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(71)出願人　503082262
ザ ティムケン カンパニー
THE TIMKEN COMPANY
アメリカ合衆国，４４７０６－０９３０ オハイオ州，キャントン，エス．ダブリュ．，デューバー アヴェニュー １８３５
１８３５ Dueber Avenue, S.W., Canton, OH 44706－0930, United States of America
(74)代理人　100104499
弁理士　岸本　達人
(74)代理人　100101203
弁理士　山下　昭彦

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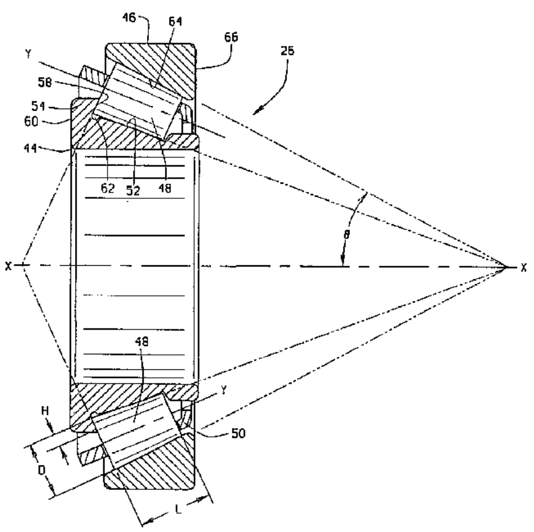
(54)【発明の名称】低磨耗性かつ出力低損失性の軸受

(57)【要約】
車両用差動装置（Ａ）のピニオンを支持するのに非常に適した円錐ころ軸受（２６、２８）であって、クラウニングされた対向する軌道面（５２、６４）を有するコーン（４４）とカップ（４６）を備えている。コーンは、軌道面の大端部に、つば面（５８）を有するスラストつば（５４）を備えている。さらに、前記軸受はクラウニングされた側面（７０）と球面である大端部（７２）をそれぞれ有する円錐ころ（４８）を有する。前記ころ（４８）は、そのクラウニングされた側面に沿って軌道面（５２、６４）と接触し、その球状端部面に沿ってつば面と接触する。ころの長さ（Ｃ）と大端部の直径（Ｄ）の比は、１．５より小さい。軌道面（５２、６４）のクラウニングところ側面（７０）のクラウニングにより、端部逃げの総量が７００μｉｎ．／ｉｎｃｈから１５００μｉｎ．／ｉｎｃｈの間となる。側面（７０）と軌道面（５２、６４）間の接触の中心は、つば面（５８）方向に補正されている。つば面（５８）の高さは、ころ（４８）の大端部（７２）の直径の３０%～４５%である。ころの球状大端部面（５２）の半径は、ころ頂点長さ



【特許請求の範囲】
【請求項１】
車両の差動装置における、軸受軸を中心とする回転を提供する円錐ころ軸受であって、前記軸受は前記軸に対して外側方向に向けられた円錐状の軌道面を備えた内輪軌道と、該内輪軌道を取り囲んでおり、且つ前記軸に対して内側方向に向けられ且つ前記内輪軌道の軌道面に向けられているとともに前記軸に対して２０°から３０°の間の角度を成している円錐状の軌道面を備えた外輪軌道と、一つの前記軌道上にあり、該軌道に対する軌道面の大端部に位置し、前記軌道面と実質的な角度を成すつば面を有するスラストつばと、前記軌道間に環状の列に配置され、それぞれが前記内輪軌道と前記外輪軌道の前記各円錐状の軌道面に沿って接触する側面と、前記スラストつばのつば面に沿って接触する大面を有する円錐ころと、を有し、前記つば面が前記軌道面によって形成される包絡面から離れる方向に、且つ、その位置している位置において、前記ころの大端部の直径の３０％以上の距離に延在していることを特徴とする円錐ころ軸受。
【請求項２】
各ころの前記側面が、おおよそ、頂点を軸受の軸上に有する円錐状包絡面内にあり、前記大端部面が、その中心を前記ころの軸上に有する球状包絡面内にあり、その半径が少なくとも前記ころ包絡面の頂点と前記大端部面の前記球状包絡面間の距離の少なくとも９０％であることを特徴とする請求項１に記載の軸受。
【請求項３】
前記つば面が、頂点を軸受軸上に有する円錐状包絡面内にあることを特徴とする請求項１に記載の軸受。
【請求項４】
前記各ころの大端部面が、接触の領域に沿って前記つば面に接触していて、前記接触の領域の中心が前記つば面に至る前記軌道面の包絡線から０．０４インチ以下であることを特徴とする請求項３に記載の軸受。
【請求項５】
前記軌道面又は前記ころの側面、又はそれら両方がクラウニングされ、前記ころの側面と前記軌道面との間の接触の領域の中心が前記ころの前記大端部方向へ補正されていることを特徴とする請求項１に記載の軸受。
【請求項６】
前記つば面と前記ころの大端部面の算術平均粗さが４μｉｎ．を超えないことを特徴とする請求項１に記載の軸受。
【請求項７】
各ころの前記大端部面の振れが５０μｉｎ．を超えないことを特徴とする請求項１に記載の軸受。
【請求項８】
差動装置が、ハウジングと、ハウジング内部で軸受軸を規定するヘッド軸受とテール軸受と、軸受軸を中心として回転するように軸受内に実装されたピニオンシャフトと、ピニオンシャフト上且つハウジング内部のピニオンと、ハウジング内部にあってピニオンと噛み合うリングギアとを含んで構成され、ピニオンシャフトの回転がリングギアを回転させる車両の差動装置において、前記ヘッド軸受が、ピニオンシャフトに取り付けられており、且つ、前記軸から外側方向に向けられている円錐状の軌道面を備えたコーンと、前記軌道面の大端部に位置するつば面を備え、前記軌道面と実質的な角度を成すスラストつばと、前記軸方向に向けて内側方向に向けられ且つ前記コーンの軌道面に向けられているとともに前記軸に関して２０°から３０°の間の角度を成している円錐状の軌道面を備えたカップと、前記コーンとカップの間に列状に配置され、前記各円錐状の軌道面にそって接触する側面と、前記スラストつばのつば面にそって接触する大端部面を有し、さらに、長さと大端部面での半径との比率が約１．５を超えない円錐ころとを含んで構成され、前記軌道面と前記ころの側面が、軌道面の両端部で逃げができるようにクラウニングされ、前記逃げは７００μｉｎ．／ｉｎｃｈから１５００μｉｎ.／ｉｎｃｈの間であり、前記つば面

が中心を前記軸上に有する円錐状包絡面内にあり、前記各ころの側面が中心を前記軸上に有する円錐状包絡面内にあり、前記各ころの大端部面が、中心を前記ころの軸上に有する球状包絡面内にあり、その半径が、前記側面によって規定される包絡面の頂点と、前記端部面の前記球状包絡線間の、前記ころの軸に沿って測定した距離の９０％より大きいことを特徴とする差動装置。

【請求項９】

前記つば面が、前記コーン軌道面によって形成される包絡面から離れる方向に向かって、前記ころの大端部の直径の３０％以上の距離に延在していることを特徴とする請求項１１に記載の差動装置。

【請求項１０】

各ころの前記大端部が、接触の領域に沿って前記つば面に接触していて、前記接触の領域の中心が前記コーン軌道面の包絡線から０．０４ｉｎ．以下であることを特徴とする請求項１１に記載の差動装置。

【請求項１１】

前記軌道面又は前記ころの側面、又はそれら両方がクラウニングされ、前記ころの側面と軌道面との間の接触の領域の中心が前記ころの前記大端部方向へ補正されていることを特徴とする請求項１１に記載の軸受。

【請求項１２】

一方における、ころの前記側面と前記コーン軌道面の間のずらした調整量と、他方における、前記ころの側面と前記カップ軌道面の間のずらした調整量の合計は、約０．０００３から０．０００２ラジアンの間であることを特徴とする請求項１４に記載の差動装置。

【請求項１３】

前記テール軸受が前記ヘッド軸受と同じ要素で構成され、且つ、それらの軸受が対向して実装され、予圧の状態に設定されることを特徴とする請求項１５に記載の差動装置。

【請求項１４】

前記つば面と前記ころの大端部面の算術平均粗さが４μｉｎ．を超えないことを特徴とする請求項１に記載の軸受。

【請求項１５】

各ころの前記大端部面の振れが５０μｉｎ．を超えないことを特徴とする請求項１に記載の軸受。

【請求項１６】

車両の差動装置において、軸受軸を中心にして回転させる円錐ころ軸受であって、該円錐ころ軸受は、前記軸から外側方向に向けられている円錐状の軌道面と、該円錐状の軌道面の大端部にスラストつばとを備えたコーンであって、前記円錐状の軌道面はクラウニングされており、且つ、頂点を前記軸上に有する円錐状包絡面内におおよそ位置しており、前記円錐状の軌道面から至る前記スラストつばは、軌道面に対して実質的な角度を成しているコーンと、前記コーンを取り囲み、さらに、前記軸方向に向けて内側方向に向き且つ前記軸に関して２０°から３０°の間の角度を成している円錐状の軌道面を備えたカップであって、該カップ軌道面はクラウニングされているとともに、およそ前記軸上で前記コーン軌道面の包絡面の頂点と大体同じ位置にある頂点を有する円錐状包絡面内にあるカップと、前記コーンとカップの間に環状の列に配置され、ころ軸と、前記軌道面に沿って接触する円錐状の側面と、前記つば面に沿って接触する大端部面をそれぞれ備えた円錐ころであって、前記側面はクラウニングされており、前記大端部面が中心を前記ころ軸上に有する球状包絡面内にあり、前記軌道面と前記各ころ側面は該側面と該軌道面の接触の領域の中心がつば面方向に補正されるようにクラウニングされている円錐ころと、を備えており、前記ころの側面と前記コーン軌道面の間のずらした調整量、及び前記ころの側面と前記カップ軌道面の間のずらした調整量の合計は、約０．０００３から０．０００２ラジアンの間であることを特徴とする円錐ころ軸受。

【請求項１７】

前記つば面が、頂点を軸受軸上に有する円錐状包絡面内にあることを特徴とする請求項１

７に記載の軸受。
【請求項１８】
前記ころの長さと前記ころの大端部での直径との比率が約１．５を超えないことを特徴とする請求項１６に記載の軸受。
【請求項１９】
前記軌道面と前記ころの側面が、前記軌道面の両端部で逃げができるようにクラウニングされ、前記逃げは７００μｉｎ．／ｉｎｃｈから１５００μｉｎ．／ｉｎｃｈの間であることを特徴とする請求項１６に記載の軸受。
【請求項２０】
前記つば面と前記ころの大端部面の算術平均粗さが４μｉｎ．を超えないことを特徴とする請求項１６に記載の軸受。
【請求項２１】
前記カップ軌道面が、内側方向に前記軸方向に向けられ且つ前記コーンの軌道面に向けられているとともに前記軸に対して２０°から３０°の間の角度を成していて、前記スラストつばのつば面が、当該スラストつばが位置している前記軌道面によって形成される包絡面から離れる方向に向かって、且つ、前記ころの大端部の直径の３０％以上の距離に延在していることを特徴とする請求項１６に記載の軸受。
【請求項２２】
各ころの前記大端部面が、接触の領域に沿って前記つば面に接触していて、前記接触の領域の中心が前記つば面に至る前記軌道面の包絡面から０．０４インチ以下であることを特徴とする請求項１６に記載の軸受。
【請求項２３】
各ころの前記大端部面の振れが５０μｉｎ．を超えないことを特徴とする請求項１６に記載の軸受。
【請求項２４】
各ころの側面が頂点を前記軸受軸上に有する円錐状包絡面内におおよそ位置し、前記大端部面が球状包絡面内にあり、その球状包絡面の半径が、前記ころ包絡面の頂点と前記大端部面の前記球状包絡面間の距離の少なくとも９０％であることを特徴とする請求項１６に記載の軸受。
【発明の詳細な説明】
【技術分野】
【０００１】
技術分野
本発明は、一般的に円錐ころ軸受に関し、より詳細には、低磨耗性かつ出力低損失性の円錐ころ軸受に関する。
【背景技術】
【０００２】
背景技術
車両の差動装置の歯車装置は、騒音と磨耗を最小限に保つために非常に正確に駆動しなければならない。典型的には、そのような歯車装置は、差動装置のハウジング内で共に回転するリングギアとピニオンを含んでなっている。ピニオンは、リングギアと噛み合うことで、ドライブシャフトからリングギアへとトルクを伝え、そして次にリングギアは、前記ハウジングから延在しているアクスルシャフトへとトルクを伝える。両歯車はらせん状の歯を持っている。すなわち、それらはハイポイドギアである。
【０００３】
ピニオンは非常に安定に回転しなければならない、言い換えれば、決まった一定の軸位置で、固定された軸の周りを回転しなければならない。この目的のために、ピニオンは、差動装置のハウジング内に対向して予圧の状態で実装されて取り付けられた一対の円錐ころ軸受中で回転するシャフト上に支持される。この配置は、ー少なくとも最初はー、必要とされる安定性を与えるが、最近用いられている円錐ころ軸受は、軸受自身中での摩擦や、

