9·08	64·7	9·4	9·5	12·0
12·0	0·75	11·08	96·4	11·4	11·5	14·0
10·0	1·0	8·77	60·4	9·2	9·3	12·0
12·0	1·0	10·77	91·1	11·2 (3)	11·3	14·0
14·0	1·0	12·77	128	13·2 (3)	13·3	17·0
16·0	1·0	14·77	171	15·2 (3)	15·3	20·0
18·0	1·0	16·77	221	17·2 (3)	17·3	22·0
20	1·0	18·77	277	19·2 (3)	19·3	24·0

Notes: (1) Assumes root radius making $d = D - 1{\cdot}23P$.
(2) These are normal 'fastener' sizes from Table (d).
(3) In this range drills at intervals of 0·25 mm (e.g. 13·25) are 'preferred' and therefore cheaper. They will provide 70% flank engagement.
(4) Recommended for union nuts etc.

Tapping drills for any other percent depth of flank engagement 'E%' may be found for metric threads to BS3643 by subtracting from the *nominal* (top) diameter an amount equal to $\frac{1 \cdot 08E}{100} \times P$. Note that tapping drills cannot be determined in terms of thread height in this case, as the height depends on the degree of rounding on the crest of the tap.

(f) Sparking plug sizes to SAE standard

Dia. mm	*Pitch mm*	*Core dia. mm*	*Tapping drill mm 60%*	*70%*
10	1·0	8·75	9·25	9·1
12	1·25	10·44	11·10	10·9
14	1·25	12·44	13·10	12·9
18	1·5	15·75	16·75	16·5

Holtzapffel screw threads

Details of these unusual threads are included for the benefit of those who own Holtzapffel lathes and are engaged in repair or restoration. The threads were originated by Holtzapffel in 1794–5, and standardised in the years 1796–1804. From 1820 most lead and guide-screws were changed to 10 tpi, and after about 1850 all screws and nuts above ½" dia. (except the mandrel nose thread) were changed to aliquot threads.

When making replacement screws, the original must be examined to see whether it is of the 'deep' (about 50°) or 'shallow (about 60°) thread form. The pitches below ½" and the 9·45 nose thread were used till 1924.

The 'screw hob' mentioned in the table is the guide bobbin on traversing mandrel screw-cutting lathes.

Tap mark letter	*Dia. in.*	*Thread no.*	*TPI*	*Hob no.*	*Tap mark letter*	*Dia. in.*	*Thread no.*	*TPI*	*Hob no.*
A	1·00	1	6·58		K	¼	8	25·71	5
B	0·875	2	8·25		L	0·24	10	36·10	6
C	¾	3	9·45	1	M	0·21	9	28·88	
D_1	⅝	4	13·09	2	N	0·210	10	36·10	
D_2	$\frac{9}{16}$	4	13·09	2	O	0·18½	11	39·83	
E	½	4	13·09	2	P	0·18	10	36·10	
F	0·45	5	16·5	3	Q	0·16¼	11	39·83	
G	0·41	6	19·89	4	R	0·15	12	55·11	
H	0·36	6	19·89	4	S	0·13½	12	55·11	
—	—	7	22·12		T	0·12	12	55·11	
I	0·33	8	25·71	5	U	0·10	12	55·11	
J	0·29	8	25·71	5					

Thread heights. 'Deep' = 1·072P. 'Shallow' = 0·866P. Crest and root is a sharp point.

Screwcutting Holtzappfel threads

The following are for a MYFORD lathe *with gearbox*, but can be used with changewheels as follows: where 8 tpi is called for in column 2, set up as shown; for 16 tpi use 25T on the mandrel; for others, set up additional wheels to cut the column 2 thread in place of idlers in the chain. The half-nuts *must* remain engaged throughout for all threads.

Holtz tpi	*Gearbox set tpi*	*Mandrel wheel*	*First stud DriveR*	*First stud DriveN*	*Second DriveR*	*Second DriveN*	*Leadscrew wheel*	*Actual pitch*	*Error*
6·58	14	50	47	60	IDLER (55)		30	6·580	none
8·25	11	60	——— ANY IDLERS ———				45	8·250	none
9·45	8	40	IDLER (50)		45	40	42	9·450	none
	or	60	IDLER (50)		45	40	63		
13·09	11	30	51	60	IDLER (50)		42	13·090	none
16·5	11	40	——— IDLERS ———				60	16·500	none
19·89	9	50	65	30	IDLER (60)		51	19·890	none
22·12	16	50	IDLER (55)		35	38	75	22·105	0·000031″/in.
25·71	20	27	——— IDLERS ———				21	25·714	0·000006″/in.
28·88	16	50	38	40	38	30	75	22·880	none
36·1	19	30	——— IDLERS ———				57	36·100	none
39·83	14	37	IDLER (65)		25	75	39	39·8461	0·0004″/in.
55·11	26	53	——— IDLERS ———				25	55·12	0·0002″/in.

Screwcutting in the lathe

The drawings on page **4**.1 etc. show the detail of the root of the thread forms – note that those of nut and bolt differ in the ISO and unified series. Tool points *should* be ground to the correct angle and the point then reduced to a flat of width 'W' after which it may be rounded. Alternatively, the amount 'D' can be ground from the point by direct measurement. However, for pitches less than about 2 mm (12 tpi) the dimensions are too small to measure sufficiently accurately, and it is usual to use a point with no more than the 'sharp edge removed'. The following table gives a close approximation to the amount of radial infeed for the tool point in the two cases, but the final sizing in accurate work must be tested by trial.

	Infeed			
	Bolts		*Nuts*	
Thread form	*Rounded*	*Sharp*	*Rounded*	*Sharp*
Whitworth	0·64P	0·8P	0·64P$-\frac{1}{2}$S	0·8P$-\frac{1}{2}$S
BA	0·6P	0·82P	0·6P$-\frac{1}{2}$S	0·82P$-\frac{1}{2}$S
Metric (SI)	0·625P	0·72P	0·625P$-\frac{1}{2}$S	0·72P$-\frac{1}{2}$S
UNF	0·613P	0·83P	0·613P$-\frac{1}{2}$S	0·72P$-\frac{1}{2}$S

Where 'S' is the difference between the drill diameter and the core diameter of the thread concerned.

If a fully-formed multi-point thread chaser is used in the toolpost, then the depth of infeed is that for the 'rounded' tool above.

External threads

The importance of the correct major or top diameter of the screw has already been referred to. As the normal split die has a tendency to 'extrude' metal as well as cut, and also may open against the retaining screws under load, it is essential to check the diameter after the first pass, as a die, 'set to gauge' may well cut large. At the same time, it is unreasonable to expect a die to 'machine down' the bar-stock. The bar diameter should always be to size or preferably a few thousandths of an inch small before using a die. For small threads (say 4BA downwards) the Swiss-type die, which is not split, is to be preferred.

HEXAGON NUT AND BOLT-HEAD DIMENSIONS

In modelling engines and machinery made prior to about 1939 most nut and bolt-heads on steam engines would be to the then 'Standard Whitworth' dimensions, though 'British Standard Fine' hexagons were used in congested areas of medium and large size diesel and gas engines – on cylinder heads and bearing caps especially. Very few horizontal oil or gas engines used BSF nuts. Mass-produced car engines and small stationary IC engines used the 'across flats' SAE standard hexagons almost universally from about 1930 onwards. The 'BSF' hexagon was, usually, that of the next smallest size of bolt – e.g. $\frac{1}{2}''$ BSF fits a $\frac{7}{16}''$ Whitworth spanner. From 1941 onwards the 'old' Whitworth hexagon was phased out, simply to save metal during the war, and both BSW and BSF bolts used the same (BSF) hexagon sizes. This practice has remained a British Standard ever since. The 'unified' threaded bolts and nuts adopted the 'across flats' spanner sizes of the SAE series.

The following 'rules' will give a rough approximation to the 'old Whitworth' hexagon proportions, covering those not listed below.

Across flats $= 1{\cdot}5D + \frac{1}{8}''$. Across corners $= 1{\cdot}55 \times$ A–F. Nut height = D.

Bolt-head height $= \frac{7}{8} \times D$. Locknut thickness $= 0{\cdot}5D$. D = Nominal bolt dia., inches.

Model nuts and bolt-heads scaled to these proportions will 'look right' and be reasonably authentic for earlier prototypes. *Note*, however, that if 'reduced head' BA bolts are used with visible nuts, then the nuts should match the head hexagon. (Tables follow.)

BSW and BSF *Dimensions in inches*

	Std. Whitworth					BS Fine				
Dia.	*Across flats*	*Across corner*	*Head height*	*Nut thickness*	*Locknut thickness*	*Across flats*	*Across corner*	*Head height*	*Nut thickness*	*Locknut thickness*
$\frac{1}{4}$	0·525	0·61	0·22	0·25	0·17	0·445	0·52	0·186	0·220	0·185
$\frac{5}{16}$	0·601	0·694	0·273	0·312	0·21	0·525	0·61	0·228	0·270	0·210
$\frac{3}{8}$	0·710	0·82	0·33	0·375	0·25	0·601	0·694	0·270	0·332	0·260
$\frac{7}{16}$	0·820	0·947	0·363	0·467	0·30	0·710	0·820	0·312	0·397	0·275
$\frac{1}{2}$	0·920	1·06	0·44	0·50	0·33	0·820	0·947	0·363	0·467	0·300
$\frac{5}{8}$	1·100	1·27	0·55	0·625	0·42	1·010	1·167	0·447	0·602	0·410
$\frac{3}{4}$	1·30	1·50	0·66	0·75	0·50	1·200	1·386	0·530	0·728	0·490
$\frac{7}{8}$	1·480	1·71	0·77	0·88	0·58	1·300	1·501	0·623	0·810	0·550
1	1·670	1·93	0·88	1·00	0·67	1·480	1·71	0·71	0·94	0·63
$1\frac{1}{4}$	2·05	2·37	1·09	1·25	0·83	1·860	2·15	0·89	1·21	0·81
$1\frac{1}{2}$	2·41	2·78	1·31	1·50	1·00	2·22	2·56	1·06	1·45	0·98
$1\frac{3}{4}$	2·75	3·19	1·53	1·75	1·17	2·58	2·98	1·27	1·72	1·16
2	3·15	3·64	1·75	2·00	1·33	2·76	3·18	1·43	1·85	1·25
$2\frac{1}{2}$	3·89	4·49	2·19	2·5	1·67	3·55	4·10	1·77	2·22	1·60
3	4·53	5·23	2·63	3·00	2·00	4·18	4·83	2·15	2·775	1·98

All dimensions in the table above are to 'maximum metal' sizes.

Unified, UNC & UNF *Inches*

Dia.	*Across flats*	*Across corners*	*Head height*	*Nut thickness*	*Locknut thickness*
$\frac{1}{4}$	$\frac{7}{16}$	0·505	0·163	0·224	0·161
$\frac{5}{16}$	$\frac{1}{2}$	0·577	0·211	0·271	0·192
$\frac{3}{8}$	$\frac{9}{16}$	0·650	0·243	0·333	0·224
$\frac{7}{16}$	$\frac{5}{8}$**	0·720	0·291	0·380	0·255
$\frac{1}{2}$	$\frac{3}{4}$	0·866	0·323	0·442	0·317
$\frac{5}{8}$	$\frac{15}{16}$	1·082	0·403	0·552	0·380
$\frac{3}{4}$	$1\frac{1}{8}$	1·30	0·483	0·651	0·432
$\frac{7}{8}$	$1\frac{5}{16}$	1·52	0·563	0·760	0·494
1	$1\frac{1}{2}$	1·73	0·627	0·874	0·562

Dimensions are to 'maximum metal' sizes.
**Bolt head $\frac{1}{16}$" smaller.

ISO Metric *Millimetres*

Dia.	*Across flats*	*Across corners*	*Head height*	*Nut thickness*	*Locknut thickness*
2	4·0	4·68	1·53	1·6	—
2½	5	4·78	1·83	2·0	—
3	5·5	6·35	2·13	2·40	—
4	7·0	8·09	2·93	3·20	—
5	8·0	9·24	3·65	4·0	—
6	10	11·55	4·15	5·0	4·7
8	13	15·01	5·65	6·5	5·0
10	17	19·6	7·18	8·0	6·0
12	19	22·0	8·18	10·0	7·0
14	22	25·4	9·18	11·0	8·0
16	24	27·7	10·18	13·0	8·0
18	27	31·2	12·21	15·0	8·0
20	30	34·6	13·21	16·0	9·0
22	32	37·0	14·21	18·0	10·0
24	36	41·6	15·21	19·0	10·0

Dimensions are to 'maximum metal' sizes.
Recommended sizes for 'metric model engineers' standard hexagons for nuts and unions will be found on page **4**.7.
The above are to the standard (regular) ISO specification.
Note. This table shows the ISO 'first preference' sizes only.

British Association (BA) *Inch dimensions*

BA no.	*Across flats*	*Across corners*	*Head height*	*Nut thickness*	*Locknut thickness*
0	0·413*	0·48*	0·177	0·213	0·157
1	0·365	0·42	0·156	0·188	0·130
2	0·324	0·37	0·139	0·167	0·123
3	0·282	0·33	0·121	0·153	0·103
4	0·248	0·29	0·106	0·135	0·094
5	0·220	0·25	0·094	0·120	0·084
6	0·193	0·22	0·083	0·105	0·073
7	0·172	0·20	0·074	0·094	0·060
8	0·152	0·18	0·065	0·082	0·052
9	0·131	0·15	0·056	0·071	n.s.
10	0·114	0·14	0·050	0·064	n.s.
11	0·103	0·12	n.s.	0·056	n.s.
12	0·090	0·10	n.s.	0·049	n.s.
13	0·083	0·10	n.s.	0·045	n.s.
14	0·069	0·08	n.s.	0·037	n.s.

Although BA threads are basically metric, the British Standard calls for all dimensions and tolerances to be in inches.
Dimensions to maximum metal sizes, n.s. = no standard.
**Commercial 0BA nuts are sometimes to ¼″ Whitworth or BSF sizes.
Owing to the difficulty of obtaining imperial size small hexagon bar many nuts are now using 'nearest metric' sizes for the hexagons. In some cases these are above the max. metal sizes given above by about 0·004″.
The diameter of cheese and Csk. head screws is the same as the across flats dimension within 0·004″, and the height of cheese-heads is the same as that of the hex. bolt head.

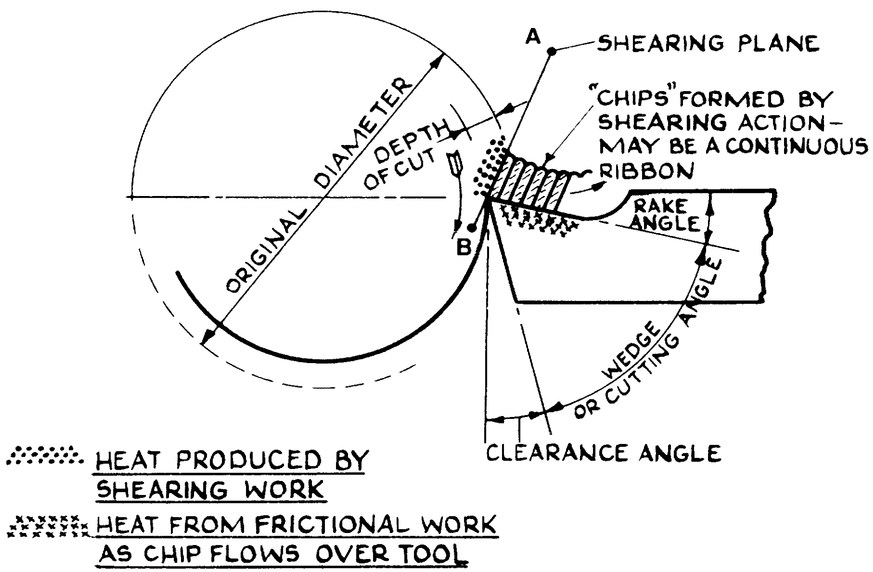
SECTION FIVE
WORKSHOP PRACTICE

TOOL FORMS, CUTTING SPEEDS FOR DRILLING, MILLING & TURNING. GRINDING WHEELS
METRIC REPLACEMENT OF GAUGE AND LETTER DRILLS

CUTTING TOOLS

In general the model engineer has control of three variables in the metal-cutting process: tool shape, cutting speed, and depth × rate of cut. A fourth choice – tool material – is less important, if only because the amateur workshop cannot afford a multitude of tool types, but a choice *may* be made between carbon steel, high-speed steel, and occasionally tungsten carbide.

The diagram below shows the generally accepted mechanism of metal removal with a single-point (e.g. lathe or shaper) tool. It makes no difference whether the work moves past the tool, or the tool past the work. The relative movement in the direction of the arrow is determined by the *cutting speed* and the point of the tool is held immersed in the body of the workpiece. The top face of the tool exerts a force on the parent metal sufficient to cause this to *shear* across the plane A-B, and the resultant 'chip' (which may be a continuous ribbon) slides across the top face of the tool. The force involved is considerable – of the order of 120 tons/sq. in. of area of cut for mild steel, and perhaps 80 tons/sq. in. for medium cast iron.

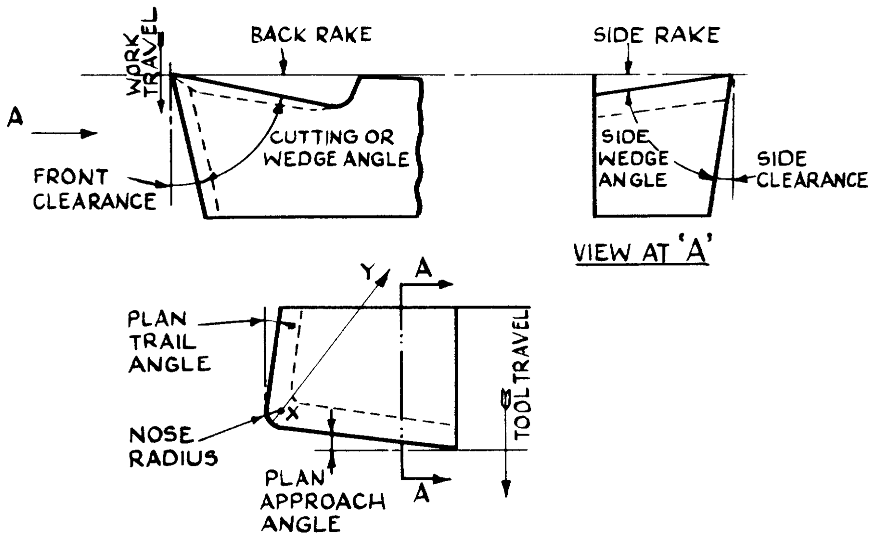


The work done appears as heat at the tool point, this work being the cutting force multiplied by the cutting speed. In addition, the top and front faces are subject to abrasion, and the very point of the tool to a crushing stress. The ability of the tool to resist abrasion and breakdown depends on its *hot hardness*; carbon tool steel is actually harder than HSS at room temperature, but at high rates of metal removal a high-speed steel tool retains its hardness at the higher temperature whereas carbon steel does not.

Tool shape

The smaller the 'cutting angle' in the diagram on page 5.1 the less will be the cutting force for a given rate of feed and depth of cut, and as the chip will slide easier over the top, the less will be the abrasion on the top face. However, the strength of the tool point to resist crushing improves as this angle is *increased*; it follows that the so-called 'correct' tool angles are necessarily a compromise between two criteria of tool failure. With some materials there is another consideration. If the angle of *rake* is large, then there will be a component of the cutting force tending to draw the tool into the work, out of control of the operator. (There is bound to be *some* slack or backlash in the tool-slide feedscrew.) As a result, though such materials (typically the brasses and malleable cast iron) could be machined most economically with a high rake angle (small cutting angle) zero or even negative rake must be used.

Two other factors should be mentioned. First, some materials are prone to 'build up' a false edge on the tools; small, dusty particles of the work material are welded onto the tool by the heat from the cutting forces. This false edge changes the design shape, but, more important, makes it impossible to achieve a good surface finish. (An increase in top rake may reduce this tendency.) Second, the surface finish of the *tool* is important; a smooth surface both reduces the friction as the chip slides over the face, and also improves the finish on the workpiece – under ideal conditions this can only be a replica of the finish on the tool.



PRINCIPAL ANGLES OF A LATHE TOOL

Finally, it must be remembered that conditions in a manufacturing works are quite different from those in the amateur's workshop. In the first case the economics of the plant demand high rates of metal removal, and the power available at the tool point is considerably greater. The amateur is more concerned with relatively small cuts, has lower power available, and likes a long tool life. This matter is referred to again later, but suffice now to mention that tool forms designed for economic industrial production are not necessarily the best for model-making.

The sketch (p. 5.2) shows the principal angles on a lathe tool. The *front clearance* or relief angle is required to ensure that the tool never rubs on the machined surface. If, as most writers recommend, the tool is set between 1% and 4% of the diameter above centreheight for diameters above $\frac{3}{8}''$ finished, then the *true* clearance will be less than that ground on the tool. The front face often is ground to one angle in the rough, and then fine ground to a slightly smaller one. This reduces the amount of metal to be removed when resharpening. So far as the workpiece is concerned it only 'sees' this second angle. As a general rule, front clearance should be as small as possible, 5° to 9° being typical.

The *top rake* is, in most tools, a compound angle in the direction of the arrow XY made up of the back and side slope of the top of the tool. (For a knife tool, it is, of course, a single angle.) However, with the depths of cut normal in the modelmaker's workshop it is doubtful whether side-rake has any effect; for deep cuts when roughing a knife-tool is usual (and is easier on the machine) and even with the 'general purpose' roughing tool shown later the side rake may be made quite small. The rake, in conjunction with the front clearance, forms the 'tool angle' (or 'wedge' angle) and this is the angle with which the workpiece is concerned. *Side clearance* is necessary on tools which cut sideways, and also on the sides of *parting tools* to avoid jamming. Side clearance is usually the same as front clearance for roughing, smaller for finishing (except on knife tools), and perhaps $\frac{1}{2}$ to 1 degree on parting tools.

The table below gives some suggested angles for HSS – those for carbon steel may be a degree or so larger, giving a sharper tool angle.

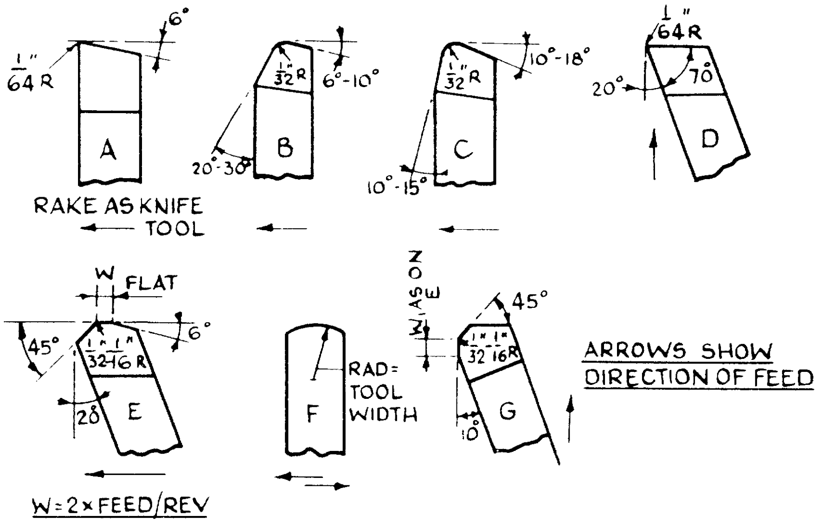
		ANGLES IN DEGREES			
Workpiece material	*Class of work*	*Back (4) rake*	*Side rake*	*Front (4) clearance*	*Side clearance*
FCMS & BDMS	Rough	6–10	16	5–9	5–9
	Finish	14–22	0	5–9	0–2
	Parting	15	0	5–9	$\frac{1}{2}$–$1\frac{1}{2}$
Tough Steel	Rough	6	12	5–9	5–9
	Finish	12	0	5–9	0–2
	Parting	5–10	0	5–9	1
Cast Iron	Rough	8	12	5–9	5–9
	Finish	6–10	0	5–9	0–2
	Parting	5	0	5–9	$\frac{1}{2}$–$1\frac{1}{2}$
Common Brass & Bronze, FC Leaded PB	Rough	0	0–3	6	5
	Finish	0	0	6	6
	Parting	0 to −2*	0	5–10	1

Workpiece material	*Class of work*	*ANGLES IN DEGREES* Back (4) rake	*Side rake*	*Front (4) clearance*	*Side clearance*
70/30 &	Rough	5	5	9	9
Ductile brasses,	Finish	2–6	2–6	6–15	9
Naval brass	Parting (1)	3	0	9	$\frac{1}{2}$–$1\frac{1}{2}$
Copper, phos.	Rough	8–18	16–25	5–9	5–9
Bronze,	Finish	10–20	16–25	5–9	0–2
Al. bronze	Parting (1)	10	0	5–9	0
Monel	Rough	4–8	10–14	5–9	5–9
	Finish	15–20	0	5–9	0–2
	Parting	6	0	5–9	$\frac{1}{2}$–1
Aluminium	Rough	8	15–22	5–9	5–9
Light alloy (2)	Finish	8	15–22	5–9	0–2
	Parting (1)	10	0	5–9	$\frac{1}{2}$–$1\frac{1}{2}$
Nickel	Rough/finish	10–20	20–30	12	10
Silver (3)	Parting	7–15	0	5–10	2–3

Notes:
*Negative rake is desirable on free-chip forming metals.
(1) Reduce rake if chips curl up in the kerf.
(2) Range of Al. alloys is so vast that only a guide can be given.
(3) 'Leaded' or free cutting nickel silvers – as for FC brass.
(4) This will be the *side* rake or clearance for a knife tool.

Tool planforms

There are still some books on turning which continue to show tool shapes designed for use in treadle lathes, but it is a mistake to replace these with planforms found in toolmakers catalogues. The latter are intended for the arduous work in production plants, and the power available to the average $3\frac{1}{2}''$ or $4''$ lathe is such that full advantage cannot be taken of them. For normal mild steel, about $1\frac{1}{2}$ cu. in. (25 cc) of metal can be removed per minute per hp available at the chuck, so that the most that can be expected on a model maker's machine is about $\frac{1}{2}$ cu. in./min (8 cc) at 100 ft/min; that is a cut of $\frac{1}{8}''$ (3 mm) deep at a feedrate of ·003″/rev (0·08 mm)! The shapes shown in the diagram opposite have been found to give good results in both carbon and HSS tools. For removing a great deal of metal I use a tool of shape 'A' at a cutting speed about 75% of normal, fed as fast as the machine will stand using the leadscrew handwheel (not power feed) with cuts of up to 0·3″ (7 mm) deep in drawn mild steel. The finish is poor, but this shape gives the maximum possible rate of metal removal. Shape B is used for normal roughing, under power feed, and gives a tolerable finish. For copper alloys, the lead angle is reduced as at 'C'. 'D' is a tool-form used for facing work – it is, of course, cocked over in the toolholder to give the 20° clearance indicated.



Finishing cuts are better taken with a relatively coarse feed and a flat or large radius face to the tool, than by the use of a very fine feed. Finishing cuts of 3 to 5 thousandths of an inch (say 0·08–0·12 mm) impose very small loads but, especially on cast materials, abrasive wear may cause the tool to cease cutting full depth; this especially applies when boring; at a feed of ·002″/rev (0·05 mm) often recommended, the tool will travel nearly 400 *feet* (120 metres) in boring a cylinder 1½″ dia × 2″ long (36 × 50 mm). Provided the tool is honed really sharp, angles are correct, and the tool itself is as stiff as possible to avoid chatter, it is better to use a feed of 6 thou/rev or even more (1½ mm/rev) and a profile as either 'E' or 'F' above. 'E' is preferred, as the actual cutting is done on a small radius and a more positive chip is formed. If chatter is experienced the speed may be reduced, but it is preferable to stiffen up the tool. For finish cuts on slender work a finer feed and higher speed may be essential. Shape 'G' is used for finish facing cuts.

These shapes are, of course, for general turning. The classical 'bent' tools are necessary for getting into awkward corners, but even here I tend to follow the shapes shown unless a definite radius or form is needed.

CUTTING SPEEDS

It cannot be emphasised too strongly that the *commercial* cutting speeds recommended in production engineering handbooks are quite unsuitable for model engineers, especially those quoted for roughing out. True, the cuts envisaged commercially are much greater, but these speeds are for an *economic* tool life – one which will give the lowest cost of production taking into account fixed charges, tool-changing time, sharpening costs, and overheads. To give an example, the economic tool life (HSS) cutting steel or cast iron in 'ordinary' class of work is about 40 minutes, and for tool-room work may be 2 hours. To increase the latter figure to (say) 8 hours might require a reduction in cutting speed of about 15%; if the cutting speed were increased by as little as 25% the life would be reduced to about 15 minutes. As might be expected, the effect is more marked with carbon steel tools than with HSS, and carbide tools do better. This example is for lathework: the 'economic life' for a milling cutter is longer, due to the higher sharpening costs, about

6 hours – but as some model engineers' endmills may have gone for some *years* between sharpening this isn't much help!

The speeds given in the table which follow are a compromise between reasonable times between resharpening and a reasonable rate of metal removal. Those who have good tool-grinding equipment, and who use their machine more than 12 hours/week may care to increase them a trifle; others may care to experiment with lower speeds. The figures given for 'finishing cuts' assume that the tool has been rehoned, if not reground, before starting work. Cutting speeds are not very critical, and at the depths of cut used by many amateurs the 'finishing' speeds could well be used for all general work. The speeds are related to arbitrary 'machinability factors', listed in Table A, below. Look up the material in Table A, and then refer to Table B, opposite.

MACHINABILITY GROUPS

Table A

Material	*Group*
Aluminium	E
Bakelite	C
Brass, common	D
Brass, free-cutting	D*
Bronze	C
Cast iron, ordinary	C
Cast iron, hard	A
Cast (tool) steel, annealed	B
Copper (drawn)	C
Duralumin	E
Gunmetal	C
Hard cast iron	A
Malleable iron	B
Monel metal	B
Muntz metal	D
Nickel silver (German silver)	C*
Phos. bronze	C
Steel, BDMS or black	C
Steel, cast (tool)	B
Steel, freecutting	D
Steel, 18–8 stainless	B
Steel, high tensile (unheattreated)	B
Silver steel (un-hardened)	B
Tufnol	E[1]
Plastics	D × 2[2]

Notes:

*'Leaded' or free-cutting varieties, speeds may be increased by 75%.

[1] Use 2° to 5° *negative* top rake for finishing.

[2] With many plastics speeds may have to be reduced to avoid re-welding of chips to the surface.

CENTRE-LATHE CUTTING SPEEDS – High Speed Steel

Table B

Machinability group				*A*		*B*		*C*		*D*		*E*	
Class of work				*R*	*F*	*R*	*F*	*R*	*F*	*R*	*F*	*R*	*F*
Dia. in.	**RPM at 100 ft/min*	*65 ft min*	*Dia. mm*										
$\frac{1}{8}$	3056	1986	3	610	760	1225	1525	2000	2300	3600	4275	—	—
$\frac{1}{4}$	1528	993	6	305	380	610	765	995	1150	1830	2140	—	—
$\frac{3}{8}$	1018	662	10	205	250	405	500	660	760	1220	1425	—	—
$\frac{1}{2}$	764	497	12	175	190	305	380	500	570	920	1070	—	—
$\frac{5}{8}$	611	397	16	125	150	245	305	400	460	730	855	3700	5000
$\frac{3}{4}$	509	330	20	100	127	205	250	330	380	610	715	3050	4000
1	382	248	25	75	95	150	190	250	285	460	535	2300	3000
$1\frac{1}{4}$	306	200	32	60	75	125	190	200	230	360	430	1840	2500
$1\frac{1}{2}$	256	166	40	50	64	100	125	165	190	310	360	1540	2000
$1\frac{3}{4}$	218	142	45	45	55	85	110	140	165	260	305	1300	1700
2	191	124	50	38	48	76	95	125	145	230	270	1150	1530
3	127	82	75	26	32	51	63	82	95	150	180	760	1000
4	95	62	100	19	24	38	48	62	71	114	135	570	760
5	76	50	125	—	—	30	38	50	57	91	105	460	600
6	63	41	150	—	—	25	32	41	47	75	75	380	500
7	56	36	180	—	—	22	28	36	42	67	78	335	450
8	48	31	200	—	—	—	—	31	36	58	67	290	380
9	42	27	225	—	—	—	—	27	32	50	59	250	340
10	38	25	250	—	—	—	—	25	28	45	53	225	300

Carbon steel tools – use $\frac{2}{3}$ of the above; *Carbide* tools – increase by 50% to 75%. *Parting off*. Reduce speeds by 75%; *Form tools*. Reduce by 50–75% depending on width. Speeds assume $\frac{5}{16}''$ or $\frac{3}{8}''$ sq. shank tools projecting from clamp by twice the width. *100 ft/m = 30 m/min. 65 ft/min = 20 m/min. (Metre/min = 0·305 ft/min.)

DRILLS

For drills from 0·1 mm up to 1·5 mm 'horological' drills may be preferred to 'jobbers' as they have a shorter and stiffer flute length. Those below 1·0 mm all have a standard 1·0 mm dia. shank. The table on page **5**.11 gives the recommended metric replacement for 'number' and 'letter' drills, which are no longer available in the U.K.

I strongly recommend the use of 'stub' length drills for tapped holes. They are much stiffer than 'jobbers' (normal) length and less prone to wander.

Stock and preferred sizes

From 1 mm dia. up to 3 mm, jobbers straight-shank drills increase in steps of 0·05 mm – e.g. 2·40, 2·45, 2·50, 2·55 mm etc. From 3 mm to 10 mm the steps are 0·1 mm, but with $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ mm also available – e.g. 7·00, 7·10, 7·20, 7·25, 7·30, etc. Above 10 mm it is advisable to refer to the stockists, as though some intermediate sizes may be available they are 'second preference' and more expensive. For example, 10·00, 10·20, 10·50, 10·80, 11·00 are preferred, and cheaper than the available 10·10, 10·25, 10·30, 10·40, 10·60, 10·70, and 10·90 listed as 'second preferences'. As time goes by it is likely that preferred drill sizes will follow the R20-40-80 series – see p. **2**.26.

Larger drills with Morse taper shanks, above 14 mm dia., rise in steps of 0·25 mm to 32 mm, and above this by steps of 0·5 mm to 50 mm and by 1 mm from 50 to 100 mm dia.

Drill points and helix angle

The standard point of 118° included angle and the normal helix or twist is found best for the general run of engineering materials. 'Straight flute' drills, preferred by many for brass, are not readily available, and for occasional use in this material it is usual to stone a very small flat on the cutting edge to reduce the 'top rake' to zero – a normal helix drill tends to walk into brass and similar metals. If much work is done in brass, the use of 'slow helix' drills is recommended. No change in point angle is necessary.

For thin materials the point may be reground to give a wider angle, so that the edge of the flutes engage before the point breaks through. As an alternative a back-up packing of the material being drilled may be set under the workpiece.

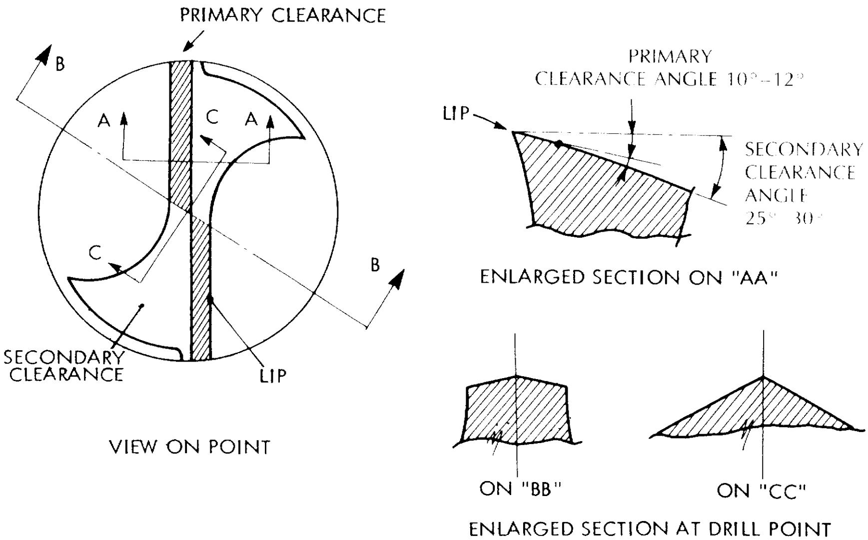
For other materials the following is recommended, but it is clearly unnecessary to obtain special drills for this work unless a great deal of drilling is done in these metals.

Drill point angles

Material	*Point angle*	*Helix*	*Remarks*
Aluminium, copper, light alloy	100°	Quick	For 'production' runs on copper, use a helix between the 'Quick' and the 'Standard'
High tensile steel	130°	Slow	
Cast iron, soft.	90°	Standard	
Medium and close grain	118°	Standard	
Plastics	90°	Standard	For 'production' runs use 'narrow land thin web' to minimise clogging

The four facet point

In this point the clearance is provided by two *flat* facets instead of the conventional cone. The main, or secondary, clearance is 30° but that at the cutting edge is a very narrow flat at 10° angle – see sketch below. This secondary or cutting edge clearance angle is carried right to the centre of the drill, so that there is a *point* on the chisel edge, unlike the pitch roof of the cone ground drill.



The 'Four-facet' drill point.

The result is a marked reduction in drill pressure needed; more important, when using a stub length drill it can be started without a centre-pop. There is also a considerably reduced tendency for even long drills to wander. However, this type of point does need a proper cutter grinder (such as the *Quorn*) and it cannot be sharpened 'off-hand'. Grinding errors are much more serious than with a cone point.

The major limitation in practice is that when used on 'greedy' materials (like brass or cast GM) the drill will tend to 'walk through' the workpiece and, when breaking through, may drag so much as to pull the chuck off its taper mandrel.

SPEEDS FOR DRILLING

The cutting process is more complex and the forces much greater* when drilling than turning, so that speeds are lower. On the other hand, the model engineer's drills are used for minutes rather than the hours at a time found in production plants, so that the following table has been drawn up with very little reduction from speeds used in manufacture. It is assumed that suitable coolant (water soluble oils are preferred for drilling) is used where appropriate. Where a dash '–' is shown the rule is 'as fast as possible'; few amateur's drilling machines run at over 4500 rpm (but see later note for very small drills).

Note: When drilling in the lathe, reduce these speeds by from 25% to 30%, to allow for the difficulty of chip clearance and coolant penetration down the hole. For carbon steel drills, divide by 2.

*A sharp ¼ in. drill requires a thrust of about 80 lb in mild steel; much more when blunt.

SPEEDS IN RPM

Table C

Drill dia.		Machinability group – Table A				
in.	*mm*	*A*	*B*	*C*	*D*	*E*
$\frac{1}{32}$	0·75	4900	—	—	—	—
$\frac{1}{16}$	1·5	2455	4300	—	—	—
$\frac{1}{8}$	3·0	1220	2140	3100	4500	—
$\frac{3}{16}$	4·75	815	1430	2040	3060	4900
$\frac{1}{4}$	6·5	601	1070	1530	2300	3650
$\frac{5}{16}$	8·5	490	855	1220	1800	2950
$\frac{3}{8}$	9·5	410	710	1020	1530	2450
$\frac{7}{16}$	11·0	350	610	870	1300	2100
$\frac{1}{2}$	12·5	310	535	760	1140	1850
$\frac{9}{16}$	14·0	270	475	670	1010	1600
$\frac{5}{8}$	16·0	245	430	610	910	1450
$\frac{11}{16}$	17·5	220	390	550	830	1300
$\frac{3}{4}$	19·0	205	360	500	760	1220
$\frac{7}{8}$	22·0	175	305	430	650	1050
1	25·5	150	265	380	570	900

Monel metal: Some authorities would put this in Group A for drilling.
Stainless steel: Work-hardening varieties – treat as Group A, but keep up the feedrate. Use tallow or turps as lubricant.
Chilled cast iron: Use about half the speed for Group A.

The motor power on many machines may limit the speed on larger drills. No harm will follow using a somewhat slower drill speed – this is preferable to reducing the feedrate. Clear chips frequently on small drills.

MICRO-DRILLS

'As fast as you can' should be applied with care to drills below 1 mm dia., as there is risk of 'whirling' under the drilling pressure. The following figures are suggested by manufacturers for use with high-speed drilling machines. Speeds are in **1000s rpm**.

