roller itself. Although bearings from 0.25 in. (6.3 mm) shaft diameter to 6.0 in. (152 mm) shaft diameter have been tested successfully, there is a practical limit to the smallest size of hollow rollers. Rollers of less than about 0.30 in. (8 mm) diameter tend to become overstressed rapidly with deflection, and are not suitable for bearings manufactured to standard tolerances. However, there appears to be no limit to the maximum size of hollow rollers. Large rollers can deflect under load with only small increases in bending stress.

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DISCUSSION

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The authors have provided an extensive description of an interesting roller bearing concept that they have developed to the point where successful application has been realized in the machine tool industry. Though not yet employed in aircraft gas turbines the interest is growing because of increased awareness of some of the favorable characteristics of the hollow roller bearing design. Some of the advantages are readily apparent, especially in light of the higher DN levels that future engine designs will dictate. Obviously, one advantage results from the reduced mass of the hollow rollers which lessen the centrifugal load levels on the outer race and favorably influence rolling contact fatigue life. Also, the reduced mass of the hollow roller would decrease the dynamically induced impact loads occurring at the interface between the roller ends and the inner race guide flange. This would result in reduced roller end wear adding to the durability of this design. In a conventional bearing, roller and wear can progress to the point where roller guidance is no longer provided by the inner race guide flanges. The cage cross bars then experience wear which can proceed to the point of fracture making the bearing inoperable. The hollow roller bearing design described by the authors has no such cage and thus would be free of this failure mode. The hollow roller bearing described by the authors has another advantage arising from the method of preload. The uniform circumferential stiffness provided by the slight radial interference at each roller location could provide a less vibration prone engine design. Conventionally preloaded solid roller bearings present a highly variable circumferential stiffness distribution that is a natural result of their two point or three point preload system. This can produce regions of low stiffness around the bearing circumference which under some circumstances would result in engine vibration beyond acceptable limits. This same uniform preload provided by the authors' hollow roller bearing design could also have a favorable effect on engine fuel consumption. The uniform preload and stiffness can potentially reduce engine rotor radial runout. This reduced radial movement of the rotor would then allow tighter radial clearances to be employed in the seals located at the interface between the rotating and stationary components along the engine gas path. The improved sealing resulting from the tighter clearances would reduce leakage which has a direct favorable effect on engine fuel efficiency.

The list of advantages of the authors' hollow roller bearing design is impressive. However, to be considered for use in an application as critical as supporting the main shaft of an aircraft gas turbine engine these advantages must be closely scrutinized and other features or drawbacks of the design fully examined. In light of this need the following comments and questions are offered for consideration by the authors.

The authors claim that cage rubbing as a source of heat generation is eliminated in their cageless design. This certainly cannot be contested, however, isn't this gain at least partially reduced by the fact that the rollers now must rub against each other since there is no cage to separate them? The relative speed between the rollers at their contacting interface is higher than at any cage/roller/raceway interface in the conventional bearing. Wouldn't this higher relative speed increase that portion of the frictional losses? In this regard, have the authors ever run a test, or are they planning a test, to provide a head to head comparison of the heat generation and operating temperature levels of their hollow roller bearing design versus a conventional bearing of the same size and quality classification?

The authors describe a temperature drop phenomena as the bearing operating speed is increased beyond a certain level in tests of both 80 mm and 14 mm bore size bearings. It is conceivable that this reduction occurred at least in part because of improved oil drainage from the rig sump and bearing cavity due simply to the increased speed. This improvement would not be realized continually as the speed is further increased because turbulence is likely to dominate the flow pattern at the higher rpm levels causing losses to again ascend. In addition to this temperature reduction phenomena did the authors observe a corresponding reduction in heat generation?

On the subject of conventional high precision bearings, the authors make the statement that in such a bearing the rollers are guided by the retainer. This is generally not true in that roller guidance is provided by guide rails or flanges normally attached to the inner race. The retainer has two functions, one is for roller retention during installation and the other is for separation of the rollers during operation.

The authors did not mention any specific method used for retaining the rollers in their cageless hollow roller bearing as would be required during the installation process. Is there any specific technique that they have developed for installation of their bearing that would minimize the chance for loss of a roller?

The authors make the point that there is a certain running-in period during the first few hours of operation of their bearing which produces a higher frictional torque than what is required for a bearing that has been running for a while. The claim is made that "certain changes become apparent in the texture of the rolling contact surfaces" after the bearing has been "run-in." Do the authors have any quantifiable evidence to substantiate this claim such as surface finish and waviness measurements or even SEM or TEM microphotographs?

In discussing the load-life characteristics of the hollow roller bearing the authors make the claim that the "roller fatigue life is related to the bending stress to the ninth power". Is this a mis-statement in that it was intended to say that the "roller fatigue life is related to the *contact* stress to the ninth power"? If it is not a mis-statement then what data can be proferred to substantiate the claim that life varies with bending stress to the ninth power?

The final question deals with the lubrication system. The authors did not offer any details as to how the lubricating oil was introduced to the bearing interior. Exactly how was this accomplished? There is evidence in some of the bearing photos of axial bore slots in the inner race. Are these slots used as oil passages? If so, does all the oil pass completely under the bearing through these slots or is some bled off through intersecting holes to direct some flow to the bearing interior for lubrication?

