United States Patent [19]

Thome

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[45] **Oct. 7, 1975**

[54] **FURNACE FAN ASSEMBLY**

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[73] Assignee: **Midland-Ross Corporation,** Cleveland, Ohio

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[21] Appl. No.: **443,340**

[52] **U.S. Cl.** .... **415/219 C;** 415/180; 308/168

[51] **Int. Cl.**² **F04D 29/40;** F04D 29/20

[58] **Field of Search** .... 415/180, 175, 204, 219 C; 64/1 V; 308/168, 169, 194

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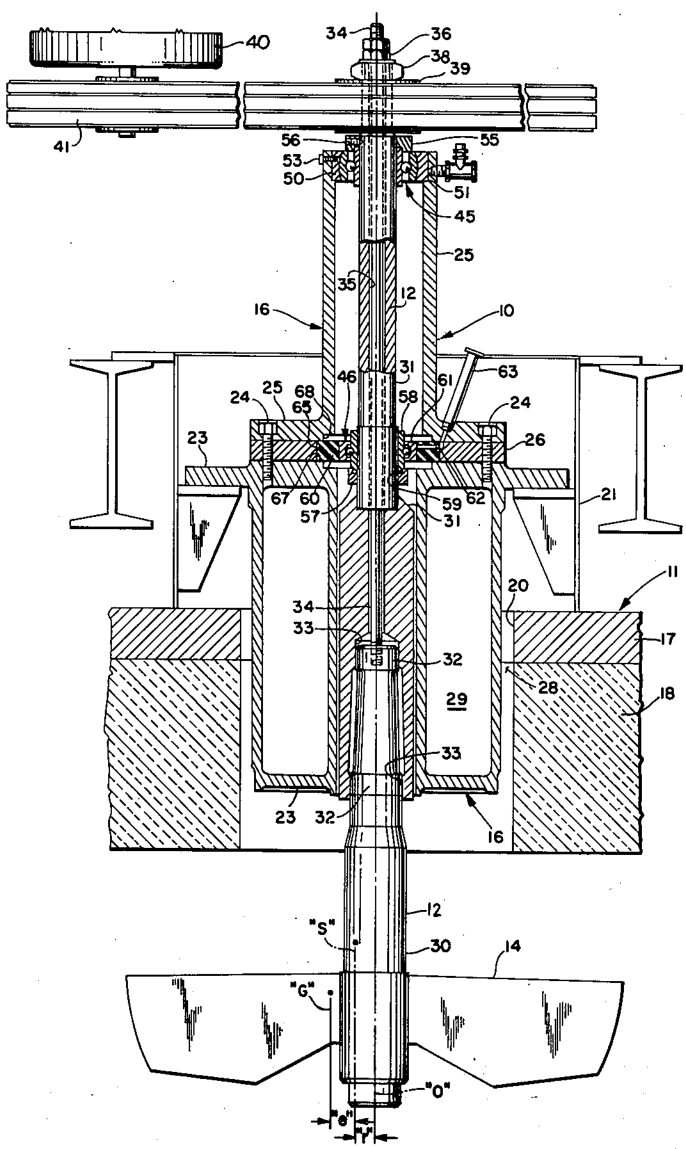
Attorney, Agent, or Firm—Frank J. Nawalanic

[57] **ABSTRACT**

A furnace fan assembly includes a unique bearing arrangement which materially simplifies the fan structure while permitting the fan assembly to rotate at relatively high speeds without transmitting damaging vibrations to the fan mounting structure. The assembly includes a shaft with a fan wheel adjacent one end thereof. The shaft is journaled at its opposite end in a first bearing and a second bearing is positioned between the first bearing and fan wheel. The second bearing is mounted in a resilient support.