Drill dia., mm	*0·2*	*0·4*	*0·6*	*0·8*	*1·0*	*1·2*	*1·4*	*1·6*	*1·8*
Aluminium	6·5	12·5	16·0	18·5	20·0	20·0	19·5	18·5	17·5
Brass	6·0	10·7	14·0	15·5	16·0	15·8	15·2	14·4	13·2
GM, bronze	2·5	5·5	8·0	9·7	10·3	11·0	10·7	10·0	9·2
Steel (EN8)	4·0	5·8	8·0	8·3	8·8	8·8	8·0	7·8	7·2
CI	2·1	3·6	5·1	6·0	6·5	6·2	5·8	5·4	

BRITISH STANDARD SIZES SUPERSEDING OBSOLETE DRILL GAUGE AND LETTER SIZES

Drill gauge or letter size	Decimal equivalent in.	RECOMMENDED ALTERNATIVE SIZE mm	RECOMMENDED ALTERNATIVE SIZE Decimal equivalent in.	Drill gauge or letter size	Decimal equivalent in.	RECOMMENDED ALTERNATIVE SIZE mm	RECOMMENDED ALTERNATIVE SIZE Decimal equivalent in.	Drill gauge or letter size	Decimal equivalent in.	RECOMMENDED ALTERNATIVE SIZE mm	RECOMMENDED ALTERNATIVE SIZE Decimal equivalent in.
80	·0135	**·35**	**·0138**	44	·0860	**2·20**	**·0866**	8	·1990	**5·10**	**·2008**
79	·0145	**·38**	**·0150**	43	·0890	**2·25**	**·0886**	7	·2010	**5·10**	**·2008**
78	·0160	**·40**	**·0157**	42	·0935	$\frac{3}{32}$ in.	**·0938**	6	·2040	**5·20**	**·2047**
77	·0180	**·45**	**·0177**	41	·0960	**2·45**	**·0965**	5	·2055	**5·20**	**·2047**
76	·0200	**·50**	**·0197**	40	·0980	**2·50**	**·0984**	4	·2090	**5·30**	**·2087**
75	·0210	**·52**	**·0205**	39	·0995	**2·55**	**·1004**	3	·2130	**5·40**	**·2126**
74	·0225	**·58**	**·0228**	38	·1015	**2·60**	**·1024**	2	·2210	**5·60**	**·2205**
73	·0240	**·60**	**·0236**	37	·1040	**2·65**	**·1043**	1	·2280	**5·80**	**·2283**
72	·0250	**·65**	**·0256**	36	·1065	**2·70**	**·1063**	A	·2340	$\frac{15}{64}$ in.	**·2344**
71	·0260	**·65**	**·0256**	35	·1100	**2·80**	**·1102**	B	·2380	**6·00**	**·2362**
70	·0280	**·70**	**·0276**	34	·1110	**2·80**	**·1102**	C	·2420	**6·10**	**·2402**
69	·0292	**·75**	**·0295**	33	·1130	**2·85**	**·1122**	D	·2460	**6·20**	**·2441**
68	·0310	$\frac{1}{32}$ in.	**·0312**	32	·1160	**2·95**	**·1161**	E	·2500	$\frac{1}{4}$ in.	**·2500**
67	·0320	**·82**	**·0323**	31	·1200	**3·00**	**·1181**	F	·2570	**6·50**	**·2559**
66	·0330	**·85**	**·0335**	30	·1285	**3·30**	**·1299**	G	·2610	**6·60**	**·2598**
65	·0350	**·90**	**·0354**	29	·1360	**3·50**	**·1378**	H	·2660	$\frac{17}{64}$ in.	**·2656**
64	·0360	**·92**	**·0362**	28	·1405	$\frac{9}{64}$ in.	**·1406**	I	·2720	**6·90**	**·2717**
63	·0370	**·95**	**·0374**	27	·1440	**3·70**	**·1457**	J	·2770	**7·00**	**·2756**
62	·0380	**·98**	**·0386**	26	·1470	**3·70**	**·1457**	K	·2810	$\frac{9}{32}$ in.	**·2812**
61	·0390	**1·00**	**·0394**	25	·1495	**3·80**	**·1496**	L	·2900	**7·40**	**·2913**
60	·0400	**1·00**	**·0394**	24	·1520	**3·90**	**·1535**	M	·2950	**7·50**	**·2953**
59	·0410	**1·05**	**·0413**	23	·1540	**3·90**	**·1535**	N	·3020	**7·70**	**·3031**
58	·0420	**1·05**	**·0413**	22	·1570	**4·00**	**·1575**	O	·3160	**8·00**	**·3150**
57	·0430	**1·10**	**·0433**	21	·1590	**4·00**	**·1575**	P	·3230	**8·20**	**·3228**
56	·0465	$\frac{3}{64}$ in.	**·0469**	20	·1610	**4·10**	**·1614**	Q	·3320	**8·40**	**·3307**
55	·0520	**1·30**	**·0512**	19	·1660	**4·20**	**·1654**	R	·3390	**8·60**	**·3386**
54	·0550	**1·40**	**·0551**	18	·1695	**4·30**	**·1693**	S	·3480	**8·80**	**·3465**
53	·0595	**1·50**	**·0591**	17	·1730	**4·40**	**·1732**	T	·3580	**9·10**	**·3583**
52	·0635	**1·60**	**·0630**	16	·1770	**4·50**	**·1772**	U	·3680	**9·30**	**·3661**
51	·0670	**1·70**	**·0669**	15	·1800	**4·60**	**·1811**	V	·3770	$\frac{3}{8}$ in.	**·3750**
50	·0700	**1·80**	**·0709**	14	·1820	**4·60**	**·1811**	W	·3860	**9·80**	**·3858**
49	·0730	**1·85**	**·0728**	13	·1850	**4·70**	**·1850**	X	·3970	**10·10**	**·3976**
48	·0760	**1·95**	**·0768**	12	·1890	**4·80**	**·1890**	Y	·4040	**10·30**	**·4055**
47	·0785	**2·00**	**·0787**	11	·1910	**4·90**	**·1929**	Z	·4130	**10·50**	**·4134**
46	·0810	**2·05**	**·0807**	10	·1935	**4·90**	**·1929**				
45	·0820	**2·10**	**·0827**	9	·1960	**5·00**	**·1969**				

DIMENSIONS OF COMBINATION CENTRE DRILLS

Inch sizes

Ref. no.	*S1*	*S2*	*S3*	*S4*	*S5*	*S6*	*S7*
Body dia., in.	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{3}{4}$
Drill dia., in.	$\frac{3}{64}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{5}{16}$
Point length – max	$\frac{5}{64}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{9}{32}$	$\frac{11}{32}$	$\frac{15}{32}$
min	$\frac{1}{16}$	$\frac{5}{64}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{13}{32}$

Metric sizes

Drill dia., mm	1·00	1·25	1·60	2·00	2·50	3·15	4·00	5·00	6·3
Body dia., mm	3·15	3·15	4·00	5·00	6·30	8·00	10·0	12·5	16·0

Drill points are ground to 118° included, and the countersink is 60° inc.

MILLING

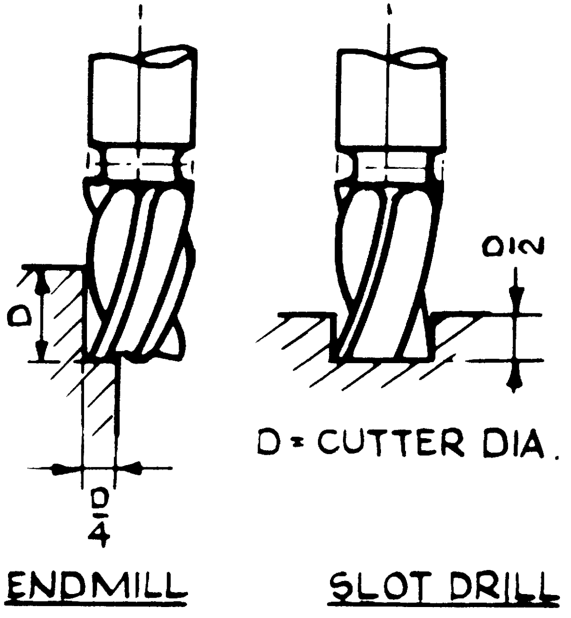
The process of chip formation in milling is totally different from that in turning. The cut is 'interrupted', takes place only during part of the revolution, and the chip thickness varies from beginning to end of the cut. This means that cutting speed cannot be divorced from 'tooth load' – the feedrate in inches/min (or m/min) divided by the number of teeth on the cutter. This latter is a variable, depending on the type of cutter.

In the case of the amateur the matter is further complicated by the fact that he is usually milling in the lathe, and with a not too rigid set up at that – especially when using side-and-face cutters on an arbor between centres. He has far less power available than the industrial user, for whom the cutters are designed. He uses endmills as facing cutters, a duty for which they are not intended; he faces some problem in keeping a steady feed at the low rates often needed; and last but not least, he seldom possesses a cutter grinder so that cutter life is a paramount consideration and his tools are seldom as sharp as they ought to be.

This being so, it is difficult to generalise in a fashion which would be useful to all modellers alike. All that can be done is to offer a basis around which experiment can be made, together with a few observations which may, perhaps, help in overcoming common difficulties. As the endmill is the most commonly used tool, the following remarks apply to these, and to the apparently similar slot drill.

Endmills are designed for *profiling*, and are at their best in the situation shown in the sketch below. The maximum width of cut is one quarter of the diameter of the cutter, and depth of cut is up to one diameter deep. This depth should not be less than about 0·006″ and the width of cut should not be greater than that which engages two teeth at once. A better result will be obtained, with probably longer cutter life, if several cuts D/4 wide are made rather than fewer cuts the full width of the tool.

If an endmill *is* used as a facing cutter, even with a very light depth of cut, then the corner may break down and rubbing will result. It is preferable to stone a 45° bevel or chamfer on the sharp corner, ensuring that clearance or relief is maintained, and to employ a *depth* of cut not less than 0·006″ even in finishing; more is better. The bevel need not exceed $\frac{1}{32}$″ even on a large cutter.



Endmills are *not* suitable for cutting slots equal in width to the cutter diameter. For this duty a *slot drill* should be used; this has the advantage that, unlike the endmill, it may be fed axially down into the work so that slots with blind ends may be cut. If slot drills are not available, then each side of the slot should be machined with a smaller cutter.

So much depends on the condition of the machine, the method of holding the cutter, and the nature of the work-holding set-up that experiment is essential. As with any other cutting tool (but more so in milling), the edges must *cut* and not rub; in general the depth of cut should be as high as the machine and cutter can stand (within the limits shown in the sketch above) and the feedrate kept *up* to those shown in the table. Keeping the feedrate steady is difficult but *must* be achieved for satisfactory results.

The tables below are based on those recommended by the Clarkson company for standard HSS 'Autolock' cutters and slot drills, the former with a cut as in the sketch, and the latter with a depth of cut equal to D/4, cutting full width. Experiments around these speeds and feeds should form a fair starting point.

Note that feedrates should be halved for long series cutters.

Endmills (4-flute)

	Machinability, from Table A									
	B			C			D or E			
Dia. in.	RPM	Feed/min in.	Feed/min mm	RPM	Feed/min in.	Feed/min mm	RPM	Feed/min in.	Feed/min mm	Dia. mm
$\frac{1}{16}$	2000	$\frac{3}{16}$	4	3800	$\frac{1}{2}$	12	*	*	*	1·5
$\frac{1}{8}$	1000	$\frac{3}{8}$	9	1920	1	25	8100*	*	*	3·0
$\frac{3}{16}$	700	$\frac{5}{8}$	15	1280	$1\frac{3}{8}$	35	5350*	*	*	4·5
$\frac{1}{4}$	500	$\frac{7}{8}$	22	960	$1\frac{3}{4}$	44	4000*	8*	200*	6
$\frac{3}{8}$	350	$1\frac{1}{8}$	28	640	$2\frac{5}{8}$	65	2700	11	280	8
$\frac{1}{2}$	250	2	50	480	3	75	2000	13	330	12·5
$\frac{5}{8}$	200	$1\frac{3}{8}$	35	385	$3\frac{1}{8}$	80	1600	13	330	16
$\frac{3}{4}$	150	$1\frac{1}{2}$	38	320	$3\frac{1}{2}$	88	1350	14	350	20
1	130	$1\frac{5}{8}$	40	240	$2\frac{7}{8}$	72	1000	12	300	25

*Cutter speed, maximum available, up to speeds given, with a 'tooth load' equal to cutter dia./1000. 'Feedrate' is then tooth load × RPM × no. of teeth.
For milling, treat phosphor bronze as a group B material, and F.C. steel as C.

Slot drills (2-flute)

	Machinability, from Table B									
	B			C			D or E			
Dia. in.	RPM	Feed/min in.	Feed/min mm	RPM	Feed/min in.	Feed/min mm	RPM	Feed/min in.	Feed/min mm	Dia. mm
$\frac{1}{16}$	2000	$\frac{1}{4}$	6	*	*	*	*	*	*	1·5
$\frac{1}{8}$	1200	$\frac{1}{2}$	12	3200	$1\frac{5}{8}$	40	*	*	*	3
$\frac{3}{16}$	800	$\frac{3}{4}$	18	2100	$2\frac{1}{8}$	55	*	*	*	4·5
$\frac{1}{4}$	600	1	25	1600	3	75	4500	9	225	6
$\frac{5}{16}$	480	$1\frac{1}{4}$	30	1200	4	100	3500	12	300	8
$\frac{3}{8}$	400	$1\frac{1}{4}$	30	1000	4	100	3000	12	300	9·5
$\frac{7}{16}$	350	$1\frac{1}{2}$	38	900	$4\frac{1}{2}$	115	2500	13	330	11
$\frac{1}{2}$	300	$1\frac{1}{2}$	38	800	$5\frac{1}{4}$	135	2200	15	380	12·5
$\frac{5}{8}$	240	$1\frac{1}{2}$	38	600	$5\frac{1}{4}$	135	1800	15	380	16

*As for endmills.
Phosphor bronze, treat as a group B material, F.C. steel as C.

Three-flute ('throwaway') cutters
These cutters may be run at the same cutting speeds as regular cutters, using tooth loads about D/125. Remember that they have *three* flutes when calculating the feedrate.

Side and face cutters

Cuts and feed depend entirely on the length and stiffness of the arbor especially if used in a lathe between centres. For cutting speeds with HSS cutters, try 33 ft/min for group B material, 60 ft/min for group C, and 80 ft/min for group D. (10, 18 and 30 metres/min.)

For single-point carbon steel fly-cutters, use $\frac{1}{3}$ to $\frac{1}{2}$ the cutting speed suggested in the table for turning.

CUTTING FLUIDS

This is a complicated subject, and only a bare outline is possible in this handbook. But most model engineers have at least some experience of the use of cutting fluids, and it is hoped that the notes which follow will supplement that experience, and help them to overcome any problems which may have arisen. The amateur machinist uses such a wide variety of materials, cut with many types of tool under such differing conditions that experiment is always advisable.

Cutting fluids serve two main purposes. First, as a coolant, to remove heat both from the tool point and the workpiece. Second, to lubricate the flow of the chips across the tool face, reducing friction and the consequent heating, and to prevent the formation of a built-up edge.

Water is the ideal coolant, and in former times soapy water was used, the soap acting as a mild lubricant. The obvious disadvantage was the inevitable rusting which occurred. The alternative, 'fatty' oils such as lard or whale oil, were excellent lubricants but had poor cooling properties. Both have now been replaced by synthetic products, of two main types.

(1) **Soluble oils**. This is a misnomer, as the oil is not 'dissolved' in the water, but dispersed in minute droplets as an *emulsion*. The oil content effects the necessary lubrication, and the main body, the water, provides the cooling. Dilutions depend on the type and the duty, ranging from 15/1 to 30/1, with 20/1 being typical. The overall cost is low, but as the emulsion can suffer from age, overheating, and contamination with tramp (machine lubricating) oil, replacement is necessary fairly often.

(2) **Neat cutting oils**. These are compounded oils, often with 'extreme pressure' additives. The choice is large, ranging from those with very high lubricating qualities (e.g. for broaching, machine tapping etc) to others blended for their cooling capacity. Though more costly than soluble oils they are very stable, and can with care be filtered and reclaimed for repeated use over long periods.

Choices

There is little doubt that the 'soluble' oils are the most effective for the relatively light cutting done by the amateur, whether for turning, drilling or milling. Emulsions will give reasonable tool life and a good finish. But there are problems. The first is the washing away of machine lubricants, on bedways, screws and bearings. Constant re-oiling is necessary.

The second is corrosion. This is *not* caused by the water content and is a quite different phenomenon from 'rust'. It is not too serious a matter for industry, where machines are in constant use but in the model engineer's workshop, where machines may stand for days (or perhaps weeks) unused, it can be troublesome.

The cause is bacterial action. (Not the bacteria associated with disease, by the way!) In the presence of air these bacteria are inactive. They are 'anaerobic'. But in the absence of oxygen they multiply, feeding on the oil content, and in so doing release corrosive products which produce, initially, black stains, and ultimately cause pitting. Drops on the machine bed do no harm, as oxygen is present, but any trapped under a machine vice, in a saddle oilway or, worst of all, soaked into a felt wiper, will very quickly set up stains or pitting.

This *cannot* be overcome by using a stronger emulsion; indeed, this may make matters worse, as it is the oil content which provides the nutrient! The *only* solution is rigorous hygiene, even to the extent of removing, washing and drying felt wipers each time the machine is used.

The alternative is to use one of the neat cutting oils – or possibly a variety for different materials; those designed for 'cooling' when roughing mild steel etc, but EP (extreme pressure) blends for tougher materials and stainless steel, and to counter the extra cost by reclaiming, filtering and cleaning the oil for re-use. Many practitioners who have fitted pump circulation systems settle for a general purpose neat oil here, but have on standby a small drip-feed applicator for special cases or for soluble oil.

Application

The ideal is a constant feed to the work – not necessarily a 'flood' which can, when working at high speed, drown the operator. If a smaller flow is used then it must be applied to the *chip*, at the point where it leaves the parent metal and slides over the tool point. (Ideally there would be a fine, high power jet *upwards* between the clearance face of the tool and the chip.) A drip onto the top of the workpiece is almost useless. In the case of a parting tool used in the rear tool-post it is imperative that the kerf be kept full, so that oil is carried round to the tool point.

When sintered carbide tools are used the coolant must be either full flood or none at all. Sporadic cooling from 'drips' will set up thermal shock at the speeds (and, hence, temperatures) at which such tooling is, or should be, run. The result will be premature failure of the cutting edge due to chipping.

Selection

The variety of blends of both types of fluid is now so great that no general guidance is possible in a 'handbook'. However, the oil manufacturers and dealers are always happy to advise and can supply data sheets. It is, however, important, to be clear about your requirements. (Perhaps the best guidance to the supplier is the machine motor power!) The fact that once a month a high alloy steel is machined is perhaps best met by using a general purpose fluid, and to mix in a little EP additive (obtainable now in $\frac{1}{2}$ litre bottles or in a spray can) when needed.

General

When mixing emulsions, oil must always be added to the water, *never* the reverse. It should be poured in slowly, with constant agitation. Tap water should be suitable, but if it is very hard, best boiled beforehand. (Mixing should be done cold.)

Disposal of both soluble and neat oils in small quantities should not be a problem, but it is unwise to use the domestic drain for gallon quantities. The local council will have a special disposal unit, and the waste oil should be taken there. No oil of any type should be disposed of into a septic tank.

Hygiene

All oils, lubricating, fuel or cutting, are liable to cause skin irritation, and reasonable precautions should be taken. Operators liable to dermatitis should use barrier cream or gloves, preferably under medical advice. Irritation on the tender skin between the fingers is an indication of possible allergy. Although overalls *can* be cleaned successfully in modern laundrette machines the oil manufacturers recommend dry-cleaning for heavily contaminated clothing. The use of splash guards on machines is an obvious precaution.

WHEELS FOR TOOL-GRINDERS

In general, hard materials should be ground on a relatively soft wheel, and vice-versa. Hard and brittle materials require a finer grade of wheel than softer or more ductile materials. Aluminium oxide is used for the majority of steels; silicon carbide (of appropriate grade) is better for cast iron, non-ferrous materials and, of course, for very hard alloys such as tungsten carbide.

The most frequently used designation for wheel specifications is that of the American Standards Association, as follows.

Prefix	A number, often absent, being the maker's abrasive record.
First letter	Type of abrasive: A for aluminium oxide, C for silicon carbide, GC green silicon carbide.
A number	This is the grain size, in grits per lineal inch. 10 to 24, very coarse; 30–60, medium; 70–180, fine; and 220–240, very fine. (Grades up to 400 can be had to order.)
Second letter	Hardness grade of the grit; ranging from A = very soft, through LMNOP = medium to Z = very hard.
A number	(Not always present) indicates the 'openness' of the structure, 1 = denser to 20+ = open.
3rd letter	Type of bonding: V = Vitrified, S = Silicate, R = Rubber, B = Resinoid.
Final number	May be absent – the maker's private marking.

Thus 58–A54–L8–V1306 means maker's grit bin no. 58, aluminium oxide, no. 54 grit, medium hardness, medium dense face, vitrified bond, maker's catalogue number 1306.

The Carborundum Company have kindly supplied the following data on suitable wheels for model engineers' tool-grinders, assumed to be 6″ wheels running at 2545 rpm or thereabouts.

For single-ended grinders,	a general purpose wheel	A46–Q5–V30W
For DE grinders.	One wheel coarse	A36–Q5–V30W
	One wheel fine	A60–O5–V30W
	or, a wheel for finer work	A100–O5–V30W
For carbide tools.	General purpose	GC80–K11–VR
	ditto (DE grinder) coarse	GC60–L11–VR
	ditto (DE grinder) fine	GC120–J11–VR

Those using home-constructed grinders should note that every wheel is marked with a 'maximum safe speed'. This should *on no account* be exceeded, and special care should be taken with small wheels mounted on series wound or 'universal' motors, the no-load speed of which can be very high indeed. This type of motor is *not* recommended.

The *only* safe speed for second-hand wheels of unknown provenance is *zero*.

Surface finish

'CLA' (Centreline Average) figures are quoted in micro-inches or in 'microns' (0·001 mm. 1 $\mu = 0{\cdot}000039''$), indicating the average height of the protuberances above a mean surface drawn through them. Typical figures (in micro-inch) are:

Hacksawn	350–850	Offhand grinding	80–250
Normal turning	30–250	Machine grinding	20–80
Fine turning	20–35	Honing	5–25
Scraping	200–300	Polishing	$\frac{1}{2}$–5

Shaft/hole fits

The following figures indicate the difference between hole and shaft size needed for the named classes of fit. These apply to shafts from $\frac{1}{8}''$ to $2''$ dia. Where the 'hole' is in a wheel boss some consideration must be given to the strength of this – a test assembly is advised. Normally the hole should be made dead to size and the shaft diameter adjusted to get the fit desired. The figures are in thousandths of an inch per inch of diameter, to which must be added the constant 'C'. Thus for a 1″ push fit the shaft must be $0{\cdot}35 + 0{\cdot}15 = 0{\cdot}5$ thousandths smaller than the hole. ('+' means the shaft is larger and '–', smaller than the hole.) 'C' may be converted directly to mm (divide by 25 is near enough) and 'thou/in.' = micron/mm.

Fit	*'C' (0·001″)*	*thou/in.*
Shrink	+0·5	+1·5
Force	+0·5	+0·75
Drive	+0·3	+0·45
Wheel keying	0	0
Push	−0·15	−0·35
Slide	−0·3	−0·45

Fit	*'C' (0·001″)*	*thou/in.*
Prec. run	−0·5	−0·65
Close run	−0·6	−0·8
Normal run	−1·0	−1·5
Easy run	$-1\frac{1}{2}$	$-2\frac{1}{4}$
Small clearance	−2	−3
Large clearance	−3	−5

An allowance for thermal expansion must be made on engine pistons.

SECTION SIX
METAL JOINING
BRAZING, SOLDERING AND ADHESIVES

BRAZING & SOFT SOLDERING

In both cases a filler material is used which has a melting point lower than that of the workpiece. This filler usually forms an alloy with the parent metal to create the bond, but in some cases a more complicated intercrystalline penetration is involved. Provided the joint is reasonably close, the strength of the joint can reach as much as twice that of the solder or filler material.

The filler material is almost always an alloy, and it is important to appreciate that alloys have seldom an *exact* melting point. If two metals are combined in alloy form the melting point will vary according to the composition, and will be *below* that of either constituent. At one particular proportion of the two materials this melting point will be at its lowest – this is known as the 'eutectic' mixture. At any other composition the alloy is likely to be in the form of a matrix of this eutectic with the remainder of the material dispersed therein – perhaps as secondary alloys. When heated, this eutectic content will melt first, the remainder at a higher temperature, or at a series of higher temperatures. Not until the highest of these is exceeded will the alloy be entirely liquid. Alloys containing more than two elements behave in a similar, but more complex fashion; there may, for example, be two different eutectics present.

For this reason most soldering and brazing alloys are quoted as having a 'melting range'. The alloy will be solid at temperatures below the lowest of these – hence the name 'Solidus' and entirely liquid above the higher – the 'Liquidus'. This point is important, for if a complex alloy with a wide melting range is heated *slowly* the eutectic may melt and migrate into the joint gap whilst the remainder is still solid or pasty. The result is at best an unsightly joint but more usually a weak and porous one.

On the other hand, advantage may be taken of this property in some cases; the most well-known being the plumber's 'wiped joint' – because of its relatively wide melting range the solder can be formed whilst pasty by 'wiping' with a moleskin or wirebrush. With hard solders, the wider melting range alloys will form fillets, whilst the close range ones form only traces of such.

Step brazing

Because the filler alloy forms a further alloy with the work it is often possible to carry out a second brazing operation without destroying a previous one, but only if the joint gaps are small – and provided the previous joint is refluxed. However, by selecting a *series* of alloys with progressively lower melting ranges it is possible to perform three or even more jointing operations in succession. If the liquidus temperature of the first-used is higher than the solidus of the second, and so on, then provided care is taken when heating there is no risk of remelting the previous joint. Manufacturers now offer ranges of such alloys specifically for this kind of duty. See *Soldering and Brazing* (Workshop Practice Series No. 9. ISBN 0 85242 845 6 Nexus Special Interests).

Flux

The surface alloying process can only take place on chemically clean metal. It is the function of the flux first to react with and remove any surface contamination, and thereafter to prevent the work (and the jointing alloy) from oxidising under the high temperature. Apart from a few fluxes used in tin-lead soldering (such as resin) the majority of fluxes are chemically active when hot, and most will decompose to an inactive state if overheated or, more important, if heated for too long. The latter point is another reason for the universal rule that heating should be rapid. It is therefore important to follow the maker's advice when selecting the flux. In some cases the nature of the parent metal will demand a special flux; one suitable for brass or even mild steel may not react with the oxides formed on stainless steel. The flux is usually applied in the form of a paste with water, and the well-known 'Easyflo' flux is, in fact, sold in this form if required. Most fluxes give off offensive, if not poisonous, fumes, and due regard must be paid to ventilation if prolonged work is envisaged.

Capillary action

It cannot be emphasised too strongly that both soft soldering and brazing are *capillary* processes; the alloy penetrates the joint in the same way that sap runs up a tree – by virtue of the surface tension of the fluid. It follows that *there must be a gap* if a proper joint is to be made. Alloy Ag1 will penetrate a gap of 0·001″ (0·025 mm) but most need twice this figure and a few even more. A very common cause of faulty joints is the use of tight riveting to hold parts together before brazing, and this must be watched. On the other hand, whilst most alloys will fill gaps up to 0·006″ (0·15 mm) the 'easy flowing' range Ag1 and Ag2 are unhappy over 0·004″ (0·1 mm). The fillet-forming step-brazing alloys Ag17, 20 and 21 will tolerate gaps up to 0·008″ (0·2 mm). Where gaps are irregular it is prudent to fill the wider parts with suitable shim material to reduce the effective gap, but a total absence of gap must be avoided.

Detailed specifications, strengths and limiting joint gaps can be found in *Soldering and Brazing*, Workshop Series No. 9, published by Nexus Special Interests.

Brazing alloys

The original 'brazing spelter', used for thousands of years, is seldom employed these days, having been replaced by the silver-bearing alloys. However, it is worth remembering that satisfactory joints *can* be made with common 50/50 brass (using borax as a flux, or Tenacity) provided the work will withstand the temperature of about 875°C. 60/40 brass may be used for steel or copper. However, no data is given on these spelters, as it is unlikely they would be used by the model engineer except in an emergency.

The table opposite is extracted, with permission, from BS 1845/1984, but also includes those deleted from the 1977 standard which are still available. There are, however, many others (Johnson Matthey Ltd list over 30) which conform to the continental DIN or US standards, or which have been developed for special purposes. It will be seen that two 'old favourites', B6 and C4, do not appear. These have not, in fact, been manufactured as such for many years. The modern equivalents are Silverflo 16 and Silverflo 24, both cadmium-free alloys. Another useful product is the 67% silver alloy made to DIN L.Ag67, marketed by J. M. & Co., Ltd as 67E. This is almost white in colour and makes a better match to steel etc than do the yellower alloys containing more copper. The melting range is 705/723°C.

GENERAL DATA ON SILVER-BRAZING ALLOYS TO BS1845/1984, WITH SOME EQUIVALENT 'TRADE MARKS'

BS 1845 No.	*Ag %*	*Cu %*	*Zn %*	*Cd %*	*Sn %*	*Mn %*	*Ni %*	*Approximate Solidus °C*	*Approximate Liquidus °C*	*Fry's Metals Ltd*	*Johnson Matthey Ltd*	*Thessco Ltd*	*Remarks*
Ag1	60	15	16	19	—	—	—	620	640	FSB No. 3	Easyflo	MX20	For all general work. Fine fillets.
Ag2	42	17	16	25	—	—	—	610	620	FSB No. 2	Easyflo No. 2	MX12	For all general work. Cheaper than Ag1.
Ag3	38	20	22	20	—	—	—	605	650	FSB No. 1	Argoflo	AG3	For wider joint gaps than Ag1 and 2. Moderate fillets.
Ag11	34	25	20	21	—	—	—	612	688	FSB No. 15	Mattibraze 34	MX4	Cheaper grades, for wider gaps. Larger fillets and wider melting range.
Ag12	30	28	21	21	—	—	—	600	690	FSB No. 16	Argoswift	MX0	
Ag9	50	15½	15½	16	—	—	3	635	655	FSB No. 19	Easyflo No. 3	MX20N	Nickel bearing, rather sluggish. For brazing tool-tips. Forms substantial fillets.
CADMIUM FREE ALLOYS													
Ag14	55	21	22	—	2	—	—	630	660	FSB No. 29	Silverflo 55	M25T	Cadmium free substitute for Ag1 and Ag2. Very fluid. Tin bearing
Ag20	40	30	28	—	2	—	—	650	710	—	Silverflo 40	M10T	
Ag21	30	36	32	—	2	—	—	665	755	FSB No. 33	Silverflo 302	M0T	
Ag13	60	26	14	—	—	—	—	695	730	FSB No. 4	Silverflo 60	H0	Low zinc. Recommended for nickel-bearing alloys.
Ag5	43	37	20	—	—	—	—	690	770	FSB No. 5	Silverflo 43	—	Useful for step brazing.
Ag7	72	28	—	—	—	—	—	MP	780	FSB No. 17	AgCu Eutectic	H12	Very fluid indeed; for vacuum brazing.
Ag18	49	16	23	—	—	7·5	4·5	680	705	FSB No. 37	Argobraze 49H	M19MN	For brazing carbide tool-tips.
Ag19	85	—	—	—	—	15	—	960	970	—	15 Mn-Ag	—	Very costly; the silver-manganese eutectic.
From BS1845/1977, but still available													
Ag10*	40	19	21	20	—	—	—	595	630	FSB No. 10	DIN Argoflo	MX10/DIN	Cheaper than Ag1 and 2 but slightly longer melting range. Fluid.
Ag15*	44	30	26	—	—	—	—	675	735	FSB No. 39	Silverflo 44	M14	Ag15–17 form a series of alloys suitable for successive step brazing.
Ag16	30	38	32	—	—	—	—	680	770	FSB No. 25	Silverflo 30	M0	
Ag17*	25	41	34	—	—	—	—	700	800	FSB No. 23	Silverflo 25	L18	

*These alloys also conform to the DIN specifications. Note that the alloy contents shown lie at the midrange of the specifications, and slight differences may be expected between makers. The same applies to the solidus and liquidus figures, which may be a few degrees up or down.

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Soft solders

Tinmans quality

BS219 grade	*Tin %*		*Antimony %*	*Melting range* *Solid °C*	*Liquid °C*	*UTS* tons/sq. in.*	*Elong. %*	*Remarks*
A	64		0·6 max	183	185	3·9	20	Very fluid. For fine work.
B	50		3·0	183	204	3·9	67	For general work. (Not zinc or brass.)
C	40	Lead-remainder	2·2	183	227	3·4	63	General work. (Cheaper than B.)
F	50		0·5 max	183	212	3·0	69	For electrical work and general sheet metalwork.
K	60		0·5 max	183	188	4·0	52	For fine electrical work.

*UTS and elongation figures are approximate, as they depend on the conditions of test and type of joint (see page 7.10 for equivalent figures in N/mm^2).

Plumbers quality

Important Note: 'Plumber's solder' proper has a high lead content and long melting range, as shown below. Grades H and J are *prohibited* from use in water systems today. 'Solder that plumbers use' on such systems is the lead-free grade, BS219–99C.

BS219 grade	*Tin %*	*Antimony %*	*Melting range* *Solid °C*	*Liquid °C*	*Remarks*
H	35	0·3	183	244	For plumbers wiped joints, auto body solder, and for dipping baths.
J	30	0·3	183	255	
99C	99·5	0·5 Cu	. . . 228 . . .		For jointing copper waterpipes.

High melting-point soft solders

Name or spec. no.	*Type*	*Solid °C*	*Liquid °C*	*Typical properties*
HT3(95a)	95·5% Tin	236	—	1·8 ton/sq. in. at 100°C, 1·3 at 150°C 0·5 at 200°C.
Comsol	AgSnPb	. . . 296 . . .		2·5 ton/sq. in. UTS, 40% elong. at 150°C.
LM5	Silver/ Cadmium	338	390	8·4 ton/sq. in. UTS at 20°C. 2·0 at 150°C. 1·2 at 200°C. 0·8 at 250°C.

Fusible alloys
The following are the approximate composition of some alloys used for (e.g.) fusible plugs in boilers and the like.

Melting point °C	*Composition*
137	56·7% Bismuth, 43·3% Tin
177	52% Cadmium, 48% Tin
180	36·5% Lead, 63·5% Tin
247	12·5% Antimony, 87·5% Lead
318	2·3% Zinc, 97·7% Lead

Solder resist
It is sometimes necessary to prevent soft solder from adhering to vulnerable parts of the workpiece. Mix fine precipitated chalk to a paint with methylated spirit (or cold water) and paint on the part. Allow to dry before soldering. Or use 'Plumber's Black Lead', obtainable from builders' merchants.

Fluxes
For *Soft Soldering* the temperature is low enough to permit simple protective coatings to be used on some metals. *Tallow*, for example, will serve when soldering newly-scraped lead surfaces. *Resin* is an excellent flux especially in fine electrical work where its non-corrosive properties are valuable; however, it is desirable that the surfaces be tinned first. (Pre-tinning *all* soft soldered joints will make a much stronger joint.) When dealing with materials which readily form oxides such as copper, brass, or steel, an active flux is needed. Provided the work can be thoroughly cleaned afterwards 'killed spirit' is excellent. It is made by dissolving zinc in hydrochloric acid till the reaction ceases, adding two further volumes of water, and finally one volume of glycerine previously mixed with one volume of alcohol. A form of killed spirit is marketed as 'Bakers Fluid'.

For all general soldering work one of the proprietary paste fluxes is best used. These are formulated by the solder manufacturers and can be relied upon. Many of these are as active as killed spirit, but less corrosive.

For *Brazing* or *Silver Soldering* an active flux is essential except when brazing copper with the phosphorus bearing alloys – the phosphorus content serves as a flux. The maker's recommendation should always be followed, as both the nature of the workpiece metal and the upper temperature limit of the operation have an effect.

The following data have been provided by courtesy of Johnson Matthey Metals Ltd.

Easy-flo, green label (or, as paste, black/white).
A standard flux which will serve for all general work where the temperature does not exceed 750°C. It is a fluoride base compound, and care should be taken to avoid prolonged exposure to fumes. The residue is water-soluble provided heating has not been prolonged. The paste form is highly recommended, and can be used in a syringe.

Easy-flo Paste, Al. Bronze grade, light blue label.
Ordinary Easy-flo flux will serve for alloys with less than 2% aluminium but for most aluminium bronzes this flux is needed. Usable with brazing alloys having liquidus below 760°C. Residue is water-soluble. It is unwise to attempt the brazing of alloys with more than 10% aluminium.

Tenacity No. 5, red label.
This flux is designed for use with higher melting-point brazing alloys, and/or where

prolonged heating is needed, as on large assemblies. It is effective on most stainless steels, but reference to the manufacturer is recommended for advice on the low nickel stainless steels. The residue is but slowly dissolved in water, and mechanical removal of flux is necessary.

Pickling

Pickling is *not* necessary for the removal of most brazing fluxes; if the work is quenched from 'black-hot' in warm water, the thermal shock will crack off most flux and the rest will usually dissolve. However, a sulphuric acid pickle will remove oxide or scale that has formed on the work and does slightly accelerate the flux removal. The often suggested 'old battery acid' is not recommended; it is almost certain to contain lead sulphate, which may react with the work. The best proportions for pickle are from 5 to 8 volumes of concentrated sulphuric acid carefully poured into 100 volumes of water; this strength of water-acid mixture can be obtained from most chemists.

(*Note* that lead-bearing free-cutting steels should *not* be acid pickled for more than five minutes – and best not at all.)

For pickling iron castings, to remove scale and adhering sand, a much weaker acid solution should be used for a longer period. Commercially, iron castings are pickled for 6 to 10 hours in 1 part acid to 30 parts water. The castings must be washed in *hot* water afterwards.

Some commercial pickles contain hydrofluoric acid, used hot. This is not recommended for model engineers.

In time, pickle will become 'loaded' with copper sulphate, shown by heavy copper deposit on steel specimens. It should then be replaced. Neutralise the acid with lime or washing soda before flushing down a drain.

ENGINEERING ADHESIVES

Three broad types of adhesive are used by model engineers: epoxy 'two part' resins such as *Araldite*; anaerobic adhesives of which *Loctite* is typical; and the cyanoacrylates. There is a considerable number of such adhesives available, especially of the epoxy type, so that only 'typical' data can be presented in this section. It must also be emphasised that for any given grade of adhesive the bond-strength can vary enormously. This strength depends amongst other things on the surface finish of the joint faces; on the nature of the two materials being joined; on the temperature at cure (and on the mixing temperature for 2-part adhesives) and on the 'environment' of the joint – temperature, humidity, pollution, etc. For any especially important joints, therefore, the manufacturers should be consulted and/or a series of trial joints made.

Epoxy adhesives

These are in the form of two more or less viscous compounds, an adhesive and a hardener. When mixed, a chemical reaction occurs between the two, resulting in a hard adhesive bonding the joint-faces. The range is very large indeed, from the fluid to the putty-like in consistency, and with an equally large range of properties.

The most commonly used come either in 2-tube packs, or double syringes which dispense the correct amount of resin and hardener, the most well known being *Araldite* though there are others. There are two variations, 'regular', which has a long pot life and is slow curing; and 'rapid', with a very short pot life and a correspondingly brief clamping and curing time. Both have a 1 to 1 ratio by volume of resin and hardener. The details which follow are those for *Araldite* as sold in retail outlets.

Araldite standard cure

Cure at 20°C for 24 hours or at 60°C for 3 hours or at 100°C for 20 minutes. Cured at room temperature, the joint may be handled after 3 to 5 hours. After 24 hours at 20°C, shear strength on a lap joint about 2200 lbf/sq. in. (15 MN/sq. metre) rising to 2 600 lbf/sq. in. (18 MN/m^2) after 3 months.

Cured at 80 to 100°C the shear strength is about 50% greater than when cured at room temperature. The adhesive is relatively brittle so that with lap joints the ratio of thickness to overlap has an influence. If thickness/lap is 0·1, shear strength is about 75% of that at ratio 0·2, and 50% of that at ratio 0·4. Scarfed joints reduce this effect. At 80°C the shear strength is about 30% of that at room temperature.

The shelf life of the unmixed constituents is up to 5 years. Pot life after mixing is about 30 minutes at 25°C.

2012 (*Araldite* rapid)

Setting ('handling') time at 15°C is about 20 minutes. Curing time to reach 1500 lbf/sq. in. (10 N/mm^2) is 70 minutes at 15°C, 60 minutes at 25°C, 25 minutes at 40°C. Shear strength after 7 days cure at 25°C is about 2 300 lbf/sq. in. (16 N/mm^2). For high strength a 'forced cure' of 24 hours at 23°C followed by 30 minutes at 80°C can be applied. This will give a shear strength of about 5 500 lbf/sq. in. (40 N/mm^2).

This grade is temperature sensitive above 50°C, the shear strength falling to 500 lbf/sq. in. (3 N/mm^2) at 100°C. Shelf life unmixed up to 3 years; pot life after mixing, 3–4 minutes at 25°C.

Epoxy putty

Supplied in the form of 'bricks' of adhesive and hardener, mixed one to one by kneading. The main application is the repair of concrete, stone, or brickwork, and it has a use as a packing under the feet of machines etc., as it can be kneaded in place yet sets as a firm support. As it is intended for use on construction sites the formulation allows a reasonable cure time even at freezing temperatures.

Temperature °C	*0*	*5*	*15*	*25*	*30*
Usable life of mixture	2 hr	$1\frac{1}{2}$ hr	1 hr	30 min	20 min
Setting time, hour	10	5	3	2	1
Cure time, hour	170	72	48	24	18

Tensile strength is 2000 lbf/sq. in. (13·8 MN/m^2), compressive strength 11 600 lbf/sq. in. (80 MN/m^2) and shear strength about 550 lbf/sq. in. (3·8 MN/m^2).

The adhesive will set in the presence of water – in some cases under water – but it is preferable that the joint surfaces be dry or at least free from loose water during application.

General notes

The epoxies are not too sensitive to gap size though an adhesive thickness of 0·05–0·1 mm is desirable. *Araldite* and the like can be used as a casting resin, or even for encapsulation, though specially designed products are available for such. Their main attribute in the model field is probably for repair of lightly loaded castings and so on. AV/HV100 and similar can be 'filled' with metallic dust or colouring matter for decorative purposes.

Cured joints may be dismantled using heat above 120°C. Given time a soak in

Nitromors or similar paint stripper will soften the adhesive. Cellulose thinners can be used to remove smears of *uncured* resin.

Anaerobic adhesives

These comprise a self-setting compound, the curing of which is inhibited by the presence of air. On installation in an air-free situation the compound will cure to a strong solid or semi-solid. This process is assisted by the catalyst effect of the metal of the joint, but in certain cases – non reactive metals, for example, or non-metallic joints – the curing must be assisted by the use of a catalysing primer or 'activator'. Maker's data sheets should be consulted when using these adhesives on: stainless steel, aluminium, hardened steel, zinc alloys, and some types of plated surfaces. A normal machined – turned or milled – surface provides sufficient abrasive key, and for the thread-locking types an effective cure can be obtained on the threads 'as made', though de-greasing will give improved performance. Most anaerobics lose from 30% to 50% of their shear strength at 100°C.

Formulations range from those with relatively weak shear strength, but with a 'sticky', high friction, characteristic after the bond is broken, for such applications as the locking of nuts, through medium strength adhesives, which will retain components like ball-races, yet not prevent disassembly by normal methods, to the so-called 'high strength' retainers for the permanent bonding of the two parts. Most are designed for use in shear, but gap-filling 'liquid gasket' is also available.

The following data refer to some of the *Loctite* range of products.