潤滑剤の掻き乱しを克服するために、ある程度のトルクを必要としている。このトルクは出力を消費する。その上、やがて軸受は磨耗し、それにより安定性を損なうので、ピニオンはリングギアと最も効率的に噛み合う位置から移動してしまう。

【０００４】

いくつかの要因が、円錐ころ軸受により要求されるこのトルクと、軸受が受ける磨耗に対して影響を与える。例えば、ころは、予圧の下、軌道面上に対して小角度をもって軌道面間にしっかりとはめられているが、そのために軸受の摩擦が増加し、それに伴い磨耗も増加する。ころの長さがその幅に比べて長い場合、軌道面とのその長い接触ラインは、ころを軌道面から離している潤滑性膜の重大な掻き乱しを引き起こしてしまう。そしてこのことはトルクを要する。典型的には、ころは、過度な端部負荷を避けるために、わずかにクラウニングが施されている。しかし、それでも、端部間でそのクラウニングが相当の程度減少すると、そこで接触のラインに沿って掻き乱しが起きてしまう。

【０００５】

典型的な円錐ころ軸受は、内輪軌道又はコーン上の軌道面の大径端部において軌道スラストつばを備えている。このつばは、ころが軌道面をせり上って放出されてしまうのを、防止している。通常、つばの高さはころの大径端部における直径の約２０％である。このことにより、接触力は比較的狭い領域に集中し、磨耗は加速される。つばところの端部の表面仕上げが雑であると粗面が残り、この粗面によって潤滑剤によって形成される保護膜が突き破られ、より大きな摩擦が生み出される。ころの大端部表面は、望ましい接触領域を得るためにわずかにＲ面状である。しかし、クラウニングがあまりに顕著であると、それによって、ころとつばの間の軸力はあまりに小さい領域にさらに集中してしまう。ころの大端部表面での振れ（runout）が顕著になると、その端部面とスラストつばとの間の油膜が不安定になり、より大きなトルクの原因となる。

【０００６】

典型的には、ころ端部面は、コーン軌道面の大端部から比較的離れたスラストつばに接触している。そしてこのことによって、その接触点と軌道面との間に比較的大きなモーメントアームが生じている。この大きなモーメントアームによって、ころを軸周りに回転させるのに比較的大きなトルクが必要となり、これが、軸受のコーンを回転させるのに必要とされるトルクに形を変える。最終的に、ころ上のクラウニングによって、典型的な軸受は、通常、ころと軌道面の間の接触の中心を、ころの端部間の途中に有する。そのため、比較的大きなモーメントアームは、ころの大端部の接触点と軌道面に沿った接触の中心点との間に存在する。このモーメントアームに沿って作用する接触力は、ころを傾ける傾向があり、そのことにより、軸受中でのトルクが増大する。

【発明の開示】

【課題を解決するための手段】

【０００７】

本発明は、対向した軌道面を備えた軌道と、一方の軌道上の軌道面の大端部につば面を有するスラストつばと、軌道間に位置し、軌道面に沿って接触する位置となる側面と、つば面に沿って接触する位置となる大端部面を有する円錐ころと、を有する円錐ころ軸受に関する。前記軌道面、前記つば面、前記ころの側面及び大端部面のすべては、ころ自体と同様に、軸受が、その内部にある流体力学的な油膜の掻き乱れを最小限に減らし、ほとんどトルクを必要とせず、また、最小限の磨耗しか起こさないように設計されている。本発明はまた、このような軸受で支持されたピニオンを有する車両用差動装置に関する。

【発明を実施するための最良の形態】

【０００８】

以下、図を参照すると、車両用の差動装置Ａ（図１）は、その内部でピニオンシャフト４と２本のアクスルシャフト６が回転するハウジング２を含んでなっている。前者は軸Ｘ周りに、後者は、軸Ｘに対して横切って延在し且つその上方にある軸Ｙ周りに回転する。

【０００９】

ピニオンシャフト４はピニオン１０を支持しており、このピニオン１０は、クロスシャフ

ト１６が貫通して延在するキャリア１４上のリングギア１２と噛み合う。クロスシャフト１６はベベルギア１８を支持しており、このベベルギア１８は、アクスルシャフト６上にあるもう一つのベベルギア２０に噛み合う。そのため、リングギア１２とそのキャリア１４が回転すると、側方のシャフト６が回る。キャリア１４は、ハウジング２に直接配置され大端部が互いに突き出る方向に向かい合うように取付けられている２つの円錐ころ軸受２４上で回転する。

【００１０】

一方、ピニオンシャフト４は、ハウジング２に間接配置されて小端部が互いに突き出る方向に向かい合うように取付けられている円錐ころ軸受２６と２８の中で回転する。軸受２６はピニオン１０に最も近いヘッドポジションを占め、一方、軸受２８はピニオン１０からより離れたテールポジションを占める。２つの軸受２６と２８は、ピニオンシャフト４の周りにぴったりと取り付けられており、さらにハウジング２にもぴったりと取り付けられており、その上、予圧状態に設定されている。結果的に、軸Ｘはハウジング２に対して固定され、ピニオン１０は、シャフト４と共に軸Ｘ周りに回転可能でありながら、ハウジング２に対して半径方向にも、軸方向にも動かないようになっている。

【００１１】

実際、ハウジング２は座ぐり３０を有している（図１）一方で、ピニオンシャフト４は、ピニオン１０の背後に隣接する部分に軸受座３２を有している。ヘッド軸受２６は、座ぐり３０内と受座３２の周囲にはめ合わされている。また、ハウジング２は、ハウジング２の外部へと開いている座ぐり３４を有し、シャフト４は、その座ぐり３４に位置する他の軸受座３６を有している。テール軸受２８は座ぐり３４内と軸受座３６の周囲にはめ合わされている。ピニオンシャフト１０は、ハウジング２から突き出ていて、そこでシャフト４の端部にねじ込まれているナット４０によってそのシャフト４上に保持されているユニバーサルジョイント３８と嵌合している。軸受座３０はピニオン１０に通じており、一方、受座３４はユニバーサルジョイント３８の端部に通じている。テール軸受２８の外側に、ハウジング２には、ユニバーサルジョイント４０の周囲の動的液体バリアを確保するシール４２が嵌合されている。

【００１２】

ヘッド軸受２６は（図２）、軸受座３２に締まりばめではめ合わされたコーン４４を含んでおり、且つ、そのコーン４４を取り囲んでいる。さらに、軸受２６は、コーン４４とカップ４６の間に一列に配列された円錐ころ４８と、コーン４４とカップ４６との間に取り付けられていて、隣接するころ４８間の適切な間隔を維持するためにころ４８を受ける保持器５０とを備えている。コーン４４は、軸受２６の内輪軌道を構成しており、一方、カップ４６は外輪軌道４６を構成している。

【００１３】

ここでコーン４４に着目すると、コーン４４は（図２）、軸Ｘに対して外側に向けられている円錐状の軌道面５２を備えている。加えて、コーン４４は、円錐状の軌道面５２の大端部にスラストつば５４と、小端部に保持つば５６を有する。スラストつば５４は、軌道面５２の大端部に沿うつば面５８と、その反対方向に背面６０を有する。実際、コーン４４は、その背面６０に沿ってピニオン１０の後背部に接するか、ピニオン１０の後背部に接触しているシムに接している。加えて、コーン４４は、スラストつば５４の基部に、軌道面５２の大端部とつば面５８を分ける逃げ溝６２を有する。一般的には、円錐状の軌道面５２は、頂点を軸Ｘ上に有する円錐状の包絡面内にある。実際には、軌道面５２はわずかにクラウニングされており、円錐状の包絡面は、クラウニングされた軌道面５２（図３）を通り抜ける平均を表す。また、つば面５８も頂点を軸Ｘ上に有する円錐状の包絡面内にあるが、つば面５８の頂角は軌道面５２の頂角に比べかなり大きい。背面６０は軸Ｘに垂直な一平面内にある。

【００１４】

カップ４６は、該カップの端部間に延在し、内側に軸Ｘ方向に向けて、且つコーン４４の軌道面５２に向けて内側方向に面している円錐状の軌道面６４を有する。軌道面６４の小

端部に、カップ４６は背面６６を有しており、この背面はハウジング２内で座ぐり３０の端部の段部に当接する。一般的には、軌道面６４は、頂点を軸Ｘ上に有する円錐状の包絡面内にある。実際、２つの軌道面５２と６４についての包絡面の頂角は軸Ｘ上で一致する。コーン軌道面５２と同じように、カップ軌道面６４はわずかにクラウニングされているため、その軌道面に対する包絡面は実際には平均である（図３）。カップ軌道面６４－又はより正確には軌道面６４に対する包絡面－は、軸Ｘと角度θをなす。カップ４６の背面６６は、軸Ｘに垂直な一平面内にある。

【００１５】

円錐ころ４８は、コーン４４とカップ４６の間に環状の列に配置されている（図２）。各ころ４８は円錐状の側面７０（図３）と大径端部面７２とを有し、その側面に沿って該ころ４８がコーン４４及びカップ４６のそれぞれの軌道面５２及び６４と接触し、その大径端部面に沿って前記ころ４８がスラストつば５４のつば面５８に接触する。円錐状の側面７０はクラウニングされている。しかし、クラウニングされた表面の平均面は、軸Ｘ上に頂点を有する円錐状の包絡面を形成し、その包絡面の頂点は、軌道面５２と軌道面６４により形成される各包絡面の頂点とほぼ同じ位置にある（図３）。これによって、ころ４８は頂点に合わせて配置されるので、ころが、個々の中心軸Ｚを中心にして回転しながら軌道面５２と６４に沿って転動するときに、純粋な転がり接触が、ころ４８の側面７０と軌道面５２及び６４との間に存在する。すなわち、この転がり接触はスピンを伴わない。個々のころ４８の大端部面７２は、中心をころ４８の軸Ｚ上に有する球面の一部を形成する。

【００１６】

テール軸受２８は、ヘッド軸受２６と同じ構成部品を有するが、ピニオン１０から離れて位置しており、より小さい荷重しか支持しないため、ヘッド軸受２６ほど大きくなくて良い。テール軸受２８のカップ４６は、ハウジング２の座ぐり３０内に、締まりばめではめ合わされていて、その背面６６は座ぐり３０の端部の段部に当接している（図１）。テール軸受２８のコーン４４は、ピニオンシャフト４上の軸受座３６を覆うように締まりばめではめ合わされていて、その背面６０は、ユニバーサルジョイント３８の端部に当接して支えている。ユニバーサルジョイント３８をピニオンシャフト４に取り付けているナット４０は、シャフト４上でジョイント３８の軸方向の位置を制御しており、それによって、２つの軸受２６と２８の設定を制御している。ナット４０は、軸受２６と２８に予圧がかかるまで締める。このことにより、軸受２６と２８は軸方向又は半径方向のクリアランス無しで作動する。

【００１７】

軸受２６と２８は、もちろん、ピニオンシャフト４の端部のピニオン１０を定まった軸方向及び半径方向の位置に保持しながら、ピニオンシャフト４を最小限の摩擦抵抗で回転可能にしている。リングギア１２と噛み合っているピニオン１０は、リングギア１２を回転させ、次にリングギア１２はアクスルシャフト６を回転させて車両を動かす。２つの軸受２６と２８は、ハウジング２に、例えばユニバーサルジョイント４０に連結しているドライブシャフトの重量などによりピニオンシャフトに加えられる半径方向荷重を伝える。それらはまた、軸方向荷重も伝える。これらの荷重は、主として、ピニオン１２とリングギア１２の係合している歯のラセン形状により発現する。ピニオン１０が車両を前進させる方向に回転している場合、前記係合している歯は、前記ピニオンを２つの軸受２６と２８の方向へと動かそうとする。しかし、ヘッド軸受２６が、ピニオン１０の軸方向への移動に抵抗し、ピニオン１０をリングギア１２と適切に噛み合うように維持する。一方、ピニオン１０が逆方向に回転する場合、テール軸受２８は前記ピニオンがハウジング２内へさらに動いてしまう傾向に抵抗する。

【００１８】

ヘッド軸受２６は、その外観において、いく分、従来の設計の単列円錐ころ軸受に似ている。しかし、違いがあり、この違いにより、軸受２６は、より少ないトルクで且つより少ない磨耗量で駆動することが可能となっている。結果として、軸受２６は、同等のサイズ

の従来の単列円錐ころ軸受と比べて、より少ない出力しか消費せず、また、非常に長い寿命を有している。

【００１９】

まず第一に、カップ４６に対する円錐状の軌道面６４の角度θはかなり大きく（図２）、２０度から３０度の範囲である。角度が大きくなると、実効軸受広がり（effective bearing spread）が、軸受の幾何学的広がり（spread）と従来の軸受の実効広がり（effective spread）を超えて増加し、この実効広がり（effective spread）における増加は、シャフト４の安定性を増す。カップ軌道面６４の角度が大きくなると、軌道面６４はシャフト４上にかかるスラスト荷重に対しより良く耐えることができるような配置になる。また、ピニオン１０は、そのらせん状の歯によって、スラスト荷重をシャフト４上に及ぼす。このスラスト荷重は、車両が前進するとき、ヘッド軸受２６の方向に向けられ、そのヘッド軸受２６により阻止される。

