

# Morgan carbon bearings

## Application data

### 1. Introduction

Carbon Bearings have been developed by Morganite Carbon Limited over the last 30 years. The generic term "Carbon Bearings" includes all possible variations of the basic material "Carbon". Therefore it covers "carbon-graphite", "metal-graphite" and "electro graphite". Carbon is self lubricating and has a low coefficient of friction at high and low temperatures. Its use as a bearing material is extensive when a conventional lubricant is not possible and also where the performance of other lubrication is poor. There are other properties which give it unique and distinctive advantages for bearings. These include:

- Carbon has a low wear rate.
- Is mechanically strong in compression.
- Has a high strength-weight ratio.
- Has high strength at elevated temperatures.
- Has a low modulus of elasticity.
- Is electrically conductive.
- Has a relatively low density.
- Has good hydrodynamic bearing properties.
- Is non oxidising to 500 °C (dependant on grade).
- Has high thermal conductivity.
- Has high resistance to thermal shock.
- Is chemically inert.
- Does not swell in water.
- Does not weld.
- Is easily machined to close tolerances.
- Is dimensionally stable.

This publication indicates how these properties can be used to advantage for bearings, and details some successful applications. In addition, our Technical Applications Department is available to advise on material selection and bearing design. To provide them with the relevant facts on which to make a preliminary assessment of your requirements, just complete our [Technical Questionnaire](#).

When a design has been resolved, the bearings can be manufactured by our skilled machinists. Alternatively special carbon blanks of a suitable shape and size can be supplied for finishing by you or your local carbon machinists.

### 2. Field of use

Carbon is used where oil or grease lubricated bearings could be used only with difficulty, if at all. Carbon can run dry or in fluids which are corrosive or unsuitable as lubricants with other bearing material.

The three principal fields are thus:

1. Where contamination is undesirable, as in textile and food machinery, i.e., by avoiding the use of oil or grease.
2. In furnace and boiler equipment where temperatures are too high for the use of conventional lubricants. Where the bearings are immersed in liquids such as hot and cold water, sea water, acidic or alkaline solutions, or in oil solvents such as petrol and benzene.

### 3. Permissible loads, speeds and temperatures

Carbon bearings run satisfactorily under conditions which are so varied that it is impracticable to give maximum load and speed figures which would be universally applicable. An approximate guide is the value of the loading characteristic, i.e., the PV factor which experience has shown to be satisfactory with the grades and operating conditions mentioned below.

$$PV = \frac{F}{A} * v \quad \text{where}$$

F = force [N]

A = projected area of bush [mm<sup>2</sup>]

v = shaft peripheral speed [m/s]

#### 3.1 Dry operation

Usually in dry operation the maximum permissible load depends on the wear rate that can be allowed. The maximum permissible speed is determined by local heating at the rubbing surface; if surface spot temperature becomes extremely high the carbon may deteriorate too rapidly. Carbon grades employed as dry bearings are chiefly carbon-graphite grades and metal-carbon grades. In some special instances electro-graphite may be used. ([Physical data of grades](#))

##### 3.1.1 Continuous Operation

###### 3.1.1.1 Carbon and Carbon-graphite

To give a long life with continuous operation i.e. steady load over long periods the product PV should not exceed 0,11 [N/mm<sup>2</sup>\*m/s]. P should not exceed, say 1,4 N/mm<sup>2</sup> and V, say 1,25 m/s. Loads above 1,4 N/mm<sup>2</sup> need careful consideration and may not always permit the maximum value of PV. In conditions where *short* period life (say 3 to 6 months) is economically permissible, a PV of 0,178 can be sustained or even exceeded.

###### 3.1.1.2 Metal-carbon

The strength of metal-carbon grades permits much higher maximum loads than for carbon or carbon-graphite grades but the limit for speed remains unchanged. The load for metal-carbon grades can reach 2,8 N/mm<sup>2</sup> provided the speed is low.

For long period running maximum PV 0,145.

For short period running maximum PV 0,216.

Too rigid use of these figures should be avoided because the pressure calculated on the projected bearing area is an arbitrary factor in the case of dry running. There is no fluid film to distribute the load. Initially, only line contact is made, and especially in large, heavy loaded bearings, reduced speed during the "bedding-in" period is recommended if possible.

##### 3.1.2 Intermittent Running

Considerably greater values for PV can be safely used for dry running when the operation is intermittent and especially when it is of short duration with long stationary periods. This applies to all carbon grades.

### 3.1.3 Working Temperature

This is the temperature caused by heat conducted to the bearings plus heat developed in the bearing by friction.

Grade	Maximum Operating Temperature °C
carbon - graphite	350
metal-carbon	350
metal-carbon	130
metal-carbon	350
electro - graphite	450 to 500 at reduced loads
electro- and carbon - graphite	500 to 1000 in reducing atmosphere

### 3.2 Wet Operation (Water, petrol, chemical liquids, etc)

The fluid reduces the friction and mechanical wear. It also helps to dissipate the friction heat. Consequently, much higher loads and speeds are possible than with dry running. If the fluid in which the bearings are running effects perfect lubrication, values of PV 1000 times those for dry running are quite practicable, which are comparable with the loading of ordinary oil-lubricated bearings. In fluids such as petrol, carbon tetrachloride, paraffin, etc., there are successful applications at over 2,1 N/mm<sup>2</sup> with speeds exceeding 1,25 m/s.

Operation in liquids usually requires special consideration, and under ideal conditions of full fluid film lubrication, PV values in the order of 36 are possible.