	Shear strength		*Time to*		*Gap range*		
Type no.	*N/sq. mm*	*lb/sq. in.*	*Handle (30% FS) min*	*90% full strength hours*	*mm*	*in.*	*Remarks*
648	20/25[4]	2800/3500[4]	5	3–6	0·10	0·004	High temperature retainer to 175°C.
641	8/12	1150/1750	10/30	4–6	0·15	0·006	'Bearing fit' medium strength retainer.
638	25/30	3600/4300	5	3–6	0·10	0·004	High strength slip-fit, up to 120°C.
601	17·5/22·5	2540/3260	10/30	4–6	0·10	0·004	Slipfit retainer.
510	10 to 12[1]	1400/1700[1]	10/30	12	0·04	0·15	Flange sealant to 4500 lbf/sq. in.
290	4 to 6·5[2]	600/1000[2]	10/30	3	0·1	0·004	Penetrating for use on in-situ threads.
270	8 to 12[2]	1150/1750[2]	10/30	$\frac{1}{2}$/2	0·25	0·010	Studlock. Med. strength, gap filling.
242	4 to 8·5[2]	600/1200[2]	10/30	$\frac{1}{2}$/5	0·25	0·010	Nut locking.
222	1·5 to 4[2]	220/580[2]	10/30	1/5	0·25	0·101	Locking adjustable screws etc.
317	18 to 21	2500/2900	$\frac{1}{2}$ to	24	0·2	0·008	A very rapid anaerobic adhesive.
	35 to 42	4900/5800[1]	1 min				Requires use of primer type NF in
							tension.[3]
542	N.A.		15 min	12	Thread clnce.		For pipe thread seal up to 150°C.

Notes: [1] Ultimate tensile strength.
[2] Shear strength at breakaway of nut from bolt.
[3] Or heat cure at 120°C for three minutes.
[4] 80% of shear strength retained at 150°C, 60% at 175°C.

Cyanoacrylate adhesives

These are very fast curing compounds, the curing process being the result of the presence of humidity on the bond surfaces. Care is needed in use, as the curing process means that *human tissue will bond almost instantly*. However, if such accident happens (e.g. between fingers) the bond may be carefully *peeled* apart. Deposits on the skin should be washed in warm soapy water and peeled off. It is especially recommended that goggles be worn when handling these materials.

The bond strength is high, and the adhesives are especially useful for bonding dissimilar materials – glass to metal, or plastic and ceramics. The compound is applied in the form of a drop or drops, sufficient to cover the surface after applying the two parts. Degreasing is desirable, but washing with detergent and water is usually sufficient. The following data are typical of the 'general purpose' type of cyanoacrylate.

Tensile strength, 30–35 N/sq. mm (4300–5000 lbf/sq. in.)
Shear strength, 20–25 N/sq. mm (2900–3500 lbf/sq. in.)
Gap filling capacity 0·1 mm (0·004″)
Handling strength in less than 60 seconds.
Full strength in about 3 hours, cured at room temperature.

The variety of this class of adhesive is now much extended, both for special application and for gap filling. Special 'toughened' cyanoacrylates (e.g. *Loctite* 480) have also been developed for use on metals.

Two-part 'Elastomeric' adhesives

A number of so-called 'multi-bond' adhesives are now available, in which the adhesive is applied to one surface and an activator to the other. Handling strength may be obtained within 1 to 2 minutes, the strength increasing with curing time up to 24 hours. Holding together or light clamping is advisable. Provided that there is no cross-contamination the adhesive may be applied to one surface and the activator to the other some hours before the joint is made. Many will tolerate slightly oily surfaces and gap-filling qualities are good – up to $\frac{1}{2}$ mm or more with many – though strength is highest with small (0·05 mm) gaps. Clamping is essential with gaps over 0·125 mm. Occasional exposure to temperatures above 150°C will not seriously affect strength (with some formulations joints may be baked after painting) but normal operating temperatures should be kept below about 60°C. Typical strengths are 7 N/mm^3 (1000 lbf/in.^{3}) in tension and 10 N/mm^3 (1400 lbf/in.^{3}) in shear, but different formulations vary quite widely. Their main advantages are that they can be used successfully in bonding dissimilar materials (e.g. glass to steel) and their ability to tolerate slightly oily surfaces.

Shelf life

The *shelf life* of the epoxy adhesives is about 3 years – the adhesive itself lasts about 5 years, but the hardener goes off first. Anaerobics should remain effective for at least 12 months, and possibly longer with reduced strength and longer curing time, but the slightest contamination with metal dust may shorten the life and reduce adhesion considerably. The cyanoacrylates have a shelf life of 6 months at room temperature but are best kept in the refrigerator (around 5°C) when the life may be extended up to 12 months.

Stress factors

These adhesives, though 'strong' relative to ordinary glues etc., are for the most part less than 20% of the ultimate strength of even soft solder. The epoxies are brittle and the working stress should be reduced by a factor of 10 if shock loads are probable, especially when (e.g.) transverse loads are applied to a butt joint. For shearing loads on all types, it is recommended that after allowing for all unknowns the working stress should not exceed $\frac{1}{3}$ to $\frac{1}{2}$ of the ultimate shear stress. It must be remembered that though heat may assist cure, prolonged exposure to temperatures above about 150°C is to be avoided. If the working temperature is at 100°C–150°C allow an *additional* factor of $\frac{1}{3}$ on the ultimate stress.

Most cyanoacrylates are vulnerable to prolonged exposure to hot water, especially if soapy or containing detergent residues. When applied to the repair of dentures early failure can be expected!

SECTION SEVEN
PROPERTIES OF MATERIALS

WITH NOTES ON TOOL HARDENING AND TABLES OF WEIGHTS OF METAL SECTIONS. METRIC REPLACEMENT OF WIRE & SHEET GAUGES

NOTE ON STEEL SPECIFICATION NUMBERS

The familiar 'EN' steel numbers, originated in BS970/1941, and revised in 1955, had become less and less rational, so that they were replaced by new designation numbers in BS970/1972. The old-fashioned EN numbers are still used by some, but the new designations, most of which are inherently descriptive of the steel analysis, should now be used.

A six 'bit' code is now used, e.g. '212M36' – which corresponds to the free-cutting steel EN8M. The letter in the middle indicates the *type* of specification, for which three different letters may be used, viz:

'M' for steel supplied to a specific set of *mechanical properties*.
'A' for steel having a specified chemical *analysis*.
'H' for steel tested to comply with a specified *hardenability*.

The same material may be supplied to any of these three, the difference being only that the supplier is required to work to the British Standard limits in the designated category.

A letter 'S' may replace the above; this means the steel is '*stainless*' (using the description very broadly – it includes heat resisting valve-steels) and in this case the category of test is specified separately from the designation code.

The first three figures serve several functions. The first is a broad description of the *type* of steel.

Nos 000 to 199 are plain carbon steels, with some manganese content.
Nos 200 to 240 are free-cutting versions of the above.
Nos 300 to 499 are all 'stainless' or valve steels.
Nos 500 to 999 are 'alloy' steels other than stainless.

In the case of plain carbon steels, the *second and third* figures indicate the manganese content, whilst for free-cutting steel the same figures show the sulphur content. Thus '175' means 0·75% manganese, and 220 stands for 0·20% sulphur. For other steels – stainless and alloy – the three figures have no significance to model engineers, though some do correspond to the American Iron & Steel Institute listings.

The final two figures of the six-bit code indicate the carbon content of all steels except stainless, where they are arbitrary; e.g. figures 60 mean 0·60% carbon. (If the carbon content exceeds 1·0%, then these last two numbers will always be 99 but high carbon 'tool' steels are not covered by BS970. 'Silver steel' is specified in BS1407.)

070M26 is a plain carbon steel supplied to specified mechanical properties, having 0·26% carbon and 0·70% manganese.

212A36 is a free-cutting steel supplied against chemical analysis, with 0·12% sulphur and 0·36% carbon.
503H40 is an alloy steel of 0·40% carbon content, supplied to a specific 'hardenability'.

There are no direct equivalents to the EN numbers, but the following can be used for guidance in seeking acceptable alternatives.

Plain carbon steels

BS970	1955	EN3	EN5	EN8	EN15	EN43A
	1972	070M20	080M30	080M40	150M36	080M50

Free-cutting steel

BS970	1955	EN1A	EN1B	EN15AM	EN8M
	1972	220M07	240M07	216M36	212M44

Alloy steels

BS970	1955	EN12	EN11	EN23	EN39A
	1972	503M40	526M60	653M31	659M15
'Vernacular'		1% Ni	$\frac{3}{4}$% Cr	3% NiCr	4% NiCr

Stainless steels

BS970	1955	EN58A	EN58M	EN56A	EN56C
	1972	302S25	303S21	410S21	416S37
'Vernacular'		'18/8'	'18/8'*	13% Cr	13% Cr*

*Free-cutting

RAPID GUIDE TO STEEL SELECTION

This list is based on the *ultimate tensile strength* only, in the normalised or hot-rolled condition. Reference should be made to steel suppliers if other conditions are to be met. Specification numbers are to BS970/1972.

UTS, T/sq. in.	28	30	32	34	36	40	45
UTS N/mm^2	430	460	500	525	550	620	700
BS No.	070M20	080M30	070M26	120M28	150M28	080M46	070M55

WORKSHOP IDENTIFICATION OF MATERIALS

Several 'standard' colour codes exist for bar-stock (most metal stockists will supply a copy) but the use of any of these would demand a workshop full of paint tins. The following abbreviated code has served well over some decades, and is offered as an adequate substitute. It will be noticed that the same colour is used for two or more materials. The colour or character of the metal distinguishes between them.

No colour	Aluminium
	Mild steel (EN2 and similar)
Yellow	Free-cutting mild steel (EN1 or 1A)
	Copper
Red	Common brass (unclassified stock)
	Stainless steel (18/8)
Red/black stripe	Cast brass
Red/yellow stripe	Free-cutting stainless
Blue	Carbon tool steel (silver steel)
	German silver
	Cast gunmetal
Green	Phosphor bronze
Green/black stripe	Cast phosphor bronze
Black	70/30 brass (CZ106)
	Cast iron
Black/white stripe	Al. alloy (spec. no. stamped on end as well)
White	Hex bar of BA nut sizes, brass or steel

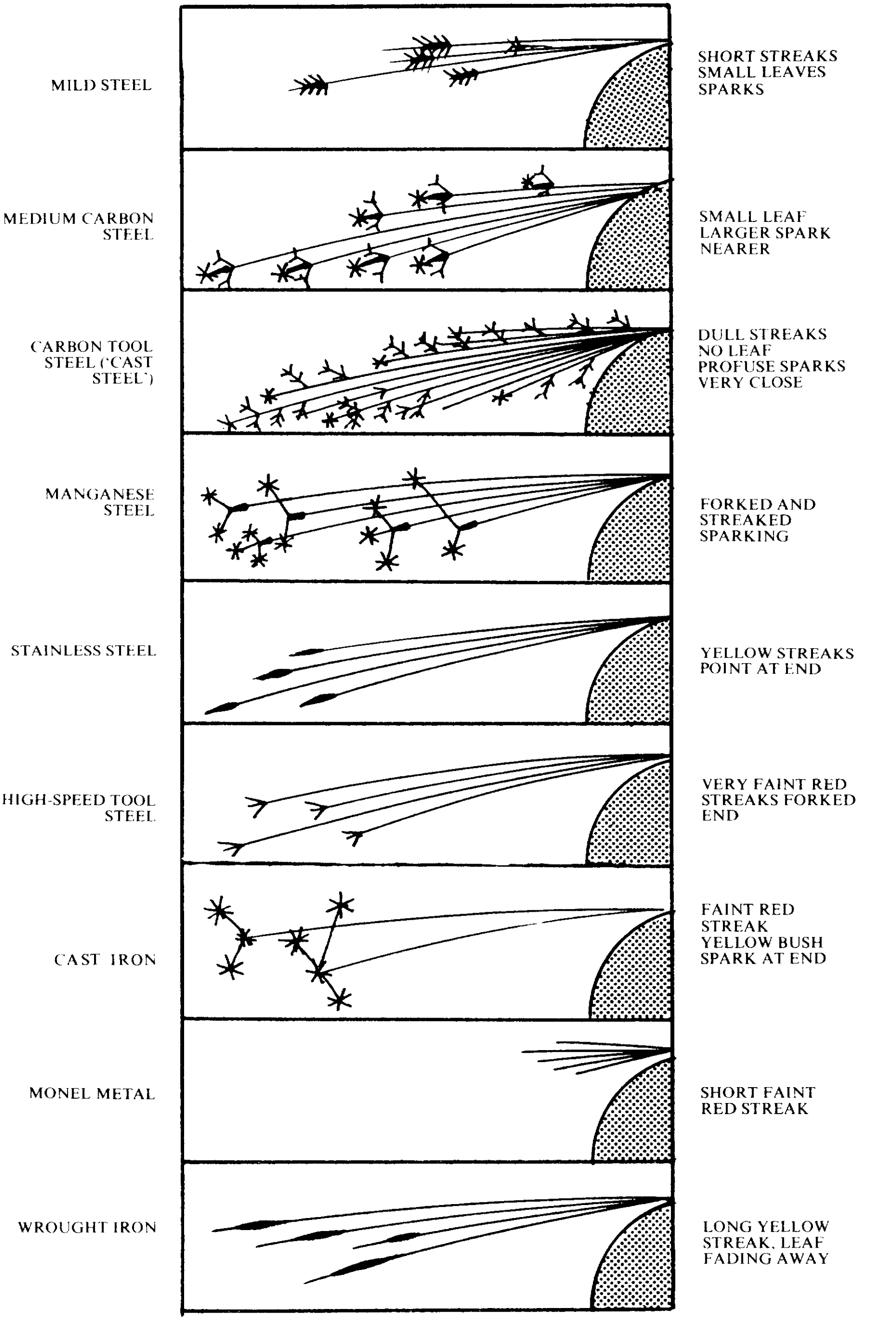
Bars are painted at each end and in the middle. A grey *overspray* between colour bands indicates a *metric size*.

Alloy steels and any special materials are coded black/white and also wrapped and labelled – small quantities only are stocked as a rule by model engineers.

THE SPARK TEST

This is a useful, if old-fashioned, method of identifying ferrous materials. The sketch on the next page gives a rough idea of the sort of spark obtained from 9 common materials, but it is better that small samples of each be kept available and a direct comparison made actually at the wheel. The spark from tool steel is about the same whether hard or soft. See the diagrams on the next page.

THE SPARK TEST FOR METALS



STRENGTH OF ENGINEERING MATERIALS

Ultimate Tensile Stress (UTS) is the stress at which the material fractures in tension. Calculated on the breaking load divided by the reduced area of the 'neck' at fracture.

Yield Stress is the stress at which the load/extension curve first deviates from a straight line. Material stressed beyond this point will not return to its original length when unloaded. Calculated on the original area of the test specimen.

Proof Stress is the stress at which the load/extension curve deviates from a straight line by a given amount; usually 0·1 or 0·2%. This criterion is used for materials which do not exhibit a sharp point of yielding. (Some US reference books refer to this criterion as a 'yield' point, and confusion can result.)

Elongation is the measure of ductility. It is stated as the percentage increase in length at fracture compared with a gauge length marked on the original unstressed material. (See also below.)

Hardness is measured by impressing a ball or pyramid point into the material and calculating a 'surface stress' from the size of the impression. The Brinell test uses a ball, the Vickers test a pyramid shaped diamond. (Other tests such as the 'Rockwell' are also used.)

'*Young's Modulus*' is the ratio between stress (load per unit area) and strain (extension under load per unit length) and denoted by 'E'.

$$E = \frac{\text{Stress}}{\text{Strain}}$$

'E' is stated in millions of lbf sq. in. or MN/mm^2. It is also known as the 'Modulus of Elasticity'.

The strength of all materials will vary according to the treatment in manufacture. Many alloy steels and all aluminium alloys require some heat treatment to bring out the full values, whilst the copper alloys vary considerably with the amount of cold working to which they have been subjected in manufacture and manipulation. The figures given in the table on page 7.8 are, in general, those which apply to the material in the condition likely to be supplied to a model engineer. Values for UTS are the mid-range for 'run of the mill' supplies. Although the Brinell number is usually used for iron and steel, these hardness figures have been quoted in Vickers Diamond numbers throughout to facilitate direct comparison. Figures for 'yield' have been given for steel, the remainder are proof stress, either 0·1% or 0·2%, depending on that called for in the appropriate BSS. This difference is of no significance in model design procedures.

Only a limited range of materials has been presented. An abridged list of steel specifications in one handbook occupies 11 pages, whilst the specifications for copper and alloys occupy two thick A5 volumes. Those in the list will serve most 'amateur engineers' for general purposes. The sulphur-bearing 'free cutting' mild steel 220M07 (EN1A) is well worth the slight extra cost over the so-called 'good commercial quality' normally supplied as 'mild steel', as it is far easier to machine. The lead bearing free-cutting steels (e.g. *Ledloy*) are not economic for model engineers. For any very highly stressed components reference should be made to the material suppliers, especially for aluminium alloys.

Ductility tests on steel

Ductility is measured by determining the extension of the specimen at fracture compared with its original length between two gauge points, usually 2″ apart. Most of this 'stretch' occurs adjacent to the fracture, where the material necks down before breaking. It follows that the ratio of gauge length to cross-sectional area of the specimen will affect the 'percentage elongation' recorded by the instrument.

The majority of tests up to recent years were conducted on specimens of 0·25 sq. in. cross section, but BS970/1972 quoted properties determined from specimens of 0·125 sq. in. area. This means that, for the identical material, the '% elongation' given in this specification will differ from that given in the earlier BS970/1955 – and from those listed in many reference books. The material is, however, no less ductile.

As a fair approximation, the figures quoted in BS970/1972 will be *about 75%* of those obtained from the larger specimens for the same ductility of material i.e. 15% in the 1972 table equates to 20% in the earlier specification.

It is unwise to compare the elongation figures between totally different types of metal, as the ratio of gauge length to cross-sectional area may be quite different in the standard tests. For example, a gauge length of 8 inches is used in some cases, and in others (e.g. copper rod) an unmachined specimen is used.

All figures for steel in this handbook refer to tests with 2" gauge length on 0·25 sq. in. cross-section; i.e., those as used in the earlier editions of BS970, except where stated.

Bright drawn steel

The majority of model engineers purchase their supplies in the bright drawn condition. In this state the steel has very different properties from those listed in reference books where the material is usually in the 'hot rolled' or 'normalised' state. The effect of bright drawing whether from the hot rolled or normalised bar is to reduce the ductility – often quite severely – and to increase the ultimate and yield strength of the steel. The effect is greater in the smaller sections. In all cases where ductility is important, and especially when any cold bending is to be carried out, it is recommended either that 'black' bar be used (machined to size if need be), or that the work be normalised before forming. (Heat to bright red and air cool.)

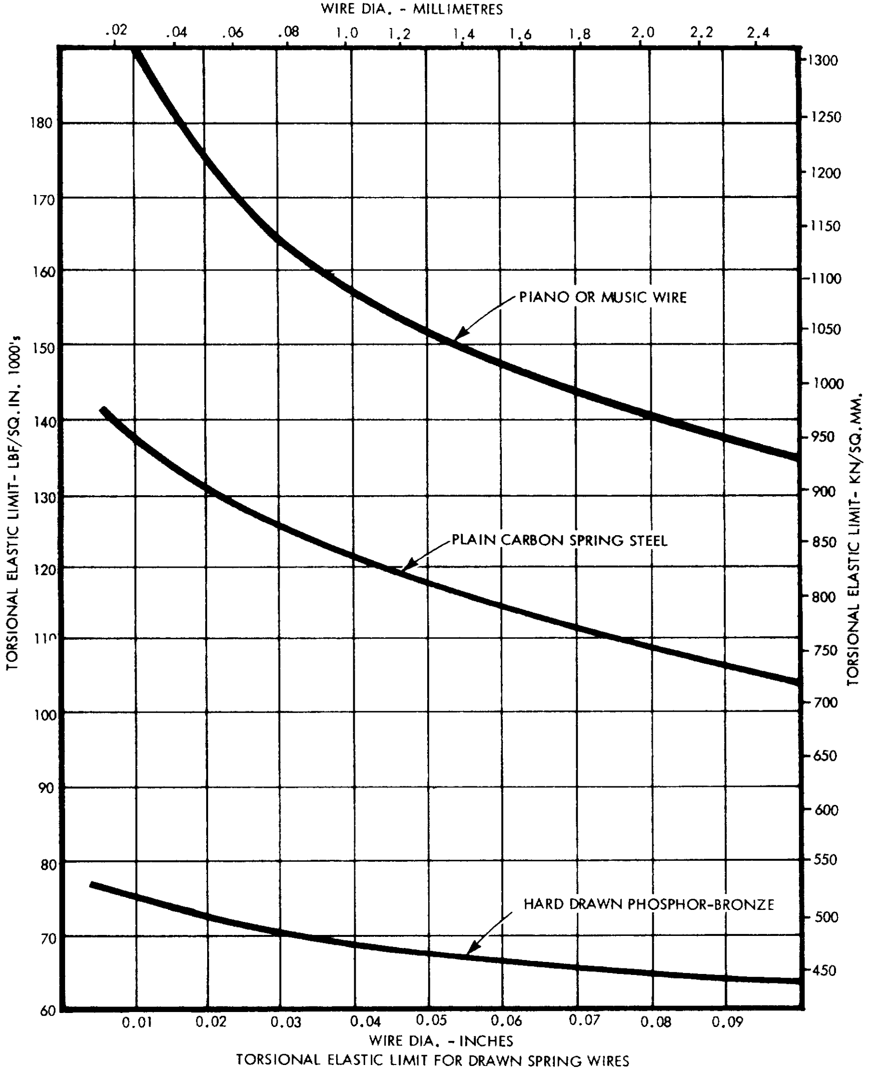
The following table shows the effect on three typical steels; changes of the same order may be expected from others . (For stresses in SI units see p. 7.10.)

Steel	*Test*	*Normalised or hot rolled*	*Bright drawn* ½" *dia.*	1" *dia.*	2" *dia.*
EN3-070M20	UTS ton/sq. in.	28 (min)	34	32	30
'20 carbon'	Yield, ton/sq. in.	14	25	24	23
	Elongation, %	26	17	17	18
EN8-080M40	UTS ton/sq. in.	35 (min)	42	40	39
'40 carbon'	Yield ton/sq. in.	18	33	31	30
	Elongation, %	21	12	13	14
EN1A-220M07	UTS ton/sq. in.	23	30	28	25
Low carbon	Yield ton/sq. in.	14	22	21	19
free cutting	Elongation, %	27	10	12	13

Elongation is measured on gauge length $= 4 \times \sqrt{\text{Area}}$ to BS970/1972 in this table.

Hard drawn spring wire

The strength of hard-drawn spring wire arises solely from work hardening, though there is some heat treatment involved in the actual drawing process – the so-called 'patenting' of the wire. It follows that the more the wire is drawn the greater will be this effect, so that higher stresses can be carried by wire of small diameter than by larger gauges. The curves show the *elastic limit in torsion* (coil springs are stressed in torsion) for the three most commonly used spring wires. The actual elastic limit may be slightly higher than shown, but is not likely to be less. It is usually recommended that the maximum *working* stress to be used in the spring formulae on pp. **13**.1–**13**.4 should not exceed 70% of the figures given by the curves.



TORSIONAL ELASTIC LIMIT FOR DRAWN SPRING WIRES

AVERAGE PROPERTIES OF ENGINEERING MATERIALS

Material	*UTS* ‡ *Ton/sq. in.*	*Yield*	*Elong" %* *see page* 7.5	*Hard'* *Vickers*	*'E'* ‡ *lb. sq. in.* × 10^6	*Specification number*	*Remarks or condition*	
Cast iron	17	—	(55)*	200	16–19	BS 1452/260	Grey CI. * = Compressive UTS, ton/sq. in.	
Malleable iron	20–25	8–10	5–10	130	11		Varies considerably.	
Steel						**BS970/55**		**BS970/72**
'20 carbon'	32	24	12	120/180	30	EN3	Bright drawn	070 M20
Free-cutting M.S.	28/30	20	12	120	30	EN1A	Cold rolled or drawn	220 M07
'40 carbon'	35	18	21	140/220	30	EN8	Normalised	080 M40
'55 carbon'	50	40	10	240/300	30	EN9	Bright drawn $\frac{5}{8}$" dia.	070 M55
Stainless '18/8'	51	36	25	180/220	28	EN58A	'Bars for machining'	302 S25
'Silver steel'†	40–60	35–50	20–35	280 max	30	P. STUBS	As supplied – ground bars	BS 1407
Copper sheet	14	3	50	50	16	C106	Annealed sheet	Deoxidised copper
	17	14	30	85	16	C106	H/hard sheet	as used
	23	20	10	100	16	C106	Hard sheet	for boiler making
	17	13	30	90	16	C106	Drawn rod	
Brass – 70/30	21	6	70	80	15	CZ105/6	Annealed sheet	Deep drawing brass
	25	19	40	120	15	CZ105/6	H/hard sheet	or
	22	14	50	100	15	CZ105/6	Drawn rod	'Cartridge brass'
Brass 63/37	22	8	50	65	13	CZ108	Annealed sheet	
(Common brass)	27	21	30	130	13	CZ108	H/hard	
	24	15	35	120	13	CZ108	Drawn rod	
Brass 60/40	23	8	45	90	14	CZ109	Annealed	
(Muntz metal)	27	14	35	125	14	CZ109	Drawn rod (up to 1" dia.)	
Brass, free cutting	22	7	45	75	13	CZ119	Annealed	
	28	18	25	140	13	CZ119	Drawn rod, 0–$\frac{3}{8}$" dia.	'Screw-brass'
	26	14	35	120	13	CZ119	Drawn rod, $\frac{3}{8}$–1" dia.	

† See note on page 7.5. ‡ For conversion to SI units see p. 7.10

AVERAGE PROPERTIES OF ENGINEERING MATERIALS – *Continued*

Material	*UTS Ton/sq. in.* ‡	*Yield Ton/sq. in.* ‡	*Elong"% see pages 94 & 95*	*Hard' Vickers*	*'E' lb. sq. in. × 10⁶* ‡	*Specification number*	*Remarks or condition*
Gunmetal	17–20	8–12	13–25	80	16	—	'Admiralty', Sandcast (varies from foundry
	14–17	7–8	13–25	—	16	—	'Leaded GM', Sandcast to foundry.)
Phosphor bronze	33	25	25	160	13·5	PB102	Drawn rod
	60	—	—	—	14	PB102	Hard-drawn wire
	16	9	5	—	—	PB1	Cast P.B. for bearings.
Nickel silver 12%	32	23	15–25	150	18·6	NS104	Drawn rod Specifications vary
	13–17	8–12	10–25	—	18·6	—	Cast German silver according to purpose
Monel metal	33–44	13–24	30–50	150	25	'400'	Drawn rod
Aluminium 'pure'	7	5	20–30	30	10	E1A	Extruded rod
	7	5	8	30	10·1	S1A	H/hard sheet
Duralumin	25	15–19	15	110	10·3	—	
Y-Alloy	10–14	8–10	1–2	85	10·2	LM14	As cast
RR56	26	20	10	*	10·5	DTD130B	Extruded bar & rod
Hiduminium 22	12·5	6	25	*	10·3	NE4	Extruded bar & rod
Hiduminium 24	12	5	18	*	10·3	DTD634A	Ductile sheet for drawing
Aluminium bronze	38	18	28	160	17	CA103	9% Aluminium. Drawn rod
Pure metals							
Iron	19	—	40	—	30		
Lead	1	—	—	—	2·2		
Tin	2	—	35	—	4		As cast.
Nickel	40	30	30	—	30		Annealed rod
Silver	20	—	41	—	11		Hard sheet

* Properties depend on the heat treatment

‡ For conversion to SI units see p. 7.10.

CONVERSION TABLE, METRIC, IMPERIAL, AND SI UNITS OF STRESS

SI Units	*'MKS' Metric Units*	*Imperial Units*	
Megapascals (MPa) or Newton/sq. mm, or MegaNewton/sq. m	*Kgf/sq. mm*	*Ton/sq. in.*	*Lbf/sq. in.*
5	0·51	0·32	730
10	1·02	0·65	1450
20	2·04	1·30	2900
30	3·06	1·94	4350
40	4·08	2·59	5800
50	5·10	3·24	7250
60	6·11	3·89	8700
70	7·13	4·53	10 150
80	8·15	5·18	11 600
90	9·17	5·83	13 050
100	10·19	6·48	14 500
200	10·38	12·95	29 000
300	30·57	19·43	43 500
400	40·76	25·90	58 000
500	50·95	32·38	72 500
600	61·41	38·85	87 000
700	71·3	45·3	101 500
800	81·52	51·80	116 000
900	91·70	58·31	130 500
1000	101·90	64·76	145 000

A few specifications quote stresses in *Hectobar*. 1 HBar = 0·1 MPa or 0·1 N/mm^2. Hence, divide stress in HBar by 10 and use the first column. See also page **1**.10.

HARDENING OF TOOL STEEL

As the hardening of 'high-speed' tool steel requires temperatures and precision in control not available to the average amateur, and in any case, HSS toolbits already hardened are normally used, this section deals only with 'high-carbon' or 'cast' steels.

Carbon tool steels contain about 1·0–1·25% carbon 'dissolved' in the iron in the form of a number of iron carbides or in 'solid solution'. The object of hardening is to convert all into a hard variety of alloy. This alloy normally exists only at high temperature and if cooled slowly it will revert to the softer variety. However, the conversion from state to state is a relatively slow process, so that if the cooling be effected rapidly the iron-carbon alloy is 'frozen' in its hard state. The first step in hardening is, therefore, to heat the metal to a temperature somewhat above the transition level, and then to cool rapidly ('quench') to room temperature.

In this state the material is very hard, but has little ductility; a small tool can shatter in pieces if dropped. Further, the material is of coarse structure and will not take a high finish necessary for cutting tools. The material can, however, be 'refined', improving both ductility and toughness, and also the grain size, by heating to a relatively low temperature; a process known as 'tempering'. The degree of tempering required depends to some extent on the particular steel used, but rather more on the purpose to which it is to be put; steel springs require more tempering than does an engraving tool. This is the second part of the process. (*Note*. In the USA the word 'temper' often refers to the prior quenching process and 'draw' to the subsequent refining.)

For steels having more than about 0·9% carbon content the 'hardening range' of temperature varies from 1390/1430°F (755–775C) for 1% carbon to 1380/1420°F (750/770C) for those over 1·1% carbon. However, the presence of other alloying elements (manganese or chromium) will modify these figures and unless it is known that a 'plain' carbon steel is being hardened the maker's advice should be sought. For Stubs silver steel the range is 770–790°C.

The necessary temperature can, however, be found by observing that as the metal is heated it reaches a point where it is no longer attracted by a magnet. This is the 'decalescence point' at which the carbon alloy transformation takes place. Test by just touching the tool with the magnet – don't let it get hot, or it will be demagnetised. A temperature about 100°F (60°C) above this will generally provide satisfactory hardening.

The transformation takes time; heating should, therefore, be *slow*, and the tool should be held *at* the hardening temperature for some time. An old rule is 'one hour per linear inch of tool section' so that a tool $\frac{1}{4}''$ square must be held for at least 15 minutes. However, when hardening thin, but wide, workpieces the rule should be applied only to the thickness, so that a $\frac{1}{8}'' \times 2''$ flat plate should be held in heat for $\frac{1}{8}$ hour, or 7–10 minutes. Even very thin sections should be held at the hardening temperature for 5 minutes if the maximum hardness is desired.

Heating may produce two unwanted effects – scaling and 'decarburising', the latter being a reduction of carbon content of the surface with resulting reduced hardness. In the case of normal tools both can be overcome by subsequent grinding; with the usual model engineer's size tools allow about $\frac{1}{32}''$ to come off after hardening. With form-tools, however, this cannot be done. One way of overcoming the problem is to coat the business end with a thick paste of chalk powder and water, and wrap this again with thin iron wire. This will reduce both scaling and decarburising. The wire should, however, be pulled off just before quenching as it will insulate the tool from the cooling effect.

Clean water is an effective coolant, but brine (1 kg per litre of water) is better. The tool must be immersed, vertically and point downwards, and a *moderately* slow up-and-down motion given – this is preferable to a violent swirling about. Rainwater should be used in hard water districts, and the bath should be between 60°F and 80°F (say 20°C). For amateurs it is far the best to quench right out, when treating small work e.g. a scriber, rather than quench the point and allow the tool to 'temper from the shank'. Then temper as a second operation.

When tempering lathe and similar tools it is desirable to keep the shank softer than the point, typical figures for a plain carbon steel being 850/860 Vickers at the point and 250/350 in the shank. The tool can, therefore, be tempered by heating the *shank* **slowly** and watching the colours 'run up' to the point. As soon as the 'colour' reaches the end the tool must be quenched in water to prevent overheating. For work which must be tempered evenly all over, a sand-bath can be used, the sand being heated evenly from below with the tool on top. Alternatively the work may be immersed in molten solder of the appropriate melting point (or even the chip-pan!) – but the work must be preheated to nearly the desired temperature first. Tinman's solder will temper at about 'middle to pale straw' and plumber's to 'dark straw'. In either case, heating should be slow. The work should be quenched in oil, or water at 50°C, agitating as for primary hardening.

'Silver steel'

The following data refer to '**Stubs** silver steel' – one of the most commonly used 'amateur' tool steels. It is, in fact, a 1·1/1·2% carbon steel with the addition of about 0·35% manganese, 0·45% chromium and 0·1/0·25% silicon. The 'permitted' range of carbon in BS 1407 is from 0·95% up to 1·25%. Performance as a cutting tool may be disappointing if the carbon content is less than 1·05%. The chromium content is intended to reduce distortion, and improve through hardening on thicker sections. The US equivalent is 'drill rod', which normally contains no chromium.

Hardening temperature – 770/790°C or 1420/1450°F, water quenched. Vickers hardness, untempered, about 950.

Tempering temperatures.	150°C (300°F)	870–940 Vickers.
	200°C (390°F)	750–800 Vickers.
	250°C (480°F)	720–770 Vickers.
	300°C (570°F)	640–700 Vickers.

For the small lathe tools used by model engineers, tempering just to 'pale yellow' (210°C) will serve well for brass etc., and to 'pale straw' for tougher materials – mild steel etc. Provided there is no shock load from 'interrupted cuts' excellent results are obtained if the tool is tempered by heating in the deep chip fryer (about 165°C) for 30 minutes, but the clamping area of the shank may have to be let down further, say to dark straw, afterwards. For scribers, gravers and the like, temper till at 'pale yellow' about $\frac{1}{8}''$ from the point. For punches and cold chisels, temper to 290°C (purple-blue) and for springs to dark blue (300°C). For 'ornamental turning' in ivory, craftsmen temper until there is just a trace of colour about $\frac{1}{4}''$ from the tool point, quench in oil, and subsequently boil in water for half-an-hour.

Rehardening. Tools which may have been overheated in work *must* be fully annealed before rehardening. If this is not done, cracking or distortion is inevitable. The tool must be heated to just above the decalescence point (760/780°C is recommended for silver steel) slowly, as for hardening, soaked at this temperature for 1 hour/inch as before, and then allowed to cool *very slowly*. If done in a furnace, allow to cool in the chamber; otherwise cover with at least 2″ all round of hot ashes.

Ground flat stock. This is an oil-hardening tool steel intended chiefly for gauges etc. It should be heated to 810/840°C and quenched in oil, giving a hardness of the order of 880 Vickers. (Clean engine oil about 20SAE will serve, or cooking oil.) For gauges, temper at 'middle straw' (240°C) using the sand-bath or furnace method, quenching in oil. Tempered at 150°C it will give tolerable performance as a cutting tool (800 Vickers). The tempering temperature should be held for rather longer than in the case of 'silver' steel.

Oil tempering. Industrially, tempering in a heated oil-bath is common and amateurs might well make use of this. The work should be heated to about 300°F (140°C) before immersion, and allowed to 'soak' long enough to reach the oil-temperature right through. Special oils may be obtained for this purpose. The main advantages are even tempering throughout and the absence of any need to quench; the tool is simply withdrawn and cooled naturally. A good domestic oven thermometer will have sufficient range of temperature measurement. The deep-frying chip pan will heat to 190/200°C.

Hardening very small work. The difficulty here is the time lost between heating and cooling, and the following procedure, taken from a very old book, has been used with success on very fine wire broaches. The point is held in the blue part of a candle flame until it reaches cherry-red and immediately plunged into the wax of the candle to 'quench'. After cleaning, it is then tempered by holding the shank in the flame until the point reaches pale yellow, and again quenched in the wax. This procedure is clearly neither very scientific, nor applicable to many workpieces, but is mentioned to 'suggest' other unorthodox methods which may succeed when difficulties are met with*.

COLOUR AND TEMPERATURE

The use of colour to estimate temperature requires some care, quite apart from the fact that the definition of (e.g.) 'cherry red' may differ by as much as 100°C between different 'authorities'! In judging hardening colours the work should be viewed in a shadowed part of the workshop, well away from direct sunlight. Tempering colours vary a little for different materials, those given below being for a plain carbon tool steel. These colours should be viewed either in diffuse daylight or under a tungsten lamp – on no account by light from a fluorescent tube.

'Hot' colours

Colour	*Celsius*	*Fahrenheit*
Barely red	520	950
Dull red	700	1290
Blood red	750	1380
Cherry red	800	1470
Bright cherry	825	1510
Red	850	1560
Bright red	900	1650
Very bright red	950	1740
Orange-red	1000	1830

'Tempering' colours

	°C	*°F*		*°C*	*°F*
Pale yellow	210	430	Brown-red	265	510
Light yellow	225	440	Light purple	275	530
Pale straw	230	450	Purple	280	540
Middle straw	238	460	Dark purple	287	550
Dark straw-yellow	245	470	Full blue	293	560
Dark straw	255	490	Dark blue	300	570
Yellowy-brown	260	500	Light blue-grey	335	640

Tempering colours will not develop properly within a flame. Heating should always be indirectly, either by heating the tool shank and allowing the colours to 'run up', or (better) in a sand or oil bath.

*Heat treatment of tool steel is dealt with in detail in the author's book *Hardening, Tempering and Heat Treatment*, Nexus Special Interests, 1985.

SPECIFIC WEIGHT & DENSITY OF MATERIALS

(a)

Metals	*Lb/cu. in.*	*Grams/cc*
Cast aluminium	0·093	2·58
Wrought aluminium	0·097	2·69
Aluminium bronze	0·281	7·78
Cast brass (av)	0·304	8·42
Drawn 60/40 brass	0·296	8·2
'Gunmetal'	0·319	8·84
Copper (sheet/rod)	0·318	8·81
Cast iron	0·260	7·20
Lead	0·410	11·36
Steel	0·285	7·89
White metal	0·264	7·31
Zinc	0·255	7·06

("Specific gravity" is identical to density in grams/cc.)

(b)

Dry woods	*Lb/cu. ft*	*Kg/m³ (g/cc × 1000)*
Ash and beech	43–53	688–850
Birch	40–46	640–740
Boxwood	57–83	915–1300
Elm	34–45	545–720
Ebony	70–83	1120–1330
Spruce fir	30–44	480–705
Larch	31–37	500–600
Lignum vitae	83	1330
Mahogany, Honduras	35	560
Mahogany, Spanish	53	850
English oak	48–58	770–930
Pine . . . red	30–44	480–700
white	27–34	430–540
yellow	29–41	460–660
Teak	41–55	660–880

(c)

Misc.	*Lb/cu. ft*	*Kg/m³ (g/cc × 1000)*
Brick	100–125	1600–2000
Concrete	120–140	1900–2250
Earth	77–120	1230–1900
Granite	164–175	2600–2800
Sand, dry	88–120	1400–1900
Ice	57·4	919·5
Ivory	117	1874

(d) Coal: Anthracite, 40–45 cu. ft/ton; Soft coal 43–48 cu. ft/ton; Coke 80 cu. ft/ton when in 1″–3″ lumps, tipped and lying naturally. (Divide by 35·4 for m^3/tonne.)

(e) **WEIGHT OF SHEET METAL**

Thickness			*M.S.*		*Copper*		*Brass*		*Aluminium*	
mm (App)	*SWG*	*in.*	*lb/ft²*	*kg/m²*	*lb/ft²*	*kg/m²*	*lb/ft²*	*kg/m²*	*lb/ft²*	*kg/m²*
9·5	—	$\frac{3}{8}$	15·3	74·7	17·4	84·9	15·8	77·1	5·1	24·9
8	—	$\frac{5}{16}$	12·8	62·5	14·5	70·8	13·2	64·5	4·33	21·1
6·38	—	$\frac{1}{4}$	10·2	49·8	11·6	56·6	10·5	51·3	3·5	17·1
—	—	$\frac{3}{16}$	7·65	37·35	8·68	42·38	7·89	38·52	2·5	12·2
—	—	$\frac{1}{8}$	5·1	24·9	5·79	28·27	5·26	25·68	1·68	8·2
3·2	10	0·128	5·21	25·43	5·90	28·8	5·71	27·88	1·7	8·3
2·5	12	0·104	4·24	20·7	4·8	23·43	4·64	22·65	1·38	6·74
2	14	0·080	3·26	15·92	3·68	17·97	3·57	17·43	1·06	5·175
1·6	16	0·064	2·61	12·74	2·95	14·4	2·86	13·96	0·848	4·14
1·2	18	0·048	1·96	9·57	2·21	10·79	2·15	10·5	0·637	3·11
0·9	20	0·036	1·47	7·18	1·66	8·104	1·61	7·86	0·477	2·329
0·7	22	0·028	1·142	5·58	1·29	6·298	1·25	6·103	0·371	1·811
0·56	24	0·022	0·897	4·379	1·015	4·955	0·982	4·794	0·292	1·426
0·45	26	0·018	0·735	3·588	0·830	4·052	0·804	3·925	0·239	1·167
0·35	28	0·014	0·571	2·788	0·644	3·144	0·625	3·051	0·186	0·908
0·3	30	0·012	0·490	2·392	0·552	2·695	0·536	2·617	0·159	0·776
0·25	32	0·010	0·408	1·992	0·461	2·251	0·447	2·182	0·133	0·649
0·15	38	0·006	0·245	1·196	0·276	1·347	0·268	1·308	0·079	0·386

Note: Weights quoted are correct for the gauges and inch thicknesses; millimetre thicknesses are approximate, but close to the metric replacement of SWG.