【００２０】

カップ軌道面の角度θが小さいと、軌道面にはスラスト荷重が強い力となって加わるが、反対に、カップ軌道面６４の角度θが大きいと、軌道面力は小さくなる。このことにより、ころの側面７０と軌道面５２、６４の間でのより短いラインでの接触が可能となり、また、出力損失が少なくてすむ。このことに関して、短い接触ラインを有するころ４８は、ほとんど動的油膜を乱さない。油膜を乱したり、かき乱すことは、出力を必要とするものである。軌道面５２と６４にかかる力が減少することにより、磨耗も減少する。

【００２１】

加えて、軸受２６は、低いＬ／Ｄ比を有する、すなわち、任意のころ４８の長さＬと、そのころ４８の大端部の直径Ｄの比が低い（図２）。典型的なヘッド軸受では、Ｌ／Ｄ比は２．０よりも大きいが、軸受２６ではＬ／Ｄ比は１．５から１．２の間である。また、長さが短いので、ころ４８は動的油膜をほとんど掻き乱さない。この短い長さにより、任意のころ４８の側面７０に沿った直径は大きくなり、このことにより、ころ側面７０と軌道面５２、６４の間の力はより大きな面積にわたって分散する。言い換えれば、このことにより、ころ４８と軌道面５２、６４の間の接触ラインの幅は広くなるので、接触圧力は少なくなる。Ｌ／Ｄ比が小さいので、軸受２６は、破片粒子をより排除できる。

【００２２】

従来の軸受に比べると、軸受２６は、コーン４２とカップ４４に高度に輪郭形成された軌道面５２と６４を有し、ころ４８の側面７０に対しても同じことが当てはまる（図３）。要するに、軌道面５２、６４ところ側面７０はクラウニングされている。従来の円錐ころ軸受が、ころの端部で圧力を最小限化する様に輪郭形成された軌道面ところ側面を有することは確かであるが、しかし、その輪郭形成では接触長さ１インチあたりの逃げで４０μｉｎ．よりも小さかった。軸受２６において、各ころ４８の両端部で、接触１インチあたりの逃げは７００μｉｎ．から１５００μｉｎ．になる。このことは、軽荷重時に軸受２６の剛性を減らすが、また、軽荷重時にころ４８が油膜を疵つけたり掻き乱すことを減らす。なぜなら、端部に近いころ側面７０は、動的油膜を過度に掻き乱すのを妨げるのに十分なだけ軌道面５２と６４から離れており、それらの領域では膜は断裂されないからである。しかし、より重い荷重（これは普通は過渡的である）は、クラウニングの作用を減じ、軸受２６をより安定にする。

【００２３】

さらに従来の軸受と比べると、コーン４２上の軸受２６は、つば面５８を備えた比較的高いスラストつば５４を有し、これは、摩耗し始める部分であるつば面５８が従来の軸受の対応する構成要素よりも大きな面積を有する。従来の軸受において、つばの高さは任意のころの大端部の直径の約２０％である。軸受２６においては、つば５４の高さＨは、任意のころ４８の大端部の直径Ｄの３０％から４５％になる（図２）。同様のことが、スラストつば５４のつば面についても当てはまり、大体、任意のころ４８の大端部の直径Ｄの３０％から４５％になる。つば面５８の大きな面積の結果、任意のころ４８の端部面７２は、大きな面積にわたってつば面５８に接触し、このことにより、磨耗が減少する。そのた

め、ピニオン１０の軸方向の位置は、より長い期間にわたって同じ位置にとどまる。また、ピニオン１０はより長い期間にわたってリングギア１２と適切に噛み合う。

【００２４】

軸受の接触表面の大きな粗さにより、動的油膜は突き破られるので、表面仕上げは低い平均粗さとなっていなければならない。このことは、それらの面間の接触が摺動と回転で特徴付けられる、つば側面５８ところ４８の大端部面７２との接触にも当てはまる。面５８と面７２に大きな粗さがあると金属－金属接触が起こり、これは低速でのトルクを増大させ、同時に温度を上昇させる。軸受２６では、つば面５８ところ４８の端部面７２の算術平均粗さが４μｉｎ．以下である。この小さい表面粗さにより、つば面に沿った動的油膜は維持され、トルクが軽減される。

【００２５】

特別な高さのつば高さとそのことによる増大した表面領域を最大限に利用するために、任意のころ４８の大端部面７２がある球面の半径は、その球面から、ころ４８の側面７０についての包絡面により形成される頂角までの、ころの軸Ｚに沿って測定した距離の９０％より大きいことが望ましい。典型的な円錐ころ軸受では、任意のころの大端部面半径は、頂点長さの８５％から９０％にある。各ころ４８の大端部面７２の半径を大きくしたことによって、大端部面７２と円錐状つば面５８の接触の面積が増大し、そしてこのことが低速でのトルクと高温を減少させる。ころ４８は、その大端部面７２において円錐状つば面５８と、長軸が円周方向に延びた、いく分楕円形状の領域に沿って接触する（図４）。

【００２６】

ころ４８の大端部面７２での振れを減らすことによって、動的油膜はより安定になり、大端部面７２とつば面５８との間に起こる金属－金属接触は起こりにくくなる。任意のころ４８の端部面７２での振れにより、端部面７２でのふらつきが生じ、振れの高位点では油膜を突き破ってしまい、端部面７２はつば面５８と接触してしまう。ころ４８の大端部面７２でのこの振れは、５０μｉｎ．を超えないことが好ましい。

【００２７】

典型的には、ピニオンシャフトを支持するヘッド軸受のころは、コーン軌道面とつば面の約０．０４μｉｎ．から０．０６μｉｎ．上方で、つば面に接触している。これにより、接触の楕円領域とコーン軌道面との間に、比較的大きなモーメントアームが発生し、ころは軌道面に沿って転動する時に、そのかなり長いモーメントアームを通って作用している、つば面に沿って摩擦により生じるトルクに打ち勝たねばならない。このトルクは、出力を消費し、このことは温度を上昇させる。軸受２６のコーン４２では、任意のころ４８の大端部面７２と円錐状つば面５８間の接触の楕円領域の中心は、コーン軌道面５２と円錐状つば面５８の各包絡線の交差する点を超えて、半径方向に０．０２ｉｎ．から０．０４ｉｎ．の間の距離Ｃにある（図４）。これにより、つば面５８に沿って摩擦に逆らって、ころ４８を回転させるのに必要なトルクが減り、そのため、軸受２６を回転させるのに必要なトルクが減る。

【００２８】

円錐ころ軸受において、ころとつば面間の摩擦接触により、ころにトルクが加わるだけでなく、軌道面間でころが傾けられる傾向がある。ここで、他のモーメントアームーころの大端部面と、ころところがそれに沿って転動する軌道面間の接触の領域の中心との間のモーメントアームーを考慮する。このモーメントの結果、ころはその軸を中心に蝶つがい状に僅かに摺動運動する、または言い換えれば、傾く。このことにより軸受のトルクを増加させる傾向がある。典型的には、わずかにクラウニングされたころとわずかにクラウニングされた軌道面間の接触の中心は、一般的に、ころの端部間の中間にある。軸受２６において、任意のころ４８の側面７０と軌道面５２、６４の接触の中心は、ころ４８の大端部面７２方向へ補正されている。これにより、より小さいモーメントアームとなり、この小さいモーメントアームによっては、軌道面５２と６４間で、ころ４８は傾けられる程度は少ない。接触の中心は、コーン軌道面をカップ軌道面に対して故意にずらして調整することで補正される。このずらした調整は０．０００３から０．０００２ラジアンであること

が望ましい。

【００２９】

好ましくは、テール軸受２８も同様に、前述の特性、つまりヘッド軸受２６を従来の軸受と区別しているような特性を有する。

【００３０】

スラストつばは、コーン４４上のコーン軌道面５２の大端部にある代わりに、カップ４６上にあってもよく、又は、少なくともカップ軌道面６４の大端部にカップ４６に当接して設けられても良い。

【図面の簡単な説明】

【００３１】

【図１】本発明によって構成され且つこれを具現化している、軸受上に実装されたピニオンシャフトを有する車両用差動装置の断面図。

【図２】ヘッドポジションにある軸受の拡大断面図。

【図３】接触面の輪郭内を強調して示してある軸受の断面図。

【図４】図３中の線４－４に沿った部分断面図であって、ころと、スラストつば面の接触している領域を隠れ線で示してある。

【図１】

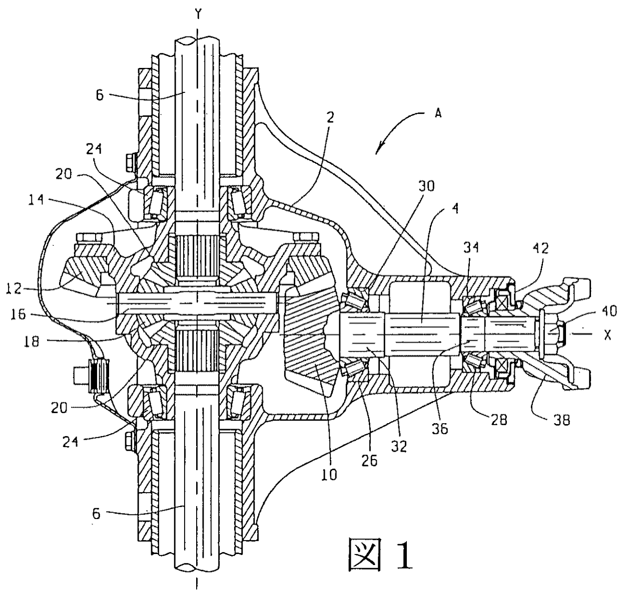


図１

【図２】

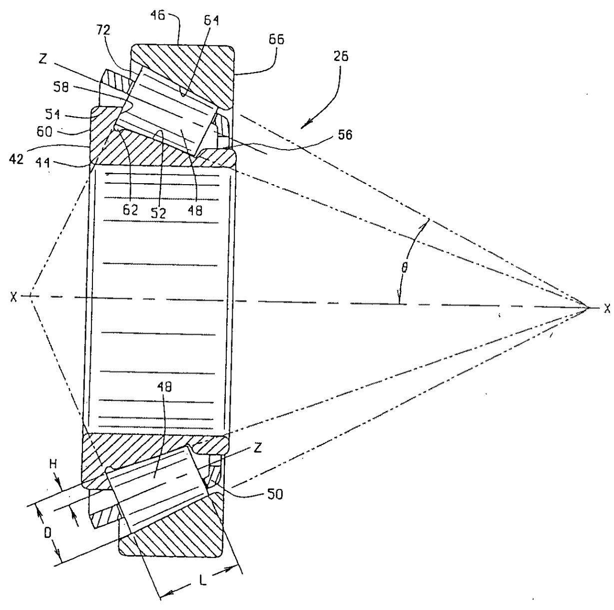


図２

【図３】

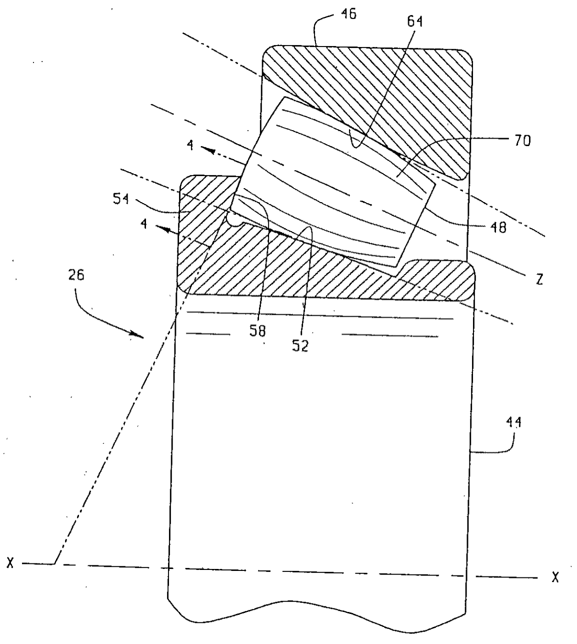


図３

【図４】

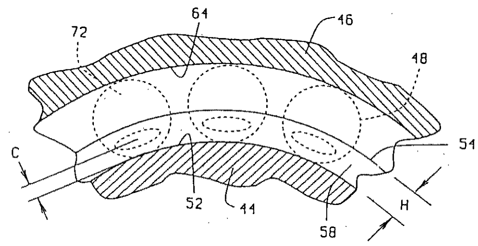


図４

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(71) Applicant: THE TIMKEN COMPANY [US/US]; 1835 Dueber Avenue, S.W., Canton, OH 44706-0930 (US).

(72) Inventor: JOKI, Mark, A.; 5930 North American Road, Dover, OH 44622 (US).

(74) Agent: SOIFER, Jonathan, P.; Polster, Lieder, Woodruff & Lucchesi, L.C., 763 South New Ballas, St. Louis, MO 63141 (US).