2 Claims, 5 Drawing Figures



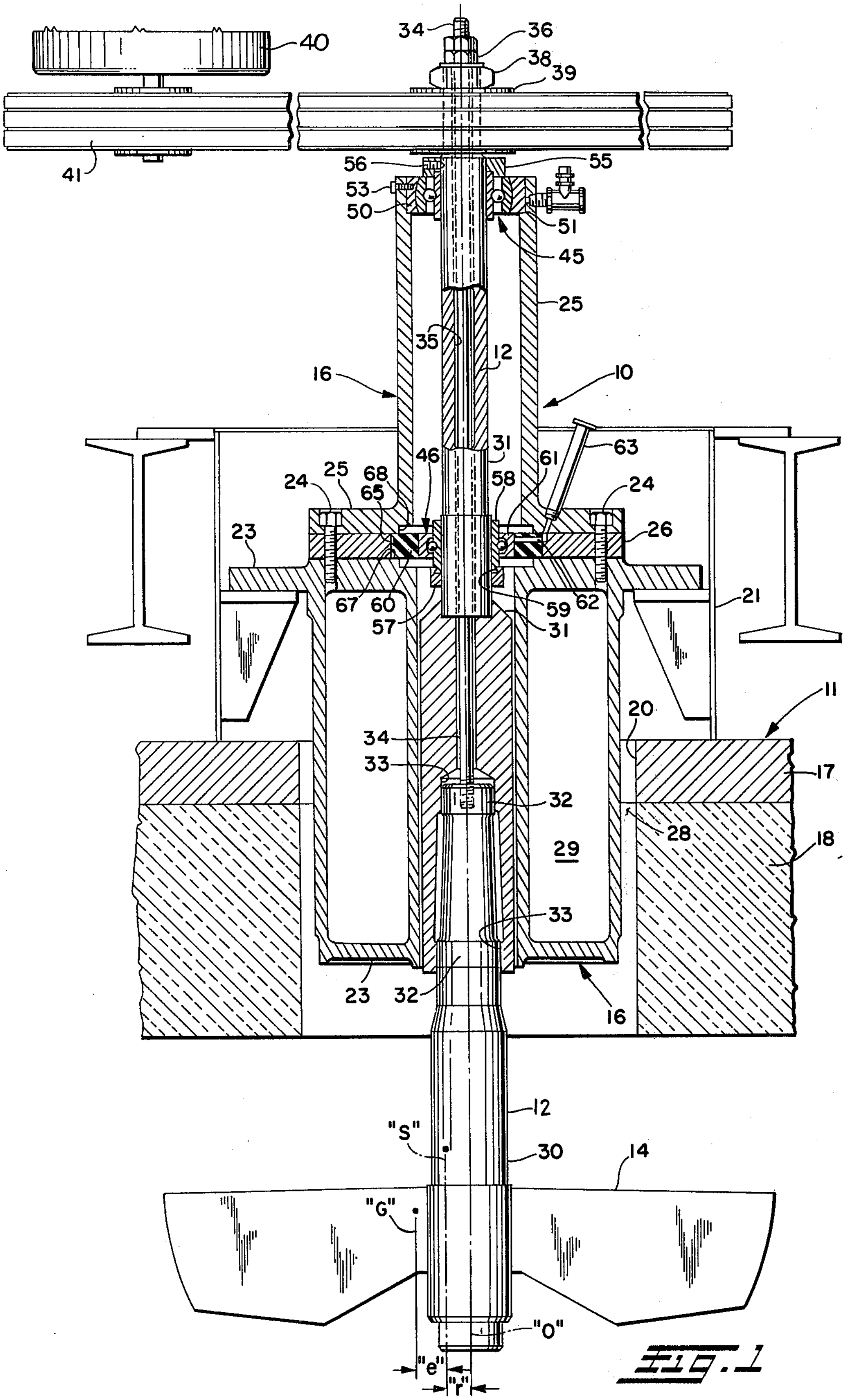


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FIG. 1

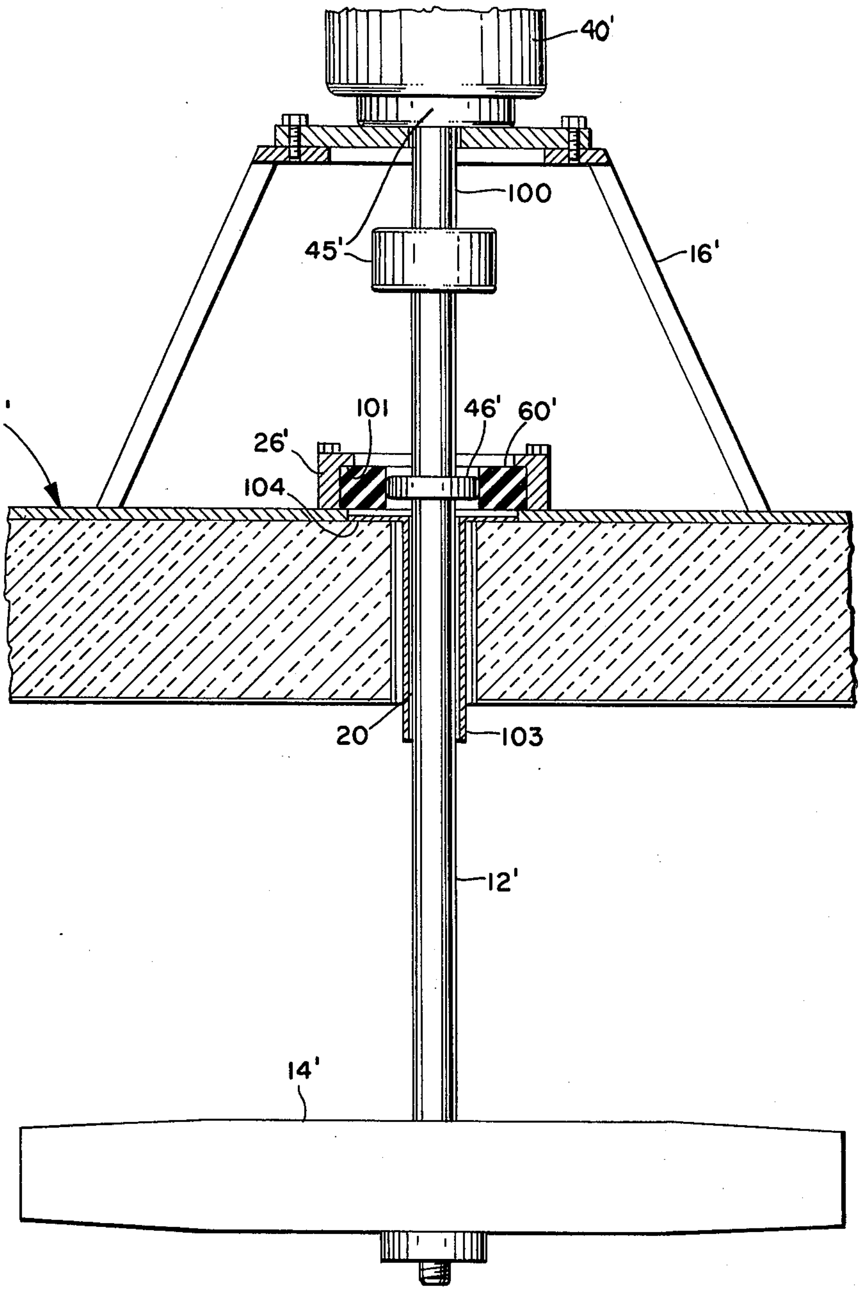

FIG. 2

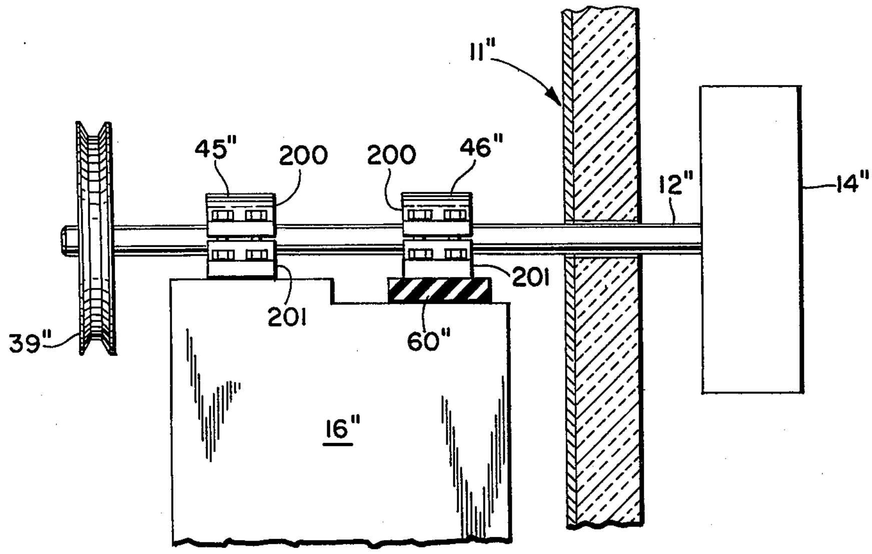

FIG. 3

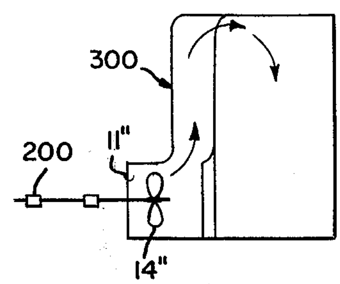

FIG. 3A

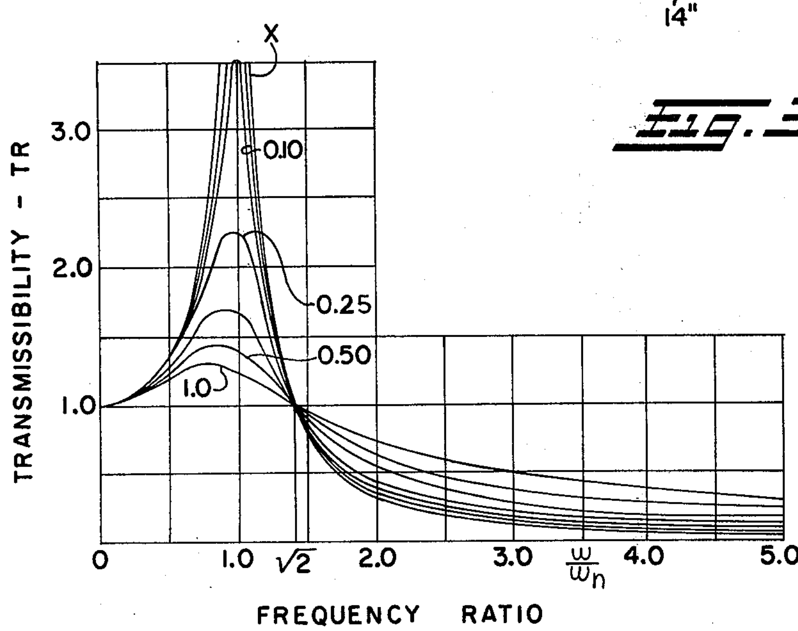

FIG. 4

FURNACE FAN ASSEMBLY

This invention relates generally to a furnace fan assembly and more specifically to the bearing support thereof.

The invention is particularly applicable to high speed furnace fans which circulate a hot gaseous medium and will be described with particular reference thereto. However it will be appreciated by those skilled in the art that the invention may have broader applications and may be applied to "low speed" fans circulating either a hot or cold gaseous medium.

High speed furnace fan assemblies basically comprise a shaft or shafts keyed to one another which have at one end a fan wheel mounted thereto integral therewith. The fan wheel may be directly positioned either vertically or horizontally within the furnace enclosure in which case the shaft extends through the furnace wall and appropriate structure, commonly referred to as a fan bung, is used to mount the assembly to the wall. Alternatively, the fan may be mounted in a duct which is in fluid communication with the furnace in which case the shaft extends through the ductwork; such arrangement being commonly referred to as a scroll-type fan. Generally, all such arrangements journal the shaft within the mounting structure by at least an outboard bearing spaced near the end of the shaft opposite the fan wheel and an inboard bearing spaced closer to the fan wheel.