(f) **WEIGHT OF BARSTOCK, kg/metre**

D. mm	*Steel*	*Brass*	*GM*	*PB*	*D. mm*	*Steel*	*Brass*	*GM*	*PB*
10	0·620	0·644	0·694	0·211	25	3·873	4·025	4·340	1·321
11	0·750	0·779	0·840	0·256	30	5·577	5·796	6·249	1·902
12	0·892	0·927	1·000	0·304	35	7·591	7·889	8·505	2·588
13	1·047	1·088	1·173	0·357	40	9·918	10·304	11·108	3·380
14	1·214	1·262	1·360	0·414	45	12·545	13·041	14·059	4·278
15	1·394	1·449	1·562	0·475	50	15·488	16·101	17·353	5·280
16	1·587	1·649	1·778	0·541	55	18·747	19·482	21·004	6·391
17	1·791	1·861	2·007	0·610	60	22·305	23·181	24·991	7·605
18	2·008	2·087	2·250	0·685	65	26·179	27·210	29·331	8·925
19	2·237	2·325	2·506	0·790	70	30·369	31·558	34·025	10·354
20	2·479	2·576	2·778	0·845	75	34·858	36·227	39·055	11·884
21	2·733	2·840	3·062	0·932	80	39·663	41·217	44·439	13·552
22	2·999	3·116	3·360	1·022	85	44·776	46·531	50·167	15·266
23	3·278	3·407	3·673	1·118	90	50·196	52·116	56·240	17·114
24	3·569	3·709	3·999	1·217	95	55·924	58·123	62·658	19·067
					100	61·968	64·403	69·429	21·127

The weights depend on (diameter)^2. Hence to find the weight of 3·5 mm dia. divide that for 35 mm by 100; for 32 mm multiply 16 m × 4. *Note*: Weight in lb/ft is very nearly $\frac{2}{3}$ × kg/metre.
For **Cast Iron** multiply steel by 0·907. For **Copper** multiply steel by 1·132.
For **square** bars, multiply weights given by 1·273; for **hexagons** by 1·102. For use with **'inch' sizes**, read the diameter as a decimal (i.e. read 25 mm as 0·25″) and divide the weight by 23·2. This gives the answer in lb/ft.

Example: $\frac{1}{2}$″ steel bar is 15·488/23·2 = 0·667 lb/ft.
Specific gravity used above: Steel – 7·89; Brass – 8·2; GM – 8·84; Al – 2·69.

WIRE AND SHEET – IMPERIAL (SWG) DIMENSIONS

Copper wire is now entirely to metric dimensions. Steel wire and rod is still available in a few gauge sizes, but most is metric. Spring wire is in transition at the present date (1995). Note that the American 'music wire gauge' ('piano wire') differs slightly from the so-called 'English' standard, given opposite.

SWG and music ('piano') wire gauge sizes

Gauge number	*SWG inches*	*Music inches*	*Gauge number*	*SWG inches*	*Music inches*
6/0	·464	·0065	23	·024	·051
5/0	·432	·007	24	·022	·055
4/0	·400	·0075	25	·020	·059
3/0	·372	·008	26	·018	·063
2/0	·348	·0085	27	·0164	·067
0	·324	·009	28	·0148	·078
1	·300	·010	29	·0136	·074
2	·276	·011	30	·0124	·078
3	·252	·012	31	·0116	·082
4	·232	·013	32	·0108	·086
5	·212	·014	33	·010	·090
6	·192	·016	34	·0092	·094
7	·176	·018	35	·0084	·098
8	·160	·020	36	·0076	·102
9	·144	·022	37	·0068	·106
10	·128	·024	38	·006	·112
11	·116	·026	39	·0052	·118
12	·104	·028	40	·0048	·125
13	·092	·030	41	·0044	·132
14	·080	·032	42	·004	·138
15	·072	·034	43	·0036	·146
16	·064	·036	44	·0032	·153
17	·056	·038	45	·0028	·160
18	·048	·040	46	·0024	—
19	·040	·042	47	·002	—
20	·038	·044	48	·0016	—
21	·032	·046	49	·0012	—
22	·028	·048	50	·001	—

METRIC DIMENSIONS FOR ENGINEERING MATERIALS

These follow the "R" series of preferred numbers, rounded to suit the nature of the product. Generally, 1st preference is R10, 2nd is R20 and 3rd is R40. However, both the 'rounding' and the preferences are still (1996) subject to adjustment to suit industrial usage. See also p. **2**.26.

Metric standards for wire and sheet

The figures in brackets show the approximate SWG equivalent. Note that rounding of the third significant figure may differ between wire and sheet.

Preferences							
First	*Second*	*Third*	*First*	*Second*	*Third*	*First*	*Second*
mm	*mm*	*mm*	*mm*	*mm*	*mm*	*mm*	*mm*
0·010			0·100 (42)			1·00 (19)	
	0·011			0·112 (41)			1·12
0·012			0·125 (40)			1·25 (18)	
		0·013			0·132		
	0·014			0·140 (39)			1·40 (17)
		0·015			0·150		
0·016			0·160 (38)			1·60 (16)	
		0·017			0·170		
	0·018			0·180 (37)			1·80 (15)
		0·019			0·190		
0·020			0·200 (36)			2·00 (14)	
		0·021			0·212 (35)		
	0·022			0·224 (34)			2·24 (13)
		0·024			0·236		
0·025 (50)			0·250 (33)			2·50 (12)*	
		0·026			0·265		
	0·028			0·280 (32)			2·80 (11)
		0·030			0·300 (31)		
0·032 (49)			0·315 (30)			3·15 (10)	
		0·034			0·335		
	0·036			0·355 (29)			3·55 (9)
		0·038			0·375 (28)		
0·040 (48)			0·400 (27)*			4·00 (8)	
		0·042			0·425 (27)		
	0·045			0·450 (26)			4·50 (7)
		0·048			0·475		
0·050 (47)			0·500 (25)			5·00 (6)	
		0·053			0·530		
	0·056			0·560 (24)			5·60 (5)*
		0·060			0·600		
0·063 (46)			0·630 (23)		0·670	6·3 (3)*	
	0·071 (45)			0·710 (22)			7·10 (2)
		0·075			0·750		
0·080 (44)			0·800 (21)			8·0 (0)	
		0·085			0·850		
	0·090 (43)			0·900			9·00 (00)
		0·095			0·950 (20)		

Preferred dimensions of drawn bars (round or square) and flats
R10 and R20 only are used. There is at present no third preference.

Choice		*Choice*		*Choice*		*Choice*	
First	*Second*	*First*	*Second*	*First*	*Second*	*First*	*Second*
mm	*mm*	*mm*	*mm*	*mm*	*mm*	*mm*	*mm*
0·10		1·0		10·0		100·0	
	0·11		1·1		11·0		110·0
0·12		1·2		12·0		120·0	
	0·14		1·4		14·0		140·0
0·16		1·6		16·0		160·0	
	0·18		1·8		18·0		180·0
0·20		2·0		20·0		200·0	
	0·22		2·2		22·0		220·0
0·25		2·5		25·0		250·0	
	0·28		2·8		28·0		280·0
0·30		3·0		30·0		300·0	
	0·35		3·5		35·0		
0·40		4·0		40·0			
	0·45		4·5		45·0		
0·50		5·0		50·0			
	0·55		5·5		55·0		
0·60		6·0		60·0			
	0·70		7·0		70·0		
0·80		8·0		80·0			
	0·9		9·0		90·0		

Note the slight differences between "wire" and "bar" below 6 mm.

Hexagon drawn bars, mm sizes
At present these are limited to those dimensions needed for hexagon nuts and bolts. Sizes will be found in the table for ISO metric threads on page **4**.13. Hexagon A-F dimensions shown in the table for Metric Model Engineer threads (MME) on page **4**.8 which do not also appear in the ISO table may be second preference and, at present, difficult to obtain in retail lengths.

SECTION EIGHT
STEAM AND THE STEAM ENGINE

PROPERTIES OF STEAM

Definitions

Saturated steam	Steam at a temperature corresponding to the boiling point of water at that particular pressure.
Dry steam	Steam containing no unevaporated water. 'Dry saturated' steam is both dry and at the boiling or 'saturation' temperature.
Superheated steam	Steam at a temperature above that corresponding to the boiling point at that particular pressure. Superheated steam will always be 'dry'.
Wet steam	Steam containing unevaporated water dispersed as mist through the vapour.
Dryness fraction	The proportion of dry steam present in wet steam. Denoted as a rule by 'q' – a decimal. Thus if $q = 0{\cdot}85$, the 'stuff' contains 85% dry steam and 15% water mist by weight.
'Stuff'	A general term denoting steam of unknown condition.
Condition	Describes the steam as 'wet', 'dry', 'superheated', etc. The word 'quality' is sometimes used, with a numerical value.
Specific volume	The volume of unit mass of steam – quoted in cu. ft/lb, cu. metres/kilogram, etc. Symbol V_S. (For water, the symbol is V_W.)
Specific energy	Sometimes called 'total heat' or 'enthalpy'. The energy content per unit mass of water, steam, or wet vapour. May be stated in heat units, BTU/lb; work units, ft lb/lb or 'energy' units joules/kg. Symbols are 'h' for water, H_S for dry saturated steam, and 'H' for superheated steam.
Volume ratio	The volume of dry saturated steam produced from unit volume of water *at the feed temperature*. For model engineering work this is quoted in cu. in./cu. in., and the feed temperature is taken as 60°F. The ratio is the same if stated in cc/cc. For untabulated values, volume ratio $= V_S/0{\cdot}0161$ in imperial units (approximately $V_S \times 1000$ for SI units).
Specific heat	The energy required to raise one pound or one kilogram of the substance through one degree temperature rise. For water, S = 1 BTU/lb/°F or 4·187 kJ/kg/°C within the limits used by model engineers.
Adiabatic	An expansion or compression process in which no heat energy is gained or lost through the walls of the container. Any work done is equal to the change in energy (heat) content of the contained gas or vapour.
Isothermal	An expansion or compression process in which the temperature remains constant. Heat must pass through the walls of the container to permit this. (As the temperature depends on the pressure an isothermal operation is not possible with saturated steam.)

TABLE 1 SKELETON STEAM TABLES

PRESSURE P			*TEMPERATURE T*		*VOLUME V_S*		*ENERGY*						*Volume ratio*
Absolute		*Gauge*					*h (Water)†*		*L (Latent)*		*H_S (Steam)*		
psi	*bar*	*in. Hg (vac)*	°C	°F	ft^3/lb	M^3/kg	*BTU/lb*	*kJ/kg*	*BTU/lb*	*kJ/kg*	*BTU/lb*	*kJ/kg*	V_S/V_W
0·087	0·006	29·88	0·01*	32·01*	3301	206·1	0	0	1075·7	2501	1075·7	2501	
1	0·07	28	38·7	102	334	20·5	69·7	163	1036·1	2409	1105·8	2572	—
2	0·14	25·9	52·3	126	173·7	10·7	94·0	220	1022·2	2376	1116·2	2596	—
3	0·21	23·9	60·8	142	118·7	7·4	109·4	255	1013·2	2355	1122·6	2610	—
4	0·28	21·8	67·2	153	90·6	5·58	121·0	283	1006·7	2339	1127·7	2622	—
5	0·34	19·8	72·4	162	73·5	4·65	130·2	302	1001·6	2328	1131·8	2630	—
6	0·41	17·8	76·7	170	62·0	3·87	138·1	321	996·6	2316	1134·7	2637	—
7	0·48	15·7	80·5	177	53·6	3·37	144·9	336	992·2	2308	1137·1	2644	—
8	0·55	13·7	83·7	183	47·4	2·96	151·0	351	988·5	2298	1139·5	2649	—
9	0·62	11·6	86·8	188	42·4	2·65	156·5	364	985·2	2292	1141·7	2656	—
10	0·69	9·6	89·6	193	38·4	2·40	161·3	375	982·5	2885	1143·8	2660	—
12	0·83	5·5	94·4	202	32·4	2·02	170·1	386	976·9	2272	1147·0	2668	—
14	0·97	1·4	98·7	210	28·0	1·75	177·7	413	972·2	2261	1149·9	2674	—
14·697	1·013	p.s.i. below	100·0	212·0	26·8	1·67	180·1	419	970·6	2257	1150·7	2676	1665
20	1·4	5·3	109	228	20·1	1·24	196·3	456	960·4	2234	1156·7	2690	1248
25	1·7	10·3	116	240	16·3	1·03	208·6	485	952·5	2215	1161·1	2700	1012
30	2·1	15·3	121	250	13·7	0·846	219·0	500	945·6	2199	1164·6	2709	851
35	2·4	20·3	126	259	11·9	0·747	228·0	530	939·6	2185	1167·6	2716	739
40	2·8	25·3	131	267	10·5	0·646	236·1	549	934·4	2173	1170·5	2722	652
45	3·1	30·3	135	275	9·4	0·587	243·5	566	929·2	2161	1172·7	2727	584
50	3·4	35·3	138	281	8·52	0·532	250·2	582	924·6	2150	1174·8	2732	529
55	3·8	40·3	142	287	7·79	0·486	256·4	596	921·0	2144	1177·3	2738	484
60	4·1	45·3	145	293	7·18	0·448	263·2	610	916·2	2131	1178·4	2741	446
65	4·5	50·3	148	298	6·66	0·414	267·5	622	912·3	2122	1179·8	2744	414

*Indicates water (liquid) at the melting point of ice. †From 32°F or 0°C.

TABLE 1 SKELETON STEAM TABLES (Contd)

PRESSURE P			*TEMPERATURE T*		*VOLUME V_S*		*ENERGY*						*Volume ratio*
Absolute		*Gauge*					*h (Water)*†		*L (Latent)*		*H_S (Steam)*		
psi	*bar*	*psi*	*°C*	*°F*	*ft^3/lb*	*M^3/kg*	*BTU/lb*	*kJ/kg*	*BTU/lb*	*kJ/kg*	*BTU/lb*	*kJ/kg*	*V_S/V_W*
70	4·8	55·3	151	303	6·21	0·388	272·7	634	908·7	2114	1181·4	2748	386
75	5·2	60·3	153	308	5·81	0·363	277·4	645	905·3	2106	1182·7	2751	361
80	5·5	65·3	156	312	5·47	0·343	281·2	656	901·9	2098	1184·0	2754	340
85	5·9	70·3	158	316	5·28	0·330	286·4	666	898·7	2090	1185·1	2756	328
90	6·2	75·3	160	320	4·90	0·306	290·7	676	895·5	2083	1186·2	2759	304
95	6·6	80·3	162	324	4·66	0·291	294·7	685	892·5	2076	1187·2	2761	289
100	6·9	85·3	164	328	4·43	0·277	298·5	694	889·7	2069	1188·2	2763	275
105	7·2	90·3	166	331	4·23	0·264	302·2	703	886·9	2063	1189·1	2766	263
110	7·6	95·3	168	335	4·05	0·253	305·7	711	884·2	2056	1189·9	2767	252
115	7·9	100·3	170	338	3·88	0·242	309·2	719	881·5	2050	1190·7	2769	241
120	8·3	105·3	172	341	3·73	0·233	312·5	727	878·9	2044	1191·4	2771	232
125	8·6	110·3	174	344	3·59	0·224	315·7	734	876·4	2038	1192·1	2773	223
130	9·0	115·3	175	347	3·46	0·215	318·8	741	874·0	2033	1192·8	2774	215
135	9·3	120·3	177	350	3·34	0·208	321·9	749	871·5	2027	1193·4	2776	207
140	9·7	125·3	178	353	3·22	0·201	324·9	756	869·1	2021	1194·0	2777	200
145	10·0	130·3	180	356	3·12	0·195	327·8	762	866·8	2016	1194·6	2778	194
150	10·3	135·3	181	358	3·02	0·189	330·6	769	864·5	2011	1195·1	2780	188
155	10·7	140·3	183	361	2·92	0·182	333·3	775	862·3	2006	1195·6	2781	181
160	11·0	145·3	184	363	2·84	0·177	336·0	781	860·1	2001	1196·1	2782	176
165	11·3	150·3	186	366	2·75	0·172	338·6	787	858·0	1996	1196·6	2783	171
175	12·1	160·3	188	371	2·60	0·162	343·7	799	853·9	1986	1197·6	2785	161
185	12·8	170·3	191	375	2·47	0·154	348·5	811	849·9	1976	1198·4	2787	153
195	13·4	180·3	193	380	2·35	0·147	353·2	821	846·0	1968	1199·2	2789	146
205	14·1	190·3	196	384	2·24	0·140	357·8	832	842·0	1958	1199·8	2790	139
215	14·8	200·3	198	388	2·14	0·134	362·1	842	838·3	1950	1200·4	2792	133

†From water at 32°F or 0°C.

THE STEAM TABLES

Because steam, even when superheated, is a *vapour* rather than a gas its behaviour cannot be predicted by using the 'Gas Laws'. For that reason, its properties have been *tabulated*, using a combination of complex thermodynamic theory and experiment. Complete tables can be purchased very cheaply, but 'skeleton' tables are given on pp. **8**.2 and **8**.3 for 'saturated' steam and on p. **8**.5–**8**.6 for superheated steam.

The following data are presented in Table 1 for **Saturated steam**.

Pressure, P. Both 'absolute' and 'gauge', in lbf/sq. in. and bar, but vacuum is quoted in in. mercury.

Temperature, T. In both °F and °C.

Specific volume, V_S. Volume of unit mass – ft^3/lb or M^3/kg.

Energy content (sometimes called 'enthalpy') in BTU/lb or kJ/kg

(a) Of water, **h**. Includes both that due to the temperature rise from 0°C or 32°F *and* that due to the feed pump work.

(b) Of evaporation ('latent heat') **L**. Energy required to convert water into steam.

(c) Total H_S = h + L. Called 'total heat' or 'total enthalpy'.

Volume ratio, V_S/V_W. A 'model engineer's' concept, being the volume of steam raised from unit volume of water. Any units can be used.

Data for **Superheated steam** is presented in Table 2; at 2A in ft lb second units and at 2B in SI units. The column headings give the 'total temperature' (i.e. the actual temperature) of the steam in °F or °C as appropriate.

Pressure. Given either in lbf/sq. in. or in bar. In 'B' the figures in lbf/sq. in. are approximate.

Specific volume, V_S, in cu. ft/lb or cu. m/kg.

Energy content, H, in BTU/lb or kJ/kg. The difference between this figure and H_S in Table 1 is the amount of heat needed to superheat the steam.

Using the steam tables

To use the tables it is only necessary to look up the operating pressure – bearing in mind that model pressure gauges are seldom likely to be accurate within better than 15% or so – and read off the quantity needed. Thus, to find the heat required to evaporate 1 lb of steam at 80 lb sq. in. gauge from feed-water at 75°F. The nearest 'gauge' reading is 80·3, and we can ignore the difference as the instrument won't read to 0·3 lb sq. in. The total heat, H_S, is given as 1187·2 BTU/lb. However, the water is already at 75°F, so this will contain some heat already. The 'specific heat' of water is 1, (4·19 kJ/kg°C in SI) so the heat content of the feed is (75 – 32) × 1 = 43 BTU/lb. Hence the boiler must provide 1187·2 – 43 = 1144·2 say 1144 BTU/lb. This ignores the work you may have done with a hand feed-pump, but this is too small to be relevant at this temperature. You would use column 'h' at higher temperatures.

Suppose the steam to be wet – a normal boiler will provide steam at about 0·88 dryness fraction. In this case the boiler only provides 'q' times the latent heat, q being the dryness fraction. So, the heat content of the steam will be (h + q.L), h is 284·7, L is 892·5. Thus the heat content of the wet steam is (0·88 × 892·5 + 284·7) = 1070·1. Subtracting the 43 BTU for the feed heat as before, the heat provided will be 1070·1 – 43 = 1027·1, say 1027 BTU/lb.

To find the volume of the steam in the two cases, read down the V_S column, to see that one pound of steam at 80·3 occupies 4·66 cu. ft/lb. In the case of wet steam, the volume occupied by the water may be ignored (it is less than 0·05%) but the actual steam present is only q × Vs = 0·88 × 4·66 = 4·1 cu. ft/lb.

In terms of 'volume ratio', one cu. in. of cold feed will produce 289 cu. in. dry steam, or 0·88 × 289 = 254 cu. in. of steam at dryness fraction 0·88.

Superheated steam

Table 2 gives figures which show both the heat content and the specific volume of steam heated above the saturation temperature. The temperatures shown are the *total* temperature – the actual temperature of the steam; superheat is sometimes quoted in 'degrees of superheat', which is the temperature *above* the saturation temperature. Thus steam at 100 lb sq. in. abs with 100°F **OF** superheat will be at 328 + 100 = 428°F (the 328 is read from the temperature column of the steam tables).

To find the heat supplied *by the superheater* for steam at 85 lb sq. in. at 500°F (a not untypical figure for a 5″ gauge loco) we see that H = 1279·1 BTU/lb. If the boiler supplies steam 0·88 dry as before, then the heat/lb entering the superheater will be 1070·1 from previous work. Hence the superheater tubes will have to supply 1279·1 – 1070·1 = 209 BTU/lb. (Note that the superheater provides about 16% of the total heat in the steam in this case.) The volume of the superheated steam is now 5·59 cu. ft/lb, having expanded

PROPERTIES OF SUPERHEATED STEAM

TABLE 2A
H in BTU/lb; V in cu. ft/lb

Pressure abs. lbf/sq. in.		*Steam Temperature, °F*				
		300	*400*	*500*	*600*	*700*
20	V	22·36	25·43	28·46	31·47	—
(5·3 gauge)	H	1191·6	1239·2	1286·6	1334·4	—
40	V	11·04	12·63	14·17	15·69	—
(25·3)	H	1186·8	1236·5	1284·8	1333·1	—
60	V	—	8·36	9·41	10·43	11·44
(45·3)	H	—	1233·6	1283·0	1331·8	1380·9
80	V	—	6·22	7·02	7·80	8·56
(65·3)	H	—	1230·7	1281·8	1330·5	1379·9
100	V	—	4·94	5·59	6·22	6·84
(85·3)	H	—	1227·6	1279·1	1329·1	1378·9
150	V	—	3·22	3·68	4·11	4·53
(135·3)	H	—	1219·4	1274·1	1325·7	1376·3
200	V	—	2·36	2·73	3·06	3·38
(185·3)	H	—	1210·3	1268·9	1322·1	1373·6

For intermediate values plot adjacent values either side on squared paper and read the desired figures.

TABLE 2B SI UNITS

P in bar; V in M^3/kg; H in kJ/kg (lbf/in.^{2} approximate)

Abs press			*Actual steam temperature °C*					*App. gauge press*	
bar	*lbf/in.^{2}*		*150*	*200*	*250*	*300*	*350*	*bar*	*lbf/in.^{2}*
2·0	(29)	V	0·96	1·08	1·20	1·32	—	1·0	(15)
		H	2770	2871	2971	3072	—		
3·0	(44)	V	0·63	0·72	0·80	0·88	—	2·0	(29)
		H	2762	2866	2968	3070	—		
4·0	(58)	V	0·47	0·53	0·60	0·65	0·71	3·0	(44)
		H	2753	2862	2965	3067	3170		
5·0	(74)	V	—	0·43	0·47	0·52	0·57	4·0	(46)
		H	—	2857	2962	3065	3168		
6·0	(88)	V	—	0·35	0·39	0·43	0·47	5·0	(74)
		H	—	2851	2958	3062	3166		
7·0	(103)	V	—	0·30	0·34	0·37	0·41	6·0	(88)
		H	—	2846	2955	3060	3164		
8·0	(117)	V	—	0·26	0·29	0·32	0·35	7·0	(103)
		H	—	2840	2951	3057	3162		
10·0	(145)		—	0·21	0·23	0·26	0·28	9·0	(132)
			—	2829	2944	3052	3158		

For intermediate values plot adjacent value on squared paper and read the desired figures.

from the 4·1 cu. ft in its wet state as it entered the wet header. (In passing, this means that the velocity of the steam leaving the superheater will be about 35% higher than at entry.)

In using the tables it may well be that figures are needed for values between those tabulated (e.g. for 100 lb sq. in. gauge, not shown in the superheat table). There will be a slight error if the figure is obtained by dividing the adjacent tabulations in proportion but it is unlikely to be significant in model work. Thus H for steam at 114·7 abs i.e. 100 gauge and 500°F works out this way at about 1277·6 BTU/lb. The correct figure is 1277·55 – a very small error. However, if it is desired to obtain more exact results a curve can be plotted on squared paper and the intermediate figures obtained that way. Note that for a given temperature the heat content of superheated steam *diminishes* as the pressure rises. (Specific heat of steam may be taken as 0·5 BTU/(lb°F) or 2·1 kJ/(kg°C) for rough calculations.) The procedure is identical when working in SI units, using °C and kJ/kg.

STEAM IN THE STEAM ENGINE

The work done in the cylinder of an engine is derived from the heat in the steam. The pressure changes are a *symptom* of the conversion of heat to work, not a cause; the engine is a *heat* engine. Suppose an engine accepts steam at 120 lb sq. in. abs, and expands this until it is reduced to 60 lb sq. in. abs. The value of H_S at 120 is 1191·4 BTU/lb. If the work done is equivalent to 60 BTU/lb (which is about right for a perfect engine with no leakage or heat loss to the cylinder) then the heat content at exhaust will be 1131·4 BTU/lb. But at 60 lb sq. in. the heat content of dry steam is 1178·4 BTU/lb; it follows, therefore, that the steam cannot be dry at exhaust – it is, in fact, about $q = 0{\cdot}95$. If the steam were initially wet, as from a normal boiler, the final condition might be as wet as $q = 0{\cdot}8$ or 20% water. This **quite apart from any cooling effect of the cylinder walls**.

The following table shows the final condition of the steam for an engine taking dry steam at 120 lb sq. in. abs, for various *true* expansion ratios (allowing for clearance volume, that is; the cutoff will be slightly earlier than the figures suggest) and also indicates the steam superheat temperature that would be necessary to ensure that steam remains dry throughout the stroke. *This does not allow for heat losses*; still higher superheat would be necessary to correct for that.

Exp. ratio	10/1	4/1	2·5/1	2/1	1·4/1
Final dryness, q	0·89	0·93	0·96	0·97	0·98
Steam temp. needed, °F	630	525	450	425	385
Cutoff (ignoring clearance)	10%	25%	40%	50%	70%

This reduction in steam condition is an inevitable consequence of work having been done and no amount of cylinder lagging will prevent it. It seems probable that the presence of this inherent wetness during expansion may accelerate condensation on the cylinder walls, owing to the presence of water droplets throughout the steam.

Heat to condenser

The heat to be absorbed by the condenser must be calculated on the steam condition at *release* – i.e. at the end of expansion, not at that of the back pressure. Thus with an engine operating as in the previous paragraph, cutting off at 50% of the stroke, the pressure at the end of expansion will be about 60 lb sq. in. abs, say 0·95 dry. The heat to be absorbed by the condenser will be $(h + qL)_{60}$ which is $262 + (0{\cdot}95 \times 916{\cdot}2) = 1132$ BTU/lb. But less, of course, by the temperature of the hot-well; say 1045 BTU/lb, allowing this to be 120°F. This is somewhat larger than the usual published figures, the latter being usually for steam turbines, which expand the steam right down to the back-pressure.

A useful overall factor of heat transmission through condenser tubes can be taken as 550 to 600 BTU per hour/sq. ft of tube area/deg F (say 11·3 to 12·3 MJ/hr m^2°C) difference in temperature between steam and mean water temperature. The water velocity through the tubes should be as high as possible – 5 to 6 ft/second (2 m/sec) is usual. The OD of the tube should be used in estimating the heat transmission area. The water quantity should be calculated so that there is a temperature rise of about 10–15°F, the mean of this figure being used in calculating 'temperature difference'; the steam temperature being, of course, that of the desired vacuum. (There is little point in using the concept of 'Log Mean Temperature Difference' in a model condenser, especially on a reciprocating engine.) The most important reason for poor performance of model condensers is the presence of oil on the steam side of the tubes, and the interior should be degreased frequently. Carbon tetrachloride will be found most effective.

Air pumps

Air leaks into the system will 'lock up' any condenser and destroy the vacuum, so that an air pump is essential. The same pump usually doubles as a condensate pump. 'Scale' pumps usually perform quite satisfactorily, but Mr K. N. Harris recommended that for a surface condenser the pump should have a capacity equal to about one sixth of the swept volume of the low-pressure cylinder of the engine. Somewhat larger pumps would be needed for jet condensers as they must handle both the condensed steam and air from the engine *and* the injected condensing water. 'Common' single acting pumps for jet condensers could well be made about one fifth of the capacity of the LP cylinder. For high-speed engines it is desirable to operate the air pump at a lower speed than the engine it serves.

THE SLIDE VALVE DIAGRAM

The lower figure in the sketch opposite shows an ordinary slide valve at mid-travel. The valve overlaps the port on the steam side by an amount known as the 'steam lap' and on the exhaust side to provide 'exhaust lap'. The latter may be, and in models usually is, zero. The amount of these laps, together with the angle of advance (the angle by which the eccentric exceeds 90° ahead of the crank) governs the timing of the steam distribution to the cylinder. The eccentric travel also has some effect. The influence of these dimensions can be studied and, in design, predicted, by the use of the 'valve diagram'. One form of this is shown in the upper figure sketch opposite, invented by M. Reuleaux.

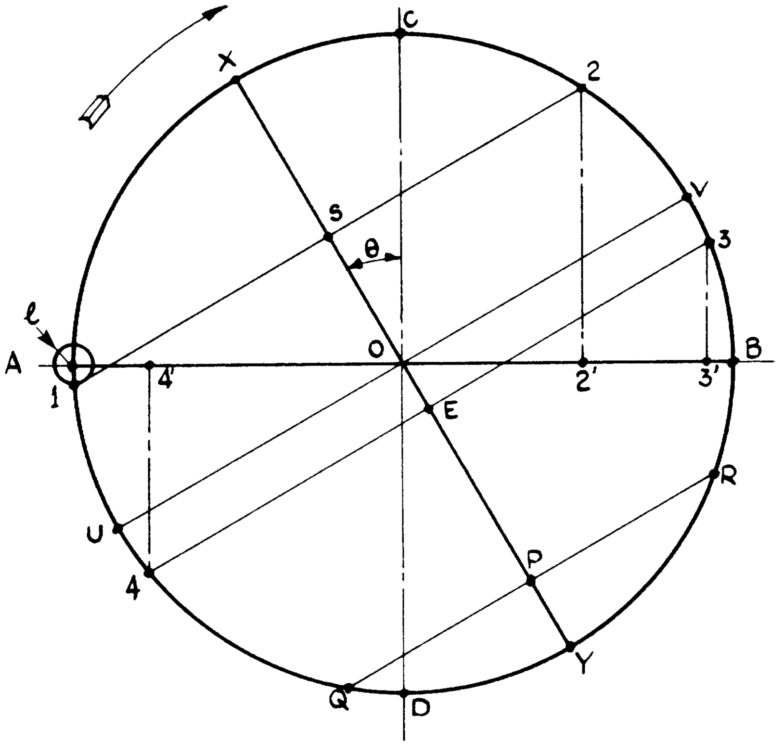
On two centrelines at right angles, AB and CD, is drawn a circle, centre O, of any arbitrary diameter. Five inches is convenient, ten inches more accurate. Then AB represents the piston travel to scale; if AB is 5″ and the stroke 2″, this means that 1″ on AB represents 0·4″ or 20% of piston travel. The line XOY is drawn at an angle θ – the 'angle of advance' – to CD. This represents the valve travel, to a scale which must be determined. Thus, if the valve travel is 0·8″ and the circle is 5″ dia., one inch on XY = 0·8/5 or 0·16″ of valve travel. Draw UOV at right angles.

The distance OS represents the steam lap to this scale, and OE the exhaust lap, if any; if this is zero, then the line 4–3 coincides with UV. OP is made equal (to scale) to the width of the steam port. The lines 1S2, UV, 4E3 and QPR are all parallel, of course, and at right angles to XOY. The arrow indicates the succession of events – A being inner dead centre of the crank.

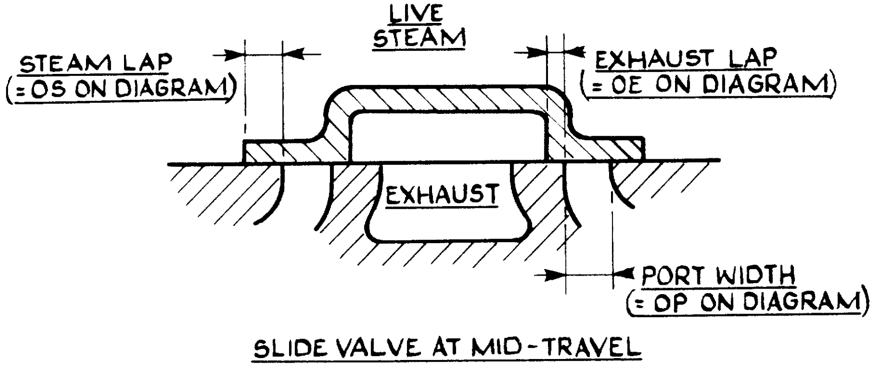
The point marked 1 is the point of steam admission – i.e. when the edge of the valve just uncovers the cylinder port to steam. 2 indicates the moment of 'cutoff' when the valve closes to steam. 3 is 'release' when the exhaust edge of the valve uncovers the port, and 4 indicates the closure of the port to exhaust, or the start of compression. The small circle, centre at A, has a radius equal to the 'lead' of the valve; that is, the amount the valve is open to steam at inner dead centre. All these dimensions are to be measured using the scale appropriate to the valve travel, as explained above.

If perpendiculars are dropped from these points onto the line AOB as shown at 2–2′, 3–3′ etc., the intersections with AOB show the position of the *piston* when the various events take place; i.e. cutoff is at 2′, and A2′ divided by AB gives this as a fraction of the stroke. Release occurs at 3′ and compression at 4′. The point 1 is so close to A that the difference is undetectable on a small diagram, and scarcely to be seen even if the circle were 20″ diameter. The diagram provides two more pieces of information. The distance SX shows the maximum opening of the port to steam – this is almost invariably less than the full port width; cutoff would have to be very late indeed for the maximum opening to be equal to the port size. On the other hand, the port is full open to exhaust for quite a proportion of the stroke – from the point R to Q; this is to be desired. 'Any fool can get steam into a cylinder – it requires a good deal of cunning to get it out again'.

Two examples may illustrate the use of the diagram.



REULEAUX VALVE DIAGRAM



SLIDE VALVE AT MID-TRAVEL

Example i
Given the valve travel, the desired point of cutoff, and a desired lead of the valve, find the angle of advance of the eccentric and the required valve lap.

Draw the lines AB and CD and the circle to some convenient diameter, and establish the scale of valve travel as explained above. Draw the small circle radius '*l*' to this scale. Mark the point of cutoff 2′ on AOB and erect a perpendicular to meet the circle at 2. Draw the line 1–2, tangential to the lead circle. Draw XOY at right angles to this line; measure the angle θ, which gives the required angle of advance. Measure OS to find the necessary steam lap.

Example ii
To determine the lead to be set on the valve to provide the correct cutoff, with a given steam lap, and to find the angle of advance.

Draw the circle and lines AB, CD. Mark off the point 2′ and from this find the point 2. Draw a circle centre 0 and radius (to scale) equal to the steam lap. Draw a line tangential to this circle to meet the point 2, and produce to meet the main circle at the point 1. Measure the radius '1' – this gives the necessary lead, the scale being that relating to the valve travel. Draw XOY at right angles to the line 1–2, this line passing, of course, through the point where 1–2 meets the construction circle. The angle between XOY and CD is the required angle of advance.

Other uses will no doubt occur to readers after a bit of practice.

The true points of cutoff, release, compression, etc., will differ slightly from those shown by this diagram, as it does not take account of the effects of the angularity of the engine connecting rod. For an account of means to allow for this see *Model Engineer* Volume 142, 2nd July 1976, but for most model work the simple analysis outlined above will suffice. Other diagrams are used for similar purposes, notably the 'Zeuner' and the 'Bilgram', the latter being very useful when designing from scratch. Details of these can be found in most early books on heat engines, notably that by Prof. D. A. Low, published in 1930. It will be found that a study of some existing valves using the simple Reuleaux diagram will help a great deal in understanding the way in which changes in travel, angle, and lap affect the engine, and readers are advised to do this before going on to use the Zeuner or the Bilgram which, though in many ways more informative, are certainly more difficult to construct.

INDICATOR DIAGRAMS

The indicator diagram, a graphical record of the pressure and volume changes in an engine cylinder, is commonplace in full size engineering. Unfortunately the small size of model cylinders has hitherto made the use of a cylinder pressure indicator very difficult, so that much valuable data has been denied to us.

However, experiments currently (1995) in hand, especially those by Prof. W. B. Hall, suggest that the indicating of model cylinder processes may soon be much easier. This being so it seems appropriate to pay some attention to the matter – even if only in *anticipation* of later developments.

The 'indicator diagram' is a simple plot of the cylinder pressure against stroke volume, and Fig. 1 shows how this might appear in an 'ideal' cylinder having a clearance volume at the end of the stroke and with instantaneous valve events.

Steam admission occurs at 'a', and the pressure rises instantaneously to the steam-chest pressure at 'b'. Admission continues until 'c', the 'point of cutoff' after which the cylinder

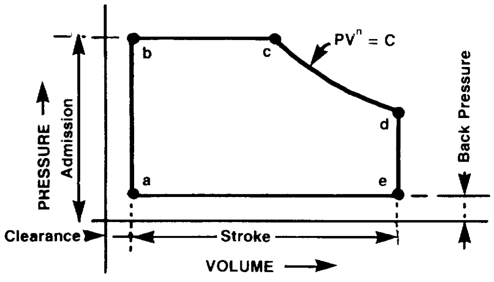


Fig. 1

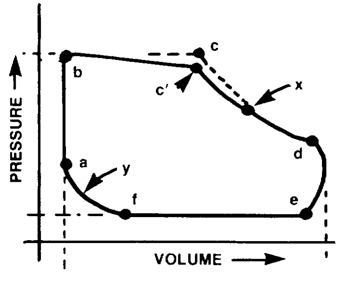


Fig. 2

contents expand following a law of the form $PV^n = C$.* At 'd' the exhaust port opens, and the pressure drops suddenly to 'e', the back pressure, which may be above or below atmospheric. (All pressures must be 'absolute'.) This diagram gives a graphical picture of what *should* happen in the cylinder, but as we shall see, the picture changes in real engines, large and small.

Now, if the pressure scale were in lbf/ft^2 (or $N.m^2$) and the volumes in cubic feet or cubic metres, then the *area* of the diagram would, to scale, represent ft lbf or newton.metres. That is, the area represents the *work done per stroke*. Such a calculation, done on a real engine, would enable the *indicated horsepower* to be calculated, and this, compared with the measured *developed* (or 'brake') HP enables the engineer to calculate the *mechanical efficiency* = BHP/IHP.

However, the 'real' indicator diagram displays marked differences. Fig. 2 shows one which might be obtained from a medium speed engine. In this case the exhaust port is closed early, at 'f', and from 'f' to 'a' the remaining exhaust steam is *compressed* into the clearance volume. Admission starts at 'a', but instead of running from 'b' to 'c' there is a fall in pressure, c—c'. This is due to 'wire-drawing' – friction in the valve and port. Expansion follows down to 'd' where the exhaust port opens (before the dead centre) but again, port friction results in a curve down to 'e' rather than a straight line.

Mean effective pressure

As before, the area of the diagram represents 'work done', and this area may be obtained by using a 'planimeter' (if you have such an instrument!) or by using Simpson's Rule (see page **2**.33). The ordinates should be fairly close together, and be measured accurately with dividers.

If this area is divided by the length of the outline on the diagram we get the height of a rectangle which would have identical area. This height, multiplied by the pressure scale, gives the *Indicated Mean Effective Pressure* (IMEP, or P_m) of the diagram, in lbf/sq. in. or N/sq. mm, depending on the units used. This is the pressure which, if uniform through the stroke, would produce the same net work as that of the diagram in Fig. 2.

*The value of 'n' for steam in this context is about **1·03**, but it is usual to assume that expansion is 'hyperbolic' – e.g. PV = constant. Note that this is *not* Boyles Law, as the temperature changes throughout the expansion.

The indicated power of the engine can then be obtained by writing:

$$IHP = (P_m \times L \times A \times N)/33\,000 \;\ldots\ldots\ldots\; (1)$$

P_m = IMEP in lbf/sq. in.
L = Stroke, in *feet*
A = Cyl. area, sq. in.
N = No. of strokes/min.

Note that 'N' must allow for both 'double acting' and number of cylinders.

In SI units, P_m will be in N/mm^2; A in sq. mm; L in metres (mm/1000) and N in strokes/*second*. Then Exp. (1) becomes:

$$P \times L \times A \times N = \text{output in } \mathbf{watts} \;\ldots\ldots\; (2)$$

Estimation of steam consumption

A very rough 'guesstimate' of the steam consumption can be made from the indicator diagram. Measure the absolute pressure and volume at a point about 'x' on Fig. 2. From the steam tables the specific volume can be found and the mass present calculated. Repeat this for the point 'y' on the compression curve. This is the mass of steam left in the cylinder after exhaust closing. Subtract this from the first figure, to give the amount of 'cylinder feed steam' per stroke. From this, lb/hr can be calculated.

This will not correspond with the *actual* value (as, for example, that measured by condensing the exhaust) because the steam at 'x' is almost certainly not in the 'dry saturated' condition, due to cylinder condensation. (The condition of the steam at 'y' is less important.) On full-size engines the discrepancy would be of the order of 75% – i.e. the estimated cylinder feed would be 75% of the actual. On models a safer figure would be 50%, but the 'guess' *can* give a guide to the size of boiler needed.

Features of the cylinder events

(1) *Lead*. This is the advance of the point of admission 'a' ahead of dead centre (see diagram on page **8**.9). The object is to get the steam port open as quickly as possible. Too little may result in excess wire-drawing, but too much can cause 'kick-back'. Slow-speed engines require very little indeed. That required on high-speed engines (especially model locos) is a matter for considerable debate, as will be seen in the correspondence columns of the model magazines! Trial and experiment is needed.

(2) *Cutoff*. This governs the 'expansion ratio' on which the thermodynamic efficiency of all steam engines depends. Early cutoff gives a larger expansion ratio, hence a higher efficiency. However, it also reduces the area of the diagram, so reducing the power per unit volume of cylinder. It will also be appreciated that if the cutoff is too early (less than 50% stroke) starting may be difficult. A compromise is always needed here.

(*Note*: The true expansion ratio will *not* be the same as the ratio cutoff/stroke, for the clearance volume must be allowed for. If the clearance is 10%, and cutoff at 50%, then the expansion ratio will be 110/60 = 1·83/1, not 2/1.)

All engines need some control of steam supply to match power demand. This may be automatic, with a governor, or manual, as by a locomotive driver. There are two main methods – see Fig. 3.