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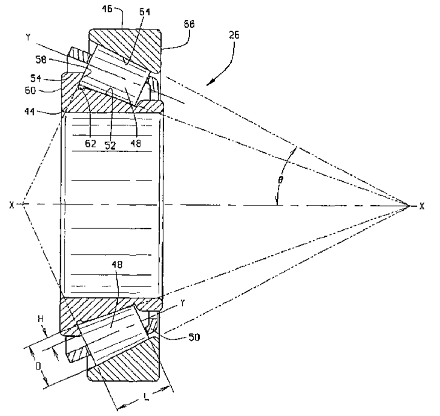
[Continued on next page]

(54) Title: BEARING WITH LOW WEAR AND LOW POWER LOSS CHARACTERISTICS



(57) Abstract: A tapered roller bearing (26, 28) that is well-suited for supporting a pinion in an automotive differential (A) has a cone (44) and a cup (46) provided with opposed raceways (52, 64) that are crowned. The cone also has a thrust rib (54) provided with a rib face (58) at the large end of its raceway. In addition, the bearing has tapered rollers (48), each having a tapered side face (70) that is crowned and a large end (72) that is spherical. The rollers (48) contact the raceways (52, 64) along their crowned side faces and the rib face along their spherical end faces. The ratio of the roller length (C) to the large end diameter (D) is less than 1.5. The crowning on the raceway (52, 64), together with the crowning of the roller side face (70) provide total end relief ranging between 700 μin. and 1500 μin. per inch. The centers of contact between the side faces (70) and raceways (52, 64) are offset toward the rib face (58). The height of the rib face (58) amounts to 30% - 45% of the diameter of the large ends (72) of the rollers (48). The radius of the spherical large end face (52) for a roller exceeds 90% of the roller apex length. The runout in the large end faces is less than 50 μin. and the center of contact between the end face (72) of each roller (48) and the rib face (58) is between ranges between 0.02 and 0.04 in. All of this contributes to low torque demands by the bearing itself and wear.

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BEARING WITH LOW WEAR AND LOW POWER LOSS CHARACTERISTICS

Technical Field

This invention relates in general to tapered roller bearings and, more particularly, to a tapered roller bearing that is characterized by low wear and low power loss.

Background Art

The gearing in an automotive differential must operate with considerable precision to keep noise and wear at a minimum. Typically, that gearing includes a ring gear and a pinion both of which rotate in a differential housing. The pinion meshes with ring gear to transfer torque from a drive shaft to the ring gear which in turn transfers the torque to axle shafts that extend from the housing. Both gears have spiral teeth, that is to say they are hypoid gears.

The pinion must rotate with considerable stability, that is to say, about a fixed axis and in a fixed axial position. To this end, the pinion is carried on a shaft that rotates in a pair of tapered roller bearings that are fitted to the differential housing where they are mounted in opposition and set to a condition of preload. While the arrangement gives the stability required - at least at the outset - the tapered roller bearings in current use require a measure of torque to overcome friction within the bearings themselves as well as the churning of the lubricant within them. This torque consumes power. Moreover, the bearings in time experience wear, and this detracts from the stability, so that the pinion may migrate from the position that provides the most effective engagement with the ring gear.

Several factors affect the torque required by a tapered roller bearing and the wear that the bearing experiences. For example, small angles on the raceway cause the rollers to fit tightly between the raceways under the preload, thus increasing the friction in the bearing and wear as well. When the rollers are long in comparison to their width, the longer lines of contact with the raceways result in significant churning of the lubricant film which separates the rollers from the raceways, and this requires torque. Typically, the rollers are slightly crowned to prevent excessive edge-loading, but even so, between the ends the crowning diminishes considerably, and here the churning occurs along the lines of contact. The typical tapered roller bearing has a thrust rib at the large diameter end of the raceway on its inner race or cone, and the rib prevents the rollers from moving up the raceway and being expelled. Normally the height of the rib amounts to about 20% of the diameter of the rollers at their large ends. This concentrates the contact forces in relatively small areas, and accelerates wear. A rough surface finish on the rib and roller ends leaves asperities that penetrate the protective film created by the lubricant, thus creating higher friction. The large end faces on the rollers are slightly radiused to provide a desirable contact area, but when the crown becomes too pronounced, it serves to further concentrate the axial forces between the rollers and rib in an area that is too small. Pronounced runout in the large end face of a roller will destabilize the oil film between the end face and the thrust rib and contribute to higher torque. Typically, the roller end faces contact the thrust rib relatively far from the large end of the cone raceway, and this creates a relatively large moment arm between the point of contact and the raceway. The large moment arm demands a relatively high torque to rotate the roller about its axis, and this translates into torque required to rotate the cone of the bearing. Finally the typical bearing, owing to the crown on the rollers, has the center of contact between the rollers and the raceways generally located midway between the ends of the rollers. Thus, a relatively large moment arm exits between the contact at the large end of the roller and the center of contact along the raceways. The contact force acting along this moment arm tends to skew the rollers, and that in turn increases the torque in the bearing.

Summary of the Invention

The present invention resides in a tapered roller bearing having races provided with opposed raceways, a thrust rib having a rib face at the large end of the raceway on one of the races, and tapered rollers between the races and having side faces along which they contact the raceways and large end faces along which they contact the rib face. The raceways, the rib face, the side and large end faces of the rollers, as well as the rollers themselves are all configured such that the bearing reduces churning of the hydrodynamic oil film within it to a minimum, demands little torque, and experiences minimal wear. The invention also resides in an automotive differential having a pinion'supported on such a bearing.

Brief Description of Drawings

Figure 1 is a sectional view of an automotive differential having a pinion shaft mounted on bearings constructed in accordance with and embodying the present invention;

Figure 2 is an enlarged sectional of the bearing in the head position;

Figure 3 is a sectional view of the bearing with its contacting surfaces exaggerated in contour; and

Figure 4 is a fragmentary sectional view taken along line 4-4 of Fig. 3 and showing the rollers and areas of contact with the thrust rib face in phantom lines.

Best Mode For Carrying Out the Invention

Referring now to the drawings, a differential A (Fig. 1) for an automotive vehicle includes a housing 2 within which a pinion shaft 4 and two axle shafts 6 rotate, the former about an axis X and the latter about an axis Y that extends transversely with respect to the axis X, but is located above it. The pinion shaft 4 carries a pinion 10 which meshes with a ring gear 12 on a carrier 14 through which a cross shaft 16 extends. The cross shaft 16 carries bevel gears 18 which mesh with more bevel gears 20 on the axle shafts 6. Thus, when the ring gear 12 and its carrier 14 rotate, the side shafts 6 turn. The carrier 14 rotates on two tapered roller bearings 24 that are fitted to the housing 2 in the direct configuration, that is with their large ends presented toward each other. The pinion shaft 4, on the other hand, rotates in tapered roller bearings 26 and 28 that are fitted to the housing 2 in the indirect configuration, that is with their small ends presented toward each other. The bearing 26 occupies the head position closest to the pinion 10, whereas the bearing 28 occupies the tail position more remote from the pinion 10. The two bearings 26 and 28 fit snugly around the pinion shaft 4 and snugly into the housing 2 and are furthermore set to a condition of preload. As a consequence, the axis X remains fixed with respect to the housing 2, and the pinion 10, while being able to rotate with the shaft 4 about the axis X, cannot be displaced radially or axially with respect to the housing 2.

Actually, the housing 2 contains (Fig. 1) a counterbore 30, whereas the pinion shaft 4 adjacent to the back face of the pinion 10 has a bearing seat 32. The head bearing 26 fits into the counterbore 30 and around the seat 32. The housing 2 also has a counterbore 34 which opens out of the housing 2 toward the exterior, while the shaft 4 has another bearing seat 36 which is located in the counterbore 34. The tail bearing 28 fits into the counterbore 34 and around the bearing seat 36. The pinion shaft 10 projects out of the housing 2 where it is fitted with a universal joint 38 that is retained on the shaft 4 with a nut 40 that is threaded over the end of the shaft 4. The bearing seat 30 leads up to the pinion 10, whereas the seat 34 leads up to the end of the universal joint 38. Beyond the tail bearing 28, the housing 2 is fitted with a seal 42 which establishes a dynamic fluid barrier around the universal joint 40.

The head bearing 26 includes (Fig. 2) a cone 44, which is fitted over the bearing seat 32 with an interference fit, and surrounds the cone 44. In addition the bearing 26 has tapered rollers 48 arranged in a single row between the cone 44 and the cup 46, and a cage 50 that fits between the cone 44 and cup 46 and receives the rollers 48 to maintain the proper spacing between adjacent rollers 48. The cone 44 constitutes the inner race of the bearing 26, whereas the cup 46 constitutes the outer race 46.

Turning now to the cone 44, it includes (Figs. 2) a tapered raceway 52 that is presented outwardly away from the axis X. In addition, the cone 44 has a thrust rib 54 at the large end of the raceway 52 and a retaining rib 56 at the small end. The thrust rib 54 has a rib face 58 along the larger end of the raceway 52 and a back face 60 presented in the opposite direction. Indeed, the cone 44 along its back face 60 abuts the back of the pinion 10 or else abuts a shim that is against the back of the pinion 10. In addition, the cone 44 at the base of its thrust rib 54 has an undercut 62 which separates the large end of the raceway 52 from the rib face 58. Generally speaking, the tapered raceway 52 lies within a conical envelope having its apex along the axis X. Actually, the raceway 52 is slightly crowned, and the conical envelope represents a mean which passes through the crowned raceway 52 (Fig. 3). The rib face 58 also lies within a conical envelope that has its apex along the axis X, but the apex angle for the rib face 58 is much greater than the apex angle for the raceway 52. The back face 60 lies in a plane that is perpendicular to the axis X.

The cup 46 has a tapered raceway 64 that extends between its ends and is presented inwardly toward the axis X and toward the raceway 52 of the cone 44. At the small end of its raceway 64 the cup 46 has a back face 66 that is against the shoulder at the end of the counterbore 30 in the housing 2. Generally speaking, the raceway 64 lies within a conical envelope that has its apex along the axis X. Indeed, the apecies for the envelopes of the two raceways 52 and 64 coincide along the axis X. Like the cone raceway 52, the cup raceway 64 is slightly crowned, so the envelope for that raceway actually represents a mean (Fig. 3). The cup raceway 64 - or more accurately the envelope for the raceway 64 - forms an angle θ with the axis X. The back face 66 of the cup 46 lies in a plane that is perpendicular to the axis X.

The tapered rollers 48 are organized in a circular row between the cone 44 and the cup 46 (Fig. 2). Each roller 48 has a tapered side face 70 (Fig. 3) along which the roller 48 contacts the tapered raceways 52 and 64 of the cone 44 and cup 46, respectively, and a large diameter end face 72 along which the roller 48 contacts the rib face 58 on the thrust rib 54. The tapered side face 70 is crowned, but nevertheless the mean of the crowned surface establishes a conical envelope that has its apex along the axis X approximately at the location of the apecies for the envelopes formed by the raceways 52 and 64 (Fig. 3). This places rollers 48 on apex, so that when they roll along the raceways 52 and 64, rotating about their individual center axes Z as they do, pure rolling contact exists between the side faces 70 of the rollers 48 and the raceways 52 and 64, that is to say, rolling contact that is characterized by the absence of spinning. The large end face 72 of each roller 48 forms a segment of a sphere having its center along the axis Z of the roller 48.

The tail bearing 28 possesses the same components as the head bearing 26, although it need not be as large as the head being 26 since, being located remote from the pinion 10, it carries a lesser load. The cup 46 of the tail bearing 28 fits into the counterbore 30 of the housing 2 with an interference fit, its back face 66 being against the shoulder at the end of the counterbore 30 (Fig. 1). The cone 44 of the tail bearing 28 fits over the bearing seat 36 on the pinion shaft 4 with an interference it, and its back face 60 bears against the end of the universal joint 38. The nut 40, which attaches the universal joint 38 to the pinion shaft 4, controls the axial position of the joint 38 on the shaft 4, and that in turn controls the setting of the two bearings 26 and 28. The nut 40 is turned down until the bearings 26 and 28 are in preload. Thus, the bearings 26 and 28 operate without any axial or radial clearances.

The bearings 26 and 28 of course enable the pinion shaft 4 to rotate with minimum frictional resistance, all while maintaining the pinion 10, which is on the end of the shaft 4, in a fixed axial and radial position. The pinion 10, being engaged with the ring gear 12, rotates the ring gear 12, and the ring gear 12, in turn, rotates the axle shafts 6 to propel the vehicle. The two bearings 26 and 28 transfer to the housing 2 radial loads that are imparted to the pinion shaft, such as by the weight of the drive shaft connected to the universal joint 40. They also transfer axial loads. These loads develop primarily by reason of the spiral of the meshing teeth on the pinion 12 and ring gear 12. When the pinion 10 rotates in the direction which propels the vehicle forwardly, the meshing teeth urge the pinion toward the two bearings 26 and 28, but the head bearing 26 resists axial displacement of the pinion 10 and maintains the pinion 10 properly meshed with the ring gear 12. On the other hand, when the pinion 10 rotates in the opposite direction, the tail bearing 28 resists the tendency of the pinion to move farther into the housing 2.

The head bearing 26 in outward appearance somewhat resembles any single row tapered roller bearing of conventional design. But differences exist, and these differences enable the bearing 26 to operate with less torque and less wear. As a consequence, the bearing 26 consumes less power than a tradition single row tapered roller bearing of equivalent size and has a greater lifespan.