As thus described, when the shaft and wheel are rotated at relatively high speeds, the fan vibrates to such an extent that damage may be done to the support structure and/or the furnace wall to which the structure is mounted and/or items such as instruments supported by the furnace wall in the vicinity of the fan. In some instances the vibration develops such high lateral forces that the shaft is permanently bent which in turn causes higher vibration forces, etc.

The prior art has recognized that such vibrations are inherent to the fan assembly and occur, basically, because the center of the mass of the fan cannot be aligned to coincide with the axial center between the bearings which journal the shaft. Attempts have been made to reduce the amplitude of the vibration by aligning and rigidizing the fan in such a manner that its critical speed, defined as the speed where maximum vibration occurs, is significantly higher than its operating speed. To accomplish this the following steps have been taken:

1. The fan wheel has been accurately mounted to the fan shaft to coincide with the center thereof and the fan shaft has been accurately aligned in an expensive aligning machine to coincide with the bearings;

2. The shaft has been significantly increased in diameter to rigidize same and expensive bearings employed to journal the shaft; and

3. Importantly, the inboard bearing is spaced as close as possible to the fan wheel to reduce the tendency of the shaft to whip.

Fan assemblies furnished in accordance with the above criteria are obviously costly to manufacture and difficult to assemble. Importantly, when the inboard bearing is moved closely adjacent the fan wheel, it becomes subjected to the hot gaseous medium within the furnace. To prevent heat failure of the bearing, an expensive water jacket surrounding the bearing is usually furnished with the fan assembly. Finally, such fan assemblies may still be suspect of failing from vibration. That is, investigations have indicated vibration will always occur because of the clearance between shaft and bearings, the tolerances encountered in fixing the fan head to the shaft and aligning same, and importantly the fact that thermal stresses produce cracks in the fan head which shift the center of mass of the fan wheel to produce an imbalance in the fan wheel assembly.

