

(19)



Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number:

0 299 692
A2

(12)

EUROPEAN PATENT APPLICATION

(21) Application number: 88306285.3

(51) Int. Cl. 4: **B24B 9/16** , **B24B 37/04** ,
B24B 41/04

(22) Date of filing: 08.07.88

(30) Priority: 14.07.87 GB 8716512

(43) Date of publication of application:
18.01.89 Bulletin 89/03

(84) Designated Contracting States:
BE DE FR LU NL

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(54) **Gemstone polishing apparatus.**

(57) In order to centre the residual imbalance of a scaife 5, the scaife 5 is carried on an air bearing 6 which is axially stiff but allows radial float, and is driven by a motor 2 which is carried on silent blocks 3 which are axially stiff but allow radial float of the motor 2. The motor 2 is driven at a speed above the first radial critical frequency of the system formed by the motor 2, motor shaft 4 and scaife 5 and its mounting.

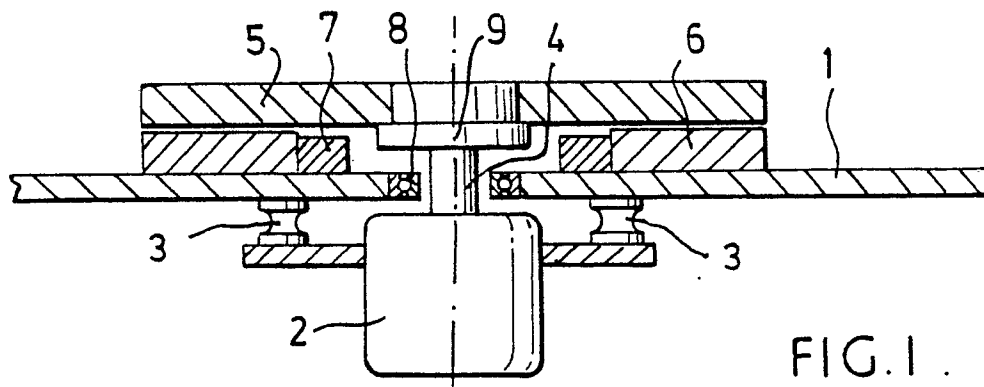


FIG. 1 .

EP 0 299 692 A2

GEMSTONE POLISHING APPARATUS

Background of the Invention

The present invention relates to a gemstone polishing apparatus comprising a scaife, a supporting structure for the scaife, a bearing for the scaife, and a drive for the scaife.

A scaife is a wheel which is used to polish gemstones, e.g. polishing facets on a brilliant-cut diamond. The scaife runs at a high speed, e.g. up to 6000 rpm (3000 rpm for standard scaifes), normally about a vertical axis, and a flat face of the scaife is used for the polishing, usually with the application of an abrasive oil to the face. Normally, the scaife has a diameter of e.g. 150 to 350 mm and a thickness of 10 to 20 mm and is made of soft steel or cast iron, so that it is heavy, for example weighing 10 to 15 kg. In the art, though not in this specification, the term "scaife" is also used to include the whole item of equipment or apparatus. The supporting structure is normally a flat table. The gemstones can be applied to the scaife in hand-held dops, but are normally mounted on polishing machines, referred to as tangs, which hold the stones in the correct orientation, and more than one tang can be used at the same time on the same scaife.

To obtain a very high quality polished diamond surface, any vibration of the scaife and table must be avoided. This is a severe problem for direct-driven scaifes where the drive is a motor generally coaxial with the scaife, as motor-induced vibrations are transmitted through the table and the scaife. This problem can be reduced by special motor drives, or by using a special motor and bearing design, or by using a normal motor with a radial-axial air bearing combination for the scaife. Such solutions are quite expensive, for example ten to twenty times the cost of a standard drive motor, and they cannot reduce the residual imbalance remaining after assembling the balanced scaife.

The Invention

The present invention provides gemstone polishing apparatus as set forth in Claim 1 and a method as set forth in Claims 17. The remaining Claims set forth optional features of the invention.

The system is run over-critically, i.e. between nodes of the critical frequency, so that the scaife is self-balancing - although the scaife (and the motor) should be separately balanced, the radial float of the scaife centres the residual imbalance.

The technique of balancing by operating at

over-critical speed has been used in the past, it is believed in turbines, in centrifugal separators and domestic spin driers.