At 'a' is the diagram of an engine having cutoff at about 50% of stroke, on full power. At 'b' power has been reduced by throttling the steam supply, so reducing the diagram height and hence its area. Cutoff is still at 50%. At 'c' there is no throttling; steam supply is still at full pressure, but the cutoff has been advanced to 25% of stroke. (Allowing for 10% clearance, the expansion ratio has been raised

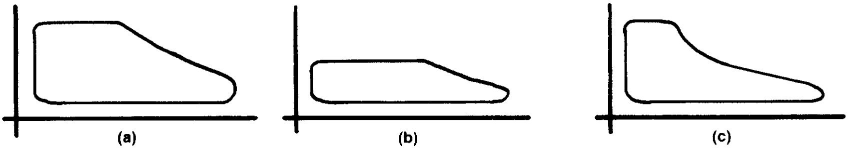


Fig. 3

from $1 \cdot 83/1$ – see above – to $3 \cdot 14/1$.) Again, the area of the diagram has been reduced.

Both methods have their advantages; 'b' is very simply done – with a throttle – whilst 'c' is more economical. The steam is used more efficiently, though a more elaborate valve-gear is needed. (Fortunately, the need for reversal provides this on road and rail locos!)

(3) *Clearance volume*. This is necessary for mechanical reasons in the cylinder itself, but much of it arises from the volumes of the ports between valve and cylinder. Clearance volume is malignant from a thermodynamic point of view, as it increases the cylinder feed to no purpose, and as 'volume' must be enclosed by an 'area' the rate of heat loss is increased.

(4) *Compression*. Thermodynamically, compression is very desirable, as the steam temperature in the clearance is increased, so reducing initial condensation of the cylinder feed. Further, the amount of cylinder feed is reduced; in effect, the exhaust steam in the clearance is being recycled to do more work. However, if compression is carried too far the clearance pressure may exceed the chest pressure, especially when throttle governing is used. A compression pressure of from 15 to 25% of chest pressure will serve for most models.

Compression is sometimes known as 'cushioning', because the pressure rise helps to decelerate the piston. This can be important at the bottom end of vertical cylinders, especially large ones, but is not a very serious consideration for models.

(5) *Wire-drawing*. This is clearly of importance, both at admission and at exhaust. However, the steam speeds through model ports are relatively low, and provided the finish on the passages is fair there is little cause for concern except in extreme cases. Restriction of the exhaust, however, will reduce both efficiency and performance.

Conclusion

It will be appreciated that the timing events which govern the shape of the indicator diagram are, to some extent, inter-related; study of the Reuleaux diagram (page **8**.9) shows this. Many forms of valve gear have been devised to enable one or more events to be fixed whilst varying the timing of others, but consideration of this subject is beyond the scope of this book. A good summary of types will be found in *Heat Engines*, by Prof. D. A. Low, obtainable from public libraries.

Size of steam pipes

Piston speeds in model engines are well below those for their full-size prototypes, so that steam speeds in 'scale' pipes will be reduced also. 'Full-sized' steam speeds are seldom met with in models. Messrs Stuart Turner Ltd suggest the following pipe sizes for their models:

Bore in.	*Stroke in.*	*Steam pipe OD*	*Exhaust pipe OD*	*Remarks*
2	2	$\frac{3}{8}''$	$\frac{1}{2}'' \times 18$ g	
$2\frac{1}{4}$	2	$\frac{3}{8}''$	$\frac{1}{2}''$	
$1\frac{1}{2}$	$1\frac{1}{2}$	$\frac{5}{16}''$	$\frac{3}{8}'' \times 18$ g	
1	1	$\frac{1}{4}''$	$\frac{3}{8}''$	2-cyl. SA 2800 rpm
$\frac{3}{4}$, $1\frac{1}{4}$, $1\frac{3}{4}$	1	$\frac{1}{4}''$	$\frac{5}{16}'' \times 18$ g	Triple expansion
1	$\frac{7}{8}$	$\frac{3}{16}''$	$\frac{5}{16}''$	Twin marine engine
$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{32}''$	$\frac{1}{4}'' \times 20$ g	
1	2	$\frac{5}{32}''$	$\frac{1}{4}''$	Slow speed beam engine

When designing models, estimate the bore of the *full-size* pipe to carry steam at 100 ft/sec. for exhaust, and scale down. For modern installations using high-pressure superheated steam, speeds of 120–150 ft/sec. might be used. Undersize steam supply pipes (within reason) will do little harm, but exhaust steam piping should always be to scale size in the bore.

Velocity of steam from a nozzle

With a purely convergent nozzle the steam velocity will increase until the supply pressure is about 1·7 times the back pressure – i.e. about 25 lb sq. in. abs for a nozzle discharging to atmosphere. At or above this point the jet velocity is the speed of sound in steam at the discharge condition – about 1400 to 1500 ft/sec. – and will not rise no matter how high the pressure is raised. This situation also applies to discharge through safety valves and any free discharge of steam. The excess energy available as the pressure is increased is all dissipated in internal friction and eddies.

If the nozzle is of the convergent-divergent type, as used in many steam turbines, the velocity *will* increase further in the divergent portion. Approximate velocities achieved at the exit from such a nozzle are:

Initial pressure abs		*Steam speed, discharging to atmosphere*		*Steam speed, to vacuum 28 in. Hg (710 mm)*	
lb/in.^{2}	*bar*	*ft/sec.*	*m/sec.*	*ft/sec.*	*m/sec.*
10	0·7	—	—	2675	815
15	1·03	low	—	2900	885
30	2·07	1605	410	3265	995
50	3·45	2110	645	3510	1150
75	5·2	2425	740	3670	1120
100	6·9	2645	805	3805	1160
125	8·6	2775	845	3905	1190

The exact speed for any pressure ratio can only be determined by the use of Mollier diagrams for steam, outside the scope of this handbook. If such a diagram is available, the ideal velocity may be found from

$$V\ (\text{ft/sec.}) = 223\cdot7\ \sqrt{H},\quad H \text{ in BThU/lb.}$$
$$V\ (\text{m/sec.}) = 44\cdot72\ \sqrt{H},\quad H \text{ in kJ/kg.}$$
$$V\ (\text{m/sec.}) = 1\cdot414\ \sqrt{H},\quad H \text{ in J/kg.}$$

in each case the value of the 'adiabatic heat drop', H, being found from the chart. The actual velocity will be somewhat less due to friction and heat losses – typically 0·9 of the 'ideal' velocity found from the above.

SECTION NINE
AIR AND GASES
A SUMMARY OF FORMULAE USED IN IC ENGINE CALCULATIONS

SUMMARY OF THE GAS LAWS

This section may be of use to those experimenting with IC and hot air engines, air compressors, and the like. Of necessity the subject is dealt with only cursorily, and those requiring deeper knowledge must face a fairly intensive study of thermodynamics, preferably under expert tuition.

The behaviour of gases can best be understood by imagining the molecules which compose the gas to behave like perfectly elastic spheres in constant random motion. Energy will be transmitted from molecule to molecule without loss by 'elastic collision' and similar collisions with the walls of the containing vessel will exert a 'force' which we recognise as a 'pressure'. The energy possessed by the gas – heat energy – is held by the molecules in the form of energy of motion or kinetic energy; this kinetic energy can be shown to be equal to a constant times the *absolute temperature* of the gas (see page **1**.11) and is what we know as 'heat' energy or 'enthalpy'.

If the gas is contained in a closed vessel and we apply heat, the temperature will rise; the velocity of the molecules will increase. There will be more frequent collisions between them and the walls of the vessel, and these collisions will be at a higher velocity; thus the force on the walls will increase – i.e. the pressure will go up. If we now diminish the volume of the vessel – in other words compress the gas – then again the number of impacts will increase as the molecules have less distance to travel; in addition, because we have done some work in compressing the gas we have increased the energy contained in the system; hence the energy of the molecules will be augmented, with an increase in velocity. The combined effect is an increase in the force exerted on the walls of the vessel – the 'pressure will rise'.

Because we are dealing with a vast multitude of molecules (something like 27×10^{18} per cu. cm) and an enormous number of collisions (about 5000 million/second at 0°C) and the mean velocity (at 0°C) is about 1500 m/s the motion is truly random and the behaviour quite 'average' so that pressure is uniform in all directions; further, though the individual molecules may be moving very quickly, they literally 'get nowhere very fast' as they all move in different directions. The molecules will move, even though the gas is stagnant. (The population of the earth is about 4×10^9 and though at least half are moving about at any one time, 'the whole population' does not move.)

The above very much simplified outline of the kinetic theory of gases applies only to *gases*, not to vapours such as steam, refrigerants, and a few gases which can be liquified by pressure, such as 'Calor gas'. For these substances rather complex modifying factors must be applied to the basic kinetic model and most of the necessary properties are nowadays tabulated. (See page **8**.4 for steam.)

Density of gases

Though it appears normal to measure quantities by volume, the mass of gas present is sometimes needed, and weighing a gas presents problems. Fortunately nature has arranged a very easy way to determine the density of a gas, by ensuring that the density (mass per unit volume) is proportional to the 'molecular weight' – the weight of one molecule of gas compared with that of an atom of hydrogen. These molecular weights are accurately known, and those for some of the more common gases are shown in the table. This includes air, which, though a mixture of gases, is given an 'equivalent' m.w.

Gas	**m.w.*
Air	29
Carbon dioxide	44
Carbon monoxide	28
Hydrogen	2
Oxygen	32
Nitrogen	28
Methane	16

*To nearest unit.

The mol

This concept is very simple – it is the *molecular weight of the gas in pounds or kilograms*, depending on the system used. It has a very important property – that this quantity of *any* gas occupies the same volume when at the same temperature and pressure. At 0°C and an absolute pressure of 760 mm Hg, 1·01 bar, or 14·7 lbf/sq. in. this volume is 359 cu. ft for the 'pound mol' and 22·4 cu. metre for the 'kilogram mol'. (Or, sometimes more convenient, 22·4 litre for the 'gram mol'.) The unit is often abbreviated simply to 'mol', but it is only prudent to add the mass unit, to be quite clear.

From this fact we can find the density of any gas quite simply; thus that for air (m.w. 29) the density (at 0°C and 14·7 lbf/sq. in.) is 29/359 = 0·081 lb/cu. ft; or 29/22·4 = 1·29 kg/cu. m. The inverse of these figures gives the 'specific volume' – e.g. 12·38 cu. ft/lb or 0·77 cu. m/kg. It is useful to memorise the volume of the mol in whatever units you are accustomed to use. The effects of temperature and pressure must be taken into account, and are dealt with in the next section.

The general gas equations

Having got this preliminary explanation out of the way we can now look at the way gases behave in numerical terms when pressures, temperatures, etc. are altered. From experiment, as well as by derivation from the kinetic theory outlined above, it can be shown that, for all 'perfect' gases the following relationship holds:

$$P \times V = M \times R \times T \qquad (1)$$

where P = pressure, ... lbf/sq. *foot*; or newton/sq. metre
V = volume, ... cu. ft; or cu. metre
M = mass of gas, ... lb; or kg
T = *absolute* temperature, . °F abs (°F + 460) or °K (°C + 273)
R = a constant

(When working in imperial units pressure must be in lb/sq. **foot** to keep the equation rational, as we are using cu. ft for volumes.)

This 'gas constant', 'R', varies from gas to gas and also differs depending on the units of mass – lb or kg. However, if we insert *molar quantities* in equation (1) the product M × R becomes a constant which is the same of *all* gases, though differing between imperial and SI units. Try it for yourself: put in $P = 2117$ lb/ft^2. $V = 359$ ft^3 and $T = 273$ and **MR** becomes **2780** ft lb/mol/°C. Or **1545** ft lb/mol/°F if using °F, and **8314** joules/kg.mol/°K in SI units.* These values are known as the *universal* or *molar* gas constants, and the 'characteristic' gas constant (per lb or kg) is found by dividing them by the molecular weight of the gas. Thus for air, mw 29, 'R' is 96 in ft lb °C units, 53·4 in ft lb °F units or 286·7 in SI units – J or N.m.

Changes in a closed cylinder

If the gas is in a cylinder, the mass present stays constant so that in equation (1) **both** M and R are constants. Hence we can rewrite this as

$$\frac{P \times V}{T} = \text{'A Constant'} \quad \ldots\ldots\ldots\ldots \quad (2)$$

from which follows the familiar expression:

$$\frac{P_1 \times V_1}{T_1} = \frac{P_2 \times V_2}{T_2} \quad \ldots\ldots\ldots\ldots \quad (3)$$

The great advantage of equation (3) over the previous ones is that we are now dealing with ratios, so that provided temperatures and pressures are *absolute* we can use any units we like so long as they are consistent – i.e. all lb/sq. in., all N/sq. mm, etc.

Example

Air in a cylinder is compressed to one fifth of its original volume and the pressure is seen to rise from 14·7 lbf/sq. in. to 90 lbf/sq. in. If the initial temperature was 60°F (= 520°F abs) what is the final temperature? (0°F = 460°F abs.)

From equation (3),

$$T_2 = \frac{P_2 V_2}{P_1 V_1} \times T_1$$

$$T_2 = \frac{90}{14 \cdot 7} \times \frac{1}{5} \times 520$$

$$= 1 \cdot 224 \times 520$$

$$= \underline{636 \cdot 5\text{°F abs or } 176 \cdot 5\text{°F}}$$

The procedure is identical when working in SI units.

'Boyle's Law'

In the above example the temperature rose during compression because work was done on the gas, thus increasing its 'internal' energy. If, however, the cylinder were cooled sufficiently to keep the temperature constant, then the conditions would be 'isothermal' – which means simply that the temperature does not change. Hence $T_1 = T_2$, and

*The 'units' of 'molar **R**' in FPS (imperial) units are ft lb/mol°K or ft lb/mol°F, but if required in heat units R must be divided by J – 'mechanical equivalent of heat'. Whether working in °F or °C molar **R** then become **1·985**. When working in SI units the magnitude of molar **R** is identical whether in work or heat units because 1 Nm = 1 J.

equation (3) becomes:

$$P_1V_1 = P_2V_2$$

or $$P_1/P_2 = V_2/V_1 \quad \ldots\ldots \quad (4)$$

Thus Boyle's Law is but a *special case* of the general equation, when T is constant, and *only* applies under this condition.

Heating a gas

Like all other substances, gases have 'specific heats' – the amount of heat required to raise the temperature of unit quantity by one degree. However, gases are different in two important respects. First, the specific heat may be in heat units per unit *mass* or per unit *volume*. We can use whichever is convenient. Second, and more important, the specific heat depends on *how* the gas is heated. If it is in a closed vessel where the volume cannot change, all the heat applied goes to increase the velocity of the molecules; the 'internal energy' is increased, that is all. But if it is heated in such a way that the *pressure* remains constant (as when you heat your workshop, for example) the volume must increase as the temperature rises (see equation (3)) and this means that *work is being done* as well as the temperature rising. It therefore requires *more* heat to raise the temperature by one degree in this case.

Gases thus have *two* specific heats, one at constant volume, called Cv and one at constant pressure, called Cp, and this applies whether it is the specific heat per pound or per cu. ft. If you are a mathematician you could work out what this difference would be; it is, in fact, equal to the gas constant, R divided by J, the mechanical equivalent of heat (1400 in °C units, 778 in °F units or unity in SI). That is:

$$Cp - Cv = R/J \quad \ldots\ldots \quad (5)$$

In addition, the ratio between these two is also a constant, thus:

$$Cp/Cv = \gamma \text{ (Gamma)} \quad \ldots\ldots \quad (6)$$

The specific heat of all gases varies with temperature, but the method of dealing with this lies beyond the scope of this book. The following figures will serve for most of the calculations likely to be needed by model engineers. Note that the value of (gamma) shown is that normally applied; the values of C shown may not quite agree with equation (6) above.

Specific heats of gases

Gas	*Cp*		*Cv*		*Gamma (γ)*
	BTU/lb/°F	*kJ/kg/°C*	*BTU/lb/°F*	*kJ/kg/°C*	
Air	0·241	1·01	0·172	0·72	1·4
CO_2	0·228	0·953	0·183	0·765	1·3
CO	0·252	1·054	0·181	0·757	1·4
H_2	3·45	14·43	2·457	12·280	1·4
O_2	0·23	0·962	0·165	0·69	1·4
N_2	0·249	1·045	0·179	0·749	1·4

Figures for CHU/lb/°C are identical to those for BTU/lb/°F.

Example
A heat exchanger contains 100 cu. ft of air at 60°F and 15 lbf/sq. in. abs. How much heat is required to raise the temperature to 700°F?

First, use equation (1) to find the mass of gas, remembering to use lbf/sq. ft and absolute temperatures. From previous work, R for air is 1544/29 in lb°F units = $\underline{53 \cdot 4}$.

Hence: $M \times R = (15 \times 144 \times 100)/(60 + 460)$ ... (Note; T in °F abs)

and $M = (1500 \times 144)/(520 \times 53 \cdot 4)$

$= \underline{7 \cdot 78 \text{ lb}}$.

The question implies heating at *constant volume* (in a closed cylinder) so that Cv must be used, 0·172 from the table.

Hence heat required will be given by:

$$H = 0 \cdot 172 \times (700 - 60) \times 7 \cdot 78$$

$$= \underline{856 \cdot 4 \text{ BTU}}.$$

If this example had been stated in SI units the procedure would have been the same, but the pressure must be in **newton/m²**, not N/mm² or MN/m², the mass in kg, volumes in m³, and temperature in °K as follows.

P = 103 420 Pa (N/m²). V = 2·83 m³. Initial temp. 289°K, final 644°K.
R = 8314/29 = 286·7 J/kg°K. C_v = 720 J/kg°K (0·72 kJ).
As before –

$$M = \frac{P \times V}{R \times T}$$

$$= \frac{103\,420 \times 2 \cdot 83}{289 \times 286 \cdot 7}$$

$$= \underline{\underline{3 \cdot 53 \text{ kg}}}$$

$$H = M \times C_v (T_2 - T_1)$$

$$= 3 \cdot 53 \times 720 (644 - 289)$$

$$= \underline{\underline{902\,270 \text{ Joule } (902 \cdot 3 \text{ kJ})}}$$

Use the conversion factors on p. **1**.8 and p. **1**.11 to compare these results.

Compression and expansion of gases

The most common error made by model engineers in considering these processes is to assume that Boyle's Law is followed. This is very rarely the case, and *never* occurs in engines and compressors, even when water-cooled. In almost all real processes compression and expansion follow the law:

$$P \times V^n = \text{a constant} \quad \ldots \ldots (7)$$

Or, more conveniently,

$$P_1 \times V_1^n = P_2 \times V_2^n = \text{etc.} \quad \ldots \ldots (7a)$$

In an 'ideal' case the index 'n' would be gamma (γ) the ratio of the two specific heats, but this is seldom realised. (Such a process would be 'adiabatic', to give it its thermodynamic name.) In a well-cooled air compressor, running at slow speeds (up to 300 rpm) 'n' is about 1·2; at higher speeds it may rise to 1·28. For the compression period of a diesel or petrol engine 'n' may be taken at from 1·33 in low speed models (up to 1000 rpm) and 1·36 for higher. During the expansion stroke in a real engine the value of 'n' depends a good deal on the type of fuel and the engine cycle – whether a true diesel or a mixture type engine. Ricardo quotes a figure of 1·26 for a petrol engine, and I found most full-size high-speed diesels to operate at about n = 1·36. In turbo-compressors the index is very high, as much as 1·75, due to blade friction causing internal heating.

It follows that the calculation of compression pressures in *model* engines can only be approximate, because they are so small that few accurate indicator diagrams have been taken and there is no body of data from which values of 'n' can be estimated. Though the use of logs makes short work of calculating indexes, the value of 'n' can be chosen to make this easier still – viz. for an IC engine n = 1·33, which is 4/3; or, for a compressor n = 1·25 = 5/4 – without invalidating the approximation. However, the procedure is simple enough. From (7a) we can see that:

$$P_2/P_1 = (V_1/V_2)^n \quad \ldots\ldots\ldots\ldots \quad (8)$$

where V_1/V_2 is the 'compression ratio'.

Example

Estimate the compression pressure for a model engine having compression ratio 6:1. Assume the pressure in the cylinder at the end of suction to be 13·5 lb sq. in. abs.

From expression (8)

$$\begin{aligned} P_2 &= P_1(V_1/V_2)^n \\ &= 13{\cdot}5 \times 6^{1\cdot33} \qquad (1{\cdot}33 = \tfrac{4}{3}) \\ &= 13{\cdot}5 \times \sqrt[3]{6^4} = 13{\cdot}5 \times \sqrt[3]{1296} = 13{\cdot}5 \times 10{\cdot}9 \\ &= \underline{147 \text{ lbf sq. in.}} \end{aligned}$$

To find temperatures the pressures can first be calculated, and then use expression (3) – which holds irrespective of the process involved. But it is easier to amalgamate these formulae and the following derivations may be useful.

$$T_2/T_1 = (V_1/V_2)^{n-1} = (P_2/P_1)^{\frac{n-1}{n}} \quad \ldots\ldots \quad (9a)$$

$$V_1/V_2 = (P_2/P_1)^{\frac{1}{n}} \quad \ldots\ldots\ldots\ldots \quad (9b)$$

These, with expression (8) will cover all probable compression and expansion problems. Note that they only apply to operations in which the *temperature varies*. For an isothermal, or constant temperature, operation the 'Boyle's' expression is used, as explained earlier, i.e. $P_1 \times V_1 = P_2 \times V_2$.

As expressions (7), (7a), (8) and (9) are all *ratios* they hold in any system of units provided that they are consistent.

Work done

The work done when a gas is compressed or expanded is given by:

$$E = \frac{P_1 V_1 - P_2 V_2}{(n - 1)} \text{ ft lb} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

P in lb/sq. ft or N/m^3
V in cubic feet or cu. metre
E in ft lbf or N.m.

If this is positive, work is done *by* the gas; if negative, the process requires work to be done *on* the gas.

If the compression or expansion is isothermal, (at constant temperature) expression (10) will not serve as, 'n' being 1 in this case, the formula makes nonsense. In this special case *only*, work done is given by:

$$E = P_1 V_1 \times 2{\cdot}3 \log \frac{V_2}{V_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

This will seldom be needed in models as this class of operation is unusual in 'work' processes.

Example
If the engine in the previous exercise has a maximum cylinder volume of 0·1 cu. ft, find the work done during the compression stroke.

From this data, $V_1 = 0{\cdot}1$, $V_2 = \dfrac{0{\cdot}1}{6} = 0{\cdot}017$.

Substitute these figures and those found above in equation (10) not forgetting to convert pressures to lbf sq. ft – i.e. multiply by 144.

Thus $$E = \frac{(144 \times 13{\cdot}5 \times 0{\cdot}1) - (144 \times 147 \times 0{\cdot}017)}{(1{\cdot}33 - 1)}$$

$$= \frac{194{\cdot}4 - 359{\cdot}8}{0{\cdot}33}$$

$$= \underline{-501{\cdot}2 \text{ ft lbf/stroke}}.$$

the negative sign indicating that work is done *on* the gas.

If working the above example in SI units the same procedure is followed, with volumes in cu. metre and pressure in newton/sq. metre. The solution comes out in newton.metres *or* joules. If working in bar or N/mm^2 the appropriate multipliers must be used – 10^5 and 10^6.

Conclusion

As mentioned at the beginning, this summary has of necessity been a 'skim over the surface'; whole books have been written on the subject, one at least running to two volumes. However, it is hoped that, brief though it may be, the section will help modelmakers to understand a little of what goes on inside the cylinders of their engines.

SECTION TEN
BOILER WORK

BOILER STEAMING RATES

The prediction of steaming rates from small boilers is very difficult, as full-size rules do not apply. More depends on the firing method than on the physical dimensions of the boiler grate area or heating surface. However, for small units such as those used for stationary engines and small power boats, actual tests suggest the following 'rules of thumb'. The steaming rate is given as water evaporated in cubic inches/100 sq. in. of exposed heating surface. The volume of steam made at any pressure can be found from the Steam Tables on pages **8**.2 and **8**.3.

Simple *pot* boiler, $2\frac{1}{2}$–$3\frac{1}{2}$ in. dia. Spirit fired		1·0
3 in. dia. centre flue, spirit fired		1·25
ditto	ceramic gas burner	2·0
3 in. dia. cross-tube type, gas burner		2·5–3·0
2 in. × 6 in. horizontal with circulating tubes, gas fired		3·0–3·5

Those with gas firing could be driven harder, but with excessive heat loss up the chimney. 'Flash steam' boilers with vapourising paraffin burners have been known to reach, or even exceed, 5 cu. in./100 sq. in. HS.

The case of *locomotive* boilers is very difficult, as so much depends on the draughting arrangements. When stationary and steaming under the blower many boilers can make far more steam than the engine could use in traffic. However, it is very easy to check the steaming rate under these conditions, and readers are referred to an article by Prof. W. B. Hall, *Model Engineer*, 6 January 1995, p. 44. Research is at present (January 1996) in hand with the object of relating steaming rate with one or more of the boiler design parameters.

BOILER MAKING

These notes refer to brazed copper boilers, though some data may be applicable to welded or rivetted shells. For information on the latter refer to the books given later in the bibliography at the end of this section.

Boiler shells

The shell or barrel of a boiler is a 'thin' cylinder, and the safe shell thickness is given by the formula:

$$t = \frac{p \times d}{2f} \times \frac{1}{E} \quad \ldots\ldots \quad (1)$$

t = thickness of shell, in. (or mm)
p = internal pressure, lbf/sq. in. (or N/sq. mm)
d = internal diameter, in. (or mm)
f = working stress, lbf/sq. in. (or N/sq. mm)
E = joint efficiency.

If the shell is a drawn tube, then E = 1. If rolled and brazed then an allowance must be made both for the slightly distorted shape at the joint and for the discontinuity of material. For the dovetailed 'brazier's joint', E may be taken as 0·8 if skilfully made; for brazed butt-joints, or those with internal or external butt-straps E is about 0·7. A joggled lap-joint is about the same. Though a sifbronzed joint may be stronger, the change in stiffness at the joint has an effect, and it is recommended that model engineers use 0·7 for all joints.

The *working stress* is often taken as the Ultimate Tensile Strength (UTS) divided by a factor of safety. However, it is possible to assess the factors which reduce the material strength to arrive at a working stress. The first constraint is that the initial deflection should not exceed 0·1% – i.e. a 5″ boiler should not expand by more than ·005″ under load. This suggests a working stress not exceeding 30% of the UTS for copper. However, when under steam the metal is at a fairly high temperature, and this reduces the strength. The most reliable working stress will be that which does not exceed 15% of the UTS (allowing for the joint efficiency) at the *test pressure* (which is applied cold) and does not exceed the figures in the table below at the *operating temperature*. The American Society of Tool and Manufacturing Engineers quotes these figures for annealed tubes.

Temperature, °F	300	350	400
Equiv. steam pressure, lbf/sq. in. (bar)	50 (3·5)	120 (8·3)	215 (14·8)
Working stress, lbf/sq. in. (N/sq. mm)	5000 (34·5)	3800 (26·2)	2500 (17·25)

Whilst these stresses may appear to be low, it must be appreciated that after brazing the copper will be fully annealed, and though some work-hardening may occur under the test loads, this cannot be accurately assessed.

The overall safety factor, referred to the UTS cold, is between 8 and 11 using these figures for 'f' in the formula.

Effect of holes in the barrel

A tube carrying internal pressure is stressed in tension, and the presence of any hole will result in a local increase in stress well above that due to the removal of stressed area. The smaller the hole, the greater this increase will be. The effect of such 'stress-raisers' can be calculated, but it is better to avoid them. (Fortunately, in copper and mild steel some stress relief occurs at the first loading, but this is less the case with stainless steel, where more care must be taken.) The most effective method is to enlarge the hole and at the same time stiffen the surrounding material, which is easily done by fitting a bush, brazed in. This has the added advantage of giving a longer thread for the fitting. *All* holes in boiler barrels should be bushed using bronze or gunmetal *not* brass.

With larger holes, as in way of the dome, the situation is different. The stress-raising factor is small, but the removal of load-carrying metal means that the steam load no longer applies a tensile force, but tries to open out the shell in bending. The brazing in of a piece of tube to form the dome doesn't help – in fact it makes matters worse. It is *imperative* that all such openings be strengthened by fitting a compensating ring round the hole. The total cross-sectional area of such rings should be from 25% to 35% greater than the load-bearing area removed in the hole, e.g. if a 1″ dia. hole is made in a shell 0·08″ thick, the area removed is 0·08 sq. in. The ring should have a diametral cross section which provides 0·1 to 0·11 sq. in., say ·375″ wide × 0·14″ thick at the thinnest part. This can be provided either as a bush for the dome to sit on, or as a ring brazed round a brazed-in dome-tube. Note that there is no reason why the ring should not be internal, or even with a ring both inside and outside, each half the thickness. The material should be the same as that used for the shell.

Flat plates

These are found at the firebox of a loco boiler, and at the ends of shell-type boilers. Calculation of the stress is very difficult indeed, but fortunately a good body of empirical knowledge has been built up during the 100-odd years of model boiler-making. Martin Evans recommends that wrapper-plates should be slightly thicker than the shell. This should also apply to the inner wrapper as well; the suggestion that this may reduce heat flow is unfounded – the resistance of the copper is less than $\frac{1}{2}$% of that due to the gas and water films on either side of the plate. Tube-plates and firebox backplates of loco boilers should be from 20% to 25% thicker than the boiler barrel. For the endplates of small 'pot' boilers it is suggested that these should be not less than one gauge of metal thicker, with a stay or stays from end to end. If there is a centre flue, this should provide sufficient staying at the low pressure usual in such plant.

Stays

All flat plates should be stayed, as should any tubes of large diameter carrying *external* pressure, unless they are very thick (see later). The size of the stay needed is often calculated by assuming that each stay bears the steam load on the unsupported area between the centre lines of adjacent rows. This is not altogether satisfactory, as it assumes that all stays are uniformly loaded. A better rule is to write that each stay carries a load given by:

$$W = 1{\cdot}2c^2p \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

W = stay load, lbf, or newton
c = pitch of rows of stays, in. or mm
p = boiler pressure, lbf sq. in. or N/mm^2

This assumes that adjacent stays may carry loads varying by ± 10%, a more likely event. Having determined this load, the *core* area at the root of the threads should be used to determine the stay diameter, using working stresses of 2500–3000 lbf sq. in. (17–21 N/mm^2) for copper, 3500–4500 (24–30 N/mm^2) for drawn GM, and 6500–7500

($45-50$ N/mm^2) for monel stays, the lower stresses for the smaller diameter stays.

There is something to be said for working the other way round – deciding the stay diameter in terms of the plate thickness, to get a minimum of 3 threads engaged in metal of 16 gauge ($\frac{1}{16}$") rising to say 6 threads in $\frac{3}{16}$" metal. Martin Evans uses this principle in his recommendation given in the table below.

Plate thickness	1·6 $\frac{1}{16}$	2·0 0·08	2·4 $\frac{3}{32}$	3·2 $\frac{1}{8}$	4·0 $\frac{5}{32}$	4·8 $\frac{3}{16}$	6 mm $\frac{1}{4}$ in.
Stay dia. and tpi	5 BA	4 BA	$\frac{3}{16}$" 40 tpi	$\frac{1}{4}$" 32 tpi	$\frac{5}{16}$" 32 tpi	$\frac{3}{8}$" 32 tpi	$\frac{3}{8}$" 26 tpi
Metric	M3	M3·5	M5	M6 × 0·75	M8 × 0·75	M10 × 1	M10 × 1

He then offers the following formula to find the pitch, modified here to use the *safe* stresses above rather than the UTS.

$$c = \sqrt{\tfrac{3}{4}\,\frac{d^2 f}{p}} \qquad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

(Symbols as in equation (2) – f = working stress)

This may be compared with the results obtained from the previous formula.

K. N. Harris in his book on boiler making gives a table of stay centres evidently based on the rule that stay centres = 6 × plate thickness.

The BA series threads have a better profile for use in copper than others but the pitch is a little coarse for 0 and 1 BA. ISO (metric) threads to BS3643 'normal' pitch will provide about 3 threads. The 'constant pitch' series of 0·75 mm and 1 mm are better for stays 6 mm and above.

Dished or domed ends

Some boilers have endplates formed over a section of a sphere, with the pressure on the concave side. The thickness may be found from:

$$t = \frac{0{\cdot}885pR}{0{\cdot}7f - 0{\cdot}1p} \text{ in. or mm} \qquad \ldots\ldots\ldots\ldots \quad (4)$$

where p = pressure, lbf/sq. in. or N/mm^2
t = thickness of plate, in. or mm
R = dome radius, in. or mm
f = working stress, lbf/sq. in. or N/mm^2

The *radius* of the domed part should not exceed the *diameter* of the boiler barrel, and the radius at the junction of dome and flange should be R/16 or more.

Flue tubes

These are subject to external pressure, and unstable. For a given thickness the collapse pressure drops steeply as the diameter increases, as shown in the following table of experimental results for 18 SWG (1·2 mm) annealed copper tube.

Tube dia., in.	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Fail pressure lbf/sq. in.	1600	1150	900	650	460	330

These results (some interpolated) were for 'long' tubes, where the support of the end-fixing was negligible, the usual condition in a fire-tube boiler. Collapse is, of course, a non-catastrophic failure as a rule, the tube simply closing in on itself without rupture unless the material is brittle, or the joint tears away at the endfixing.

For model boilers, Martin Evans presents the following recommendations for a working pressure of 100 lbf/sq. in. (7 bar).

OD in.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}-\frac{5}{8}$	$\frac{3}{4}-1\frac{1}{8}$	$1\frac{1}{4}-1\frac{1}{2}$
SWG thick†	24	22	20	18 (16)	16 (14)

†See p. 7.18 for metric equivalents to SWG.

In view of the collapse figures given previously it is suggested that the larger sizes are a trifle on the thin side, and the figures in brackets might be used instead. The additional thickness will have no effect on heat transfer – indeed, no harm would follow from using a thicker gauge throughout if this were more readily to hand. One problem with flue-tubes is abrasive wear when brushing out.

Large flues, as for Cornish or Lancashire boilers, are more of a problem, as there is little data on such tubes at steam temperatures. Extrapolating from other data, it would appear that a 2″ flue ought to be of the order of 12 SWG for 100 lbf/sq. in., but stiffening rings should be brazed on outside the flue, spaced about $1\frac{1}{4}$ to $1\frac{3}{4}$ tube diameters apart. These are in the form of 'washers' about $\frac{1}{8}$″ thick and $\frac{3}{16}$″ to $\frac{1}{4}$″ wide (radially). Care should be taken not to distort the flue in fitting them, and that the brazing is full strength with good penetration. Such flues should on no account be constructed by rolling and brazing; *any* deviation from the circular form will risk initiating collapse, and in this connection, care should be taken not to overtighten 3-jaw chucks when trimming tube ends. The three flats which may be formed will seriously weaken the tube.

Firetube diameter

The most economical length/diameter ratio, for larger model and full-size boilers with induced draught, appears to be between 70 and 90; any increase in length above this may result in more steam being needed to produce the draught than is provided by the longer tube. Tests on a full-sized locomotive showed that the last 25% of tube length evaporated only 10% of the total steam provided by the tube as a whole. In the case of model boilers, where high steam rate is more important than thermal efficiency the ratio may be much smaller. Martin Evans uses the formula

$$d = \sqrt{\frac{L}{65}} \qquad \begin{array}{l} d = \text{tube internal dia. in. or mm} \\ L = \text{tube length, in. or mm} \;\ldots\ldots\ldots\; (5) \end{array}$$

which gives ratios around 25–30, for loco boilers.

Vertical boilers, with natural, or mild draught from a blower, may have L/d ratios as small as 15, largely due to mechanical difficulties in making the boiler with very small diameter tubes if a greater ratio were used.

Superheater flues are, of course, proportioned more to carry the superheater elements than with any practical or theoretical heat transfer considerations in mind.

SAFETY VALVES

Recent research has resulted in a considerable reduction in the trial and error needed in designing a safety valve once the steaming rate is known. It has been established that the mass flow through the valve is given by:

$$M = 51{\cdot}4 \times A_f \times C_d \times (p + 14{\cdot}7) \text{ lb/hour} \quad \ldots\ldots\ldots \quad (6)$$

where p = lifting pressure under the valve, lbf/in.^{2}
A_f = area of flow between valve and seat, sq. in.
(Allowing for the seat angle, if any.)
C_d = is a coefficient of discharge, 0·9 in this case.

The pressure p will, of course, be 'normal boiler pressure + 10%, as required for the safety valve test.

Having established the area required the designer may then use a small diameter valve with high lift, or vice versa. Experiment has supported theory in suggesting that **low lift** valves have some advantages. It is recommended that the ratio 'bore dia./vertical lift' should be as high as 30 where possible, but on no account be lower than 10. Where space, or the 'scale outline', permits a ratio even higher than 30 can be advantageous.

After deciding the D/L ratio the valve diameter can be calculated to pass the estimated steam flow. From this the spring **force** needed to hold the valve on its seat at the working pressure, and the spring **rate** required to permit the required *vertical* lift to provide the area A_f can be worked out. (See p. **13**.1, or *Spring Design & Manufacture* Nexus Workshop Practice Series No. 19.) With mitre or ball valves the effect of the effective seat angle must be allowed for, and in the case of 'Salter' type valves, any lever ratio between valve and spring.

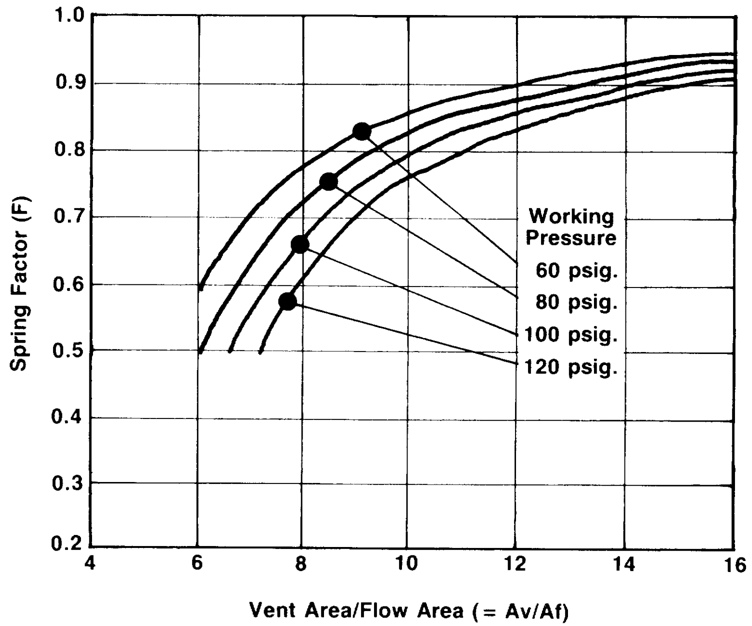
If the valve discharges direct to atmosphere there are no further complications, but most locomotive valves are enclosed. This leads to two considerations. First, the valve head, or ball, can cause an obstruction within the case. The area of flow around the valve head should be large, and in any case, never less than $1\frac{1}{2}$ times the area of the vents to atmosphere.

Second, these vents themselves do cause an obstruction to the flow, and set up a **back pressure** within the valve casing. This back pressure *acts as an additional spring, preventing the valve from achieving the design lift at the design over-pressure*. To allow for this the **spring rate** must be reduced.

Our work showed that this back pressure was a mathematical function of the ratio between the vent area (A_v) and the flow area through the lifted valve (A_f). The calculation is somewhat laborious, but the resulting corrections have been reduced to a simple chart, shown opposite. Suppose the ideal spring rate to be 50 lbf/in., the working pressure 100 lbf/sq. in., and A_v/A_f is 10, then from the chart the correction, (F) is 0·8. The required spring rate is thus $0{\cdot}8 \times 50 = 40$ lbf/in.

It will be observed that the effect becomes very critical as the ratio A_v/A_f falls below 10, and this figure is recommended as the minimum design requirement. If the ratio can be held at 15 or over, then a uniform correction of 0·9 can be applied without exact reference to the chart for most designs. It is clear that the greater the ratio D/L referred to earlier the easier it will be to accommodate a larger vent area.

A final practical comment may be worthwhile. During the research it was found that the use of a ball valve did lead to problems in design. Although a mitre seated valve might be a little more difficult to make it was then far easier to accommodate the required clearances and vent areas, as well as leaving more space for the spring. The one proviso that must be made, however, is that the guide wings on such a mitre valve must be



properly formed, so that the wings do not occupy more than 25%–30% of the area of the bore. The mere 'filing of three flats' once commonly used is quite inadequate!

BOILER TESTING

Rules for boiler testing have undergone considerable revision over the last few years. They have been codified by the Northern Association and Southern Federation of Model Engineers to satisfy their respective Insurance Policies, with special reference to locomotives, road or rail. However, the rules laid down should be taken to apply to **all** boilers, large or small; even a simple 'pot' boiler can contain a considerable amount of lethal energy.

The codes occupy a dozen or so pages and copies can be obtained from the Secretary of the local club or the relevant Association. The following is a summary of the salient points of the rules of the Southern Federation for *copper* boilers. Some note on those made of steel follow later.

(1) The constructor of any boiler not built to a published design must produce design drawings and calculations for approval. Boiler testers must satisfy themselves that both the specifications and thicknesses are correct.
(2) All boilers must carry an identification mark, to be recorded on the test certificate(s). On change of ownership the new owner is responsible for submitting the boiler for re-examination and, if the tester so decides, retest.
(3) The boiler must be subjected to a **visual inspection**, both external and, so far as possible, internal, when stripped of all insulation. Special attention is directed to silver-soldered joints. 'Used' boilers must be examined for corrosion and wasting.
(4) An **hydraulic test** must be carried out at *twice* the working pressure. The pressure-gauge used must have been calibrated and certified against an accurate master gauge.

The test must be applied slowly, with examinations for leakage and distortion at intervals. After reducing the pressure slowly to zero, the test shall be repeated, with the test pressure held for 20 minutes whilst a further examination is made.

(5) A test **under steam** must follow, when a further examination must be made, together with a water-gauge and safety valve test.

(6) The **safety valve test** shall ensure that the valve(s) are correctly set to operate at the working pressure and are able to maintain the pressure at less than **10%** above working pressure under all circumstances. The gauge fitted to the boiler must be marked with a red line to show the working pressure *as indicated by the calibrated test gauge*, not that shown on the model gauge.

(7) **Retests** are to be carried out at intervals (not stated, but presumably every two years) when the hydraulic test is reduced to $1\frac{1}{2}$ the declared working pressure. The superficial examination may be carried out, at the Inspector's discretion, without removing the cladding. The steam tests remain the same.