Metal-carbon grades are advisable for full fluid film lubricated conditions. Their more homogeneous texture supports a continuous lubricant film. Should the film momentarily rupture they are mechanically and thermal more resistant and do not damage the journal surface. Generally, lubricated carbon bearings follow normal bearing design practice, except that they can utilize much more mobile fluids as lubricants than can any metal bearings. This is because they can approach much more closely to the "boundary lubricated" condition without risk of seizure. For low loads and speeds the plain carbon or carbon-graphite grades are adequate, but for high loads and speeds the metal-carbon grades are much superior.

#### 3.2.1 Electrolytic Action

Carbon alone is inert even to potent chemical liquids. However, when immersed together with metals in an electrolyte, precautions must be taken to ensure that the combination is not excessively electrolytically active. It is sometimes necessary to make a special choice of shaft and housing metal or to insulate the bearing from its housing. The suitability of a proposed combination of materials cannot usually be forecast and can only be resolved by experiment.

Carbon bearings are successful with many different liquids but because of the great variety of possible combinations of material and operating conditions, it is not practicable to give a reliable list. We are always prepared to supply sample pieces of carbon for a test. One interesting and important example is the choice by this test of a carbon-graphite grade impregnated with Babbitt alloy, for use in

sea or salt water in conjunction with phosphor bronze housings and shafts, a combination which has proved entirely satisfactory in practice.

## 4. Coefficient of friction

The coefficient of friction, i.e., the ratio of the force required to overcome friction to the pressure normal to the sliding surface, is lower for carbon-graphite sliding dry on metallic surfaces than for most combinations of metals working under the same conditions.

### 4.1 Static Friction

This depends on the mating materials, the hardness and condition of the surfaces, and on other factors so that it is impracticable to give precise figures without individual experiment. Under some operating conditions it might be appreciably higher for a metal-carbon grade than for an electrographitic grade.

Generally, the static friction of carbon on metals is several times the dynamic friction, but appreciably less than the static friction of metal on metal. For some of the electrographitic grades the coefficient of static friction approaches that of dynamic friction.

### 4.2 Dynamic Friction

#### 4.2.1 Dry Operation

The coefficient of dynamic friction depends on the materials and the conditions of the surfaces as well as on the operating and loading conditions.

For the general range of carbon grades the coefficient of friction may be 0,25 at heavy loads and diminish to 0,10 at light loads. A further important fact is that the friction tends to diminish with continued and prolonged operation because of the polishing action of the carbon.

Such variables are not present in ball bearings; their coefficient of friction remains practically unchanged for all conditions of load and speed, including stopping and starting. Therefore, when considering a change from ball, roller or conventional lubricated sleeve bearings to carbon bearings, the large increase in coefficient of friction must be borne in mind.

Although the variable factors prevent definite figures for the coefficient of dynamic friction being given they are usually within the range 0,08 to 0,3.

The power absorbed by the bearing friction is:

$$P = \frac{\mu * F * d * \pi * n}{60} \text{ where}$$

P = friction power [W]

$\mu$  = coefficient of friction

F = total force on bearing [N]

d = diameter of bearing [m]

n = shaft speed [rev/min]

#### 4.2.1.1 Noise of Dry Operation Bearings

In dry bearings momentary variations in frictions are likely to be considerably greater than in lubricated bearings. Furthermore, there is no lubricant fluid present helping to absorb the vibrational forces which such variations produce.

Particular care must, therefore, be taken to avoid designs and materials that may cause excessive noise. It is essential that any necessary precautions be taken in the beginning; it is usually too late once the bearings are installed. The grade of carbon has an influence. Further, resonance effects of the mechanical parts and their arrangement adjacent to the bearings should be considered. The frequency range within which serious resonance may occur is smallest with the electro-graphite grades, and becomes progressively wider with the carbon-graphite grades and, finally, the metal-carbon grades.

#### 4.2.2 Wet Operation

When the lubricant separating bearing and shaft exists only as an adsorbed film, boundary lubrication prevails and the coefficient of friction with carbon bearings is variable, but in practice with heavy loads usually lies within the range of 0,01 to 0,10.

With complete fluid lubrication the hydrostatic pressure is sufficient to support the applied load. The friction varies directly as the viscosity and for thin lubricants, such as petrol, is very low. It also depends on the clearance and the ratio of the bearing length to diameter as well as on the load and speed.

The performance of carbon bearings is affected by the surface condition of the carbon which should be impervious.

The shaft surface finish should be of a high order, as described in [Section 9](#), and the shaft alignment be as perfect as possible. Preferably the bearing should be self-aligning.

#### 4.2.3 Air lubrication

Carbon bearings have in several instances been successfully employed at very high speeds with air lubrication. Impervious metal-carbon material is necessary for this application, together with very high surface finishes on bearing bore and shaft journal.

Suitable materials for the journal are hardened nitrided steel or "Stellite" puddled on the carbon steel or stainless steel.

These special applications should be referred to our [Technical Department](#).

## 5. Wear

### 5.1 Dry Operation

Wear is the most important limiting factor in unlubricated carbon bearings. It is mechanical and thermal in origin.

The PV values quoted in this brochure are based on the bearing wear of 0,25 microns/h.

The shaft surface is of considerable influence, and a finely finished and non-corrodible surface is recommended. The finish of the bearing bore is of less importance because the carbon is softer and more readily wears to the same smoothness as the shaft.

The formation or presence of rust on the journal is harmful and should be avoided. Rust is abrasive and can appreciably reduce the bearing life. On the other hand, the fine film of graphite developed on shaft surfaces has a beneficial lubricant effect and lengthen the bearing life.