In polishing, axial position stability is important, the necessary axial stiffness must be introduced, for instance using the rear face or backplane of the scaife as an air bearing surface.

In a first approximation, the first radial critical frequency of the scaife is given by $f = 2\pi \sqrt{k/m}$, where k is the radial stiffness and m is the mass of the system. The running frequency must be higher than the first radial critical frequency, but the amount depends upon the desired running accuracy and the damping in the system. Higher damping requires a larger difference between the running frequency and the first radial critical frequency. As to achieve perfect auto-centering, the unbalance vector has to be opposite to the displacement vector, a 180° shift or phase angle. For low damping, the magnitude of unbalance is very high when going through the critical frequency, but above critical frequencies the magnitude soon becomes very small, with a phase angle very close to 180° . High damping, however far above critical frequency, maintains a residual magnitude, while the phase angle goes gradually towards 180° . The running speed can be calculated by predetermining a minimum phase shift. Hence a system with very low damping will result in a running speed about 20 to 30% above the critical frequency, while high damping systems will result in a running speed of 5 to 10 times the critical frequency. The advantage of using damping is avoiding a large unsafe balance magnitude when running through the critical frequency. The disadvantage is a larger residual unbalance and a weaker spring system. Also, the higher the applied drag force due to the polishing action of the stone, the greater the difference required between the frequencies as the grinding force may affect the position of the axis of rotation.

The radial float may be free, i.e. not against any elastic force, in ideal conditions, but in general there will be some radial elastic restraint or stiffness (see the equation above for the first radial critical frequency).

To minimise the effect of vibration modes due to an elastic drive system or an elastic mount of the drive motor, it is advisable to use a light-weight motor and to have the centre of gravity of the scaife, motor and motor shaft approximately on the plane of support of the axial scaife bearing - this takes advantage of the low power needed to drive the scaife (less than 1 kw) and the heavy weight of the scaife.

An axial bearing pre-load can be applied e.g.

by direct magnetic action on the scaife or air pressure cells, and/or by the deadweight of the motor, and/or pre-loading an air bearing by use of the motor mounting means.

The motor drive shaft may be rigidly fixed to the scaife or there may be a universal or flexible coupling, i.e. a flexible or swivel coupling where rotary motion is permitted between the motor drive shaft and the scaife about transverse axes, or even a coupling which permits radial float. Such a universal, flexible or floating coupling must be able to transmit torque, and must provide (if necessary in association with other means) a stiffness corresponding to an over-critical drive, and in some cases, depending upon the pre-load system, must be capable of transmitting preload to the axial scaife bearing.

In general, whether using a universal, flexible or floating coupling or another arrangement, there may be means for preventing substantial radial float of the scaife, at will. Such means make it possible to autobalance the system at will by running the scaife over-critical without using the means; by actuating the means, the scaife can be run under-critically but balanced and for instance can bear high tangential loads such as may occur during multi-tang polishing. An advantage of such an arrangement is that during start-up, one does not have to go through the critical speed. A disadvantage is that the motor alignment must be more accurate because when the radial float is prevented, misalignment cannot be taken up, resulting in excitation forces and hence increased vibration. Various means for preventing radial float at will, are disclosed below. An alternative, at least in theory, is to use a fluid (e.g. air) radial bearing whose radial stiffness can be changed by changing the fluid pressure.

Preferred Embodiments

The invention will be further described, by way of example, with reference to the accompanying drawings, in which the Figures illustrate four different embodiments of the invention in side view, partly in axial section.

Figure 1

Figure 1 shows a gemstone polishing apparatus having a supporting structure including a table 1 carrying a drive motor 2 (such as a high quality AC motor with limited mechanical and magnetic imbalance) by way of flexible couplings in the form of silent blocks 3 which are axially stiff but radially weak. The motor shaft 4 is rigidly fixed to a scaife

5. In order to provide a bearing which is axially stiff but allows radial float of the scaife, the scaife has an axial air bearing one part of which is the rear or bottom face of the scaife 5 and the other part of which is indicated at 6. Axial pre-load is provided by the deadweight of the motor 2, shaft 4 and scaife 5, but if desired, further axial pre-load can be provided by the silent blocks 3 or by magnets 7 exerting direct magnetic action on the scaife (which will be ferromagnetic). In order to limit the radial float of the scaife 5, there is an anti-friction bearing 8 around the shaft 4. In the case of a soft steel or cast iron scaife 5 having a diameter of 300-350 mm and a thickness of 10-20 mm, a radial gap can be provided between the bearing 8 and the shaft 4 (when the shaft is central) of about 0.2 to 0.3 mm, for example, i.e. at least the sum of the eccentricities of the scaife 5 and of the shaft 4.