To begin with, the angle θ of the tapered raceway 64 for the cup 46 is quite large (Fig. 2), ranging between 20° and 30°. The greater angle increases the effective bearing spread well beyond the geometrical spread of the bearings and beyond the effective spread of traditional bearings as well, and the increase in effective spread enhances the stability of the shaft 4. The greater angle of the cup raceway 64 positions the raceway 64 to better resist thrust exerted on the shaft 4, and the pinion 10, owing to the spiral of its teeth, exerts a thrust load on the shaft 4 - a thrust load that is directed toward and resisted by the head bearing 26 when the vehicle is propelled forwardly. Whereas a lesser angle θ for a cup raceway translates a thrust load into high forces at the raceways, the greater angle θ for the cup raceway 64 reduces raceway forces. This in turn allows for shorter lines of contact between the side faces 70 of the rollers and the raceways 52 and 64 and less power loss. In this regard, the rollers 48 with their shorter lines of contact, disturb the hydrodynamic oil film less, and to disrupt or churn an oil film requires power. The reduced forces at the raceways 52 and 64 also reduce wear.

In addition, the bearing 26 has a low L/D ratio, that is to say the ratio between the length L of any roller 48 and the diameter D of the large end for that roller 48 (Fig. 2). In the typical head bearing the L/D ratio exceeds 2.0, whereas in the bearing 26 the L/D ratio ranges between 1.5 and 1.2. Again, with a shorter length, the rollers 48 impart less churning to the hydrodynamic oil film. The shorter length results in a greater diameter along the side face 70 of any roller 48, and that distributes the forces between the roller side faces 70 and raceways 52 and 64 over a greater area. In other words, it increases the width of the lines of contact between the rollers 48 and raceways 52 and 64, so that the contact stresses are less. The lesser L/D ratio also enables the bearing 26 to better reject debris particles.

In contrast to conventional bearings, the bearing 26 has the raceways 52 and 64. of its cone 42 and cup 44 highly profiled and the same holds true for the side faces 70 of its rollers 48 (Fig. 3). In short, the raceways 52 and 64 and the roller side faces 70 are crowned. To be sure, conventional tapered roller bearings have their raceways and roller side faces profiled to minimize stresses at the end of the rollers, but the profiling results in less than 40 μin. of relief per inch of contact length. In the bearing 26 the relief amounts to 700 μin. to 1500 μin in. per inch of contact at both ends of each roller 48. This reduces the stiffness of the bearing 26 at light loads, but also causes the rollers 48 to plow or churn less of the oil film at light loads, because the rollers side faces 70 near their ends are separated from the raceways 52 and 64 sufficiently to avoid excessive churning of the hydrodynamic oil film and thus in those regions do not disrupt the film. However, heavier loads, which are normally transient, diminish the crowning and give the bearing 26 greater stability .

In further contrast to more traditional bearings, the bearing 26 on its cone 42 has a relatively high thrust rib 54 which provides the rib face 58 with greater surface area as it begins to wear than its counterparts in conventional bearings. In a conventional bearing the height of the rib is about 20% of the diameter of the large end of any roller. In the bearing 26, the height H (Fig. 2) of the rib 54 amounts to between 30% and 45% of the diameter D of the large end of any roller 48. The same holds true with regard to the rib face on the thrust rib 54 - it amounts to about 30% to 45% of the diameter D for the large end of any roller 48. As a consequence of the larger surface area on the rib face 58, the end face 72 of any roller 48 contacts the rib face 58 over a greater surface area, and this reduces wear. Hence, the axial position of the pinion 10 will remain the same over a greater duration, and the pinion 10 will mesh properly with the ring gear 12 over a greater duration.

High asperities in contacting surfaces of a bearing penetrate the hydrodynamic oil film, so the surface finish should have a low average roughness, and this holds particularly true along the rib face 58 and the large end face 72 of the rollers 48 where the contact between those faces is characterized by sliding and spinning. When high asperities exist along the faces 58 and 72, metal-to-metal contact occurs which increases torque at low speed and produces high temperatures as well. In the bearing 26, the arithmetic average roughness of the rib face 58 and of the end face 72 on the rollers 48 is 4 μin. or less. This low surface roughness preserves the hydrodynamic oil film along the rib face and reduces torque.

In order to best utilize the extra rib height and the increased surface area it provides, the radius of the sphere in which the large end surface 72 of any roller 48 lies should exceed 90% of the distance from that spherical surface to the apex formed by the envelope for the side face 70 on the roller 48, with the distance of course being measured along the roller axis Z. In the typical tapered roller bearing the large end face radius for any roller falls between 85% and 90% of the apex length. By increasing the radius of the large end face 72 or each roller 48, the area of contact between the large end face 72 and the conical rib face 58 increases, and this reduces torque at low speed and high temperature. The rollers 48 at their large end faces

72 contact the conical rib face 58 along somewhat elliptical regions, with the major axis of the ellipses extending circumferentially (Fig. 4).

By reducing runout in the large end face 72 of the rollers 48, the hydrodynamic oil film becomes more stable and less chance for metal-to-metal contact between the large end faces 72 and the rib face 58 exists. Runout in the end face 72 of any roller 48 produces wobble at that end face 72, and the high point can break through the oil film and bring the end face 72 into contact with the rib face 58. The runout at the large end face 72 of a roller 48 should not exceed 50 µin.

Typically, the rollers for the head bearing on which a pinion shaft is supported contact the rib face about 0.04 in. 0.06 in. above the cone raceway and the rib face. This creates a relatively large moment arm between the elliptical region of contact and the cone raceway, and as the roller rolls along the cone raceway it must overcome the torque generated by the frictional force along the rib face acting through the relatively long moment arm. The torque consumes power that increases operating temperature. In the cone 42 of the bearing 26, the center of elliptical region of contact between the large end face 72 of any roller 48 and the conical rib face 58 lies a distance C between 0.02 in. and 0.04 in. radially beyond the intersection of the envelopes for the cone raceway 52 and the conical rib face 58 (Fig. 4). This reduces the torque required to rotate the rollers 48 against the friction along the rib face 58 and thus reduces the torque required to rotate the bearing 26.

Not only does the frictional contact between a roller and the rib face in a tapered roller bearing impose a torque on the roller, but it also tends to skew the roller between the raceways. Here another moment arm comes into consideration - the moment arm between the large end face of the roller and the centers of the region of contact between roller and the raceways along which it rolls. As a consequence of the moment, the roller is likely to pivot slightly about its center - or in other words skew, which tends to increase bearing torque. Typically the centers of contact between a slightly crowned roller and the slightly crowned raceways lie generally midway between the ends of the roller. In the bearing 26, the centers of contact between the side face 70 of any roller 48 and the raceways 52 and 64 are offset toward the large end face 72 of the roller 48. This produces a smaller moment arm and with it less tendency of the roller 48 to skew between the raceways 52 and 64. The center of contact is offset by an intentional misalignment of cone raceway to cup raceway. That misalignment should range between 0.0003 and 0.002 radians.

Preferably, the tail bearing 28 likewise has the foregoing characteristics, that is the characteristics which distinguish the head bearing 26 from conventional bearings.

In lieu of being on the cone 44 at the large end of the cone raceway 52, the thrust rib may be on the cup 46, or at least positioned against the cup 46 at the large end of the cup raceway 64.

Claims:

1. A tapered roller bearing in a differential for a vehicle, for accommodating rotation about a bearing axis, said bearing comprising an inner race having a tapered raceway that is presented outwardly away from the axis; an outer race surrounding the inner race and having a tapered raceway presented inwardly toward the axis and toward the raceway on the inner race, at an angle of 20° to 30° with respect to the axis; a thrust rib on one of the races and having a rib face located at the large end of the raceway for that race and presented at a substantial angle with respect to that raceway; tapered rollers arranged in a circular row between the races, with each having a side face along which it contacts the tapered raceways of the inner and outer races and a large face along which it contacts the rib face of the thrust rib, wherein the rib face extends away from the envelope formed by the raceway at which it is located a distance greater than 30% of the diameter of the large ends of the rollers.

2. A bearing according to claim 1 wherein the side face of each roller lies generally within a conical envelope having its apex along the bearing, axis, and the large end face lies in a spherical envelope having its center along the axis of the roller and a radius that is at least 90% of the distance between the apex of the roller envelope and spherical envelope of the large end face.

3. A bearing according to claim 1 wherein the rib face lies within a conical envelope having its apex on the bearing axis.

4. A bearing according to claim 3 wherein the large end face of each roller contacts the rib face along an area of contact, and the center of the area of contact is no greater than 0.04 inches from the envelope of the raceway that leads to the rib face.

5. A bearing according to claim 1 wherein the raceways or the side faces of the rollers or both are crowned; and wherein the centers of the

regions of contact between the side faces of the rollers and raceways are offset toward the large end of the rollers.

6. A bearing according to claim 1 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 µin.

7. A bearing according to claim 1 wherein the runout of the large end face on each roller does not exceed 50 µin.

8. In a differential for a vehicle, with the differential including a housing, head and tail bearings in the housing where they define a bearing axis, a pinion shaft mounted in the bearings for rotation about the bearing axis, a pinion on the pinion shaft and within the housing, and a ring gear in the housing and meshing with the pinion, whereby rotation of the pinion shaft will rotate the ring gear, and wherein the head bearing comprises; a cone fitted to pinion shaft and having a tapered raceway that is presented away from the axis, and a thrust rib provided with a rib face located on the thrust rib at the large end of the raceway and oriented at a substantial angle with respect to the raceway; a cup having a tapered raceway that is presented inwardly toward the axis and the cone raceway at an angle of 20° to 30° with respect to the axis; and tapered rollers organized in a row between the cone and the cup and having tapered side faces along which the rollers contact the raceways and large end faces along which the rollers contact the rib face of the thrust rib wherein the ratio between the roller length and the diameter of the roller at its large end does not exceed about 1.5; wherein the raceways and the side faces of the rollers are crowned to provide reliefs at both ends of the raceways, and the reliefs are between 700 µin. and 1500 µin. per inch; the rib face lying within a conical envelope having its center at the axis; the side face of each roller lying in a conical envelope having its center along the axis; the large end face of each roller lying within a spherical envelope having its center along the axis of the roller and a radius greater than 90% of the distance between the apex for the

envelope defined by the side face and the spherical envelope of the end face measured along the axis of the roller.

9. A differential according to claim 11 wherein the rib face extends away from the envelope formed by the cone raceway a distance greater than 30% of the diameter of the large ends of the rollers.

10. A differential according to claim 11 wherein the large end of each roller contacts the rib face along an area of contact, and the center of the area of contact is not more than 0.04 in. from the envelope of the cone raceway.

11. A bearing according to claim 11 wherein the raceways or the side faces of the rollers or both are crowned, and the centers of the regions of contact between the side faces of the rollers and the raceways are offset toward the large ends of the rollers.

12. A differential according to claim 14 wherein the sum of the misalignments between the side face of a roller and the cone raceway, on one hand, and the side face of that roller and the cup raceway, on the other, is between about 0.0003 and 0.002 radians.

13. A differential according to claim 15 wherein the tail bearing comprises the same elements that are in the head bearing; and wherein the bearings are mounted in opposition and set to a condition of preload.

14. A bearing according to claim 1 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 μin.

15. A bearing according to claim 1 wherein the runout of the large end face on each roller does not exceed 50 μin.

16. A tapered roller bearing in a differential for a vehicle, for accommodating rotation about a bearing axis said bearing comprising: a cone having a tapered raceway that is presented outwardly from the axis and a thrust rib at the large end of the tapered raceway, the tapered raceway being crowned, yet generally lying in a conical envelope having its

apex along the axis, the thrust rib having a rib face to which the tapered raceway leads, with the rib face being presented at a substantial angle with respect to the raceway; a cup surrounding the cone and having a tapered raceway that is presented-inwardly toward the axis at an angle of 20° to 30° with respect to the axis, the cup raceway being crowned and generally lying in a conical envelope that has its apex along the axis at generally the same location as the apex for the envelope of the cone raceway; tapered rollers arranged in a circular row between the cone and cup with each having a roller axis and also tapered side face along which it contacts the raceways and a large end face along which it contacts the rib face, the side face being crowned and the large end face lying in a spherical envelope having its center along the roller axis, the crowning of the raceways and the side face of each roller being such that the centers of the regions of contact between the side face and the raceways are offset toward the rib face, wherein the sum of the misalignments between the side face of each roller and the cone raceway and the side face of that roller and the cup raceway is between about 0.0003 and 0.002 radians.

17. A bearing according to claim 17 wherein the rib face lies within a conical envelope having its apex along the bearing axis.

18. A bearing according to claim 16 wherein the ratio between the roller length and the diameter of the roller at its large end does not exceed about 1.5.

19. A bearing according to claim 16 wherein the raceways and the side faces of the rollers are crowned to provide reliefs at both ends of the raceways, and the reliefs are between 700 μin. and 1500 μin. per inch.

20. A bearing according to claim 16 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 μin.

21. A bearing according to claim 16 wherein the cup raceway is presented inwardly toward the axis and toward the cone raceway, at an

angle of 20° to 30° with respect to the axis and wherein the rib face of the thrust rib extends away from the envelope formed by the raceway at which it is located a distance greater than 30% of the diameter of the large ends of the rollers.