Broadly speaking, with continuous operation the wear is directly proportional to the load, to the temperature and to the square of the speed. The carbon materials, under various operating conditions generally compare as follows: Low loads, up to 0,5 N/mm<sup>2</sup> – all Morgan carbon grades are satisfactory.

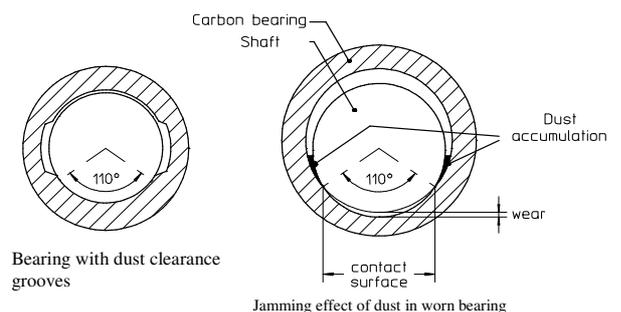
Medium loads, 0,5 to 2,8 N/mm<sup>2</sup>, speeds to suit – metal-carbon grades wear less than non-metal grades  
Heavy loads up to 7,0 N/mm<sup>2</sup> – only metal-carbon grades are suitable depending upon operating conditions.

Ambient temperature reaching 300 °C – metal-carbon grades superior to non-metal between 150 °C and 300 °C (excepting Babbitt impregnated material).

Ambient temperature above 300 °C up to 450 °C max in air – electro-graphite grades are superior to carbon-graphite grades for high PV conditions, metal-graphite grades are superior to metal-carbons. As the temperature is raised the permissible PV for electro-graphite grades must be reduced to obtain a reasonable life. In this temperature range PV's up to 0,055 [N/mm<sup>2</sup>\*m/s] are possible.

When the carbon bush is stationary carrying a rotating shaft and the clearance is too small, the dust arising from wear, accumulating at the edges of the rubbing area, can cause jamming, and possibly split the bearing if the angle of the contact between bearing and shaft exceeds 110° (see [Sketch 1](#)).

This risk can be eliminated by clearance grooves in the bearing sides but this is not always practicable because the location and extent of the area of support is not always determinable and may change as wear proceeds. The best practice is to allow adequate clearances, as specified in [Section 8](#), to give room for the wear dust to clear away. When the shaft is fixed and the bush rotates a localised rubbing area cannot form on the bore. There will always be line contact and the higher specific load will produce a greater rate of wear. Even with small running clearances the contact angle is negligible and no jamming can occur so long as provision is made so that the wear dust can escape from the end of the bearing.



Sketch 1: Bearing design for dry operation

### 5.2 Wet Operation

Under proper conditions the wear is negligible with fully lubricated conditions. The construction may sometimes be designed to make fuller use of the working fluid as a lubricant, thereby permitting very high loads and speeds. For example, on some submersible pumps a Michell thrust bearing design is highly successful with the water lubricated Michell pads made in a metal-carbon grade. However, care should be taken to ensure an adequate constant supply of fluid to the bearing. A condition of

intermittent fluid supply, possibly involving a cyclic wet or dry running, should be avoided. The transition stage from wet to dry will cause temporary high friction and possible high rate of wear.

## 6. Bearing grade recommendations

Typical mechanical properties of the various classes of carbon grades are given in the data-base "[Material Selection](#)". Individual characteristics and physical characteristics are described there in detail.

## 7. Design arrangements

There are certain general precautions to be taken in designing any bearing arrangement. If appreciable shaft deflection exists, say, due to considerable distance between bearing centres, or distortion due to temperature - a self-aligning design should be used. The greater the length of carbon bearings in relation to the bore diameter, the greater is the need to ensure some form of automatic self-alignment. [Sketch 2](#) shows an inexpensive arrangement sometimes practicable although not ideal from a mechanical point of view. For belt conveyor rollers and such arrangements, a typical design for self-aligning bearings is shown in [Sketch 3](#).

Another design of self-aligning bearing is shown in [Sketch 4](#). Good alignment should also be sought in replacing ball or roller bearings by carbon bearings. With the latter, a length to bore ratio of 1,5:1 is normal, although in some smaller and lightly loaded bearings a ratio of 2:1 is common. With a dry stationary bearing length to bore ratio of under 1:1 (length shorter than bore diameter) is not recommended, and Therefore it is not usually possible to substitute carbon bearings for ball or roller bearings retaining the same length dimensions.

With wet bearing length to bore ratios down to 0,25:1 are used successfully. With certain types of conveyor, short rollers or wheels, the length of the revolving carbon bearing should be twice the bore diameter, in the interest of wheel stability.

Specific points which should be observed are as follows:

- Flanges to take axial thrust should be avoided. When fitted, say by shrink fitting, there will be considerable stress in the carbon between the body of the bush and the free flange which may cause breakage at the neck. A thrust load is supported better by increasing the wall thickness of the bush and taking end thrust of the shaft against the end of the bush, or by a separate carbon washer of suitable thickness and diameter. See [Sketches 3, 4 and 5](#) for typical designs.
- To avoid jamming, oil, if used for neighbouring parts, should not be allowed to reach the carbon. The ingress of dust or grit should also be avoided at all costs as it prevents the carbon polishing the shaft and will cause rapid wear.
- In conditions of severe dust, a check labyrinth or some design of dust cap should be employed at each end of the bearing to prevent the ingress of dust into the clearance (see [Sketch 6](#)).
- Replacement bushes should be in the same grade as engraved on the end of the bush.
- Manufacturing limitations determine the following maximum possible sizes. Two or more equal lengths may be fitted in tandem to make up longer bushes.