It is thus an object of the subject invention to provide a furnace fan assembly which does not produce excessive or objectionable vibrations when the fan wheel is rotated at predetermined speeds.

This object along with other features of the subject invention is achieved in a furnace fan assembly which comprises a shaft and a fan wheel on the shaft adjacent one end thereof and in contact with the hot gaseous medium within the furnace. A drive mechanism is provided to rotate the shaft and likewise the fan wheel at some predetermined speed, and the shaft and fan wheel are supported by some form of mounting structure. This structure includes a first bearing adjacent the end of the shaft opposite the fan wheel which bearing journals the shaft therein. A second bearing is also applied to the shaft in a position on the shaft between the first bearing and the fan wheel. The second bearing is affixed to a resilient mount. When the fan wheel rotates, the second bearing moves in an orbital manner relative the axis of the first bearing while the shaft rotates within the bearings. This motion of the shaft which is achieved because of the resilient bearing support substantially reduces the critical speed of the fan wheel and shaft to a value significantly below the operating speed of the fan assembly. In accordance with known vibration theory, rotating a mass above its critical speed significantly reduces the vibration amplitude which might otherwise occur and the fan assembly is thus stabilized. Furthermore, the resilient bearing support besides permitting the fan assembly to achieve a significantly low critical speed acts as a dampener against excessive vibrations when the furnace fan is accelerated past its critical speed.