22. A bearing according to claim 16 wherein the large end face of each roller contacts the rib face along an area of contact, and the center of the area of contact is no greater than 0.04 inches from the envelope of the raceway that leads to the rib face.

23. A bearing according to claim 16 wherein the runout of the large end face on each roller does not exceed 50 μin.

24. A bearing according to claim 16 wherein the side face of each roller lies generally within a conical envelope having its apex along the bearing axis, and the large end face lies in a spherical envelope having a radius that is at least 90% of the distance between the apex of the roller envelope and spherical envelope of the large end face.

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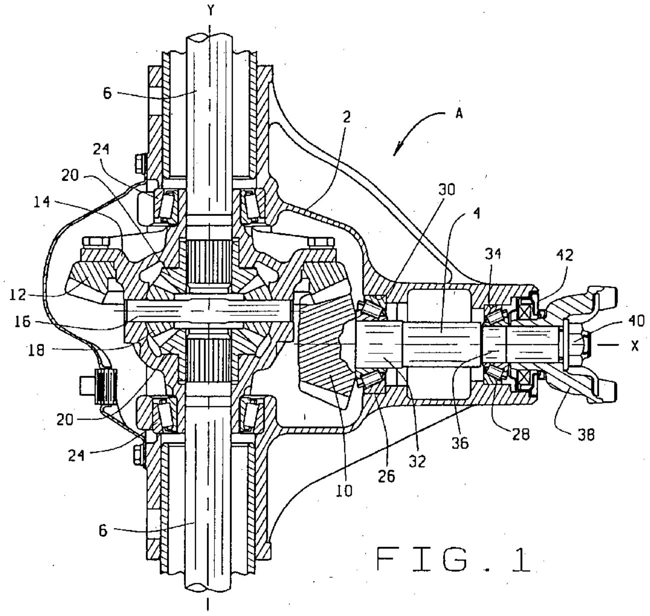
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FIG. 1

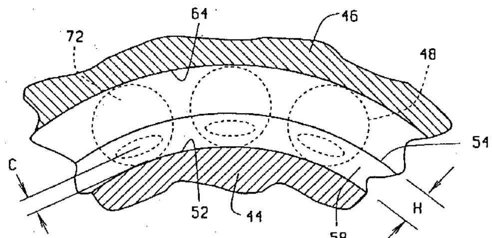
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FIG. 4

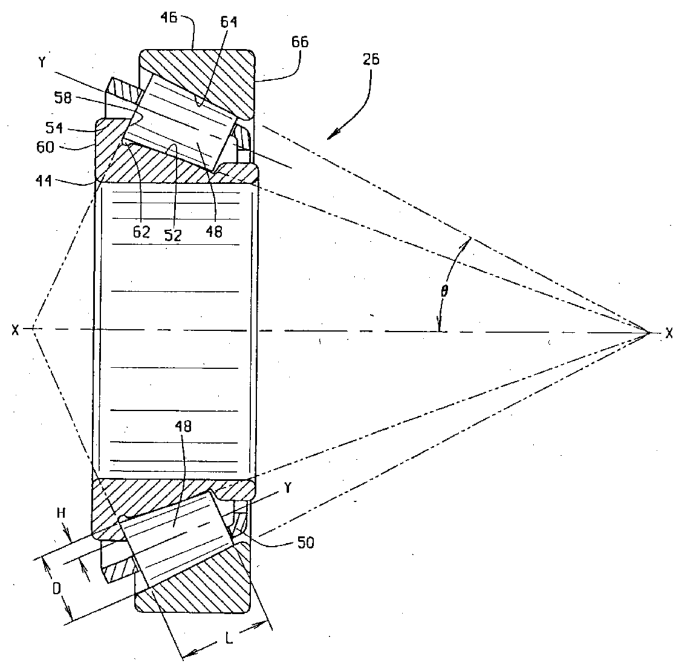

FIG. 2

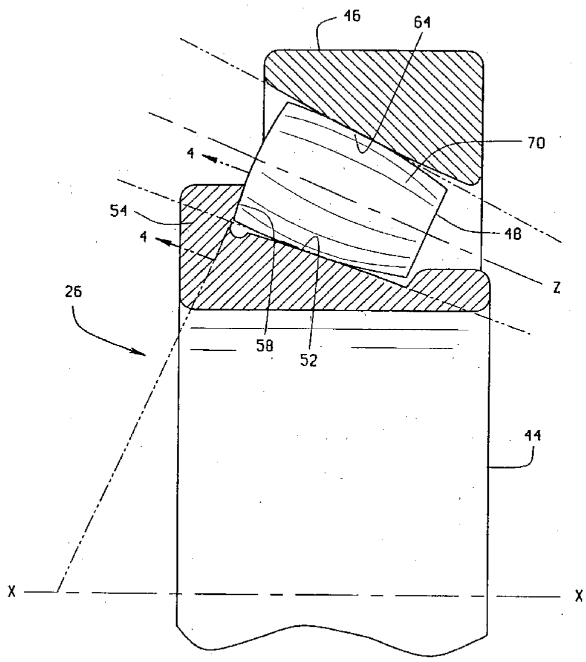

FIG. 3

【国際公開パンフレット（コレクトバージョン）】

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(71) Applicant: THE TIMKEN COMPANY [US/US]; 1835 Dueber Avenue, S.W., Canton, OH 44706-0930 (US).

(72) Inventor: JOKI, Mark, A.; 5930 North American Road, Dover, OH 44622 (US).

(74) Agent: SOIFER, Jonathan, P.; Polster, Lieder, Woodruff & Lucchesi, L.C., 763 South New Ballas, St. Louis, MO 63141 (US).

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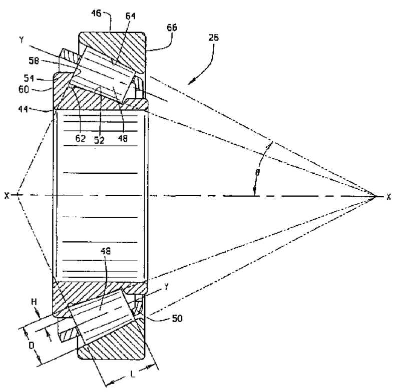
(54) Title: BEARING WITH LOW WEAR AND LOW POWER LOSS CHARACTERISTICS



(57) Abstract: A tapered roller bearing (26, 28) that is well-suited for supporting a pinion in an automotive differential (A) has a cone (44) and a cup (46) provided with opposed raceways (52, 64) that are crowned. The cone also has a thrust rib (54) provided with a rib face (58) at the large end of its raceway. In addition, the bearing has tapered rollers (48), each having a tapered side face (70) that is crowned and a large end (72) that is spherical. The rollers (48) contact the raceways (52, 64) along their crowned side faces and the rib face along their spherical end faces. The ratio of the roller length (C) to the large end diameter (D) is less than 1.5. The crowning on the raceway (52, 64), together with the crowning of the roller side face (70) provide total end relief ranging between 700 μin. and 1500 μin. per inch. The centers of contact between the side faces (70) and raceways (52, 64) are offset toward the rib face (58). The height of the rib face (58) amounts to 30% - 45% of the diameter of the large ends (72) of the rollers (48). The radius of the spherical large end face (52) for a roller exceeds 90% of the roller apex length. The runout in the large end faces is less than 50 μin. and the center of contact between the end face (72) of each roller (48) and the rib face (58) is between ranges between 0.02 and 0.04 in. All of this contributes to low torque demands by the bearing itself and wear.

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BEARING WITH LOW WEAR AND LOW POWER LOSS CHARACTERISTICS

Technical Field

This invention relates in general to tapered roller bearings and, more particularly, to a tapered roller bearing that is characterized by low wear and low power loss.

Background Art

The gearing in an automotive differential must operate with considerable precision to keep noise and wear at a minimum. Typically, that gearing includes a ring gear and a pinion both of which rotate in a differential housing. The pinion meshes with ring gear to transfer torque from a drive shaft to the ring gear which in turn transfers the torque to axle shafts that extend from the housing. Both gears have spiral teeth, that is to say they are hypoid gears.

The pinion must rotate with considerable stability, that is to say, about a fixed axis and in a fixed axial position. To this end, the pinion is carried on a shaft that rotates in a pair of tapered roller bearings that are fitted to the differential housing where they are mounted in opposition and set to a condition of preload. While the arrangement gives the stability required - at least at the outset - the tapered roller bearings in current use require a measure of torque to overcome friction within the bearings themselves as well as the churning of the lubricant within them. This torque consumes power. Moreover, the bearings in time experience wear, and this detracts from the stability, so that the pinion may migrate from the position that provides the most effective engagement with the ring gear.

Several factors affect the torque required by a tapered roller bearing and the wear that the bearing experiences. For example, small angles on the raceway cause the rollers to fit tightly between the raceways under the preload, thus increasing the friction in the bearing and wear as well. When the rollers are long in comparison to their width, the longer lines of contact with the raceways result in significant churning of the lubricant film which separates the rollers from the raceways, and this requires torque. Typically, the rollers are slightly crowned to prevent excessive edge-loading, but even so, between the ends the crowning diminishes considerably, and here the churning occurs along the lines of contact. The typical tapered roller bearing has a thrust rib at the large diameter end of the raceway on its inner race or cone, and the rib prevents the rollers from moving up the raceway and being expelled. Normally the height of the rib amounts to about 20% of the diameter of the rollers at their large ends. This concentrates the contact forces in relatively small areas, and accelerates wear. A rough surface finish on the rib and roller ends leaves asperities that penetrate the protective film created by the lubricant, thus creating higher friction. The large end faces on the rollers are slightly radiused to provide a desirable contact area, but when the crown becomes too pronounced, it serves to further concentrate the axial forces between the rollers and rib in an area that is too small. Pronounced runout in the large end face of a roller will destabilize the oil film between the end face and the thrust rib and contribute to higher torque. Typically, the roller end faces contact the thrust rib relatively far from the large end of the cone raceway, and this creates a relatively large moment arm between the point of contact and the raceway. The large moment arm demands a relatively high torque to rotate the roller about its axis, and this translates into torque required to rotate the cone of the bearing. Finally the typical bearing, owing to the crown on the rollers, has the center of contact between the rollers and the raceways generally located midway between the ends of the rollers. Thus, a relatively large moment arm exits between the contact at the large end of the roller and the center of contact along the raceways. The contact force acting along this moment arm tends to skew the rollers, and that in turn increases the torque in the bearing.

Summary of the Invention

The present invention resides in a tapered roller bearing having races provided with opposed raceways, a thrust rib having a rib face at the large end of the raceway on one of the races, and tapered rollers between the races and having side faces along which they contact the raceways and large end faces along which they contact the rib face. The raceways, the rib face, the side and large end faces of the rollers, as well as the rollers themselves are all configured such that the bearing reduces churning of the hydrodynamic oil film within it to a minimum, demands little torque, and experiences minimal wear. The invention also resides in an automotive differential having a pinion'supported on such a bearing.

Brief Description of Drawings

Figure 1 is a sectional view of an automotive differential having a pinion shaft mounted on bearings constructed in accordance with and embodying the present invention;

Figure 2 is an enlarged sectional of the bearing in the head position;

Figure 3 is a sectional view of the bearing with its contacting surfaces exaggerated in contour; and

Figure 4 is a fragmentary sectional view taken along line 4-4 of Fig. 3 and showing the rollers and areas of contact with the thrust rib face in phantom lines.

Best Mode For Carrying Out the Invention

Referring now to the drawings, a differential A (Fig. 1) for an automotive vehicle includes a housing 2 within which a pinion shaft 4 and two axle shafts 6 rotate, the former about an axis X and the latter about an axis Y that extends transversely with respect to the axis X, but is located above it. The pinion shaft 4 carries a pinion 10 which meshes with a ring gear 12 on a carrier 14 through which a cross shaft 16 extends. The cross shaft 16 carries bevel gears 18 which mesh with more bevel gears 20 on the axle shafts 6. Thus, when the ring gear 12 and its carrier 14 rotate, the side shafts 6 turn. The carrier 14 rotates on two tapered roller bearings 24 that are fitted to the housing 2 in the direct configuration, that is with their large ends presented toward each other. The pinion shaft 4, on the other hand, rotates in tapered roller bearings 26 and 28 that are fitted to the housing 2 in the indirect configuration, that is with their small ends presented toward each other. The bearing 26 occupies the head position closest to the pinion 10, whereas the bearing 28 occupies the tail position more remote from the pinion 10. The two bearings 26 and 28 fit snugly around the pinion shaft 4 and snugly into the housing 2 and are furthermore set to a condition of preload. As a consequence, the axis X remains fixed with respect to the housing 2, and the pinion 10, while being able to rotate with the shaft 4 about the axis X, cannot be displaced radially or axially with respect to the housing 2.