Maximum available length 100 mm for diameters up to 250 mm.

Maximum available length 65 mm for diameters between 250 and 300 mm.

For strength in assembly the wall thickness should be greater than that of metal bushes, the following being recommended:

Bore [mm]	Wall thickness [mm]
Up to 12,5	3
12,5 to 25	4,5
25 to 50	6
50 to 75	8
75 to 100	9,5
100 to 150	12,5
150	16

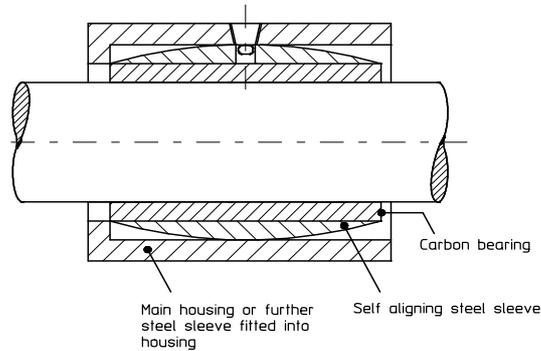
- Machining Tolerances

Carbon bush tolerances are as follows:

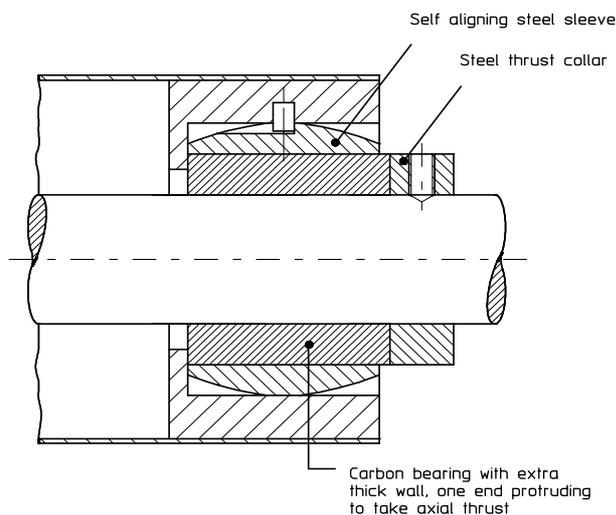
Diameter [mm]	Tolerances for I.D. and O.D. [mm]
Up to 75	0,025
75 to 150	0,050
150 to 300	0,075

Length [mm]	Tolerance [mm]
Up to 100	0,025

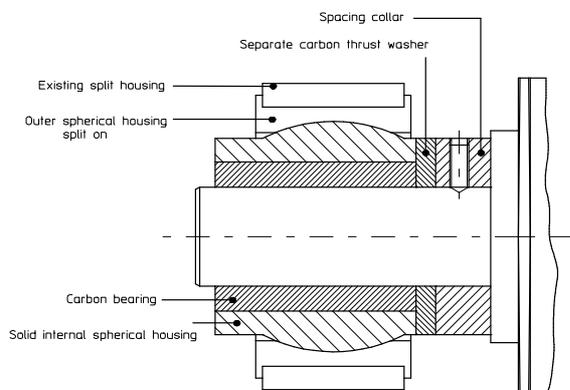
For bearings exceeding 100mm long, the bearing will be made up of two or more length fitted in tandem, each length having a tolerance of 0,25mm.



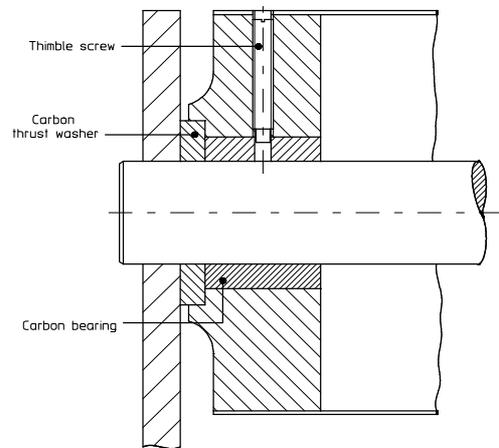
Sketch 2: Simple arrangement for self-aligning bearings



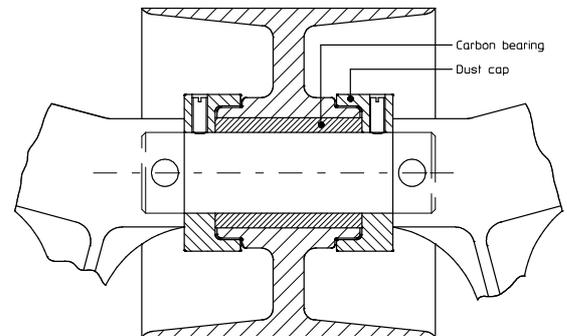
Sketch 3: An arrangement of self-aligning carbon bearing for conveyor rollers, etc. illustrating provision for axial thrust taken on end face bearing



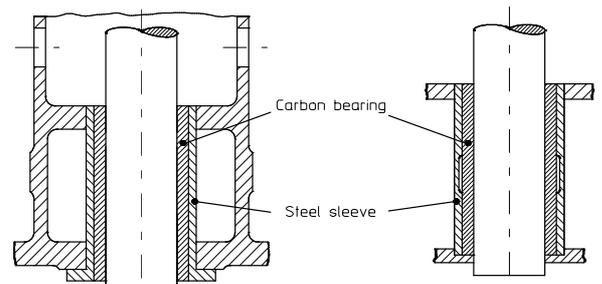
Sketch 4: Arrangement for self-aligning carbon bearings incorporating separate carbon thrust washer