(8) Other clauses cover the issue of Test Certificates (without which the loco may not be accepted on the club track), the acceptance of commercially made boilers, and basic procedures when steaming the locomotive on the track or road.

Steel boilers are covered in a separate section. The main differences cover (a) Test requirements on materials. (b) The welder must be 'suitably qualified'. (c) Special conditions for stainless steel boilers. Other procedure is similar to that for copper boilers. There are special arrangements for boilers tested under the rules of 'reputable' insurance companies and which conform to BS requirements as to design.

Author's comments

Whilst the code may well satisfy the Federation's insurance companies, and hence be adequate for 'club running' it is important to appreciate that owners of both road and rail locomotives owe a 'duty of care' to the public, whether when running for their own pleasure or when carrying passengers. Any mishap can lead to an official enquiry or, perhaps worse, to litigation.

In my view clause (5) is loosely worded (what sort of test?) and clause (6) above (as laid down in detail in the code) is quite inadequate. It may be that the Boiler Inspector's *instructions* give more detail, but it is most important that both constructors and users should know what is expected of the test.

It is suggested that the following sub clauses should be considered for future issues of the code:

(a) After initial raising of steam the blower valve should be kept wide open until the test is completed.

(b) The Inspector must satisfy himself that the fire is at the normal depth before the safety valve test commences.

(c) The water level shall be maintained at or just above the normal level (by use of pump or injector) until the safety valve(s) start to blow off. Water feed must then be shut off completely.

(d) The accumulation of pressure shall be observed over a period, *either* for five minutes *or* until the water level has dropped to the lowest safe level, whichever is the shorter. During this period the accumulation of pressure must not exceed 10% of the declared working pressure as measured on the Inspector's gauge.

(e) If the pressure is *still rising* at the end of this period the Inspector must refuse to certify the boiler. **However**, if the water level is such that it is safe to do so, the Inspector may, at his discretion, continue the test for a total period of *ten* minutes, and if the 10% condition is met, certify the boiler with a note to the effect that it was an 'extended test'.

So far as the code for **steel** boilers is concerned similar considerations apply. Further, the use of the term 'suitably qualified welder' is inadequate. The correct designation is 'certified welder', and there should be a requirement that his registration number is stamped on the fabrication.

It may be found that larger scale models with steel boilers must be insured with one of the specialist insurance companies, in which case the inspection requirements will be more rigorous.

Copper tubes and pipes

Metric sizes of copper tube				
Nominal OD	*Preferred thicknesses BS 2871*			
3	0·5	0·6	0·8	
4	0·5	0·6	0·8	
6	0·5	0·6	0·8	1·0
8	0·6	0·8	1·0	
10	0·6	0·8	1·0	
12	0·8	1·0	1·2	
15	1·0			
16	0·8	1·2	(1·5)	
18	1·0			
20	1·0	1·2	1·5	
25	1·0	1·2	1·5	(2)
30	1·2	1·5	2·0	
38	1·2	1·5	2·0	
44·5	1·5	2·0	2·5	
50	1·5	2·0	2·5	
76·1	2·0	(2·5)	(3·8)	
89	2·4	3·0		
108	2·5	(3·0)	(4·9)	
133	2·4	3·0	(6·5)	
159	2·5	3·0	3·5	

Common stock sizes of boiler tubes (not all from same supplier)		
Nominal OD in.	*Thickness SWG/mm*	
1	18/1·2	16/1·6
$1\frac{1}{4}$	16/1·6	
$1\frac{1}{2}$	16/1·6	
$1-\frac{5}{8}$	14/2·0	
2	16/1·6	
$2\frac{1}{2}$	16/1·6	
3	16/1·6	
$3\frac{1}{4}$	16/1·6	
$3\frac{1}{2}$	16/1·6	
$3\frac{3}{4}$	13/2·5	
4	13/2·5	16/1·6
$4-\frac{3}{8}$	13/2·5	
$4\frac{1}{2}$	13/2·5	
$4\frac{3}{4}$	10/3·0	13/2·5
5	10/3·0	13/2·5
6	10/3·0	
$6\frac{1}{4}$	10/3·0	

Figures in brackets are to British Standard but may not be generally available. Thick-wall copper tube to BS 1306 and BS 61 is still available in inch sizes.

Copper sheet

Although some copper sheet is still available in SWG, metric thickness is now an almost universal rule. In the table below those marked ** are 'second preference' and may not be easy to come by. See also p. **7**.18.

Comparative thickness

Inch	*SWG*	*mm*	*SWG*	*mm*
$\frac{3}{16}$	6	5·0 (4·5**)	19	1·0
$\frac{5}{32}$	8	4·0	20	0·9**
$\frac{1}{8}$	10	3·15	21	0·8
$\frac{3}{32}$	13	2·5	22	0·71**
$\frac{1}{16}$	16	1·6	24	0·56**
$\frac{3}{64}$	18	1·25		

Note that when making flanged plates the **exact** dimensions of the sheet and mating tube should be *measured* as all are subject to tolerances.

Boiler manholes

The normal size of an inspection manhole in a boiler shell is 12" × 16" (305 × 405 mm). The short axis should lie along the axis of the boiler.

Threads on Stuart-Turner boiler fittings

Dia. of thread, in.	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
tpi	40	40	40	32	26	26	26	26

Bibliography

Model Locomotive Boilers	Martin Evans, MAP Ltd.
Model Boilers & Boilermaking	K. N. Harris, MAP Ltd.
Strength of Copper Tubes	Copper Development Association
BS 3274, Tubular Heat Exchangers	British Standards Institution

SECTION ELEVEN
PISTON AND GLAND SEALS
O RINGS & PISTON RINGS

'O' RINGS

Standard 'O' rings

The British Standard for inch sizes (BS 1806/1989) corresponds very closely to that of the Society of Automotive Engineers (USA). In this standard the 'nominal' ring diameter is the actual diameter of the cylinder or rod to which the ring is applied; the actual diameters of the rings are a few thou larger or smaller than the nominal OD or ID respectively. Three cord diameters are in common use by model engineers, and the dimensions of rings likely to be needed are given in the table below. Those marked * *may* give satisfactory service in sliding seals at the pressure used in models, but are not suitable for high-pressures; the next larger size of cord should be used in such cases. Cords larger than those in Table 1 (up to 0·275″ cord × 26″ OD) are available. 'C' and 'D' refer to the diagram shown later.

Table 1 below gives details of 'inch' dimensioned rings, and Table 2 those for metric. However, rings can accommodate *small* deviations for the nominal shaft or cylinder size – see later note – so that in some cases 'inch' rings can be used in metric housings.

Table 1. BS 1806

Cord 0·070″ dia. ± 0·003″ ($\frac{1}{16}$″ nominal). Larger sizes available from some suppliers

BS No.	004	005	006	007	008	009	010	011	012	013*	014*
Rod dia. (C)	$\frac{5}{64}$	$\frac{7}{64}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
Cyl. dia. (D)	·203	·234	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$

Cord 0·103″ dia. ± 0·003″ ($\frac{3}{32}$″ nominal). Larger sizes available from some suppliers

BS No.	110	111	112	113	114	115	116	117*	118*	119*	120*
Rod dia. (C)	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
Cyl. dia. (D)	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	1·06	1·125	1·1875

Cord 0·139″ dia. ± 0·004″ ($\frac{1}{8}$″ nominal). Larger sizes available from some suppliers

BS No.	210	211	212	213	214	215	216	217	218	219	220	221	222	223*	224*
Rod dia. (C)	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
Cyl. dia. (D)	1	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{5}{8}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2

Cord 0·210″ dia. ± 0·005″ ($\frac{3}{16}$″ nominal). Larger sizes available from some suppliers

BS No.	325	326	327	328	329	330	331	332	333	334
Rod dia. (C)	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$
Cyl. dia. (D)	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{7}{8}$	3

Table 2. Metric Dimension Rings to BS 4518

1·6 ± 0·08 mm cord dia., suffix 16. Sizes up to 40 mm OD See note*

BS No.	0031	0041	0051	0061	0071	0081	0091
Rod dia. (C)	3·5	4·5	5·5	6·5	7·5	8·5	9·5
Cyl. dia. (D)	6·0	7·0	8·0	9·0	10·0	11·0	12·0

2·4 ± 0·08 mm cord dia., suffix 24. Sizes up to 74 mm OD

BS No.	0036	0046	0056	0066	0076	0086	0096	0106	0116	0126	0136
Rod dia. (C)	4·0	5·0	6·0	7·0	8·0	9·0	10·0	11·0	12·0	13·0	14·0
Cyl. dia. (D)	8·0	9·0	10·0	11·0	12·0	13·0	14·0	15·0	16·0	17·0	18·0

3·0 ± 0·10 mm cord dia., suffix 30. Sizes up to 250 mm OD

BS No.	0195	0215	0225	0245	0255	0265	0275	0295	0315	0345	0365	0395
Rod dia. (C)	20	22	23	25	26	27	28	30	32	35	37	40
Cyl. dia. (D)	25	27	28	30	31	32	33	35	37	40	42	45

5·7 ± 0·12 mm cord dia., suffix 57. Sizes up to 500 mm OD

BS No.	0443	0453	0493	0523	0543	0553	0593	0623	0643
Rod dia. (C)	45	46	50	53	55	56	60	63	65
Cyl. dia. (D)	55	56	60	63	65	66	70	73	75

**Note*: Metric rings of this section are not recommended for use on pistons but may give good service on glands.

All 'O'-rings will accommodate a slight mismatch between the nominal ID or OD and the dimensions of the cylinder or rod which is being sealed. Messrs James Walker Ltd offer the following guidance.

To fit an undersize shaft the groove in the housing may be made not more than 3% undersize on the diameter. To fit an oversize cylinder the groove in the piston or ram may be made up to 5% oversize.

When used as a flange sealer the ring may be stretched by no more than 2% to fit a mismatch, but should not be reduced radially by more than 1%.

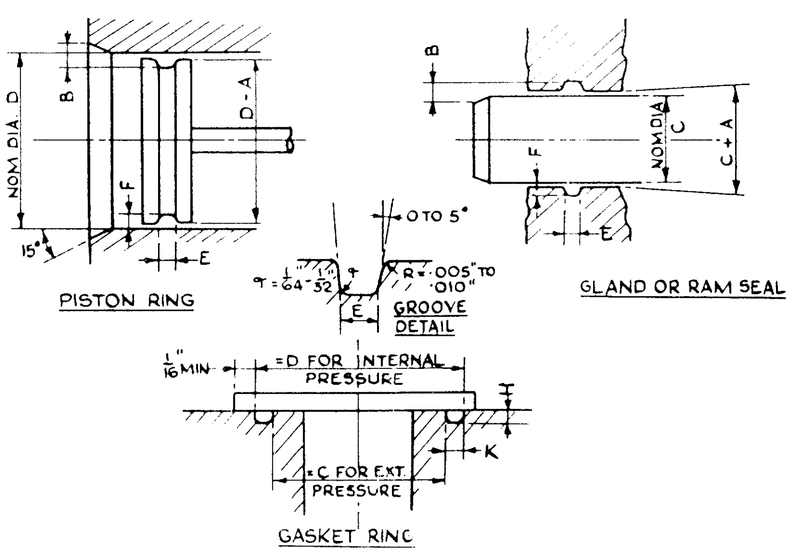
Groove sizes

The groove depth is smaller than the cord dia. to give a degree of 'pinch', and its width is larger, both to allow the cord to expand sideways and to move a trifle in the groove, the sealed pressure holding it against the downpressure side of the groove. In experimenting, the groove width should never be made less than an amount equal to cord dia. *plus* the allowed 'pinch'. Best practice is to make the cross-sectional area of the groove about 16%–18% greater than that of the cord.

Two dimensions for each are given in Table 3, p. **11**.4. That marked 'SAE' is the Society of Automotive Engineers' recommendation for sealing up to 1500 lbf/sq. in.; that marked 'ME' follows the recommendations of Mr Arnold Throp for steam pressures in models. The latter require close attention to limits in machining both bore and groove, if the pinch is not to be lost in tolerances. The only reason for using small degrees of pinch is to reduce friction, and any figure between the two will 'work'. Piston and rod clearances should be normal, the 'maximum clearance' shown being that which, if exceeded, is found to cause extrusion of the ring under high pressures. The table also shows dimensions for grooves when rings are used as gaskets sealing static joints.

Mr Throp's work was done entirely on the 'inch' section rings to BS1806, but metric rings to BS4518 are now in increasing use. Table 4 shows the *maker's* recommendations for the dimensions of the housing. These correspond to the 'SAE' figures for 'inch' size rings, and some relaxation of 'pinch' might be permissible. It is recommended, however, that experiments be carried out before adopting deeper grooves (dimension F) to reduce the pinch. *Note* that most makers recommend that 1·6 mm cord rings (corresponding to $\frac{1}{16}''$) are not suitable for sliding applications. However, as the $\frac{1}{16}''$ cord has been successful on small (below $\frac{3}{8}''$ dia.) sizes there seems to be no reason why 1·6 mm cord should not be used up to 10 mm dia.

The sketch shows the recommended groove shape, and also indicates the desirable chamfer to facilitate the entry of the ring without damage. Such a chamfer should always be included where the design permits.



Selection

For a ring sliding in a bore (e.g. a piston ring) the *nominal* OD of the ring is to correspond to the *actual* bore of the cylinder. For a ram or piston rod, the nominal ID of the ring should correspond to the actual diameter of the ram or rod. For gaskets sealing pressure engaged on the ID of the ring, the OD of the groove should correspond to the nominal OD of the ring; for externally applied pressure the reverse holds. For sliding seals where a non-standard diameter is needed, select the nearest *smaller* ring and slightly increase the pinch – i.e. reduce the groove depth.

Whilst 'O'-rings will seal oscillating or rotary glands they are liable to more rapid wear in this application. Little or no advantage is found in oscillating glands requiring very low friction (e.g. Governor valve spindles) as although the moving friction is low, the 'stiction' at breaking loose on initiating movement is obtrusive.

Table 3. Inch Dimensions

Ring size (cord dia.)	*Sliding seal groove*				*Gasket groove*		*Clearance (max)*	*Bevel*
	Width, E		*Depth, F*					
Nom	*SAE*	*ME*	*SAE*	*ME*	*H*	*K*	*A*	*B*
$\frac{1}{16}''$	0·094″	0·090″	0·061″	0·065″	0·056″	0·095″	0·005″	0·085″
$\frac{3}{32}''$	0·141″	0·125″	0·093″	0·098″	0·086″	0·140″	0·005″	0·120″
$\frac{1}{8}''$	0·188″	0·160″	0·124″	0·132″	0·115″	0·190″	0·006″	0·160″
$\frac{3}{16}''$	0·281″	—	0·186″	—	0·175″	0·280″	0·007″	0·230″

Table 4. Metric Dimensions

Cord dia. nominal	*Sliding seal groove*		*Gasket groove*		*Max clearance diametral A*	*Bevel B*
	Width, E	*Depth, F*	*H*	*K min*		
1·6 mm	2·3–2·5	1·18–1·25	1·2–1·3	2·4	0·12	0·60
2·4 mm	3·2–3·4	1·97–2·09	1·7–1·8	3·7	0·14	0·65
3·0 mm	4·0–4·2	2·5–2·65	2·2–2·3	4·5	0·15	0·70
5·7 mm	7·5–7·7	4·95–5·18	4·4–4·5	8·1	0·18	0·93

Installation

Surface finish must be as high as is attainable, not forgetting the inside of the groove; the ring will slide back and forth in operation and wear on the inside will reduce the 'pinch' in time. Ground rod as used for piston rods *must* be polished before fitting, as the grinding marks will cause rapid wear. Tolerances – i.e. the inevitable differences between the correct size and the size as machined – should be plus or minus about 0·005″ on the axial width of the groove; tolerance on the groove depth should be such that the pinch is slightly high if ME figures are used, and slightly low if using those of the SAE. This ensures proper sealing, if perhaps more friction. Even with SAE grooves the friction will be less than with ordinary packing. With large cylinders, ($1\frac{3}{4}''$ bore or over) $\frac{1}{8}''$ 'O'-rings may tend to roll over in service, and though $\frac{3}{16}''$ rings are available it is probably better to use normal piston-rings. In all cases the diametral clearance "A" must be controlled to the practicable minimum to avoid risk of 'extrusion' of the ring.

PISTON RINGS

For a detailed examination of the design and manufacture of piston rings see the series on this subject in the Model Engineer, *starting on p. 85, issue 3961, 21 Jan. 1994.*

Measurement of wall pressure

The mean pressure exerted by the ring on the cylinder wall may be measured in two ways.

(i) By wrapping a wire around the ring and exerting a tension as shown in Fig. 1a until the free gap is closed. Then the mean pressure is given by:

$$p = 2F_t/(w \times D) \quad \ldots\ldots \quad (1)$$

where

p is in lbf/in.^{2} or N/mm^2
F_t in lbf or newton
D is cyl. dia., in. or mm
w is axial width of ring, in. or mm

(ii) By compressing the ring using a diametral force as in Fig. 1b, until the diameter at right angles to the gap is exactly that of the cylinder. Then mean pressure is given by:

$$p = 0{\cdot}881F_{dd}/(w \times D). \text{ Symbols as before} \quad \ldots\ldots \quad (2)$$

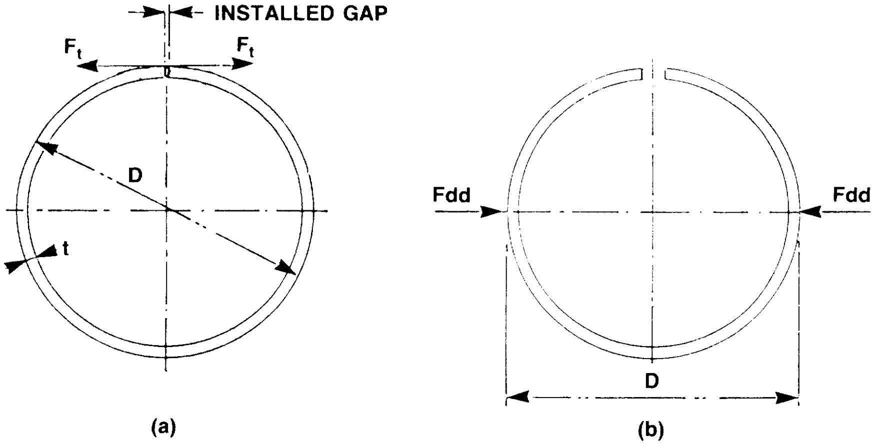


Fig. 1

Calculation of wall pressure

The mean wall pressure generated by a ring of uniform section can be estimated from:

$$p = \frac{g}{D} \times \frac{E}{7{\cdot}06} \times \frac{1}{(D/t - 1)^3} \quad \ldots\ldots \quad (3)$$

where:

p = wall pressure, lbf/in.^{2}
g = free gap − installed gap, in.
D = nom. dia. of ring (cyl. bore) in.
E = Young's modulus (transverse) lbf/in.^{2}
t = radial thickness of ring, in.

Material

The usual material for model rings is cast iron, to BS 1452, grade 17 (or grade 260 in the metric specification). This has a nominal ultimate tensile strength of 17 T/in.^{2}. However, tensile strength is irrelevant. The ring is stressed in *bending*, and cast iron does not conform to classical bending theory so far as the ultimate stress is concerned. The more appropriate test is the *transverse test*, where the material is loaded in bending. The ultimate transverse stress for grade 17 material is about **29 ton/in.^{2}** (say 455 N/mm^2). The value of 'E' (Young's modulus) under these conditions is about $14{\cdot}2 \times 10^6$ lbf/in.^{2} (say 100 000 N/mm^2).

Both phosphor bronze and brass have been used for piston rings in rare cases. The relevant values are:

Material	*UTS lbf/in.^{2}*	*Proof stress lbf/in.^{2}*	*'E' lbf/in.^{2}*
70/30 brass	53 700	33 600	15×10^6
Ph. Br. CuSn5	74 000	56 000	$15{\cdot}5 \times 10^6$

The limiting condition with these metals is the *proof stress* ('yield') and both materials behave normally in bending. The value of 'E' is such that both materials give wall pressures which are comparable to iron rings of the same section.

Wall pressures

There is ample evidence which shows that piston ring wall pressures used in models have been too high in the past. This probably arises from the use of imperfect rings, which were expected to 'wear in', but (provided reasonable precautions are taken) this need not be necessary. Again, many commercial rings seem to be designed for use in IC engines of the highest performance rating likely to arise – leading to excessive pressures in all normal model IC engines, and very much too high for steam plant.

There is also a misunderstanding as to the operation of a piston ring. Provided that the wall pressure is reasonable, and (even more important) that the ring fits the cylinder, gas or steam pressure leaking to the space *behind* the ring will provide adequate sealing pressure.

The following design wall pressures, all in lbf/sq. in. are suggested; the lower figures can be used on those models which can be 'run in' on the bench, the higher where the rings have not been given a final skim on the outside diameter.

Stationary steam engines	6 to 10
Locomotives, road and rail	8 to 12
'Classical' gas and petrol engines	6 to 8
'Workhorse' IC engines (e.g. for locos)	12 to 15
'High performance' IC engines	20 upwards

It is always prudent to start with the *lowest* wall pressure, increasing this only if experience shows this to be necessary. Piston ring friction is one of the highest losses in any engine.

Ring design

From expression (3) above it will be seen that the ratios g/D (free gap/diameter) and D/t (dia./radial thickness) all have direct effects, the latter according to a cube law. The design process can be simplified by using a constant ratio of g/D, and then deriving D/t from this. This then allows minor adjustments to be made simply by altering the gap during manufacture. (It is always prudent to make a couple of experimental rings first.)

For IC engines the late Prof. Dennis Chaddock used a ratio g/D of 4. Steam engine practice is to use 10, as the wall pressure required is much lower. These ratios lead to equation (3) appearing as follows.

$$g = D/4: \quad p = \frac{4E}{7 \cdot 06} \times \frac{1}{D/t(D/t - 1)^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

$$g = D/10: \quad p = \frac{E}{70 \cdot 6} \times \frac{1}{(D/t - 1)^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Naturally these gap ratios can be varied at will, but it will be seen that from either expression the only number to be determined is the radial thickness, as the bore is known. The 'Chaddock rule' leads to thicker rings, but as the free gap is relatively large the risk of breakage during installation is small. On the other hand, the stresses in the thinner rings used in steam plant mean that, despite the smaller free gap, installation breakage is rare. (Except that due to accident!)

The *axial width* of piston rings is a matter of choice, but wide rings do tend to put up losses. A good compromise is to make the width $\frac{2}{3}$ to $\frac{3}{4}$ of the radial thickness, with a lower limit of $\frac{1}{16}$" (1·5 mm) but rings as narrow as $\frac{1}{32}$" (0·75 mm) have been used by very skilled machinists.

Running gap

Apprehensions about leakage past the ring gap are unjustified. A slight running gap is essential, as ring butting can do enormous damage. A minimum installed gap of 0·001" (0·03 mm) is recommended and in most applications a worn s.ate gap of five times this figure will not cause trouble.

Manufacture

Space prevents any discussion of this in detail, but two very important points are emphasised.

(1) The fit of the ring to the bore is vital, and it is essential, whatever manufacturing method is used, that the ring be given a *final sizing skim*. This can be done using a fixture similar to that shown in Fig. 3. This should be self-explanatory, but the 'sizing sleeve' is made to fit the *rings* before skimming, not the cylinder bore.

(2) The 'fit' of the ring in its groove is very important, and the surface finish of the sides of both ring and groove should receive careful attention. The ring must be free in the groove, but not sloppy.

Oil scraper rings

These are needed only on high-speed totally enclosed IC engines, where large quantities of oil may be thrown up onto the bore. The wall pressures needed can be found only by experiment (and should be as low as possible), but *may* be from twice to three times that of the pressure rings in some cases. This is achieved by using the normal pressure ring section, but machining either a step or a groove in the rubbing surface. In fact, the ring pressure is not as important as the free passage of oil *from* the ring, and most difficulties are due to the holes in the piston skirt being far too small. These should not be less than $0 \cdot 6\,w$ in diameter and be pitched as close together around the piston skirt as prudence will permit (see Fig. 2).

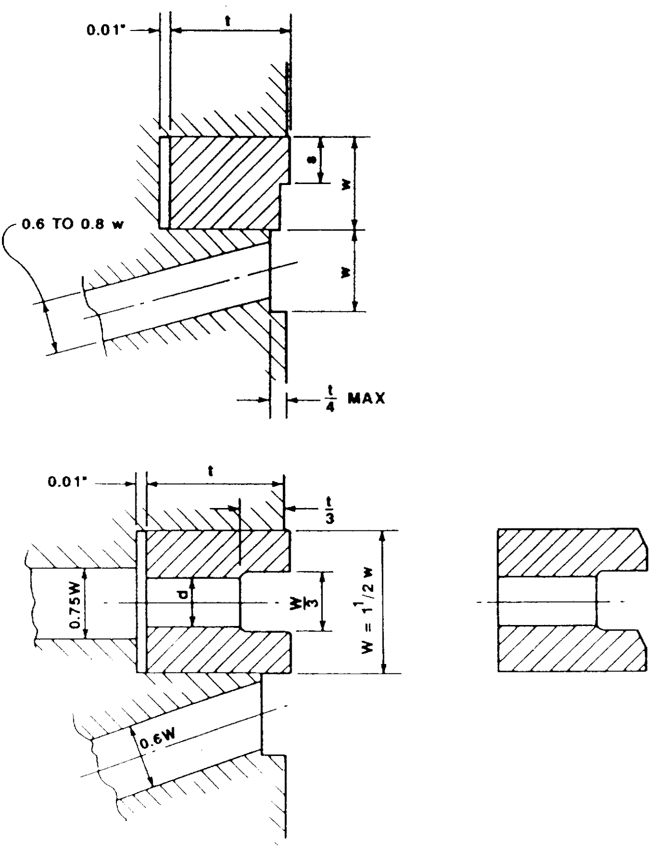


Fig. 2

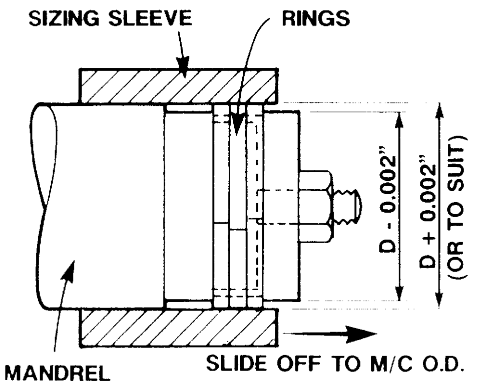


Fig. 3. MACHINING FIXTURE

SECTION TWELVE
ELECTRICAL MEMORANDA

COLOUR CODES

Colour code for cables

The obsolete code for flexible cables is shown in brackets. Many appliances and some plugtops may be found marked in this code, and care should be taken when connecting them.

Conductor carrying:	*Non-flexible cable code*	*Flexible cable code*
Earthing	Green (green/yellow)	Green/yellow stripe (green)
Live, AC single phase	Red	Brown (red)
Neutral, 1- or 3-phase	Black	Blue (black)
3-phase, phase R	Red	Brown (red)
phase Y	Yellow (or white)	Brown (yellow)
phase B	Blue	Brown (blue)
DC positive, 2-wire	Red	Brown
DC negative, 2-wire	Black	Blue
DC positive, 3-wire	Red	—
DC 'middle' 3-wire	Black	—
DC negative, 3-wire	Blue	—

An orange conductor may be found in 4- or 5-core cables, for use when (e.g.) wiring 2-way switches, remote indicators and the like.

Note the possibility of confusion with the new code for flexible conductors carrying three-phase supplies. It is recommended that the red, yellow, and blue phases be identified at both ends of the flexible and colour sleeves fitted.

Colour code for fuses

13 amp plugtops. Not all manufacturers code their fuses, but all are marked. Codes used by some are: 2 or 3 amps, blue; 5 amps, red; 13 amp, brown. For small glass (radio type) fuses coloured dots on the glass indicate the rating: dark blue, 1 amp; light blue, $1\frac{1}{2}$ amp; purple, 2 amp; white, 3 amp. Fuses in the milliamp range should be marked in digits on the end cap.

Colour code for small resistors

The colour code is gradually being replaced by a digital indicator, and both are given overleaf. Colour coded resistors have four (occasionally five) colour bands. The first – at the end of the component – indicates the first digit of the value; the second band the second digit; the third band the multiplying factor; and the fourth band, the tolerance on the value. If there is a fifth band, this indicates the grade of 'stability' of the resistor material. If only 3 bands are present, the tolerance is 20%.

Colour	*First figure*	*Second figure*	*Multiplier*	*Tolerance*	*Stability*
Pink	—	—	—	—	high
Silver	—	—	0·01	10%	—
Gold	—	—	0·1	5%	—
Black	—	0	1	—	—
Brown	1	1	10	1%	—
Red	2	2	100	2%	—
Orange	3	3	1000	—	—
Yellow	4	4	10 000	—	—
Green	5	5	100 000	—	—
Blue	6	6	1 000 000	—	—
Violet	7	7	10 000 000	—	—
Grey	8	8	100 000 000	—	—
White	9	9	1 000 000 000	—	—

Thus a resistor with bands yellow, violet, red, silver would be a 4 700 ohm component 10% tolerance of normal stability, and red, yellow, silver, gold, pink would be high stability, 5% tolerance 0·24 ohms.

The current BS 1852 code is as follows: There are two digits and one letter to indicate the ohmic value and a further letter to show the tolerance. The first figure represents the first digit of the ohmic value and the second figure the second digit. The first letter is R when the value is in ohms, K when the value is in kilohms (1000s of ohms) and M when the value is in megohms (millions of ohms). The *position* of this letter indicates the location of the decimal point (e.g. 6K8 is 6·8 kilohm – 6800 – 3M9 is 3·9 megohm and 4R7 = 4·7 ohm). The final letters indicate tolerances as follows: F, 1%; G, 2%; H, 2½%; J, 5%; K, 10%; M, 20%.

Examples:

R33M = 0·33 ohm, 20%. 4R7K = 4·7 ohm, 10%. 270RJ = 270 ohm, 5%.
6K8G = 6 800 ohm, 2%. 47KK = 47 000 ohm, 10%. 3M9M = 3·9 megohm, 20%.
1ROM = 1·0 ohm, 20%.

Colour coding for capacitors is similar, but the 'stability' code is usually absent, and imported codes may vary.

FUSES

Fuses are intended to protect the house or shop wiring from overheating should a fault develop in the system; they are *not* as a rule suitable to protect the actual apparatus, though they may sometimes do so.

Modern practice is to fuse 'single pole' – that is, a fuse is fitted in the 'line' or 'positive' lead only. The earlier practice of double-pole fusing – with a fuse in each lead – could result in the neutral supply being broken leaving both 'line' lead and the apparatus connected to it at mains potential above earth. Fusing of transformer circuits requires special care, and in many cases fuses will be required in both primary and secondary circuits, especially if supplying silicon rectifiers. In the case of 'VARIAC' variable output transformers the makers should be consulted. Auto transformers should *never* be fused in the 'neutral' on the output side, as an open circuit here automatically raises the other output lead to mains potential.

Fuses for use in reactive circuits (including transformers and motors) should be of the 'anti-surge' type, which will carry a high starting current for a short time – about three times as long as can a normal fuse; this will permit the current surge to pass without harm on switching on.

In circumstances where an equipment failure might cause a dead short circuit across the house mains, the use of High Rupturing Capacity (HRC) cartridge fuses is recommended. These reduce the risk of the fuse failing explosively.

Fusewire

In an emergency, ordinary wire may be used to replace fuselinks if the correct replacement is not available. However, the fusing current does depend to some extent on the type of holder, and the temporary replacement should be refitted with the proper cartridge or link as soon as possible.

Fusing current amps	*Copper*		*Tin*		*Lead*	
	Dia., in.	*SWG*	*Dia., in.*	*SWG*	*Dia. in.*	*SWG*
1	0·0021	47	0·0072	37	0·0081	35
2	0·0034	43	0·0113	31	0·0128	30
3	0·0044	41	0·0149	28	0·0168	27
5	0·0062	38	0·0210	25	0·0236	23
10	0·0098	33	0·0334	21	0·0375	20
15	0·0129	30	0·0437	19	0·0491	18
30	0·020	25	0·072	15	0·080	14

CABLE SIZES

Non-flexible PVC insulated main wiring cable, 2-core and earth, 600/1000 volt rating. (The normal house-wiring cable.)

		Enclosed in pipe or trunking		*Surface laid*	
Conductor nom. area sq. mm	*No. & dia. of wires mm*	*Current rating (amps)*	*Voltage drop/metre (millivolt)*	*Current rating (amps)*	*Voltage drop/metre (millivolt)*
1·0	1/1·13	14	40	16	40
1·5	1/1·38	18	27	20	27
2·5	1/1·78*	24	16	28	16
4·0	7/0·85*	32	10	36	10
6·0	7/1·04*	40	6·8	46	6·8

*Twin cable only – separate earth cable required.

Flexible rubber or PVC insulated twin cords.

Conductors		*Current rating amps*	*Max permitted suspended mass on twin cord* kg*
Area sq. mm	*No. & dia. of wires mm*		
0·5	16/·020	3	2
0·75	24/·020	6	3
1·0	32/·020	10	5
1·5	30/·025	15	5
2·5	50/·025	20	5
4·0	56/·030	25	5

*Assumes proper cord-grips at both ends of cable.

Note: All cable ratings should be checked against the local electricity company's regulations.

CHARACTERISTICS OF SMALL ELECTRIC MOTORS

No single-phase AC motor is inherently self-starting, but once near synchronous speed with the mains frequency the motor will pull up to full speed. In early days it was not uncommon for fractional horsepower (fhp) motors to be started by 'pulling on the belt' but current types are fitted with an arrangement to convert the machine to a two-phase motor for starting. Under these conditions the current taken is high, as the extra starting winding is very inefficient. In all save the smallest motors, therefore, a speed-sensitive switch is fitted internally to disconnect or alter the arrangement of starting winding, the machine running thereafter as a simple single-phase induction motor. (Exceptions are the very small 'shaded pole' motors used in fans and tape-recorders, and in the so-called 'universal' motor, referred to later.)

The types of fhp motor are distinguished by the method of converting to 2-phase working for starting. These are:

Split-phase in which the necessary phase-shift to the starting winding is obtained by a combination of inductive and resistive characteristics in the winding.

Capacitor start/induction run where a capacitor provides the phase-shift at starting, disconnected when running.

Capacitor start-and-run in which capacitance is used both in starting and running. In some cases a centrifugal switch may be used to connect extra capacitance at starting, to improve the starting torque.

The centrifugal switch usually opens at about 75% of full speed, and recloses somewhat lower.

Typical starting characteristics of fhp motors are as follows:

Type of motor	*Starting torque as % of full-load torque*	*Starting current as % of full-load current*
Capacitor start induction run	250/350%	450/550%
Split phase	175/200%	800/1100%
Capacitor start-and-run	about 50%	400%
Shaded pole	about 50%	200%
3-phase	200/250%	600%

Approximate full-load currents of typical single-phase fhp motors are as follows:

Output		*Split phase & capacitor start*	*Capacitor start-and-run*
Watts	*hp*	*Amps*	*Amps*
90	$\frac{1}{8}$	1·4	1·0
120	$\frac{3}{16}$	1·6	1·1
180	$\frac{1}{4}$	2·3	1·6
250	$\frac{1}{3}$	2·9	2·0
370	$\frac{1}{2}$	3·8	2·7
550	$\frac{3}{4}$	5·1	3·8
750	1	7·0	Not made

From the above it will be seen that to get the starting torque provided by a $\frac{1}{2}$hp capacitor start motor in one of split phase design it would be necessary to fit a $\frac{3}{4}$hp machine, and the starting current would be about $2\frac{1}{2}$ times greater.

For applications where the starting torque is of little consequence the capacitor start-and-run motor is about 40% more efficient on full load. The split-phase motor is an economical solution for driving machines with clutch start (provided the rotor inertia will cope with the clutch – see later) being cheaper than the capacitor start machine. The latter is essential for the driving of machine tools where the machine is started and stopped by switching the motor.

There is another consideration to be borne in mind – the heat generated at starting. The higher the starting current, the greater the heat generated in the windings. The cooling of the machine – usually by fan – is designed to cope with the normal full-load current plus a certain amount of heating from the starting winding. Makers therefore recommend a limit to the number of starts per hour, usually by specifying the maximum 'starting time' for any one start together with the total starting time per hour. Thus a capacitor start induction run machine might be rated for 3 second starts, and 45 seconds total starting time per hour. This means in effect, that only 15 starts could be recommended if each took 3 seconds; or 90 starts, if it took the motor only $\frac{1}{2}$-second to come up to speed each time.

Typical recommendations are (for 1450 rpm motors):

3-phase motor .... 60 starts/hr (say 1 sec. each)
Split phase .... 10 starts of 0·5 sec./hr
Capacitor start induction run .... 15 to 20 starts of 3 sec./hr
Capacitor start-and-run .... 60 starts/hr (say 1 sec. each)

These recommendations assume that the motor is running under full load continuously once started, so that in the case of model engineering applications some latitude may be expected. Nevertheless, this factor must be borne in mind when selecting a motor type. Note that the rating of the starting winding (or the capacitor) governs the *length* of the start, and if the machine is to start on load (e.g. driving a compressor) then a 3 second start is quite probable. It will be observed that the low starting torque available with the 'capacitor run' machine is a serious limitation, as it will take four times as long to accelerate a given load as will a split phase motor of the same power.

Experienced model engineers will point out that they have exceeded these starting specifications 'for years'. This may be so, and there is a considerable margin for error. However, modern motors are very much lighter than their counterparts of some 30 years ago, and this has meant that some of these margins are now less than they were.

In a number of cases machines are now being supplied with motors larger than formerly, to deal with the frequent starts usual in model engineering.

'Universal' motors

These machines are modified series-wound DC motors. Their main use is on low-power drives ($\frac{1}{10}$hp or less) in cases where speed variation is required, and in portable electric tools (up to 1hp) where advantage is taken of their compactness. The notable characteristics are: (a) They develop up to $2\frac{1}{2}$ times the full load torque at zero speed. (b) The speed drops steeply with increase in load – and unloaded motors *can* literally run away. (c) The speed can be controlled simply by a variable series resistance, or more economically by the use of a 'Variac' type of transformer. The second of these characteristics makes them somewhat difficult to use on a centre lathe, as the speed can seldom be predicted on load without elaborate electronic controls. For portable tools this is not so great a disadvantage, as naturally a small drill should be run faster than a large one. They should *never* be used for tool grinders.

FHP MOTOR CONNECTIONS

The standard colours for coil leads on British made motors are:

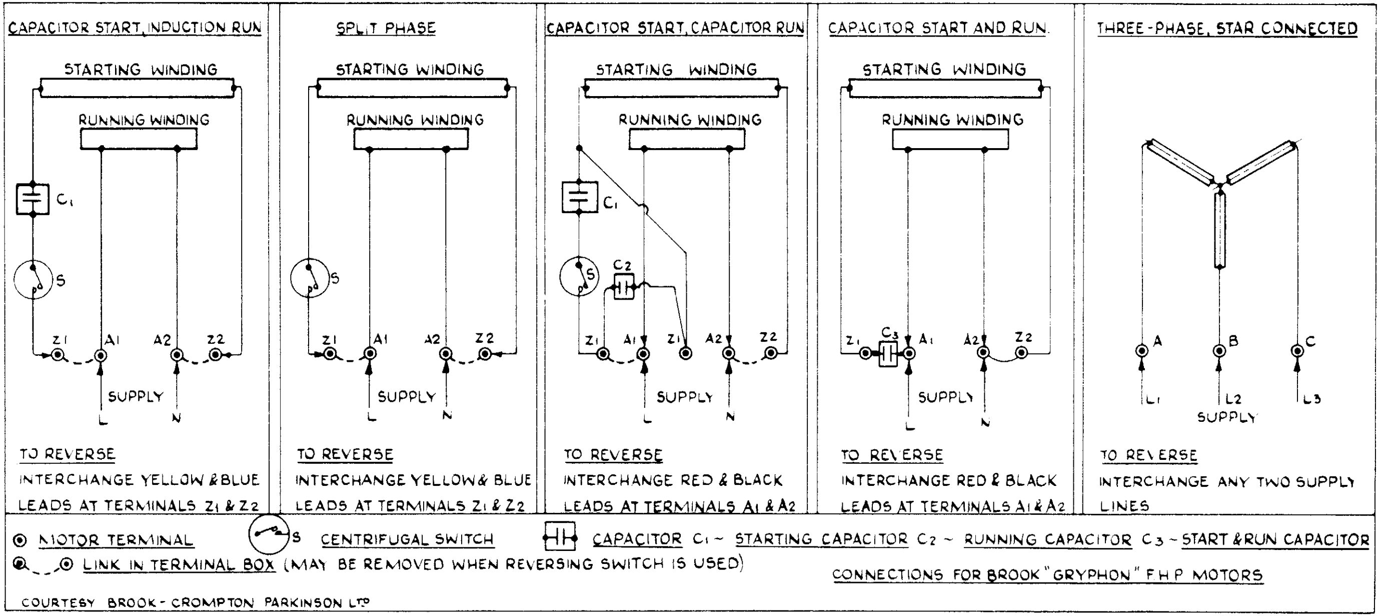
Single-phase motors,	Running winding, red and black		
	Starting winding, yellow and blue		
Three-phase motors,	*Phase*	*Live end*	*Neutral (star point)*
	A or red	Red	Black
	B or yellow	Yellow	Brown
	C or blue	Blue	White

Through the courtesy of Messrs Crompton Parkinson Ltd the following wiring diagrams are given, also showing the changes needed to reverse the rotation of the motor. Note that the capacitors are usually outside the motor casing and are connected with leads that *may* have colours corresponding to the coil ending colours but which are not connected to the same terminals as the coils. Care is needed to identify these correctly.

Note: Terminal codes to BS 4999 Pt 108 are now coming into use.