Actually, the housing 2 contains (Fig. 1) a counterbore 30, whereas the pinion shaft 4 adjacent to the back face of the pinion 10 has a bearing seat 32. The head bearing 26 fits into the counterbore 30 and around the seat 32. The housing 2 also has a counterbore 34 which opens out of the housing 2 toward the exterior, while the shaft 4 has another bearing seat 36 which is located in the counterbore 34. The tail bearing 28 fits into the counterbore 34 and around the bearing seat 36. The pinion shaft 10 projects out of the housing 2 where it is fitted with a universal joint 38 that is retained on the shaft 4 with a nut 40 that is threaded over the end of the shaft 4. The bearing seat 30 leads up to the pinion 10, whereas the seat 34 leads up to the end of the universal joint 38. Beyond the tail bearing 28, the housing 2 is fitted with a seal 42 which establishes a dynamic fluid barrier around the universal joint 40.

The head bearing 26 includes (Fig. 2) a cone 44, which is fitted over the bearing seat 32 with an interference fit, and surrounds the cone 44. In addition the bearing 26 has tapered rollers 48 arranged in a single row between the cone 44 and the cup 46, and a cage 50 that fits between the cone 44 and cup 46 and receives the rollers 48 to maintain the proper spacing between adjacent rollers 48. The cone 44 constitutes the inner race of the bearing 26, whereas the cup 46 constitutes the outer race 46.

Turning now to the cone 44, it includes (Figs. 2) a tapered raceway 52 that is presented outwardly away from the axis X. In addition, the cone 44 has a thrust rib 54 at the large end of the raceway 52 and a retaining rib 56 at the small end. The thrust rib 54 has a rib face 58 along the larger end of the raceway 52 and a back face 60 presented in the opposite direction. Indeed, the cone 44 along its back face 60 abuts the back of the pinion 10 or else abuts a shim that is against the back of the pinion 10. In addition, the cone 44 at the base of its thrust rib 54 has an undercut 62 which separates the large end of the raceway 52 from the rib face 58. Generally speaking, the tapered raceway 52 lies within a conical envelope having its apex along the axis X. Actually, the raceway 52 is slightly crowned, and the conical envelope represents a mean which passes through the crowned raceway 52 (Fig. 3). The rib face 58 also lies within a conical envelope that has its apex along the axis X, but the apex angle for the rib face 58 is much greater than the apex angle for the raceway 52. The back face 60 lies in a plane that is perpendicular to the axis X.

The cup 46 has a tapered raceway 64 that extends between its ends and is presented inwardly toward the axis X and toward the raceway 52 of the cone 44. At the small end of its raceway 64 the cup 46 has a back face 66 that is against the shoulder at the end of the counterbore 30 in the housing 2. Generally speaking, the raceway 64 lies within a conical envelope that has its apex along the axis X. Indeed, the apecies for the envelopes of the two raceways 52 and 64 coincide along the axis X. Like the cone raceway 52, the cup raceway 64 is slightly crowned, so the envelope for that raceway actually represents a mean (Fig. 3). The cup raceway 64 - or more accurately the envelope for the raceway 64 - forms an angle θ with the axis X. The back face 66 of the cup 46 lies in a plane that is perpendicular to the axis X.

The tapered rollers 48 are organized in a circular row between the cone 44 and the cup 46 (Fig. 2). Each roller 48 has a tapered side face 70 (Fig. 3) along which the roller 48 contacts the tapered raceways 52 and 64 of the cone 44 and cup 46, respectively, and a large diameter end face 72 along which the roller 48 contacts the rib face 58 on the thrust rib 54. The tapered side face 70 is crowned, but nevertheless the mean of the crowned surface establishes a conical envelope that has its apex along the axis X approximately at the location of the apecies for the envelopes formed by the raceways 52 and 64 (Fig. 3). This places rollers 48 on apex, so that when they roll along the raceways 52 and 64, rotating about their individual center axes Z as they do, pure rolling contact exists between the side faces 70 of the rollers 48 and the raceways 52 and 64, that is to say, rolling contact that is characterized by the absence of spinning. The large end face 72 of each roller 48 forms a segment of a sphere having its center along the axis Z of the roller 48.

The tail bearing 28 possesses the same components as the head bearing 26, although it need not be as large as the head being 26 since, being located remote from the pinion 10, it carries a lesser load. The cup 46 of the tail bearing 28 fits into the counterbore 30 of the housing 2 with an interference fit, its back face 66 being against the shoulder at the end of the counterbore 30 (Fig. 1). The cone 44 of the tail bearing 28 fits over the bearing seat 36 on the pinion shaft 4 with an interference it, and its back face 60 bears against the end of the universal joint 38. The nut 40, which attaches the universal joint 38 to the pinion shaft 4, controls the axial position of the joint 38 on the shaft 4, and that in turn controls the setting of the two bearings 26 and 28. The nut 40 is turned down until the bearings 26 and 28 are in preload. Thus, the bearings 26 and 28 operate without any axial or radial clearances.

The bearings 26 and 28 of course enable the pinion shaft 4 to rotate with minimum frictional resistance, all while maintaining the pinion 10, which is on the end of the shaft 4, in a fixed axial and radial position. The pinion 10, being engaged with the ring gear 12, rotates the ring gear 12, and the ring gear 12, in turn, rotates the axle shafts 6 to propel the vehicle. The two bearings 26 and 28 transfer to the housing 2 radial loads that are imparted to the pinion shaft, such as by the weight of the drive shaft connected to the universal joint 40. They also transfer axial loads. These loads develop primarily by reason of the spiral of the meshing teeth on the pinion 12 and ring gear 12. When the pinion 10 rotates in the direction which propels the vehicle forwardly, the meshing teeth urge the pinion toward the two bearings 26 and 28, but the head bearing 26 resists axial displacement of the pinion 10 and maintains the pinion 10 properly meshed with the ring gear 12. On the other hand, when the pinion 10 rotates in the opposite direction, the tail bearing 28 resists the tendency of the pinion to move farther into the housing 2.

The head bearing 26 in outward appearance somewhat resembles any single row tapered roller bearing of conventional design. But differences exist, and these differences enable the bearing 26 to operate with less torque and less wear. As a consequence, the bearing 26 consumes less power than a tradition single row tapered roller bearing of equivalent size and has a greater lifespan.

To begin with, the angle θ of the tapered raceway 64 for the cup 46 is quite large (Fig. 2), ranging between 20° and 30°. The greater angle increases the effective bearing spread well beyond the geometrical spread of the bearings and beyond the effective spread of traditional bearings as well, and the increase in effective spread enhances the stability of the shaft 4. The greater angle of the cup raceway 64 positions the raceway 64 to better resist thrust exerted on the shaft 4, and the pinion 10, owing to the spiral of its teeth, exerts a thrust load on the shaft 4 - a thrust load that is directed toward and resisted by the head bearing 26 when the vehicle is propelled forwardly. Whereas a lesser angle θ for a cup raceway translates a thrust load into high forces at the raceways, the greater angle θ for the cup raceway 64 reduces raceway forces. This in turn allows for shorter lines of contact between the side faces 70 of the rollers and the raceways 52 and 64 and less power loss. In this regard, the rollers 48 with their shorter lines of contact, disturb the hydrodynamic oil film less, and to disrupt or churn an oil film requires power. The reduced forces at the raceways 52 and 64 also reduce wear.

In addition, the bearing 26 has a low L/D ratio, that is to say the ratio between the length L of any roller 48 and the diameter D of the large end for that roller 48 (Fig. 2). In the typical head bearing the L/D ratio exceeds 2.0, whereas in the bearing 26 the L/D ratio ranges between 1.5 and 1.2. Again, with a shorter length, the rollers 48 impart less churning to the hydrodynamic oil film. The shorter length results in a greater diameter along the side face 70 of any roller 48, and that distributes the forces between the roller side faces 70 and raceways 52 and 64 over a greater area. In other words, it increases the width of the lines of contact between the rollers 48 and raceways 52 and 64, so that the contact stresses are less. The lesser L/D ratio also enables the bearing 26 to better reject debris particles.

In contrast to conventional bearings, the bearing 26 has the raceways 52 and 64. of its cone 42 and cup 44 highly profiled and the same holds true for the side faces 70 of its rollers 48 (Fig. 3). In short, the raceways 52 and 64 and the roller side faces 70 are crowned. To be sure, conventional tapered roller bearings have their raceways and roller side faces profiled to minimize stresses at the end of the rollers, but the profiling results in less than 40 μin. of relief per inch of contact length. In the bearing 26 the relief amounts to 700 μin. to 1500 μin in. per inch of contact at both ends of each roller 48. This reduces the stiffness of the bearing 26 at light loads, but also causes the rollers 48 to plow or churn less of the oil film at light loads, because the rollers side faces 70 near their ends are separated from the raceways 52 and 64 sufficiently to avoid excessive churning of the hydrodynamic oil film and thus in those regions do not disrupt the film. However, heavier loads, which are normally transient, diminish the crowning and give the bearing 26 greater stability .

In further contrast to more traditional bearings, the bearing 26 on its cone 42 has a relatively high thrust rib 54 which provides the rib face 58 with greater surface area as it begins to wear than its counterparts in conventional bearings. In a conventional bearing the height of the rib is about 20% of the diameter of the large end of any roller. In the bearing 26, the height H (Fig. 2) of the rib 54 amounts to between 30% and 45% of the diameter D of the large end of any roller 48. The same holds true with regard to the rib face on the thrust rib 54 - it amounts to about 30% to 45% of the diameter D for the large end of any roller 48. As a consequence of the larger surface area on the rib face 58, the end face 72 of any roller 48 contacts the rib face 58 over a greater surface area, and this reduces wear. Hence, the axial position of the pinion 10 will remain the same over a greater duration, and the pinion 10 will mesh properly with the ring gear 12 over a greater duration.

High asperities in contacting surfaces of a bearing penetrate the hydrodynamic oil film, so the surface finish should have a low average roughness, and this holds particularly true along the rib face 58 and the large end face 72 of the rollers 48 where the contact between those faces is characterized by sliding and spinning. When high asperities exist along the faces 58 and 72, metal-to-metal contact occurs which increases torque at low speed and produces high temperatures as well. In the bearing 26, the arithmetic average roughness of the rib face 58 and of the end face 72 on the rollers 48 is 4 μin. or less. This low surface roughness preserves the hydrodynamic oil film along the rib face and reduces torque.

In order to best utilize the extra rib height and the increased surface area it provides, the radius of the sphere in which the large end surface 72 of any roller 48 lies should exceed 90% of the distance from that spherical surface to the apex formed by the envelope for the side face 70 on the roller 48, with the distance of course being measured along the roller axis Z. In the typical tapered roller bearing the large end face radius for any roller falls between 85% and 90% of the apex length. By increasing the radius of the large end face 72 or each roller 48, the area of contact between the large end face 72 and the conical rib face 58 increases, and this reduces torque at low speed and high temperature. The rollers 48 at their large end faces

72 contact the conical rib face 58 along somewhat elliptical regions, with the major axis of the ellipses extending circumferentially (Fig. 4).

By reducing runout in the large end face 72 of the rollers 48, the hydrodynamic oil film becomes more stable and less chance for metal-to-metal contact between the large end faces 72 and the rib face 58 exists. Runout in the end face 72 of any roller 48 produces wobble at that end face 72, and the high point can break through the oil film and bring the end face 72 into contact with the rib face 58. The runout at the large end face 72 of a roller 48 should not exceed 50 μin.

Typically, the rollers for the head bearing on which a pinion shaft is supported contact the rib face about 0.04 in. 0.06 in. above the cone raceway and the rib face. This creates a relatively large moment arm between the elliptical region of contact and the cone raceway, and as the roller rolls along the cone raceway it must overcome the torque generated by the frictional force along the rib face acting through the relatively long moment arm. The torque consumes power that increases operating temperature. In the cone 42 of the bearing 26, the center of elliptical region of contact between the large end face 72 of any roller 48 and the conical rib face 58 lies a distance C between 0.02 in. and 0.04 in. radially beyond the intersection of the envelopes for the cone raceway 52 and the conical rib face 58 (Fig. 4). This reduces the torque required to rotate the rollers 48 against the friction along the rib face 58 and thus reduces the torque required to rotate the bearing 26.

Not only does the frictional contact between a roller and the rib face in a tapered roller bearing impose a torque on the roller, but it also tends to skew the roller between the raceways. Here another moment arm comes into consideration - the moment arm between the large end face of the roller and the centers of the region of contact between roller and the raceways along which it rolls. As a consequence of the moment, the roller is likely to pivot slightly about its center - or in other words skew, which tends to increase bearing torque. Typically the centers of contact between a slightly crowned roller and the slightly crowned raceways lie generally midway between the ends of the roller. In the bearing 26, the centers of contact between the side face 70 of any roller 48 and the raceways 52 and 64 are offset toward the large end face 72 of the roller 48. This produces a smaller moment arm and with it less tendency of the roller 48 to skew between the raceways 52 and 64. The center of contact is offset by an intentional misalignment of cone raceway to cup raceway. That misalignment should range between 0.0003 and 0.002 radians.

Preferably, the tail bearing 28 likewise has the foregoing characteristics, that is the characteristics which distinguish the head bearing 26 from conventional bearings.

In lieu of being on the cone 44 at the large end of the cone raceway 52, the thrust rib may be on the cup 46, or at least positioned against the cup 46 at the large end of the cup raceway 64.