Sketch 5: Method of fitting carbon bearings & trust washers to conveyor rollers



Sketch 6: Typical design showing carbon bearing protected by dust caps



Sketch 7: Typical arrangement for fitting carbon bearings to centrifugal pump, showing indirect fitting of carbon into steel sleeve

## 8. Fitting methods

Four methods of fitting carbon bearings can be employed according to requirements:

- Shrink-fitting
- Press-fitting
- Slack-fitting
- Split-bearing fitting

### 8.1 Shrink-fitting

This employs heat expansion of the metal housing and dropping the cold carbon bush into it, the O.D. of the

carbon bush being larger than the I.D. of the cold housing. This enables an interference fit to be obtained between carbon bush and metal housing much in excess of that obtainable by press-fitting.

Shrink-fitting is employed when large and/or heavily loaded carbon bearings are employed in conditions of large temperature changes.

Shrink-fitting requires that both the O.D. of the carbon bush and the I.D. of the metal housing must have unbroken surfaces (i.e., housing provides complete support to the bush along its entire length), otherwise cracking of the carbon bush may result.

For shrink-fitting, the dimensions of the bush and housing should be so arranged that there is residual interference between carbon and housing at the maximum working temperature. When employed thus the carbon will (up to the temperature where I.D. of the housing becomes greater than the O.D. of the free carbon bush), behave with the apparent coefficient of expansion of the metal housing. This fact is important, when considering the running clearance between shaft and carbon bearing and the material of the shaft. For high temperatures, say 250 to 500 °C, housing material of relatively low coefficient of expansion, about  $10 \cdot 10^{-6} \text{ m}/(\text{m} \cdot ^\circ\text{C})$ , should be employed, such as cast-iron or steel, otherwise, as with say aluminium ( $25 \cdot 10^{-6} \text{ m}/(\text{m} \cdot ^\circ\text{C})$ ), the initial interference between carbon bush and housing would have to be so large that it would crush the carbon when shrinking-in.

It will be found that below 25 mm diameter it will be necessary to use very close tolerances for bush O.D. and housing I.D. to maintain sufficient interference at, say 400 °C operating temperature, and interference up to 0,4% of bearing O.D. is good practise.

Shrink fitting directly into housings having high coefficient of expansion such as bronze ( $18 \cdot 10^{-6} \text{ m}/(\text{m} \cdot ^\circ\text{C})$ ) or aluminium ( $25 \cdot 10^{-6} \text{ m}/(\text{m} \cdot ^\circ\text{C})$ ) should be used only for convenience of fitting or for a very small temperature range. Where it is necessary to fit a carbon bush, say, into aluminium for a wider temperature range, the carbon bush should first be shrunk into a steel sleeve which would then be fitted in the usual way into the aluminium housing. Similarly, if the housing into which the carbon bearing is to be shrink-fitted is too large to heat, or inconvenient to heat due to risk of distortion, then this method of first shrink-fitting the bearing into a steel sleeve can be adopted (see [Sketch 7](#)).

It should be remembered that the working temperature limits for the various grades of carbon are:

Plain carbon, carbon-graphite and metal carbon grades (excluding Babbitt impregnation)	350 °C
Electrographitic grades	450 to 500 °C

Running clearances within the limits of working temperature remain practically constant when the coefficient of expansion of journal and housing are alike. When the coefficient of expansion of the shaft material is greater than that of the housing then greater initial clearance must be allowed between the cold shaft and bearing.

The required tolerances and temperatures can be calculated as follow:

$$\Delta\alpha = \alpha_H - \alpha_B$$

$$I_{C \min} = t_W * 1,1 * \Delta\alpha * d_B$$

$$t_{f+f} = \frac{I_C}{\Delta\alpha * d_B}$$

$$t_S = \frac{I_C - 0,1 * I_C (\min 0,05 \text{mm})}{\alpha_H * d_B}$$

$\Delta\alpha$  = Differential coefficient of expansion of housing to carbon

$\alpha_H$  = Coefficient of expansion of housing material [ $\text{m}/(\text{m} \cdot ^\circ\text{C})$ ]

$\alpha_B$  = Coefficient of expansion of carbon bush material [ $\text{m}/(\text{m} \cdot ^\circ\text{C})$ ]

$I_{C \min}$  = Minimum cold interference required

$I_C$  = Cold interference of carbon bush [mm]

$t_W$  = Working temperature [°C]

$t_{f+f}$  = Temperature [°C] at which carbon bush will become fit-and-fit with housing

$t_S$  = Shrinking-in temperature [°C]

$d_B$  = Bush O.D. [mm]

## 8.2 Press-fitting

This should only be employed with smaller bearing sizes and where the working temperature range is likely to be small.

A maximum interference rate of 0,05mm per 25mm, is usually allowed with maximum allowance of 0,15mm.

Even so, the bushes must be pressed into their housings carefully, i.e., bush square with housing bore and application of steady pressure, otherwise the bush will be found to be cracked after assembly. Carbon bushes must not be hammered into position.