Though the air bearing is shown as extending to the outer periphery of the scaife 5, it can for instance have an outer diameter of 110-285 mm. The air bearing should be effective enough to limit axial movement of the scaife 5 to less than ± 1 micron. One of the advantages of the use of the axial air bearing underneath the scaife 5 is the high stiffness and the fact that bearing errors of the drive have negligible effect on the vertical vibration of the scaife, and hence one can use a traditional, cheap motor drive. The vertical movement of the polishing surface of the scaife is nearly completely determined by the flatness of the lower side of the scaife (and if a multipad air bearing is used, that error is averaged out to about one third of its actual value) and the non-parallelism of the scaife 5, which can be kept very low. Hence the bearing on the back of the scaife 5 provides good vertical running accuracy.

The motor 2 is of relatively light weight whereas the scaife 5 is heavy, and the centre of gravity of the rotationally rigid system formed by the motor 2, shaft 4 and scaife 5 is at 9, approximately on the plane of the axial bearing or just slightly below the plane.

In this case, the first radial critical frequency is given by $F = 2\pi \sqrt{k/m}$ where k is the radial stiffness of the silent blocks 3 and m is the mass of the rigid motor/shaft/scaife system. The motor 2 is run at a frequency higher than this first radial critical frequency.

Purely as an example, the scaife running speed can be 3,000 rpm, with a scaife mass of 15 kg and a mass of the motor 2 and the remainder of the system of 7 kg. The silent blocks 3 have a damping ratio of 0.3. At the working speed, the phase angle is assumed to be 165° (instead of the theoretical 180°). The speed ratio is theoretically 3, giving a natural frequency of 1000 rpm or 17 Hz, or $\omega = 17 \times 2\pi = 105$ r/sec. The silent blocks 3 have

a radial stiffness $k = m\omega^2 = 22 \times 105^2 = 240$ N/mm. The axial stiffness is determined by the stiffness of the axial bearing and is preferably from half to one tenth of the radial stiffness of the silent blocks 3, the choice depending upon mounting accuracy and bearing pre-load.

If the air bearing is an aerostatic bearing, an air pressure reservoir is necessary to avoid running the motor 2 without pressurised air. The motor 2 is only energised if a certain minimum air pressure is present. The reservoir should contain at least sufficient air to pressurise the bearing during running out (i.e. as the scaife 5 slows to a stop), and preferably for sufficient time to polish a facet (or even a whole stone) plus the running out time. The electrical resistance of the air gap can be checked to avoid starting the motor 2 before the electrical resistance reaches a threshold, e.g. 100 ohms. Such measures are not necessary when using an aerodynamic (e.g. herringbone) air bearing.

Figure 2

Figure 2 shows an arrangement similar to that of Figure 1, but the shaft 4 carries a universal joint 10 which permits rotary motion between the shaft 4 and the scaife 5 about transverse axes, whilst transmitting torque (a splined arrangement can be used). However, the radial movement of the scaife 5 can be blocked using a magnetic clamp 11. When blocked, the system is very similar to that of Figure 1 but is not sensitive to misalignment of the motor 2. However, in a different arrangement, the silent blocks 3 can be replaced by rigid blocks to prevent lateral float (as well as axial float) of the motor 2 and the scaife 5 can be run under-critically so that it can bear high tangential loads e.g. for multi-tang polishing.

Figure 3

Figure 3 shows a somewhat different arrangement in which the motor shaft 4 is connected to the scaife 5 by a torque transmitter 12 which permits limited movement in any radial direction and limited axial movement. There is a radial spring system 13 which applies an elastic bias to radial movements of the scaife 5. In addition, there is an axially-movable clutch plate 14 which can be made to bear against the underside of the scaife 5. The motor 2 is carried on the table 1 by means of blocks 15 which allow axial movement as well as radial float, and a controlled preload is applied to the casing of the motor 2 by an arrangement indicated schematically as a pivoted arm 16 having a pre-load applied by a spring 17 which can be

reduced by means of a coil 18. During the stop and start cycle, the motor 2 is lifted up and keeps the scaife 5 in balance due to the engagement of the clutch plate 14 with the underside of the scaife 5. During over-critical running, the air bearing can be pre-loaded by the deadweight of the motor 2 via the torque transmitter 12, possibly additionally using a magnetic pre-load as in Figures 1 and 2. The first radial critical frequency is determined by the mass of the scaife 5 and the spring constant of the spring system 13.