Claims:

1. A tapered roller bearing in a differential for a vehicle, for accommodating rotation about a bearing axis, said bearing comprising an inner race having a tapered raceway that is presented outwardly away from the axis; an outer race surrounding the inner race and having a tapered raceway presented inwardly toward the axis and toward the raceway on the inner race, at an angle of 20° to 30° with respect to the axis; a thrust rib on one of the races and having a rib face located at the large end of the raceway for that race and presented at a substantial angle with respect to that raceway; tapered rollers arranged in a circular row between the races, with each having a side face along which it contacts the tapered raceways of the inner and outer races and a large face along which it contacts the rib face of the thrust rib, wherein the rib face extends away from the envelope formed by the raceway at which it is located a distance greater than 30% of the diameter of the large ends of the rollers.

2. A bearing according to claim 1 wherein the side face of each roller lies generally within a conical envelope having its apex along the bearing, axis, and the large end face lies in a spherical envelope having its center along the axis of the roller and a radius that is at least 90% of the distance between the apex of the roller envelope and spherical envelope of the large end face.

3. A bearing according to claim 1 wherein the rib face lies within a conical envelope having its apex on the bearing axis.

4. A bearing according to claim 3 wherein the large end face of each roller contacts the rib face along an area of contact, and the center of the area of contact is no greater than 0.04 inches from the envelope of the raceway that leads to the rib face.

5. A bearing according to claim 1 wherein the raceways or the side faces of the rollers or both are crowned; and wherein the centers of the

regions of contact between the side faces of the rollers and raceways are offset toward the large end of the rollers.

6. A bearing according to claim 1 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 μin.

7. A bearing according to claim 1 wherein the runout of the large end face on each roller does not exceed 50 μin.

8. In a differential for a vehicle, with the differential including a housing, head and tail bearings in the housing where they define a bearing axis, a pinion shaft mounted in the bearings for rotation about the bearing axis, a pinion on the pinion shaft and within the housing, and a ring gear in the housing and meshing with the pinion, whereby rotation of the pinion shaft will rotate the ring gear, and wherein the head bearing comprises; a cone fitted to pinion shaft and having a tapered raceway that is presented away from the axis, and a thrust rib provided with a rib face located on the thrust rib at the large end of the raceway and oriented at a substantial angle with respect to the raceway; a cup having a tapered raceway that is presented inwardly toward the axis and the cone raceway at an angle of 20° to 30° with respect to the axis; and tapered rollers organized in a row between the cone and the cup and having tapered side faces along which the rollers contact the raceways and large end faces along which the rollers contact the rib face of the thrust rib wherein the ratio between the roller length and the diameter of the roller at its large end does not exceed about 1.5; wherein the raceways and the side faces of the rollers are crowned to provide reliefs at both ends of the raceways, and the reliefs are between 700 μin. and 1500 μin. per inch; the rib face lying within a conical envelope having its center at the axis; the side face of each roller lying in a conical envelope having its center along the axis; the large end face of each roller lying within a spherical envelope having its center along the axis of the roller and a radius greater than 90% of the distance between the apex for the

envelope defined by the side face and the spherical envelope of the end face measured along the axis of the roller.

9. A differential according to claim 11 wherein the rib face extends away from the envelope formed by the cone raceway a distance greater than 30% of the diameter of the large ends of the rollers.

10. A differential according to claim 11 wherein the large end of each roller contacts the rib face along an area of contact, and the center of the area of contact is not more than 0.04 in. from the envelope of the cone raceway.

11. A bearing according to claim 11 wherein the raceways or the side faces of the rollers or both are crowned, and the centers of the regions of contact between the side faces of the rollers and the raceways are offset toward the large ends of the rollers.

12. A differential according to claim 14 wherein the sum of the misalignments between the side face of a roller and the cone raceway, on one hand, and the side face of that roller and the cup raceway, on the other, is between about 0.0003 and 0.002 radians.

13. A differential according to claim 15 wherein the tail bearing comprises the same elements that are in the head bearing; and wherein the bearings are mounted in opposition and set to a condition of preload.

14. A bearing according to claim 1 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 μin.

15. A bearing according to claim 1 wherein the runout of the large end face on each roller does not exceed 50 μin.

16. A tapered roller bearing in a differential for a vehicle, for accommodating rotation about a bearing axis said bearing comprising: a cone having a tapered raceway that is presented outwardly from the axis and a thrust rib at the large end of the tapered raceway, the tapered raceway being crowned, yet generally lying in a conical envelope having its

apex along the axis, the thrust rib having a rib face to which the tapered raceway leads, with the rib face being presented at a substantial angle with respect to the raceway; a cup surrounding the cone and having a tapered raceway that is presented-inwardly toward the axis at an angle of 20° to 30° with respect to the axis, the cup raceway being crowned and generally lying in a conical envelope that has its apex along the axis at generally the same location as the apex for the envelope of the cone raceway; tapered rollers arranged in a circular row between the cone and cup with each having a roller axis and also tapered side face along which it contacts the raceways and a large end face along which it contacts the rib face, the side face being crowned and the large end face lying in a spherical envelope having its center along the roller axis, the crowning of the raceways and the side face of each roller being such that the centers of the regions of contact between the side face and the raceways are offset toward the rib face, wherein the sum of the misalignments between the side face of each roller and the cone raceway and the side face of that roller and the cup raceway is between about 0.0003 and 0.002 radians.

17. A bearing according to claim 17 wherein the rib face lies within a conical envelope having its apex along the bearing axis.

18. A bearing according to claim 16 wherein the ratio between the roller length and the diameter of the roller at its large end does not exceed about 1.5.

19. A bearing according to claim 16 wherein the raceways and the side faces of the rollers are crowned to provide reliefs at both ends of the raceways, and the reliefs are between 700 μin. and 1500 μin. per inch.

20. A bearing according to claim 16 wherein the arithmetic mean roughness of the rib face and the large end faces of the rollers does not exceed 4 μin.

21. A bearing according to claim 16 wherein the cup raceway is presented inwardly toward the axis and toward the cone raceway, at an

angle of 20° to 30° with respect to the axis and wherein the rib face of the thrust rib extends away from the envelope formed by the raceway at which it is located a distance greater than 30% of the diameter of the large ends of the rollers.

22. A bearing according to claim 16 wherein the large end face of each roller contacts the rib face along an area of contact, and the center of the area of contact is no greater than 0.04 inches from the envelope of the raceway that leads to the rib face.

23. A bearing according to claim 16 wherein the runout of the large end face on each roller does not exceed 50 μin.

24. A bearing according to claim 16 wherein the side face of each roller lies generally within a conical envelope having its apex along the bearing axis, and the large end face lies in a spherical envelope having a radius that is at least 90% of the distance between the apex of the roller envelope and spherical envelope of the large end face.

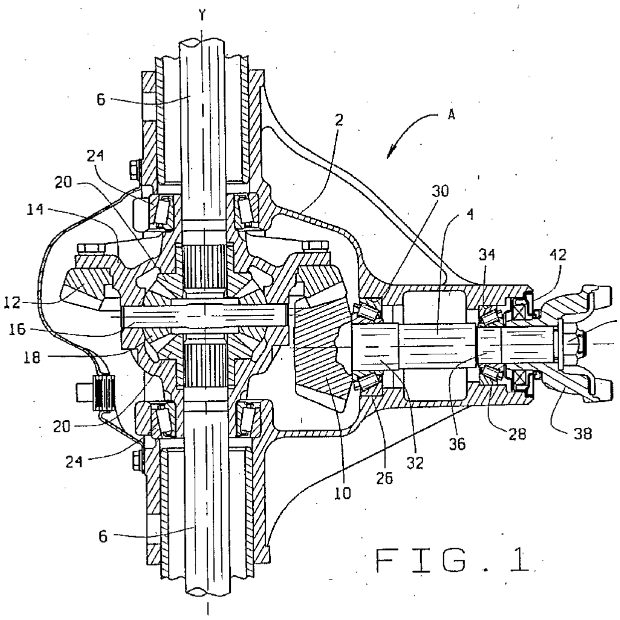
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FIG. 1

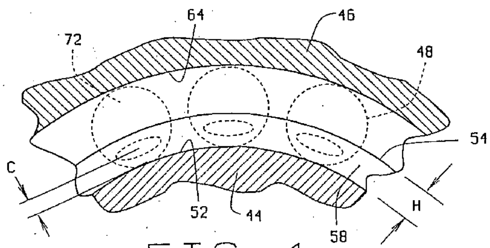
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FIG. 4

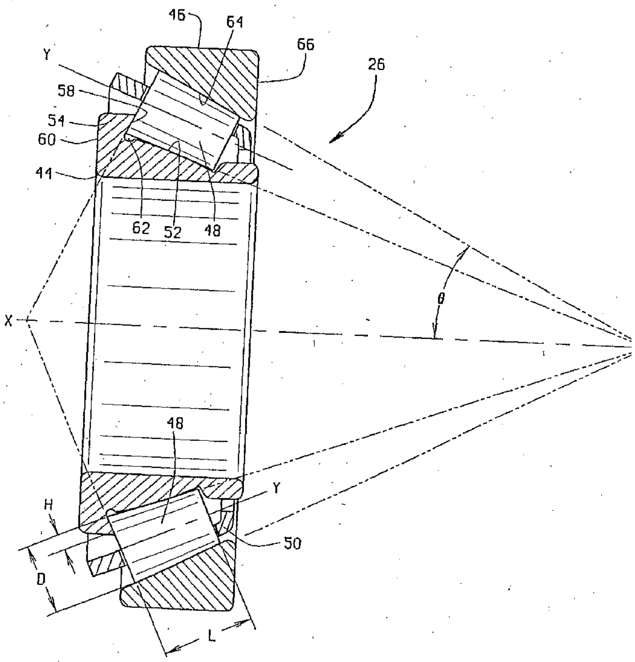

FIG. 2

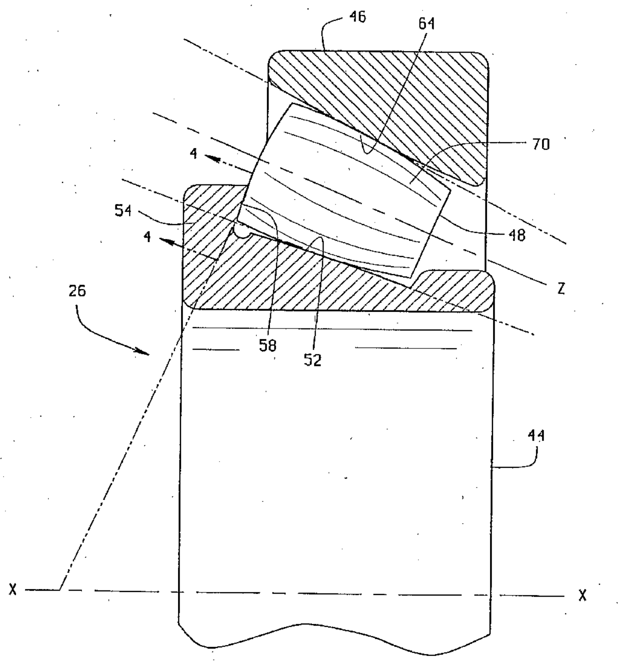

FIG. 3

【国際調査報告】

INTERNATIONAL SEARCH REPORT

Ional Application No
PCT/US 02/14809

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F16C19/36 F16H1/14

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F16C F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

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P,A	US 6 379 049 B1 (SHIBAZAKI KENICHI ET AL) 30 April 2002 (2002-04-30) the whole document ---	1-4
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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
2 August 2002	08/08/2002

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Orthlieb, C

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INTERNATIONAL SEARCH REPORT

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C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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(81)指定国　AP(GH,GM,KE,LS,MW,MZ,SD,SL,SZ,TZ,UG,ZM,ZW),EA(AM,AZ,BY,KG,KZ,MD,RU,TJ,TM),EP(AT,BE,CH,CY,DE,DK,ES,FI,FR,GB,GR,IE,IT,LU,MC,NL,PT,SE,TR),OA(BF,BJ,CF,CG,CI,CM,GA,GN,GQ,GW,ML,MR,NE,SN,TD,TG),AE,AG,AL,AM,AT,AU,AZ,BA,BB,BG,BR,BY,BZ,CA,CH,CN,CO,CR,CU,CZ,DE,DK,DM,DZ,EC,EE,ES,FI,GB,GD,GE,GH,GM,HR,HU,ID,IL,IN,IS,JP,KE,KG,KP,KR,KZ,LC,LK,LR,LS,LT,LU,LV,MA,MD,MG,MK,MN,MW,MX,MZ,NO,NZ,OM,PH,PL,PT,RO,RU,SD,SE,SG,SI,SK,SL,TJ,TM,TN,TR,TT,TZ,UA,UG,UZ,VN,YU,ZA,ZM,ZW

(72)発明者　ジョキ，マーク，エイ．
アメリカ合衆国，４４６２２　オハイオ州，ドーヴァー，ノース　アメリカン　ロード　５９３０
Ｆターム(参考) 3J101 AA16　AA25　AA32　AA42　AA54　AA62　BA02　FA31　GA11

【要約の続き】
の９０％より大きい。大端部面における振れは５０μｉｎ．以下であり、各ころ（４８）の端部面（７２）とつば面（５８）間の接触の中心は０．０２から０．０４ｉｎ．の間である。これらすべてのことが、軸受自身による低トルク性要求と低摩耗性に寄与する。