## 8.3 Slack-fitting

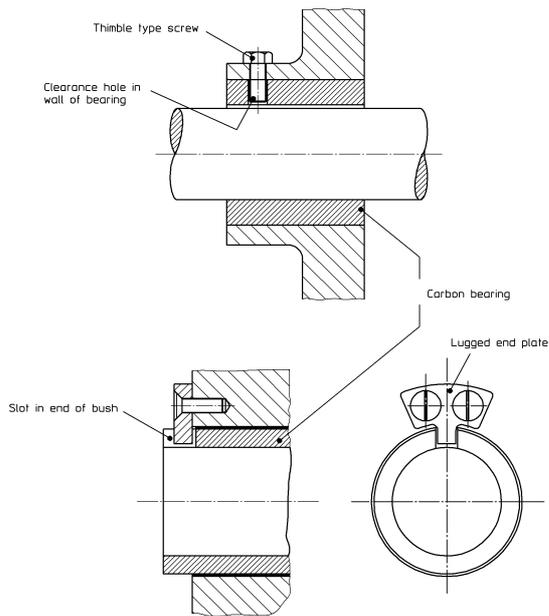
Where the loading is not onerous and provided that generous wall thickness is employed, the carbon bush may be fitted with a small clearance in the housing, i.e., slide fit.

The bush maybe keyed at one or both ends of the housing by means of lugged key plates, the lugs of which engage in slots in the end of the bush (see [Sketch 8](#)), or, alternatively, a thimble screw spaced well away from the bush end, opposite to loaded area (see [Sketch 8](#))

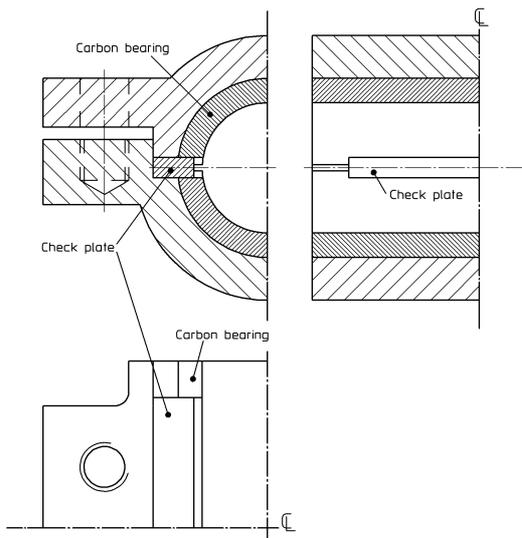
The bush maybe left free to rotate provision only being made to prevent the bush from moving endwise out of the housing.

## 8.4 Split bushes

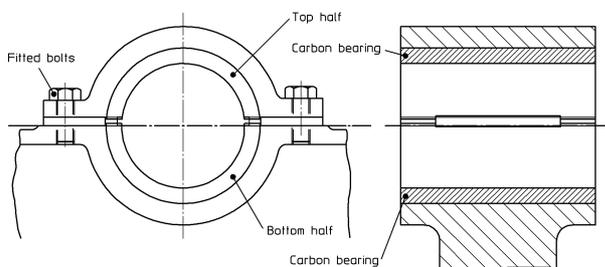
Use of split bushes should if possible be avoided. They are more fragile than solid bushes. If the bearing design makes splitting necessary the halves should be located by check plates (see [Sketches 9 and 10](#)). The dimensions of the carbon half bushes and the surrounding shell must be so arranged that the carbon is not subjected to undue pressure when the bearing cap is tightened.



Sketch 8: Methods of retaining carbon bearings



Sketch 9: Housing arrangement for split carbon bearing



Sketch 10: Typical arrangement for split carbon bearing showing check plates

## 9. Clearance and tolerances

### 9.1 Dry operation

The running clearance for dry bearings should in no case be less than 0,05 mm and is usually 0,3% of the shaft diameter.

Recommended tolerances for dry bearings are:

Housings:	H7
Bore after assembly:	D9
Shaft:	f8
Corresponding clearances:	D9-f8

Dry bearings will not jam if these tolerances are respected. However, as wear increases the angle of contact with a rotating shaft will gradually increase. To avoid risk of jamming, worn bearings should be serviced before a contact angle of  $110^\circ$  is reached on the bearing area (see [Sketch 1](#)). With high loads, giving quicker wear, the initial clearance should be correspondingly increased.

### 9.2 Wet operation

With lubrication there is no accumulation of dust and the cooling is usually good. Clearances can correspond more to those of ordinary bearing practise and the table below should be followed. Entry of liquid between the bearing surfaces is assisted, if necessary, by, say, three axial grooves in the bore suitably placed in relation to the loaded area. Depending on the bearing size a depth of 1mm to 2mm is sufficient.

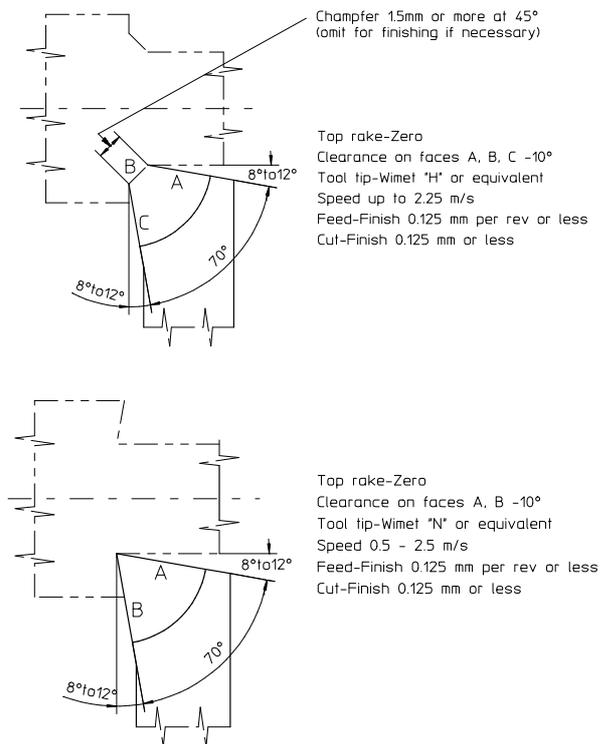
In certain applications finer running clearances may be necessary and the tolerances employed must be chosen on the merits of the case. If extremely fine running clearances are required, or if the assembled bore diameter is to be held to a close tolerance, it may be necessary finally to machine the bore of the bush to the required size after assembly into its housing.