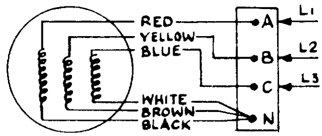
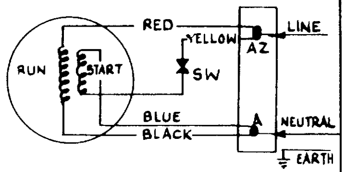
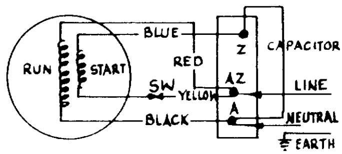
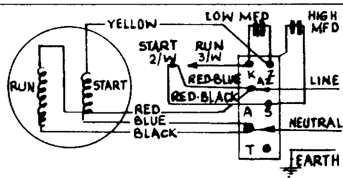
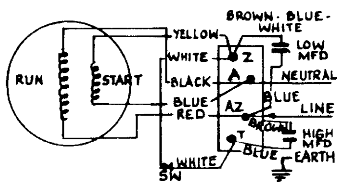
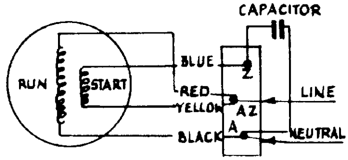
On **3-phase** motors U, V, W replace A, B, C. Star point (if brought out) marked N.

On **single phase** motors the main winding is now U1, U2 etc. and the starting winding Z1, Z2. Generally suffix 1 indicates the normal 'line' connection on the main winding. C indicates a capacitor terminal.



CONNECTIONS FOR CROMPTON PARKINSON FHP ELECTRIC MOTORS

COURTESY BROOK CROMPTON PARKINSON LTD

MOTOR TYPE	CONNECTION DIAGRAM	TO REVERSE
THREE-PHASE		Interchange any two supply leads to motor.
SPLIT PHASE START		Change over blue and yellow starting winding leads at terminals A & AZ.
CAPACITOR START INDUCTION RUN		Change over blue and yellow starting leads at terminals AZ & Z.
CAPACITOR START CAPACITOR RUN (SPECIAL)		Change over blue and yellow starting winding leads at terminals A & Z.
CAPACITOR START CAPACITOR RUN		Change over blue and yellow starting winding leads at terminals A & Z.
CAPACITOR START-AND-RUN		Change over blue and yellow starting winding leads at Z and AZ.
NOTE: SW = SWITCH CLOSED AT START, OPEN TO RUN.		

These diagrams show alternative markings found in some early terminal boxes.

Reversing switches

These differ considerably in their internal connections as well as in mechanical arrangement. The following are typical of two types, but it is imperative to check with the switch actually supplied. A connection diagram is usually fixed inside the cover.

Dewhurst drum type. Single phase.
Supply – neutral to terminal 3, line to terminal 1.
Run winding – A or A1 (black) to 5, AZ or A2 (red) to 7.
Start winding – blue to 6, yellow to 2.
Earth – to earth terminal on casing.

Three-phase.
Supply – to terminals 1, 3 & 4.
Motor – to terminals 2, 6 & 8. Reverse connections to 2 and 6 if the motor runs the wrong way.

Rotary switch (Santon). Single phase.
Supply – neutral to F–, line to F+.
Motor – starting winding, blue to A1, yellow to A2.
running winding, black to terminal adjacent to A1.
red to terminal adjacent to A2.

Reversing D.C. motor. (shunt or compound wound).

Refer to switch and/or motor manufacturers, as in some cases the compounding winding is not brought out. Arrange the switch to reverse the armature and series field, if any, keeping the shunt field connections the same polarity. Arcing may occur on the reversing contacts, and the motor manufacturer's advice should be sought regarding the fitting of a discharge resistor across the motor terminals.

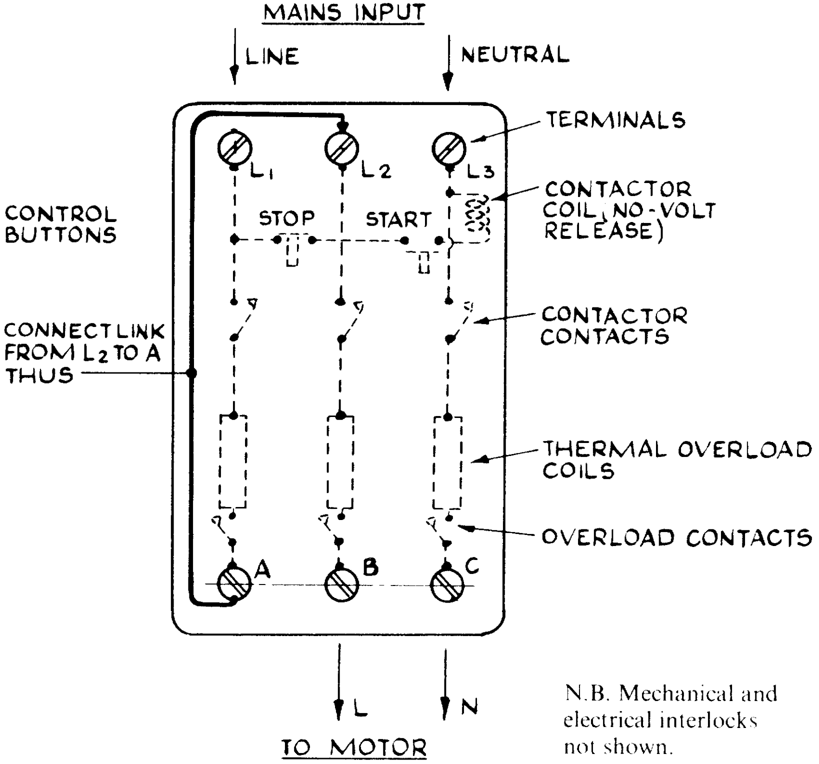
Series or universal motor. Drum type switch.
Supply – neutral (or +) to 4, line (or –) to 1.
Brushes – A to 6, AA to 2. (If not marked, either way round for trial, reverse if wrong rotation.)
Field coils – Y to 8, YY to 7 (trial as above if not marked).

MOTOR PROTECTION

The fitting of a simple fuse will *not* protect a single-phase motor; if the fuse is large enough to withstand the starting current (even with anti-surge type) it will carry sufficient current to burn out the motor. (Special dual-rated fuses *are* available for larger 3-phase machines.) The push-button type starter fitted with thermal overload cutouts is designed to cope with this situation.

Such starters can also serve to protect the driven machine and the operator. If a power failure occurs the workshop will be in darkness and it is easy to forget to switch off the lathe motor. If the supply is restored later, with no protection, the motor will restart with almost certain damage to the machine. The starters recommended for machine tools should be fitted with a '*no-volt release*' which opens the switch as soon as the supply fails and will not reclose until the button is again pressed to start. (Fuses are also required by regulation, to protect the shop wiring and the mains supply against a short-circuit or failure to earth.) A proper starter should be wired to all motors driving machine tools, especially those with auto-feed. The starter is wired between the socket outlet or isolater and the reverse switch if fitted. The latter can then be used for all normal machine operation. (Reversing push-button starters, with overload and no-volt release, are available, but expensive.) An additional advantage of this type of starter is that a remote stop-button can be fitted if desired; some would regard this as essential if children are active in the parent's workshop.

Most of such PB starters are designed for 3-phase, but will work equally well on single or 2-phase. However, it is not sufficient merely to wire through two of the connections. If this is done, the starter will see an 'unbalanced load' and react accordingly. The correct way to adapt to single phase is to wire two of the overload coils in series; this will be shown on the diagram attached to the starter, but in cases where such a diagram is absent, the sketch below shows a typical connection diagram. (This will differ in detail with different makes.)



3-PHASE PUSH-BUTTON STARTER CONNECTED FOR SINGLE-PHASE

Some practitioners prefer to omit the link between L_2 and A, connecting the motor 'line' terminal to A on the starter but the arrangement shown is recommended.

Thermal overload protection

In circumstances where the expense or complication of a PB starter is not worthwhile, motors can be fitted (when manufactured) with an internal overload switch which reacts to excessive winding temperatures, or to high current, sometimes with an external reset button, sometimes self-resetting after a time interval. The extra cost is several times less than that of a contactor type starter.

GENERAL POINTS ON MOTOR INSTALLATION

Motors fitted with centrifugal switches **cannot** be reversed until the motor speed has dropped *well* below the running speed, and preferably come to rest. If the drum switch is reversed with the centrifugal switch in the 'run' position, the motor will continue to rotate in the same direction. **Three-phase** motors can, however, be 'slugged' – if the drum switch is reversed the reversed winding acts as a brake and then reverses the rotation. This may seem to be an advantage but with centre-lathes the very rapid deceleration *can* unwind the chuck from the mandrel nose, so care should be exercised.

Modern motors have little inertia in the rotor, and in starting a machine with a high mandrel speed, the starting switch may cut in and out repeatedly. The remedy in this case is to warm up the machine at a lower speed for a few minutes first. In the case of clutch start machines, the remedy is to fit a flywheel to the motor pulley or change the aluminium motor pulley for an iron one and slightly slacken the first motion drive belt.

Lubrication

Most motors are now fitted with 'sealed for life' ball-bearings, but sleeve bearing machines need periodic lubrication. For normal speed motors, Shell Voluta 27 (or Vitrea 27) is satisfactory. For motors in cold situations, Tellus 11 may be used. Overoiling is to be avoided at all costs, as oil in the starting switch is a common cause of trouble, and may also damage the insulation. For rechargeable ball-bearings, Shell Alvania R.A. is a suitable lubricant.

Location

Drip-proof motors to IP22 are usually employed, as the old 'screen protected' is no longer acceptable, and totally enclosed fan cooled (TEFC) machines are hardly necessary in the amateur's machine shop. However, drip-proofing cannot operate upside down, and if the motor is to be mounted more than say 40° off the correct way up the end frames should be rotated (with due regard to any internal wiring) to bring it within this limit of the vertical. With machines on resilient mounts, this adjustment can be made there. On centrelathes there is sometimes a risk of swarf being thrown into the motor endshield through the ventilation holes. A simple sheet metal guard behind the chuck will avoid this danger. If cutting fluid is used in 'flow' rather than 'drip', conditions then enclosure to specification IP23 is advisable.

Earthing

Both common prudence and the IEE wiring regulations require all apparatus not specially insulated from all contact with live leads *and* having a fully insulated casing to be solidly connected to the earth conductor of the mains wiring system. It is not sufficient – indeed, can be downright dangerous – to connect equipment to a separate earth, such as a nearby water main. (Such a practice can also cause interference with radio and television sets, as well as parasitic hum in hi-fi equipment.) The earth conductor should be not less than one square millimetre in area or equal to the area of the 'line' conductor if this is greater, and should be bonded to the motor, the driven machine, *and* to associated metalwork such as drip-trays or guards even if attached metalically to the machine. It is important to note that the 'neutral' of either single or three-phase supplies is *not* necessarily at earth potential at the 'consumer's meters' and should be regarded as being 'live' when working on the system.

Size of motors

The physical dimensions of nearly all makes of British motors are standardised, though some motors for special applications may depart from this (e.g. motors from washing machines, refrigerators, etc.). The relevant British Standard is BS No. 2048. The screwed connection to terminal boxes, where fitted, is usually 20 mm conduit thread. The shafts are usually 0·0002″ to 0·0007″ down on the nominal diameter, and the spigots of end- or flange-mounted motors usually dead size to 0·0003″ down in diameter. Mountings of most makes are interchangeable, though some may require longer bolts than others.

TO RESTORE SULPHATED ACCUMULATORS

Empty the acid from the cell(s) and wash out with distilled water, taking care that no deposit is lodged between plates. Prepare a solution of sodium sulphate (Glauber's salts) and distilled water in the proportion of 1 of salt to 5 of water by weight and fill the cells to normal level. Charge at the ordinary charging rate for twice the normal charging time. Empty, wash out with distilled water and refill with fresh acid of specific gravity 1·208, or as recommended by the battery manufacturer. Discharge through a resistor or other load at about one tenth of the 'ten-hour rate' for 24 hours. Recharge at the normal rate and time and correct the specific gravity. (This may necessitate the removal of some liquid from the cells.)

This treatment is moderately successful with cells that are neglected, or sub-standard due to sulphation, but will not revive accumulators which are very badly sulphated or whose plates are disintegrating. It is, however, a cheap expedient to try out before going to the expense of replating.

Note that the initial recharging must be done on a charging board or trickle charger (the rate is not critical) and *not* by installing in a motor-car.

SECTION THIRTEEN
GENERAL

SPRING DESIGN, GEAR TEETH, SAFE LOAD IN STUDS AND BOLTS, FOUNDRY NOTES, MACHINING ECCENTRICS, BELT DRIVES, MODEL TRACK AND WHEEL STANDARDS, WOODRUFF KEYS, HEX. SOCKET SCREWS, PIPE FLANGES ETC.

SPRING DESIGN

Coil springs

The unsatisfactory results experienced by many model engineers when designing coil springs is largely due to the fact that the simple spring formulae assume that the wire is a straight 'bar' in torsion, whereas it is, in fact, a helix. This is allowed for in the expressions given below. It must, however, be appreciated that model springs are so small that the normal variations in wire diameter and coil diameter do have a measurable effect. Nevertheless, use of the corrected formulae will save a great deal of time wasted in trial springs.

The corrected expressions are as follows:

$$f_s = \frac{8k\,W\,D}{\pi\,d^3} \qquad (1)$$

OR

$$d = \sqrt[3]{\frac{8k\,W\,D}{\pi\,f_s}} \qquad (2)$$

$$R = \frac{c\,G\,d^4}{8\,D^3\,n} \qquad (3)$$

OR

$$n = \frac{c\,G\,d^4}{8\,D^3\,R} \qquad (4)$$

c = correction factor for rate.
k = correction factor for stress.
W = max load, lbf or newton.
D = mean coil dia., in. or mm.
d = wire dia., in. or mm.
f_s = allowable shear stress, lbf/sq. in. or N/mm^2.
n = number of free coils.
G = torsional modulus of material, lbf/sq. in. or N/mm^2.
R = 'spring rate', Lbf/inch of deflection or Newton/mm deflection.

Values of c and k are given below.

D/d	*2*	*4*	*6*	*8*	*10*	*12*	*14*	*16*	*18 up*
k	2·1	1·4	1·25	1·18	1·15	1·12	1·10	1·08	1·07
c	1·3	1·09	1·07	1·05	1·04	1·04	1·03	1·03	1·02

Springs having D/d less than 3 are difficult to wind, and those above 20 are somewhat fragile. The 'ideal' spring will have D/d between 7 and 9. (5 to 14 is acceptable).

(*Spring Design and Manufacture*, published by Nexus Special Interests, ISBN 0 85242 925 8, gives many charts and nomograms which greatly facilitate these calculations.)

To use formulae (1) to (4) it is thus necessary to 'guess' the ratio; as the curve is relatively flat, k and c for D/d = 10 may be used for the first approximation. The design is then checked after winding a trial spring, to find out how much the spring will 'unwind' after coiling – this alters the effective mean diameter D.

Values of f_s and G depend on the material used. It is normally considered good practice to use a maximum stress not exceeding 70% of the torsional elastic limit. If this is not known, then a safe assumption is to use 40% of the *ultimate tensile strength*. For a given material, the smaller the wire, the higher is likely to be the torsional strength, as shown on p. 7.7. The table below gives comparative values for other materials. A low stress is no disadvantage other than cost of material or space limitation.

Material (in 'hard' or 'spring' temper)	*Elastic limit lbf/sq. in. (4)*	*Modulus G millions lbf/sq. in. (4)*	*Suggested working stress lbf/sq. in. (4)*
Carbon steel	110 000	11·4	80 000
'Music wire'	141 000	12·0	90 000
Cr. Va. steel (1)	160 000	11·5	120 000
5% phos. bronze	65 000	6·0	45 000
70/30 brass, CZ106	40 000	6·1	30 000
BeCopper, CB101 (2)	80 000	7·0	60 000
Monel (3)	65 000	9·5	48 000

Notes: (1) Heat treated condition.
(2) Typical value; depends on manufacturing method.
(3) Stress for operation at 400°F (200°C) about 60% of these figures.
(4) For values in SI units (N/mm^2) divide by 145.

Design procedure

Ascertain the mean diameter D, bearing in mind that this will be the (OD – wire dia.) or (ID + d) and making an allowance for clearances in the housing.

Determine the maximum working load, W. Allow for any additional load that may occur when installing; if this is less than 15% above the normal working load it may usually be ignored.

Use equation (2) to find the appropriate diameter of wire, taking k = 1·15 in the first instance. From this, find the nearest larger commercial gauge of wire, wind a few coils on the appropriate mandrel, and measure the 'as wound' mean diameter D. Ascertain the true value of K from the table.

Use expression (1) and check the stress with the new values. If this is within +10% of the design figure, proceed; if not, alter the diameter D and start again. (If dimensions are constrained so that D cannot be altered, use another spring material.) Note that it doesn't matter if the working stress f_s is *lower* than 75% of the elastic limit.

To find the number of coils it is necessary to determine the 'rate' of the spring – the increase in force per inch of spring deflection. The more coils, the more 'springy' this will be, but the working *load* will not be altered. Generally, a spring with less than 5 or

6 free coils is not recommended.

Using the appropriate value of c from the table, apply equation (4) to find 'n'. Two coils must be added to this figure when winding, to allow for the inoperative end coils, usually ground off flat, or to form the hooks of a tension spring.

This 'rate' will decide the free length of the spring – and hence the pitch when winding – so that the spring may exert the correct force when compressed to the working length. Alternatively, it determines the length of the adjusting screw in cases where the spring has to be pre-loaded as in a safety valve.

Owing to the comparatively large effect of quite small errors in mean diameter, and the commercial tolerances on wire diameter, together with the somewhat variable amount of unwinding that occurs with amateur-made springs, a final test with scales or weighing machine is essential. Even with commercially made, machine-wound springs, a tolerance of about 5% on both load and rate must be expected.

There is evidence that tempering cold coiled carbon steel and music wire springs improves the performance. As little as three or four minutes at 200 to 250°C has an effect, and up to 20 minutes will give maximum performance. The oven at gas mark 8 (450°F) will both roast the joint and temper the springs; a normal chip-frying pan will achieve 200°C, just.

Axlebox springs

As it is very important that such springs be uniform in both load carrying and spring rate it is best to have these wound commercially on proper spring-winding machines. The design procedure above can be used to assess the probable size of the spring, but it is best to leave the *actual* design to the spring manufacturer. He will need to know the limiting dimensions (with a tolerance if possible) the load to be carried, and the probable deflections when running, together with the nature of the duty.

Example

Design a phosphor-bronze safety-valve spring assuming a mean diameter of 0·2″, for a valve of area 0·012 sq. in., blow-off pressure 100 lb sq. in. (Hence W = ·012 × 100 = 1·2 lbf.)

Use k = 1·15 in expression (2).

$$d = \sqrt[3]{\frac{8 \times 1\cdot15 \times 1\cdot2 \times 0\cdot2}{\pi \times 40\,000}}$$

$$= \sqrt[3]{0\cdot0000176} = \underline{0\cdot026'' \text{ dia.}}$$

Nearest commercial sizes are 22 or 24 gauge; use 22 = 0·028″ dia.

Wound on a $\frac{3}{16}$″ dia. mandrel the mean diameter measured 0·237″.

Hence D/d = 8·46; say k = 1·17, c = 1·05.

Check in expression (1).

$$f_s = \frac{8 \times 1\cdot17 \times 1\cdot2 \times 0\cdot237}{\pi \times 0\cdot028^3}$$

$$= \underline{38\,599 \text{ lbf/sq. in.}}$$

This is low, but safe; if 24 gauge wire is used the stress will be too high.

Check the number of coils. The 'rate' is not important; from the drawing the spring requires about $\frac{3}{4}$ inch in the working position, so assume that the free length will be 1″.

The compression will thus be $\frac{1}{4}''$ for a load of 1·2 lb, hence R = 4·8.

Use expression (4).

$$n = \frac{1\cdot05 \times 6\cdot6 \times 10^6 \times (0\cdot028)^4}{8 \times (0\cdot237)^3 \times 4\cdot8}$$

$$= \underline{8\cdot33, \text{ say nine coils.}}$$

To which add two for the ends, making 11 coils to be wound, to a free length (after grinding flat) of one inch. Set lathe changewheels for 11 tpi.

Check the 'lift'. Assume the valve lifts by an amount equal to 10% of the diameter ($\frac{5}{32}'' = 0\cdot156''$) say 0·015″. Then additional spring-force is $0\cdot015 \times 4\cdot8 = 0\cdot072$ lbf. The valve area is 0·012″, so the additional steam pressure will be $0\cdot072/0\cdot012 =$ 6 lbf/sq. in. or 6% of the boiler-pressure. This is satisfactory.

(*Note:* the 'valve area' in this case is the passage area *minus* the area of the spring rod. Hence the figure of 0·012 sq. in. instead of the passage area of 0·0192.)

In this example, the spring finished at 1·062″ free length when wound, and had to be compressed to 0·71″ to reach the desired 'blow-off' pressure. This is a normal tolerance on a hand-wound spring.

Conclusion

The expressions and corrections, and the tabular values of stress are equally applicable to tension and to compression springs. It should be remembered that compression springs will expand in diameter, and tension springs contract, when under load, and the coils will tend to unwind slightly. The expressions are *not* valid for helical springs used in *torsion*; in this case the wire is stressed in bending, and reference should be made to the book mentioned on p. **13**.2 when dealing with this class of spring.

LEAF SPRINGS

A leaf spring may be treated as a modified simply-supported beam loaded at the centre. In many applications (e.g. loco springs) a beam of sufficient strength to carry the load would either be too deep to give the necessary 'springiness' or too wide to be accommodated in the available space. To overcome this problem the spring is, in effect, a wide one cut up into slices, these slices then being placed one on top of the other, to produce the well-known laminated spring. As the bending stress is least at the ends and greatest at the centre each leaf is made shorter than the one above, giving a more or less uniform stress along the length. It is, however, usual to have the two top leaves of equal length, as an insurance against catastrophe should the master leaf fail.

The stress in a laminated spring is given by:

$$f = \frac{3\,W\,L}{2\,n\,b\,t^2}$$

Deflection at the centre is given by:

$$d = \frac{3\,W\,L^3}{8\,n\,b\,t^3\,E}$$

f = stress, lbf/sq. in. or N/mm².
W = centre load, lbf or newton.
L = span, inches or mm.
n = no. of leaves.

b = width of leaf, in. or mm.
t = thickness of one leaf, in. or mm.
d = deflection at the centre, in. or mm.
E = Young's modulus for the material in the appropriate units.

Because the analysis is approximate, and does not allow for the fact that laminating the spring distributes the load along part of the length of the upper leaves instead of concentrating it at the centre, the actual deflection may be 5% to 10% greater than is given by the formula. *Note* that 'L' is measured under normal working load – usually with the top leaf flat or nearly so.

It is not possible to make a 'true scale' spring using the prototype material and stress. If this is done the leaves will be much too thin, and the deflection far too small. If the stress is reduced to 1/scale (i.e. $\frac{1}{12}$ for 1 in./ft) the deflection will still be too small – about 1/scale again. To overcome this it is usual to employ working stresses well below the normal and materials which have a much lower value of 'E' than that of the prototype spring steel. The figures below give values for some typical materials. As model engineers may have to design other than 'scale' springs, a suggested normal working stress is given; this should be divided by the scale when used in scale applications.

Material	*Working stress lbf/sq. in.*	*Young's modulus lbf/sq. in.*
Carbon steel, spring temper	75 000	29 000 000
70/30 brass, CZ106, hard temper	28 000	16 000 000
5% phos. bronze, PB102, hard temper	70 000	18 000 000
Beryllium-copper, CB101, WH temper	80 000	18 000 000
'TUFNOL' (fine fabric laminate)	8 000	1 000 000 to 1 300 000

Divide by 145 to obtain values in N/mm^2.

In all the above materials the method of manufacture may affect the values somewhat, and it is advisable to make a sample single-leaf spring and check deflection against load; the result may then be substituted in the deflection formula to establish the true value of 'E'.

It is important to note that the leaves *must* be free to slide one upon the other if the behaviour of the laminations is to be correct. It is useless to solder together a number of very thin leaves in the hope of achieving 'scale appearance'.

The expressions above differ from those found in some books dealing with full-sized practice. The leaves of such springs are very rigidly anchored together at the centre. The spring then behaves as if it were two independent cantilevers supported at the centre anchorage.

Model springs cannot be sufficiently rigidly fixed at the centre, and a closer approximation is to treat them as if they were beams simply supported at the ends. This results in different constants in the formulae.

This contrast in theoretical treatment emphasises the need for experiment in model leaf spring design.

For a more detailed examination of the design process see *Spring Design and Manufacture*, Workshop Practice Series No. 19 Nexus Special Interests, ISBN 0-85242-925-8.

GEAR TEETH

Where the word 'pitch' appears alone this refers to the *circular pitch*, denoted by 'p', which is the length of the arc on the pitch circle between similar faces of successive teeth.

Diametral pitch is the number of teeth in the gear divided by the pitch diameter, and is denoted by 'P', or 'DP'. Unless otherwise stated the DP is taken to be in inch units.

Module, denoted by 'm', is the pitch diameter divided by the number of teeth; it is thus the reciprocal of DP. The module may be either metric or imperial, and the units should always be stated. It is, however, more used in metric terms than is DP.

From the above it follows that:

$$p = \pi/DP = \pi m$$
$$DP = \pi/p = 1/m$$

The **Addendum** is the height of the tooth above the pitch circle; the **Dedendum** is the depth of the tooth space below the pitch circle. In general the addendum is equal to $0{\cdot}3183p = 1/DP = m$, and the dedendum is given by $0{\cdot}3683p = 1{\cdot}157/DP = 1{\cdot}157m$, for 20° involute spur gears.

Hence, the outside diameter of the gear-blank before cutting will be $(t + 2)/DP$, where t is the number of teeth [or $OD = (t + 2)m = p(t + 2)/\pi$]. The above apply for gears of a reasonable number of teeth; if the number is less than about 16 teeth, then a correction should be made to avoid interference, and the cutter manufacturer should be consulted.

The following table shows the relationship between p, DP, and m, over the range of pitches likely to be used by model engineers. p and DP are for inch units, m in millimetre units.

DP *in.*	*p* *in.*	*m* *mm*	*DP* *in.*	*p* *in.*	*m* *mm*	*DP* *in.*	*p* *in.*	*m* *mm*
8	·3927	3·1749	**15**	·2094	1·6933	25·4	·1237	**1**
8·378	$\frac{3}{8}$	3·0318	**16**	·1963	1·5875	**26**	·1208	·9769
8·467	·3711	**3**	16·76	$\frac{3}{16}$	1·516	**28**	·1122	·9071
10	·3142	2·54	16·93	·1855	**1½**	**30**	·1047	·8467
10·053	$\frac{5}{16}$	2·5265	**18**	·1745	1·4111	**32**	·0982	·7847
10·16	·3092	**2·5**	**20**	·1571	1·2700	**34**	·0924	·7470
12	·2618	2·1166	**22**	·1428	1·1545	**40**	·0785	·6350
12·566	$\frac{1}{4}$	2·0212	**24**	·1309	1·0583	**50**	·0628	·5080
12·7	·2472	**2**	**25**	·1257	1·0160	50·27	$\frac{1}{16}$	·5053
14	·2244	1·8143	25·133	$\frac{1}{8}$	1·0106	50·8	·0618	$\frac{1}{2}$

For a complete list of pitches, refer to BS436 – machine cut gears. The same specification provides complete details of gear strength calculations and should always be used for load-carrying gears of any importance.

Gear cutters

Most model engineers will use profiled milling cutters to form gear teeth; unfortunately this means that a different cutter will be needed not only for each pitch used, but also for a range of sizes of gear. The range of size (number of teeth) that can be cut with each cutter is as follows:

BS 436 involute form

Cutter number	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*
Teeth in gear	135–up	55–134	35–54	26–34	21–25	17–20	14–16	12 & 13

Berner's epicycloidal form (constant addendum clock teeth)

Cutter number	*1*	*2*	*3*	*4*	*5*	*6*
Teeth in wheel	135 up	55–134	35–54	26–34	21–25	17–20
Cutter number	*7*	*8*	*9*	*10*	*11*	
Teeth in pinion	14–16	12–13	11–12	8–9	6–7	

Similar cutter numbers apply to BS 978 cycloidal teeth.

Some manufacturers can supply cutters with more limited range, which give more accurate profiles at the extremes of the group. Thus a 3A cutter might serve 35–44 teeth, and 3B, 45–54.

Gear blanks

The overall diameter of the gear blank is found by adding *two* to the number of teeth and dividing by the diametral pitch, assuming the normal 20° involute form. Thus a 40-tooth wheel of no. 10 DP will be 42/10 = 4·200″ blank diameter. For circular pitch wheels the same calculation can be made, using the 'equivalent' DP given in the table above).

SOUND

Decibels

A decibel is $\frac{1}{10}$ of a bel, which is a measure of power *ratio*; an increase by 1 bel means that the power of the signal (sound or otherwise) has gone up 10 times (1 = log 10) and +2 bel means an increase of 100 times (2 = log 100). The bel is too large for normal use and the decibel (dB) is used instead. An increase in sound power of +1 dB is the smallest increase that can be detected by a trained ear, and represents a power increase of 1·25/1. The following table shows the power ratios represented by various changes in decibel (dB) measurements.

dB	+1	+2	+3	+4	+5	+6	+7	+8	+9	+10	+20	+30
Power ratio	1·25	1·6	2·0	2·5	3·2	4·0	5·0	6·0	8·0	10·0	100	1000

dB	−1	−2	−3	−4	−5	−6	−7	−8	−9	−10	−20	−30
Power ratio	0·8	0·63	0·5	0·4	0·31	0·25	0·2	0·17	0·125	0·1	0·01	0·001

Example – a change from 87 to 90 dB measured means a doubling of the sound power; a change from 90 to 84 dB means the power is reduced to one quarter of the original.

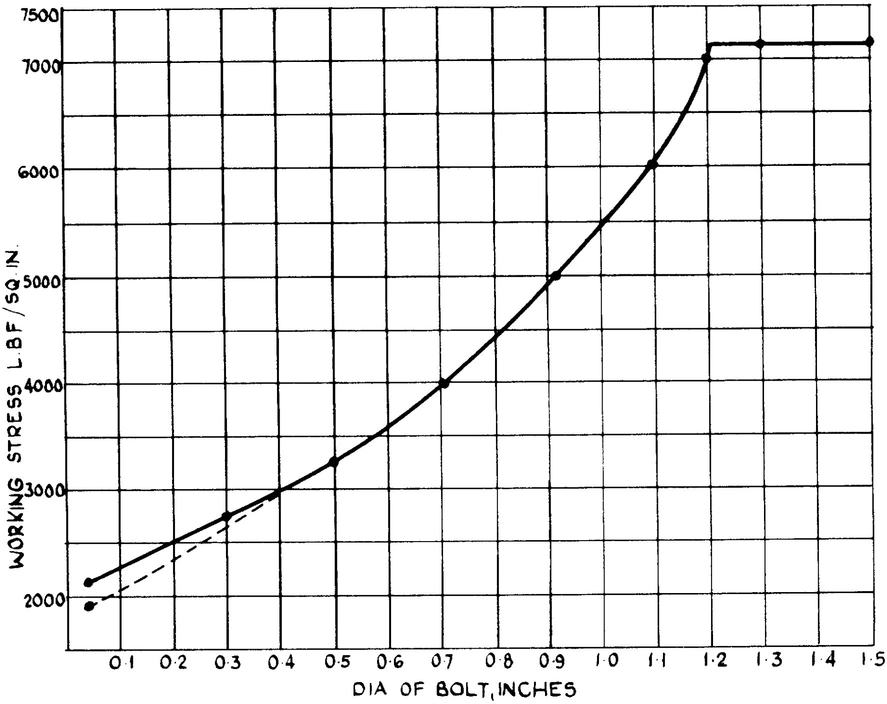
In absolute figures, 35 dB is very quiet indeed; 60 dB is 'loud but tolerable'; 80 dB is 'offensive'. 125 dB is lethal.

To prevent long-term damage to hearing, ear muffs should be worn at noise levels above 75 dB. Repeated exposure to 85 dB can cause deafness in later years.

SAFE LOAD IN SCREWS AND BOLTS

The working load in a bolt is determined by a nominal stress applied to the core diameter, this stress allowing a reasonable safety factor over the yield strength or 0·1% proof stress of the material. However, the major stress in a bolt below about 1″ dia. is that resulting from the tightening of the nut, and the smaller the diameter the greater this stress will be. Indeed, any bolt in mild steel smaller than $\frac{3}{8}$″ diameter can be overtightened to failure.

An old rule of thumb for *mild steel* is that working stress $f = 2500 + 3000\,D^2$ lbf sq. in., where D = bolt dia. This is shown graphically below. For other materials of ultimate strength different from 25 tons/sq. in., the safe stress should be altered in proportion.



Safe working stress in **mild steel** studs and bolts allowing for tightening stress under normal loads. Based on ENI material with UTS 25 ton/sq. in.
Dotted part of the curve recommended for commercial 'all thread' stud-rod.
For values in N/mm^2 divide lbf/in^2 by 145.

The table opposite shows working loads in bolts for various nominal stresses, but it is recommended that no threaded fastener made of mild steel below 0 BA (or 6 mm) be stressed above 2500 lbf sq. in. Note that this table refers to *mild* steel only.

Dia.	Approx. working load lbf at a stress of (lbf/sq. in.)									
	1000	*2000*	*3000*	*4000*	*5000*	*6000*	*7000*	*8000*	*9000*	*10 000*
$\frac{5}{8}$" WHIT			610	816	1020	1224	1418	1620	1824	2040
$\frac{1}{2}$" WHIT			360	484	605	726	850	970	1090	1210
$\frac{1}{2}$" BSF		276	415	554	692	831	969	1110	1246	1385
$\frac{3}{8}$" WHIT		135	204	272	340	408	478	546	614	680
$\frac{3}{8}$" BSF		152	228	304	380	456	532	608	684	760
$\frac{5}{16}$" WHIT		92	136	184	230	275	322	378	414	—
$\frac{5}{16}$" BSF		102	152	203	254	305	356	406	457	—
$\frac{1}{4}$" WHIT	27	54	81	108	135	162	189	216	—	—
$\frac{1}{4}$" BSF	32	63	95	126	158	190	221	253	—	—
$\frac{3}{16}$" WHIT	14	28	42	57	70	84	98	112	—	—
$\frac{3}{16}$" BSF	17	34	51	68	85	102	120	137	—	—
$\frac{1}{8}$" × 40	6·8	13·6	20·4	27	34	41	48	—	—	—
0 BA	28	56	84	112	140	168	196	224	252	280
1 BA	21·7	43	65	87	108	130	152	174	195	217
2 BA	16·9	33·8	51	68	84	101	118	135	152	—
3 BA	12·6	25·2	38	50	63	75	88	100	113	—
4 BA	9·6	19·4	29	38	48	58	67	76	—	—
5 BA	7·5	15	22·5	30	37	45	53	60	—	—
6 BA	5·7	11·4	17·1	23	28	34	40	—	—	—
7 BA	4·5	9·0	13·5	18	22·5	27	31	—	—	—
8 BA	3·4	6·8	10·2	13·6	17	20	—	—	—	—
9 BA	2·5	5·0	7·5	10	12·5	15	—	—	—	—
10 BA	2·0	4·0	6·0	8	10·0	12	—	—	—	—
12 BA	1·12	2·24	3·4	4·5	5·6	—	—	—	—	—
14 BA	0·63	1·26	1·89	2·5	3·1	—	—	—	—	—
16 BA	0·39	0·78	1·17	1·56	—	—	—	—	—	—

(N/mm^2 = 145 × lbf/in.^{2}; 1 lbf = 4·45 newton.)

STRENGTH OF HIGH-TENSILE SOCKET HEAD MACHINE SCREWS

These screws are made from oil-hardened and tempered HT steel having a UTS of between 75 and 90 tonf/sq. in. (1180–1370 N/mm^2), a yield (0·2% proof) stress of 34 tonf/sq. in. (1060 N/mm^2) and an elongation figure of about 10%. These figures vary a little with the size of screw, as does the hardness, which is typically about 400 brinell.

The recommended *maximum* tightening torque for the smaller sizes is given in the table below, together with the *induced tightening stress*. Note that this is *not* the safe working load, but that solely due to tightening the bolt.

Imperial

Nom. dia.	$\frac{3}{16}$ W	$\frac{3}{16}$ F	$\frac{1}{4}$ W	$\frac{1}{4}$ F	$\frac{5}{16}$ W	$\frac{5}{16}$ F	$\frac{3}{8}$ W	$\frac{3}{8}$ F
Torque, lbf in.	44	50	108	144	276	276	285	588
Load, lbf	1538	1753	2662	2951	4588	4956	6813	7364

BA

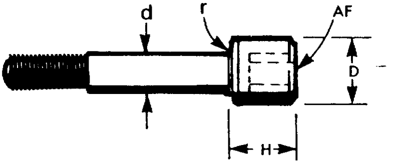
Nom. dia.	0 BA	1 BA	2 BA	3 BA	4 BA	5 BA	6 BA	7 BA	8 BA
Torque, lbf in.	144	80	49	35	24	19	12	$8\frac{1}{2}$	5
Load, lbf	2850	2210	1730	1290	990	780	590	465	360

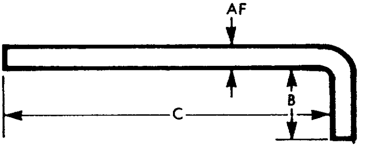
ISO metric

Nom. dia.		M2	M2½	M3	M4	M5	M6	M8	M10	M12
Torque	N.metre	0·6	1·21	2·1	4·6	9·5	16·0	39·0	77·0	135·0
	lbf in.	5·3	10·7	18·6	41	84	142	345	682	1200
Load	K/newton	1·55	2·6	4·0	6·7	11·1	15·6	28·7	45·7	66·7
	lbf	347	5808	890	1500	2480	3480	6400	10 200	14 900

Tightening torques for *plated* threads should be about 75% of those given to induce the same tightening stress.

Dimensions of socket-head capscrews and wrenches





Screws, inch and BA sizes

Thread 'd'	4 BA	3 BA	2 BA	1 BA	0 BA	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
Dia. 'D'	·219	·250	·312	·312	·375	·375	·438	·563	·625	·750
Height 'H'	·142	·161	·187	·209	·236	·250	·312	·375	·438	·500
Hex. AF	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$

Screws, metric sizes

Thread 'd'	M3	M4	M5	M6	M8	M10	M12	M14	M16	M20
Dia. 'D'	5·5	7·0	8·5	10·0	13·0	16·0	18·0	21·0	24·0	30·0
Height 'H'	3·0	4·0	5·0	6·0	8·0	10·0	12·0	14·0	16·0	20·0
Hex. AF	2·5	3·0	4·0	5·0	6·0	8·0	10·0	12·0	14·0	17·0

Wrenches
There is a wide tolerance on dimensions B and C, and a 'long' pattern is listed in the standard which provides 'C' about 50% greater. There are also special lengths designed for particular machines or applications. In general, length 'C' should not be exceeded, to reduce the risk of overtightening the screw.

Wrenches, inch sizes

Hex. AF	·035	·050	$\frac{1}{16}$	$\frac{5}{64}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{7}{8}$
Arm. 'B'	·375	·53	·56	·63	·65	·75	·85	·94	1·03	1·13	1·25	1·38
Arm. 'C'	1·3	1·7	1·7	1·8	2·0	2·25	2·5	2·75	3·0	3·25	3·743	4·25
Grubscrew dia.	8 BA	6 BA	4/5 BA	3 BA	1/2 BA $\frac{3}{16}''$	0 BA $\frac{1}{4}''$	$\frac{5}{16}''$	$\frac{3}{8}''$ $\frac{1}{8}''$ BSP	$\frac{7}{16}''$	$\frac{1}{2}''$ $\frac{1}{4}''$ BSP	$\frac{3}{8}''$ BSP	$\frac{1}{2}''$ BSP
Capscrew dia.	—	—	8 BA	6 BA	4/5 BA	3 BA	1/2 BA $\frac{3}{16}''$	0 BA $\frac{1}{4}''$	$\frac{5}{16}''$	—	$\frac{3}{8}''$ $\frac{7}{16}''$	$\frac{1}{2}''$

Wrenches, metric sizes

Hex. AF	0·71	0·89	1·27	1·5	2·0	2·5	3·0	4·0	5·0	6·0	8·0	12·0
Arm. 'B'	7	10	12	14	16	16	19	22	25	27	32	36
Arm. 'C'	28	30	40	40	45	50	50	64	70	80	100	120
Grubscrew dia.	M1·4 M1·8	M2·0	M2·5	M3	M4	M5	M6	M8	M10	M12	M16	M24
Capscrew dia.	—	—	M1·4	M1·6 M2·0	M2·5	M3	M4	M5	M6	M8	M10	M14

Note that on all capscrews there is a fillet at 'r' amounting to about d/20 rad. The recesses must be countersunk accordingly.

FOUNDRY DATA

Shrinkage of castings

The following allowances should be used when making patterns for sand castings. Special rules engraved with divisions making the correct allowance are available, but care should be taken not to get them mixed up with the workshop rules.

Metal	*Allowance per foot*
Cast iron*	$\frac{1}{16}''$ to $\frac{1}{8}''$ ($\frac{1}{100}-\frac{1}{200}$)
Brass	$\frac{3}{16}''$ ($\frac{1}{60}$)
Bronze, GM	$\frac{5}{32}''$ ($\frac{1}{75}$)
Steel*	$\frac{3}{16}''$ to $\frac{1}{4}''$ ($\frac{1}{60}-\frac{1}{50}$)
Aluminium	$\frac{3}{16}''$ ($\frac{1}{60}$)

*(Castings above $\frac{3}{4}''$ (20 mm) thick use the higher figure.)

Note that if a wooden pattern is made from which an aluminium or brass pattern is to be cast for later use in machine moulding, a *double* contraction must be allowed, one for the pattern casting and a second for the final casting. The shrinkages for the two metals should be added.

Wherever possible it is recommended that the advice of the foundry making the castings be sought, as shrinkages vary considerably with casting methods, metal analysis, and pouring temperatures. The above figures will serve as a guide for 'backyard foundries'.

Taper or 'draw' on patterns

For easy withdrawal from the mould, patterns should be made with a draw or taper of about $\frac{1}{16}''$ per foot ($\frac{1}{200}$) on all surfaces which are vertical as they will stand in the mould.

This will serve for vertical draws of up to 4″ (100 mm) in depth. Above this, use $\frac{3}{32}$″ to $\frac{1}{8}$″/foot ($\frac{1}{125}-\frac{1}{100}$). For draws less than 1″ (25 mm), $\frac{1}{32}$″/foot ($\frac{1}{350}$) will suffice.

Weight of casting from the pattern

A rough guide to casting weight may be had by weighing the pattern and multiplying by the following factors. The effect of cores must be allowed for by separate calculation.