In accordance with another aspect of the subject invention, the use of a resilient bearing support for the inboard bearing materially simplifies the fan assembly because (a) the inboard bearing may be located beyond the furnace enclosure to thus significantly simplify the cooling apparatus heretofore employed to shield such bearing from high heat, (b) the shaft may be reduced in diameter to effect a cost saving, and (c) the fan assembly may be manufactured and easily assembled within significantly larger tolerances than heretofore possible. Finally, imbalance of the fan resulting from thermal stress cracks will not adversely affect the life of the fan assembly.

It is thus another object of the subject invention to provide a furnace fan assembly which is simple in design, economic to manufacture and easy to assemble.

Yet another object of the subject invention is to provide a high speed furnace fan assembly for circulating a hot gaseous medium which has a greater life expectancy than heretofore possible.

The invention may take physical form in certain parts and arrangement of parts, embodiments of which will be described in detail herein and illustrated in the accompanying drawings which form a part hereof and wherein:

FIGS. 1, 2 and 3 illustrate different mounting arrangements of the subject furnace fan assembly;

FIG. 4 is a known graph indicative of vibration forces transmitted by rotating shaft assemblies; and

FIG. 3A is a schematic of the scroll fan arrangement shown in FIG. 3.

Referring now to the drawings wherein the showings are for the purpose of illustrating embodiments of the invention only and not for the purpose of limiting same, FIG. 1 illustrates a preferred embodiment of a furnace fan assembly 10 extending vertically downwardly into an insulated enclosure 11, which enclosure defines a furnace. Fan assembly 10 generally comprises a shaft 12, a fan wheel 14 adajcent one end of shaft 12 and a mounting structure 16, also known as a fan bung or cartridge unit for suspending fan wheel 14 within enclosure 11.

Insulated enclosure 11 is shown as a typical construction defined by an exterior lining 17 of sheet metal construction and a layer of refractory material 18 secured to the interior of lining 17. An opening 20 is provided in enclosure 11 for receiving a portion of mounting structure 16 and appropriate super structure indicated generally at 21 extends from lining 17 to support mounting structure 16 as by being bolted or welded thereto. The portion of insulated enclosure 11 shown in FIG. 1 is representative of a removable top wall of a furnace which may be employed for various purposes such as annealing or heat treating, carburizing, carbon baking, etc.

Mounting structure 16 generally includes a lower flanged housing portion 23 bolted as at 24 to an upper flanged housing portion 25. Lower housing portion 23 extends at least into opening 20 in close fitting relation thereto with a slight clearance 28 between lower housing portion 23 and opening 20 which may be packed with a suitable material to prevent the gaseous medium from escaping therethrough. In FIG. 1, lower housing portion 23 is illustrated as a water jacket casting having an interior chamber 29 into which a coolant is circulated through suitable pipe connections in a known manner and thus not shown or described in detail herein. While it is a specific object of the invention that a water jacket need not be employed in mounting structure 16, the embodiment shown in FIG. 1 is contemplated to be employed in furnaces at relatively high heat, typically 1700°–1800°F., and it may be desired to employ such a cooling arrangement as a matter of design choice when furnace temperature exceeds 1200°F.

Suspended within enclosure 11 by mounting structure 16 is shaft 12 and fan wheel 14. Fan wheel 14 is generally a relatively heavy casting, iron or steel, which may be secured to shaft 12, adjacent one end thereof by weld or other means. In FIG. 1, fan wheel 14 is cast with an integral shaft portion 30 extending therefrom which is secured to an upper shaft portion 31 to form a two-piece shaft 12. Shaft portions 30, 31 are joined in end-to-end aligned relation with one anotehr by means of respective interfitting male and female tapered end sections or seats 32, 33 thereon which are drawn together into tight interfitting engagement by a draw rod 34 which extends through a bore 35 in upper shaft portion 31 and is threaded into the tapered male end 32 of lower shaft portion 30. The draw rod 34 is tightened to effect a tight interfitting of the tapered end sections 32, 33 with one another, by means of a draw nut 36 threaded onto the projecting upper end of the draw rod 34. Draw nut 36 bears against a puller-thread protecting nut 38 which assures the axial position of a sheave 39 keyed to the end of shaft 12 opposite the fan wheel end. Suitable motor means comprising a motor 40 and belts 41 engaging sheave 39 is provided for rotating shaft 12 and fan wheel 14.