Recommended tolerances for lubricated operation are:

Housings:	H7
Bore after assembly:	E9 to D9
Shaft:	f8
Corresponding clearances:	E9-f8 to D9-f8

### 9.3 Machining

Whilst it is normal practise for carbon bearings to be supplied fully machined to size and ready for fitting, in certain cases machining may be necessary, e.g., machining bores after assembly, prototype bearings, etc. [Sketch 11](#) shows the type of tool necessary, and also the recommended machining technique for centre lathe turning.



Sketch 11 Recommended turning tool profiles

## 10. Shaft material and finish

The shaft material and surface finish are of great importance and considerably influence the rate of wear of the carbon. It is generally agreed that the harder and more polished the surface the better the results.

When mater is present or moisture forms on the bearing from condensation or some other source, non-rusting material must be used. Generally speaking, shafts should be non-rusting iron or steel. If ordinary steels are employed they should be plated with nickel or chromium. Also with acids and alkalis, even in very low concentrations, the material should not corrode electrolytically and an experiment or preliminary test, as mentioned in [Section 2](#), should be made.

Shaft materials used for carbon bearings are:

### Mild steel

Attractive because plentiful and low in price, but it is not a good material for working against carbon. It rusts easily and gives satisfactory results only when constantly above a temperature which ensures freedom from rust.

### Grey cast iron

Little used because of low mechanical strength and inclination to rust easily. Otherwise it is one of the best materials for working against carbon, giving low friction and long life.

### Austenitic cast iron

Nearly as good as grey cast iron and has the advantage of being non-rusting and is stronger and more ductile than cast iron. Costs more than cast iron or steel but is cheaper than stainless steel.

### Stellite

A proprietary alloy of 55% cobalt, 15% tungsten, 30% chromium puddled on steel with oxy-acetylene flame. Stellite is by far the best material for working against carbon, but it should be given a high degree of finish. Both Stellite 1 and Stellite 6 are particularly suitable as they are "puddled" onto steel using an oxy-acetylene flame. Hardness is retained at all working temperatures, which is an important quality for reducing wear.

### Stainless steel

Gives good results. Hardened varieties preferred having Brinell about 400.

### Sprayed stainless steel

A recent and comparatively cheap deposited protective surface on mild steel, by the metallisation process. Can be applied to large diameter shafts and complicated shapes. The deposit appears able to resist frictional wear at moderate loads and speeds.

### Nitrided steel

Good results obtainable at hardness of 500 to 600 Brinell. Has a surface of uniform hardness and is usually sufficiently non-rusting

### Chromium plated steel

Excellent results are obtainable with chromium plating but in mass production plating faults, particularly porosity are not always avoidable. Slivers of plating may become detached and cause rapid wear of the carbon bush. For most operating conditions the final deposit after grinding should not be less than 0,125mm and be preferably between 0,2mm and 0,3mm with fine ground or lapped finish. There should be an underlying deposit of nickel and the total thickness before grinding should allow sufficient margin to give the required final thickness. The care required to obtain a satisfactory deposit is rather an obstacle in some cases to the use of deposited chromium.

The following shaft materials should be avoided:

### Case hardened steel

The surface hardness is seldom uniform and this may cause undue wear of the carbon bush.

### Aluminium and its alloys

These are not recommended. Some use has been made of specially deep-anodised aluminium but a considerable risk of corrosion and wear remains.

### Bronzes

All bronzes and, generally speaking, most other non-ferrous metals are to be avoided for dry bearings.

#### *Shaft finish*

The machining should be as fine as possible. If the surface is rough it will be polished in time by the carbon but the rate of wear of the carbon will be initially high. For heavy loadings, wet and dry, a fine finish, i.e., a roughness of 0,25 to 0,5 microns, is recommended. Fine turning, i.e., a roughness of 2 microns is considered suitable for easy applications, but is recommended only for dry applications.

## 11. Some successful applications

### 11.1 Dry operations

#### **Conveyors**

In food, textile, and chemical industries where elimination of free oil or grease is necessary.

#### **Scroll Conveyors**

Handling powders, where conventional bearings are not possible.

#### **Foundry Slat Conveyors**

Where temperature and conditions render conventional bearings unsuitable.

#### **Chain Grate Stokers**

Industrial and electricity generating station plant. Bearing temperatures prohibit conventional lubrication.

#### **Boiler Plant**

Flue gas, induced and forced draft damper and spot blower bearings on boiler equipment, where temperatures prevent normal lubricated bearings being employed.

#### **Furnace and Oven Plant**

In furnace roller and furnace charger roller bearings in the steel industry, bearings on oven conveyors and continuous belt ovens, etc., where the use of normal lubricated bearings is impracticable due to temperature.

#### **Veneer and Paper-Drying Machines**

Contamination by conventional lubricants must be eliminated, and also temperatures are sufficiently high to cause lubrication difficulties with normal bearings. Carbon bearings very satisfactory.