Pattern material	*Cast iron*	*Brass*	*GM*	*Aluminium*
Baywood	8·8	9·9	10·3	3·2
Mahogany	8·5	9·5	10·0	3·1
Pear	10·9	12·2	12·8	3·9
Yellow pine	13·1	14·7	15·4	4·7
Aluminium	2·85	3·2	3·35	1·0

ECCENTRIC TURNING IN THE 3-JAW CHUCK

The size of packing needed to set work eccentric in a self-centring chuck can be predicted with reasonable accuracy, the main source of error being the fact that the jaws usually are ground to a small cylindrical contact face. This error increases with increase in eccentricity, but is small enough to be ignored for normal steam engine eccentrics. There is a risk of the sharp corners of the chuck jaws marking the work, and it is advisable to fit a piece of shim-brass say ·006″ thick all round the work. This thickness must be allowed for in using the expression given below, by adding twice the shim thickness to the diameter of the eccentric. This method is best used by machining the OD of the eccentric sheave first, and using the packing method to bore the hole for the shaft. The boss is then machined by fitting the work to a mandrel in the usual way.

If 'E' is the desired eccentricity (i.e. half valve travel)
'D' is the OD of the sheave, plus twice the shim thickness
'T' is the thickness of the packing to be set under one jaw.
Then:

$$T = 1\cdot 5E\left[1 - \tfrac{1}{2}\frac{E}{D} - \tfrac{3}{8}\left(\frac{E}{D}\right)^3\right]$$

This expression will result in an eccentricity accurate to about one part in 800 with the average 3-jaw chuck jaws, and over the normal range of engine eccentrics, i.e. a $\frac{1}{4}$″ throw eccentric should be within 0·0005″ of the correct figure, certainly within 0·001″.

Care must be taken to see that the eccentric is set flat in the 3-jaw, to ensure that the hole is bored true to the plane of eccentricity. The packing can, of course, be made exactly right by checking with a dial indicator.

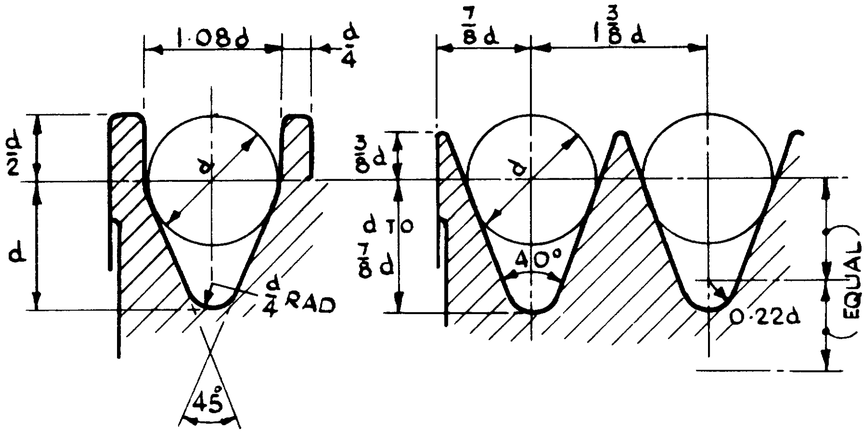
BELT DRIVES

Grooves for round belts

Round ropes or belts of leather, cotton, or plastic from $\frac{1}{8}$″ to $\frac{1}{4}$″ dia. (3–6 mm) should have grooves of about 30° to 35° included angle for light drives. Above $\frac{1}{4}$″ up to $\frac{1}{2}$″ dia. (6–12 mm) in leather the groove may be wider, up to 40°. The total depth of groove should be about $1\frac{1}{4}$ × cord diameter, allowing for a radius at the root of $\frac{1}{4}$ to $\frac{1}{3}$ cord dia. Jockey pulleys should have no vee, and should be machined with a semicircular groove of radius about 0·6 × rope dia.

Model rope drives

These should follow the scaled down full-size groove practice. The drawing below shows both the classical 45° flanged groove and the later 40° flangeless groove, the latter being associated with the firm of William Kenyon & Sons and later becoming general practice. The proportions shown are those for a 1″ rope, and grooves for other sizes may be scaled accordingly. Note that in practice the rope flattens against the walls of the vee when driving.



Grooves for cotton ropes

Vee-belts for machine drives (See sketch on p. **13**.14.)

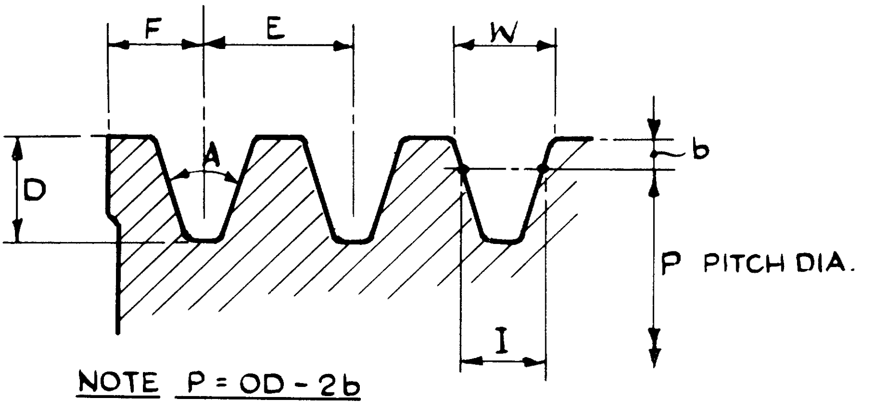
The table shows the dimensions of standard (metric) pulleys, kindly provided by Messrs J. H. Fenner, Ltd. (Note that the dimension 'b', which determines the pitch dia. from the OD may differ from this on some cheap die-cast pulleys made elsewhere.) The true speed ratio depends on the PCD, not the OD of the pulleys.

Belt section	*PCD P mm*	*A deg.*	*Top width W mm*	*D mm*	*E mm*	*F mm*	*b mm*	*Pitch width l mm*
Z (M)	50–90 91 up	34 38	10·18 10·39	9·75	12	8	2·75	8·5
A	70–125 126 up	34 38	13·29 13·58	12·5	15	10	3·75	11·0
B	125–199 200 up	34 38	16·75 17·1	15·5	19	12·5	4·5	14

Correct tension for vee-belts

The force required to cause a deflection of 16 mm at mid span of the belt should be:

For 'Z' size belts, 0·5 to 0·8 kg.
For 'A' size belts, 1·0 to 1·5 kg.
For 'B' size belts, 2·0 to 3·1 kg.



Grooves for vee belts (See previous page.)

HEATING VALUES OF AVERAGE FUELS

The heating value of fuels varies according to source and, in the case of petroleum products, blend and analysis. The following values may be taken as a guide, but for accurate figures the fuel suppliers should be consulted.

Solid fuel

			Approx. theoretical air requirement	
Fuel	*BTU/lb*	*kWh/kg**	*Lb/lb fuel*	*Cu. m/kg fuel*
Dry wood	6 800	4·39	5·1	3·9
Soft coal	11 100	7·16	8·7	6·7
'Steam coal'	14 500	9·35	11·3	8·7
Anthracite	14 800	9·55	11·4	8·8
Coke	13 000	8·39	10·2	7·9

Liquid fuel

Fuel	*BTU/lb*	*kWh/kg*	*Lb air/ lb fuel*	*M^3 air/† kg fuel*	*Sp. gr.*	*Energy* ft lb/cu. in.*	*Energy* N.m/litre*
Petrol	20 340	13·12	14·9	11·5	0·72	48·6	1·069
Kerosene	20 100	12·96	14·6	11·3	0·79	48·9	1·076
Diesel oil	19 765	12·75	14·4	11·09	0·87	49·0	1·078
Alcohol (meths)	10 200	6·58	8·0	6·16	0·82	47·0	1·034
50% ether/petrol	16 830	10·86	13·0	10·0	0·727	49·0	1·078

*These figures give the energy release per cubic inch or litre of cylinder volume filled with a 'correct' air/fuel mixture. The remarkable similarity between different fuels will be noted.

†At 15°C and atmos. pressure.

Gaseous fuel

Fuel	*BTU/ft^3*	*kWh/m^3**	*Air reqd vol/vol*	*Vol. gas/mass liquid* *Ft^3/lb*	*Litre/kg*	*Boiling point, °F*	*(°C)*
Methane CH_4	995	9·88	9·52	—	—	—	—
Calor butane C_4H_{10}	3200	32·96	30·0	6·51	82·6	31	(−0·5)
Calor propane C_3H_8	2500	25·75	23	8·6	109	−43·7	(−42)

*1 kWh = 3600 kJ.

If the specific gravity of a petroleum based liquid fuel is known, an approximation to the calorific or heating value is given by:

$$H = 22\,320 - 3780d^2 \text{ BTU/lb } (\times 2\cdot326 \text{ for kJ/kg}).$$

Where d is the specific gravity at 60°F. This does *not* apply to alcohol fuels, or mixtures containing alcohols.

MODEL RAILWAY WHEEL AND GAUGE STANDARDS

Track and wheel standards have evolved over many years, and those published by Martin Evans (the then Editor of the *Model Engineer*) in 1960 are still in common use today. However, the last few years have seen a number of important developments, not least the introduction of ground-level track carrying turnouts and cross-overs. Further, multiple gauge track, originally limited to $2\frac{1}{2}/3\frac{1}{2}$ or $3\frac{1}{2}/5$ inch, has now developed into triple – $3\frac{1}{2}/5/7\frac{1}{4}$ inch gauges. This has led to difficulties, especially when stock having fine (i.e. 'true') scale wheels is used. This matter is discussed in some detail in the *Model Engineer*, vol 90, pp. 99, 113, 228 et seq.

In addition, the increasing use of club tracks for public passenger hauling (whether for fun or charitable fund-raising) has emphasised the 'duty of care' laid on both track and locomotive operators, the more so as even a bruised toe-nail can now lead to claims for compensation!

Finally, the never ending pursuit of perfection, which is the hallmark of model engineering in all its branches, has had its effect.

The result is that there is (at the time this is written) an active debate going on, both about the wheel profile standards to be adopted and about the present practice as to check-rails, curve radii and loading gauges. This will take time to resolve.

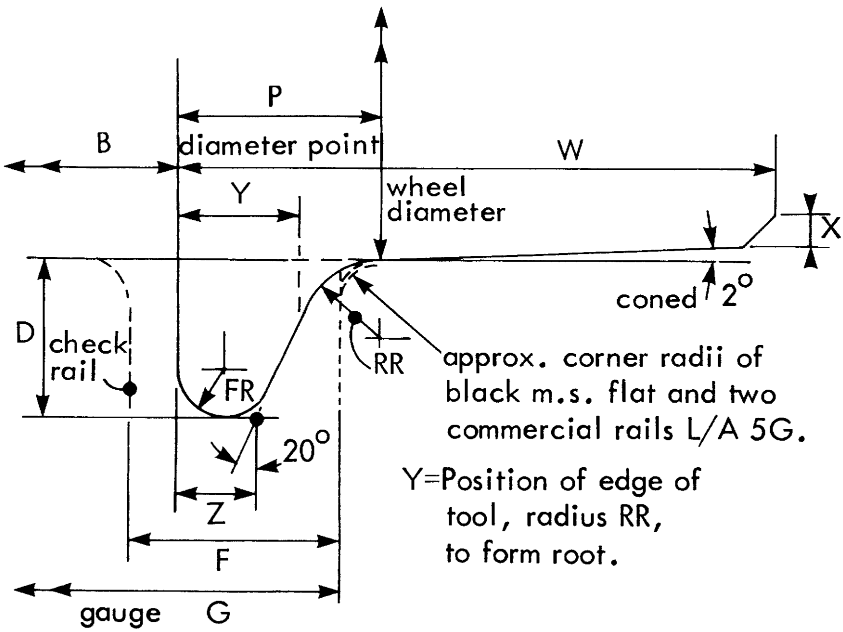
The data given below *have* proved satisfactory for elevated trackwork, and is repeated from previous editions with but one change – the note on 'minimum radius of curve'. However, those concerned with laying down new multiple gauge track with points or crossings are urged to consult the document TN3 issued by the Health & Safety Executive (*Guidelines on Miniature Railways*) and also to seek advice from one or other of: The Southern Federation; the Northern Association of Model Engineering Societies; or the $7\frac{1}{4}$ inch Gauge Society.

(a) Track standards

Gauge	*Scale to 1 foot*	*Buffer height from rail*	*Weight of an average express loco.*	*Average train weight*	*Average speed†*	*Loading gauge*	*Minimum radius of curves**	*Recommended radius for curves*
$1\frac{1}{4}$ in.	7 mm	24·2 mm	7 lb	30 lb	2 mph	$63 \times 94\frac{1}{2}$ mm	4 ft	6 ft
$1\frac{3}{4}$ in.	10 mm	34·6 mm	20 lb	80 lb	$3\frac{1}{2}$ mph	90×135 mm	5 ft	10 ft
$2\frac{1}{2}$ in.	$\frac{17}{32}$ in.	$1\frac{27}{32}$ in.	50 lb	2 cwt	5 mph	$4\frac{3}{4} \times 7\frac{1}{8}$ in.	8 ft	16 ft
$3\frac{1}{2}$ in.	$\frac{3}{4}$ in.	$2\frac{19}{32}$ in.	100 lb	9 cwt	7 mph	$6\frac{3}{4} \times 10\frac{1}{8}$ in.	14 ft	30 ft
5 in.	$1\frac{1}{16}$ in.	$3\frac{21}{32}$ in.	$2\frac{1}{2}$ cwt	18 cwt	10 mph	$9\frac{9}{16} \times 14\frac{3}{8}$ in.	18 ft	40 ft
$7\frac{1}{4}$ in.	$1\frac{1}{2}$ in.	$5\frac{3}{16}$ in.	6 cwt	$2\frac{1}{2}$ tons	12 mph	$13\frac{1}{2} \times 20\frac{1}{4}$ in.	30 ft	60 ft
$9\frac{1}{2}$ in.	2 in.	$6\frac{7}{8}$ in.	10 cwt	4 tons	15 mph	18×27 in.	40 ft	100 ft
$10\frac{1}{4}$ in.	$2\frac{1}{4}$ in.	$7\frac{3}{4}$ in.	17 cwt	5 tons	17 mph	$20\frac{1}{4} \times 30\frac{1}{4}$ in.	50 ft	150 ft

*Should not be less than 20 × longest rigid wheelbase.
†It is recommended that trains run on tracks in public parks should not exceed 6 to 8 mph.

(b) Wheel Standards



(b) Wheel standards

Gauge	*Back to back*	*Tyre width*	*Flange depth*	*Root radius*	*Flange radius*	*Chamfer*	*Tread diameter point*	*M/Cg dimension*	*M/C^g dimension*	*Flange way*
G	*B*	*W*	*D*	*RR*	*FR*	*X*	*P*	*Y*	*Z*	*F*
2½″	2·281	0·268	0·085	0·035	0·020	0·015	0·090	0·055	0·034	0·093
63·5 mm	58	6·8	2·2	0·9	0·50	0·40	2·3	1·4	0·90	2·3
3½″	3·281	0·375	0·110	0·050	0·030	0·020	0·126	0·076	0·051	0·130
89 mm	83	9·5	2·8	1·3	0·75	0·50	3·2	1·9	1·3	3·3
5″	4·687	0·535	0·140	0·070	0·045	0·030	0·176	0·106	0·077	0·190
127 mm	119	13·6	3·6	1·8	1·2	0·80	4·5	2·7	1·95	4·8
7¼″	6·800	0·776	0·203	0·100	0·065	0·040	0·254	0·154	0·110	0·270
184 mm	172	19·7	5·2	2·5	1·7	1·00	6·4	3·9	2·8	6·9
9½″	8·900	1·017	0·266	0·133	0·086	0·050	0·336	0·203	0·146	0·350
241 mm	225	25·8	6·8	3·4	2·2	1·30	8·5	5·15	3·7	8·9
10¼″	9·600	1·097	0·287	0·144	0·093	0·060	0·363	0·219	0·158	0·380
260 mm	244	27·8	8·0	3·7	2·4	1·50	9·2	5·55	4·0	9·7

ORSAT FLUE GAS ANALYSIS APPARATUS

The solutions required for this apparatus are made as follows:

To absorb CO_2.	One part of potassium hydroxide by weight dissolved in $2\frac{1}{2}$ parts by weight of water.
To absorb oxygen.	Dissolve one part by weight of pyrogallic acid in 2 parts of hot water. To this add three parts of the same potassium hydroxide solution as used for CO_2.
To absorb CO.	Dissolve one part by weight of cuprous chloride in seven parts of hydrochloric acid. Add 2 parts of clean copper chippings, and allow to stand for 24 hours. Then add 3 parts of water.

Note: Distilled water should be used for all solutions.

COEFFICIENTS OF THERMAL EXPANSION AND MELTING POINTS

Coefficients vary with detail differences in composition, and are also affected by the temperature range. The following are for average materials between 15 and 250°C, and will serve for most model engineering purposes.

Figures give the expansion in thousandths of an inch per inch, or microns per millimetre, per degree rise.

Material	*Per °F*	*Per °C*	*Melting, °C*
Aluminium	·0128	·0255	658
Brass	·0104	·0189	800–1000
Bronze	·0098	·0177	900–1050
Copper	·0093	·0167	1083
Duralumin	·0125	·0226	650
German silver	·0102	·0184	1030–1150
Glass (average)	·0048	·0086	500–1000
Gunmetal	·0101	·0181	995
Grey cast iron	·0059	·0102	1140–1200
Hard tool steel	·0078	·0140	—
Invar steel	·0002	·00036	—
Mild steel	·0065	·0110	1250–1500
Nickel	·0071	·0128	1452
Phosphor bronze	·0093	·0168	1000
Stainless steel	·0059	·0107	—
Wood (oak) transverse	·0030	·0054	—
Wood (oak) longitudinal	·0027	·0049	—
Zinc	·0162	·0292	420

SIZE OF WOODSCREWS

Gauge no.	*Dia. in.*	*Pilot drill mm*	*Slot width in.**	*Gauge no.*	*Dia. in.*	*Pilot drill mm*	*Slot width in.**
2/0	0·060		0·015	7	0·150	2·3	0·043
0	0·063		0·018	8	0·164	2·4	0·050
1	0·070		0·020	9	0·178	2·6	0·050
2	0·082		0·023	10	0·192	2·8	0·055
3	0·094	1·5	0·030	12	0·220	3·2	0·060
4	0·108	1·6	0·034	14	0·248	3·5	0·070
5	0·122	1·8	0·039	16	0·276	3·8	0·070
6	0·136	2·1	0·043	18	0·304	4·2	0·076

*There is a tolerance on the slot width; figures given are the mean of max and min metal.

The diameter of a woodscrew should not exceed $\frac{1}{10}$ of the width of the wood accepting the screwed portion. The depth of the pilot hole should be less than the penetration of the screw by about $1\frac{1}{2}$ screw diameters. There should ideally be between 4 and 6 screw diameters of thread engagement as a minimum. Shanks should have a clearance hole. Pilot holes may be 0·5 mm smaller for softwood – no pilot is necessary in small diameters.

When using brass screws in hard wood, 'tap' the hole first part way using a steel screw.

For those expecting to drive many screws with power tools a range of hardened screws is available. These have a double-start thread and a self-centring point. Diameters range from 2·4 mm (no. 3) up to 5·6 mm (no. 12) in lengths up to 4″ – 100 mm, depending on diameter.

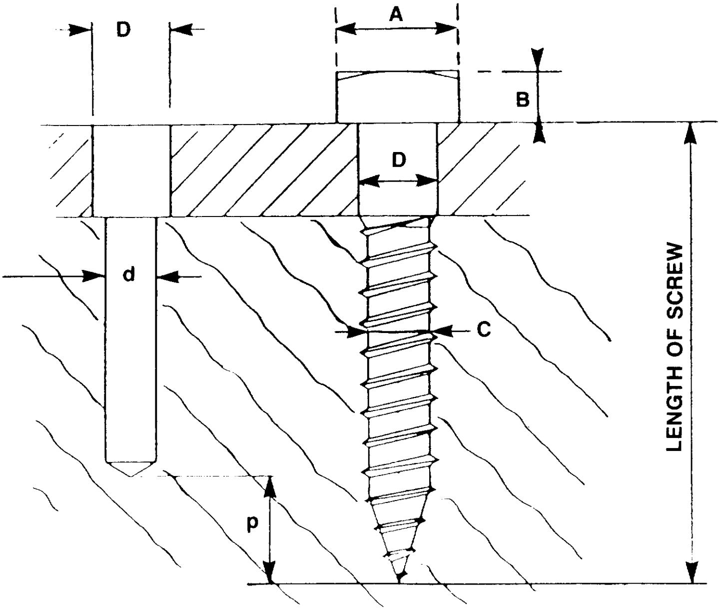
COACH SCREWS

These are known as 'spanner screws' in some areas. Pilot holes are essential, as are slight clearance holes to the length of plain shank. The across flats dimension corresponds to the BSF hexagon sizes shown on page **4**.12. The material is mild steel, UTS about 28 T/in^2 – 430 N/mm^2.

Dimensions of coach screws (*See p.* **13**.*20.*)

Size in.	*Pilot drill* Hardwood	*Pilot drill* Softwood	*Drive, P mm*	*A in.*	*B in.*	*C in.*	*D in.*
$\frac{1}{4}$	4·5 mm	3·8 mm	9	0·437	0·172	0·170	$\frac{1}{4}$
$\frac{5}{16}$	5·8	5·0	10	0·520	0·212	0·220	$\frac{5}{16}$
$\frac{3}{8}$	7·7	6·6	14	0·592	0·254	0·290	$\frac{3}{8}$
$\frac{1}{2}$	9·3	8·0	18	0·810	0·360	0·350	$\frac{1}{2}$

The use of grease on the threads is almost essential, especially with hardwoods.

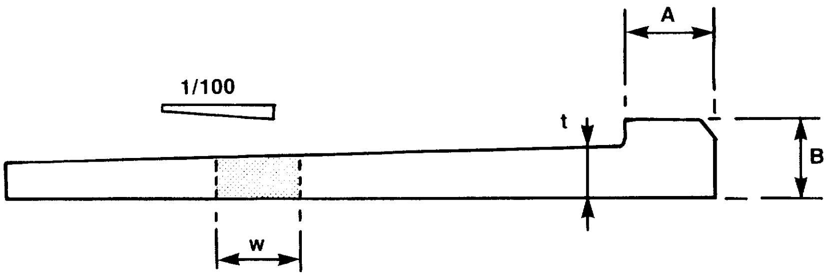


(*See p.* **13**.*19*)

KEYS AND KEYWAYS

Taper keys

The appearance of models is greatly enhanced if proper key-heads are seen on shafts, even if a grub-screw is used on the wheel. The following proportions for *full-size* keys are in line with BS 46. (Metric keys to BS 4235 have the same proportions, but shaft sizes are at much closer intervals.)



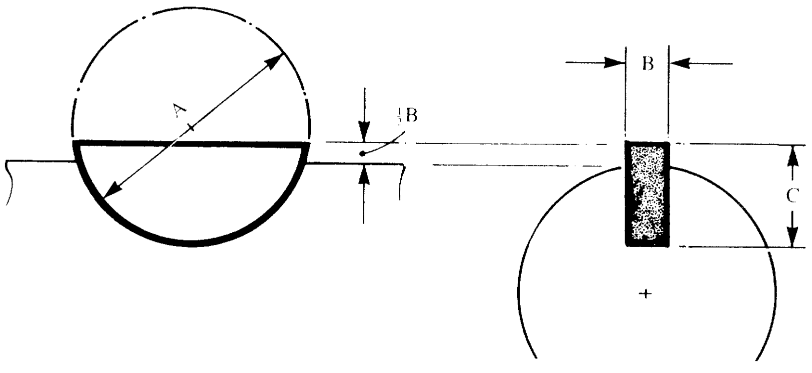
Shaft dia., in.	$\frac{1}{4}$–$\frac{1}{2}$	$\frac{1}{2}$–$\frac{3}{4}$	$\frac{3}{4}$–1	1–1$\frac{1}{4}$	1$\frac{1}{4}$–1$\frac{1}{2}$	1$\frac{1}{2}$–2	2–2$\frac{1}{2}$
w × t, in.	$\frac{1}{8}$ sq.	$\frac{3}{16}$ sq.	$\frac{1}{4}$ sq.	$\frac{5}{16} \times \frac{1}{4}$	$\frac{3}{8} \times \frac{1}{4}$	$\frac{1}{2} \times \frac{5}{16}$	$\frac{5}{8} \times \frac{7}{16}$
Shaft dia., in.	2$\frac{1}{2}$–3	3$\frac{1}{2}$	3$\frac{1}{2}$–4	4–5	5–6	8	10
w × t, in.	$\frac{3}{4} \times \frac{1}{2}$	$\frac{7}{8} \times \frac{5}{8}$	$1 \times \frac{3}{4}$	$1\frac{1}{4} \times \frac{7}{8}$	$1\frac{1}{2} \times 1$	$2 \times 1\frac{3}{8}$	$2\frac{1}{2} \times 1\frac{5}{8}$

Where two keys are called for (at 120°) the above sizes are used, but in some cases they may be reduced to $\frac{3}{4}$. The length of the key is made to carry the shear stress due to the torque.

The nominal thickness 't' is that just in front of the head. The taper on the thickness is $\frac{1}{100}$. BS 56 lays down dimensions for the head, but these approximate to **A** = **1.6t** and **B** = **1.8t** when scaling down for model purposes.

Feather, or sliding, keys are parallel and of square section, side = w above.

Woodruff keys and cutters



Key no.	203	303	403	204	304	404	305	405	505
A, in.	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$
B, in.	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$
C, inc.	·171	·171	·171	·203	·203	·203	·250	·250	·250

Key no.	406	506	606	507	607	807	608	808
A, in.	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1
B, in.	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{4}$
C, inc.	·313	·313	·313	·375	·375	·375	·438	·438

The cutter number is the same as the key number. The last two digits give the nominal diameter 'A' in eighths of an inch, and the preceding digits the nominal width 'B' in thirty seconds.

The key should be a tight fit to the *shaft* and a close slide fit to the *hub* or *boss*. There should be a clearance of from 0·007″ to 0·010″ between the flat top of the key and the root of the slot in the hub. See BS 46/1958 for details of tolerances on keys and cutters.

THERMOCOUPLES

The EMF (voltage) developed by a thermocouple depends on the difference in temperature between that of the *hot* junction between the two wires, and that of the *cold* junction. If the hot couple is linked to the temperature indicator using the thermocouple wires, then the cold junction is at the indicator terminals; if however, copper wire is used for this connection, then the cold junction is at the joint between couple wires and the copper. The true temperature will be that shown on the indicator *plus* that at the cold junction.

The wires forming the hot couple must be insulated from each other except at the actual joint, and small ceramic beads can be obtained for this purpose. Ideally the hot junction should be fusion welded, but tolerable results can be obtained for occasional use by thoroughly cleaning the ends of the wire, twisting together very tightly, heating to bright red and hammering the joint. Copper-Constantan joints may be brazed after twisting, as they are not used for temperatures above 500°C.

The EMF in *millivolts* developed by couples is given in the table; together with suitable standards for calibration.

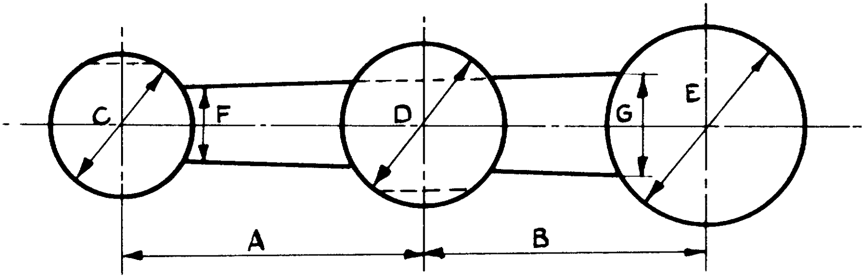
T = Copper-Constantan to BS 1828 (cheapest, but limited range)
J = Iron-Constantan to BS 1829
K = NiChrome-NiAluminium BS 1827

T °*C*	*T*	*J*	*K*	*Calibration points*
0	0	0	0	
50	2·02	2·58	2·02	
100	4·24	5·27	4·10	100° boiling water (vapour)
150	6·63	8·01	6·13	
200	9·18	10·78	8·13	218° m.p. of pure tin
250	11·86	13·56	10·16	
300	14·67	16·33	12·21	
350	17·58	19·09	14·29	327° m.p. of pure lead*
400	20·59	21·85	16·40	
500	26·10	27·39	20·65	
600	—	33·11	24·91	649° m.p. of pure aluminium*
700	—	39·15	29·14	
800	—	45·53	33·30	801° m.p. pure salt (NaCl)
900	—	51·20	37·4	
1000	—	—	41·3	1083° m.p. of pure copper
1200	—	—	48·9	
1400	—	—	55·9	

*Reducing atmosphere advisable, to avoid effects of metal oxides.

Mercury-in-glass thermometers are available calibrated up to 400°C (750°F). These can be used in tempering salt or oil baths, or (in a suitable pressure pocket) for steam temperature measurement. The latter is often a better guide to pressure than the average pressure gauge.

PROPORTIONS OF TRADITIONAL BALL HANDLES

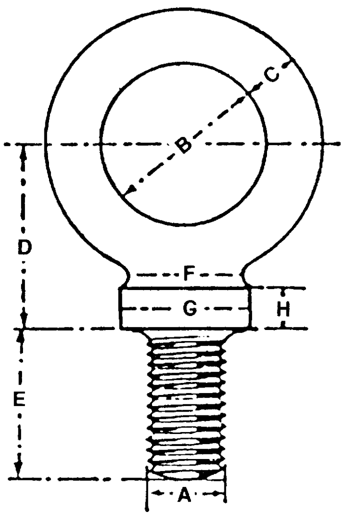


A *in.*	*B* *in.*	*C* *in.*	*D* *in.*	*E* *in.*	*F* *in.*	*G* *in.*
1	$\frac{7}{8}$	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{9}{16}$
$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$\frac{7}{16}$	$\frac{5}{8}$
$2\frac{1}{8}$	2	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{11}{16}$
$2\frac{7}{8}$	$2\frac{5}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{16}$
$3\frac{3}{4}$	$3\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{13}{16}$
$4\frac{3}{8}$	$4\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$\frac{11}{16}$	$\frac{15}{16}$

Metric handles should follow the same relative proportions.

STANDARD LIFTING EYEBOLTS

(*Dimensions on next page*)



The weight of the prototype should be estimated, and the model lifting eye proportioned to scale from the table. The 'safe load' lifting capacity assumes that the thread is screwed fully home.

Safe load		*Dimensions in inches*							
Lbf	*kN*	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*
150	0·75	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{4}$
250	1·1	$\frac{5}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{4}$
400	1·7	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{4}$
800	3·7	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	1	$\frac{3}{4}$	1	$\frac{5}{16}$
1300	6	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$
2000	9	$\frac{3}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{3}{8}$	$\frac{7}{16}$
2700	12	$\frac{7}{8}$	$1\frac{3}{4}$	$\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$
3700	16	1	2	$\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$\frac{1}{2}$
4800	21	$1\frac{1}{8}$	$2\frac{3}{8}$	1	3	2	$1\frac{5}{8}$	2	$\frac{5}{8}$
6000	27	$1\frac{1}{4}$	$2\frac{3}{8}$	1	3	2	$1\frac{5}{8}$	2	$\frac{5}{8}$
9000	40	$1\frac{1}{2}$	3	$1\frac{1}{8}$	$3\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{13}{16}$	$2\frac{1}{4}$	$\frac{11}{16}$

(Though there are metric specifications for *hooks* within this range, it appears that, at time of writing, none for eyebolts has yet been written.)

Test load on link chains for hoisting

Test load in tons = 12 d^2, where d = dia. of metal in link, inches.

Colours on old drawings

19th century practice was to indicate metals by colours, especially on arrangement drawings. The following were used.

Cast iron .... Paynes grey
Wrought iron .... Prussian blue
Steel .... Purple
Brass .... Gamboge with a little red added
Lead .... Dark indigo
Copper .... Crimson lake with gamboge added
Brick .... Crimson lake with burnt sienna
Stone .... Light sepia + blue
Wood .... Pale tint of sienna

Musical notes and speed

Speed can often be estimated by reference to a musical note. The frequency of notes about 'middle C' on the piano are:

Note	A	B	C	D	E	F	G
Vib/sec.	220	246·9	261·6	293·6	329·6	349·2	391·9

Each octave above the 'middle' range gives notes of twice the frequency below, and each octave below gives notes of half the frequency. Thus the frequency of the note 'C' in the various octaves is: 32·7, 65·4, 130·8, **261·6 (middle C)**, 523·2, 1046·4, 2092·8, 4185·6 vibrations per second.

Human horsepower

A man in good health can work for 8 hours/day at about the following hp.

Pushing or pulling horizontally	0·097
Winding a crank	0·081
Hoist 40 lb wt with single pulley	0·054
Ditto, by direct hand lifting	0·045
Digging a trench up to 5 ft deep	0·014
Treadling a machine	0·09

The pendulum

The periodic time of an ideal pendulum (one having a point mass suspended on a weightless cord) is given by $t = 2\pi\sqrt{L/g}$, where t is in seconds, L is the effective length and g the local value of 'gravity'. Both L and g must be in the same units.

In a real pendulum major adjustment is made by altering L, and fine adjustment by adding small weights to the bob.

Cleaning and restoring woodwork

The following will both clean polished or varnished wood and also 'feed' or revive the timber. The surface should be cleaned with normal methods first, to economise the solution.

Mix 4 parts raw linseed oil
4 parts pure turpentine oil (*not* 'white spirit')
4 parts vinegar
1 part methylated spirits.

Shake up well. Apply liberally with a soft cloth, wipe off whilst still damp. Apply a second, or several, further coats allowing each to dry before rubbing over with a dry cloth – the last coat being left for several hours. 3 or 4 coats will usually suffice for 'normally dirty' woodwork. An annual application will keep tool cabinets or furniture in excellent condition. (*Note:* This will not remove burns or stains in the wood itself.)

Lacquer for use on steel etc., after bluing

Mix shellac, 1 ounce, Dragon's blood (a red dye) ¼ ounce and dissolve in 2 pints water. Shake vigorously and filter through fine blotting-paper. Apply evenly or dip; rapid drying, but leave some hours before handling.

Painting new galvanised sheet

The surface of new or little weathered galvanised sheet must be 'killed' before painting. Apply a solution of phosphoric rust-killer let down with 6 to 10 parts of water, followed by a water-rinse. Paint when dry.

To prevent scaling on brass and bronze

Make a saturated solution of Borax in methylated spirit. Dip the parts in this and allow to dry before heating. This is reasonably effective up to 700°C or so. It can be removed with hot water after the work has cooled.

PIPE FLANGES

Nothing destroys the sense of authenticity of a model more than the use of coned unions in places where full-size practice is to use flanges. Unions are used only on small lubricating oil pipes, boiler fittings and cylinder drains, and on small, low pressure fuel lines. Coned unions *are* found on low pressure water pipes up to about 1½ inch diameter, but these are of quite different appearance from the 'motor-car' type commonly employed by model engineers.

The number of bolts is always a multiple of *four*, so that valves etc. may be turned through 90° when necessary. Six-bolt flanges are non-standard and very rare.

The table below gives the *full-size* dimensions of both cast iron and screwed-on or welded steel flanges for steam pressure up to 250 lbf/sq. in. The difference between these and those limited to a maximum of 100 lbf/sq. in. is too small to be noticed when scaled down to model size. All dimensions in inches.

Bore	*1*	*2*	*3*	*4*	*5*	*6*	*8*	*10*	*12*	*14*	*16*
Flange OD	$4\frac{3}{4}$	$6\frac{1}{2}$	8	9	11	12	$14\frac{1}{2}$	17	$19\frac{1}{4}$	$21\frac{3}{4}$	24
PCD	$3\frac{7}{16}$	5	$6\frac{1}{2}$	$7\frac{1}{2}$	$9\frac{1}{4}$	$10\frac{1}{4}$	$12\frac{3}{4}$	15	$17\frac{1}{4}$	$19\frac{1}{2}$	$21\frac{3}{4}$
No. of bolts	4	4	8	8	8	12	12	12	16	16	20
Bolt dia.	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1
Flange thickness											
C.I.*	$\frac{9}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
M.S.	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{8}$

**Note:* Cast-iron may be used only up to 150 lbf/sq. in.

Fluorescent fluid for spirit levels etc.

This is a 0·2% solution of *fluorescine* (obtainable to order from chemists) in 70/30 alcohol/water mixture. Dissolve the fluorescine in the water first and then add the alcohol. (Industrial alcohol, not 'meths'.)

Silica gel

These moisture absorbing crystals become inactive over time. The material may be re-activated by heating to 110°C–120°C for some hours. (Up to 160°C will do no harm.)

Crystals can be obtained which change colour when they are exhausted, but revert to the original when re-activated.

SIZE OF CLOCK KEYS

Small square socket-keys are sometimes required. These can be made from clock keys, which are surprisingly cheap, in both steel and brass. Sizes are as under; the 'size' is the size of the square arbor, in mm, to fit the key.

No.	*Size*	*No.*	*Size*	*No.*	*Size*
000	1·75	5	3·5	12	5·25
00	2·00	6	3·75	13	5·50
0	2·25	7	4·00	14	5·75
1	2·50	8	4·25	15	6·00
2	2·75	9	4·50	16	6·25
3	3·00	10	4·75	17	6·50
4	3·25	11	5·00	18	6·75
				19	7·00

These are modern keys. Differences of up to ±0·3 mm have been found on 19th century examples.

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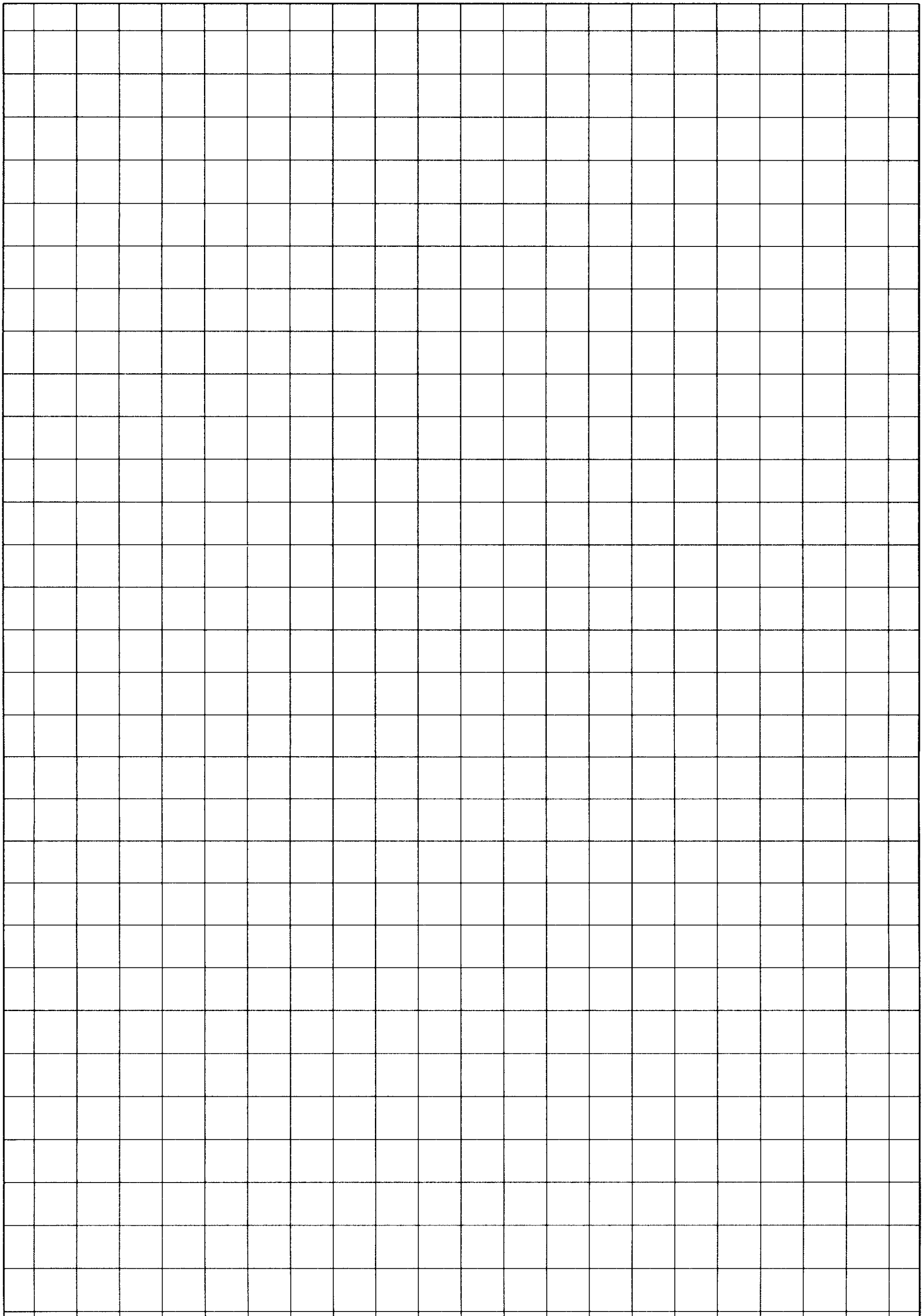
Entries in **bold** type indicate section headings. *Italic* – other headings



All professional engineers have a little book in which they jot down those notes of facts, figures and formulae which they feel that they may need in the future. It is more convenient to look up one's own record than to wade through a lot of associated, but irrelevant, material, especially as this may involve a trip to a reference library. Over a period of years, the contents of such notebooks grow to cover a wealth of vital information and the time saved can be considerable.

During his professional career, Tubal Cain filled three such books and, as a lifelong model engineer, he is in a unique position to select (and add to) the material most useful to the amateur engineer.

The Model Engineer's Handbook comprises a compilation of those tables, facts, procedures and data which the author himself found valuable in his model engineering activities, and it provides a veritable mine of information to which the reader will return again and again. The book also contains helpful explanations of the 'whys and hows' of using many of the entries. Not the least of its attributes is the use, where appropriate, of data and calculations in both imperial and SI units so that all generations of model engineers can feel at home.

In this, the third, edition all existing data have been brought up to date or rearranged for greater clarity. Over 60 pages of new matter have been added and, in addition, space has been provided for readers' own notes and sketches. These improvements add materially to the value of a volume that has already been accepted as being a unique reference book. It should find a place in the library of all model makers – beginner and expert alike.

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