Shaft 12 is journaled within mounting structure 16 by a first bearing 45 closely adjacent the end of shaft opposite the fan wheel end and a second bearing 46 spaced in between the shaft ends. In FIG. 1, first bearing 45 is self-aligning ball bearing which primarily supports shaft 12 and fan wheel 14 within mounting structure 16 against vertical and lateral movement. This is accomplished by first bearing 45 (which importantly is shown as a ball bearing for reasons to be explained hereafter) which is illustrated to be of the self-aligning type and which is fixed to upper housing 25 by an annular seating ledge 51 therein. Spherical support collar 50 is suitably secured in place against rotation within seating ledge 51 by a set screw 53. A second support collar 55 secured to shaft 12 by set screw 56 prevents shaft 12 from moving axially downwardly.

Second bearing 46 is preferably a ball bearing although other types of bearings such as needle or sleeve bearings may be employed. Importantly second bearing 46 is supported in a resilient mount 60 preferably an elastomer of known high temperature synthetic material although a low temperature material, such as neoprene, may be used in certain installations. In FIG. 1, second bearing 46 is axially secured in place on upper shaft portion 31 by a retaining collar 57 fastened thereon and within which the inner race member 58 of bearing 46 has a dove-tail fit at its lower end as shown at 59. Resilient mount 60 is shown to be a rubber-type bushing which receives the outer race 61 of second bearing 46 in a press fit manner. The rubber bushing 60 may be provided with a radial passage 62 extending therethrough which communicates with a grease tube arrangement 63 for lubricating second bearing 46. Alternatively the second bearing may be of the type having prepacked lubricant therein.

The outer periphery 65 of rubber bushing 60 is securely clamped between the flanges of the lower and upper housing portions 23, 25. This may be accomplished by a number of different arrangements and is easily accomplished by providing the upper housing portion 25 with an integral, annular retainer portion 26 at its flanged end which is shown indented at 68 at its interior. Retainer portion 26 has a thickness slightly less than resilient bushing 60 and a boss 67 is provided at the flange end of lower housing portion 23. Boss 67 and indention 68 extend down the end flat faces of rubber bushing 60 a slight distance (⅛–¼inches) so that rubber bushing 60 is slightly compressed and thus secured along its outer periphery and edges when housing portions 23, 25 are bolted together as at 24. With the resilient mount 60 thus secured between a fixed framework and second bearing 46, shaft 12 becomes yieldably supported in a cushioned manner for purposes to be explained hereafter.

The furnace fan assembly 10′ shown in FIG. 2 is similar to that of FIG. 1 and like parts will be identified by like numbers followed by a prime (′) where applicable and will not be described in further detail herein. The arrangement illustrated in FIG. 2 is especially suitable for use with large size fans wherein fan shaft 12′ is coupled at its upper end through a flexible or universal joint-type coupling connected directly to the shaft **100** of motor **40'** and at its bottom end connected directly to fan shaft **12'**. The bearings (not shown) in the motor **40'** rotatably secure motor shaft **100** and the universal joint permits skewed alignment of fan shaft **12'** with motor shaft **100.** Thus the universal joint and the fixed bearings in motor **40'** are herein designated as the first bearing **45'** as it should be viewed as the functional equivalent of the first bearing **45** shown in FIG. **1.** The resilient mount **60'** is to that of FIG. **1** and is held compressed to a fixed support by an annular spacer-retainer **26'** to firmly grip the outer edges and outer periphery of the rubber-type bushing. Spacer-retainer **26'** should be viewed as the functional equivalent of retainer portion **26.** This arrangement includes a radiation shield **103** in place of the water jacket of FIG. **1.** Shield **103** may comprise a tube which receives shaft **12'** in close fitting relationship and has a flanged end **104** for mounting purposes. If desired, the space between shield **103** and opening **20'** in enclosure **11'** may be filled with suitable packing material in a known manner for cooling purposes. Alternatively, a plurality of radiation shields may surround the shaft with or without insulation therebetween as known to those skilled in the art.

FIG. **3** illustrates still another embodiment of the furnace fan assembly **10''** of the subject invention which is similar to that shown in FIGS. **1** and **2** and like parts will be identified by like numbers followed by a double prime ('') and will not be described in further detail herein. The furnace fan assembly **10''** in FIG. **3** is especially suitable for use with large scroll-type fan arrangements known to those skilled in the art and schematically illustrated as **300** in FIG. **3A** wherein the fan extends horizontally through a duct or insulated enclosure **11''** which is in communication with the furnace. In this instance the mounting structure **16''** consists of a solid stepped base for supporting first and second bearings **45''**, **46''**. Bearings **45''**, **46''** may comprise ball bearings, each of which is mounted into half-moon supports **200, 201** which in turn are secured to the mounting structure **16''**. More particularly, the first bearing **45''** would comprise a self-aligning bearing similar to that shown in FIG. **1** and the second bearing **46''** would be affixed to resilient mount **60''** which would simply comprise a block of resilient material. Alternatively, first and second bearings **45''**, **46''** could comprise known pillow block bearings.