Figure 4

Figure 4 shows an arrangement in which, with a heavy scaife 5 and well engineered mounting of the scaife 5 onto the motor drive, the radial bearing is on the housing of the motor 2. Because the motor housing makes only very small oscillatory movements, there is no air bearing and the silent blocks 3 are replaced by axially-stiff rolling balls 21 and the radial stiffness is purely determined by a radial working spring or spring and damper system 22 (shown schematically). The ball bearing system is carried by an annular race 23 hung on pre-stressed bolts 24 and engages a motor housing flange 25.

As the unbalance exentricity becomes quite small when running over critically, and the proper frequency can be kept low by low spring stiffness and damping, the force transferred to the table 1 (which is approximately determined by the product of the spring stiffness and the displacement) can be kept low and hence the table excitation is very low, giving smoother running equipment.

The present invention has been described above purely by way of example, and modifications can be made within the spirit of the invention. The invention also consists in any individual features described herein or any combination of such features or any generalisation of such features or combination.

Claims

1. A gemstone polishing structure comprising a scaife (5), a supporting structure (1), an axial bearing (6 or 21,23) supporting the scaife (5), and a drive (2) for the scaife (5), characterised in that the drive (2) is arranged to drive the scaife (5) at a speed above its first radial critical frequency, and the axial bearing (6,21,23) is axially stiff but allows radial float of the scaife (5).

2. The apparatus of Claim 1, wherein the drive is a motor (2) generally coaxial with the scaife (5) and mounted on the supporting structure (1).

3. The apparatus of Claim 2, wherein the motor (2) is mounted on the supporting structure (1) by elastic mounting means (3 or 15).

4. The apparatus of Claim 2 or 3, wherein the motor drive shaft (4) is rigidly fixed to the scaife (5).

5. The apparatus of Claim 4, wherein the centre of gravity (9) of the scaife (5), motor (2) and motor shaft (4) is approximately on the plane of support of the axial scaife bearing (6).

6. The apparatus of any of Claims 1 to 3, wherein the motor (2) is connected to the scaife (5) by way of a flexible coupling (10,13).

7. The apparatus of any of Claims 1 to 3, wherein rotary motion is permitted about transverse axes between the motor drive shaft (4) and the scaife (5).

8. The apparatus of Claim 6 or 7, wherein the flexibility of said flexible coupling (10,13), or rotary motion about said transverse axes, can be blocked.

9. The apparatus of any of the preceding Claims, and comprising means (11,14) for preventing radial float of the scaife (5), at will.

10. The apparatus of any of the preceding Claims, wherein the axial scaife bearing is an axial fluid bearing (6).

11. The apparatus of any of Claim 10, wherein one part of the axial fluid bearing (6) is the rear face of the scaife (5).

12. The apparatus of any of the preceding Claims, wherein the scaife (5) is at least in part ferro-magnetic, and a bearing pre-load is obtained by direct magnetic action on the scaife (5) by a magnetic means (7) fixed to the supporting structure (1).

13. The apparatus of any of the preceding Claims, wherein the drive is a motor (2) generally coaxial with the scaife (5) and mounted on the supporting structure by mounting means (3,15) which apply an axial pre-load to the scaife axial bearing (6).

14. The apparatus of any of the preceding Claims, wherein the radial float of the scaife (5) is limited by the bearing (8) around the scaife drive shaft (4).

15. The apparatus of any of the preceding Claims, wherein a radial spring system (22) applies an elastic bias to radial movements of the scaife (5).

16. The apparatus of Claim 15, wherein the radial spring system (22) includes damping.

17. A method of polishing a gemstone, comprising using a scaife (5) which is supported by an axial bearing (6) or (21,23), characterised in that the axial bearing (6) or (21,23) can float radially and in that the scaife (5) is driven at a speed above its first radial critical frequency.