It will be necessary that satisfactory answers to these and related questions be provided before serious consideration could be given to the development of this interesting bearing for a specific application in an aircraft gas turbine engine.

Authors' Closure

The authors wish to thank the discussers sincerely for their useful comments which provide an insight into the considerations for the potential applications of the hollow roller bearing in aircraft gas turbines.

While this paper does touch on the subject of high speed bearings briefly, it was mainly intended to illustrate the use of hollow roller bearings for machine tool applications where accuracy and stiffness are the primary considerations. However, we are pleased to answer the questions presented by the discussers which are directed toward the high speed characteristic of this bearing.

The absence of the cage has not resulted in roller to roller rubbing problems because the preload maintains control of the rollers and reduces sliding to a minimum. It has been observed stroboscopically that the hollow rollers tend to space themselves apart as the bearing rotates. A roller passing through the load zone becomes more deflected and forces adjacent rollers to make room for it. As the rollers progress around the bearing, a gap is maintained between each roller, allowing an adequate oil film on the roller surface. This action takes place within a few revolutions of the bearing and roller to roller gaps of 0.002/0.004 in. are usually measured.

In regard to heat generation in the hollow roller bearing, it is believed that the constant flexing of the roller causes hysteresis losses. However, any heat generated seemed to be dispelled by the increased surface area from the hollow bore of the rollers. We have found that the hollow roller bearing generates the same amount of heat or slightly less than a solid roller bearing of similar size. In one particular test of a machine tool spindle, hollow roller bearings were substituted for the ball bearings previously used. Permitted temperature rise was 25°F @ 1000 rpm, using grease lubricant. The first test exceeded this figure by 5°F. The bearings were modified by increasing the roller hollowness with a resulting temperature rise of less than the 25°F allowed, however, the hollow roller bearing did not run as cool as the original ball bearing.

The temperature drop phenomena referred to in the paper was observed in three different sized test bearings. Each was tested in a different rig, but air-oil mist was used for lubrication in each case. There was no oil sump, the mist being allowed to escape after passing through the bearing. As the bearing speed was increasing, the temperature dropped momentarily and then continued rising at the same rate as before. We believe the oil-mist provided only marginal lubrication so that for each bearing design, preload, etc., there may be a certain speed at which the rollers suddenly develop a complete oil film on the raceway surface, producing the temperature drop. Heat generation in the test bearings was not measured, only the outer race temperature was monitored.

Roller guidance in conventional bearings is usually accomplished by one of two ways: either with the cage, or the race flanges. Most Torrington bearings, including our needle bearings, heavy duty roller bearings, thrust bearings, and spherical bearings all use the cage to both guide and contain the rollers. Generally, these bearings have rollers whose length is several times their diameter and flange guidance is not suitable. Therefore, it seemed natural to us to mention that rollers are cage guided in conventional bearings.

Presently, two methods of retaining the rollers are used during bearing assembly. Grease is the most popular method of holding the rollers into the outer race until the inner race is installed. It is simple, effective, and readily dissolves when oil lubricant enters the bearing. Sometimes, a dummy sleeve is used to hold the rollers in place. This sleeve is pushed out by the inner race during assembly. However, it can be difficult to remove the sleeve, particularly if it is inside the machine housing, and for that reason this method is not often used. Other devices, such as a finger type cage which would enter each hollow roller to hold it in place against the race, have been discussed but no actual designs have been considered as yet.

Running-in of a preloaded hollow roller bearing takes place in the first 10 to 40 hours of operation. The time required depending upon the speed of rotation. A surface change does take place during this run-in period. The ground surface of a bearing raceway consists of many small, sharp peaks and valleys. As these sharp peaks are flattened down, rolling friction is reduced and an oil film is maintained to end any further wear. Surface roughness (Ra) measurements show a reduction of 3 to 4 microin. in the roller path of a used bearing race. While the ground surface of the race measured 10.5 microin., the roller path adjacent to it measured only 7 microin.

Bearing life is determined by the amount and number of stress cycles that can be endured. Whether the contact stress or the bending stress is the limiting factor, the life calculations are the same. Because the roller is constantly flexing, its bending fatigue life determines the bearing life. The surface contact stresses are low and therefore have a longer fatigue life. We refer you to the *Fatigue Design Handbook*, published by SAE in 1968, pages 23–27. Fatigue strength exponents are given for bending stress and vary from the 8th power to as high as the 20th power. There is no absolute number that we know of today. However, we believe the 9th power exponent is conservative, although our fatigue test data indicates that it may be possible to raise this number safely.

Finally, coming to the lubrication systems, we find that the hollow roller bearing works best with a continuous supply of lubricant. It can be introduced either through the outer race or the inner race, depending upon the bearing design and application. In either case, we prefer to inject the oil directly to the roller complement right at the middle of the raceway, thus assuring that the lubricant reaches the raceway contact surface and is spread out by the passing rollers forming a uniform oil film. Even though the oil hole is in the raceway and subject to contact stresses, we have never experienced surface spalling associated with the oil hole.

In the case of the turbine test bearing, the oil is introduced via axial grooves in the bearing bore through which most of the oil flows for cooling purposes. Every fourth groove contains a small radial hole (six in all) to carry oil centrifugally to the roller complement.