#### **Rollers on Micro-Switches**

Presence of oil or grease undesirable. Complete carbon roller contacts very satisfactory.

#### **Textile Machinery**

Cloth drying plant, stentering machines, synthetic fibre spinning, etc. Contamination by oil or greases must be avoided at all costs.

#### **Soaking Pit Covers**

In the steel industry where temperatures prohibit normal bearings. Main support roller bushed with carbon bearings highly satisfactory.

#### **Cigarette Manufacturing Equipment**

Carbon bearings on sealing plant and also curing ovens, etc., where contamination must be avoided.

## 11.2 Wet Operation

#### **Centrifugal pumps**

Impeller steady bearings operating immersed in various liquids. Carbon used for steady bearings in several instances.

#### **Gear Pumps**

Main gear bearings, being lubricated by the liquid being pumped.

#### **Fuel Metering Pumps**

Bearings operating immersed in fuels, such as petrol, paraffin, kerosene, etc.

#### **Submersible Pump Units**

Carbon bearings used in many positions of such equipment, both in the pump and drive motor, including main motor bearings and thrust pads which operate immersed in filtered water.

#### **Glandless Centrifugal Pumps**

Main impeller bearings lubricated by the liquid being handled.

#### **Central Heating Circulating Pumps**

Main impeller bearings in carbon operating immersed in hot water.

#### **Bottle Cleansing Plant**

Carbon bearings are employed in situation subjected to cleansing fluids, which prevent the use of conventional lubricated bearings.

#### **Chemical Plant**

Mixing vessels, agitators, etc., carbon used as steady bearings in a variety of chemical liquids.

#### **Textile Industry**

Dye Jigs where carbon bearings operate both immersed in and above the dyeing solution. Wet spinning of synthetic fibres, where carbon bearings are splashed with acidic liquids.

#### **Automatic Washing Machines**

Carbon bearings used as main impeller bearings, operating immersed in hot water

#### **Liquid Measuring Meters**

Miscellaneous carbon bearings in this equipment, running in various liquids.

## Further details of some typical applications

### Description and performance

#### **Chain Grate Stoker**

Formerly, grease lubricated cast iron bearings were employed. Cast iron bearings needed replacement at approximately 18-month interval, after 9,5mm radial wear. Shafts also badly worn and scored needed "building-up"

and re-grinding. Considerable consumption of grease and frequent maintenance.

In one case after several years, carbon bearings have worn only 0,08mm and shafts are highly polished with negligible wear. No maintenance required and use of grease obviate. In another stoker, carbon bearings have run for 17 years, without replacement.

#### **Flue Gas Damper**

Formerly, unlubricated cast iron bearings employed. Excessive and rapid bearing and shaft wear, with frequent seizures with cast iron bearings. Wear of carbon bearings negligible over long operating periods. No shaft wear and no seizures experienced.

#### **Ten-ton Furnace Charging Machine**

Carbon bearings proven highly satisfactory. No conventional type of bearings possible because of high operating temperature. Carbon bearings have been in service for seven years to date.

#### **Veneer Drying Oven**

A total of 850 carbon bearings are fitted to this oven, each pair of bearing supporting a 400mm long hollow steel roller. Each bearing is mounted in a self-aligning housing. Carbon bearings have been in service for eight years to date without the need of any maintenance. Before the employment of carbon bearings, machines were fitted with grease lubricated ball bearings which required frequent attention.

#### **Rust Proofing Conveyor**

Sever hundred carbon bearings are fitted to a 4,6m long double roller suspension conveyor. Conveyor passes through various stages of bonderising cycle, i.e., caustic solution 90 °C, water at 80 °C, Phosphate solution at 90 °C, Chromic acid rinse at 80 °C, and, finally, drying oven at 230 °C. Carbon bearings originally fitted in 1937 and are still giving satisfactory performance with very few replacements being necessary. Original oil-retaining sintered bronze bearings gave constant trouble – excessive wear, high friction and frequent seizures due to high temperature.

#### **Submersible Pump Motor**

Carbon bearings employed as 260kW motor main bearing operating completely immersed in circulating distilled water. Bearings, therefore, copiously water-lubricated. Highly satisfactory results obtained with negligible bearing or shaft wear after several years continuous service. Original bearings employed were lignum vitae which gave trouble due to swelling. Frequent seizures and excessive wear were experienced.

#### **Vertical Centrifugal Pump**

Carbon bearings employed as steady bearings, running completely immersed in circulating 10% caustic solution. Negligible bearing or shaft wear after several month continuous service. Previous pumps employed Bakelised Fabric composition bearings, which gave trouble due to swelling.

#### **Aircraft Fuel Pump “Swash Plate” Type**

Carbon bearings employed as main rotor bearings operating completely immersed in circulating fuel, i.e., petrol, paraffin, kerosene, etc. Highly successful application. Carbon bearings found to be the most suitable type of bearings for these pumps.

## **12. Morgan seals for bearings**

In certain instances, where operating conditions render it difficult to apply either conventional or carbon bearings it is frequently possible to effect a solution by means of standard Morgan carbon seals. Such seals are able to seal against the ingress to the bearing clearance of abrasive dust and grit, deleterious chemical liquids, etc., and also capable of holding oil or grease in conventional bearings up to 100 °C.

## **13 Questionnaire Form**

As mentioned in the foreword, our Technical Department is always available to assist in solving difficult bearing problems. The selection of a suitable grade of carbon and also the design of the bearing is dependent on operating conditions. To obtain full benefit from our Technical Department the fullest possible details, together with any available drawing or sketches illustrating the application, should be submitted.