An understanding of the operation of the furnace fan assemblies illustrated may best be had by a brief explanation of the fundamental theories of vibration which are believed applicable hereto. In accordance with known vibration anaylsis of whirling shafts, the center of gravity G of the fan wheel is displaced a radial distance e from the centerline of the shaft S which in turn is displaced a radial distance r from the fixed bearing's center O when the shaft rotates. This is shown in FIG. **1** in exaggerated form for explanatory purposes only. When the fan assembly begins to rotate, the force through G (i.e., a centrifugal force) is radially outwardly and is maintained in equilibrium by the inherent stiffness of the shaft. (The force exerted by the shaft mass through its center of gravity will not be considered for ease of explanation.) This force through G increases in a radially outwardly direction as the rotational speed ω of the shaft increases until the critical speed of the shaft and wheel is reached. The critical speed is defined to occur when the rotational speed of the fan wheel and shaft reaches the natural frequency of lateral vibration of the fan assembly defined as ω_7 which in turn is a function of the rigidity of the shaft and wheel and the mass thereof. When the critical speed of the assembly is exceeded the force through the center of gravity G is directed radially inwardly and the vibrations or amplitude thereof are reduced. This may best be shown by reference to the known vibration graph shown in FIG. **4** where the rotational speed ω of the fan assembly is plotted as a function of ω_7 on the horizontal axis and the amplitude of vibration is expressed as a known force transmissibility ratio TR. The family of curves illustrated represents systems with various degrees of dampening and the steepest curve X shown, peaking at infinity, represents a theoretically undampened system.

Applying the above analysis to prior art assemblies which may be characterized as having a rigid, fixed second bearing support, calculations verify the occurrence of service failures in that the shaft will permanently bend in creep action. Such calculations are based on a fan wheel weight of 100 lbs., a typical rotating speed of 1200 rpm, a center of gravity displacement e of only 0.030 and a reasonable lateral force magnification factor of 4. These values are believed to be representative of actual service conditions of prior art fans. For example, the 0.030 inch dimension quoted, which is a static dimension, may easily occur when it is considered that such dimension is the cumulative total of bearing clearance, shaft internal taper alignment, initial imbalance, cracks in the fan wheel and shaft straightness. Once the shaft bends, the lateral or centrifugal force exerted by the fan wheel increases rapidly resulting in damaging vibrations transmitted to the mounting structure, the insulated enclosure, etc. Fans have been removed from service with their shafts permanently bent as much as ¼ inches or more in some cases.

Furthermore, as noted hereinbefore, prior art attempts at reducing vibrations in furnace fans were directed toward shifting the operating speeds of the fan significantly to the left of the $\omega/\omega_7 = 1$ value shown in the graph of FIG. **4.** This was accomplished by increasing the natural frequency of resonance of the fan assembly which in turn was physically achieved by rigidizing the fan assembly. More particularly, the shaft diameter was considerably increased to at least 3 inches and as much as 6–7 inches, heavy duty bearings were employed and importantly the forward rigid bearing was moved as close as possible to the fan wheel to reduce the "beam length" and correspondingly the deflection of the shaft from the fan wheel. Furthermore, painstaking steps were taken to align the shaft within the bearings and the fan wheel to the shaft. All of these steps significantly increased the cost of such fans and required that associated hardware such as water jackets be supplied to fans furnished as a matter of course. While such steps did improve the operation of the fan, excessive fan vibrations have still occurred under some conditions. These conditions which may, in time, cause escessive fan vibrations occur under extremely high fan speeds approaching 1600–1700 rpm and at high temperatures because, among other things, thermal stresses will produce cracks in the fan wheel which in turn will cause the center of gravity of the fan wheel to shift. Thermal stress cracks are believed to become a significant problem when furnace temperature exceeds 1400°F.

In accordance with the subject invention, the fan shaft and wheel are yieldably instead of rigidly supported by virtue of the resilient mount **60** which is shown in each of the embodiments illustrated. This resilient mounting is believed to accomplish two important functions. First, the resilient mount **60** permits the critical speed of the shaft **12** and fan wheel **14** to be considerably reduced to a value typically between 200 and 400 rpm. That is, shaft **12** rotates within the bearings **45, 46** while the fan wheel and shaft are orbiting about or relative to the first bearing **45.** This orbiting path which the shaft and fan wheel assume may be viewed as a cone with its base defined by the movement of the fan wheel and the cone apex at or approximately near the center of first bearing **45.**

It should be noted in connection with the movement of the shaft and fan wheel that all embodiments show first bearing **45** to be a ball bearing which is an ideal bearing for permitting such cone shaped movement. Obviously, the rigidity of the system of the subject invention is considerably reduced from that of the prior art explained above the critical speed significantly decreased. In accordance with the above analysis, when the critical speed is exceeded the force acting through the center of gravity of the fan wheel is directed inwardly with the result that the orbital path of shaft and fan wheel is significantly reduced to stabilize the system. In terms of design criteria this occurs when ω/ω_7 is greater than the square root of 2 and may be expressed in terms of operating speeds of the embodiments illustrated as an operating speed of at least 800–900 rpm.

The second feature of the resilient mount **60** is that it provides a dampener to the fan assembly which permits the shaft and fan wheel to pass through critical speed without imparting excessive vibrations to the mounting structure **16** and/or insulated enclosure **11.** The dampening factor of the resilient mount as same relates to the graph shown in FIG. **4** is not exactly known but is believed to be a function of the position of the mount on a shaft, the hardness of the resilient mount and the dimensions thereof. Fans manufactured in accordance with the invention have utilized rubber bushings **60** having a hardness between durometer readings of 40–70 which were mounted outboard of enclosure **11** and approximately 1–½ inches in thickness and depth. Such fans have not exhibited excessive vibrations when passing through the critical speed. In this connection it is believed that because the critical speed of the fan assembly is very low, there is rapid acceleration past the critical speed at a time when the mass centrifugal force is at relatively low value.

As a direct result of the motion characteristics thus imparted to the fan wheel and shaft, several design modifications may be made to the fan assembly. These include: (1) a reduction in shaft diameter to typical diameters of 1½ to 3 inches; (2) elimination of tedious assembly and alignment problems because the first bearing **45,** whether in the form of a self-aligning bearing or universal joint, permits easy and quick assembly of the unit; (3) the elimination, if desired, of the water cooling jacket because the second bearing **46** can now be mounted beyond the insulated enclosure **11**; and (4) the ability of the fan assembly **10** to sustain thermal cracks in the fan wheel as the occurrence of such cracks will increase vibration but not adversely affect fan operation above critical. In connection with the last point mentioned, it should be noted that the subject invention is directed to only the first or lowest critical speed of the assembly. It is not known if there are other higher critical speeds in the embodiments described since fan operation at speeds approaching upper limits of 1700–1800 rpm has not exhibited the presence of any higher critical speeds.

The invention has been described with reference to a preferred embodiment. Obviously, modifications and alterations will occur to others upon reading and understanding the specification. It is my intention to include all such modifications insofar as they come within the scope of the present invention.

It is thus the essence of the invention to provide a furnace fan assembly having a resiliently supported fan wheel and shaft which permits the shaft and fan wheel to be rotated at predetermined speeds without experiencing objectionable vibrations.

Having thus disclosed the subject invention, I claim:

1. A furnace fan assembly for circulating a hot gaseous medium in a furnace, said assembly comprising:
- a shaft;
- a fan wheel on said shaft adjacent one end thereof and in contact with said hot gaseous medium;
- motor means for rotating said shaft and fan wheel from rest to a predetermined speed above critical; and
- means for supporting said shaft and said fan wheel and permitting said shaft and fan wheel to pass through a critical speed between 200–400 rpm, said means including
- first bearing means including a self-aligning type bearing adjacent the end of said shaft opposite said fan wheel for journaling said shaft in a manner which permits said shaft to rotate in a conical path with the apex of said conical path at the center of said bearing,
- a second bearing receiving said shaft and positioned on said shaft between said first bearing and said fan wheel, and
- resilient means supporting said second bearing for permitting orbital movement of said second bearing relative the axis of said first bearing while said shaft rotates within said second bearing, said resilient means also dampening vibration of said shaft and said fan wheel while same orbit about said first bearing.

2. The furnace fan assembly of claim **1,** wherein said first bearing is defined as including said motor means rotatably driving a motor shaft and a universal joint directly driven by said motor shaft and driving said fan shaft.

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UNITED STATES PATENT AND TRADEMARK OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 3,910,717

DATED : October 7, 1975

INVENTOR(S) : William L. Thome

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 3, line 23, change "super structure" to -- superstructure --. Line 58, change "anotehr" to -- another --.

Column 4, line 11, after "is" insert -- a --.

Column 5, line 10, before "to" insert -- similar --. Line 12, change the words "to firmly grip" to -- which firmly grips --.

Column 6, line 61, change "escessive" to – excessive --.

Column 7, line 24, after "above" insert -- and --. Line 48, change "1-1/2" to -- 1-1 1/2 --.

Signed and Sealed this

thirtieth Day of *December 1975*

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks