PETITORIO

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SUPERINTENDENCIA

03 002760 00

SUPERINDUSTRIA Y COMERCIO
Radicacion : 03C32760 00000023 Folios: 98
Fecha (RCB): 2003-01-17 13:46:42
Tramite : 011 PCT-PATENT 379 FASE NACI 411 PRESENTAC
Dependencia: 2020 DIVISION DE NUEVAS CREACIONES

FORMULARIO ÚNICO DE SOLICITUD DE PATENTE
PCT
ENTRADA EN FASE NACIONAL

1 SOLICITUD DE :

[x] Patente de Invención   [ ] Patente de Modelo de Utilidad

[ ] Capitulo I.   [x] Capitulo II.

2 SOLICITANTE (71)	Nombre: DALMINE S.p.A. Dirección: Piazza Caduti 6 Luglio 1.944, 1, I-24044 Dalmine ITALIA Nacionalidad o Domicilio: ITALIA Lugar de Constitución: ITALIA Teléfono: Fax: E-mail:	IDENTIFICACIÓN C.C. [ ] NIT [ ] C.E. [ ] OTRO [ ] Cual: Número:
3 REPRESENTANTE O APODERADO	Sustitución: DORA MARIA GOMEZ DIAZ Nombre: ALIRIO GÓMEZ LOBO Dirección: CARRERA 7 No. 32 - 85 OFICINA 501 Teléfono: 2859058 - 2452462 Fax: 2859919 E-mail: agomezlo@latino.net.co	IDENTIFICACIÓN C.C. [x] NIT [ ] C.E. [ ] OTRO [ ] Cual: Número:17.025.640 Bogotá T.P.: 7.822 C.S. de la J.
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5 PCT (86)

Solicitud Internacional No. PCT/EP01/08188 Fecha: 16 JULIO DE 2001
Publicación Internacional No. WO 02/06715 A1 Fecha: 24 ENERO DE 2002

6 Título (54)

JUNTA ROSCADA INTEGRAL PARA TUBOS

7 Clasificación Internacional (51): F16L 15/00

8 Prioridad	(33) País de Origen	(32) Fecha	(31) Número de Solicitud
SI [x] NO [ ]	ITALIA	17/07/00	MI2000A001614

9 Comprobante de pago No. 03 - 12 Y 03 - 13 Fecha: ENER 02/2003

(10) **ANEX S**

x	Comprobante de pago de la taza de presentación de la solicitud $400.000.00
x	Comprobante de pago por reivindicación de prioridad $112.000.00
	Documento que acredita la existencia y representación legal cuando el solicitante sea persona jurídica
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	Otros

(11) **FIGURA CARACTERÍSTICA:**

(12) **Solicito la concesión de la patente,**

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RECIBO OFICIAL DE CAJA : 03 - 13
FECHA : ENERO 2 DE 2003

CONSIGNACION

DEPOSITANTE	TIPO PAGO	BANCO	CUENTA	No. PAGO	FECHA PGO	Vr. PAGO
ALIRIO GOMEZ LOBO	CONSIGNACION	BANCO POPULAR	050-00110-6	9559318	02/01/2003	400,000.00
ALIRIO GOMEZ LOBO	CONSIGNACION	BANCO POPULAR	050-00110-6	9559319	02/01/2003	112,000.00

CONCEPTO

CANT.	RENTISTICO	CONCEPTO	TOTAL CONCEPTO
1	50005-01-01 SOLICITUDES	1 TRAMITES DE SOL. DE PATENTE DE IN	400,000.00
		TOTAL : $	400,000.00

SON: CUATROCIENTOS MIL PESOS

SUPERINDUSTRIA Y COMERCIO
Radicacion : 03002760 00000000 Folios: 90
Fecha (AMD): 2003-01-17 11:40:42
Tramite : 011 PCT-PATENT 379 FASE NACI 411 PRESENTAC
Dependencia: 2020 DIVISION DE NUEVAS CREACIONES

RESPONSABLE : ______________________

RECIBO DE CAJA APLICADO AL EXPEDIENTE No. ______________________

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RECIBO OFICIAL DE CAJA : 03 - 12
FECHA : ENERO 2 DE 2003

CONSIGNACION

DEPOSITANTE	TIPO PAGO	BANCO	CUENTA	No. PAGO	FECHA PGO	Vr. PAGO
ALIRIO GOMEZ LOBO	CONSIGNACION	BANCO POPULAR	050-00110-6	9559318	02/01/2003	400,000.00
ALIRIO GOMEZ LOBO	CONSIGNACION	BANCO POPULAR	050-00110-6	9559319	02/01/2003	112,000.00

CONCEPTO

CANT.	RENTISTICO	CONCEPTO	TOTAL CONCEPTO
1	50005-01-01 SOLICITUDES	82 INVOCACION DE UNA PRIORIDAD EN NU	112,000.00
		TOTAL : $	112,000.00

SON: CIENTO DOCE MIL PESOS

SUPERINDUSTRIA Y COMERCIO
Radicacion : 03002760 00000000 Folios: 36
Fecha (AMD): 2003-01-17 11:48:42
Tramite : 011 PCT-PATENT 379 FASE NAC1 411 PRESENTAC
Dependencia: 2020 DIVISION DE NUEVAS CREACIONES

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folio 5 y 6 tarjetas
202050
8-Julio 2003

(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
24 January 2002 (24.01.2002)

PCT

(10) International Publication Number
WO 02/06715 A1

(51) International Patent Classification[7]: F16L 15/00, 15/04

(21) International Application Number: PCT/EP01/08188

(22) International Filing Date: 16 July 2001 (16.07.2001)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
MI2000A001614 17 July 2000 (17.07.2000) IT

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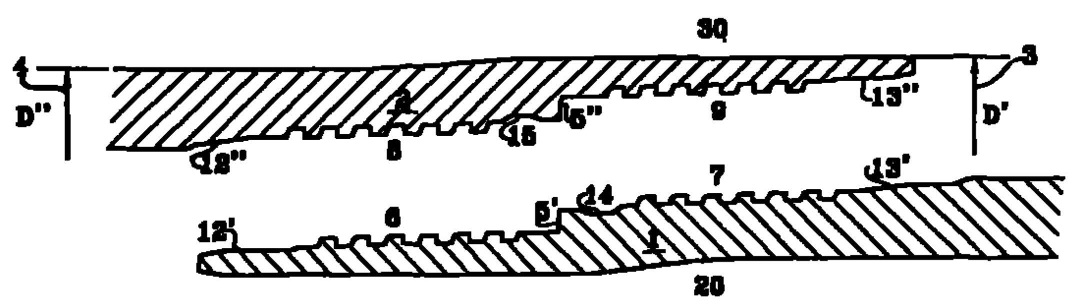
(81) Designated States (*national*): AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, US, UZ, VN, YU, ZA, ZW.

(84) Designated States (*regional*): ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GW, ML, MR, NE, SN, TD, TG).

Published:
— *with international search report*
— *before the expiration of the time limit for amending the claims and to be republished in the event of receipt of amendments*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) Title: PIPE INTEGRAL THREADED JOINT



(57) Abstract: Pipe integral threaded joint formed by a male element (1) and a female element (2) respectively placed on the external surface and on the internal surface of two truncated cone-shaped parts of the threading (6, 7, 8, 9) having the same conicity value and being radially spaced, both being respectively divided by annular shoulders (5, 5') which are parallel and orthogonal with respect to the pipe axis. They are provided two sealing surfaces (12, 12', 13, 13'), one of them being cone-shaped and the other being spherical. After the screwing of the two elements (1, 2) said two annular shoulders (5, 5') are in contact. Each male (1) and female (2) element comprises, respectively along its external and internal surface, a circumference cavity (14, 15) placed between a part of the threading and the annular shoulder (5, 5') in order to allow the expansion of the lubricating fat.

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~~PIPE INTEGRAL THREADED JOINT~~

Field of the Invention

The present invention concerns a pipe integral threaded joint having reduced diametrical dimensions, in particular for pipes employed in the gas or oil fields exploit industry. Said pipes can be used both as ducts for pumping the gas or the oil and as a coating for the wells.

Technical background

The natural gas and oil exploit industry employs pipes having a fixed length which must be connected to each other's ends in order to reach the huge depths where hydrocarbon mines are generally located.

In the most commonly employed drilling technique, wells starting from the earth or the sea surface are drilled in order to reach the oil or gas field. Said wells can have a depth of several thousands meters. During drilling, said wells are coated inside with metal pipes for their entire length. The segments of metal pipe, which have a length of about 10 metres, are connected to each other by means of threaded joints. Therefore, these pipes form a tubular line, having the same diameter for its entire length but in the joints, whose external diameter is generally at least 1 inch (25, 4 mm) bigger than the one of the line.

In order to form a coating for the entire depth of the well several lines are used having, due to mechanical resistance and to the geological characteristics of the site, the smaller diameter the bigger is the depth of the line, in order to form a "telescopic" structure. It follows that, since the diameter at the bottom of the well varies according to the pressure and the flow of the fluids to be extracted, the deeper the well, the bigger its diameter will be on the surface. As unfavourable consequence of the aforesaid, the drilling cost is high and, furthermore, these well coatings require a large amount of material, thus being very expensive. A smaller well diameter also means shorter drilling and coating times. Therefore, it is essential to minimize the well diameter and the coating pipe diameter with an equal amount of extracted fluids.

Once ended the drilling, inside the coated well it is inserted another tubular line, which must pump the gas or the oil out of the underground field. This line, which follows the entire depth of the well and can therefore reach a length of several thousands metres, is also formed by connecting pipes having a length of about 10 metres by means of the aforesaid joints. Usually, also this second kind of line has the same diameter for its entire length but in the joints, whose external diameter is generally bigger, as it happens in the previous case.

In both the aforesaid cases, the pipes are connected by means of threaded joints, which can be integral, and in this case an end of the pipe has a male threading and the other has a female threading, or can be of a coupling type, and in this case both ends of the pipe have a male threading, being connected by means of a female threaded coupling on both sides. Usually, the presence of the joint involves an increase of the external diameter of the line where the joints are; the bigger the external diameter of the joint if compared to the pipe, the bigger are the general dimensions of the line and the bore.

After the oil companies have required to minimize the oil and gas mining costs, remarkable efforts were carried out in order to reduce the diameter of the wells and therefore the diameter of the pipes.

In order to reduce the external diameter of the line and therefore the drilling costs and the amount of required material, they are used threaded joints having reduced diametrical dimensions, which can be divided into three different types, according to the performance required and to the maximum dimensions allowed. A first type, which is often called "semi-flush", is a coupling joint whose external diameter does not exceed over 6% the pipe external diameter. A second type, usually called "near-flush", is an integral joint whose external diameter is about 2-3% larger than the pipe external diameter. A last type, called "flush", is an integral joint whose external diameter corresponds to the pipe external diameter.

The choice among the different types of joints depends on the load on the pipe line, the pressure acting internally and/or externally, its length and its maximum diametrical dimensions allowed with regard to the diameter of the well.

If the joint diameter is reduced it is necessary to find new solutions able to compensate for the lower structural resistance. In fact, in the parts close to the joints the efficacy is necessarily less than in the pipe, because the structural elements such as the threading, the seals and the shoulders are placed in the thickness of the pipe wall and this involves a reduced section in the critical parts of the male or female element.

A minimization of the pipe breaking causes is essential because said breaks, above all after the pipes have been laid in the underground, and it is therefore nearly impossible for the operators to replace a broken joint, can have extremely serious economical consequences and can cause environmental problems, particularly if the oil or gas field contains aggressive elements.

Therefore, in the past much has been done to improve joints and to reach an optimal efficacy level, balancing the different needs, which are sometimes in contrast among them, of minimum dimensions, maximum structural resistance and tightness against the output and/or input of the fluids. In fact, the pipes undergo several forces, namely compression, traction, flexion and the pressure produced by external fluids and/or by fluids circulating inside the pipes.

The joints must also have an optimal resistance to screwing and seizing.

Structural and sealing problems are often worsened by the fluid temperature, by their corrosiveness or by the environmental conditions of the digging areas.

In the past several solutions have been proposed in order to satisfy the aforesaid conditions.

The international patent application WO-A-93/18329 concerns a joint with reduced diametrical dimensions having a central shoulder. Said shoulder is provided, both on the male element and on the female element, with a projection and a cavity which are parallel to the pipe axis, having homologous surfaces to be coupled in order to block the two joint elements. On the shoulder projections two sealing surfaces are provided. The shoulder divides two radially offset threading parts, which are cone-shaped or cone-shaped and cylindrical.

This joint is very efficient, but has a particularly complex structure involving very high production costs.

The European patent application EP-A-767335 concerns a joint with reduced diametrical dimensions provided with a central shoulder and two metal seals placed on the end of the male and the female elements. The shoulder divides two radially offset threading parts which are cone-shaped and cylindrical.

This joint's structure is simpler than the previous one, but it is always relatively complex because of the cone-shaped, cylindrical threading. Furthermore, the two seals at the ends of the joints hermetically seal its inside, thus pressurizing the lubricating fat for the screwing, with possible negative effects on the joint tightness.

Summary of the Invention

A first object of the present invention is to overcome the aforesaid drawbacks of the prior art joints by means of a new "near-flush" joint which, though having the same dimensions of the prior art joints, does not involve the aforesaid problems.

It is a particular object of the present invention to provide for an integral joint having reduced diametrical dimensions and lower production costs, though ensuring high resistance and operational tightness.

A further object of the present invention is to provide an embodiment of this joint which simplifies its installation.

These objects are achieved by a pipe integral joint which, according to claim 1, comprises a male element whose external surface is provided with two radially spaced, truncated cone-shaped threaded portions, said two portions being divided by a first annular shoulder lying on a plane orthogonal to the pipe axis and a female element whose internal surface is provided with two radially spaced, truncated cone-shaped threaded portions, said two portions being divided by a second annular shoulder lying on a plane orthogonal to the pipe axis, each of said male and female elements being provided with two sealing surfaces respectively placed on the opposite ends with respect to the threaded portions, said respective two threaded portions being able to mutually and reversibly screw one inside the other in order to produce a contact between said two annular shoulders, characterized in that the corresponding two threaded portions of each male and female element have the same conicity value and in that a first one of said respective two sealing surfaces of said male and female elements has a conical and a second one has a spherical shape.

Thanks to this embodiment, the joint allows an easy installation of the pipe line, with a reduced risk of damage of the joints, though ensuring an optimal resistance of the pipe line at the joints.

Preferred embodiments of the present invention are described in detail in the dependent claims.

BRIEF DESCRIPTION OF THE DRAWINGS

Further features and advantages of the present invention will become apparent by means of the following detailed description of a preferred but not exclusive embodiment of an integral joint, given as a non limitative example, with reference to the following drawings, wherein:

Figure 1 shows a cross-section along an axial plane of the joint according to the invention, with the two elements in a detached position;

Figure 2 shows the joint of figure 1 with the two elements in an assembled position;

Figure 3 shows an enlarged view of a detail of the joint threading of figure 1;

Figure 4 shows a cross-section along an axial plane of another embodiment of the joint according to the invention, with the two elements in a detached position;

Figure 5 shows the joint of figure 4 with the two elements in an assembled position.

Detailed description of the invention

With a reference to the aforesaid figures, the joint according to the present invention comprises two elements or pipes, namely the male element 1 and the female element 2. The joint is formed by an internal part 20 wherein the fluids flow, for example natural gas or oil or another analogous fluid under pressure, and an external part 30 which can also be filled with different fluids or liquids, usually also pressurized. The external diameter 3 of pipe 2 in the joint area having the maximum diameter with a dimension D' is slightly larger than the external diameter 4 of pipe 2 in the central area far away from the joint, with a dimension D". The two aforesaid pipe areas having different diameters 3 and 4 are connected in a gradual way. Dimension D' is larger than dimension D" of a value equal to or smaller than 3%.

The female element 2 of the joint has an internal threading divided into two portions, or steps, 8 and 9 with a conical generatrix at an angle to the pipe axis. Said two parts 8 and 9 are radially offset and the threading can be perfect for its entire length or can have an imperfect portion. Said embodiment is shown in figures 1 and 2.

In another embodiment of the present invention shown in figures 4 and 5, the threading portions 7 and 8 can respectively have ends with an imperfect threading.

Said two portions 8 and 9 have the same conicity, with values comprised between 6.25 and 12.5%. This range turned out to be optimal, because on the one hand lower values would involve too long threadings and a difficult coupling, and on the other end higher values would involve a reduced number of teeth and therefore an insufficient bearing capacity of the threading.

In the connecting area between the threaded part and the internal surface of the pipe, away from the joint, the female element is provided with an area 12" having a cone-shaped surface with respect to the pipe axis. The conicity of this surface is comprised between 12.5% and 25% in order to ensure a good tightness with the corresponding contacting surface of the male element, thus reducing the sliding time during the screwing step.

This range turned out to be optimal with respect to the threading conicity value and was able to reduce the negative influence of traction loads. In the connecting area between the two threaded portions 8, 9, the female element is provided with a shoulder 5" substantially lying on a perpendicular plane with respect to the joint axis, whose area is not less than 25% of the area of the section of the pipe.

Between the shoulder 5" and the beginning of the threaded part 8 there is provided a cavity 15 which extends along the whole internal circumference of the female element 2. The cavity ensures an expansion tank for the fat used to lubricate the joint, which is present in the two threaded portions 8, 9 and is entrained by the push generated by the sliding of the elements 1 and 2 during the screwing. Said solution limits the development of an excessive fat pressure, caused by the presence of a double metal seal at the ends of the joint, with a following reduced stress of the joint.

At its end the female element 2 is provided with a spherical surface 13" which, contacting a cone-shaped area 13' which it faces after its screwing with the male element 1, ensures the tightness of the joint to the external pressure.

The threading tooth profile is of the "hooked" type, with a loading flank 10 having a negative α angle with values comprised between 0 and $-10°$ and an entering flank 11 having a positive β angle with values comprised between 10 and 30°. These ranges of values have remarkable advantages though maintaining an easy installation of the joint. A loading flank with negative angle allows an efficient coupling of the two elements of the joint and reduces the danger of a possible withdrawal of the joint because of too high traction loads. An entering flank with positive but not too large angle allows an efficient co-operation of the threading to the resistance to compression loads.

In the area of the external surface close to the threaded portion of the female element, the male element 1 is provided with a threading placed in a perfectly reciprocal way, having portions shaped in an exactly analogous way with respect to said threading portion of the female element.

The element 1 is provided with two threaded portions 6, 7 separated by a shoulder 5' and by a circumference cavity 14, between said shoulder 5' and the beginning of the threaded portion 7. The presence of the two fat expansion cavities 14 and 15 allows to limit the pressure increase of the fat after the screwing of the joint, thus avoiding an excessive stress on the joint and improving its functionality.

The connecting area between the external surface of element 1 and the beginning of the threaded portion 7 has a cone-shaped surface 13' with a conicity comprised between 12.5 and 25%. This surface exerts pressure against the surface 13" after the screwing of the joint and the dimensions and tolerances are selected so that the metal-metal contact guarantees the necessary tightness to avoid the entering of the liquid or fluid under pressure outside the joint.

Analogously, at its end, the male element 1 has a spherical surface 12' which, after the screwing with the female element 2, exerts pressure against the conical surface 12' of said female element. Also in this second area a pressure is generated by the metal-metal contact between the two elements which ensures the necessary tightness against the pressure of the fluid inside the pipe. The choice of two areas at the ends of the joint where metal seals are provided which have, respectively, corresponding spherical and conical surfaces, according to the invention, makes the joint less sensitive to pressure loads and turned out to be optimal for thin pipes. In fact, considering the slimness of the ends provided with the sealing surfaces 12' and 13', the pressure, respectively internal or external to the pipe, acting on the aforesaid ends easily causes their bending. Therefore, a spherical sealing surface is able to keep an ptimal contact, unlike a truncated cone-shaped seal, which, in this case, because of the rotation imposed by the

- ~~bending of the end,~~ does not keep the contact on the whole sealing part.

The shape of the thread of the male element 1 is the same as the aforesaid one of the female element 2. Advantageously, the thread has a perfect profile.

In the other embodiment according to the present invention and shown in figures 4 and 5, an end of one or two of both threaded portions 6, 7 of the male element, for example the area 6' close to the shoulder 5', can have a thread with an imperfect profile. The corresponding part 8 of the female element on the side of the shoulder 5" facing area 6" has a perfect threading. The area 8' at the end of the threaded portion 8 axially opposite to the area 6', namely the one close to the sealing surface 12", also has a thread with an imperfect profile. However, this area can also be provided with a perfect thread.

The same kind of proceeding can also be applied to the threaded portions 7 and 9, wherein portion 7 is provided with a perfect thread on the side of shoulder 5' and part 9 is provided with an area 9' having an imperfect thread on the side of shoulder 5". In combination or alternatively to this embodiment, also part 7 can be provided with an area 7' having an imperfect thread on the side of the sealing surface 13' and part 9 can be provided with a perfect thread on the side of the sealing surface 13".

From the aforesaid it is clear which are the advantages of the joint according to the present invention, which, though having diametrical dimensions smaller than those of the "semi-flush" type of the prior art and only slightly larger than those of the "flush" type, ensures optimal performances and working efficiency.

CLAIMS

1. Pipe integral threaded joint comprising a male element (1) provided on its external surface with two radially spaced, truncated cone-shaped threaded portions (6, 7), said two portions (6, 7) being divided by a first annular shoulder (5') lying on a plane orthogonal to the pipe axis and a female element (2) provided on its internal surface with two radially spaced, truncated cone-shaped threaded portions (8, 9), said two portions being divided by a second annular shoulder (5") lying on a plane orthogonal to the pipe axis, each of said male (1) and female (2) elements being provided with two sealing surfaces (12', 13', 12", 13") respectively placed at axial ends which are opposite to the corresponding threaded portions (6, 7, 8, 9), said corresponding two threaded portions (6, 7, 8, 9) being able to screw mutually and reversibly one inside the other in order to produce a contact between said two annular shoulders (5', 5"), characterized in that the respective two threaded portions (6, 7, 8, 9) of each male (1) and female element (2) have the same conicity value and in that a first one of said respective two sealing surfaces (12', 13', 12", 13") of said male and female elements has a conical shape and a second one has a spherical shape.

2. Joint according to claim 1, characterized in that at least one of said male (1) and female (2) elements comprises along its entire surface a circumference cavity (14, 15) between one of said two threading parts and the corresponding annular shoulder (5', 5"), said cavity (14, 15) being adapted to receive the expansion of the lubricating fat of the joint.

3. Joint according to claim 2, characterized in that each of said male (1) and female (2) elements comprises along its entire surface said circumference cavity (14, 15), placed between one of said two threaded portions and the corresponding annular shoulder (5', 5").

4. Joint according to claim 3, characterized in that the dimension (D') of the external diameter (3) of said female element (2) at the joint is larger than the dimension (D") of the external diameter (4) of the same female element (2) of a value equal to or smaller than 3%.

5. Joint according to claim 4, characterized in that said corresponding two threaded portions (6, 7, 8, 9) of said male (1) and female (2) elements each have

the same conicity value, comprised between 6.25 and 12.5%.

6. Joint according to claim 5, characterized in that said corresponding conical sealing surfaces (13', 12") of said male (1) and female (2) elements have a conicity value comprised between 12.5% and 25%.

7. Joint according to claim 6, characterized in that the profile of the threading tooth has a loading flank (10) with a negative (α) angle and an entering flank (11) with a positive (β) angle.

8. Joint according to claim 8, characterized in that said negative (α) angle has a value comprised between 0 and –10° and said positive (β) angle has a value comprised between 10 and 30°.

9. Joint according to claim 8, characterized in that said corresponding two threaded portions (6, 7, 8, 9) of each of the male (1) and female (2) elements is provided with a perfect thread for its entire length.

10. Joint according to claim 8, characterized in that at least one of said two threaded portions (6, 7, 8, 9) of at least one of the male (1) or female (2) elements is provided with a first end (6', 7') with an imperfect thread.

11. Joint according to claim 10, characterized in that the corresponding threaded portion of the other of said at least one of male (1) and female (2) elements is also provided with a second area (8', 9') having an imperfect thread placed at the axially opposite end with respect to said first end.

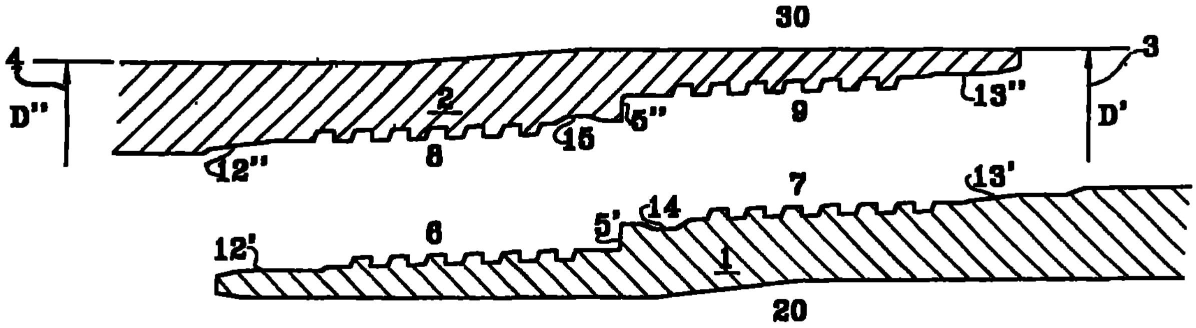

FIG. 1

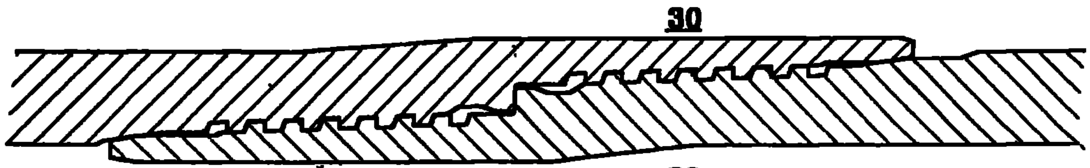

FIG. 2

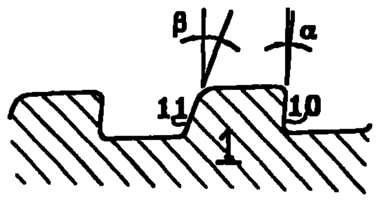

FIG. 3

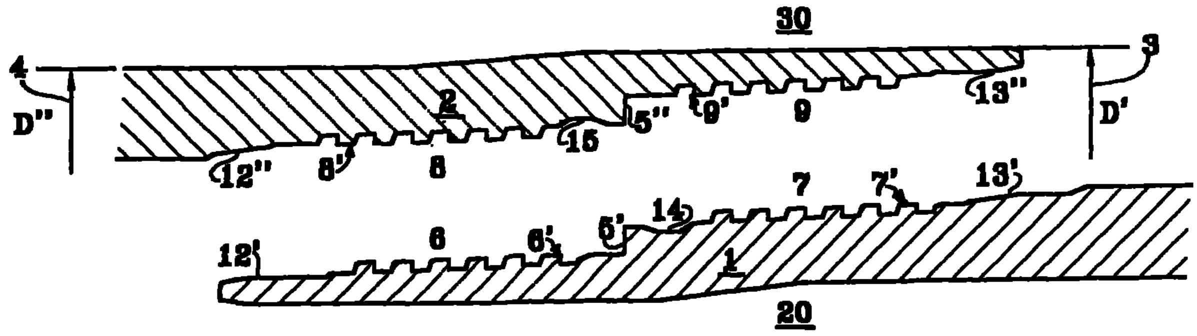

FIG. 4

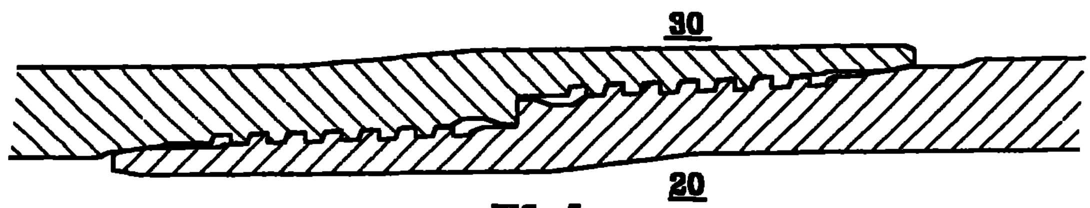

FIG. 5

INTERNATI NAL SEARCH REP RT

In ... nal Application No
F.../EP 01/08188

13

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F16L15/00 F16L15/04

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 F16L

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, WPI Data, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

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Y	US 3 870 351 A (MATSUKI NORIO) 11 March 1975 (1975-03-11) figures 1,2	1-11
A	EP 0 767 335 A (VALLOUREC OIL & GAS, SUMITOMO METAL IND (JP)) 9 April 1997 (1997-04-09) cited in the application figures 1,3	1-11

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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
10 December 2001	20/12/2001

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	Schroeder, R

INTERNATI NAL SEARCH REP RT

Int nal Application No
PCT/EP 01/08188

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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A	EP 0 916 883 A (SUMITOMO METAL IND) 19 May 1999 (1999-05-19) figures 1B,,3; table 3	5,6

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United States Patent [19]

Matsuki

[11] **3,870,351**

[45] **Mar. 11, 1975**

[54] **THREADED TUBE JOINT STRUCTURE FOR CASING, PARTICULARLY OIL WELL TUBING**

[75] Inventor: **Norio Matsuki**, Osaka, Japan

[73] Assignee: **Sumitomo Metal Industries Limited**, Osaka, Japan

[22] Filed: **Mar. 30, 1973**

[21] Appl. No.: **346,302**

[30] **Foreign Application Priority Data**

Mar. 31, 1972 Japan 47/32953

[52] **U.S. Cl.** 285/334, 285/351, 403/343

[51] **Int. Cl.** E21b 17/08, F16l 15/00

[58] **Field of Search** 285/333, 334, 351, 355, 285/DIG. 10, 332.2; 403/343

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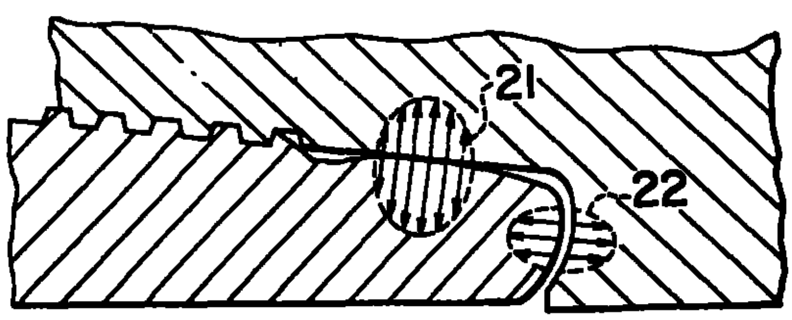
Primary Examiner—Werner H. Schroeder
Assistant Examiner—Moshe I. Cohen
Attorney, Agent, or Firm—Watson, Cole, Grindle & Watson

[57] **ABSTRACT**

A threaded tube joint structure for casing, particularly oil well tubing, which is more perfect in the sealing performance than any other joint. This sealing performance is obtained with a first metal-to-metal contact seal between the outer surface of a lip of a male threaded member and a corresponding inner surface of a female threaded member and a second metal-to-metal contact seal between the end face of the lip and a corresponding face of the female threaded member. This structure permits the realization of the intended alignment of the reference axes of the abutting faces of the second seal because these faces resist deviation from their fixed positions relative to each other.

7 Claims, 8 Drawing Figures



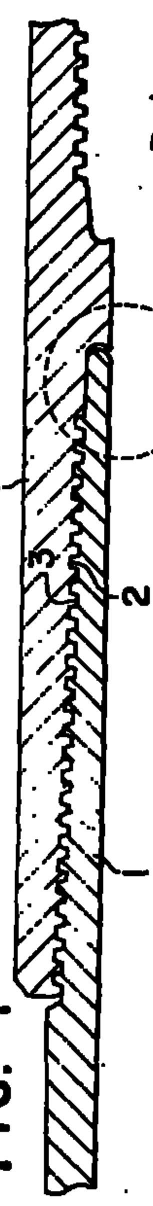


FIG. 1

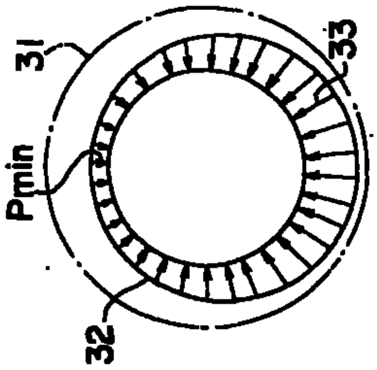


FIG. 4

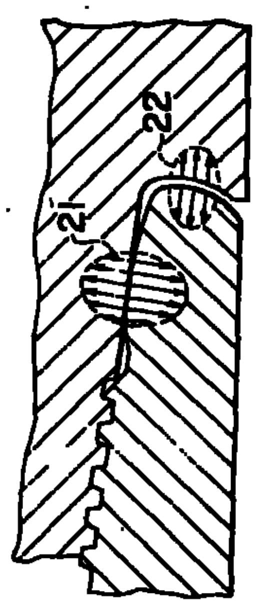


FIG. 3

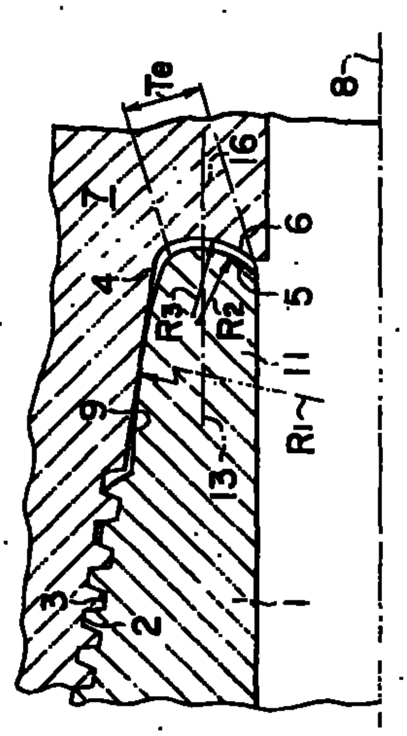


FIG. 2

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FIG. 5

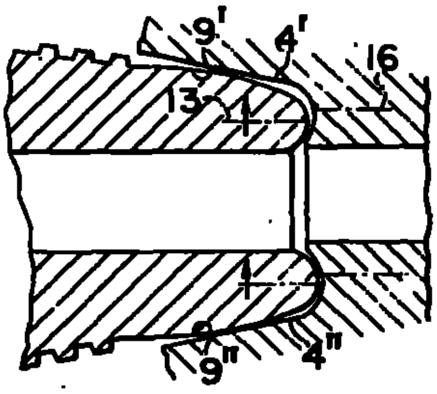


FIG. 6

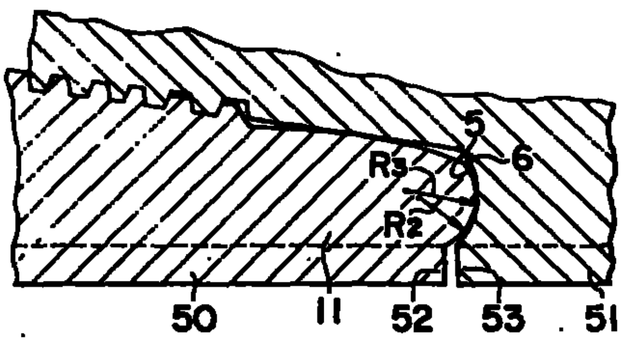


FIG. 7

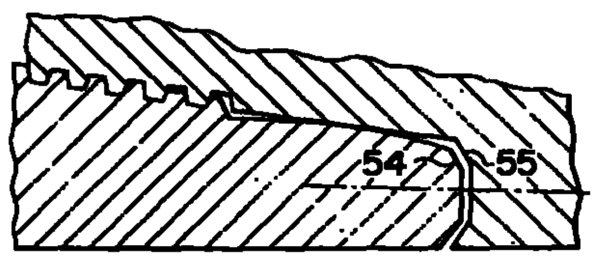
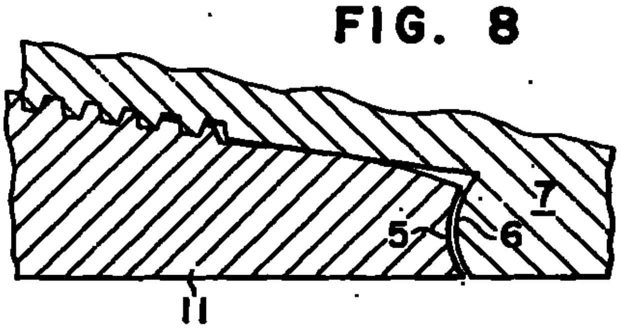


FIG. 8



THREADED TUBE JOINT STRUCTURE FOR CASING, PARTICULARLY OIL WELL TUBING

This invention relates to threaded connections for tubular products such as are extensively used in the oil and gas industry in connection with the exploitation of underground deposits of natural resources such as oil and gas.

The conditions under which tubing is used and the requirements of such tubing have been getting progressively severe through the years. Natural oil and gas wells currently being driven have extended to depths in excess of 5,000 meters. The type of the thread of tubular products employed for such wells has undergone changes through many years and various new types of threaded joints have been proposed and actually used. It will be understood that in a string of tubing extending to a depth in excess of 5,000 meters, the uppermost tube joint must withstand the total weight of the tubing even through no external force is exerted and the tubing is simply suspended. Since the ordinary tubing commonly used today weighs from 6 to 50 kilograms per one meter of the tubing length, the tremendous loads which must be carried by the threads will be appreciated. If the tubing is also used for the purpose of drilling the well, the coupling must be able to withstand high torque load in addition to the tensile load.

One difficulty encountered with tubing is that when it is subjected to a great axial tension load, the coupling between mating threaded elements tends to become loose, adversely affecting the leak-tight property of the coupling. Therefore, improvements in connection with the seal of threaded tube joints have been desired.

It has been known and is disclosed, for instance, in U.S. Pat. No. 3,210,096 and U.S. Pat. No. 3,224,799 to seal against leakage between mating threaded elements, i.e., through the threaded tube joint, by providing upon make-up of the coupling a metal-to-metal seal with interfitting or interengaging faces of male and female threaded elements of the coupling. In this joint structure, the end face of the end portion (or lip) of the male or female threaded element constitutes the metal-to-metal seal line in a plane normal to the normal tubing axis, with the mating face of the latter element serving as a stop shoulder to prevent excessive make-up. It is also known and disclosed, for instance, in British Patent specification No. 1,220,856, U.S. Pat. No. 3,489,437, U.S. Pat. No. 3,359,013 and U.S. Pat. No. 3,050,318 to make the end face of the male element a concave frusto-conical surface including the tubing axis and the mating face of the female element a convex frusto-conical surface so that the convex face of the female element acts to enlarge the concave face of the male member depending upon the magnitude of the make-up force. Further, it is known and disclosed, for instance, in U.S. Pat. No. 2,992,019 and British Patent specification No. 1,220,856 to form upon make-up of the joint a metal-to-metal seal between a conical pin surface of the lip of a male threaded member and the corresponding conical box surface of a female member, either one of the interengaging surfaces having a straight profile and the other having an arcuate profile.

However, in these known threaded joint structures the end face of the male threaded member and the mating face of the female threaded member either lie in a plane normal to the tubing axis or in a conical surface including the tubing axis. Therefore, the contact pressure obtainable by interengaging these surfaces is relatively low and tends to be insufficient even if these surfaces are urged against each other with a very strong make-up force because the contact obtainable is no other than an area contact. Such low contact pressure at the end of the male threaded member is insufficient to seal against leakage of fluid. Also, it is hardly expected to obtain the designed alignment of the end face of the male threaded member and the mating face of the female threaded member by the mere make-up of the joint. Since uniform sealing pressure over the metal-to-metal seal is not ensured and nonuniform distribution of the sealing pressure is anticipated, it is very likely that leakage will occur in a low sealing pressure zone. It is also well known in the art that in order to ensure leakage-tightness against oil and natural gas with a threaded coupling provided with a metal-to-metal seal, it is necessary to have a uniform sealing pressure over the entire seal line and that the sealing pressure should be higher than the pressure of the fluid to be sealed against. Thus, it is very important that the sealing pressure is uniform over the entire seal line and sufficiently high. Further, an excessively high local sealing pressure is undesirable because it is prone to cause seizure of metal surfaces due to friction, thus leading to leakage. Accordingly, it is desired to be able to correct circumferential distribution of the sealing pressure so as to render it uniform. If this can be done, a most effective seal both with respect to the prevention of seizure and to the fluid-tightness can be obtained.

With the above considerations in mind, in the present invention a particular importance is attached to the coaction of the end face of the male threaded member and mating face of the female threaded member and to the shape or profile of the metal-to-metal contact surfaces.

In accordance with the invention, a first metal-to-metal contact seal is formed upon make-up of the joint between an outer surface of a thread-free end portion or lip of the male threaded member and a corresponding inner surface of the female threaded member, the outer surface of the lip extending forwardly of the forward end of the thread and being substantially spherical or having a substantially arcuate profile and the corresponding inner surface of the female threaded member having a straight or convex profile. At the same time, a second metal-to-metal contact seal is formed between the end face of the lip of the male member and a corresponding face of the female member, the end face of the lip substantially having a profile of a convex or concave surface of revolution and the corresponding face of the female member having a profile substantially complementary to that of the mating end face of the male member so that an extremely high contact pressure is developed at the end of the make-up by virtue of the principles of the contact between convex and concave surfaces. In addition, the accomplishment of the intended alignment of the reference axes of the interengaging surfaces is facilitated because those surfaces substantially have the nature of surfaces of revolution and accordingly resist deviation from their fixed positions relative to each other, which usefully affects the afore-mentioned first seal, that is the seal between the outer surface of the lip of the male member, and the corresponding inner surface of the female member, by reducing the flexure of the lip toward the tubing axis. This has the effect of redistributing and equalizing the sealing pressure over the sealing area of the first seal.

It will be understood from the afore-mentioned principles in the make-up of the threaded coupling that the roles of the end face of the male threaded member and the corresponding face of the female member are very important.

One of the roles that is important is to ensure uniform and high contact pressure (as indicated at 22 in FIG. 3) of the metal-to-metal contact at the end face of the male member. To this end, this contact should be nearly a line contact. Secondly, the lip 11 of the male member should be tough and not flex inwardly or buckle even if the make-up force becomes excessive. Thirdly, upon the striking of the end face of the male member against the mating face of the female member the sealing pressure of the already formed first seal should not be reduced nor its uniformity disturbed. Instead, it is desirable that the sealing pressure is increased and its uniformity is improved. The mating faces that can meet the above requirements must be convex and concave surfaces of revolution.

FIG. 1 is a sectional view showing principally a left half of a tube joint for explaining an embodiment of the present invention.

FIG. 2 is an enlarged sectional view of a portion encircled with the dotted line in FIG. 1.

FIGS. 3, 4 and 5 are, respectively, views for explaining the action of the seal pressure at the time of making up a coupling.

FIGS. 6, 7 and 8 are respectively enlarged sectional views, each showing a different embodiment of the present invention and corresponding to FIG. 2.

As is most clearly shown in FIG. 2, the thread of the coupling 7 terminates in a non-threaded tapered surface, which covers an axial dimension corresponding to 2 to 3 times the thread pitch, has the same taper as the thread and terminates in turn in a shoulder 6, which is adapted to engage with the end face 5 of the male threaded member. On the other hand, the threaded portion of the male threaded member 1 terminates at its forward end in a lip portion 11, which also covers an axial dimension corresponding to 2 to 3 times the thread pitch. From the ground as will be described hereinafter, the outer surface of the lip 11 has an arcuate profile with a relatively large radius of curvature R_1. As is also shown in the same figure, the end face of the male threaded member having the male thread 2 has a convex arcuate profile with the radius of curvature R_2, and the mating shoulder face 6 of the coupling 7, or female threaded member having the female thread 3, has a concave arcuate profile with a radius of curvature R_3. The centers of the convex and concave surfaces with respective radii R_2 and R_3 are located at suitable positions within the male threaded member. In the case of the FIG. 2 structure, the center of the convex surface of R_2 is positioned below that of the concave surface of R_3, and R_2 is preferably no greater than R_3.

The outer surface 9 of the lip 11 of the tubing section 1 and the inner taper surface 4 of the coupling 7, both of which approach each other as the male threaded member and female threaded member are interthreaded by the make-up, are adapted to be brought into metal-to-metal contact with each other before the end face 5 of the male threaded member is seated in the mating shoulder face 6 of the female threaded member. To this end, the outer surface 9 of the lip is made spherical with the radius R_1 such that its crown is located at a position about midway between the forward end of the male thread and the end of the male threaded member. In this manner, a metal-to-metal contact is first brought about between the thread-free pin and box surfaces 9 and 4, and then a second metal-to-metal contact is produced between the end face 5 of the male threaded member and the mating shoulder face 6 of the female threaded member. With further make-up, the metal-to-metal sealing pressure builds up at a suddenly increased rate, and the sudden increase of the make-up torque can be detected by a torque gauge to automatically stop the make-up operation upon reaching of a predetermined make-up torque, thereby preventing the buckling and plastic deformation of the lip 11.

In the novel joint structure according to the invention, the pitch line 13 of the male threaded member and the pitch line 16 of the female threaded member are perfectly aligned by the application of the make-up force virtue of abutting convex and concave surfaces substantially having the nature of a surface of revolution to prevent deviation from the fixed relative position, so that a firm and powerful metal-to-metal contact between these surfaces 5 and 6 can be ensured. In particular, by setting R_2 to be equal to or smaller than R_3 a circular line contact can be ensured between these contact surfaces, so that a metal-to-metal seal of an extraordinarily high sealing pressure, can be obtained. Generally, R_2 and R_3 are suitably 1.0 to 2.5 times the thickness Te of the end of the male threaded member, and R_1 is suitably 50 to 300 millimeters. With such values, the second seal, that is, seal 22 at the end of the male threaded member, can be formed as shown in FIG. 3. Further, with a convex or concave surface of revolution at the end face of the lip 5, the lip can withstand excessive make-up forces and will not flex toward the tubing axis (downwardly in connection with FIG. 3) to lose the metal-to-metal seal effect.

Furthermore, the end face of the male threaded member receives reaction force from the mating face of the female threaded member, and this is effective in correcting the nonuniformity of the sealing pressure distribution in the main seal due to machining errors, so that it is possible to obtain a threaded tube joint of a double seal structure having a firmer and more uniform seal. To discuss this respect in more detail, the cutting of the thread and the machining of the seal surface are usually done with different tools. Therefore, there can be more or less discrepancy between the axis of the thread and the axis of the seal surface. The eccentricity between the thread pitch plane 31 and seal surface pitch plane 32 results in nonuniformity of the seal pressure distribution as shown in FIG. 4, which shows an extreme example. In this case, the effective sealing pressure is the minimum contact pressure P_{min}. The inner surfaces 4 and 6 of the female member, however, are often machined with the same tool because of their inferior processibility. In such case, it is easy to obtain good alignment of the axes of the thread pitch plane and seal surface pitch plane. Accordingly, the powerful forced contact between the faces 5 and 6 of the male and female members due to the make-up causes automatic displacement of the end 5 of the male member, for instance in the direction of the arrows in FIG. 5, thus increasing the relatively low sealing pressure over the area indicated at 9' and 4' and reducing relatively high sealing pressures over an opposite area as indicated at 9'' and 4''. In this way, the nonuniformity of the sealing pressure-distribution as indicated at 33 in FIG. 4 can be automatically corrected, and uniform distribution of the sealing pressure can be ensured.

Still further, by forming a seal film with a thickness of 0.1 to 0.3 millimeter (100 to 300 microns) on the male member end face 5 or female member shoulder face 6 by applying thereto liquid resin such as tetrefluroethylene by means of spraying or with a suitable brush and heating the coating with an infrared heater or other heater to a temperature of 60° to 80°C for a short time, both metal-to-metal contact and metal-to-resin contact can be formed between the faces 5 and 6, thereby increasing the seal area and hence rendering the seal effect more perfect.

FIG. 6 shows another embodiment, which is applicable where the tubing section 1 has a thickness of 8 millimeters or more. It is the same as the preceding embodiment except that the tubing section 1 and coupling 7 have respective inner peripheral portions 50 and 51, whose opposing surfaces 52 and 53 lie in planes normal to the tubing axis, or in conical surfaces.

FIG. 7 shows a further embodiment, in which the end face of the male member and the mating face of the female member have respective profiles constituted by a plurality of straight line segments and approximating the arcuate profiles of R_2 and R_3 in the embodiment of FIG. 2.

FIG. 8 shows a still further embodiment, in which the end of the male member has a concave face and the mating shoulder of the female member has a convex face.

The embodiments of FIGS. 6 to 8 have the same effect as the embodiment of FIG. 2.

What is claimed is:

1. A threaded tube joint for casing, particularly oil well tubing, including male and female threaded tube members, each having a substantially cylindrical wall and said joint having a structure having:

a first metal-to-metal contact seal formed upon make-up of the joint between an outer surface of a thread-free end portion of the male threaded tube member having a longitudinally extending reference axis within its wall and a corresponding inner surface of the female threaded tube member having a longitudinally extending reference axis within its wall, said outer surface of the male member having a substantially spherical profile, said inner surface of the female member having a straight profile; and

a second metal-to-metal contact seal formed also upon make-up of the joint between the end face of the wall of said end portion of the male member and a corresponding face of the wall of the female member, one of said faces of the male and female members substantially having with respect to its reference axis a profile of a convex surface of revolution for aligning the reference axis of the male member with the reference axis of the female member, and the other of said faces of the male and female members substantially having with respect to its reference axis a profile of a concave surface of revolution so as to be complementary to that of the one face.

2. The threaded tube joint of the structure having the first and second seals as claimed in claim 1, wherein the surface of the male member and the mating surface of the female member constituting said second seal have respective radii of curvature R_2 and R_3, R_2 being at least no greater than R_3, said radii of curvature both being within a range of 1.0 to 2.5 times the average thickness of said end portion of the male member.

3. The threaded tube joint according to claim 1, wherein one of the interengaging surfaces of said second seal is provided with a coating of such resin material as tetrafluoroethylene.

4. The threaded tube joint according to claim 1, wherein the male and female members are large in wall thickness and have respective inner peripheral portions facing each other at a predetermined gap, the facing surfaces of said inner peripheral portions lying in planes normal to the tubing axis.

5. A threaded tube joint of a structure having the first and second seals as claimed in claim 1, wherein alignment of the male member with the female member by the second seal prevents a flexure of the male member toward the tubing axis, caused by the action of the first seal at the lip of the male member, and equalizes the sealing pressure over the sealing area of the first seal.

6. The threaded tube joint according to claim 1, wherein the interengaging surfaces of said second seal have respective profiles each constituted by a plurality of straight line segments approximating arcuate profiles and each corresponding to the other of said interengaging surfaces.

7. The threaded tube joint according to claim 1, wherein the male and female members are large in wall thickness and have respective inner peripheral portions facing each other with a perdetermined gap between them, the facing surfaces of said inner peripheral portions both lying within complementary conical planes.

* * * * *

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US005462315A

United States Patent [19]

Klementich

[11] **Patent Number:** 5,462,315

[45] **Date of Patent:** Oct. 31, 1995

[54] **STABILIZED CENTER-SHOULDER-SEALED TUBULAR CONNECTION**

[75] Inventor: Erich F. Klementich, Houston, Tex.

[73] Assignee: Marubeni Tubulars, Inc., Houston, Tex.

[21] Appl. No.: 913,105

[22] Filed: Jul. 14, 1992

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 848,508, Mar. 9, 1992, Pat. No. 5,415,442.

[51] Int. Cl.[6] F16L 35/00

[52] U.S. Cl. 285/24; 285/331; 285/334; 29/237

[58] Field of Search 285/333, 334, 285/355, 390, 24; 29/237; 411/411, 306, 414, 423

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5,154,452	10/1992	Johnson	285/333
5,154,452	10/1992	Johnson	285/355

FOREIGN PATENT DOCUMENTS

2308854	11/1976	France.
2599811	12/1987	France.
8404352	11/1984	WIPO.

Primary Examiner—Dave W. Arola
Attorney, Agent, or Firm—Conley, Rose & Tayon

[57] **ABSTRACT**

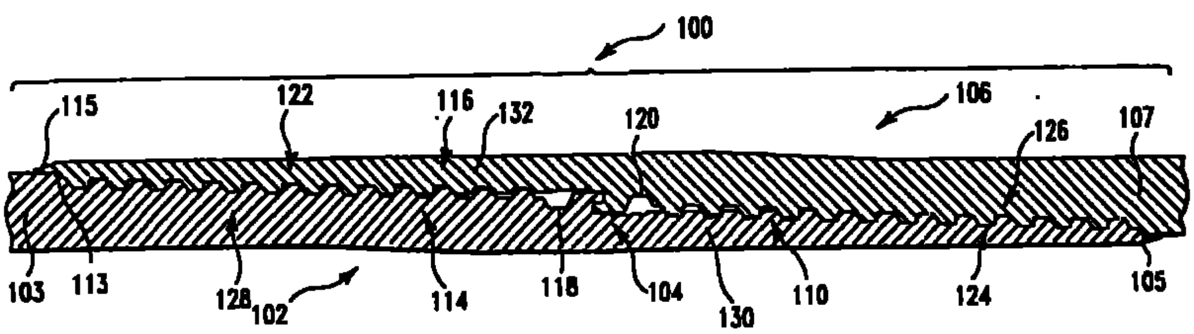
The present invention pertains to a tubular connection having a stabilized center-shoulder seal. The stabilization of the center-shoulder seal is achieved using at least one run-out section of thread adjacent to the center-shoulder seal configuration of each connection member. The run-out section of thread is typically used in combination with a run-in section of thread on the corresponding connection member.

Preferably, the center shoulder seal configuration of each connection member is positioned between a run-out thread section and a run-in thread section. (The corresponding connection member will comprise the corresponding sections of run-in and run-out threads, respectively, so there is an interengaging set of run-in and run-out threads). Even more preferably, in addition to the connection having one set of run-out and run-in thread sections adjacent to the center-shoulder seal, the connection has a second set of run-out and run-in threads adjacent to the exterior edges of the connection (A double run-out, run-in stabilization of the center-shoulder seal). The double set of run-out, run-in sections of thread provides the most stable connection.

The most preferred center-shoulder seal design is a "locked double shoulder" design, wherein the cross section of the assembled center-shoulder area exhibits an interlocking double shoulder configuration.

The most preferred thread profile is a single hook thread profile wherein the load flank angle is a negative angle and the stab flank angle is about zero degrees. The preferred thread profile comprises chamfered stab flanks at least at locations which will come into contact upon initial telescoping of a connection pin member into a connection box member. The thread is typically a variable width thread having a progressively increasing thread width throughout a continuous section of thread.

102 Claims, 10 Drawing Sheets



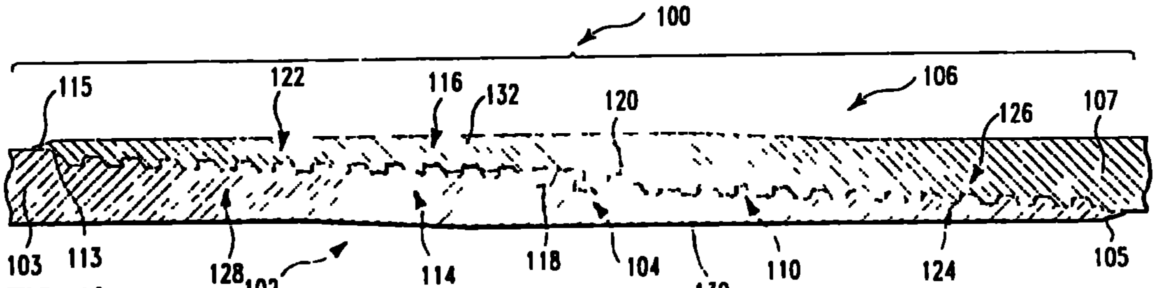


FIG. 1A

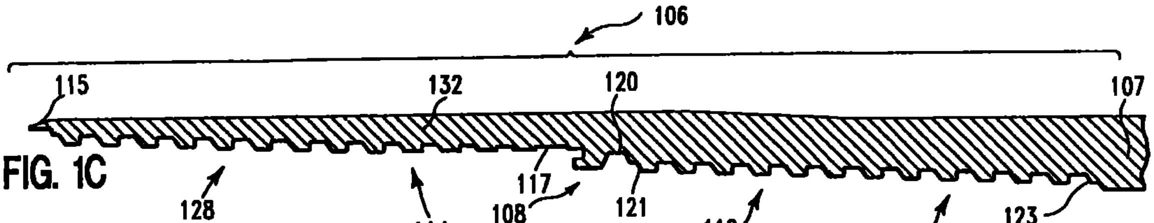


FIG. 1C

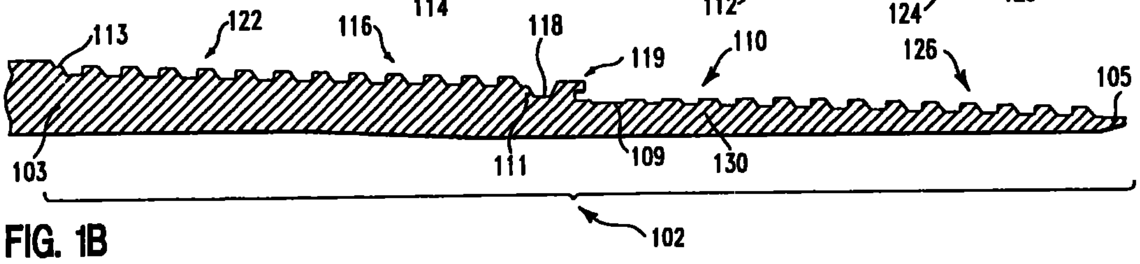


FIG. 1B

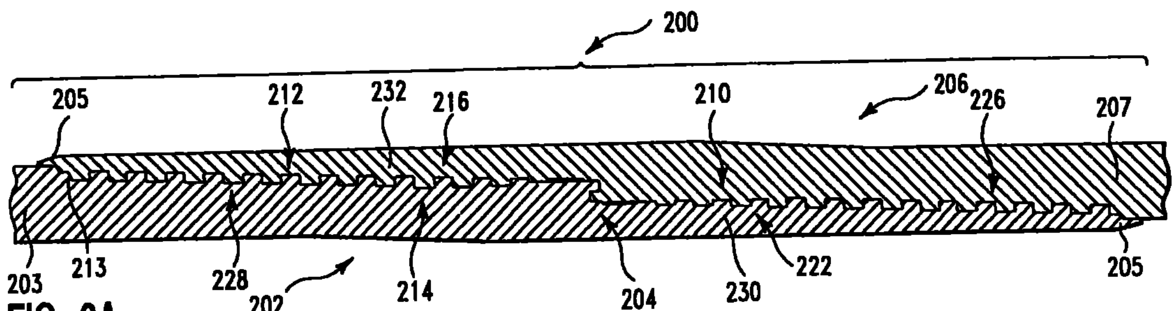


FIG. 2A

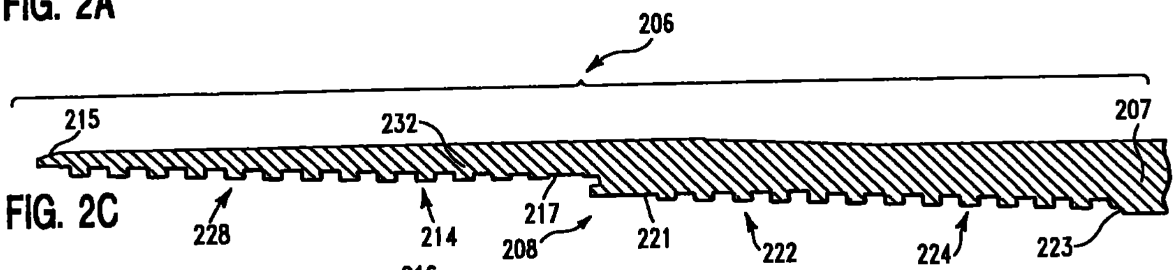


FIG. 2C

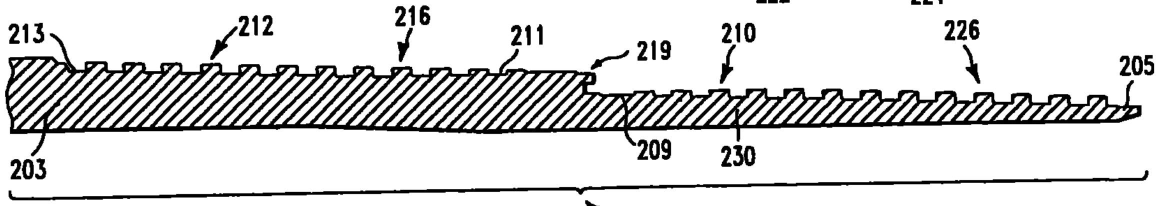


FIG. 2B

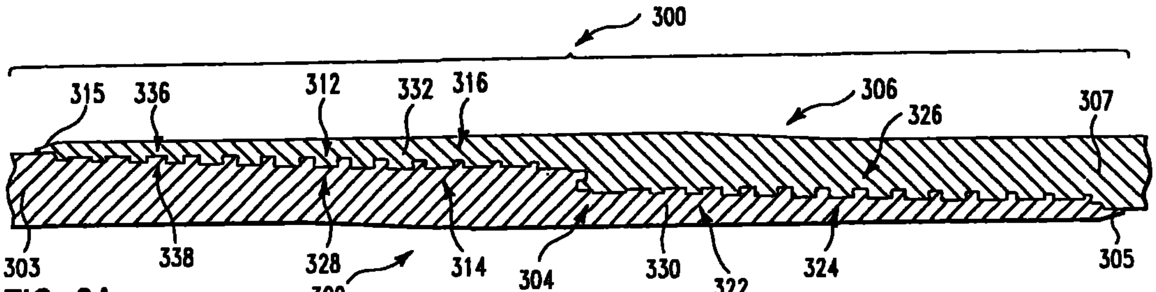


FIG. 3A

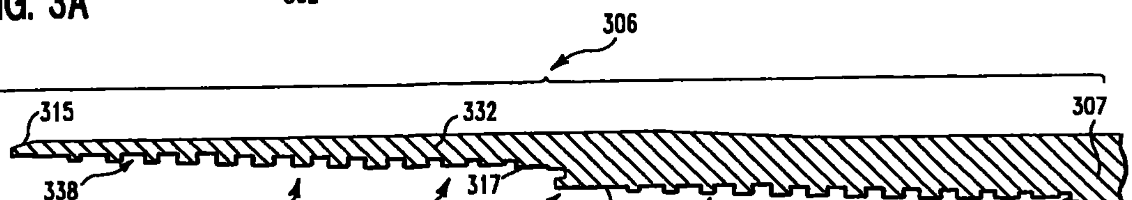


FIG. 3C

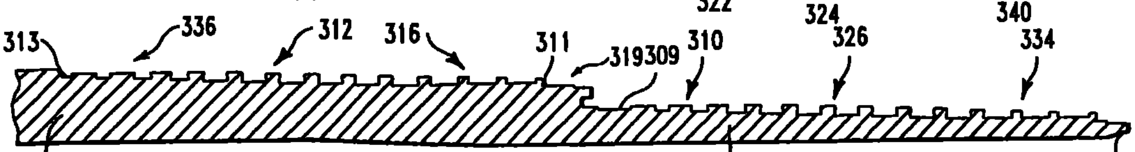


FIG. 3B

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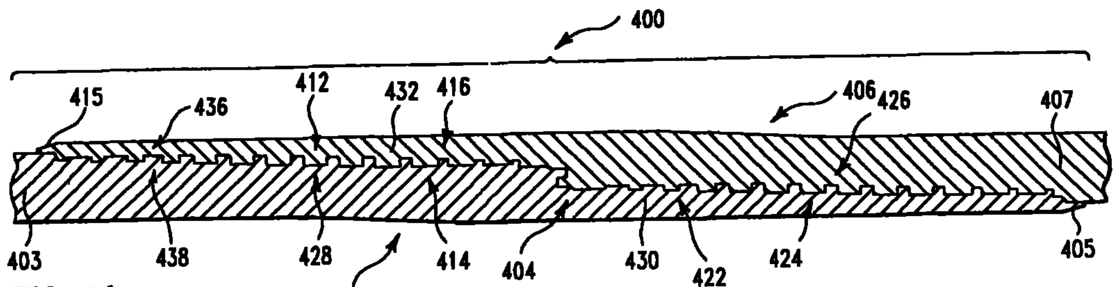

FIG. 4A

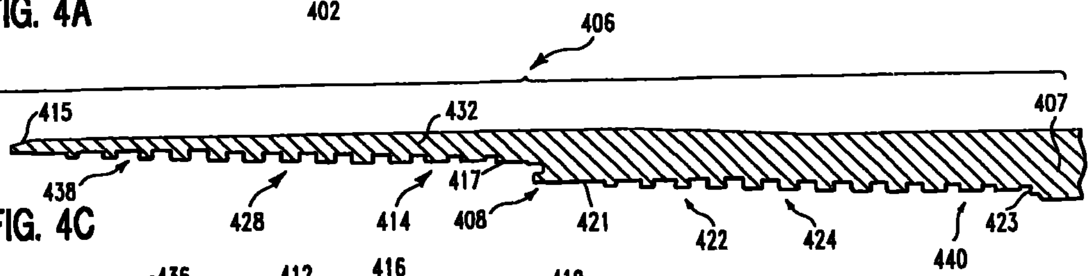

FIG. 4C

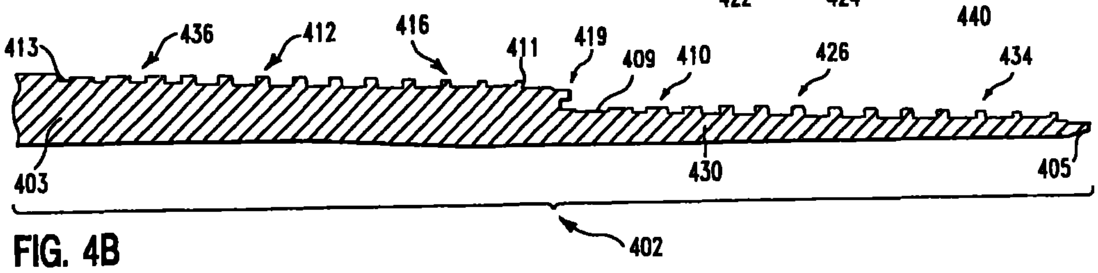

FIG. 4B

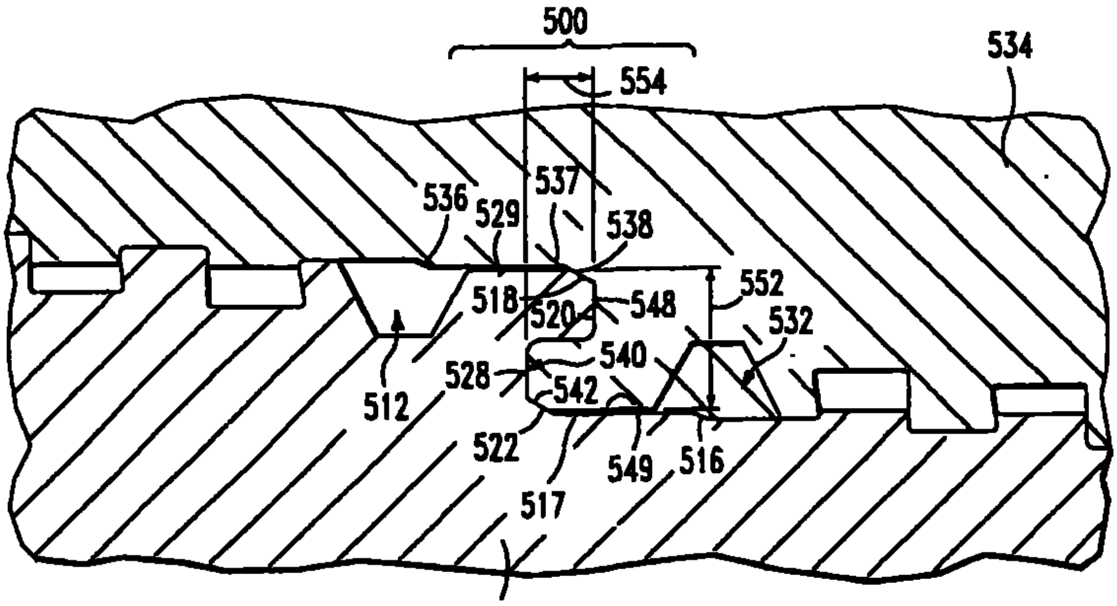

FIG. 5A

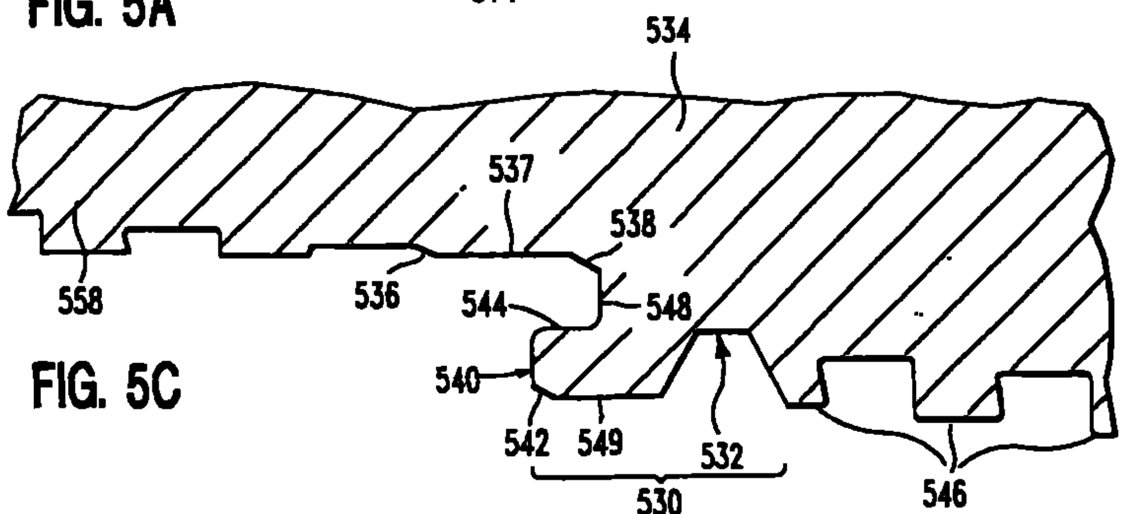

FIG. 5C

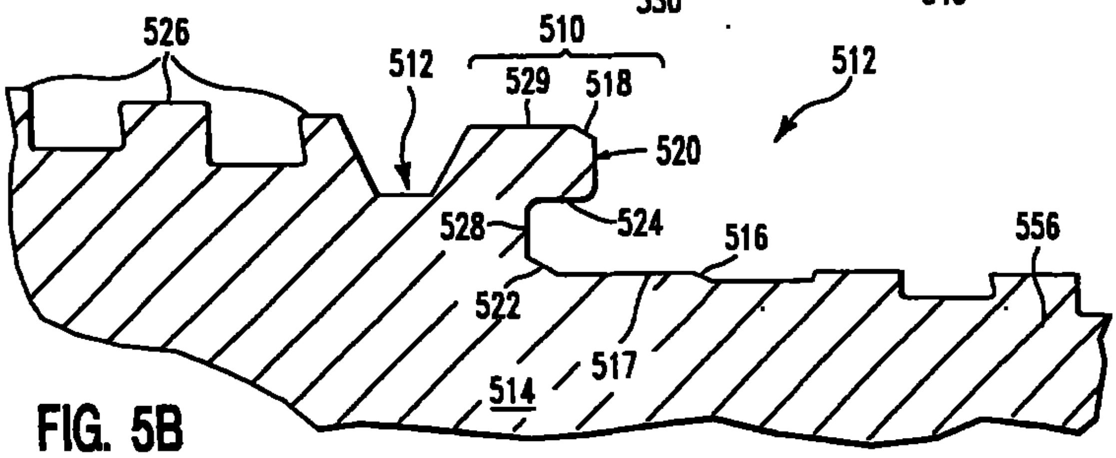

FIG. 5B

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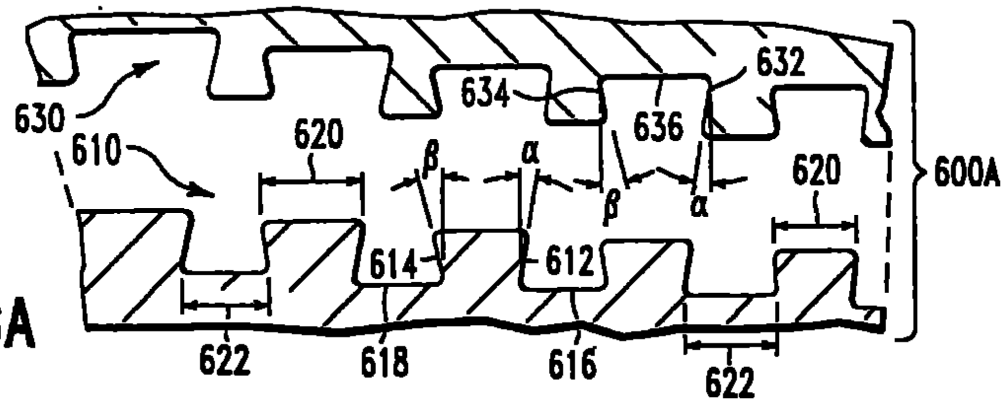

FIG. 6A

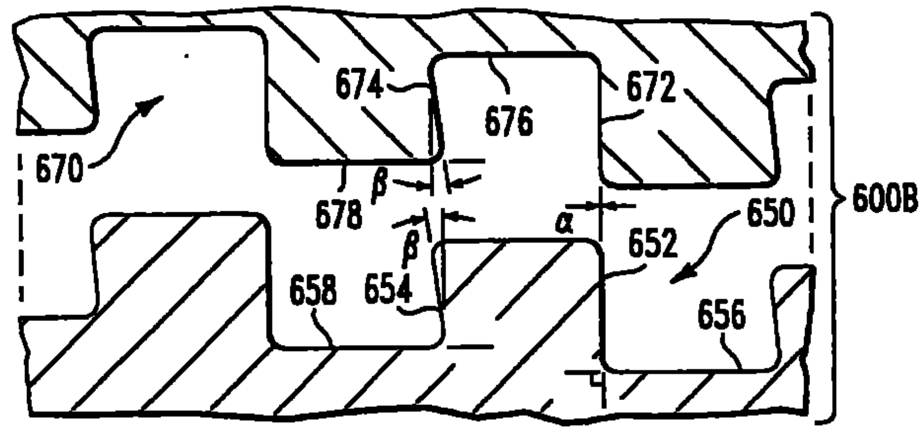

FIG. 6B

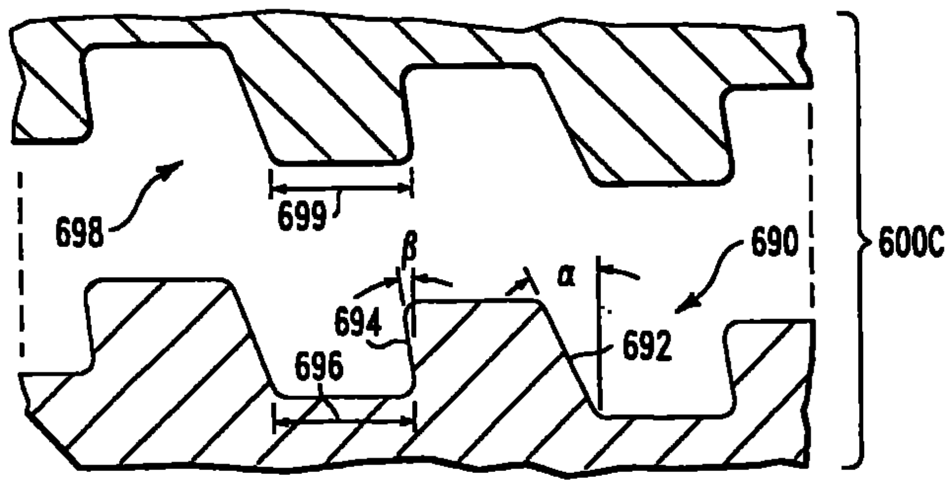

FIG. 6C

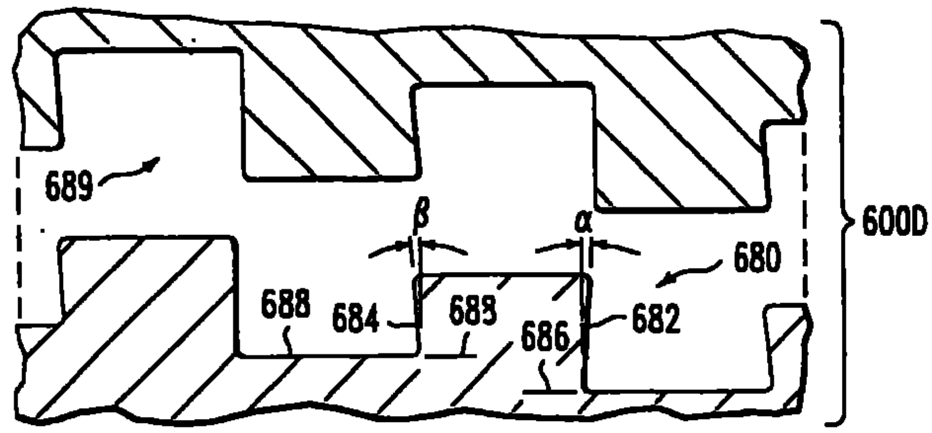

FIG. 6D

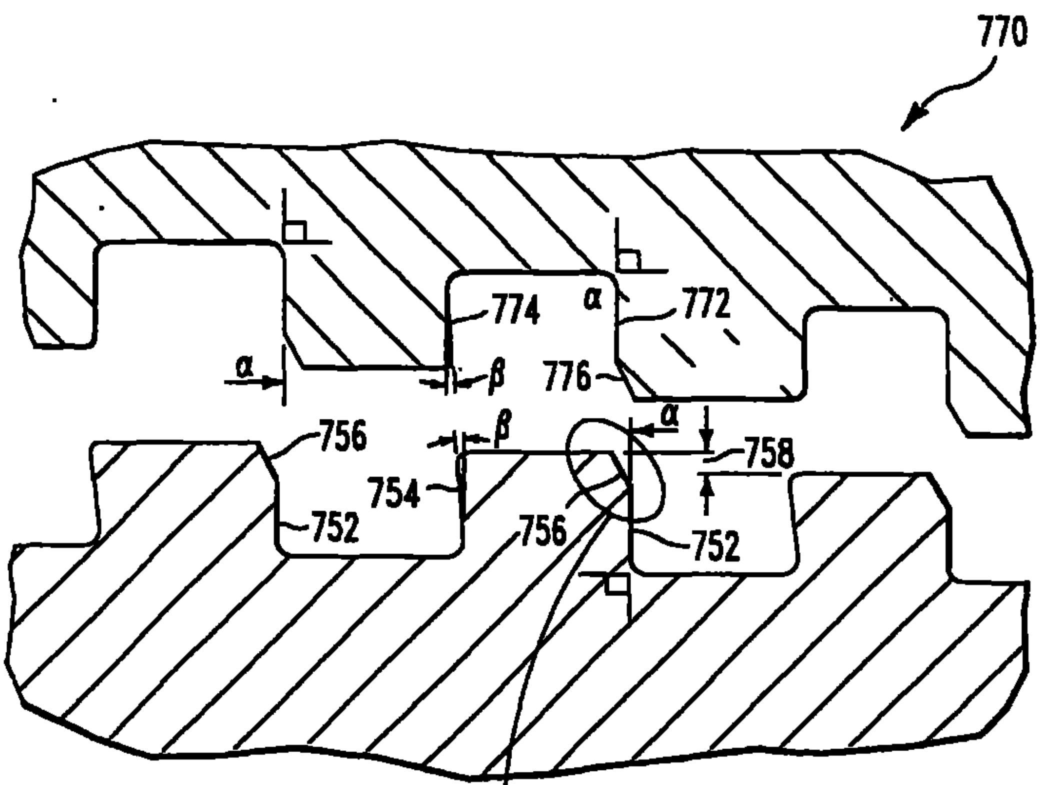

FIG. 7A

FIG. 7B + FIG. 7C

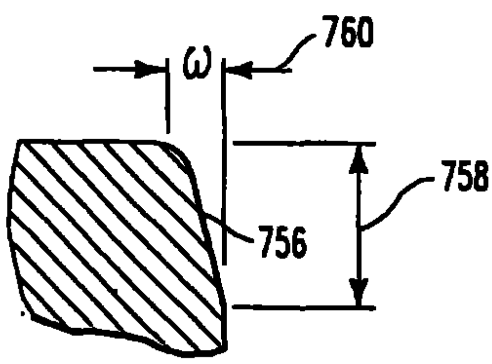

FIG. 7B

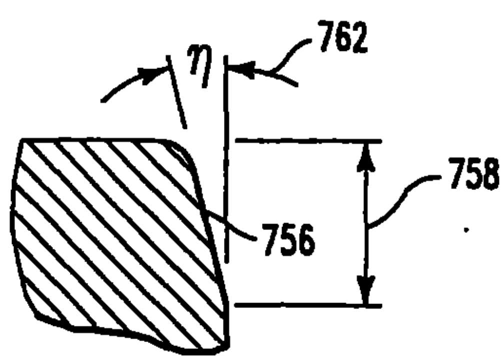

FIG. 7C

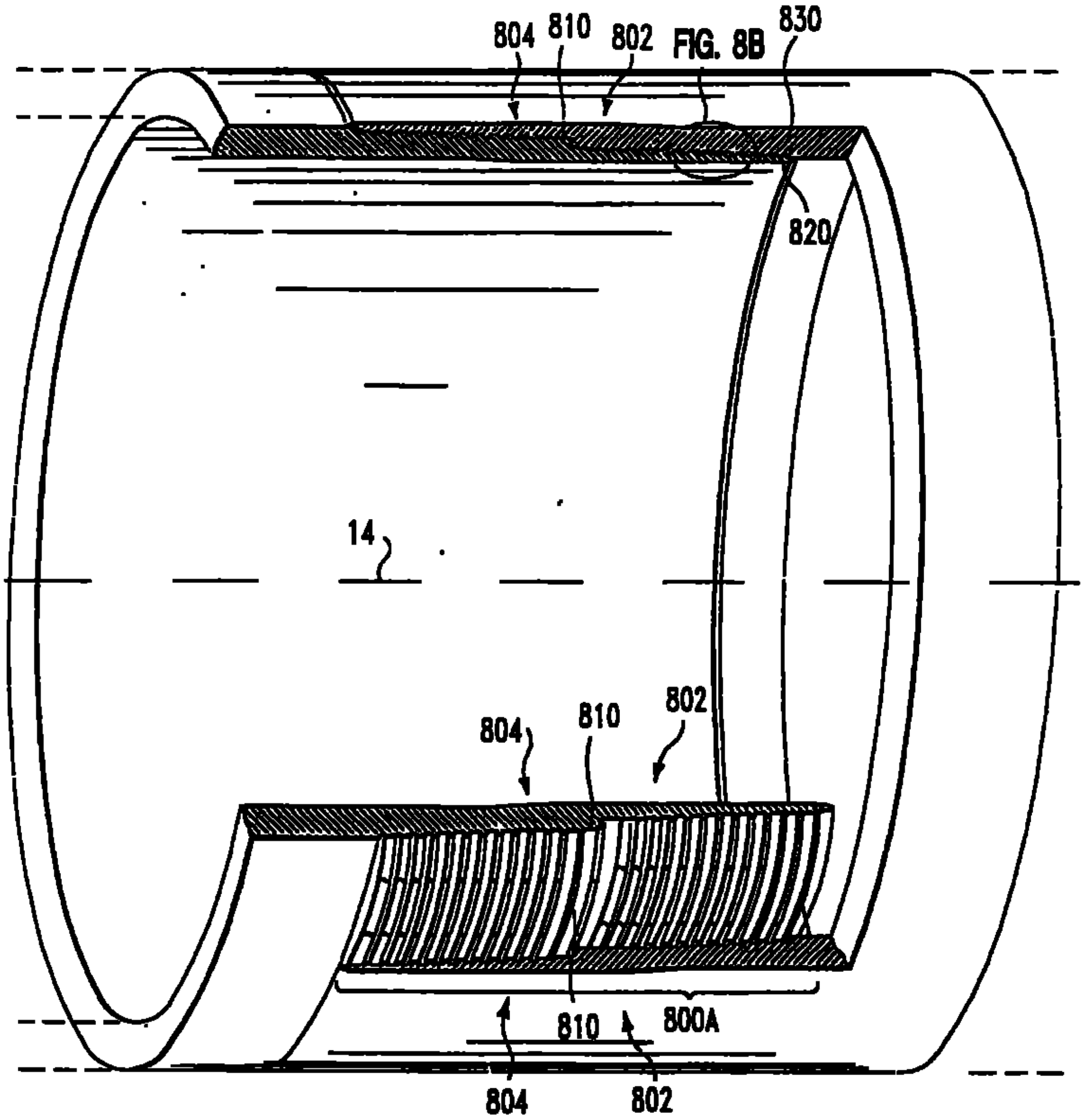

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FIG. 8B
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FIG. 8A

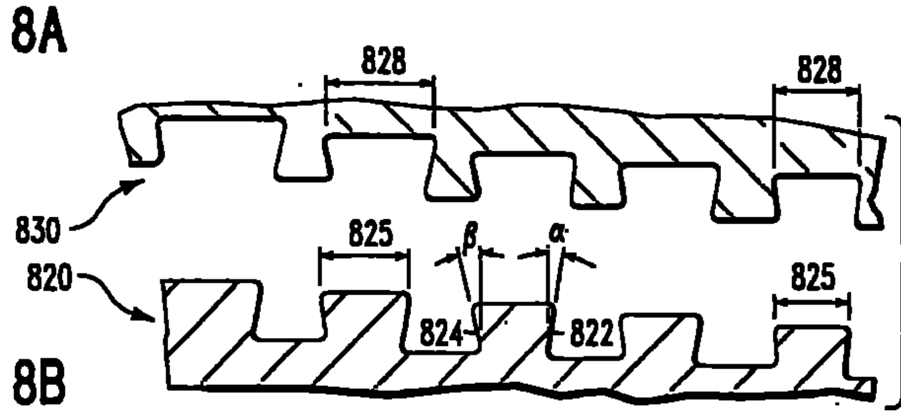

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FIG. 8B

FIG. 9B

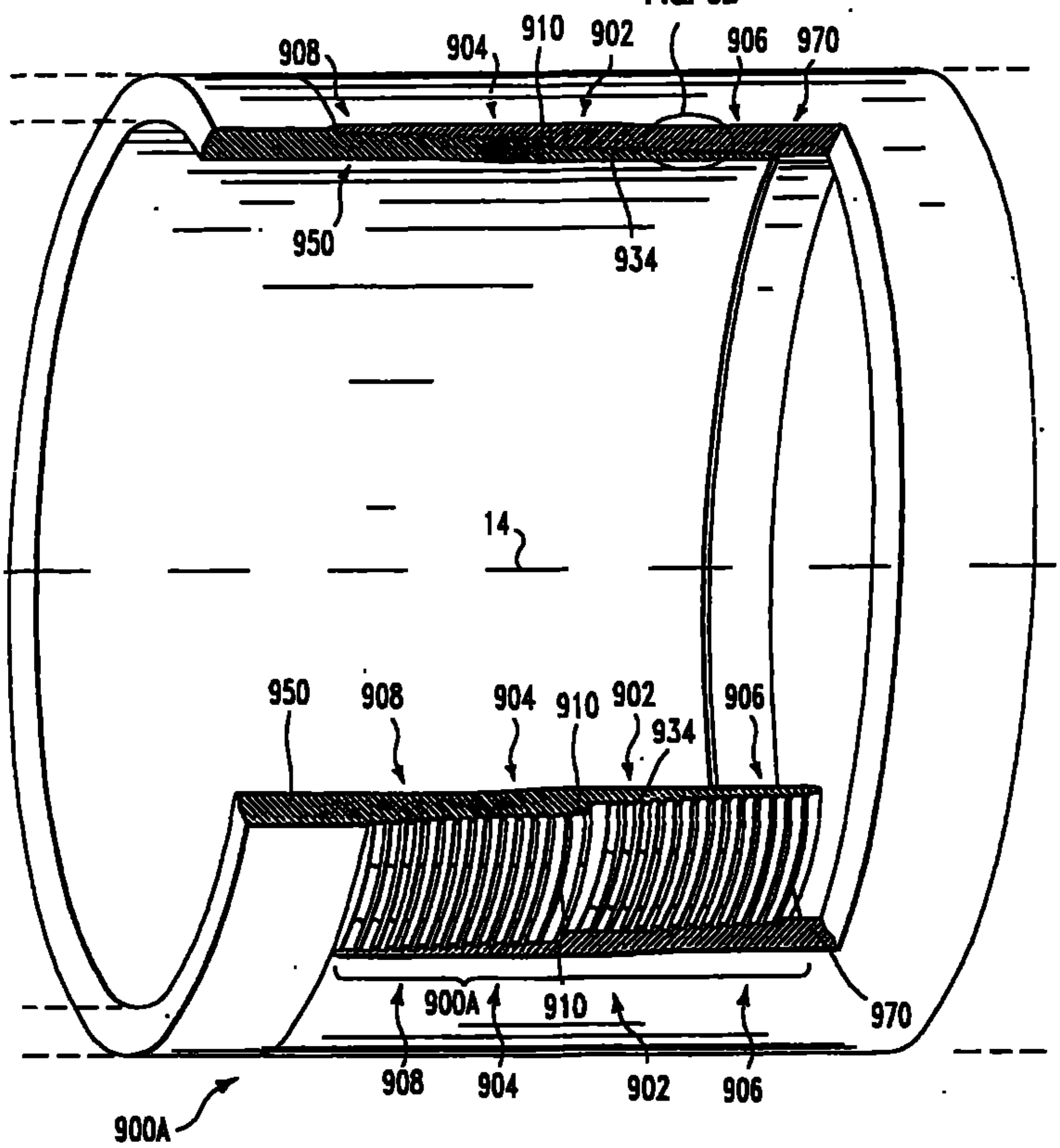
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FIG. 9A

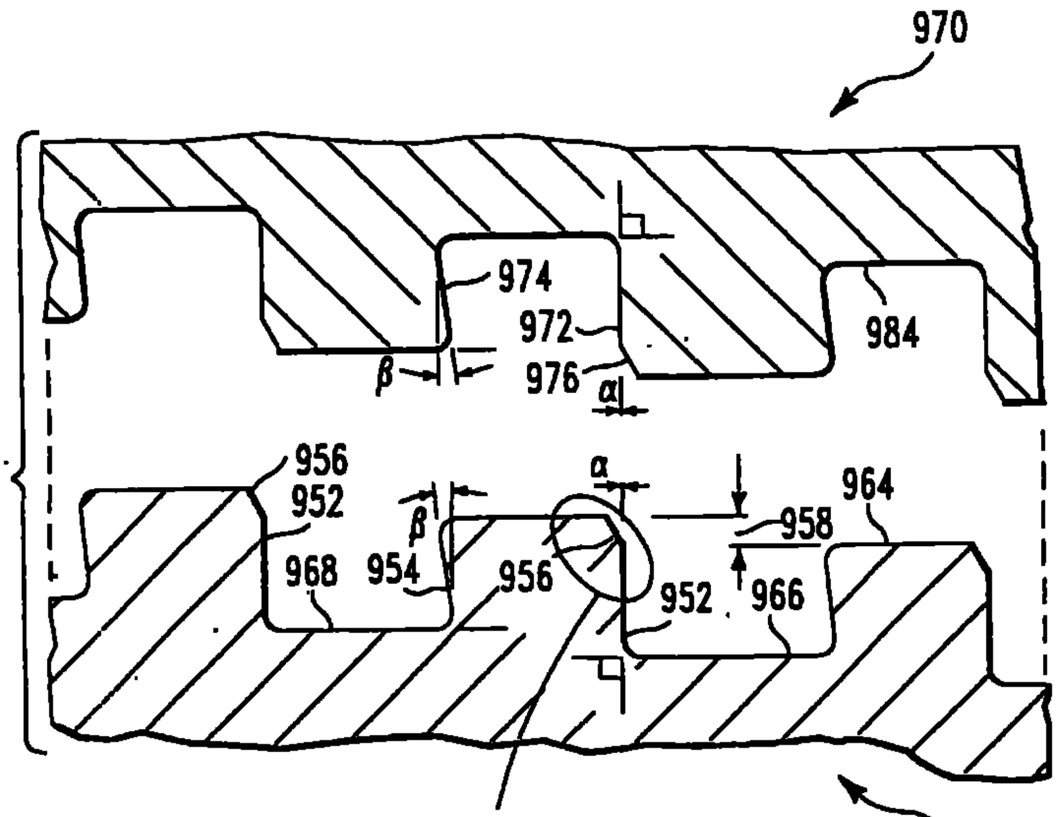

FIG. 9B

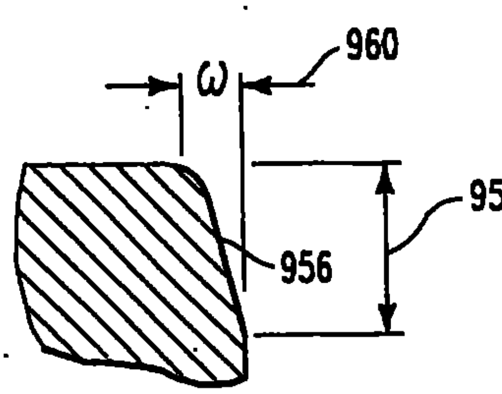

FIG. 9C

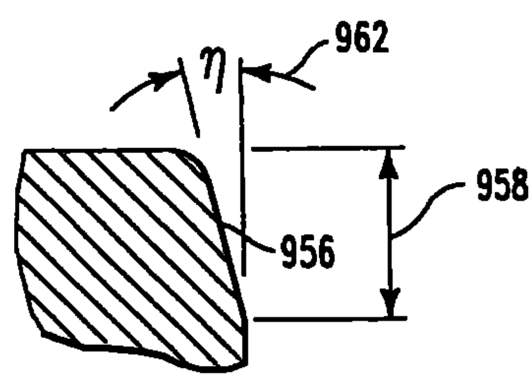

FIG. 9D

STABILIZED CENTER-SHOULDER-SEALED TUBULAR CONNECTION

This application is a continuation-in-part of U.S. application, Ser. No. 07/848,508 filed Mar. 9, 1992 and now U.S. Pat. No. 5,415,442 issued May 16, 1995.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to tubular connections of the kind commonly used in the oil industry. In particular, the tubular connection of the present invention comprises a stabilized center-shoulder seal, wherein stabilization is provided by specialized thread constructions adjacent the center-shoulder seal. The invention also pertains to a specialized center-shoulder seal design. Further, the invention pertains to a thread profile which provides a self-centering function upon makeup of the tubular connection. This thread profile can be used with the stabilized center-shoulder seal connection as well as other connection designs known in the art.

2. Background Art

The present invention is applicable broadly for use in joining pipes, couplings and the like, and has particularly been developed for use in oil industry tubular connections, since these connections experience more severe and demanding conditions than those observed for other industries and applications. In particular, the trend in the oilfield is to minimize pipe diameter and to conserve borehole diameter. The smaller the diameter of the borehole for the well, the greater the conservation of energy and reduction of drilling costs. However, in utilizing smaller diameter pipe, it is important that the connections between adjacent tubular pipes be transparent in geometry and performance characteristics, i.e. the geometry and performance characteristics of the connection be substantially the same as that of the tubular pipe itself. Preferably, the pipe string should behave as if the pipe connections were not there. The ideal connection would have the same geometry as the pipe, such as the same inside and outside diameters, and would have the same performance characteristics such as tension rating, compression rating, internal and external pressure ratings, torsion resistance, and bending resistance, for example.

Two types of oilfield connections, namely flush joints and slim line connections, have been utilized to conserve hole diameter. The outer diameter of a flush joint connection is generally the same as the outside diameter of the tubular pipe. The outside diameter of a slim line connection is generally 2 to 3.5% greater than that of the pipe. Although the objective of such prior art pipe has been to achieve total geometric and performance characteristic transparency, such has not been possible. Often, the performance characteristics of prior art pipe have been sacrificed to achieve greater geometric transparency.

Various standards have been set for evaluating the efficiency of a pipe connection. One such efficiency is determined by the load capacity of the pipe and pipe connection in the axial direction. This tension efficiency is calculated by comparing the fracture strength of the connection with the fracture strength of the tubular pipe. Most prior art flush joint and slim line connections achieve an efficiency in the range of 65% to 75%, respectively. This efficiency, of course, varies with the type of connection, the pipe diameter and thickness, and the method of manufacture of the connection.

Prior art connections include the basic elements of threads, shoulders, and seals. The following prior art patents describe various features which have been used to produce flush and slim-line connections.

U.S. Pat. No. 671,274 to W. C. Fischer, issued Apr. 2, 1901, shows a dovetail-shaped thread design which can be applied to the outside of one tubular and the inside of another tubular to be connected together.

U.S. Pat. No. 1,474,375 to J. J. Moore, issued Sep. 20, 1923 shows what applicant has described as single hook thread design. The thread is used to fasten the forward end of the after-body of a torpedo to an air flask body.

U.S. Pat. No. 1,927,656, issued Sep. 19, 1933 to G. M. Eaton et al., describes a pipe joint for fastening together the consecutive pipe lengths of a pipe string in a manner to obtain strength at the joint without increasing the wall thickness (or with a minimum increase thereof) in the portion of the pipe in which the screw threads are cut. One of the two pipes to be joined is machined on the outside surface of the pin and the other is machined on the inside surface of the box so the two pipes can be screwed together to provide what would today be called a slim line connection. The threaded portion of each pipe is machined on a taper with the depth of the turns of the threads being the greatest at the center of the threaded portion and decreasing gradually in both directions to vanishing points at the ends of the threaded portions.

U.S. Pat. No. 2,062,407, issued Dec. 1, 1936 to G. M. Eaton et al., discloses a preferred form of pipe joint wherein the height of the threads in the threaded portion of the joint is at a maximum in the central area of the threaded portion of the joint and decreases gradually in both directions to vanishing points, somewhat similar to the joint described above. However, the thread design differs in that the vanishing of the threads corresponds with the intersection of thread fabrication lines which are not only based on a taper relative to the pipe longitudinal axis but also a line which is cylindrical or parallel to the pipe longitudinal axis. Specifically, the root of the thread at the pipe exterior edge position of the pin is located along a line parallel with the pipe longitudinal axis while the crest of the thread at the same location is located along a line at a constant taper with the pipe longitudinal axis. The root of the thread at a position more interior from the edge of the pipe is located along a line at a constant taper with the pipe longitudinal axis, while the crest of the thread at the same location is located on a line of parallel taper to the root for a distance. Finally, at the most interior thread location on the pin, the thread root continues on a taper with the pipe longitudinal axis while the crest of the thread is constructed along a line parallel with the pipe longitudinal axis. Thus, at both the exterior and interior edges of the threaded connection, the thread disappears where thread root and thread crest intersect. The thread construction of the box is complimentary to that of the pin so the two threaded members can be interengaged.

U.S. Pat. No. 2,207,005 to R. D. Haas, issued Jul. 9, 1940, pertains to a single hook thread form (as defined herein) wherein the pin member load flank is undercut to provide a cam action under pressure to draw the connection members together and establish a sealing-off function. The thread form is constructed on a taper to the longitudinal axis of the connection.

U.S. Pat. No. 2,488,566 to R. S. Sperry, issued Nov. 22, 1949 described a tapered thread structure having a lead on one flank of the helical thread which differs from the lead on the other flank of the helical thread, to provide a constantly increasing thread width from one end of the thread structure to the opposite end. This variable width thread structure is said to enable easy disassembly of a rod or pin from a socket, permitting separation of pin from socket without the need to completely unscrew the parts, while providing a wedge-like, snug interfit between the threads of the parts when coupled together. The preferred embodiment for use of the variable width thread structure is on the surface of a pull rod used in combination with a tubular work piece to be formed.

U.S. Pat. No. 3,109,672, issued Nov. 5, 1963 to William F. Franz describes a threaded pipe joint comprising a pipe member having a cylindrical outer surface and a tapered buttress thread at the end thereof vanishing along the outer cylindrical surface, providing a length of fully formed and a length of vanishing threads. The complementary coupling member having thread machined on the internal surface thereof exhibits a full height tapered thread throughout the entire length thereof. The complementary threads on each member have following flanks in bearing relationship and substantially normal to the longitudinal axis of the joint and leading flanks in bearing relationship, which leading flanks have a larger flank angle than the following flanks. Crests and roots are truncated to provide flat crests and roots which are parallel to the longitudinal axis of the joint. Crest truncations of fully formed threads exceed root truncations to provide a predetermined amount of void between the crests and roots of the complementary threads throughout the length of the joint when the pipe and coupling are in hand tight engagement. After power make up, the crests of the coupling threads engage the roots of the vanishing pipe threads, but voids still remain between crests and roots of the fully formed threads, thus preventing the development of deleterious hoop stresses at the end of the coupling during power make-up.

U.S. Pat. No. 3,989,284, issued Nov. 2, 1976 to Thomas L. Blose (see also U.S. Pat. No. Re. 30,647, issued Jun. 16, 1981) describes a tubular connection (pipe joint) designed to produce hoop tension in the pin member and hoop compression in the box member. The connection exhibits threads having a dove-tail interfit. To achieve the dove-tail interfit, the angles formed between the thread flanks and the root wall of the thread are less than about 85°. In another described embodiment, the load flank of the thread exhibits an angularity from the direction of the connection on axis of less than about 85° while with the stab flank extends radially, in axial radial planes. The pin thread stab flank is the thread flank which is on the leading edge as the pin is telescoped (stabbed) into the box. The load flank is the trailing flank of the thread. In all cases, the threads are designed with clearances such that there is a "dove tail interfit" between pin and box drawing the pin wall outwardly and the box wall inwardly to produce hoop tension in the pin member and hoop compression in the box member in a made up connection, i.e. there is a wedging interfit of threads in the made up joint.

Each thread member is disclosed in the '284 Blose patent as having progressively changing axial width along substantially the entire helical length thereof. Although the preferred embodiment appears to have the threads constructed on a line parallel to the pipe longitudinal axis, the possibility is disclosed of having the threads constructed on a line having constant taper from the pipe longitudinal axis. Also described is the use of an inner shoulder seal between threaded sections of the connection. In one embodiment, the inner shoulder seal is placed between two thread sections which fall on two different lines, each parallel to the pipe longitudinal axis, creating a multiple-step thread with an inner shoulder seal. The inner shoulder seal is shown and

described as having a major mating (sealing) surface which is parallel to the pipe longitudinal axis or a major mating (sealing) surface which is at an axial taper.

U.S. Pat. No. 4,009,893, issued Mar. 1, 1977 to Schatton et al., discloses box and pin members having two axially spaced thread sections being separated by a radial step zone which includes, for one member, eg. the pin member, an annulus having an axial undercut, a frusto-conical end face and a radially outward bulging peripheral surface, which sealingly engages a slightly contoured surface of the box member in the step zone, while the end face of the annulus bears against a complementary, slightly contoured face in the step zone of the box member. These stop faces supplement complementary thread stop faces respectively at the end of the pin member and box member. It is advised that a flat annulus having good sliding properties, such as teflon, be interposed between the pin member annulus end face and the complementary bearing face of the box member. In one embodiment, there is a space between the pin member annulus end face and the complementary bearing face of the box member, and a sealing sleeve is interposed to fill this space and the undercut space of the pin member annulus.

U.S. Pat. No. 4,076,436 to D. T. Slatoret al., issued Feb. 28, 1979 describes specially designed stress relief grooves for threaded end portions of tools and for collars on reciprocating or rotating drill string mounted tools.

U.S. Pat. No. 4,161,332, issued Jul. 17, 1979 to Thomas L. Blose describes a pipe joint including pin and box members having interengaged two-step threads and interengaged annular shoulders. One of the annular shoulders of the connection is located axially between the first and second pair of threads. This shoulder is constructed so that the pin member stab flank of the shoulder is at a negative angle to the pipe longitudinal axis (preferably at about 5 degrees) to provide a semi-dovetail interfit of the shoulder on makeup of the joint. The pin member threads in general exhibit a negative angle load flank (preferably about 15 degrees) and a negative stab flank angle (typically about 30 degrees), to provide a semi-dovetail interfit with corresponding box threads; the pin member stab flank is typically not engaged, there being about 0.020 inches clearance with the corresponding box threads on normal make-up.

U.S. Pat. No. 4,224,607 to T. L. Blose, issued Jul. 17, 1979, discloses a cylindrical threaded connector having a small amount of taper to produce a controlled radial interference to prevent inadvertent thread disengagement.

U.S. Pat. No. 4,373,754 to C. A. Bollfrass, et at., issued Feb. 15, 1983, describes a threaded connector having multi-stepped, non-tapered thread sections. The thread profile has at least one "hooked" flank to provide improved tensile strength. The wall thickness of the female or box member of the connector is controlled to improve thread loading and to enable inwardly radial movement with the male or pin member to prevent "pulling out" of the threads under axial tensile loading. The outer thread step of the pin member may only partially engage the threads of the box member such that the critical reduction of the tube wall thickness at the thread roots is a minimum.

U.S. Pat. No. 4,384,737 to H. E. Reusser, issued May 24, 1983 discloses a tubular connection including a coupling or integral socket into which pipe sections are threaded. The outer surface of the pipe sections includes an unthreaded nose portion at the outer end of the threaded portion. The nose portion terminates with an annular surface and includes a plurality of convex surfaces of revolution extending circumferentially of the nose portion, all of which surfaces are adapted to engage the unthreaded portion of the coupling or socket to establish a fluid-tight seal. The present invention does not utilize such a pin nose design. Another feature of the Reusser invention (which is present in all claims which do not pertain specifically to the above-described pin nose design) is that surface-to-surface thread contact exists between either the minor diameter flat portions of thread on each member or between the major diameter flat portions of thread on each member, but that there not be contact between both the minor and major diameter flat portions. The present invention does not use this method of controlling stresses created during make-up of the connection. The connection of the present invention exhibits surface-to-surface thread contact between the crests and roots of successive threads for all full height threads and the majority (of not all) partial height threads.

U.S. Pat. No. 4,398,756, issued Aug. 16, 1983 to Duret et al., describes a cylindro-conical pipe joint wherein approximately 15% to 25% of the joint is threaded upon a continuous construction line which is at a taper to the longitudinal axis of the pipe, with the remaining approximately 75% to 85% of the threaded portion of the joint being threaded upon a continuous construction line parallel to the longitudinal axis of the pipe. The position of the threading which is on a taper to the pipe longitudinal axis is that which forms the entry to the connection when the pin and box are initially being telescoped together. Thus, clearance is provided by the taper of the box entry to permit ease in initial telescoping of the pin into the box, prior to actual engagement of the threads. Another feature of the Duret et al. invention is that the female (box) member has an internal shoulder against which the end face of the male (pin) member seals. The connection of the present invention relies on other sealing surfaces within the central structure of the connection, permitting the end face of the pin member to extend openly at the edge of the connection.

U.S. Pat. No. 4,570,982 to T. L. Blose, issued Feb. 18, 1986, describes a tubular connection including pin and box members having a two step thread profile with the leading thread step of the pin member being a smaller diameter than the second thread step. The thread profile of the members has a negative angle load flank and the thread steps are axially separated from one another by a reverse angle torque shoulder having one or two adjacent conical surfaces. The negative load flank angle of the threads and the reverse shoulder angle combine to lock the members together and prevent differential radial separation of the engaged conical sealing surfaces.

U.S. Pat. No. 4,591,195 issued May 27, 1986 to Chelette et al. discloses a center shoulder seal located between two threaded portions of a pipe joint, wherein the threaded portions are stepped threads constructed on a line parallel to the longitudinal axis of the pipe. The center shoulder seal is fabricated so the pin portion of the seal contains an undercut groove which can be used in combination with a sealing ring of rigid plastic or rubber material to provide a seal capable of retaining hot gases or liquids.

U.S. Pat. No. 4,603,889 to J. W. Welsh, issued Aug. 8, 1986, discloses a threaded fastener having an axis adapted to thread connect to two bodies. The fastener is threaded on the inside and the two bodies are threaded on the outside. The threading has first and second load flanks which face axially oppositely. The first flank defines a second pitch, said pitches being different and characterized in that when the fastener is screwed onto body threads the first flank tightens against one body thread when the second flank tightens against the other body thread.

U.S. Pat. No. 4,671,544, issued Jun. 9, 1987 to Donald J. Ortloff discloses a threaded pipe connection having two portions of tapered threads with tapered sealing surfaces located between the threaded portions. The tapered sealing surfaces engage a resilient seal member located in a groove about midway between the ends of one of the seal member surfaces; when the joint is made up, the sealing surfaces form independent metal-to-metal seals on both sides of the resilient seal member. In a preferred embodiment, the threaded portions are in the form of two steps, one on each side of the center sealing surface described above; the threads are wedge-shaped threads that increase in width progressively in one direction within each step of threads.

U.S. Pat. No. 4,676,529, issued Jun. 30, 1987 to Patrick E. McDonald describes a pipe joint wherein the pin and box members have two stepped portions of threads, each constructed on a line parallel with the longitudinal axis of the pipe, and having metal-to-metal sealing surfaces located between the threaded portions. The sealing surfaces provide three engaging shoulders between the pin and box; the two sets of outer shoulders have space between them at the time the set of inner shoulders is in contact. Between the set of inner shoulders and each set of outer shoulders, there is a groove; in one location the groove is in the box portion of the connection and in the other location the groove is in the pin portion of the connection. When the connection make-up torque is beyond the yield point of the metal of the set of inner shoulders, the metal extrudes into the grooves in the box and pin members to form interlocking surfaces which hold the box and pin from relative rotation.

U.S. Pat. No. 4,696,498 to K. L. Church, issued Sep. 29, 1987, describes a tubular connection having two members, one threaded on the inside (box) and one threaded on the outside (pin). Each member has a double set of steep tapered hook threads, two internal pressure seals including a compressible primary metal-to-metal seal and a supplementary seal made up of a resilient corrosion resistant material. Additionally, the connection has a primary resilient corrosion resistant external pressure seal, and a reverse shoulder engagement between the end of the pin member and a shoulder on the box member to hold the primary metal sealing surfaces in sealing engagement.

U.S. Pat. No. 4,703,954, issued Nov. 3, 1987 to Ortloff et al. describes a threaded pipe connection comprising a box having tapered internal threads and a pin having tapered external threads, wherein the threads are dovetail shaped in cross section (wedge shaped thread) and increase in width in one direction on the box and in the other direction on the pin. This design is improved over that described in U.S. Pat. No. 3,989,284 (U.S. Pat. No. Re. 30,647) in that high stress concentrations are avoided in the box between the first thread and the end of the box by making the length of the box between the first thread and the end of the box equal to or greater than 5% of the pipe diameter and by providing clearance-between the crest of the first full thread on the box and the root of the last thread on the pin and between the crest of the last full thread on the pin and the root of the first full thread on the box.

U.S. Pat. No. 4,703,959 to E. E. Reeves et al., issued Nov. 3, 1987 discloses a threaded pipe connection with compressible seal ring. The connection comprises a pin member threaded on the outside and a box member threaded on the inside. The members have threading which is tapered, with generally dovetail-shaped threads that increase in width in one direction on the pin and in the opposite direction on the box so that the pin and box threads engage to form thread seals between the flanks of the threads and between the roots and crests of the threads. The improvement is said to comprise an annular groove intermediate the ends of the threads on one of the box and pin to form a cavity between the groove and the threads on the other of the box and pin and a ring-shaped seal member of compressible material located in the annular groove.

U.S. Pat. No. 4,753,460, issued Jun. 28, 1988 to Lawrence Y. Tung, discloses a tubular connection having a small diameter thread set and a large diameter thread set, the small thread set having a lesser number of threads than the large set. A primary torque shoulder is located between the thread sets and a secondary torque shoulder is located behind the large thread set. In the preferred embodiment, the initial partial thread on the pin of both sets and the last partial thread of the small set on the box has a cylindrical root contour (the root contour is constructed on a line parallel with the longitudinal axis of the pipe), whereas the remainder of threads exhibit a tapered root contour. The longitudinal taper angle of the pin thread sets are dissimilar to the box thread sets, there being a maximum separation of the sets near the center of the connection. At the center, a seal is provided; the last threads adjacent to the center shoulder on the threaded sets are hooked (the angle of the pin load flank is negative with respect to the pipe longitudinal axis). In all embodiments described, the threads exhibit a negative load flank and a positive stab flank. The center shoulder area of the pin exhibits two flanks, one of which is positive (near the small diameter step of threads) and one of which is negative (near the center of the shoulder and about midway between the two steps of threads).

U.S. Pat. No. 4,822,081 to T. L. Blose, issued Apr. 18, 1989, describes a driveable two-member tubular connection, and in particular a thread form for the connection. The thread form utilizes converging, trapped thread flanks and mating shoulders and end faces with dissimilar angles. A two step thread design is illustrated, wherein variable width "trapped" threads are present on each step of threads. The thread crests and roots are described as making contact before the thread flanks are touching. This Blose et al. invention is concerned with the benefit of the thread stab flanks in assisting the shoulders/end faces in resisting axial compressive loading and with the different angles between mating shoulders and end faces to reduce deformation of the shoulders upon axial compressive loading.

U.S. Pat. No. 4,893,844 to K. D. Chelette et al., issued Jan. 16, 1990, discloses a tubular connection with ventable seal. A deforming seal member is disposed between two surfaces such that a space is provided to at least one side of the sealing member. The connection comprises box and pin members having hooked threads, a stop shoulder between the two members and various seal means.

U.S. Pat. No. 4,917,409, issued Apr. 17, 1990 to Doyle E. Reeves describes a threaded connection employing a tapered, wedge shaped thread. The threads increase in width in one direction on the box and in the other direction on the pin. The improvement over the prior art appears to be in the ability of the thread design to be used with pipe thread lubricants; to accommodate such lubricants, the thread is designed to eliminate the radial clearance between the thread roots and crests so that the possibility of creating an expansible chamber upon connection make-up that can entrap the liquid or paste-like thread lubricant therein, to produce a false torque reading is greatly minimized, if not entirely reduced. The wedge-shaped thread can be a dovetail or semi-dovetail thread profile. A variable width thread construction (wherein thread width increases progressively in a given direction) is used in combination with the dovetail or semi-dovetail thread profile (as in the Blose U.S. Pat. No. 3,989,284). The specification and original claims of Reeves discloses that the load and stab flanks of the thread fully engage "at substantially the same time upon rotational make-up of the threaded connection" Col. 4, lines 43–48. Although only continuation applications were filed during prosecution (no continuation-in-part applications which would permit the introduction of new matter), the claims were amended in a subsequent continuation application to contain the limitation that the thread flanks did not fully engage at substantially the same time. Thus, there is presently some confusion as to exactly what is taught by Reeves.

Another concept disclosed and dependently claimed in Reeves is that it is desirable to increase the thread width at the widest portion in the range of approximately four times the thread width at the narrowest portion at the same thread location. There is no discussion as to why this provides an advantage over prior art (which is said to show less of an increase in thread width).

U.S. Pat. No. 4,944,538 to S. R. Read, issued Jul. 31, 1990, describes a threaded pipe joint having improved seal ring entrapment.

U.S. Pat. No. 4,958,862 to C. Capelli et al., issued Sep. 25, 1990, discloses a tubular connection comprising a pin member having at least one external tapered thread and a box member with complementary internal threads. An end part of the pin member beyond each thread has a conical seal surface and the box member has a conical seal surface with the same taper as that of the pin member, so that upon tightening of the joint a seal is created between pin and box at the forward end region of the pin member.

U.S. Pat. No. 5,029,906 to K. D. Chelette et al., issued Jul. 9, 1991, pertains to a deformable sealing member used to provide a pressure vent for mating tubular members. A typical threaded coupling comprises pin and box members each having a stepped thread design of hooked configuration on at least one side of a stop shoulder.

U.K. Patent Specification 137,777 of W. Pickard, published in 1920, shows a variety of thread designs which can be used to connect two or more parts or objects laterally or which, on entering a material, will become anchored therein and resist separation in all directions at right angles to the longitudinal axis of the connection. FIG. 8 of the disclosure shows a single hook thread profile. FIG. 10 shows a double hook thread profile.

U.K. Patent Application of K. Maruyama, published Nov. 4, 1981, describes a fluid tight thread joint for oil or gas well pipes. The joint comprises box and pin members; the thread on the pin starts from the pin end as a rounded triangular thread and becomes a buttress thread; the thread on the box member is formed in a corresponding manner.

DEFINITIONS

For convenience and accuracy of description, the following definitions have been adapted for use in describing the connection of the present invention.

The "pin member" of a connection is understood to mean the male portion of a tubular member which is threaded on its external surface to be interengaged with threads on a box member to provide a connection.

The "box member" of the connection is understood to mean the female portion of a tubular member which is threaded on its internal surface to be interengaged with threads on a pin member to provide a connection.

A connection having a "two-step thread" is understood to mean a connection having two threaded portions wherein one threaded portion has a greater radius from the longitudinal axis of the pipe than the other threaded portion.

A "center shoulder seal" is understood to mean a section of a connection disposed between at least two threaded portions, which section exhibits directly contacting surfaces between the pin and box members, or between the pin and box members and a third component such as an annular sealing sleeve, so that the section provides a seal for preventing the passage of fluid through (across the threads of) the assembled connection.

A "thread root" is understood to mean the thread location at which the wall of the tubular member has been machined away to its maximum depth. The thread root defines the major diameter of a box thread and the minor diameter of a pin thread.

A "thread crest" is understood to mean the thread location at which the wall of the tubular member has been machined to its minimum depth (in some cases, none of the wall will have been machined away at the crest of the thread). The thread crest defines the major diameter of a pin thread and the minor diameter of a box thread.

A "run-out thread" is understood to mean a portion of thread having its roots machined on a taper with respect to the longitudinal axis of the tubular member, but having its crests machined parallel to the longitudinal axis of the tubular member; gradually the construction (machining) lines of the crests and roots of the thread intersect and the thread disappears. In an assembled tubular connection, a run-out thread on a pin member or box member typically corresponds with a run-in thread on the corresponding box member or pin member, respectively.

A "run-in thread" is understood to mean a portion of thread having its roots machined parallel to the longitudinal axis of the tubular member, but having its crests machined on a taper with respect to the longitudinal axis of the tubular member; from an initial point, the construction lines of the crests and roots of the threads diverge, ultimately achieving a full height thread. In an assembled tubular connection, a run-in thread on a pin member or box member typically corresponds with a run-out thread on the corresponding box member or pin member, respectively.

A "thread stab flank" is understood to mean the more forward or leading flank of the thread when the pin is telescoped into the box.

The "thread load flank" is understood to mean the trailing flank of a thread upon telescoping of the pin into the box.

A thread "flank angle" is understood to mean the angle formed between the thread flank and a line constructed at the root of the thread flank, which line is perpendicular to the longitudinal axis of the connection (tubular). For ease in understanding, flank angles are defined in geometric terminology. For example, for a connection member cross-sectional drawing showing the connection member with its leading, threaded end toward the fight hand side and its trailing threaded end (typically followed by extended tubular) toward the left hand side, a positive flank angle is one wherein rotation of the perpendicular line toward the thread flank is in a clockwise direction, and a negative flank angle is one wherein rotation of the perpendicular line toward the thread flank is in a counter-clockwise direction. In the connection drawings illustrated throughout this application the connection pin member is shown at the bottom of the drawings with its leading edge on the right hand side while the connection box member is shown at the top of the drawing with its leading edge on the left hand side. Thus, the box member drawing can be reoriented to have its leading edge on the fight hand side for purposes of viewing the direction of rotation of the line constructed perpendicular to the connection longitudinal axis at the root of the thread flank. As described previously, the stab flank of the thread of each connection member is the thread flank which is leading or most forward as the connection members are telescoped together.

A "corresponding connection member portion" is understood to mean the portion of a connection member in contact with the member being discussed, upon assembly of the connection. For example, when a portion of the pin member of the connection is discussed, its corresponding connection member portion is the portion of the box member which will be in contact with that portion of the pin member upon assembly of the connection.

"Complementary engagement" is understood to mean the interengagement of corresponding members wherein both corresponding members are threaded in sections having essentially equal thread section length on each member, and wherein the entire height of thread on at least one member per threaded section is completely encompassed within the thread height of the other corresponding member threaded section.

A "lead-in thread" is understood to mean a starting portion of full-height or nearly full height threads which are machined on a taper to the longitudinal axis of the tubular member and beginning adjacent to the center shoulder seal configuration. In an assembled tubular connection, a lead-in thread on a pin member or box member corresponds with a run-out thread on the corresponding box member or pin member, respectively.

SUMMARY OF THE INVENTION

In accordance with the present invention, a tubular connection comprising at least two connection members and a center shoulder seal has been developed in a manner which provides stabilization of the center shoulder seal and improved efficiency (strength of connection/strength of tubular) of the connection.

Also in accordance with the present invention, a center shoulder seal design which provides multiple metal-to-metal sealing surfaces is described. This seal design comprises a locked double shoulder configuration.

In addition, the invention pertains to a thread profile which provides a self-centering function during make-up (assembly) of the connection.

Although the present invention can be applied to connections comprising tubulars and couplings, the most frequent use of the invention will be in the form of a tubular connection comprising two connection members, and thus the invention is described in this form below.

With reference to a tubular connection comprising two connection members, namely a pin member and a box member, the stabilization of the center shoulder seal is achieved using at least one run-out section of thread adjacent to the center shoulder configuration of each connection member. Typically, the at least one run-out section of thread is located adjacent the center shoulder configuration on that portion of the connection member at which the wall thickness is less. The run-out section of thread is typically used in combination with a run-in section of thread on the corresponding connection member. However, a run-out section of thread can be used in combination with a full height lead-in thread on the corresponding connection member.

Preferably, a run-out section of thread and a run-in section of thread are used together on the same connection member, with the center shoulder configuration located between the two sections of thread. The corresponding connection member will also comprise the corresponding sections of run-in and run-out threads, respectively, so there is one set of run-in and run-out threads adjacent the center shoulder seal. Even more preferably, in addition to the connection having one set of run-out and run-in thread sections adjacent to the center shoulder seal, the connection has a second set of run-out and run-in threads adjacent to the terminal ends of the connection. Such a double set of run-in, run-out thread sections stabilizes the center shoulder seal and the entire connection. The double set of run-in, run-out sections of threads also provides the most efficient connection.

When a connection member exhibits a single run-out thread adjacent to one side of the center shoulder configuration and a full height lead-in thread adjacent to the other side of the center shoulder configuration (with corresponding thread sections on the other connection member), a relief groove may be disposed between the center shoulder configuration and the full height thread. The relief groove permits proper interengagement of the threads upon assembly of the connection and facilitates machining of the full height lead-in thread. When a single set of run-out, run-in threads is used, with the center shoulder seal located between them, or when a double set of run-out, run-in threads is used, it is typically not necessary to use such a relief groove.

The most preferred center shoulder seal design is a "locked double shoulder" seal which provides an interlocking design, wherein a longitudinal cross section of the assembled pin and box contacting surfaces (in the center shoulder area) exhibits a reverse block S shape having extended tails on the S. Typically the assembled locked double shoulder configuration cross-section is two to three times as "high" in the radial direction (perpendicular to the tubular longitudinal axis) as it is "wide" in the axial direction (parallel to the connection longitudinal axis). In addition, the height is typically ¼ to ⅓ of the pipe body wall thickness. This seal design provides four (4) or five (5) zero clearance, metal-on-metal sealing surfaces and two (2) or three (3) close tolerance sealing surfaces which can be plugged with thread compound, on normal assembly, depending on amount of designed interference and tolerancing.

Although the locked double shoulder center seal is preferred, other center shoulder designs which are known in the art can also be stabilized using the run-out, run-in thread constructions described above adjacent to the center shoulder. Examples of other shoulder designs which can be stabilized include, not by way of limitation, square shoulder; hook shoulders (wherein the shoulder exhibits at least one negative angle relative to the connection longitudinal axis; square shoulders with a frustrum seal; annular shoulders (shoulder on a taper with the connection longitudinal axis); shoulders comprising grooves or an annulus having an axial undercut which employ a sealing sleeve, ring or annulus; and shoulders designed to extrude a portion of their mass laterally into adjacent grooves upon assembly of the connection.

The profile of the individual thread which can be used to make up the run-out, run-in thread sections is preferably selected from the group comprising double hook thread and single hook thread; although other thread profiles standard in the industry can be used.

The double hook thread profile is preferred in terms of mechanical performance. The double hook thread exhibits a negative load flank angle and a positive stab flank angle. A double hook thread profile in which the absolute sum of the load and stab flank angles (independent of the sign of the load and stab flank angles) totals less than about 20 degrees will function well in the present invention. Preferably, the negative load flank angle does not exceed about 8 degrees and the positive stab flank angle does not exceed about 5 degrees. Most preferably, the negative load flank angle ranges from about zero degrees to less than 5 degrees and the positive stab flank angle ranges from about zero degrees to less than 2 degrees, with the absolute sum of the angles being less than 5 degrees.

The single hook thread comprises a negative load flank angle ranging from about zero degrees to about 15 degrees and a stab flank angle ranging from zero degrees to a negative angle of about 35 degrees. Preferably the negative load flank angle is less than 8 degrees and the negative stab flank angle is less than 20 degrees. More preferably, the negative load flank angle ranges from about zero degrees to less than about 8 degrees, while the stab flank angle ranges from about zero degrees to a positive angle of less than about 8 degrees. Most preferably, the negative load flank angle ranges from about zero degrees to less than about 8 degrees, while the stab flank angle ranges from zero degrees to a positive angle of less than 5 degrees.

The double hook and single hook thread profiles described above are preferably used in combination with a variable width thread, wherein the thread on each member changes progressively in axial width, with the thread width of the pin member progressively increasing from the leading or forward end toward the following or trailing end of each step of threads, and the thread width of the box member increasing in the opposite direction.

The make-up of the connection having single hook or double hook threads is greatly improved by providing a chamfer on the stab flank of the thread profile. This chamfer provides a self-centering function during telescoping of the pin member into the box member. In addition, the chamfer provides increased surface contact area between the members at the time of telescoping, reducing the overall load at any particular contact point and reducing damage to the connection on make-up. The axial distance of travel or width of the chamfer is preferably about 0.005 inches or less, with the degree of the chamfer being a function of the thread pitch and taper. The degree of chamfer is such that the radial height of the chamfer (perpendicular to the connection longitudinal axis) is approximately equal to the step height between the chamfered stab flank and the preceding load flank. The height of the stab flank minus the height of the preceding thread load flank is typically called the step height.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1A shows a schematic of a longitudinal cross-sectional view of one embodiment of the assembled connection of the present invention, wherein the connection comprises a locked double shoulder center-shoulder seal, a single run-out thread section on the pin member adjacent one side of the center shoulder seal and a full height lead-in thread section on the pin member adjacent the other side of the center shoulder seal, with corresponding thread configurations on the box member.

FIG. 1B shows a schematic of a longitudinal cross-sectional view of the pin member shown in FIG. 1A.

FIG. 1C shows a schematic of a longitudinal cross-sectional view of the box member shown in FIG. 1A.

FIG. 2A shows a schematic of a longitudinal cross-sectional view of a second embodiment of the assembled connection of the present invention wherein the connection comprises a locked double shoulder configuration forming the center-shoulder seal and a set of run-out and run-in thread sections on each side of the center shoulder seal.

FIG. 2B shows a schematic of a longitudinal cross-sectional view of the pin member shown in FIG. 2A.

FIG. 2C shows a schematic of a longitudinal cross-sectional view of the box member shown in FIG. 2A.

FIG. 3A shows a schematic of a longitudinal cross-sectional view of a third embodiment of the assembled connection of the present invention, wherein the connection comprises a locked double shoulder center-shoulder seal and a double set of run-out and run-in thread sections, with one set being adjacent to the center-shoulder seal and the other set being disposed at the terminal ends of the pin and box members.

FIG. 3B shows a schematic of a longitudinal cross-sectional view of the pin member shown in FIG. 3A.

FIG. 3C shows a schematic of a longitudinal cross-sectional view of the box member shown in FIG. 3A.

FIG. 4A shows a schematic of a longitudinal cross-sectional view of a fourth and most preferred embodiment of the assembled connection, wherein the connection comprises a locked double shoulder center-shoulder seal, a double set of run-out and run-in thread sections, and the preferred thread profile (a single hook, variable-width thread having chamfered stab flanks).

FIG. 4B shows a schematic of a longitudinal cross-sectional view of the pin member shown in FIG. 4A.

FIG. 4C shows a schematic of a longitudinal cross-sectional view of the box member shown in FIG. 4A.

FIG. 5A shows a schematic of an enlarged view of the longitudinal cross-section of the locked double shoulder center-shoulder seal of the assembled connection shown in FIG. 1A.

FIG. 5B shows a schematic of a longitudinal cross-sectional view of the pin member of FIG. 5A.

FIG. 5C shows a schematic of a longitudinal cross-sectional view of the box member shown in FIG. 5A.

FIG. 6A shows a schematic of a longitudinal cross-sectional view of a portion of a section of double hook thread profile of the present invention having a negative load flank angle and a positive stab flank angle.

FIG. 6B shows a schematic of a longitudinal cross-sectional view of a single hook thread profile of the present invention having a negative load flank angle and zero degree (fight angle) stab flank.

FIG. 6C shows a schematic of a longitudinal cross-sectional view of a conventional single hook thread profile of the present invention having a negative load flank angle and a negative stab flank angle.

FIG. 6D shows a schematic of a longitudinal cross-sectional view of a preferred double hook thread profile of the present invention having a negative load flank angle and a stab flank angle which is slightly positive, approaching zero degrees.

FIG. 7A shows a schematic of a longitudinal cross-sectional view of a preferred single hook thread profile of the present invention having a negative load flank angle, a zero degree (right angle) stab flank and a chamfered stab flank. Most preferably this thread profile would be a variable width thread, wherein the thread on each member changes progressively in axial width.

FIG. 7B is an enlarged cross-sectional view of a chamfered stab flank of FIG. 7A particularly illustrating the width, ω, of the chamfer.

FIG. 7C is an enlarged cross-sectional view of a chamfered stab flank of FIG. 7A particularly illustrating the chamfer degree, n, of the chamfer.

FIG. 8A shows a schematic of a longitudinal cross-sectional view of the assembled connection of FIG. 2, but with the thread having the double hook thread profile shown in FIG. 5A, and having different leads in the load and stab flanks, thereby producing a variable thread crest/root width on each thread.

FIG. 8B shows a schematic of a longitudinal cross-sectional view of a portion of the double hook threaded profile of the pin and box members shown in FIG. 8A.

FIG. 9A shows a schematic of a longitudinal cross-sectional view of the assembled connection of FIG. 3, but with the thread having the single hook thread profile.

FIG. 9B–9D show a schematic of a longitudinal cross-sectional area of a portion of the single hook thread profile of the pin and box members shown in FIG. 9A.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The tubular connection of the present invention is applicable broadly for use in joining pipes, couplings and the like, but will be described in terms of oil industry tubular connections, since this is one of the highest value uses for the invention. In particular the invention is most typically used in flush joint or slim line tubular connections, where it is desired to minimize any variation in the outside and inside diameter dimensions between the connections and the assembled tubular members. The two major uses for the connection of the present invention are as a flush joint liner, typically having a flush outside diameter, and a liner tie-back, typically a formed connection having a maximum of +3.5% variation in outside diameter.

The tubular connection of the present invention utilizes a center seal (between at least two threaded sections of connection) to prevent fluids (liquids or gases) from passing through (across the threads of) the connection. The preferred embodiments of the invention will be described :as having a locked double shoulder center shoulder seal, which will be described in detail subsequently. However, other center seal designs such as, for example, square shoulder, hook shoulder, square shoulder with frustrum seal, and annular shoulder can be used in the stabilized center-shoulder-sealed tubular connection of the present invention.

In accordance with the present invention, the center shoulder seal is stabilized so that it can withstand various forces placed on the assembled length of tubulars without significant damage to and/or leakage through the seal. The stabilization is provided by using various combinations of thread sections adjacent to the center shoulder seal. In particular, stabilization of the center shoulder seal is achieved using at least one run-out section of thread adjacent to the center shoulder seal configuration of each connection member. The run-out section of thread on one connection member is typically used in combination with a run-in section of thread on the corresponding: connection member.

However, a run-out section of thread on one connect! ion member can be used in combination with a full height lead-in thread on the corresponding connection member.

FIGS. 1 through 4 illustrate four different preferred embodiments of the connection of the present invention. Each of the four preferred connections illustrated in FIGS. 1–4, have certain common features which will now be described with respect to embodiment one of FIG. 1. Referring now to FIGS. 1A, 1B and 1C, the pipe joint or threaded connection of the present invention, generally designated 100 in FIG. 1A, includes a pin member 102 formed on the end of tubular member 103 and a box member 106 formed on the end of another tubular member 107. Either one or both of the tubular members 103, 107 define a longitudinal axis 14 (see FIGS. 8A or 9A) which is the longitudinal centerline of the connection 100. The box member 106 has an internal thread structure formed therein and is adapted to interengage a complementary external thread structure formed on pin member 102, for mechanically connecting tubular members 103 and 107 which form a part of a string of tubular pipe (not shown). Connection 100 further includes a center shoulder seal 104 located at approximately the longitudinal mid-point of connection 100. The external thread structure of the pin member 102 extends from its pin nose or leading terminal end 105 to a location 109 adjacent first center shoulder configuration 119 and then continues on the other side 111 of first center shoulder configuration 119 to the end of the external threads at the trailing terminal end 113 of connection 100. The complementary internal thread structure of the box member 106 extends from the box leading terminal end 115 to a location 117 adjacent second center shoulder configuration 108 and then continues at the other side 121 of second center shoulder configuration 108 to the end of the internal threads at trailing terminal end 123 of connection 100.

The center shoulder seal shown on connection 100 at 104, disposed between matched sets of connection threaded sections, prevents fluids (liquids or gases) from passing through (across the threads of) the connection. The preferred embodiments of the invention will be described as having a locked double shoulder center-shoulder seal, which will be described in detail subsequently. However, other center seal designs such as, for example, square shoulder, hook shoulder, square shoulder with frustrum seal, and annular shoulder can be used in the stabilized center-shoulder-sealed tubular connection of the present invention.

Referring again to FIG. 1, which illustrates embodiment one of the present invention, tubular connection 100, the center shoulder seal 104 is a locked double shoulder made up from a first center shoulder configuration 119 on pin member 102 and a second center shoulder configuration 108 on box member 106. The center-shoulder-seal-stabilizing thread sections (run-out thread sections) on each connection member are located on pin member 102 at 110 and on box member 106 at 114. With reference to pin member 102, run-out thread section 110, adjacent shoulder seal configuration 119, extends from position 109 adjacent seal configuration 119 to intersect full height tapered thread section 126 which extends to pin 102 leading terminal end 105. With reference to box member 106, run out thread section 114, adjacent shoulder seal configuration 108, extends from position 117 adjacent seal configuration 108 to intersect full height tapered thread section 128 which extends to box 106 leading terminal end 115. The complementary thread section which interengages run-out thread section 110 on pin member 102 is full height lead-in thread section 112 on box member 106. The complementary thread section which interengages run-out thread section 114 on box member 106 is full height lead-in thread section 116 on pin member 102. The full height lead-in thread sections 112 and 116 on box member 106 and pin member 102, respectively extend as full height thread sections 124 and 122, respectively, providing full height tapered threads to the end of these threaded sections at 123 and 113, respectively. The taper on threaded sections extending from leading terminal ends 105 and 115 of pin 102 and box 106, respectively, is slightly greater than the taper on threaded sections 112–124 and 116–122, which extend toward the trailing terminal ends 123 and 113 of box 106 and pin 102, respectively.

Upon assembly of connection 100, run-out thread section 110 of pin member 102 is interengaged with full height lead-in thread section 112 of box member 106 and full height thread section 126 of pin member 102 is interengaged with the full height thread section 124 of box member 106. Upon assembly of connection 100, run-out thread section 114 of box member 106 is interengaged with full height lead-in thread section 116 of pin member 102 and full height thread sections 128 of box member 106 interengage with full height thread section 122 of pin member 102.

Pin member 102 comprises a thread relief groove 118 between full height lead-in thread section 116 and first center-shoulder configuration 119. Corresponding box member 106 comprises a thread relief groove 120 between full lead-in thread section 112 and second center-shoulder configuration 108. The thread relief groove facilitates machining the full height lead-in threads 116, 122 on pin member 102 and 112, 124 on box member 106, and enables complete complementary engagement of any corresponding thread profile. Relief grooves 118, 120 also provide a depository for thread compound and ensure that there is no hydraulic lockup of the thread.

Helical full height thread sections 112–124 and 116–122 are machined on a constant taper relative to the longitudinal axis of connection 100. Helical full height thread sections 126 and 128 are also machined on a constant taper relative to longitudinal axis 14 of connection 100. However, the taper of these latter thread sections is greater than the taper of their corresponding member thread sections 112–124 and 116–122, respectively, to provide reduced stress at the terminal ends of connection 100 upon assembly. Thread sections 110 and 114 are run-out thread sections which are machined having thread crest construction lines parallel to longitudinal axis 14 of connection 100 and having thread root construction lines machined on the constant taper with respect to the longitudinal axis. However, individual crest surfaces and individual root surfaces typically are machined parallel to the connection longitudinal axis. It is possible to leave individual run-in truncated thread crests on a taper, but preferred to machine them to be parallel to the connection longitudinal axis.

Upon assembly, the interengaged connection 100 exhibits complete complementary engagement of run-out thread sections 110 and 114, each located on that side of center-shoulder seal 104 at which each connection member wall thickness is less, namely thin wall sections 130, 132, respectively. In addition, a slight interference fit is provided in the threads to assure maximized interengagement of the threads. In an interference fit, at corresponding engagement points on the pin and box members 102 and 106, the pin member 102 is machined slightly larger than the box member 106. Interference fit tapered threads are well established in the art. Complete complementary engagement of the run-out thread section provides a more efficient connection by completely using the available thread flank bearing surfaces, while simultaneously providing limitation of the stresses generated at the thin wall sections 130 on pin member 102 and 132 on box member 106. In addition, the run out thread sections at these locations on the pin and box members permit maintenance of maximum connection wall thickness in the areas, adjacent center shoulder 104, thereby providing a more efficient connection.

For purposes of illustration, embodiment one of the invention has been shown in FIG. 1 with a single hook thread profile having a negative load flank angle and a negative stab flank angle; the crests and roots of individual threads are parallel to the connection longitudinal axis. Thread profiles other than a single hook can be used as well, as hereinafter described.

For purposes of comparison with the prior art, embodiment one of the present invention, in the structure of a +3 percent over pipe body diameter, formed-box slim line connection, exhibits an efficiency (strength of connection (joint)/strength of pipe) of about 85%, compared with the best known +3 percent formed box connection in the industry, which exhibits an efficiency of only about 75%. When the structure is a flush-joint connection, embodiment one of the present invention provides an efficiency of about 75% compared with the best known-in-the industry flush-joint connection efficiency of 65%.

Referring now to FIG. 2A, which illustrates embodiment two of the present invention connection, tubular connection 200 is made up from pin member 202 machined on the exterior end of tubular 203 and from box member 206 machined on the interior end of tubular 207. The center shoulder seal 204 of tubular connection 200 is a locked double shoulder configuration, formed from a first center shoulder configuration 2 19 on pin member 202 and from second center shoulder configuration 208 on box member 206.

FIG. 2B, which shows pin member 202, comprises run-out thread section 210 which extends from a location 209 adjacent first center shoulder configuration 219 on the thin-walled portion 230 of pin member 202 to thread section 226 of full height tapered thread which extends to leading end 205 of pin member 202. Pin member 202 also comprises run-in thread section 216 which extends from a location 211 adjacent first center shoulder configuration 219 on the thick-walled portion of pin member 202 to thread section 212 of full height tapered thread which extends to trailing end 213 of pin member 202.

FIG. 2C, which shows box member 206, comprises run-out thread section 214 which extends from a location 217 adjacent second center shoulder configuration 208 on the thin-walled portion 232 of box member 206 to thread section 228 of full height tapered thread which extends to leading end 215 of box member 206. Box member 206 also comprises run-in thread section 222 which extends from a location 221 adjacent second center shoulder configuration 208 on the thick-walled portion of box member 206 to thread section 224 of full height tapered thread which extends to trailing end 223 of box member 206.

Upon assembly of connection 200, run-out thread section 210 of pin member 202 interengages run-in thread section 222 of box member 206 and full height threads 226 of pin member 202 interengage full height threads 224 of box member 206. In addition, run-out thread section 2 14 of box member 206 interengages run-in thread section 216 of pin member 202 and full height thread section 228 of box member 206 interengages full height thread section 212 of pin member 202.

Helical full height thread sections 212 and 226 on pin member 202 and full height thread sections 224 and 228 on box member 206 are machined on a constant taper with respect to the longitudinal axis of connection 200, but having the individual surfaces of their roots and crests parallel to the longitudinal axis of connection 200. Run-out thread section 210 on pin member 202 has its minor diameter (roots) machined on a continuation of the constant taper of the minor diameter of adjacent full height thread section 226, while its major diameter (crests) is machined parallel to the connection 200 longitudinal axis. Run-in thread section 216 on pin member 202 has its minor diameter (roots) machined parallel to the connection 200 longitudinal axis, while its major diameter (crests) is machined as a continuation of the constant taper of the major diameter of adjacent full height thread section 212. Corresponding run-out and run-in thread sections of box member 206 are constructed in like manner, with the roots forming the major diameter and the crests forming the minor diameter.

Although sequential thread crests and roots are machined as described above, individual crest and root surfaces are typically machined parallel to the longitudinal axis of the connection. The exception to this is run-in thread sections, wherein the individual crest surfaces can be machined as a continuation of the constant taper of the major diameter of adjacent full height threads or can be machined parallel to the connection longitudinal axis. Individual crest surfaces machined parallel to the connection longitudinal axis are preferred. As with connection 100 shown in FIG. 1, the tapers of full height threaded sections 226 of pin 202 and 228 of box 206 are greater than the tapers of corresponding full height thread sections 224 of box 206 and 212 of pin 202, respectively.

Also, for purposes of illustration, embodiment two of the invention has been shown in FIG. 2 with a single hook thread profile having a negative load flank angle and a zero degree stab flank angle; however, other thread profiles can be used as well as hereinafter described.

In embodiment two, wherein a set of run-out, run-in thread sections is used adjacent each side of the center-shoulder seal 204, there is minimal unused (nonengaged) thread surface in the assembled thread sections. The connection 200 exhibits complete complementary engagement of the corresponding run-out and run-in thread section pairs, making these tapered threads highly efficient and providing even greater stabilization of the center-shoulder seal 204 than that provided in embodiment one shown in FIG. 1.

For comparison purposes with the prior art, embodiment two of the present invention shows approximately the same improvement in efficiency over the known prior art as that exhibited by embodiment one shown in FIG. 1.

Referring now to FIG. 3A, which illustrates embodiment three of the present invention connection, tubular connection 300 is made up from pin member 302 machined on the exterior end of tubular 303 and from box member 306 machined on the interior end of tubular 307. The center shoulder seal 304 of tubular connection 300 is a locked double shoulder configuration, formed from first center shoulder configuration 319 on pin member 302 and from second center shoulder configuration 308 on box member 306.

FIG. 3B, which shows pin member 302 comprises run-out thread section 310 which extends from a location 309 adjacent first center shoulder configuration 319 on the thin-walled portion 330 of pin member 302 to thread section 326 of full height tapered thread, which extends to run-in thread section 334 which extends to leading end 305 of pin member 302. Pin member 302 also comprises run-in thread section 316 which extends from a location 311 adjacent first center shoulder configuration 319 on the thick-walled portion of pin member 302 to thread section 312 of full height tapered thread, which extends to run-out thread section 336 which extends to trailing end 313 of pin member 302.

FIG. 3C, which shows box member 306, comprises run-out thread section 314 which extends from a location 317 adjacent second center shoulder configuration 308 on the thin-walled portion 332 of box member 306 to thread section 328 of full height tapered threads, which extends to run-in thread section 338 which extends to leading end 315 of box member 306. Box member 306 also comprises run-in thread section 322 which extends from a location 321 adjacent second center shoulder configuration 308 on the thick-walled portion of box member 306 to thread section 324 of full height tapered thread, which extends to run-out thread section 340 which extends to trailing end 323 of box member 306.

Upon assembly of connection 300, run-out thread section 310 of pin member 302, interengages run-in thread section 322 of box member 301, full height thread section 326 of pin member 302 interengages full height thread section 324 of box member 306, and run-in thread section 334 of pin member 302 interengages run-out thread section 340 of box member 306. In addition, run-out thread section 314 of box member 306 interengages run-in thread section 316 of pin member 302, full height thread section 328 of box member 306 interengages full height thread section 312 of pin member 302, and run-in thread section 338 of box member 306 interengages run-out thread section 336 of pin member 302.

Helical full height thread sections 312 and 326 on pin member 302 and 324 and 328 on box member 306 are on a constant taper with respect to the longitudinal axis of connection 300, but having the individual surfaces of their roots and crests parallel to the longitudinal axis of connection 300. Run-out thread sections 310 and 336 on pin member 302 and run-out thread sections 314 and 340 on box member 306 have their roots machined on a continuation of the constant taper of the adjacent full height thread sections. Run-out thread sections 310 and 336 on pin member 302 and run-out thread sections 314 and 340 on box member 306 have their crests machined parallel to the longitudinal axis of connection 300. Run-in thread sections 316 and 334 on pin member 302 and run-in thread sections 322 and 338 on box member 306 have their roots machined parallel to the longitudinal axis of connection 300, while their crests are machined on a continuation of the constant taper of the respective major or minor diameter of the adjacent full height thread sections.

Embodiment three of the invention shown in FIG. 3 exhibits a double hook thread profile; however, other thread profiles can be used as well, as hereinafter described.

This double set of run-out and run-in threads (one set adjacent the center-shoulder seal 304 and a second set at the terminal ends of connection 300) provides maximum joint efficiency and maximum stability for center-shoulder seal 304 of the connection 300. The double set of run-out and run-in threads also provides the smoothest load transfer between pin and box members 302, 306, respectively, thereby not only increasing the connection's rated load capacity but also its overload capacity (i.e., safety factor).

For comparison purposes with the prior art, embodiment three of the present invention, in the structure of a +3.5 percent over pipe body diameter, formed box slim line connection exhibits an efficiency (strength of connection (joint)/strength of pipe) of about 95%. When the structure is a 3 percent formed box connection, the efficiency decreases to about 90%. The best known-in-the-industry +3 percent formed box connection provides only a 75% efficiency.

Regarding the above-described embodiment three of the present invention, in the structure of a flush outside diameter connection, the efficiency provided is about 77%. The best known-in-the-industry flush connection provides only a 65% efficiency.

Referring now to FIG. 4A, which illustrates embodiment four of the present invention connection, tubular connection 400 is made up from pin member 402 machined on the exterior end of tubular 403 and from box member 406 machined on the interior end of tubular 407. The center shoulder seal 404 of tubular connection 400 is a locked double shoulder configuration, formed from first center shoulder configuration 419 on pin member 402 and from second center shoulder configuration 408 on box member 406.

FIG. 4B, which shows pin member 402 comprises run-out thread section 410 which extends from a location 409 adjacent first center shoulder configuration 419 on the thin-walled portion 430 of pin member 402 to thread section 426 of full height tapered thread, which extends to run-in thread section 434 which extends to leading end 405 of pin member 402. Pin member 402 also comprises run-in thread section 416 which extends from a location 411 adjacent first center shoulder configuration 419 on the thick-walled portion of pin member 402 to thread section 412 of full height tapered thread, which extends to run-out thread section 436 which extends to trailing end 413 of pin member 402.

FIG. 4C, which shows box member 406, comprises run-out thread section 414 which extends from a location 417 adjacent second center shoulder configuration 408 on the thin-walled portion 432 of box member 406 to thread section 428 of full height tapered threads, which extends to run-in thread section 438 which extends to leading end 415 of box member 406. Box member 406 also comprises run-in thread section 422 which extends from a location 421 adjacent second center shoulder configuration 408 on the thick-walled portion of box member 406 to thread section 424 of full height tapered thread, which extends to run-out thread section 440 which extends to trailing end 423 of box member 406.

The principal difference between this fourth embodiment of the invention and the third embodiment illustrated in FIG. 3 lies in the thread profile. The thread profile of this fourth embodiment is a specialized self-centering thread profile wherein the stab flank of all full-height threads is chamfered, as shown in FIG. 7 and described in detail subsequently with reference to FIG. 7. Thus, each stab flank of full-height thread sections 412 and 426 of pin member 402 and each stab flank of full-height thread sections 424 and 428 of box member 406 is chamfered. The chamfer width (axial distance) is preferably about 0.005 inches or less and the chamfer height (perpendicular to the connection or member longitudinal axis) is approximately equal to the step height (height of the stab flank minus the height of the preceding thread load flank). Presence of this chamfer on the stab flank of all full-height threads on both pin member 402 and box member 406 results in a self-centering action upon assembly of connection 400; in particular, the self-centering action occurs during the initial telescoping (stabbing) of pin member 402 into box member 406.

Upon initial assembly of connection 400, after initial telescoping, the stab flanks of pin member 402 threads come to rest on the stab flanks of box member 406 with the chamfered portion of the stab flanks providing the self-centering action described above. Upon completion of assembly, to power tight position, run-out thread section 410 of pin member 402, interengages run-in thread section 422 of box member 401, full height thread section 426 of pin member 402 interengages full height thread section 424 of box member 406, and run-in thread section 434 of pin member 402 interengages run-out thread section 440 of box member 406. In addition, run-out thread section 414 of box member 406 interengages run-in thread section 416 of pin member 402, full height thread section 428 of box member 406 interengages full height thread section 412 of pin member 402, and run-in thread section 438 of box member 406 interengages run-out thread section 436 of pin member 402. At power tight position, the load flanks, roots and crests of full-height threads of each member are in sufficient contact to provide sealing engagement, while the stab flanks of these threads exhibit a slight clearance (typically about 0.002 inches).

Helical full height thread sections 412 and 426 on pin member 402 and 424 and 428 on box member 406 are machined on a constant taper with respect to the longitudinal axis of connection 400, but having the individual surfaces of their roots and crests parallel to the longitudinal axis of connection 400. Run-out thread sections 410 and 436 on pin member 402 and run-out thread sections 414 and 440 on box member 406 have their sequential roots machined on a continuation of the constant taper of the adjacent full height thread sections. Run-out thread sections 410 and 436 on pin member 402 and run-out thread sections 414 and 440 on box member 406 have their sequential crests machined parallel to the longitudinal axis of connection 400. The individual root and crest surfaces of run-out thread sections are machined parallel to the connection longitudinal axis. Run-in thread sections 416 and 434 on pin member 402 and run-in thread sections 422 and 438 on box member 406 have their sequential roots machined parallel to the longitudinal axis of connection 400, while their sequential crests are machined on a continuation of the constant taper of the respective major or minor diameter of the adjacent full height thread sections. Although the individual root surfaces of these run-in thread sections are machined parallel to the connection (connection member) longitudinal axis, the individual crests can be machined either on a taper or parallel to the connection longitudinal axis, as previously discussed with reference to run-in thread sections.

Embodiment four of the invention, shown in FIG. 4 exhibits a thread profile having a chamfered stab flank with a positive stab flank angle of less than about 2 degrees, with reference to a line perpendicular to the preceding individual thread root (and the connection longitudinal axis). The load flank angle is a negative angle of about 3 degrees or less. In the most preferred embodiment of the invention, the stab flank angle is zero degrees and the load flank angle is a negative angle of about 3 degrees.

The efficiency of the connection illustrated in FIG. 4 is essentially the same as that of the connection illustrated in FIG. 3.

FIG. 5A is an enlarged view of a center-shoulder seal 500 which is shown in embodiment one of FIG. 1. Assembled center shoulder seal 500 is a locked double shoulder seal made up from first center shoulder configuration 510 on pin member 514 and from second center shoulder configuration 530 on box member 534. Thread relief groove 512 on pin member 514 and thread relief grooves 532 on box member 534 are made necessary by the use of full height lead-in threads 526 on pin member 514 and full height lead in threads 546 on box member 534, respectively. When full height lead-in threads are not used, for example in embodiments two, three or four (as illustrated by connections 200, 300 and 400), use of thread relief grooves is optional. The thread profile shown in FIG. 5A is a preferred double hook of the kind shown in FIG. 6D.

FIGS. 5B and 5C show first center shoulder configuration 510 of pin member 514 and second center shoulder configuration 530 of box member 534, respectively. Each connection member comprises a ramp, 516 on pin 514 and 536 on box 534. Connection pin member 514 center shoulder configuration 510 comprises shoulder 520 having first undercut surface 528, first sealing surface 518 and fourth sealing surface 522. Connection box member 534 center shoulder configuration 530 comprises shoulder 540 having second undercut surface 548, second sealing surface 542 and third sealing surface 538.

Upon assembly of center shoulder seal 500, first sealing surface 518 on shoulder 520 of pin configuration 510 is lined up by and guided into position on corresponding first ramp 536 of box center shoulder configuration 530. Simultaneously, second sealing surface 542 on shoulder 540 of box center shoulder configuration 530 is lined up by and guided into position on corresponding second ramp 516 of pin center shoulder configuration 510. As the assembly progresses, a metal-to-metal seal is created as first sealing surface 518 on shoulder 520 moves into place against third sealing surface 538 on box center shoulder configuration 530. Simultaneously, a metal-to-metal seal is created as second sealing surface 542 on shoulder 540 moves into place against fourth sealing surface 522 on pin center shoulder configuration 510. As shoulders 520 and 540 move into position, typically they begin to bend slightly, closing the gap between surface 524 on first center shoulder configuration 510 and surface 544 on second center shoulder configuration 530, thereby producing a zero clearance, metal-to-metal surface seal between at least a portion of surfaces 524 and 544. This latter metal-to-metal seal will not occur if there is substantial (greater than 0.005 in.) clearance between surface 524 and surface 544 prior to the above-described bending action.

Upon final power tightening (torquing) of the assembled connection, the leading surface of shoulder 520 contacts first undercut surface 548 on box center shoulder configuration 530, while the leading surface of shoulder 540 contacts second undercut surface 528 on pin center shoulder configuration 500, thereby creating two additional zero clearance, metal to metal sealing surfaces.

Thus, a total of at least four (4) and possibly five (5) metal-to-metal seals are created during assembly (make-up) of the connection: two metal-to-metal seals at the exterior beveled edges of the locked double shoulder seal 500 where sealing surfaces 518 and 538 are in contact and where sealing surfaces 522 and 542 are in contact; and, three zero clearance, metal to metal sealing surfaces at the interior of the locked double shoulder seal 500, one at the wall surface where sealing surfaces 524 and 544 contact, another zero clearance surface where the face surface of pin shoulder 520 contacts box undercut surface 548, and another zero clearance surface where the face surface of box shoulder 540 contacts pin undercut surface 528. Thus, center-shoulder seal 500 forms multiple metal-to-metal seals, having mating metal-to-metal sealing surfaces which allow a build-up of stored energy within the seal upon power tightening of the assembled connection, such that upon the application of various loads on the pipe and also the connection, the seal will continue to perform and maintain sealing engagement.

Typically a thread compound will be used on the surfaces of the threaded connection and this thread compound will bridge any surface defects and clearances less than about 0.005 inches, to provide an improved sealing capability. Such a thread compound will assist the intended metal-to-metal, zero clearance seals described above should surface defects or machining tolerances result in a clearance between metal surfaces of less than about 0.005 inches. A thread compound can also provide a hydraulic seal between surfaces which do not exhibit zero clearance after power tight make-up of the connection. For example, hydraulic seals can be provided between surfaces 529 of pin 514 and 537 of box 534 as well as between surfaces 517 of pin 514 and 549 of box 534 using thread compound. Additional hydraulic seals can be provided for each step of connection threads, since the thread surfaces which do not exhibit metal-to-metal contact are close fit surfaces which can be bridged using thread compound.

A description of the location and dimensions of the present invention locked double shoulder center seal relative to the cross-sectional area of the connection follows. It is desirable to balance the cross-sectional areas of the tubular walls which make up the connection in the area of the locked double shoulder seal, thus providing a more efficient connection. The formula which follows permits calculation of the preferred diameter of a circle formed by the radial center of the locked double shoulder seal (radial center relative to the longitudinal axis of the connection). This preferred diameter is essentially equal to the connection outside diameter squared plus the connection inside diameter squared minus the constant k, all divided by two, with the square root being taken of the result. The constant k is equal to two times the squared height of interlocked shoulders 520 and 540. As shown on FIG. 5A, the radial center of the locked double shoulder seal is located at contacting surfaces 524 and 544 of shoulder 520 and shoulder 540, respectively. The height of interlocked shoulders 520 and 540 is dimension 552. The minimum height 552 of interlocked shoulders 520 and 540 is a function of the wall thickness of the tubular (pipe) used to form the connection. Typically the radial height 552 of interlocked shoulders 520 and 540 is two to three times the longitudinal width, dimension 554 on FIG. 5A, of shoulders 520 and 540. In addition, the height 552 of the cross section of interlocking shoulders 520 and 540 is typically ¼ to ⅓ of the tubular member (pipe) wall thickness.

The connection of the present invention establishes a constant bearing force on the various seals provided by the center-shoulder seal 500. Not only do the curving edges of locked double shoulder seal 500 at 518,538 and 522, 542 interferingly engage to form a metal-to-metal seal, but upon power tightening of the connection, the facing surface of shoulder 520 in contact with undercut surface 548 and the facing surface of shoulder 540 in contact with undercut surface 528 are loaded and the annular contact between surfaces 524 and 544 typically establishes a seal due to a bending reaction on the shoulders from the seal 518 contacting seal 538 and seal 542 contacting seal 522 and from the Poisson's effect of the metal upon final power tightening.

Other center shoulder designs which are known in the art can also be stabilized using the adjacent run-out, run-in thread section constructions described above. Examples of other center seal designs which can be stabilized include, not by way of limitation, square shoulders; hook shoulders (wherein the shoulder exhibits at least one negative angle relative to the connection longitudinal axis); square shoulders with a frustum seal; annular shoulders (shoulder on a taper with the connection longitudinal axis); shoulders comprising grooves or an annulus having an axial undercut which employ a sealing sleeve, ring or annulus; and shoulders designed to extrude a portion of their mass laterally into adjacent grooves upon assembly of the connection.

FIGS. 6A through 6D illustrate four preferred thread profiles for use in combination with the other features of connections of the present invention.

For convenience and accuracy of description, the following geometric conventions will be used in describing the angles of the load and stab flanks of the threads of the present invention. The angle α represents the stab flank angle and the angle β represents the load flank angle of the threads. These angles are defined in geometric terminology with reference to a line constructed perpendicular to the longitudinal axis of the connection (tubular) and at the root of the thread flank. A degree of a negative flank angle is the rotation, in degrees, of the perpendicular line in the counterclockwise direction to meet the thread flank. A degree of positive flank angle is the rotation in degrees of the perpendicular line in the clockwise direction to meet the thread flank. These conventions for defining stab and load flank angles do not necessarily comport with the general conventions used in machining pipe threads.

FIG. 6A illustrates corresponding pin 610 and box 630 cross-sectional profiles for a portion of full height double hook thread 600A. The double hook thread exhibits a negative load flank angle β and a positive stab flank angle α. The thread profile of full height threads from a portion of pin member 610 is shown as it would appear within a threaded tubular section of the present invention. Stab flank 612 forms a positive stab flank angle α with respect to a line perpendicular to root wall 616, which is parallel to the connection longitudinal axis 14, and load flank 614 forms a negative load flank angle β with respect to a line perpendicular to root wall 618, which is parallel to the connection longitudinal axis.

The thread profile of full height double hook threads from a portion of box member 630 is shown as it would appear within a threaded tubular section of the present invention. For example, in a manner similar to the pin thread, stab flank 632 forms a positive angle α with respect to a line perpendicular to root wall 636, and load flank 634 forms a negative angle β with respect to a line perpendicular to root wall 636. To permit interengagement of the double hook threads of pin member 610 with the double hook threads of box member 630, it is necessary that the thread width be constantly increasing from the leading edge (edge initially telescoped into the box member) of pin member 610, and that the thread width be constantly decreasing (the root width be constantly increasing) from the corresponding edge of box member 630, as shown in FIG. 6A. Thus, the thread profile crest widths 620 of pin member 610 are progressively lengthened, while the thread root widths 622 are progressively narrowed, with the reverse occurring on box member 630. One method of producing threads with such dimensions is to machine the thread stab flanks with a slightly greater (longer) lead than the load flanks. The amount of lead increase (ΔL) required to permit the threads to mesh (interengage) upon telescoping (stabbing) is a function of the thread taper (T) and the stab and load flank angles (α and β, respectively), and can be calculated according to the following formula:

$$\Delta L = \tfrac{1}{2} T (\tan|\alpha| + \tan|\beta|)$$

When there is more than one step of threads in the connection (for example, a two step thread wherein one section of threads is followed by a center-shoulder seal, followed by a second section of threads located at a greater axial distance from the longitudinal axis of the connection), the thread width can be constantly increasing from the leading edge of the pin member all of the way to the end of the connection; or, alternatively and preferably, the thread width is increased progressively within each step of threads, beginning with the minimum width thread at the pin leading edge step of threads and the minimum width thread beginning again at the forward edge of the second step of pin threads.

The preferred double hook thread, in terms of mechanical performance, exhibits a positive stab flank angle and a negative load flank angle, the absolute sum of which (without regard to sign on the angle) ranges from about 5 degrees to about 20 degrees. Typically, the negative load flank angle does not exceed about 8 degrees and the positive stab flank angle does not exceed about 4 degrees. However, in terms of ease of machining and minimizing the lead adjustments, ΔL, the absolute sum of the positive stab flank angle and negative load flank angle should range from slightly greater than 0 degrees to less than 5 degrees, with the negative load flank angle ranging from about 1 degree to about 4 degrees. When this reduced absolute sum of load and stab flank angles is used, it is not necessary to use constantly increasing thread width to permit interengagement of connection member threads. Preferably the load flank angle is a larger (greater) absolute angle than the stab flank angle, since for thread profiles having crests and roots which are parallel to the connection longitudinal axis, the load flank length is less than the stab flank length, as can be observed in viewing load flank 614 and stab flank 612. Also, when the connection is operating under the most severe service conditions, the thread load flanks experience a greater applied force than the stab flanks.

FIG. 6B illustrates corresponding pin 660 and box 670 cross-sectional profiles of one embodiment of single hook thread, 600B. The thread profile of full height threads from a portion of pin member 650 is shown as it would appear within the threaded tubular section of the present invention. The distinguishing feature of this embodiment of single hook thread is that its stab flank 652 exhibits a zero degree angle α with respect to a line perpendicular to both the connection longitudinal axis and the thread root wall, while its load flank 654 exhibits a negative angle β with respect to a line perpendicular to both the connection longitudinal axis and the thread root wall. For example, referencing the thread profile of pin member 650, stab flank 652 forms a right angle with reference to the connection longitudinal axis, or zero degree angle α with respect to a line perpendicular to root wall 656, while load flank 654 forms negative angle β with respect to a line perpendicular to root wall 658. In a corresponding manner, referencing the thread profile of box member 670, stab flank 672 forms a zero degree angle α with a line perpendicular to root wall 676, while load flank 674 forms a negative angle β with a line perpendicular to root wall 676.

The single hook thread exhibits a negative stab flank angle which ranges from zero (0) degrees to about 45 degrees, and exhibits a negative load flank angle which ranges from greater than 0 degrees to about 15 degrees. If the load flank angle is a large negative angle, and the stab flank angle is less than the load flank angle, thereby producing thread crests wider than adjacent thread roots, the use of a constantly increasing thread width from the leading edge of the pin, as described above for a double hook thread, may be necessary to permit proper interengagement of the pin and box members of the connection upon initial telescoping (stabbing).

A more preferred single hook thread profile, due to ease of machining, is the profile wherein the negative load flank angle ranges from greater than 0 degrees to less than 5 degrees and the negative stab flank angle ranges from zero degrees to about 2 degrees, with the absolute sum of the angles being less than 5 degrees. When these reduced negative load flank and stab flank angles are used, it is not necessary to use constantly increasing thread width to permit interengagement of connection member threads. The necessary clearance is provided by making the thread crests narrower than the thread roots.

FIG. 6C illustrates another thread profile embodiment 600C suitable for use in the connection of the present invention. This thread profile is particularly suitable for a connection having a center-shoulder seal (such as the locked double shoulder) in combination with a single run-out thread and full height lead-in thread on the corresponding member, on each side of the center-shoulder seal. The thread profile of full height threads from a portion of the pin member 690 is another embodiment of a single hook thread as it would appear within the threaded tubular section of the present invention. This particular embodiment of a single hook thread features a load flank 694 of pin member 690 thread which exhibits a large negative angle β and a stab flank 692 which exhibits a large negative angle α. Generally, angle α is greater than angle β, and consequently, dimensions can be formulated with flank to flank (load to stab) bearing threads, having clearance between thread roots and crests, for example, with clearance between root 696 of pin member 690 and crest 699 of box member 698. This is in contrast with the double hook thread of FIG. 6A and the single hook thread of 6B, which are more suitable in a form wherein the thread roots and crests provide bearing surfaces with clearances between corresponding stab and load flanks. Flank to flank bearing threads of the kind shown in FIG. 6C can be used to modify the interference in the run-out section of threads.

The angles of the single hook thread profile shown in 6C typically are such that the negative load flank angle ranges from about 5 degrees to about 15 degrees and the negative stab flank angle ranges from about 5 degrees to about 35 degrees, with the stab flank angle being larger than the load flank angle by about 10 to 15 degrees.

FIG. 6D illustrates cross sectional profiles of a preferred double hook thread 600D. The preferred double hook thread exhibits a positive stab flank angle α ranging from nearly a zero degree angle to an angle of less than 2 degrees, and a load flank angle β which has been reduced to a small negative angle, ranging from nearly a zero degree angle to an angle of less than 5 degrees, with the absolute sum of the flank angles being less than 5 degrees. The thread profile of full height threads from a portion of pin member 680 is shown as it would appear within a threaded tubular section of the present invention. The distinguishing feature of the preferred double hook thread is that both its positive angle stab flank 682 and its negative angle load flank 684 exhibit a nearly zero degree angle with respect to a line perpendicular to the connection longitudinal axis and the thread root wall. For example, stab flank 682 forms an angle α which ranges from a right angle with reference to the connection longitudinal axis, up to a positive angle of less than 2 degrees with a line perpendicular to root wall 686; and, load flank 684 forms an angle β which ranges from a near fight angle with reference to the connection longitudinal axis, up to a negative angle of less than 5 degrees with a line perpendicular to root wall 688. Corresponding box member 689 thread exhibits the same ranges of stab flank angle and load flank angle ranges as those just described. It is not necessary to use progressively increasing thread width to permit connection member thread interengagement (proper meshing after stabbing) when a preferred double hook thread profile of the kind shown in FIG. 6D is used. This thread profile is essentially a modified square thread.

The most preferred thread profile for use in the present invention is one having a negative load flank angle and a zero degree or positive stab flank angle with a chamfered stab flank, which provides self centering action upon initial telescoping of connection members. Typically a single hook thread profile (to which this thread profile is similar) requires a variable width thread to permit proper interengagement of the threads. FIGS. 7A–7C illustrate a thread profile having a negative load flank angle and a zero degree, chamfered stab flank in combination with a variable width thread. The thread profile of pin member 750 comprises a load flank 754 having a negative angle, β, and a chamfered stab flank 702 having a zero degree angle, α; the chamfer 756 exhibits a width, ω, and a chamfer degree, η, is a function of thread pitch and taper. The height 758 of chamfer 756 is approximately equal to or less than the height of stab flank 752 minus the height of the preceding load flank 754, or the "step height." The thread profile of box member 770 comprises a load flank 774 having a negative angle, β, and a chamfered stab flank 772 having a zero degree angle, α; the chamfer 776 exhibits a width, ω, and a chamfer degree, η, with a chamfer height which is the same as the chamfer height of pin member 750. Box member 770 chamfer width, degree, and height are determined in the manner described for the corresponding pin member 750, with the chamfer degree being a function of thread width and taper and the resulting chamfer height being approximately equal to or less than the step height. An increased chamfer height provides increased self centering action and is preferred; however, a chamfer height greater than the step height provides no advantage. A chamfer width, ω, of less than 0.005 inches is preferred.

In sizing the connection of the present invention, the location of the radial centerline of the center-shoulder seal is determined with reference to the outside and inside diameters of the pipe. The take out point of the connection thread sections is determined by the minimum required wall thickness at the terminal ends of the connection. The length of a thread section is a function of the angle of taper of the threads and the change in diameter of the thread. Since the thread sections on each side of the center-shoulder seal should bear approximately one half of the load on the connection, the amount of engaged thread length may be calculated to ensure that the threads do not shear off under load. Knowing these parameters, one skilled in the art can then determine the thread taper.

FIG. 8A illustrates an assembled tubular connection 800A of the present invention. Connection 800A includes a set of run-out, run-in threads, set 802 and set 804, on each side of a center-shoulder seal 810, which is shown as a locked double shoulder configuration. The thread profile shown is a double hook which is shown enlarged on a portion of pin member 820 and box member 830 in FIG. 8B. Double hook threads having a positive stab flank angle α and a negative load flank angle β, the absolute sum of which is greater than about 8 degrees, require variable, constantly increasing thread crest width (i.e., a lead adjustment between the stab and load flanks) for proper meshing of the flanks, after the threads are telescoped, during hand tight make up. The use of variable width threads permits clearances between flanks in an assembled connection to be kept at a minimum (typically no more than about 0.003 inches). Thus, when a thread compound containing particulate matter is used during assembly of the connection, an additional seal can be formed between thread flanks due to the bridging action of the particles between flank clearances. Generally, such a seal can be formed only if the flank clearances are less than about 0.005 inches.

FIG. 8B shows portions of pin member 820 and box member 830, respectively. Stab flank 822 of pin member 820 exhibits a positive angle α of about 5 degrees, for example, and load flank 824 exhibits a negative angle β of about 8 degrees, for example. The variable width full height thread profile shown on the portion of pin member 820 exhibits increasing crest width 825 which corresponds with the variable decreasing width thread on box member 830, which exhibits increasing root width 828.

When double hook threads having constant width crests and roots (not shown) are used in the connection of the present invention, a greater flank clearance (above 0.005 inches, depending on the design) is required, making sealing at the thread flanks uncertain. This creates a potential for fluid leakage across the threads. Consequently, either the preferred double hook thread profile wherein the absolute sum of the flank angles is less than 8 degrees or a conventional single hook thread profile must be used to achieve thread sealing when the thread profile exhibits constant width crests and roots.

FIG. 9 illustrates another embodiment of an assembled tubular connection 900A of the present invention. Connection 900A includes a double set of run-out, run-in threads, set 902 and set 904, adjacent center-shoulder seal 910 combined with a second set of run-out, run-in threads, set 906 and set 908, adjacent the terminal ends of the connection. Preferably center-shoulder seal 910 is a locked double shoulder configuration. Connection 900A is shown with a single hook thread profile, having a chamfered stab flank.

As shown in FIG. 9B–9D the thread profile of a portion of pin member 950 includes a single hook thread having a stab flank 952 exhibiting a fight angle with reference connection 900A longitudinal axis 14 (or a 0 degree angle α with reference to a line perpendicular to the longitudinal axis and to and root wall 966, and having a load flank 954 exhibiting a negative angle β with respect to a line perpendicular to the connection longitudinal axis 14 and thread root wall 968. The negative load flank angle β is preferably in the range from about 1 degree to about 10 degrees. The thread profile of box member 970 is complementary to that of pin member 950.

The most preferred embodiment of the present invention is connection 900A shown in FIGS. 9A–9D which utilizes a double set of run-out, run-in thread sections and a locked double shoulder center seal, but having a variable width thread with a single hook thread profile including a chamfered stab flank, as shown in FIGS. 9B–9D. This embodiment provides the smoothest flow of forces between the pin member 950 and box member 970; provides high tension efficiency; and, also provides excellent sealing engagement for preventing fluid penetration into the threads and through (across the threads of) the connection. The negative load flank of the single hook thread causes the two thread zones on each side of the center-shoulder seal move as a unit, and the chamfered stab flank provides self-centering action during initial telescoping of the pin member 950 into the box member 970.

The run-out, run-in thread sections of the present invention fade in the thread load gradually and then fade out the thread load gradually. Full height threads like those typically used in the prior art cause the thread load to abruptly increase, thereby transferring a large force abruptly, which in turn caused undesirable bending moments within the connection.

The tension efficiency of a connection depends upon the extent to which the pin and box member are formed. Generally, the more forming, the greater the efficiency which can be achieved. Thus, greater efficiencies may be achieved in slim line connections which may have up to a 3.5% greater outside diameter than that of the tubular pipe than can be achieved in flush joint connections which have the same outside diameter as the pipe.

The double set of run-in, run-out threads of connection 900A is preferred since this provides maximum utilization of available pipe wall without the waste of space in the form of clearances and gaps in the threads for which there is no complementary engagement. Further, it is desirable not to have wasted space in the form of relief grooves, cut-outs, or other metal reduction. However, the relief grooves of the kind shown at 118 and 120 of connection 100 are not critical because they are located in the larger cross-sectional areas of pin 102 and box 106. Thus, they do not become critical in the overall efficiency of the connection.

With the single hook thread profile, as shown in FIGS. 9A–9D and recommended for use in connection 900A, an interference between the major and minor thread diameters, i.e. thread crests 964 of pin 950 and thread roots 984 of box 970, can be developed on power fight make up. Furthermore, interference between adjacent load flanks and minimal clearance between adjacent stab flanks can be provided on power fight make up. Any interferences of the roots and crests of the threads causes a hoop stress in the pin and box members 650 and 670, respectively. Consequently, upon power tight make-up, if any interference is present, the thread metal tends to move due to Poisson's effect causing the load flanks to butt up and engage. The interference will vary depending upon the pipe diameters and the grade of the pipe. With reference to FIG. 9A, preferably, the diametrical interference is maximized at thread sections 902 and 904 adjacent center-shoulder seal 910 and is gradually reduced to a minimum at thread sections 906 and 908 at the terminal ends of connection 900A. This gradual reduction in diametrical interference is designed to protect the thin walled member thread sections such as 934 of pin member 950 at the terminal ends of the connection. The larger cross-sectional areas adjacent the center-shoulder seal 900A are more rigid and therefore can have greater diametrical interference and absorb greater hoop stress. The cross-sectional areas of both pin member 950 small step and the box member 970 large step adjacent each side of the center shoulder seal 910 are designed to be approximately equal. The inner diametrical interference between the thread crests and roots is controlled to achieve linear stresses in the greater cross-sectional area of the box and pin members as well as achieve linear stresses at the thinner cross-sectional areas of the pin and box members adjacent their terminal ends. A certain amount of diametrical interference is required to achieve a tight interengaging fit of the threads. The general rule is that the interference will vary between 0.001 and 0.002 inches per inch of diameter, with less interference at the terminal ends of the connection than adjacent the center-shoulder seal.

The preferred center seal is one that is isolated to the loads placed on the pipe string. In other words, center shoulder seal 910 of connection 900A should remain stable and insensitive to varying loads placed on the connection. The locked double shoulder seal shown at 910 of connection 900A includes redundant sealing systems, i.e. a multiplicity of metal-to-metal seals, to achieve an effective sealing engagement such that the seal does not deactivate upon the application of various loads to the connection such as internal and external pressure, axial tension or compression, and torsional and bending moments.

The preferred embodiments of the present invention, as described above and shown in the Figures are not intended to limit the scope of the present invention, as demonstrated by the claims which follow, since one skilled in the art can, with minimal experimentation, extend the scope of the embodiments to match that of the claims.

What is claimed is:

1. A tubular connection comprising at least two members, wherein each member comprises at least two threaded sections having a center shoulder disposed between said at least two threaded sections, said center shoulders sealingly engaging to form a center shoulder seal configuration, said at least two threaded sections including at least one stabilizing thread section present on each connection member adjacent said center shoulder seal configuration engaging threads on the other connection member, said stabilizing thread section comprising a full run-out thread section for stabilizing said center shoulder seal configuration, said threaded sections including threads having means for stabilizing said members during make-up of the connection.

2. The tubular connection of claim 1, wherein the profile of said threaded sections is a double hook thread profile and wherein said at least two threaded sections include full height threads and said stabilizing means includes a chamfer on the stab flank of said full-height threads.

3. The tubular connection of claim 1 wherein said center shoulder seal configuration is a hook shoulder thread profile and wherein said at least two threaded sections include full height threads and said stabilizing means includes a chamfer on the stab flank of said full-height threads.

4. The tubular connection of claim 1 wherein said center shoulder seal configuration is a square shoulder with frustum seal.

5. The tubular connection of claim 1 wherein said center shoulder seal configuration includes annular shoulders.

6. The tubular connection of claim 1 wherein said center shoulder seal configuration is a shoulder comprising grooves.

7. The tubular connection of claim 1 wherein said center shoulder seal configuration is a shoulder comprising an annulus having an axial undercut.

8. The tubular connection of claim 1 wherein each of said center shoulders forms a groove and said center shoulders extrude a portion of their mass laterally into said grooves upon assembly of the connection.

9. A tubular connection comprising:

two members having terminal ends, each member having at least two threaded sections with a center shoulder disposed between said at least two threaded sections, said center shoulders sealingly engaging to form a center shoulder seal configuration;

said at least two threaded sections including at least one stabilizing thread section present on each said connection member engaging threads on the other connection member, said at least one stabilizing threaded section including a run-out thread section located adjacent said center shoulder seal configuration for stabilizing said center shoulder seal configuration;

each member having said run-out thread section located between said center shoulder seal configuration and said leading terminal end; and

said threaded sections including threads having means for centering one member with respect to the other member make-up of the connection.

10. The tubular connection of claim 9, wherein each said connection member includes a full-height lead-in thread section opposite said run-out thread section on the other connection member.

11. The tubular connection of claim 10, wherein each connection member includes a thread relief groove located between said center shoulder and said full height lead-in thread section.

12. The tubular connection of claim 11, wherein said members each have trailing ends and further include at least one additional stabilizing run-out thread section between said center shoulder seal configuration and said trailing end of each connection member.

13. The tubular connection of claim 12, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

14. The tubular connection of claim 11, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

15. The tubular connection of claim 10, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

16. The tubular connection of claim 10, wherein said members each have trailing ends and further include at least one additional stabilizing run-out thread section between said center shoulder seal configuration and said trailing end of each connection member.

17. The tubular connection of claim 16, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

18. The tubular connection of claim 17, wherein the profile of said threaded sections is a single hook thread profile, and wherein said centering means includes full-height threads having chamfered stab flanks.

19. The tubular connection of claim 17, wherein the profile of said threaded sections is a double hook, and wherein the absolute sum of the positive stab flank and negative load flank angle is less than 5 degrees.

20. The tubular connection of claim 17, wherein the profile of said threaded sections is a single hook, wherein the negative load flank angle is less than 15 degrees, and wherein the stab flank angle ranges from zero degrees to a negative angle of about 35 degrees.

21. The tubular connection of claim 20, wherein said negative load flank angle is less than 4 degrees, wherein the absolute sum of said negative load flank angle and said stab flank angle is less than 5 degrees, and wherein said centering means includes full-height threads having chamfered stab flanks.

22. The tubular connection of claim 16, wherein the profile of said threaded sections is a single hook thread profile and wherein said centering means includes full-height threads having chamfered stab flanks.

23. The tubular connection of claim 16, wherein the profile of said threaded sections is a double hook, and wherein the absolute sum of the positive stab flank and negative load flank angle is less than 5 degrees.

24. The tubular connection of claim 16, wherein the profile of said threaded sections is a single hook, wherein the negative load flank angle is less than 15 degrees, and wherein the stab flank angle ranges from zero degrees to a negative angle of about 35 degrees.

25. The tubular connection of claim 24, wherein said negative load flank angle is less than 4 degrees, wherein the absolute sum of said negative load flank angle and said stab flank angle is less than 5 degrees, and wherein said centering means includes full-height threads having chamfered stab flanks.

26. The tubular connection of claim 9, wherein each said member includes a run-in thread section at a location corresponding with said run-out thread section on the corresponding other connection member.

27. The tubular connection of claim 26, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

28. The tubular connection of claim 26, wherein said members each have trailing ends and further include at least one additional stabilizing run-out thread section between said center shoulder seal configuration and said trailing end of each connection member.

29. The tubular connection of claim 28, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

30. The tubular connection of claim 29, wherein the profile of said threaded sections is a single hook thread profile, and wherein said centering means includes full-height threads having chamfered stab flanks.

31. The tubular connection of claim 29, wherein the profile of said threaded sections is a double hook, and wherein the absolute sum of the positive stab flank and negative load flank angle is less than 5 degrees.

32. The tubular connection of claim 29, wherein the profile of said threaded sections is a single hook, wherein the negative load flank angle is less than 15 degrees, and wherein the stab flank angle ranges from zero degrees to a negative angle of about 35 degrees.

33. The tubular connection of claim 32, wherein said negative load flank angle is less than 4 degrees, wherein the absolute sum of said negative load flank angle and said stab flank angle is less than 5 degrees, and wherein said centering means includes full-height threads having chamfered stab flanks.

34. The tubular connection of claim 28, wherein the profile of said threaded sections is a single hook thread profile, and wherein said centering means includes full-height threads having chamfered stab flanks.

35. The tubular connection of claim 28, wherein the profile of said threaded sections is a double hook, and wherein the absolute sum of the positive stab flank and negative load flank angle is less than 5 degrees.

36. The tubular connection of claim 28, wherein the profile of said threaded sections is a single hook, wherein the negative load flank angle is less than 15 degrees, and wherein the stab flank angle ranges from zero degrees to a negative angle of about 35 degrees.

37. The tubular connection of claim 36, wherein said negative load flank angle is less than 4 degrees, wherein the absolute sum of said negative load flank angle and said stab flank angle is less than 5 degrees, and wherein said centering means includes full-height threads having chamfered stab flanks.

38. The tubular connection of claim 9, wherein each member's at least two threaded sections are comprises of a two step thread.

39. The tubular connection of claim 38, wherein said center shoulder seal configuration comprises a locked double shoulder configuration.

40. The tubular connection of claim 1 or claim 9 wherein said center shoulder seal configuration comprises a locked

double shoulder configuration.

~~41. The tubular connection of claim 1 or claim 9, wherein~~ the profile of said threaded sections is a single hook.

42. The tubular connection of claim 1 or claim 9, wherein the profile of said threaded sections is a double hook, and wherein the absolute sum of the positive stab flank and negative load flank angle is less than 5 degrees.

43. The tubular connection of claim 1 or claim 9, wherein the profile of said threaded sections is a single hook, wherein the negative load flank angle is less than 15 degrees, and wherein the stab flank angle ranges from zero degrees to a negative angle of about 35 degrees

44. The tubular connection of claim 1 or claim 9 wherein said center shoulder seal configuration is a square shoulder.

45. The connection of claim 9 wherein said center seal shoulder configuration includes a first shoulder having a first undercut surface on said box member and a second shoulder having a second undercut surface on said pin.

46. The connection of claim 45 wherein said first shoulder sealingly engages said second undercut surface and said second shoulder sealingly engages said first undercut surface.

47. The connection of claim 46 wherein said first and second undercut surfaces each have a portion thereof which interferringly engages said second and first shoulders respectively.

48. A tubular connection comprising two tubular members, wherein each connection member comprises threaded sections having a center shoulder disposed between two of said threaded sections, said center shoulders sealingly engaging to form a center shoulder seal configuration, each connection member having a leading terminal end and a trailing end, said threaded sections including a first run-out thread section adjacent said center shoulder seal configuration for engaging threads on another connection member and located between said center shoulder seal configuration and said leading terminal end, a second run-out thread section adjacent the trailing end of each connection member; a run-in threaded sections on each connection member located opposite said run-out threaded section on the other connection member; and said threaded sections having threads with chamfered stab flanks

49. The tubular connection of claim 48, wherein said center should seal configuration comprises a locked double shoulder configuration

50. A tubular connection comprising:

pin and box members having engaging threaded sections, said box member having a first center shoulder and said pin member having a second center shoulder, said first and second center shoulders forming a locked double shoulder configuration.

said first center shoulder including a first shoulder and a first undercut surface, said first shoulder and first undercut surface forming a surface extending therebetween;

said second center shoulder including a second shoulder and a second undercut surface, said second shoulder and second undercut surface forming a second surface extending therebetween;

said first shoulder sealingly engaging said second undercut surface and said second shoulder sealingly engaging said first undercut surface; and

said threaded sections including threads having means for centering said pin member within said box member during make-up.

51. The tubular connection of claim 50 wherein said first shoulder includes a first exterior sealing surface and said first undercut surface includes a first interior sealing surface; said second shoulder includes a second exterior sealing surface and said second undercut surface includes a second interior sealing surface; whereby said first exterior sealing surface forms a metal-to-metal seal with said second interior sealing surface and said second exterior sealing surface forms a metal-to-metal seal with said first interior sealing surface.

52. The tubular connection of claim 51 wherein said first and second external sealing surfaces are formed by beveled surfaces on said first and second shoulders.

53. The tubular connection of claim 51 wherein said box and pin members include ramp surfaces for aligning and guiding said first and second exterior sealing surfaces into sealing engagement with said second and first interior sealing surfaces respectively.

54. The tubular connection of claim 50 wherein said first and second surfaces are generally parallel to the axis of the connection.

55. The tubular connection of claim 50 wherein said first surface provides a seal with said second surface.

56. The tubular connection of claim 50 wherein said box member further includes a first interior ramp surface extending from said first undercut surface and a first exterior ramp surface extending from said first shoulder and said pin member further includes a second interior ramp surface extending from said second undercut surface and a second exterior ramp surface extending from said second shoulder, said first interior ramp surface sealing with said second exterior ramp surface and said second interior ramp surface sealing with said first exterior ramp surface.

57. A tube and coupling connection comprising:

a tube member and a coupling member having engaging threaded sections, said tube member having a first center shoulder and said coupling member having a second center shoulder, said first and second center shoulders forming a locked double shoulder configuration;

said first center shoulder including a first shoulder and a first undercut surface;

said second center shoulder including a second shoulder and a second undercut surface;

said first shoulder sealingly engaging said second undercut surface and said second shoulder sealingly engaging said first undercut surface; and

said threaded sections including threads having means for centering said tube member within said coupling member during make-up.

58. A method of stabilizing a tubular connection which comprises at least two tubular members having a leading terminal end with each member having at least two threaded sections with threads having chamfered stab flanks and having a center seal disposed between said at least two threaded sections, said method comprising:

engaging the chamfered stab flanks of the threads to center one tubular member with respect to the other tubular member;

sealingly engaging the center seals to form a center seal configuration;

utilizing a run-out thread section as one of said at least two threaded sections adjacent said center seal configuration on each connection member for engaging threads on another tubular member, and

locating said run-out thread section adjacent said center seal and between said center seal configuration and said leading terminal end to stabilize the center seal configuration.

59. The method of claim 58, including utilizing an additional run-out thread section near a trailing end of each connection member.

60. The method of claim 58 or claim 59, including utilizing a run-in thread section on one member with said run-out thread section at a corresponding location on the other member.

61. Threads disposed on pin and box members of a tubular connection comprising:

first and second pin thread sections having different diameters on the pin member;

first and second box thread sections on the box member, said first pin and box thread sections and said second pin and box thread sections adapted for threaded engagement;

each of said threads on said thread sections including chamfered stab flanks as the stab flanks for at least the individual thread members on the pin and box which come into contact upon initial telescoping of the pin member into the box member of the tubular connection; and

said first pin thread section adapted to stab through said second box thread section for engagement with said first box thread section whereby said chamfered stab flanks on said first pin and box thread sections contact and said chamfered stab flanks on said second pin and box thread sections contact for centering the pin member within the box member.

62. The threads of claim 61, wherein the threads include tapered threads and the height of said chamfer on said tapered threads is about equal to the step height from the preceding load flank to said stab flank of said tapered threads.

63. The threads of claim 61, wherein said threads have a load flank angle which is a negative angle of less than 15 degrees.

64. The threads of claim 63, wherein said threads have a stab flank angle which ranges from a negative angle of less than 8 degrees to a positive angle of less than 8 degrees.

65. The threads of claim 64, wherein said stab flank angle ranges from an angle of zero degrees to a positive angle of less than 3 degrees.

66. The threads of claim 65, wherein the absolute sum of said negative load flank angle and said zero degree to positive stab flank angle is less than 5 degrees.

67. The threads of claim 66 wherein said stab flank angle is essentially a zero degree angle.

68. Threads disposed on pin and box members of a tubular connection, each of said threads including chamfered stab flanks as the stab flanks for at least the individual thread members on the pin and box which come into contact upon initial telescoping of the pin member into the box member of the tubular connection, said threads being variable width threads having progressively increasing thread width.

69. The threads of claim 68, wherein the threads include tapered threads and the height of said stab flank chamfer on said tapered threads is about equal to the step height from the preceding load flank to said stab flank of said tapered threads.

70. A tubular connection member on a tubular pipe body and having a terminal end, at least two threaded sections and a center shoulder seal configuration disposed between said at least two threaded sections, said at least two threaded sections including at least one stabilizing thread section adjacent said center seal configuration, said stabilizing thread section comprising a first run-out thread, said threaded sections including threads having chamfered stab flanks.

71. The tubular connection member of claim 70, wherein said member includes a second run-out thread section adjacent said tubular pipe body.

72. The tubular connection member of claim 70 or claim 71, wherein said center-shoulder seal configuration comprises a double shoulder.

73. The tubular connection member of claim 72, wherein said threaded sections include a thread profile having a negative load flank angle of less than 15 degrees and a stab flank having an angle ranging form zero degree angle to a positive angle of less than 8 degrees.

74. The tubular connection member of claim 73, wherein said at least two threaded sections include full height threads, at least the full height threads being tapered and wherein the full height thread stab flanks which will come into contact upon initial telescoping of a connection pin member into a connection box member are chamfered.

75. The tubular connection member of claim 74, which the height of said chamfer is about equal to the step height between the preceding load flank and said chamfered stab flank.

76. The tubular connection member of claim 75, wherein said thread is a variable width thread having progressively increasing width as said threads progress away from said terminal end.

77. The tubular connection member of claim 70 or claim 71 wherein said threaded sections include a thread profile having a negative load flank angle of less than 15 degrees and a stab flank having an angle ranging from a zero degree angle to a positive angle of less than 10 degrees.

78. The tubular connection member of claim 77, wherein said at least two threaded sections includes full height threads, at least the full height threads being tapered and wherein the full height thread stab flanks which will come into contact upon initial telescoping of a connection pin member into a connection box member are chamfered.

79. The tubular connection member of claim 78, wherein the height of said chamfer is about equal to the step height between the preceding load flank and said chamfered stab flank.

80. The tubular connection member of claim 79, wherein said threads are variable width threads having progressively increasing width as said threads progress away from said terminal end.

81. A connection comprising:

a tubular box member having a terminal end and another end integral with a tubular pipe body;

a tubular pin member having a terminal end and another end integral with another tubular pipe body;

said pin and box members having a thick-walled portion adjacent said tubular pipe bodies and a thin-walled portion adjacent said terminal ends of said pin and box members;

said pin and box members each having a center shoulder between said thick-walled and thin-walled portions, said center shoulders sealingly engaging to form a center shoulder seal configuration;

said pin and box thin-walled portions having a run-out thread section adjacent said center shoulder seal configuration;

said pin and box thick-walled portions having a run-in thread section adjacent said center shoulder seal configuration;

said thread sections having means for centering said pin member within said box member; and

wherein upon the assembly of the connection, there is threaded engagement between said run-out thread sections and said run-in thread sections.

82. The connection of claim 81 wherein said pin and box thin-walled portions have a run-in thread section adjacent said terminal end and said pin and box thick-walled portions have a run-out thread section adjacent said tubular pipe body, whereby upon assembly of the connection, said run-out thread sections threadingly engage said run-in thread sections.

83. The connection of claim 82 wherein said pin and box thin-walled portions and thick-walled portions include a full height thread section disposed between said run-in thread section and said run-out thread section, whereby upon the assembly of the connection, there is threaded engagement between said full height thread sections.

84. The connection of claim 83 wherein said threaded engagements are complementary engagements.

85. The connection of claim 83 wherein the width of the threads of each of said portions constantly increases from said terminal ends to said pipe bodies.

86. The connection of claim 83 wherein the profile of said threaded sections is a single hook.

87. The connection of claim 86 wherein said single hook profile exhibits a negative load flank angle in the range of zero to five degrees and a zero degree stab flank angle.

88. The connection of claim 86 wherein said single hook profile exhibits a negative load flank angle and a negative stab flank angle.

89. The connection of claim 88 wherein said negative load flank angle ranges from zero degrees to less than five degrees and said negative stab flank angle ranges from zero degrees to less than two degrees with the absolute sum of said angles being less than five degrees.

90. The connection of claim 81 wherein said center shoulder seal configuration is a locked double shoulder seal.

91. The connection of claim 90 wherein said locked double shoulder seal is two to three times as high in the radial direction than it is wide in the axial direction.

92. The connection of claim 91 wherein said height is ¼ to ½ the thickness of the pipe body.

93. The connection of claim 81 wherein said center shoulder seal includes annular shoulders on a taper with the longitudinal axis of the connection.

94. The connection of claim 81 wherein the profile of said threaded sections is a double hook.

95. The connection of claim 94 wherein said double hook profile exhibits a negative load flank angle and a positive stab flank angle.

96. The connection of claim 95 wherein said negative load flank angle ranges from zero degrees to less than five degrees and said positive stab flank angle ranges from zero degrees to less than two degrees with the absolute sum of said angles being less than five degrees.

97. The connection of claim 81 wherein said pin and box members are made of formed metal.

98. The connection of claim 81 wherein the connection is a flush joint connection with the outer diameter of said box member being generally the same as the outer diameters of said tubular pipe bodies.

99. The connection of claim 81 wherein the connection is a slim-line connection with the outer diameter of said box member being no more than 3.5% greater than the outer diameters of the tubular pipe bodies.

100. The connection of claim 81 wherein the crests and roots of the threads interferringly engage upon power tight make-up.

101. The connection of claim 81 wherein there are clearances between adjacent stab flanks upon power tight make-up.

102. The connection of claim 81 wherein said load flanks butt up and engage upon power tight make-up.

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(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 767 335 A1

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
09.04.1997 Bulletin 1997/15

(51) Int Cl.6: F16L 15/06, E21B 17/042

(21) Application number: 96420301.2

(22) Date of filing: 23.09.1996

(84) Designated Contracting States:
AT BE CH DE DK ES FI FR GB GR IE IT LI LU MC NL PT SE

(30) Priority: 03.10.1995 US 538436

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(54) Threaded joint for tubes

(57) A joint of the integral type includes a male element 1 and a female element 2 with a diameter close to that of the tubes. Each of the elements has two shifted frustoconical threaded sections 4, 5, 6, 7 with vanishing threads at the two ends, by virtue of convergences of frustoconical envelope surfaces 16, 17, 18, and 19 and cylindrical surfaces 20, 21, 22, 23. A central abutment 24 cooperates with the negative-flank threads to reinforce the male/female connection and two internal 27 and external 28 ring-shaped metal/metal tightness zones, preventing the penetration of fluid into the threaded zones from the ends. The threaded joint may be used with tubes for very deep wells for which the reduction of the diameter of the assemblies makes it possible to lower costs, and other tubes for which a problem of the same type may arise.

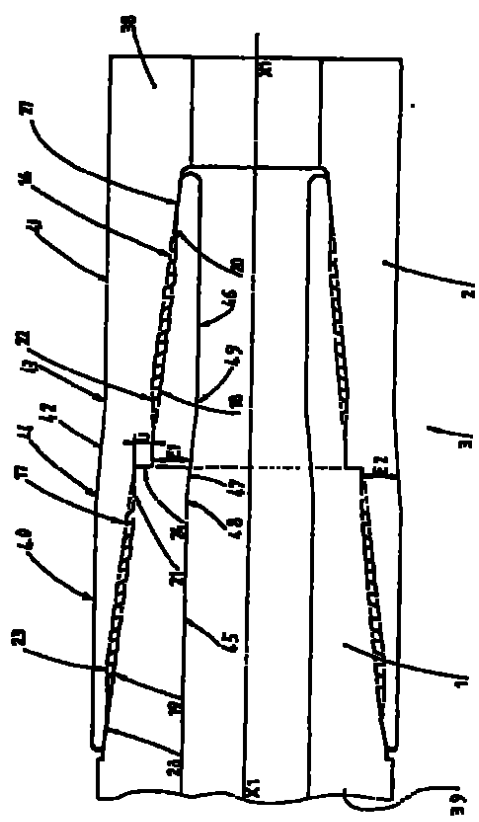

Fig:2

EP 0 767 335 A1

Description

The present invention relates to joints or assemblies of tubes to be joined by means of threadings, and concerns tubes used in industry and, in particular, assemblies or threaded junctions used in stringlines for tubing or for lines of tubular production accessories or for a casing for the operation or prospecting or exploitation of oil or gas wells, as well as threaded assemblies or junctions used for any purpose where it may be necessary to assemble pipelines or tubular accessories such as, for example, in geothermal or steam plants. The threaded assembly according to the invention is particularly useful in the assembly of metal tubes used for the casing of oil or gas wells, as explained below. The terms assembly or joint or junction will be used in the text as having the same meaning, except for particular contexts. By pipes, is meant any type of tubes or tubular components or tubular accessories currently in existence or likely to be put to use in industry, said tubes generally being metal tube.

Numerous types of assemblies are known for petroleum or gas carrying tubes that yield satisfactory results from the viewpoint of mechanical characteristics and tightness, even under tough conditions of use. Certain of these assemblies involve the use of tubes equipped with male frustoconical threadings at both ends, which are assembled by means of couplings having two corresponding female frustoconical threadings. This manner of assembly offers the advantage of rendering the two components of the assembly rigid, due to the existence of the positive interference that one can create between the male threading and the female threading.

However, the external diameter of these couplings is greater than that of the corresponding tubes and, when these assemblies are used with casing tubes, requires the drilling of bore holes with increased diameter. In the case of very deep wells, with a depth exceeding 4,000 meters, the initial diameter of the well first casing strings and consequently the diameter of the well in the vicinity of the surface may be twice as large using these couplings as it could be using slim couplings having an external diameter only slightly larger than the corresponding tubes of the casing strings.

To prevent this difficulty, one may use assemblies without a coupling or a sleeve. In this case, the tubular elements each have one male threaded end and one female threaded end, making for a thin assembly. These assemblies or junctions are generally called integral assemblies or junctions, in contrast to assemblies or junctions employing a coupling or a sleeve.

Such integral assemblies are generally made on tubes whose diameter at the end corresponding to the female threading is expanded, and whose diameter at the end corresponding to the male threading is reduced. This is done in order to have sufficient material in the thickness of the tubes to ensure the geometric and mechanical strength of the assembly that joins the tubes.

It is furthermore possible to reinforce the strength of the male-female assembly by using threadings in two successive sections instead of only a single one, with a change of dimensions when passing from one section to the other. Thus, U.S. patent No. 4,624,488 describes a threaded assembly comprising two cylindrical threaded sections having different threading diameters. This makes it possible to place a central ring-shaped abutment between these two sections. This abutment makes it possible to achieve sufficient tightening of the threadings while at the same time avoiding excessive onscrewing. In the case of threads with negative load flanks, the abutment makes it possible to tighten these threads on their negative flanks and this reduces risks of extraction due to the action of traction stresses that may or may not be combined with strong pressures.

U.S. Patent No. 4,570,982 describes a similar assembly comprising two threaded sections whose threading is cylindrical or frustoconical. The load flanks of the threads there have a negative inclination, and the abutment between the threaded sections has a generatrix that is inclined in the opposite direction. A sealing surface is provided on each element adjacent to the central abutment so as to ensure metal-to-metal tight contact due to the combined action of the abutment and the negative load flanks.

In the case of the two patents heretofore described, the thickness of each of the two components in the central zone, on the side of the abutment where its diameter is smaller, has a value that is further reduced by the height of the threads of the adjacent threading section. Thus, each component of the assembly has a zone provided with a critical thickness that is reduced when compared to the nominal wall thickness of the corresponding component.

One way of reducing the height of the frustoconical threads at the end of the components of a junction, in order to increase the mechanical strength thereof, is described in British Patent GB 1,587,836. In Figures 6 and 7 of that patent, one can see threading ends comprising so-called vanishing threads. However, this patent does not apply to making thin and high resistance assemblies.

Patent Application PCT WO 93/18329 describes a threaded assembly for tubes, having a tight connection made by use of a stabilized central shoulder. This assembly comprises two frustoconical threadings that are shifted with respect to each other, one of which presents vanishing threads at least at its end near the central shoulder. As shown in Figures 5 (A, B, C) or 10 (A, B, C), each component of the central shoulder is folded in the shape of an annular hook, the two hooks being suitable for penetrating each other. According to the description, this interpenetration makes it possible to create several tight metal/metal seals in series, which then retain very good tightness. The use of vanishing threads facilitates the assembly by reducing the risk of jamming male and female threads and it also makes it possible to increase the useful section of each of the two components of the assembly in the vicinity of the central zone.

Although the particular and complex structure of the central shoulder described in this PCT patent application may be designed to play a double role as abutment and seal joint, experience shows that it is difficult for such a component to perform these two functions in a satisfactory fashion. As a matter of fact, an abutment must preferably have high strength if one wishes to stop the advance of the male element into the interior of the female element at a certain point so as to prevent excessive on-screwing. One can achieve this result relatively easily when the central abutment is in a zone where the two components of the assembly have a large cross-section and are made so that they are tightly connected together. On the other hand, to get good sealability, it is necessary to have an elastic tightening of the surfaces in contact, because otherwise one runs the risk of being able to obtain tightness only by way of plastic deformation. In this case, the junction rapidly loses its sealibility qualities in the course of successive screwing and unscrewing operations. This sealibility loss is essentially due to the fact that surfaces are damaged by galling.

The aforementioned PCT Patent Application indicates that the use of lubricants, such as API lubricant, makes it possible to improve the sealibility of the joint. But this type of lubricant has the disadvantage, particularly at relatively high temperatures, of leaving solid deposits in the ring-shaped folds, such as those shown in Figures 5 (A, B, C). These deposits then obstruct a correct interpenetration of the hooks. Furthermore, the threaded assembly described in this PCT document employs specific threading shapes, in particular, in the form of dovetails that are more difficult and more expensive to manufacture and to use than standard thread forms known for this type of application.

Considering these problems, the present inventors looked into the possibility of making a thin threaded assembly for tubes, in particular, for petroleum carrying tubes and other tubes used most often for drilling at great depths, with a simple and effective structure. The present inventors first tried to make a central abutment structure as strong as possible in resisting deformation in order to permit setting the depth of advance of the male element into the female element with great precision. In the vicinity of the central assembly zone, the present inventors tried to obtain a critical thickness for each of the two male and female elements that would be as great as possible, while resorting to assemblies whose outside diameter is close to or a little greater than that of a mid-portion of the tube body on which the assembly is cut.

The possibility of making metal/metal seals that would also be effective against internal and external over-pressures and that, in particular, would be capable of protecting the threadings of the assembly over their entire length against the penetration of fluids coming from inside or outside was also examined. The present inventors furthermore researched the possibility of giving these metal to metal seals the desired elasticity so that they may retain their tightness in spite of the elastic deformations resulting from traction or axial compression that may or may not be combined with internal or external over-pressures or even elastic bending of the assembly with respect to its axis.

The present inventors also explored the possibility of preserving this kind of sealability over many screwing and unscrewing cycles.

Finally, the present inventors explored the possibility of achieving all of these results by means of simple machining methods, permitting easy mounting of the assembly in a single operation, where the central abutment makes it possible not only to control the mechanical tightening conditions of the assembly but also the conditions for obtaining sealability.

It is an object of the invention to provide a threaded joint for tubes of the integral joint type that achieves some or all of the above results.

The threaded joint, intended to connect two tubes according to the invention, consists of a male element and a female element placed at the ends of the tubes to be joined. The male element is equipped with an external threading and the female element is equipped with an internal threading corresponding to the threading of the male element. Each of these threadings consists of two radially shifted frustoconical sections, with the two sections of each element being separated by a ring-shaped abutment surface. Each section is equipped with vanishing threads at each of its ends.

The threadings are advantageously made up of threads whose load flank has a negative slope.

The height of the vanishing threads in each threading end zone is limited by the convergence of an external or internal frustoconical surface, situated along the prolongation of a mid-portion of the threading and an interior or exterior cylindrical surface parallel to the axis of the tube.

The height of the ring-shaped abutment surface is essentially equal to the radial spacing of two cylindrical surfaces that limit the heights of the vanishing threads adjacent to said central abutment surface. At the end of the screwing operation, engagement of the abutment surfaces of the male and female elements causes the tightening of the load flanks of the male threads against the female threads. When these threads have load flanks with negative angles, the radial component of the tightening force enhances the radial cohesion of the male and female elements to each other. The abutting abutment surfaces form a shoulder-like central abutment. The term abutment or abutment surface or shoulder will be used hereinafter as having the same meaning, except for particular contexts as the case may be.

On the male element and on the female element, in each of the threading-free zones situated beyond the end of each of the threaded sections opposite to the central abutment surface, there is positioned a seal surface. The two sealing surfaces situated on the same side of the central abutment being dimensioned, positioned, and designed to correspond to each other and to constitute a ring-shaped, fluid tight metal/metal contact zone on the assembled joint.

In this way, the two sealing surfaces which, on each side of the joint, constitute a ring-shaped tightness zone against the, respectively, internal and external pressure of the fluid circulating in the tubes or outside the tubes, are tightened elastically against each other when the male element is made to abut against the female element at the ring-shaped central abutment.

These sealing surfaces may be constituted by all surfaces which create, during the assembly of the joint, an elastic deformation that ensures metal/metal contact, all of the sealing surfaces not having necessarily the same shape.

These seal surfaces are advantageously formed as frustoconical surfaces. The angles of these frustoconical surfaces may advantageously be between .8° and 12°, the angle being measured with respect to the common longitudinal axis of the joint. These values are by no means restrictive. The values of the angles corresponding to one pair of bearings are not necessarily the same as the values corresponding to the other pair. Generally speaking, these bearings are individually designed to obtain on the installed joint a tightness that results from a deformation that remains within the elastic domain.

As described later on, the different factors cooperating in making the threaded joint according to the invention permit one to obtain a joint with a diameter close to or only a little greater than that of the tube, which has excellent mechanical characteristics and excellent tightness.

As noted earlier, the joint comprises two frustoconical threading sections. At each of its ends, each section has a small number of threads that are called vanishing threads, for which the very small initial height of each thread increases from the start of the threading in an essentially linear fashion until it attains the height which is that of the threads in the mid-portion of the threading. This result is obtained for the two threading sections of the male component by machining the roots of the threads at the start of the large-diameter zone of each of these sections with a constant slope and, hence, a diameter that decreases from the point of departure and up to the end of the mid-portion of the section. One machines the crests of the corresponding male threads to have a constant diameter, and thus to have a height that increases from the point of departure of the threading up to the start of the mid-portion and then, up to the end of the section with the constant thread height, extending parallel to the slope of the roots of the threads.

Starting from the small-diameter end of the mid portion, one machines the roots of the male threads with constant diameter, whereas the crests of the male threads continue to be machined at constant slope up to the intersection of the frustoconical surface and the cylindrical surface.

For the female element, machining is done in such a way that the male and female threadings will penetrate each other in the vanishing zones of the threads, even though the residual height of the latter is very small. To achieve this, one does the opposite of what has just been described for the male element. For each of the two sections of the female threading, starting with the beginning of the large-diameter zone, one machines the roots of the threads with a constant diameter up to the beginning of the mid-portion and then at a constant slope, including the final zone with vanishing threads. The crests of the threads are machined at constant slope in the zone of increasing height and in the mid-portion, then with a constant diameter in the final zone with the vanishing threads.

One can see that, in the assembly thus obtained, the critical thickness zone in the mid-portion of the male element corresponds to the diameter of the intersection of the frustoconical envelope surface of the roots of the male threads with the cylindrical envelope surface of the crests of the male threads of the threading section that is closest to the end of the corresponding tube. For the female element, the critical thickness zone in the mid-portion of the female element corresponds to the diameter of the intersection of the frustoconical envelope surface of the roots of the female threads with the cylindrical envelope of the crests of the female threads of the threading section that is closest to the end of the corresponding tube. One can see that the radial distance between the two diameters thus defined corresponds to the maximum height that can be given to the ring-shaped abutment zone between the two threaded sections. The use of vanishing threads, as described, makes it possible, for a certain abutment height, to give the critical thickness of each element a value that will be as great as possible. The number of vanishing threads at each end of each threading section depends on the height of the threads at the mid-portions and the pitch. In practice, each of these terminal zones comprises most often 3 to 4 threads; but this value is not in any way restrictive.

The mechanical strength of the two elements of the joint with respect to each other depends to a good deal on the profile of the threading. One advantageously uses threadings whose load flank ensures metal/metal contact on the joint in the screwed condition with a negative inclination, that is to say, where the load flank of the thread has a generatrix that is inclined toward the outside of the thread. The negative angle is most often between 3° and 20° with respect to a plane perpendicular to the axis of the element, and it is preferably between 5° and 15°. As indicated earlier, this inclination makes it possible, by tightening the male and female elements after they have been made to abut, to tighten the threaded zones against each other and thus to prevent the risks of disjointing at the threads.

The central shoulder abutment, whose height one determines as a function of the dimensions of the male and female elements and the threadings, can have a zero slope with respect to a straight plane extending perpendicular to the axis of the component. It can also have an inclination between 0° and about 20° with respect to such a straight plane. This inclination is oriented in the direction toward the inside of the male element. This inclination contributes to tightening the two components of the assembly against each other in the central part.

The stabbing flank is preferably inclined toward the interior of the thread with respect to a straight plane extending perpendicular to the axis, at an angle that is greater than or equal to the angle of the load flank, comprised, for example, between 8° and 30°.

To facilitate the stabbing of the male threads in the female threads, one may advantageously equip the stabbing flanks with a chamfer close to the thread crest. This chamfer, which has an angle greater than that of the stabbing flank, has a slope that is increased with respect to the zone of that same flank that is closer to the root of the thread.

The height of the threads in the mid-portion is, for example, about 6 to 12 % of the thickness of the tube. The slope of each of the two threading sections is for example, about 1/16 to 1/10 with respect to the axis; none of these values is restrictive.

If one gives the central abutment a height that is not excessive, then it is possible to make a joint whose critical thickness in the male and female components in the vicinity of the abutment is sufficient, particularly thanks to the two frustoconical threading sections with vanishing threads, to provide excellent mechanical characteristics.

The height of the central abutment is for example, about 10 to 35% of the thickness of the tube.

As noted earlier, one cannot achieve good metal/metal tightness in the vicinity of the shoulder due to the great rigidity of the components in this zone. On the contrary, it is possible, by putting the seal surfaces on the joint as explained earlier, to achieve metal/metal tightness under optimum conditions at the two ends of the assembly. A ring-shaped internal tight metal/metal contact zone is made between the small-thickness end of the male element, beyond the small-diameter end of the threading, and the large-thickness end of the female element, and a second ring-shaped tight metal/metal contact zone is made between the small-thickness end of the female element, beyond the threading, and the large-thickness corresponding end of the male element. The opposite seal surfaces of the male and female elements have geometries, dimensions, and positions such that, during the screwing of the joint, the end of the thin-wall element will be elastically deformed in the ring-shaped tight metal/metal contact zone with a slight variation of its diameter, while the diameter of the thick wall remains essentially unchanged.

One preferably gives these seal surfaces such a profile that the tightness metal-metal contact will take place in a narrow but well-defined ring-shaped zone.

The extension of the seal surfaces along the axis of the joint is determined so that the relative movements of one of the elements of the assembly with respect to the other one resulting from the traction or compression efforts exerted on the ends, or resulting from the external or internal over-pressures, do not modify the tightness performance of these external and internal seals, even if there are measurable displacements between the two bearings of each of the ring-shaped tight metal/metal contact zones.

Due to these ring-shaped metal/metal seal zones, one can prevent the penetration and the contact with threadings of pressurized fluids coming from the inside or outside. One can thus prevent any and all risk of corrosion of threadings or of abutment zones, as well as any risk of disjointing due to the effect of pressure.

To improve the quality of the tightness surfaces and, in particular, their resistance to galling, one can in the known fashion administer a surface treatment to the male and female elements of the assembly, for example by phosphatation with zinc or with manganese, or by depositing copper or any other metal, or by the application of a solid lubricating layer such as, for example, a layer of molybdenum bisulfide or any other treatment most often accompanied with suitable lubrication.

For certain uses, one can provide complementary tightness, consisting of a ring, for example, made up of elastomer, which one houses in the immediate vicinity of the edge of the distal end of the male element in a suitable groove. One can thus protect the tightness surfaces which ensure metal/metal connection against any attack by a corrosive fluid coming from the inside. Depending on the placement of this ring, one can also ensure the geometric continuity of the internal diameter (flush joint) and one can avoid any turbulence zone in the fluid stream.

Each tube to be assembled is equipped at one end with a male element and at its other end with a female element.

A more complete appreciation of the invention and many of the attendant advantages thereof will be readily obtained as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawings, wherein:

Figure 1 is a longitudinal half-section view of the male and female elements of the joint according to the invention, half cut away, along their X1-X1 axis and not assembled. The male and female threadings are represented only by the profiles of the envelope surfaces of the crests and roots of the male and female threads;

Figure 2 is a cross-section view of the joint according to the invention with the male and female elements assembled;

Figure 3 is a partial cut-away view along axis X1-X1 of the profile of a male frustoconical threading section of the joint according to the invention; and

Figure 4 is a partial cut-away view of the end zone of the male element of the joint according to the invention, showing the profile of the seal surfaces situated at the end of the male element.

Figure 1 is a longitudinal half-section view of the two elements, male element 1 and female element 2, situated at the end of two tubes 38, 39, only whose ends are shown, thereby forming joint 3, shown schematically in Figure 2, between the two tubes 38, 39. These tubes have the same inside and outside diameters, but for clarity of figure 1, the female element 2 is voluntarily represented shifted away from its axis X1-X1.

Each male and female element has two threading sections, respectively threading sections 4, 5, for male element 1 and threading sections 6, 7 for female element 2, between which is located a ring-shaped abutment surface or shoulder 24-1, 24-2. The mid-portions of these threading sections 4, 5 and 6, 7 are frustoconical.

The four frustoconical threading sections 4, 5 of the male element, and 6, 7 of the female element, have at each of their ends a zone of vanishing threads in which the heights of the threads diminishes to a zero value. The decrease in the height of the threads may be accomplished either by machining the crests of the threads to a constant diameter with respect to the axis of the male or female element, as can be seen in zones 8 and 9 of the male threading sections 4 and 5 as well as in zones 10 and 11 of the female threading sections 6 and 7, or by machining constant diameter thread roots with respect to the axis, as in zones 12 and 13 of male threading sections 4 and 5 as well as in zones 14 and 15 of female threading sections 6 and 7. One can thus see that, by assembling the two male and female elements, their threads engage completely in the corresponding housings, both in the mid-portions of the threadings and in the end zones with the vanishing threads.

As shown in Figure 2, in these end zones, the crests and roots of the vanishing threads are limited toward the outside or toward the inside by the convergence of an external frustoconical surface 16, 17 or an internal frustoconical surface 18, 19, each of which prolongs the surface of the mid-portion of the threading, and an internal cylindrical surface 20, 21 or an external cylindrical surface 22, 23. One can see that the difference between the diameters of these cylindrical surfaces 21 and 22 corresponds to the radial height "D" of the ring-shaped abutment or shoulder 24 in the central zone of assembly 3. This ring-shaped abutment or shoulder 24 is made up by having the two surfaces 24-1, 24-2 of male element 1 and female element 2 (Figure 1) abut against each other. As a non-limiting example the radial height « D » can be between 10 and 35% of the thickness of the tube.

It is to be noted that abutment 24 does not perform any tightness or sealing function. The generatrix of abutment surface 24-1, 24-2 is perpendicular to axis X1-X1 of the assembly. One can eventually incline this generatrix with respect to the axis X1-X1 by an angle of about 90 to 70°, this generatrix being oriented toward the inside and inner end of the male element when going from its larger diameter point to its smaller diameter point.

The load flanks of the male threads, such as 30 (Figure 3), have a generatrix with a negative inclination A of about -3 to -20°, and preferably -10° ± 5°, with respect to a line extending perpendicularly to axis X1-X1 of the element. Upon screwing, the cooperation between these threads with negative load flank and abutment 24 makes it possible to tighten male element 1 and female element 2 against each other. This practically eliminates the risks of disjointing or separation at the threads.

The shoulder (or abutment) surface 24-1 on the male and 24-2 on the female, perpendicular to the X1-X1 axis, increase, for a given radial difference D, the mechanical strength of the joint, and consequently makes it also possible to give the critical thicknesses E1 on male and E2 on the female, a value as large as possible. The absence of metal/metal tightness surfaces in the central zone -- surfaces whose effectiveness is, as already explained, not satisfactory due to the rigidity of this zone -- makes it possible to move the two frustoconical threading sections 4, 5 and 6, 7 closer to each other, and thus to improve the joining action between the two male and female elements.

To facilitate the insertion or stabbing of the male threads in the female housings, one advantageously gives the stabbing flanks, such as 31, a slope whose generatrix has a positive inclination that is greater in absolute value than that of the negative inclination of the load flank angle, the value of the inclination being about +8° to +30°. In the case of this example, one thus gives the load flanks a negative inclination of -10° and the stabbing flanks an inclination of about +25°. One may possibly further facilitate stabbing by making chamfers in the upper ends of the stabbing flanks, thereby producing a slope that is increased by about 15 to 35° with respect to the adjacent parts of the stabbing flanks which extend to the bases of the stabbing flanks.

As shown in Figures 1, 2 and 4, two fluid tight metal to metal seal surfaces 27, 28 are arranged at the internal and external ends of the joint, beyond the ends of the threaded portions. In the example shown, these seal surfaces are essentially frustoconical surfaces on the male and female elements having the desired inclination so as to come into contact with each other, said contact being accompanied by an elastic deformation of the respective thin ring-shaped wall. This elastic deformation

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is in compression for the distal end of the male element and in expansion for the distal end of the female element.

Figure 4 shows the distal end of male element 1, fully screwed to the bottom in female element 2, in a half-section along axis X1-X1. It clearly shows the interference that is provided between tightness surfaces 27-1 and 27-2, which are shown for clarity of the explanation with overlap, the last vanishing threads appearing at 32. In fact, the elastic deformation by way of compression of the distal end 33 of male element 1 allows its tightness surface 27-1 to elastically press against the corresponding surface 27-2 of the thick portion of female element 2, while the elastic deformation takes place almost exclusively on distal end 33.

The ring-shaped tight metal/metal contact zone thus made on the inside of the assembly protects the threadings against the penetration of fluids contained inside the tubes. To prevent risks of corrosion of the tightness surfaces themselves, one may, if necessary, put an elastomer ring 29, filling the space between distal end 33 of male element 1 and the internal end 35 of the female element. This deformable ring will advantageously be dimensioned to fill the volume between end 33 of male element 1 and the internal end 35 of the female element and will ensure the continuity of passage from the inside diameter 36 of tube 38 having the female element to the inside diameter 37 of the male element.

The structure of the external metal/metal seal surface 28, made between the thin-wall distal end 34 of female element 2 and the thick wall part of male element 1, is similar to the one that has just been described for the bearing 27. At the time of tightening the joint there is interference between the tightness surfaces that come into contact against each other, accompanied by elastic expansion of the thin female wall.

The double protection of the threading thus obtained turns out to be particularly effective at preventing any risk of over-pressure inside the threadings. The elasticity of the thin distal ends of male element 33 and female element 34 allows them to keep up with any relative movement of the end zones of one assembly element with respect to the other, particularly in case of traction or compression effects.

It is important to note that the only surfaces to abut against each other are ring-shaped abutment surfaces 24-1, 24-2. In no case does distal end 33 of male element 1 abut, during the screwing of the joint, against the internal end 35 of female element 2. The same is true of distal end 34 of female element 2 with an opposing surface at the end of male element 1.

Depending on the dimensions of the tubes on which the integral joint according to the invention is to be made, one could, if necessary, make the outside diameter of the end of female element 2 so as to have additional thickness at the level of threading section 7 and end 34. This can be done by expanding, over at least one portion of the end, the outside diameter 40 of this female element 2 prior to the manufacture of the elements of the joint. Thus, Figure 1 shows a female element 2 that has retained the original diameter 41 of the tube over a portion of the length. This diameter 41 is joined to the expanded diameter 40 of end by an essentially frustoconical connection shape 42 between regions labelled 43 and 44.

Likewise, one may proceed in a similar fashion in the case of the inside diameter 45 of tube 39 in which male element 1 is made. In this case, as shown in Figure 1, one reduces the inside diameter 46 over at least a portion of the end. Here, this diameter is reduced at the level of threading section 4 and at end 33. The connection is made, as before, in the form of a truncated cone at 47 between regions labelled 48 and 49. In this way, one gets greater strength without the inconvenience of using a coupling or sleeve.

The expansion of outside diameter of the female element 2 and the reduction of the inside diameter of the male element 1 are also shown in Figure 2.

The integral joint according to the invention, thus built, makes it possible to combine very great mechanical strength due to the extremely compact structure of this assembly, in its central zone, with excellent tightness, due to the placement of ring-shaped tight metal/metal contact zones, employing tightening by elastic deformation of the seal surfaces. As indicated earlier, appropriate surface treatments, well known in the art, make it possible to considerably improve the galling resistance of the tightness surfaces and their aptitude to undergo numerous screwing-unscrewing operations without losing their tightness.

Obviously, numerous modifications and variations of the present invention are possible in light of the above teachings. It is therefore to be understood that the invention may be practiced otherwise than as specifically described herein.

Claims

1. A threaded joint for tubes, comprising:

   - a male element having external threading comprising two radially shifted frustoconical sections of threadings having load flanks with a negative angle, said two frustoconical sections of said external threading being separated by a first ring-shaped abutment surface, said male element having at opposite axial ends thereof external surfaces forming tightness surfaces, each of said two frustoconical sections of said external threading having vanishing threads at opposite ends thereof, and
   - a female element having internal threading comprising two radially shifted frustoconical sections of threadings having load flanks with a negative angle, said two frustoconical sec-

tions of said internal threading being separated by a second ring-shaped abutment surface, said female element having at opposite axial ends thereof internal surfaces forming tightness surfaces, each of said two frustoconical sections of said internal threading having vanishing threads at opposite ends thereof,

wherein said male and female elements are mutually configured such that said male and female elements may be united by threading said external and internal threadings until said first and second abutment surfaces abut one another, wherein:

a) crests and roots of each of said vanishing threads are radially limited by a convergence of a frustoconical surface of one of said male and female elements and a cylindrical surface of the other of said male and female elements,

b) the abutting first and second abutting surfaces form a central abutment, and

c) the tightness surfaces of said male and female elements mate in pairs to form two fluid tight seal surfaces having ring shaped contact zones.

2. The threaded joint of claim 1 wherein the cylindrical surfaces include two radially shifted cylindrical surfaces meeting at the first ring-shaped abutment surface and separating said external threading, and two radially shifted cylindrical surfaces meeting at the second ring-shaped abutment surface and separating said internal threading.

3. The threaded joint of claim 1 wherein a generatrix of each of said first and second abutment surfaces is inclined, with respect to the axis of the joint, said inclination angle being between 90 and 70°.

4. The threaded joint of claim 1 wherein a radial height of each said abutment surface is between 10 and 35% of a thickness of the tube.

5. The threaded joint of claim 1 wherein the height of the threads in the mid-portion is between 6 to 12 % of the thickness of the tube.

6. The threaded joint of claim 1 wherein the negative angle of the load flanks of each said threading has a value between 3° and 20°.

7. The threaded joint of claim 1 wherein stabbing flanks of each said threading have a positive angle between 8° and 30°.

8. The threaded joint of claim 7 wherein the stabbing flanks have a chamfer having a positive inclination between 15° and 35° with respect to an adjacent part of stabbing flanks.

9. The threaded joint of claim 1 wherein said frustoconical sections of said male and female elements and said tightness surfaces are sloped such that a thickness of said male and female elements decreases toward distal ends thereof, wherein each of said seal surfaces comprises one of said tightness surfaces on a thin distal end of the respective one of the male and female elements mating with another of said tightness surfaces on a thick end of the respective one of the male and female elements such that the thin distal end is elastically deformed when said male and female elements are united by threading said external and internal threading until said first and second abutment surfaces abut one another.

10. The threaded joint of claim 1 wherein said tightness surfaces are frustoconical.

11. The threaded joint of claim 1 including a tightness ring housed in said female element such that the tightness ring occupies a space between the distal end of said male element and an internal end of said female element when said male and female elements are united by threading said external and internal threading until said first and second abutment surfaces abut one another.

12. The threaded joint of claim 1 wherein the slope of the frustoconical threadings is between 1/16 and 1/10 with respect to the axis of the tube.

13. The threaded joint of claim 1 wherein the female element is formed on a tube whose outside diameter is larger than an outside diameter of the mid-portion toward the distal end of the female element.

14. The threaded joint of claim 1 wherein the male element is formed on a tube whose inside diameter is smaller than an inside diameter of the mid-portion toward the distal end of the male element.

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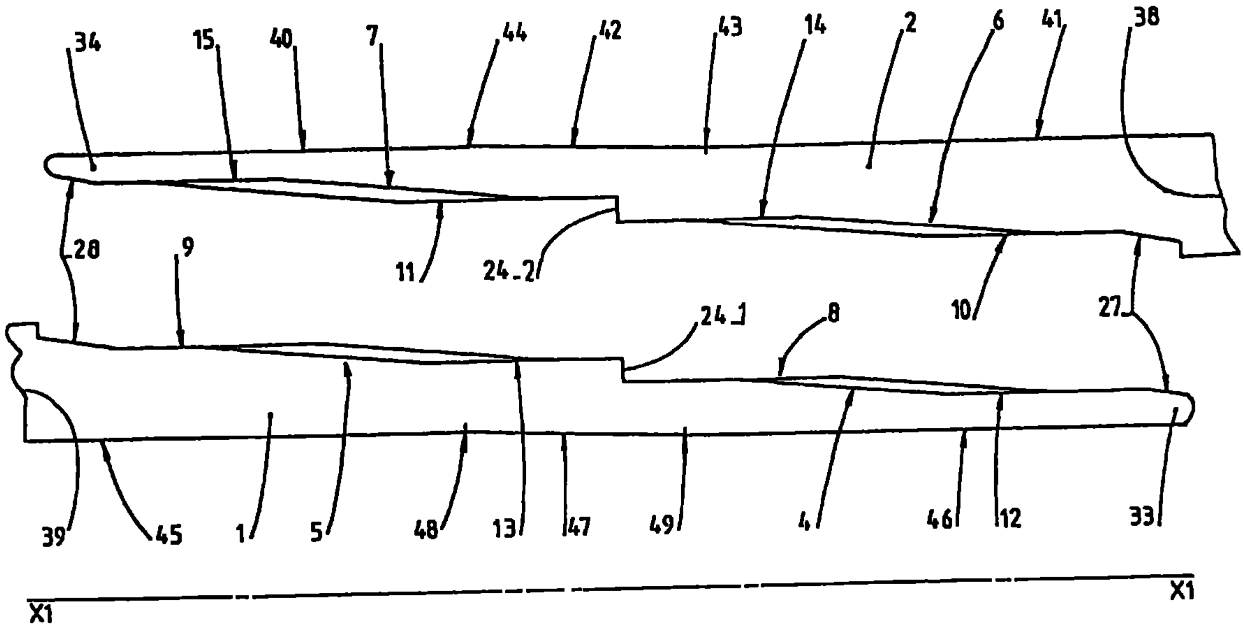
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Fig 1

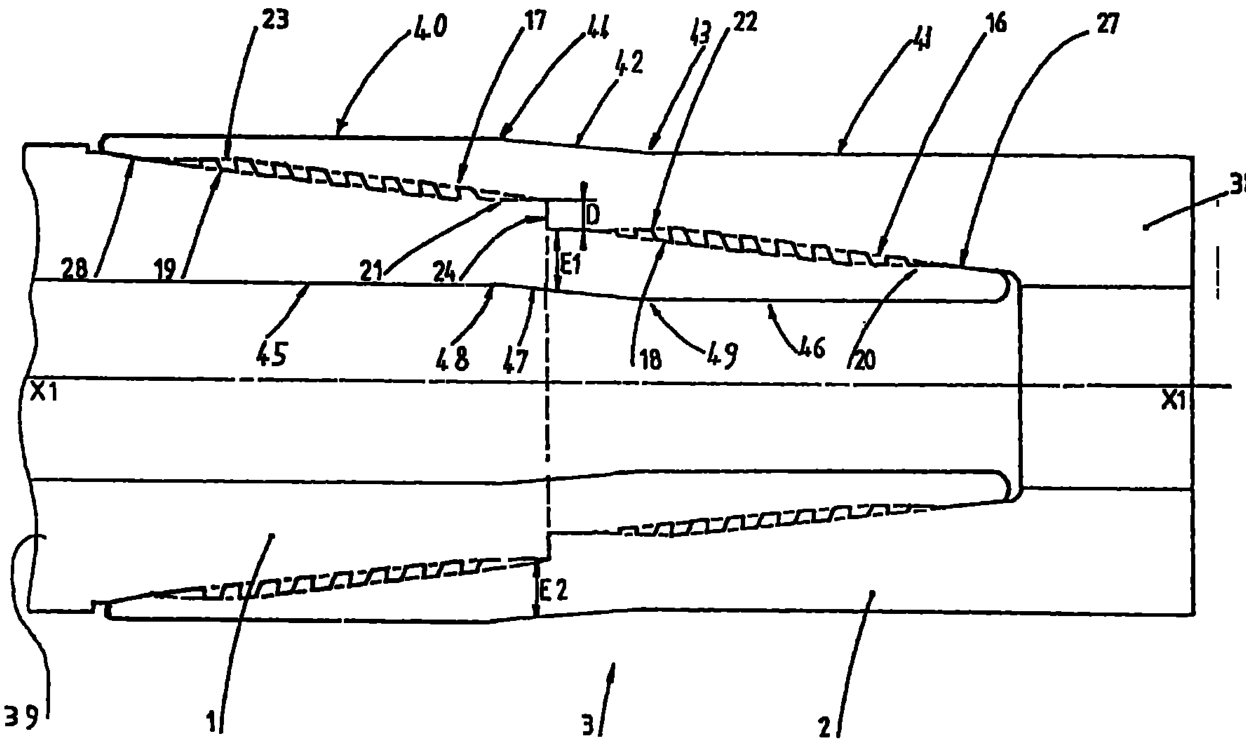
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Fig:2

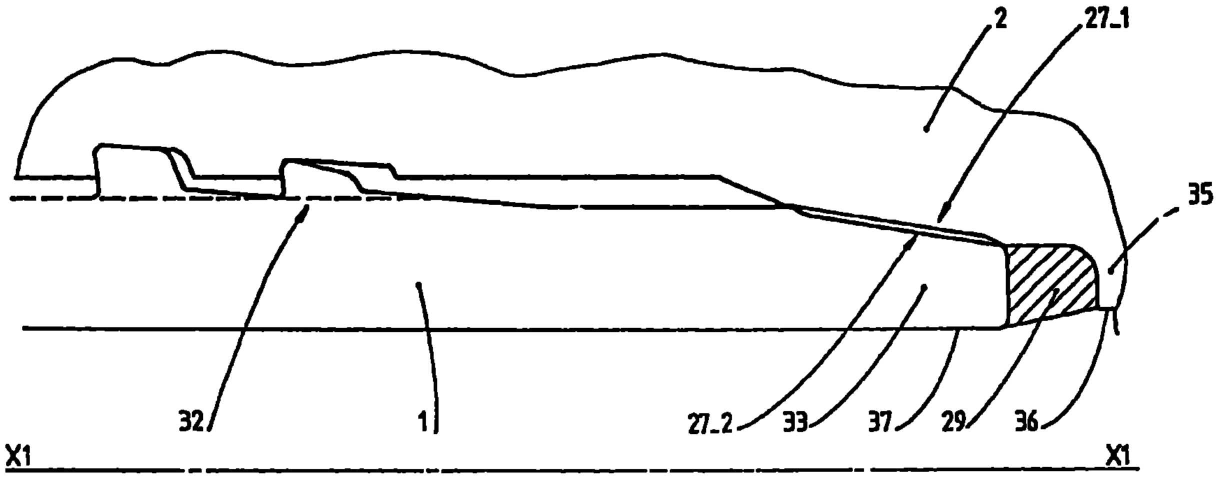

Fig: 4

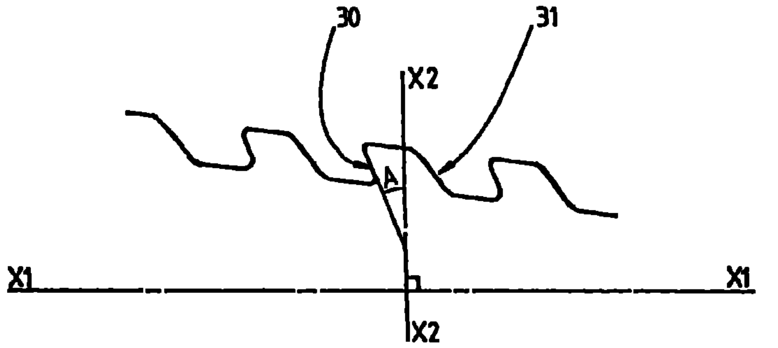

Fig: 3

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 96 42 0301

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
Y	US-A-4 253 687 (MAPLES JOHN H) 3 March 1981 * claims; figures 1-4,6 * ---	1-3,6,7,9,13,14	F16L15/06 E21B17/042
Y,D	WO-A-93 18329 (MARUBENI TUBULARS INC) 16 September 1993 * figures 1,2 * -----	1-3,6,7,9,13,14	
			TECHNICAL FIELDS SEARCHED (Int.Cl.6)
			F16L E21B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	17 December 1996	Budtz-Olsen, A

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

#39

United States Patent [19]

McDonald

[11] Patent Number: 4,676,529

[45] Date of Patent: Jun. 30, 1987

[54] PIPE JOINT

[75] Inventor: Patrick E. McDonald, Houston, Tex.

[73] Assignee: Hydril Company, Houston, Tex.

[21] Appl. No.: 843,385

[22] Filed: Mar. 24, 1986

[51] Int. Cl.⁴ F16L 55/00

[52] U.S. Cl. 285/92; 285/334; 285/350; 285/382

[58] Field of Search 285/334, 333, 355, 350, 285/390, 382, 92

[56] References Cited

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Primary Examiner—Richard J. Scanlan, Jr.
Attorney, Agent, or Firm—Vaden, Eickenroht, Thompson & Boulware

[57] ABSTRACT

A pipe joint of the free-running thread type is provided with engaging shoulders on the pin and box that are positioned for the metal adjacent the shoulders to be stressed by make-up torque beyond its yield joint and extruded laterally of the shoulders into grooves to mechanically lock the joint in its made-up position.

7 Claims, 5 Drawing Figures

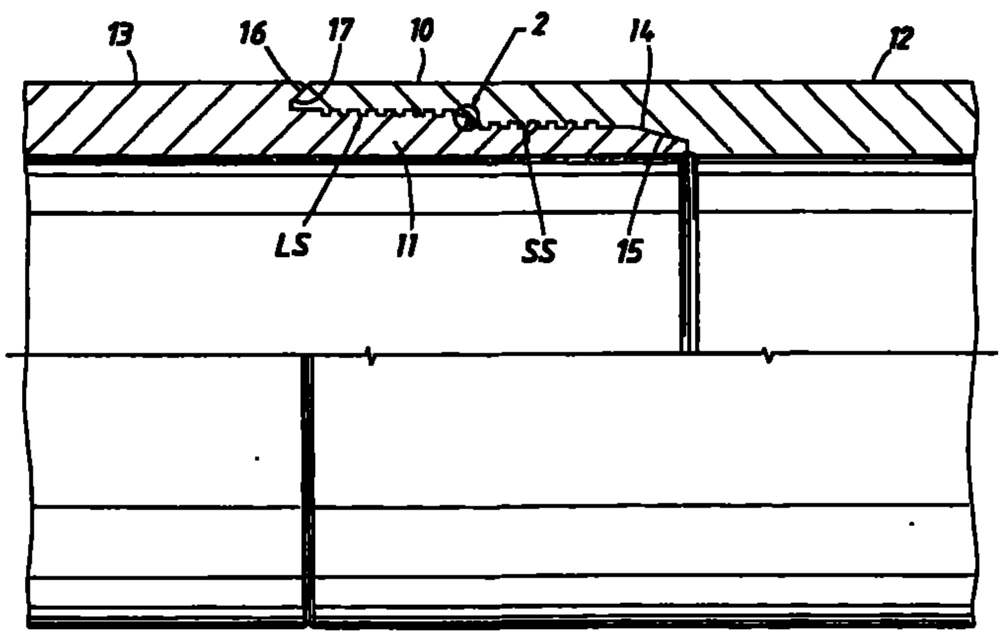

FIG. 1

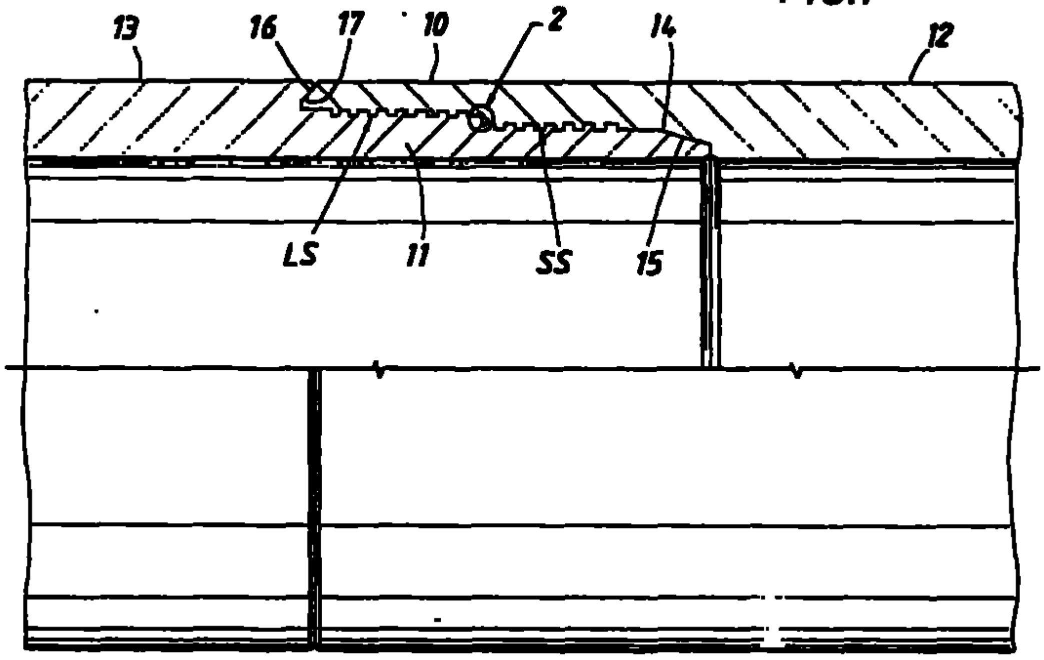

FIG. 2A

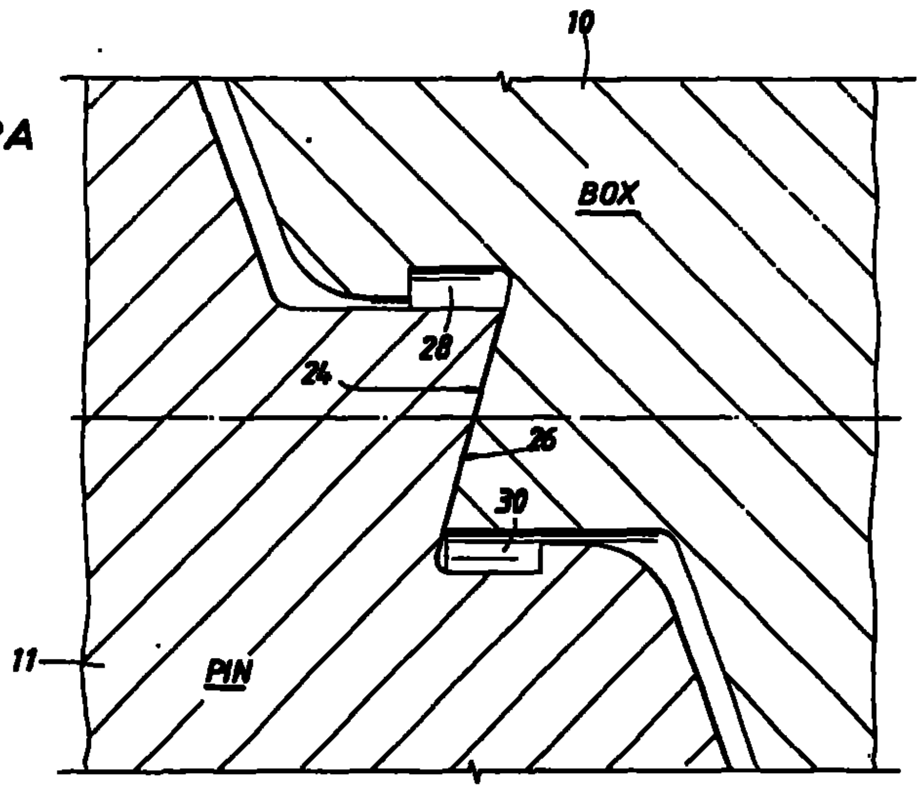

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FIG. 2B

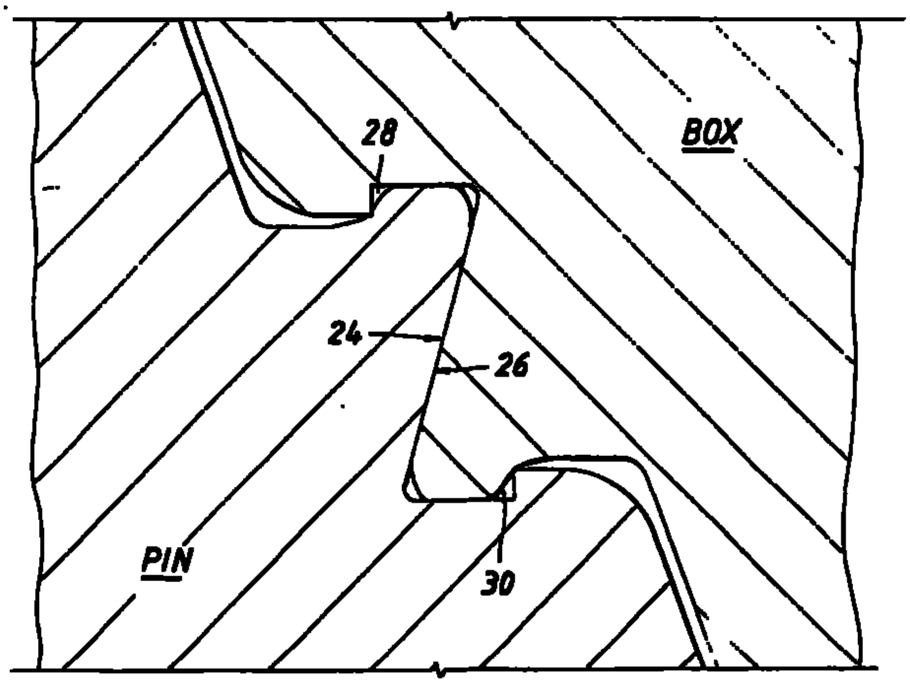


FIG. 3A

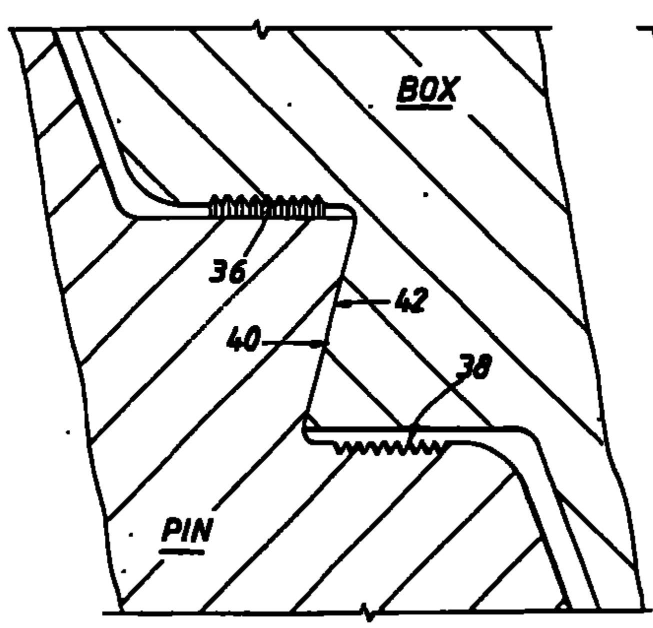
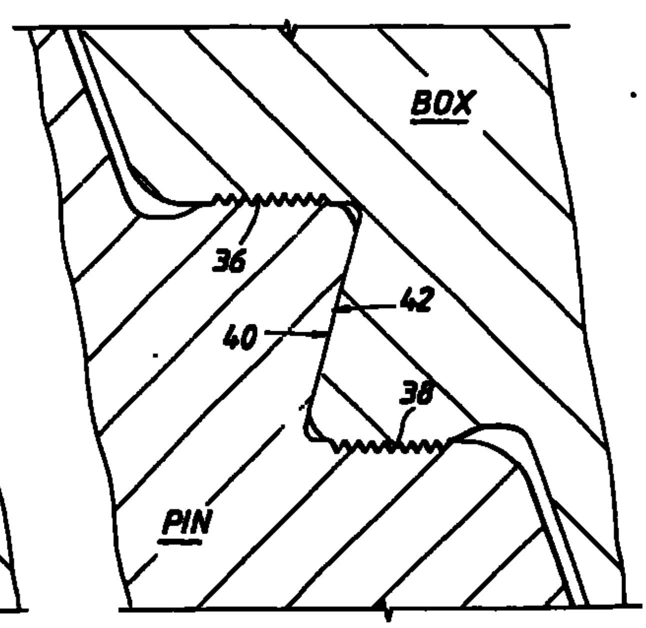


FIG. 3B



PIPE JOINT

This invention relates generally to a pipe joint for use with oil country tubular goods, and, in particular to a pipe joint having free running threads.

Threaded joints of the type that use cylindrical or tapered threads of a modified buttress type are designed to provide clearance between the flanks of the threads to allow the joints to be made-up with little torque required until the sealing and shouldering surfaces engage. There may be just one pair of shoulders that not only act as torque shoulders to limit the distance the pin can move into the box, but also define sealing surfaces to effect sealing of the joint. Alternatively, there may be two or more pairs of engaging surfaces where one or more act to seal the joint and one or more act as torque or stop shoulders. Or the joint may have two sets of engaging sealing surfaces and separate torque shoulders that can also provide a seal.

In any event, after the joint is made-up hand-tight, the torque required to completely make-up the joint serves to force the sealing and/or shouldering surfaces of the joint together. The normal forces developed between the surfaces and the threads create the friction that resists conditions or forces tending to cause the connection to unscrew. As the frictional force between the box pin is increased, the back off torque required to loosen the connection is also increased. Means for improving locking of the pin and box in the fully made-up position will enhance the sealing characteristics of the connection by reducing the probability of inadvertent back off.

Therefore, it is an object of this invention to provide an improved pipe joint having threads of the free-running type described above that will lock the pin and box against relative rotation when the joint is made-up.

It is another object of this invention to provide an improved pipe joint of the type described above and a tubular member having threads for forming the pipe joint that includes a cavity on either the box or pin or on both into which a portion of the other member is extruded when the joint is made-up to form interlocking surfaces that prevent relative rotation between the two members, and thereby lock the pin and box in the fully made-up position.

It is another object of this invention to provide an improved pipe joint of the type described above in which the box and pin have engaging shoulders positioned to stress the metal adjacent the shoulders beyond the yield point and extrude the metal laterally of the shoulders into adjacent cavities in the box and pin to lock the box and pin from relative rotation.

These and other objects, advantages, and features of this invention will become apparent to those skilled in the art from a consideration of this specification, including the attached drawings and appended claims.

In the Drawings:

FIG. 1 is a partial longitudinal cross-section of a typical pipe joint having free-running threads embodying the improvement of this invention.

FIG. 2A is a sectional view of an enlarged scale of the portion of the joint of FIG. 1 within circle 2 when the joint is hand-tight.

FIG. 2B is the same view as FIG. 2A after the joint is made-up.

FIGS. 3A and 3B are views similar to FIGS. 2A and 2B of an alternate embodiment of the invention.

While the following description primarily uses the well known, free-running two-step, non-tapered, shouldered connections of the Hydril Company to illustrate the features of the invention, it will be understood that the invention described herein can be used with any type threaded connection having engaging shoulders where it is desirable to lock the pin and box together to resist relative rotation when the joint is made-up, such as in a pipe line.

The pipe joint shown in FIG. 1 includes box 10 and pin 11. The box and pin are formed on the ends of tubular members 12 and 13, respectively. The box has internal threads and the pin has external threads that mate with the threads on the box to allow the pin to be screwed into the box to make up the joint and connect the tubular members. In this joint, the threads are straight or cylindrical and they are stepped so that half of the threads have a smaller major and minor diameter than the other threads. The threads are shown in the drawing as being square, but in practice they are usually a modified buttress type, well known to those skilled in the art.

When made-up as shown in FIG. 1, conical sealing surface 14 on the end of pin 11 engages conical sealing surface 15 at the base of the box. Usually these sealing surfaces have about a 14° taper. At the upper end of box 10, conical sealing surface 16 is in sealing engagement with matching conical sealing surface 17 on pin 11. In the joint shown, sealing surfaces 16 and 17 are tapered at about 60° from the longitudinal axis of the joint.

As previously discussed, the metal-to-metal seal between these surfaces is lost with only a small degree of relative rotation of the box and pin. To prevent the pin and box from backing off, the present invention provides means for locking the pipe joint in a made-up condition.

In the embodiment shown, both the pin and box have annular shoulders located between the large step threads (LS) and the small step threads (SS) that engage as the joint is made-up. These shoulders and the adjacent portions of the box and pin are shown on an enlarged scale in FIGS. 2A and 2B.

Engaging shoulder 24 of box 10 and shoulder 26 of pin 11 are tapered along a plane that intersects the longitudinal axis of the pipe joint at an acute angle to provide the shoulders with frusto-conical mating surfaces. Preferably, the angle of inclination is between 75° and 90°. The angles of taper of the shoulders do not need to be the same, particularly if the mismatch will provide force components that urge the extruded metal to move in the desired direction.

Annular grooves 28 and 30 are located in the box and pin, respectively, on opposite sides of engaging shoulders 24 and 26. The shoulders are positioned so that they engage, as shown in FIG. 2A, before the joint is made-up so that make-up torque will create stress in the box and pin adjacent the shoulders that exceed the yield point of the metal from which the pin and box are made. As the metal yields, it will extrude laterally of the normal force between the two shoulders, i.e., the metal of the box adjacent shoulder 24 will extrude into cavity 30 and the metal of the pin adjacent shoulder 26 will extrude into cavity 28 as shown in FIG. 2B. Since any relative rotation between the box and pin tending to loosen the joint requires relative longitudinal movement of the box and pin and such longitudinal movement is prevented by the interlocking surfaces of the grooves and the extruded metal, the joint is effectively locked in the fully made up position.

In the alternate embodiment shown in FIGS. 3A and 3B, a plurality of side by side annular cavities form knurled surfaces 36 and 38 on the box and pin, on opposite sides of engaging surfaces 40 and 42. The extruded metal is forced into the cavities and lock the pin and box against relative rotation as shown in FIG. 3B when the joint is made up.

From the foregoing it will be seen that this invention is one well adapted to attain all of the ends and objects hereinabove set forth, together with other advantages that are obvious and that are inherent to the apparatus and structure.

It will be understood that certain features and subcombinations are of utility and may be employed without reference to other features and subcombinations. This is contemplated by and is within the scope of the claims.

Because many possible embodiments may be made of the invention without departing from the scope thereof, it is to be understood that all matter herein set forth or shown in the accompanying drawings is to be interpreted as illustrative and not in a limiting sense.

What is claimed is:

1. A metal pipe joint comprising a box having internal threads and a pin having external threads for mating with the internal threads of the box to make-up the joint, a shoulder on the box and a shoulder on the pipe for engaging the shoulder on the box as the joint is made up with sufficient force to extrude the metal of one of the box and pin laterally, and a cavity in the other of the box and pin positioned to receive the extruded metal and form interlocking surfaces between the extruded metal and the cavity wall to hold the box and pin from relative rotation.

2. The pipe joint of claim 1 in which one of the shoulders is frusto-conical.

3. The pipe joint of claim 2 in which both shoulders are frusto-conical.

4. The pipe joint of claims 1, 2 or 3 in which the cavity is one of a plurality of cavities forming a knurled surface.

5. For use in forming a pipe joint, a threaded tubular member adapted for connecting to another threaded member to form a threaded connection between the two members, comprising a tubular body, free running screw threads on the body, a cavity on the body and a shoulder on the body adjacent the cavity to engage a shoulder on the other threaded member that is positioned when the threaded member is connected to the other threaded member for a portion thereof to be extruded into the cavity to engage at least one of the walls of the cavity and prevent relative rotational between the members after they are connected.

6. The threaded tubular member of claim 5 in which the shoulder lies in a plane that intersects the longitudinal axis of the member at an acute angle.

7. The threaded tubular member of claims 5 or 6, in which the threads of the member are straight and are divided into a large step and a small step having a major diameter and a minor diameter smaller than the major and minor diameters of the large step and in which the shoulder is located between the large and small step.

* * * * *

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(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 916 883 A1

(12) EUROPEAN PATENT APPLICATION
published in accordance with Art. 158(3) EPC

(43) Date of publication:
19.05.1999 Bulletin 1999/20

(21) Application number: 97924279.9

(22) Date of filing: 30.05.1997

(51) Int. Cl.[6]: F16L 15/04

(86) International application number:
PCT/JP97/01855

(87) International publication number:
WO 98/54501 (03.12.1998 Gazette 1998/48)

(84) Designated Contracting States:
DE DK FR GB IT NL SE

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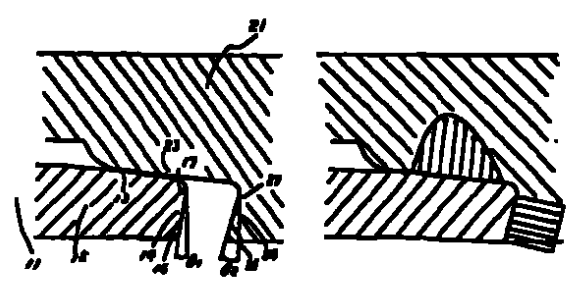
(54) SCREW JOINT FOR OIL WELL PIPE

(57) A screw joint for an oil well pipe is equipped with either one, or both, of an inner surface shoulder portion and an outer surface shoulder portion comprising a seal portion and a torque shoulder portion, and satisfies further the following conditions 1 and 2:1) effective diameter of seal formation surface before fastening is greater on the pin side than on the box side; and 2) inclination of torque shoulder formation surface is smaller on the pin side than on the box side, and difference Δθ of the inclinations is at least 0.50 and not greater than 4.0°. The screw joint for the oil well pipe of this invention is excellent in gas sealability and clearance corrosion resistance. The production cost is equal to that of conventional screw joints, and handling is easy, too. This screw joint is extremely suitable for probing and mining of burried matters at a depth and at a higher pressure and burried matters containing high corrosive impurities such as gases and crude oils.

Fig.5



(a) (b)

EP 0 916 883 A1

Description

TECHNICAL FIELD

[0001] The present invention relates to a threaded connection for connecting oil country tubular goods used for exploration and exploitation of natural gas and crude oil deposited underground, and particularly, to a threaded connection for the oil country tubular goods, suited for drawing up the deposits from deep oil wells under high pressure conditions and gas fields and oil fields containing large amounts of corrosive impurities.

BACKGROUND TECHNOLOGY

[0002] Threaded connections are in widespread use as connection for connecting the oil country tubular goods used for exploration of natural gas fields, oil fields, and the like located at depths of several thousands meters, and for exploitation of deposits therefrom. There are two types of threaded connections for the oil country tubular goods. One is an integral type of threaded connection provided with a pin at one end of a pipe, and a box at the other end of the pipe so that respective ends of two pipes are connected with each other by screwing the pin into the box. The other is a coupling type of threaded connection for connecting respective ends of two pipes with each other, each pipe provided with a pin at both ends, by means of a coupling provided with two boxes. The followings are performance required of these threaded connections in common:

- ability to withstand axial tensile force caused by weights of connected pipes themselves
- ability to withstand internal pressure caused by a fluid flowing inside the pipes, and external pressure caused by fluids flowing outside the pipes
- durability to enable repeated uses as frequent scores of times
- ability to resist corrosion and erosion caused by fluids flowing inside and outside the pipes

[0003] Lately, depths of oil wells have increased, and more often, oil is exploited from oil wells in severe environments, for example, in the polar regions. Consequently, in an increasing number of cases, the oil country tubular goods and the threaded connections are used in such severe environments. Furthermore, deposits which were not intended for exploitation before have since come to be exploited. These include , for example, a deposit which is corrosive to steel and contains significant amounts of oxidizing impurities. Accordingly, threaded connections capable of satisfying requirements described above in such severe environments are called for, and the performance of the threaded connections has since been improved at a high pace.

[0004] Fig. 1 is a schematic illustrating a coupling type threaded connection provided with a seal portion and a torque shoulder portion. In the figure, an example of the threaded connection for the oil country tubular goods provided with the torque shoulder portion internally is shown. Fig. 1 (a) is a view showing pipes body 10 and a coupling 20 making up the threaded connection. Fig. 1 (b) is a partially enlarged view of a threaded portion, the seal portion, and the torque shoulder portion, as shown in Fig. 1 (a).

[0005] The torque shoulder portion is provided to prevent build-up of such a high contact pressure as to cause plastic deformation of the seal portion. The threaded connection is constituted such that a make-up torque can be controlled to an appropriate value by contacting a torque shoulder forming face 24 on the coupling side against a torque shoulder forming face 14 on the pipe side. Such a type of threaded connection provided with the torque forming faces 14 and 24, close to the inner surface of a pipe, as described above or the torque shoulder portion is called an internal shoulder.

[0006] Fig. 2 is a schematic illustrating an example of a threaded connection for the oil country tubular goods, provided with both the internal torque shoulder and an external torque shoulder. Fig. 2 (a) is a sectional view showing the pipe body and the coupling after the threaded connection is made up. Fig. 2 (b) is a partially enlarged view of the torque shoulder forming portion on the outer side of the threaded connection shown in Fig. 2 (a), showing inclinations of the torque shoulder forming faces on a pin and a box, respectively, for forming an external torque shoulder. The external torque shoulder is formed when the torque shoulder forming face 54 disposed at the root of a male thread section on the outer surface of the pipe is contacted against the torque shoulder forming face 64 disposed at the tip on the box.

[0007] Among the coupling type threaded connections, provided with a seal portion and a torque shoulder portion, an internal shoulder type threaded connection is described hereafter.

[0008] In Fig. 1 (b), male threads 12 provided on the surface of a tip portion of the pipe 10 and female threads 22 provided on the inner surface of the coupling 20 are mated with each other. A seal forming face 13, which is a rotatable face, is provided at the tip of a pin 11. The seal forming face 13 has a curved surface, tapered in the axial direction of the pipe or substantially resembling a circular arc of a large curvature radius. A box 21 of the coupling 20 is provided with a seal forming face 23, which is a rotatable face, at the root of a threaded portion thereof, and the seal forming face 23 has a curved surface tapered in the axial direction of the pipe. The seal portion is formed when the seal forming face

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13 of the pipe body is brought into contact with the seal forming face 23 of the coupling for making up the threaded connection.

[0009] Owing to metal-to-metal seal accomplished at the seal portion, gas tightness of the threaded connection is maintained against internal pressure caused by a fluid flowing inside the oil country tubular goods, and external pressure caused by fluids flowing outside same.

[0010] The torque shoulder portion is formed when a torque shoulder forming face 14 on the pin, disposed beyond the seal forming face 13 of the pipe body, closer to the tip, is contacted against a torque shoulder forming face 24 disposed on the inner side of the seal forming face 23 on the box of the coupling.

[0011] In the case of the threaded connection for the oil country tubular goods, corrosion resistance against crevice corrosion causing leakage of fluids or rupture of the threaded portion is called for in addition to seal tightness against fluids under high pressure as described in the foregoing. Crevice corrosion occurs when a corrosive fluid intrudes into minuscule crevices formed between pin members and box members, stagnates therein, and consequently are enriched. When the threaded connection is loaded with high stress and placed under the conditions described above, crevice corrosion such as general corrosion, stress corrosion cracking, or the like occurs inside the threaded connection. No crevice corrosion occurs, however, in case that there exists no minuscule crevice on either the inner surface or the outer surface side of the pipe in which a corrosive fluid exists.

[0012] Threaded connections provided with the seal portion and the torque shoulder in shapes described hereafter have been proposed as improvements on the threaded connection shown in Fig. 1 for enhancing crevice corrosion resistance. These threaded connections are broadly classified into four classes, that is, one provided with contact faces coated with a softer material, one using a packing, one having enhanced seal tightness of the seal portion, taking advantage of elastic deformation of the pin at the tip thereof, and one having enhanced gas tightness of the seal portion by means of a device for improved contact of the torque shoulder forming faces.

[0013] The threaded connections with contact faces coated with a softer material as disclosed in Japanese Patent Laid-open (Kokai) No. 56-109975, No. 1-199088, and the like have a drawback in that gas tightness thereof is impaired in the course of repeated making up and disconnection of the threaded connection due to exfoliation and wear-out of the coating material with the result that these threaded connections are unable to withstand repeated uses as frequent as scores of times.

[0014] In the case of the threaded connection using a packing as disclosed in Japanese Patent Laid-open (Kokai) No. 58-142475, same has a drawback of extra time and expenses required in production, fitting, and the like of the packing. Further, in the case of this type of threaded connection without a function for controlling a torque for make-up at an appropriate value, which is naturally required of the torque shoulder, strict torque control is required at a work site when making up connection. Consequently, operativity of the threaded connection is poor. In addition, same poses a problem of susceptibility to crevice corrosion because crevices may be formed between the packing and the torque shoulder depending on manufacturing tolerance.

[0015] In the case of the threaded connection having enhanced gas tightness, taking advantage of elastic deformation occurring at the tip of the pin (for example, as disclosed in Japanese Patent Laid-open (Kokai) No. 60-26878), substantial bending deformation needs to be caused to occur to a lip portion at the tip of the pin in order to obtain a seal face having a sufficient contact pressure in a seal forming portion. For this purpose, the lip portion at the tip of the pin needs to be pressed by a large force to the torque shoulder forming face on the box. As a result, excessive plastic deformation occurs on the torque shoulder forming face, particularly, at a portion 26 in close vicinity of the pipe surface.

[0016] Fig. 3 shows an example of the threaded connection having enhanced gas tightness by devising improved contact on the torque shoulder forming faces (refer to Japanese Patent Laid-open (Kokai) No. 52-11765). With the threaded connection, swelling of the pin in close vicinity of the tip thereof is caused to occur in the direction of the outside diameter of the pipe through contact between the torque shoulder forming face on the pin and same on the box so that seal tightness of the seal portion is enhanced. More specifically, a seal forming face 13 on the pin is rendered a rotatable face having a surface in the shape of a circular arc of a large curvature radius in sectional view, and a seal forming face on the box is rendered a circular cone (a rotatable face having a surface of a tapered straight line in sectional view). Furthermore, a torque shoulder forming face 14 on the pin and a torque shoulder forming face 24 on the box are rendered to have a convex surface and a concave surface, respectively, each having a differentiated curvature radius, R2 and R3. In this connection, the curvature radius R2 of the torque shoulder forming face on the pin is rendered smaller than the curvature radius R3 of the torque shoulder forming face on the box, and in addition, the center of the curvature with the radius R2 is positioned on the inner side towards the pipe than that of the curvature with the radius R3. With adoption of such a constitution as described above, bending of the lip portion 15 on the pin towards the inner surface of the pipe is reduced, and even when nonuniform contact pressure is developed in the peripheral direction of the seal portion due to manufacturing tolerances, uniform contact pressure is automatically redistributed. It is claimed for this reason that satisfactory seal tightness is provided at two locations, namely, the torque shoulder portion and the seal portion.

[0017] In the threaded connection described, contact at the torque shoulder portion is linear contact along a circular face at portions 16 and 26 in close vicinity of the inner surface of the pipe. Consequently, contact pressure at such contact portions becomes very high, creating a problem of gas tightness being impaired due to excessive deformation, similarly to the case of the threaded connection as disclosed in Japanese Patent Laid-open (Kokai) No. 60-26878 described above.

[0018] The invention has been developed to solve the problems described above, encountered by the conventional threaded connections. Thus, it is an object of the invention to provide a threaded connection capable of preventing initiation of crevice corrosion without bringing about an increase in manufacturing cost thereof and deterioration in operativity, and also of demonstrating a high degree of gas tightness even after repeated uses as frequent as scores of times.

DISCLOSURE OF THE INVENTION

[0019] The threaded connection for the oil country tubular goods, according to the invention, has the advantage of less intense crevice corrosion, and an ability to maintain gas tightness sufficient for practical application even after repeated uses as frequent as scores of times.

[0020] In summary, the threaded connection according to the invention includes (A), (B), and (C) described hereafter:

(A) A threaded connection for the oil country tubular goods, wherein a seal portion is formed when a seal forming face 13 provided on the outer surface of the end portion of a pin 11 of a pipe body, with male threads formed thereon, is fitted to a seal forming face 23 provided on the inner surface of a box 21 of a coupling or a pipe body, with female threads formed thereon, while an internal torque shoulder is formed when a torque shoulder forming face 14 provided at the tip of the pin is butted against a torque shoulder forming face 24 provided on the box, opposite to the torque shoulder forming face 14, and satisfying the following conditions (1) and (2):

(1) The seal forming faces on the pin and box, respectively, are rotatable faces, and the pitch diameter of the seal forming face 13 on the pin prior to making up the threaded connection is larger than a pitch diameter of the seal forming face 23 on the box prior to making up the threaded connection; and

(2) An inclination (θ 1) of the torque shoulder forming face on the pin prior to making up the threaded connection is smaller than an inclination (θ 2) of the torque shoulder forming face on the box prior to making up the threaded connection, the difference ($\Delta\theta$) between the both inclinations being in the range of 0.5 to 4.0°.

(B) A threaded connection for the oil country tubular goods, wherein with reference to Fig. 2 (b), a seal portion is formed when a seal forming face 53 provided on the outer surface of a pin 11 of a pipe body, close to the base thereof, with male threads formed thereon, is fitted to a seal forming face 63 provided on the inner surface of a box 21 of a coupling or a pipe body close to these tips thereof, with female threads formed thereon, while an external shoulder is formed when a torque shoulder forming face 64 provided at the tip of the box is contacted against a torque shoulder forming face 54 provided on the outer surface of the pin of the pipe body, close to the base thereof, opposite to the torque shoulder forming face 64, and satisfying the conditions (1), and (2) described above.

(C) A threaded connection for the oil country tubular goods, wherein all of the seal portion, the internal shoulder and the external shoulder as described in (A) and (B) above are provided.

The threaded connections for the oil country tubular goods, described in (A), (B), and (C) above, are preferably to satisfy the following condition (3) as well:

(3) The difference ($\Delta\theta$) between the inclination (θ 1) of the torque shoulder forming face on the pin prior to making up and the inclination (θ 2) of the torque shoulder forming face on the box prior to making up is proportional to D/t, and δ r, respectively. In case the value of D/t or δ r is large, the difference in inclination ($\Delta\theta$) between the torque shoulder forming face on the pin and the same on the box is increased, and in case the value of D/t or δ r is small, the difference in inclination ($\Delta\theta$) between the torque shoulder forming face on the pin and the same on the box is reduced

wherein

D : outside diameter of the tip of the pin or box
t : wall thickness of the tip of the pin or box

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δ r = (interference on the seal forming face in the direction of the diameter) - (interference on the thread portion in the direction of the diameter).

[0021] The threaded connection for the oil country tubular goods, described in the foregoing, is unsusceptible to crevice corrosion, maintaining gas tightness sufficient for repeated uses as frequently as scores of times. Furthermore, manufacturing cost thereof is held down owing to its simple structure, and it is easy to handle.

[0022] The pitch diameter of the seal forming face on the pin and box, respectively, is defined as follows. Said diameter is a diameter of the respective seal forming faces at predetermined base positions prior to make-up, that is, an average distance between the central axis of the pipe or the box and the base positions for the respective seal forming faces. The base position refers to a region corresponding to parts where the seal forming face on the pin and same on the box come in contact with each other when the threaded connection is completed, and means the position prior to fitting.

BRIEF DESCRIPTION OF THE DRAWINGS

[0023]

Fig. 1 is a view of a conventional coupling type threaded connection provided with a seal portion and a torque shoulder portion, showing an internal shoulder type by way of example. Fig. 1 (a) is a view showing a pipe body and a coupling, making up a threaded connection. Fig. 1 (b) is a partially enlarged view of a threaded portion, a seal forming portion , and a torque shoulder forming portion in Fig. 1 (a).

Fig. 2 is a view showing a coupling type threaded connection provided with both the internal shoulder and an external shoulder. Fig. 2 (a) is view showing the pin and the box, making up the threaded connection, and Fig. 2 (b) a partially enlarged view of the external shoulder shown in Fig. 1 (a).

Fig. 3 is a view illustrating a conventional threaded connection wherein seal tightness of a seal portion is enhanced as a result of swelling in the direction of the outside diameter of the pipe occurring to a portion of a pin, close to the tip thereof, when a torque shoulder forming face (convex) on the pin and a torque shoulder forming face (concave) on a box come into contact with each other with the center of curvature of respective faces being disposed at a differentiated point.

Fig. 4 is a view illustrating the torque shoulder forming face on the pin bending by θ a relative to the torque shoulder forming face on the box due to bending deformation in the direction of shrinkage in diameter occurring to a seal forming face of a larger pitch diameter.

Fig. 5 is a view showing a threaded connection according to the invention, provided with a seal portion and a torque shoulder. Fig. 5 (a) is a view showing a seal forming face and a torque shoulder forming face on a pin and a box, respectively, prior to making up the connection. Fig. 5 (b) is a view showing distribution of contact pressure developed at the seal portion and the torque shoulder after the connection is made up.

BEST MODE FOR CARRYING OUT THE INVENTION

[0024] As described in the foregoing, the threaded connection for the oil country tubular goods, according to the invention, is characterized in satisfying conditions (1) and (2) given hereinafter. The threaded connection according to the invention has performance characteristics of preventing initiation of crevice corrosion and demonstrating high degree of gas tightness even after repeated uses as frequent as scores of times provided that said conditions are satisfied. Furthermore, the threaded connection does not entail any increase in manufacturing cost, and is easy to handle.

(1) The pitch diameter of the seal forming face 13 on the pin prior to making up the threaded connection is larger than a pitch diameter of the seal forming face 23 on the box prior to making up the threaded connection; and

(2) An inclination (θ 1) of the torque shoulder forming face on the pin prior to making up the threaded connection is smaller than an inclination (θ 2) of the torque shoulder forming face on the box prior to making up the threaded connection, the difference ($\Delta\theta$) between the both inclinations being in the range of 0.5 to 4.0°.

[0025] The reason why the threaded connection according to the invention is unsusceptible to crevice corrosion and can withstand frequent repeated uses is described hereafter.

[0026] Fig. 4 is a view illustrating the torque shoulder forming face on the pin bending by θ a relative to the torque shoulder forming face on the box due to shrinkage deformation occurring to the seal forming face on the pin. As shown in the figure, in the threaded connection according to the invention, the seal forming face and the torque shoulder forming face are disposed in close vicinity of the tip of the pin, and consequently, a lip portion 15 at the tip of the pin undergoes bending deformation in the direction of shrinkage in diameter due to contact pressure building up at the seal forming faces 13 and 23 when making up the threaded connection. The bending deformation causes the torque shoulder forming face 14 on the pin to bend by θ a relative to the torque shoulder forming face 24 on the box towards the inner side of the pipe. This brings the torque shoulder forming face on the pin in intimate contact with the torque shoulder forming face on the box, building up a large contact pressure between the both faces.

[0027] In the conventional type threaded connection wherein the torque shoulder forming face on the pin and box, respectively, bends by an identical angle, contact pressure developed at portions denoted by 16 and 26 of the torque shoulder region, on the inner side thereof towards the pipe, is low due to bending deformation occurring in the form described above. Therefore, when a large tensile stress is applied to the pipe body including the threaded connection already made up, crevices are apt to be formed on the inner face side of the torque shoulder because contact pressure scarcely exists there.

[0028] On the other hand, in the case of the threaded connection according to the invention, the difference Δ θ in inclination between the torque shoulder forming face on the pin and same on the box is set beforehand. Furthermore, as the difference Δ θ is restricted within the range of 0.5 to 4.0° , the both faces come into contact with each other after bending deformation taking place due to the connection being made up so as to be flush with each other or overlapped by each other on the inner face side of the torque shoulder. Consequently, large contact pressure is developed, and distribution of contact pressure in most preferable pattern is obtained.

[0029] With reference to Fig. 5, description in further detail is given hereafter. Fig. 5 is a view showing a structure of the threaded connection, in close vicinity of the internal shoulder wherein an inclination (θ 1) of the torque shoulder forming face on the pin is rendered smaller than an inclination (θ 2) of the torque shoulder forming face on the box. Fig. 5 (a) is a view showing the seal forming face and the torque shoulder forming face on the pin and the box, respectively, of the threaded connection prior to being made up. Fig. 5 (b) is a view showing distribution of contact pressure (results of computation) built up on the seal portion and the torque shoulder after the threaded connection is made up. It is confirmed from Fig. 5 that evenly distributed contact pressure is obtained by preestimating the bending deformation of the torque shoulder forming face on the pin, as described above, and reducing the inclination of the torque shoulder forming face on the pin to that extent.

[0030] The structure of the threaded connection according to the invention is applicable to a threaded connection provided with an external shoulder. As shown in Figs. 2 (a) and 2 (b), even distribution of contact pressure is obtained at the external shoulder by differentiating the inclination of the torque shoulder forming face on the pin from same on the box.

[0031] The threaded connection according to the invention is hereafter described in greater detail. [Δ θ : difference in inclination between the torque shoulder forming face on the pin and same the box]

[0032] The seal forming face on the pin is a rotatable face with a surface shaped in a tapered curve or a circular arc of a large curvature radius, and the seal forming face on the box is a rotatable face with a surface shaped in a tapered curve.

[0033] The pitch diameter of the seal forming face on the pin is rendered larger than that on the box. As described in the foregoing, the pitch diameter refers to the diameter of the seal forming face on the pin in the base position and same on the box in the base position, respectively.

[0034] In case that the pitch diameter of the pin to be fitted into the box, is larger than that of the box, and the pin is forced to be fitted into the box, shrinkage deformation occurs to the tip portion of the pin, having a low rigidity. The shrinkage deformation occurring to the tip portion causes the tip portion to undergo bending deformation towards the inner side surface of the pipe. Consequently, the torque shoulder forming face on the pin dips to an extent of such bending deformation (refer to Fig. 4).

[0035] In the threaded connection shown in Fig. 5(a), the inclination θ 1 of the torque shoulder forming face 14 on the pin is smaller than the inclination θ 2 of the torque shoulder forming face 24 on the box, by Δ θ (= θ 2 - θ 1). The difference Δ θ in inclination may preferably be set to be equivalent to or slightly larger than a change θ a in inclination actually taking place as a result of the bending deformation described in the foregoing. As shown in Fig. 4, the change θ a in inclination due to the bending deformation actually taking place is substantially equivalent to a bending angle θ ' of the torque shoulder forming face 14 on the pin when bending deformation is caused to occur to the lip portion 15 located close to the tip of the pin due to contact pressure developed at the seal forming faces. Herein, the bending angle refers to an angle of bending occurring to the lip portion 15, located close to the tip of the pin, when subjected to a lateral load or bending moment.

[0036] Contact pressure in the torque shoulder region is distributed as follows by differentiating the inclination of the torque shoulder forming face on the pin from same on the box:

(a) substantially even distribution of contact pressure across the torque shoulder forming faces 14 and 24 on the pin and the box, respectively; or

(b) distribution of contact pressure such that contact pressure remains high at portions denoted by 16 and 26, on xx US

the inner side surface of the pipe, and gradually declines closer to the seal forming faces 17 and 27.

[0037] Accordingly, stress concentration does not occur in such a way as to cause excessive plastic deformation at 16 and 26 in close vicinity of the inner side surface of the pipe. Further, contact pressure sufficient to prevent crevices from being formed in said regions is secured. Hence, the threaded connection for the oil country tubular goods, capable of withstanding repeated uses as frequent scores of times, and nearly preventing crevice corrosion is provided.

[0038] Since the torque shoulder forming face on the pin and same on the box are kept in contact with each other at substantially even contact pressure, localized plastic deformation due to like stress concentration does not occur. In case that a tensile stress is applied to the pipe body including the threaded connection made up in such a condition described above, contact pressure throughout the torque shoulder is uniformly reduced, and consequently, a critical value for tensile stress causing opening in the torque shoulder portion becomes by far higher than that for a threaded connection provided with a torque shoulder formed in a conventional shape. Thus, the torque shoulder is formed, which is unsusceptible to opening even when a tensile stress equivalent to yield strength of the pipes body is applied provided that contact pressure is sufficiently high when the threaded connection is made up. Subsequently, the threaded connection is provided with excellent crevice corrosion resistance.

[0039] In the case of a threaded connection wherein the torque shoulder forming face 14 on the pin has an inclination identical to that for the torque shoulder forming face 24 on the box, even distribution of contact pressure at the torque shoulder faces is not achieved. With such a threaded connection as above, contact pressure developed on the seal forming faces becomes high while contact pressure on the inner side surfaces of the pipes becomes low. Consequently, in case of tensile stress caused by the pipes' own weight, and the like coming into play after the threaded connection is made up, the contact pressure on the inner side surface of the pipes is reduced very low, causing opening in such a region of the threaded connection in an extreme case. When the threaded connection is used under such a condition for many hours, there occurs ingress of a corrosive fluid under a high internal pressure into the interior of the threaded connection through crevices formed between 16 and 26 on the inner side surface of the pipes inside the threaded connection. As a result, corrosion occurs to the seal portion and also to a threaded portion subjected to a high stress, resulting in leakage of the fluid or rupture of the threaded connection in the worst case. With the threaded connection according to the invention, such troubles can be prevented until the number of times when same is put to use reaches at least some scores.

[0040] Even in case that a compressive stress in place of a tensile stress is applied, contact pressure developed across the torque shoulder surfaces is substantially uniform. Accordingly, the torque shoulder is unsusceptible to localized plastic deformation due to stress concentration occurring therein. Thus, there occurs no excessive plastic deformation to such an extent that repeated use thereof is rendered impractical. Herein, the extent that repeated use thereof is rendered impractical refers to a condition wherein the torque shoulder is no longer able to function as torque controller and substitute seal due to permanent deformation of the torque shoulder on the box, in particular, and the threaded connection can not be properly made up, failing to demonstrate intended performance thereof.

[0041] An exceedingly large $\Delta\theta$ is not preferable. In case of $\Delta\theta$ being exceedingly large, stress concentration occurs to the inner side of the torque shoulder, resulting in occurrence of localized plastic deformation. There exists an appropriate range for $\Delta\theta$ as described hereinafter.

[Appropriate Range for $\Delta\theta$]

[0042] The bending angle θ' (refer to Fig. 6) of bending deformation of the lip portion 15 on the pin in the direction of shrinkage thereof, occurring when the seal forming face 13 on the pin is fitted onto the seal forming face 23 on the box, can be found by a computation formula based on the classic shell theory. Further, a correlation between the bending angle θ' found by the computation formula and the change in inclination θa actually occurring has already been established from the results of tests. Hence, the change in inclination θa of the torque shoulder forming face on the pin which actually occurs can be found by finding the bending angle θ' of the tip of the pin. The difference in inclination $\Delta\theta$ between the torque shoulder forming face on the pin and same the box needs to be set to be equivalent to or slightly larger than the change in inclination θa.

[0043] With the threaded connection for the oil country tubular goods in industrial application, the difference in inclination $\Delta\theta$ is to fall within a range of 0.5 to 4.0°. In the event that same is less than 0.5°, sufficient sealing effect can not be obtained. On the other hand, in the event of same exceeding 4°, localized plastic deformation occurs due to stress concentration occurring on the inner side surface of the torque shoulder as described in the foregoing. Thereupon, not only the number of times when the threaded connection can be put to repeated use decreases, but also same becomes more susceptible to formation of crevices.

[0044] The threaded connection having the angles described above can be worked on with precision sufficient for practical use, but without need for special means of precision control, by use of a computer-controlled NC lathe.

[Selection of Most Optimum Value for Δ θ]

[0045] As described in the foregoing, the difference in inclination Δ θ remaining in the range of 0.5 to 4.0 ° suffices for providing performance sufficient for practical application. However, depending on conditions of the pipes and the threaded connection, more optimum values may preferably be selected.

[0046] In the case of the internal shoulder, Δ θ may preferably be made proportional to (the outside diameter of the pin at the tip : D)/(the wall thickness of the pin at the tip : t) (hereinafter referred to as D/t). In the case of D/t being a large value, the internal shoulder is more susceptible to bending because of its low rigidity against bending. Therefore, the difference in inclination Δ θ may preferably be made larger. Thus, depending on magnitude of D/t, the most optimum value for Δ θ may preferably be selected.

[0047] Δ θ may preferably be made substantially proportional to

δ r { = (interference of the seal forming portion) - (interference of the thread forming portion)}. The interference being a difference in pitch diameter between the pin and the box, the larger the δ r is, the larger a shrinkage occurring becomes. As described above, the interference of the seal forming portion has effect on both the shrinkage of the pin at the tip and the bending of the torque shoulder forming face. However, in the event that the interference of the seal forming portion is equivalent to that of the thread forming portion, only uniform shrinkage occurs, causing no bending. The interference of the thread forming portion in close vicinity of the pipe body causes only uniform shrinkage deformation. In such a case, both the interference of the seal forming portion and same of the thread forming portion cause only shrinkage deformation, and maintain equilibrium.

[0048] In other words, a difference in the interference δ r, obtained by deducting the interference of the thread forming portion, not contributing to bending, from the interference of the seal forming portion, causes bending deformation to occur. The Δ θ which is set to be substantially equivalent to the bending angle of bending occurring in the direction of the inner side surface of the pipes is substantially proportional to δ r. Accordingly, the most optimum value for the difference in inclination Δ θ between the torque shoulder forming face on the pin and same on the box may preferably be selected so as to be substantially proportional to δ r.

[0049] Conditions described in the foregoing, under which the internal shoulder operates, are also applicable to the external shoulder.

[0050] As shown in Fig. 2 (b), in case that the pitch diameter of the seal forming face 63 on the box is rendered smaller than that of the seal forming face 53 on the pin, the torque shoulder forming face 64 on the box is bent outwardly when the threaded connection is made up, changing inclination thereof. In the case of the external shoulder being under the influence of bending stiffness of the coupling unlike the case of the internal shoulder, the outside diameter and the wall thickness of the box at the tip are denoted by D and t, respectively. The tip of the box undergoes bending deformation due to bending stiffness thereof being lower than that at the root of the pin because the tip of the box is thinner in wall thickness and a free end. An inclination θ 2 of the torque shoulder forming face on the box may preferably be set to be smaller than an inclination θ 1 of same on the pin by said change θ a in inclination or by an angle slightly larger than θ a. Hereupon, substantially uniform contact pressure or contact pressure slightly higher at 56 and 66 on the outer surface and slightly lower at 57 and 67 on the respective seal forming faces is provided.

[0051] As a result, in the event that a tensile stress is applied to the threaded connection provided with the external shoulder, opening on the external side thereof is prevented. All other effects of the internal shoulder, as described above, are applicable in whole to the external shoulder as well.

[0052] All the conditions specified in the invention are also applicable straight to the threaded connection provided with both the internal shoulder and the external shoulder.

[0053] The coupling type threaded connection is described in the foregoing by way of example, but it is needless to say that the invention is applicable to the integral type threaded connection as well. In such a case, the pipe body may be provided with a pin at one end, comprising male threads, a seal forming face, and a shoulder forming face, and a box at the other end, comprising female threads, a seal forming face, and a shoulder forming face, after surfacing. Thus, the construction exactly same as that of the threaded connection according to the invention can be applied thereto.

Embodiments

[0054] The effects of the invention are described hereafter with reference to embodiments of the invention.

[0055] Tables 1, 2, 3, and 4 are lists showing the inclinations of the torque shoulders, and particulars of the threaded connections according to the embodiments of the invention, and threaded connections for comparative examples, on which tests were conducted. With reference to the threaded connections marked A through J shown in Tables 1 and 2, D/t and δ r are in common at 17.78 and 0.4 mm, respectively. With reference to the threaded connections marked by K and L shown in Tables 3 and 4, D/t is 13.78, and δ r is 0.199 mm for the former, and D/t 21.72, δ r 0.528 for the latter, representing values either higher or lower than same for the threaded connections marked by A through J. In the case of the threaded connection K having both D/t and δ r lower in value than same for the threaded connections A through

J, the difference $\Delta\theta$ in inclination between the respective torque shoulder forming faces thereof is lower in value substantially in proportion to the D/t and δr while in the case of the threaded connection L having D/t and δr higher in value, the difference $\Delta\theta$ in inclination is higher in value proportionally.

[0056] The threaded connections according to the embodiments of the invention, in Tables 1 through 4 are the threaded connections shown in Figs. 5 and 2, including three types consisting of the internal shoulder type, the external shoulder type, and the internal shoulder combined with the external shoulder type. The comparative examples shown in Table 2 include a threaded connection provided with the torque shoulder forming face on the pin and the box, each having an identical inclination, a threaded connection provided with the seal surface, and the like, coated with a softer metal, a threaded connection having the thin lip portion provided with smooth grooves (as disclosed in Japanese Patent Laid-open No. 60-26878), and a threaded connection (refer to Fig. 3) provided with the torque shoulder forming face on the pin and the box, having a convex surface and a concave surface, respectively, each having a differentiated curvature radius.

Table 1

Item	In common with threaded connections marked A, B, C, D, E, F, G, H, I, and J
outside diameter of a pipe	177.8 mm
wall thickness of a pipe	10.36 mm
outside diameter of a coupling	194.49 mm
material for a coupling and pipe	L 80 type carbon steel (yield strength: 56.2 kgf/mm^2)
thread shape	trapezoidal thread
thread pitch	5.08 mm
taper of a threaded portion	1/16
thread ridge height	1.575 mm
interference of a threaded portion in the direction of dia.)	0.2 mm
taper of a seal portion	1/10
length of a seal portion	6.5 mm
interference of a seal portion (in the direction of dia.)	0.6 mm
80% of tensile load value	245.2 tons
80% of internal pressure value	5.24 kgf/mm^2

Table 2

	markings for threaded connections	inclination of torque shoulder on the pin (θ 1)	inclination of torque shoulder on the box (θ 2)	remarks
embodiments of the invention	A	15.0°	16.0°	internal shoulder
	B	15.0°	16.5°	internal shoulder
	C	16.0°	15.0°	external shoulder
	D	16.5°	15.0°	external shoulder
	E	15.0° 16.0°	16.0° 15.0°	combination of internal shoulder with external shoulder *1
comparative examples	F	16.0°	16.0°	internal shoulder
	G	15.0°	16.0°	external shoulder
	H	15.0°	16.0°	coated with Cu *2
	I	15.0°	16.0°	thin lip portion *3
	J	6 mm	8 mm	torque shoulder faces of differentiated curvature radius *4

*1 a threaded connection provided with both the internal shoulder and the external shoulder (the same type as one shown in Fig 2)

*2: a threaded connection having a seal forming face and a torque shoulder forming face, on the box, coated with Cu

*3: a threaded connection provided with a thin-walled lip portion formed by cutting smooth grooves on the inner surface of both the lip portion at the end of the pin and a torque shoulder on the box such that seal tightness is enhanced owing to elastic deformation (of the same type as one disclosed in Japanese Patent Laid-open No. 60-26878)

*4 a threaded connection provided with a torque shoulder forming face in a concave shape disposed on the box, opposite to a torque shoulder forming face in a convex shape, disposed at the tip of the pin (the same type as one shown in Fig. 3)

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Table 3

	markings for threaded connections	
	K	L
outside diameter of a pipe	88.9 mm	273.05 mm
wall thickness of a pipe	6.45 mm	12.57 mm
outside diameter of coupling	96.94 mm	293.54 mm
material for the pipe and coupling	L 80 carbon steel (yield strength: 56.2 kgf/mm^2	
thread shape	trapezoidal thread	
thread pitch	4.233 mm	6.35 mm
taper of threaded portion	1/18	
thread ridge height	1.016 mm	1.982 mm
interference of a threaded portion (in the direction of dia.)	0.221 mm	0.272 mm
taper of a seal portion	1/10	
length of a seal portion	6.5 mm	
interference of a seal portion (in the direction of dia.)	0.42 mm	0.8 mm
80% of tensile load value	75.2 tons	462.8 tons
80% of internal pressure value	6.53 kgf/mm^2	4.14 kgf/mm^2

Table 4

	embodiments of the invention	
markings for threaded connections	K	L
inclination of the torque shoulder forming face on the pin (θ 1)	15.0°	15.0°
inclination of the torque shoulder forming face on the box (θ 2)	15.5°	17.5°
remarks	internal shoulder	internal shoulder

[0057] The following tests were conducted on the threaded connections described above to evaluate performance of the respective threaded connections.

[Test on Repeated Making-up of Threaded Connections]

[0058] Lubrication grease was applied to the threaded portions and non-threaded portions, and then, making-up and disconnection of the respective threaded connections were repeated ten times, and the changes, having occurred on the torque shoulders, and the changes on the seal portions were both examined.

[Test on Crevice Occurrence under Tensile Stress]

[0059] After the tests on repeated making-up of threaded connections described above, the threaded connections were made up, and with pipes under load of a tensile force (245.2 tons) applied in the axial direction thereof so as to cause a stress equivalent to 80% of yield strength of the pipe, the inner surface and outer surface of the respective torque shoulders were checked for presence of crevices.

[Corrosion Test under Composite Load]

[0060] A gas mixture composed of hydrogen sulfide and carbon dioxide was sealed into the pipes including the respective threaded connections with the tensile force applied as at the crevice occurrence test described above. The pressure of the gas mixture was 5.24 kgf/mm^2, causing the pipes a stress equivalent to 80% of the yield strength of the pipe. After holding the respective threaded connections in such a condition described for 500 hours, the threaded connections were disconnected to check up on corrosion sites.

[0061] Tables 5 and 6 are lists showing results of the three tests described. Tables 5 and 6 show that in the cases of the threaded connections for comparative examples, the result of at least one test among the three tests was found unsatisfactory. On the other hand, in the cases of the threaded connections according to the invention, the result of every test was found satisfactory. Table 6 shows results of tests conducted on the threaded connections marked K and L wherein the difference in inclination $\Delta\theta$ was varied in proportion to D/t and δ r. It was confirmed from the results that it was effective to vary $\Delta\theta$ in proportion to D/t and δ r.

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Table 5

	markings for threaded connections	results of test on repeated making up of connections	crevices on the shoulder under axial tensile stress	results of corrosion test under composite stress load	
				leakage	corrosion sites
embodiments of the invention	A	no change	none	none	none
	B	no change	none	none	none
	C	no change	none	*1	*1
	D	no change	none	*1	*1
	E	no change	none	none	none
comparative examples	F	no change	yes	none	at the seal portion
	G	no change	yes	*1	*1
	H	coating worn out	yes	yes (4.62) *2	*3
	I	the inner side of the shoulder deformed	yes	yes (3.23) *3	*3
	J	galling on the shoulder face	none	none	on the box shoulder

*1 In the case of the external shoulder type, only a tensile test in the axial direction was conducted.
*2 figures in parentheses indicate pressure values (kgf/mm²) at which leakage occurred. Corrosion tests could not be conducted.
*3 In case of leakage occurring at internal pressure tests, the tests were suspended forthwith.

Table 6

	markings for threaded connections	results of test on repeated making-up	crevices on the shoulder under axial tensile stress	corrosion test results under composite stress load	
				leakage	corrosion sites
embodiments of the invention	K	no change	none	none	none
	L	no change	none	none	none

INDUSTRIAL APPLICABILITY

[0062] In the threaded connection for the oil country tubular goods, according to the invention, no crevice is formed either on the internal side of the torque shoulder in the case of the internal shoulder type, or on the external side of the torque shoulder in the case of the external shoulder type when used even under conditions subjected to load of a tensile stress due to the weight of connected pipes themselves, and the like. This is why same is unsusceptible to crevice corrosion caused by ingress of fluids flowing inside or outside of the pipes into the interior of the threaded connection through crevices in the torque shoulder region. Further, the threaded connection is able to withstand repeated uses as frequent as scores of times because excessive plastic deformation does not occur to the torque shoulder. In addition, the threaded connection can be manufactured by normal machining via a normal number of process steps, precluding a possibility of an increase in manufacturing cost.
[0063] Hence, the threaded connection according to the invention is quite suited for use at oil wells, and the like under increasingly severe exploitation conditions.

Claims

1. A threaded connection for the oil country tubular goods, wherein a seal portion is formed by fitting a seal forming face (13), provided on the outer surface of an end portion of a pin (11) of a pipe body, with male threads formed thereon, into a seal forming face (23), provided on the inner surface of a box (21) of a coupling or a pipe body, with female threads formed thereon, while an internal shoulder is formed by butting a torque shoulder forming face (14), provided at the tip of the pin against a torque shoulder forming face (24) provided on the box, opposite to the torque shoulder forming face (14), said threaded connection for the oil country tubular goods satisfying the following conditions (1) and (2):

   (1) The seal forming face on the pin and box, respectively, is a rotatable face, and the pitch diameter of the seal forming face (13) on the pin, prior to making up the threaded connection, is larger than a pitch diameter of the seal forming face (23) on the box, prior to making up the threaded connection;
   and
   (2) An inclination (θ 1) of the torque shoulder forming face on the pin, prior to making up the threaded connection, is smaller than an inclination (θ 2) of the torque shoulder forming face on the box, prior to making up the threaded connection, the difference (Δ θ) between the both inclinations being in the range of 0.5 to 4.0°.

2. A threaded connection for the oil country tubular goods, wherein a seal is formed by fitting a seal forming face (53), provided on the outer surface of a pin (11) of a pipe body, close to the base thereof, with male threads formed thereon, into a seal forming face (63), provided on a inner surface of a box (21) of a coupling or a pipe body close to these tips thereof, with female threads formed thereon, while an external shoulder is formed by butting a torque shoulder forming face (64), provided at the tip of the box against a torque shoulder forming face (54), provided on the outer surface of the pin of the pipe body, close to the base thereof, opposite to the torque shoulder forming face (64), said threaded connection for the oil country tubular goods satisfying the following conditions (1), and (2):

   (1) The seal forming face on the pin and box, respectively, is a rotatable face, and the pitch diameter of the seal forming face (53) on the pin, prior to making up the threaded connection, is larger than a pitch diameter of the seal forming face (63) on the box, prior to making up the threaded connection;
   and
   (2) An inclination (θ 1) of the torque shoulder forming face on the pin, prior to making up the threaded connection is larger than an inclination (θ 2) of the torque shoulder forming face on the box, prior to making up the threaded connection, a difference (Δ θ) between the both inclinations being in the range of 0.5 to 4.0°.

3. A threaded connection for the oil country tubular goods, wherein a seal is formed in respective regions by fitting a seal forming face (13), provided on the outer surface of an end portion of a pin (11) of a pipe body, with male threads formed thereon, into a seal forming face (23), provided on the inner surface of a box (21) of a coupling or a pipe body, with female threads formed thereon, and by fitting a seal forming face (53), provided on the outer surface of a pin (11) of a pipe body, close to thereof, provided with male thread, into a seal forming face (63), provided on the inner surface of a box (21) of a coupling or a pipe body, close to these tips thereof, with female threads formed thereon, while an internal shoulder and an external shoulder are formed respectively, by butting a torque shoulder forming face (14), provided at the tip of the pin against a torque shoulder forming face (24), provided on the box, opposite to the torque shoulder forming face (14), and by butting a torque shoulder forming face (64), provided at the tip of the box against a torque shoulder forming face (54), provided on the outer surface of the pin of

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the pipe body, close to the base thereof, opposite to the torque shoulder forming face (64), said threaded connection for the oil country tubular goods satisfying the following conditions (1) and (2):

(1) The seal forming faces on the pin and box, respectively, are rotatable faces, and the pitch diameter of the seal forming faces (13, 53), on the pin prior to making up the threaded connection, are larger than a pitch diameter of the seal forming faces (23, 63) on the box, prior to making up the threaded connection;
and
(2) An inclination (θ 1) of the torque shoulder forming faces on the pin, prior to making up the threaded connection, is smaller than an inclination (θ 2) of the torque shoulder forming faces on the box, prior to making up the threaded connection, a difference (Δ θ) in inclination between the both sides being in the range of 0.5 to 4.0°.

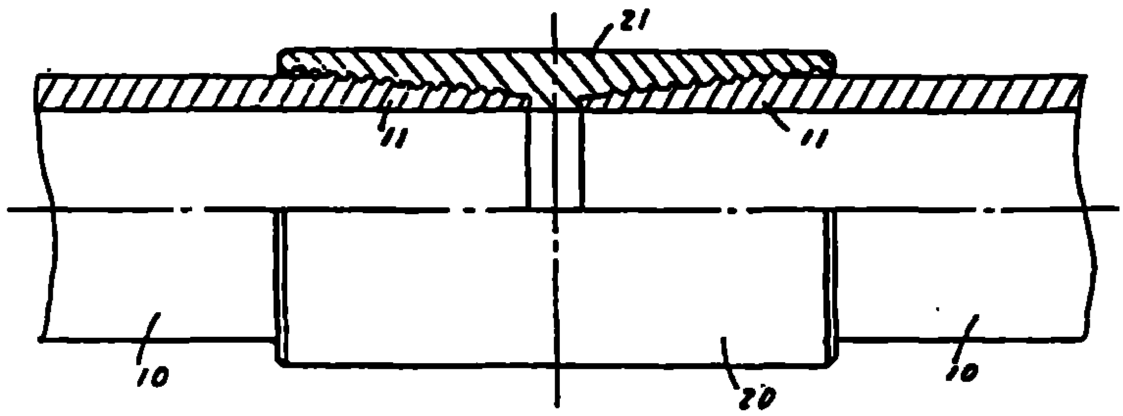
4. A threaded connection for the oil country tubular goods, according to Claim 1, 2, or 3, satisfying the following condition (3):

(3) The difference (Δ θ) between the inclination (θ 1) of the torque shoulder forming face on the pin, prior to making up and the inclination (θ 2) of same on the box, prior to making up, is proportional to D/t, and δ r, respectively. In case the value of D/t or δ r is large, the difference in inclination (Δ θ) between the torque shoulder forming face on the pin and the same on the box, is increased, and in case the value of D/t or δ r is small, the difference in inclination (Δ θ) between the torque shoulder forming face on the pin and the same on the box is reduced
wherein

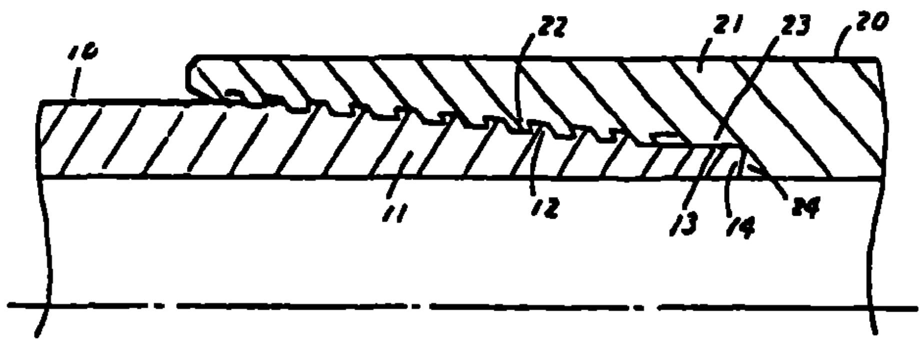
D: the outside diameter of the tip of the pin or box
t: wall thickness of the tip of the pin or box

δ r = (interference on the seal forming face in the direction of the diameter) - (interference on the thread portion in the direction of the diameter).

Fig.1



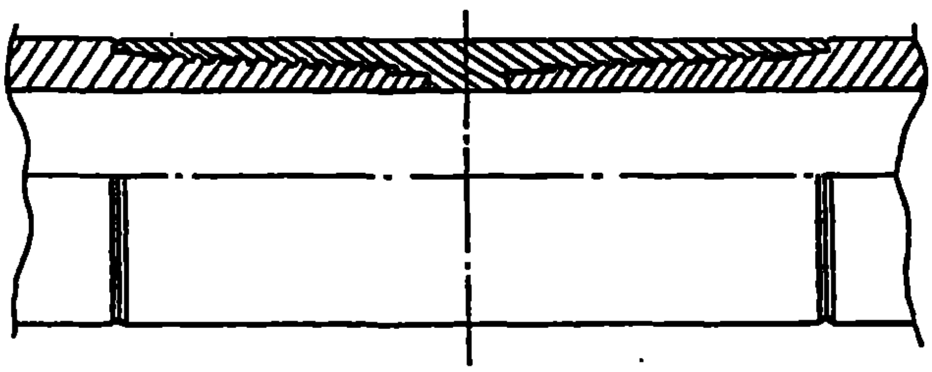
(a)



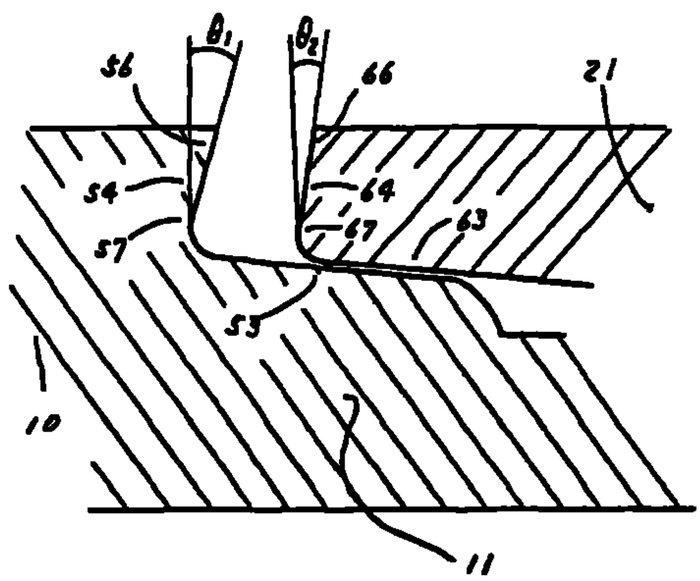
(b)

57 50

Fig.2



(a)



(b)

Fig.3

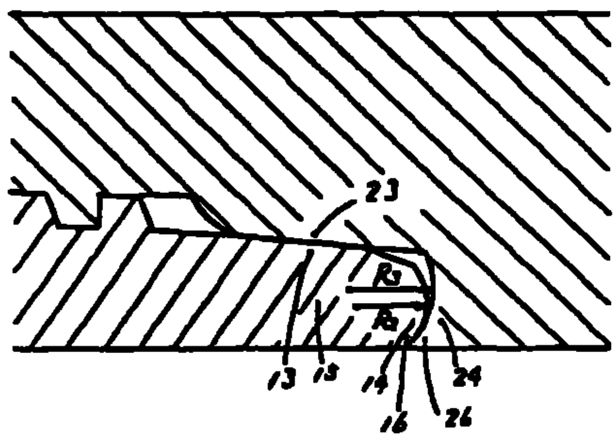


Fig.4

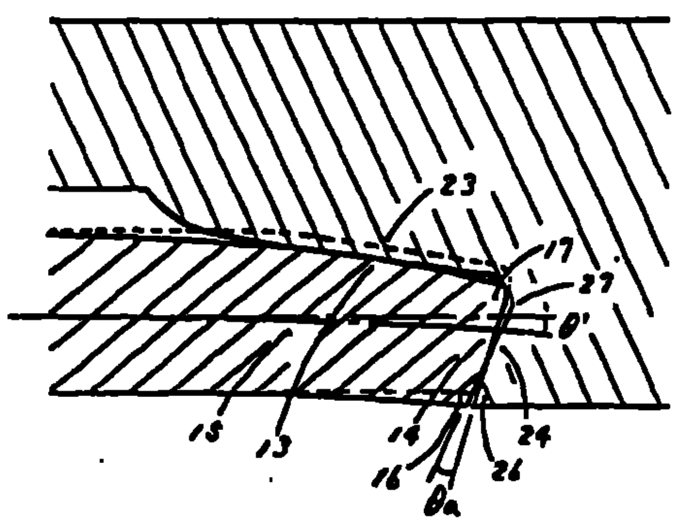


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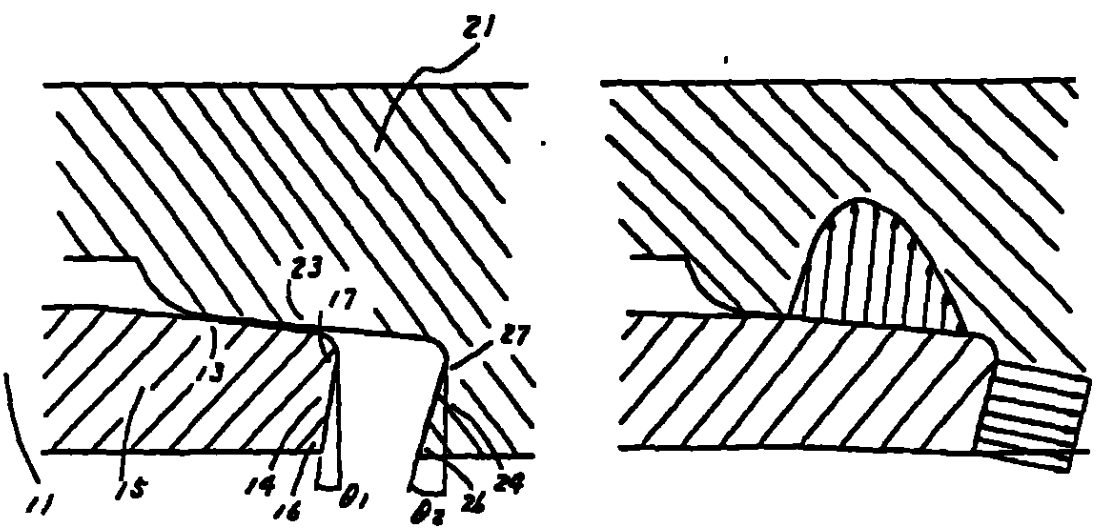
Fig.5



(a)

(b)

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP97/01855

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl[6] F16L15/04

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)

Int. Cl[6] F16L15/04

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

Jitsuyo Shinan Koho 1940 - 1997 Jitsuyo Shinan Toroku Koho 1996 - 1997
Kokai Jitsuyo Shinan Koho 1971 - 1997
Toroku Jitsuyo Shinan Koho 1994 - 1997

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP, 63-6293, A (NKK Corp.), January 12, 1988 (12. 01. 88), Page 1, lower left column, line 5 to lower right column, line 13 (Family: none)	1 - 4
Y	JP, 52-11767, B (Sumitomo Metal Industries, Ltd.), April 2, 1977 (02. 04. 77), Page 2, left column, line 11 to right column, line 23 (Family: none)	1 - 4

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier document but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
August 22, 1997 (22. 08. 97)	September 2, 1997 (02. 09. 97)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)

PATENT COOPERATION TREATY

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From the
INTERNATIONAL PRELIMINARY EXAMINING AUTHORITY

To:

Gervasi, Gemma
NOTARBARTOLO & GERVASI S.P.A.
Corso di Porta Vittoria, 9
I-20122 Milano
ITALIE

NOTARBARTOLO & GERVASI MILANO
RECEIVED
20 SET. 2002
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NOTIFICATION OF TRANSMITTAL OF THE INTERNATIONAL PRELIMINARY EXAMINATION REPORT

(PCT Rule 71.1)

Date of mailing (day/month/year) 18.09.2002

Applicant's or agent's file reference
2388PTWO

IMPORTANT NOTIFICATION

International application No.	International filing date (day/month/year)	Priority date (day/month/year)
PCT/EP01/08188	16/07/2001	17/07/2000

Applicant
DALMINE S.P.A. et al.

1. The applicant is hereby notified that this International Preliminary Examining Authority transmits herewith the international preliminary examination report and its annexes, if any, established on the international application.

2. A copy of the report and its annexes, if any, is being transmitted to the International Bureau for communication to all the elected Offices.

3. Where required by any of the elected Offices, the International Bureau will prepare an English translation of the report (but not of any annexes) and will transmit such translation to those Offices.

4. REMINDER

The applicant must enter the national phase before each elected Office by performing certain acts (filing translations and paying national fees) within 30 months from the priority date (or later in some Offices) (Article 39(1)) (see also the reminder sent by the International Bureau with Form PCT/IB/301).

Where a translation of the international application must be furnished to an elected Office, that translation must contain a translation of any annexes to the international preliminary examination report. It is the applicant's responsibility to prepare and furnish such translation directly to each elected Office concerned.

For further details on the applicable time limits and requirements of the elected Offices, see Volume II of the PCT Applicant's Guide.

For the purpose of deciding whether the claimed invention is patentable or not, the elected Offices may apply criteria additional to or different from the criteria on which the international preliminary examination report is based (see Articles 27(5), 33(5)). Additional criteria may include e.g. exemptions from patentability and the requirements of enabling disclosure and of clarity and support of claims.

Name and mailing address of the IPEA/	Authorized officer
European Patent Office - P.B. 5818 Patentlaan 2 NL-2280 HV Rijswijk - Pays Bas Tel. +31 70 340 - 2040 Tx: 31 651 epo nl Fax: +31 70 340 - 3016	Daniels, H Tel.+31 70 340-3718

Form PCT/IPEA/416 (July 1992)

PATENT COOPERATION TREATY

PCT

INTERNATIONAL PRELIMINARY EXAMINATION REPORT

(PCT Article 36 and Rule 70)

AP 53

Applicant's or agent's file reference 2388PTWO	FOR FURTHER ACTION	See Notification of Transmittal of International Preliminary Examination Report (Form PCT/IPEA/416)
International application No. PCT/EP01/08188	International filing date *(day/month/year)* 16/07/2001	Priority date *(day/month/year)* 17/07/2000
International Patent Classification (IPC) or national classification and IPC F16L15/00		
Applicant DALMINE S.P.A. et al.		

1. This international preliminary examination report has been prepared by this International Preliminary Examining Authority and is transmitted to the applicant according to Article 36.

2. This REPORT consists of a total of 6 sheets, including this cover sheet.

☒ This report is also accompanied by ANNEXES, i.e. sheets of the description, claims and/or drawings which have been amended and are the basis for this report and/or sheets containing rectifications made before this Authority (see Rule 70.16 and Section 607 of the Administrative Instructions under the PCT).

These annexes consist of a total of 2 sheets.

3. This report contains indications relating to the following items:

- I ☒ Basis of the report
- II ☐ Priority
- III ☐ Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
- IV ☐ Lack of unity of invention
- V ☒ Reasoned statement under Article 35(2) with regard to novelty, inventive step or industrial applicability; citations and explanations suporting such statement
- VI ☐ Certain documents cited
- VII ☐ Certain defects in the international application
- VIII ☐ Certain observations on the international application

Date of submission of the demand	Date of completion of this report
12/02/2002	18.09.2002
Name and mailing address of the international preliminary examining authority: European Patent Office - P.B. 5818 Patentlaan 2 NL-2280 HV Rijswijk - Pays Bas Tel. +31 70 340 - 2040 Tx: 31 651 epo nl Fax: +31 70 340 - 3016	Authorized officer Schroeder, R Telephone No. +31 70 340 4650

~~I. Basis of the report~~

1. With regard to the elements of the international application *(Replacement sheets which have been furnished to the receiving Office in response to an invitation under Article 14 are referred to in this report as "originally filed" and are not annexed to this report since they do not contain amendments (Rules 70.16 and 70.17)):*

Description, pages:

1-8 as originally filed

Claims, No.:

1-10 with telefax of 05/08/2002

Drawings, sheets:

1/2,2/2 as originally filed

2. With regard to the language, all the elements marked above were available or furnished to this Authority in the language in which the international application was filed, unless otherwise indicated under this item.

   These elements were available or furnished to this Authority in the following language: , which is:

   ☐ the language of a translation furnished for the purposes of the international search (under Rule 23.1(b)).
   ☐ the language of publication of the international application (under Rule 48.3(b)).
   ☐ the language of a translation furnished for the purposes of international preliminary examination (under Rule 55.2 and/or 55.3).

3. With regard to any nucleotide and/or amino acid sequence disclosed in the international application, the international preliminary examination was carried out on the basis of the sequence listing:

   ☐ contained in the international application in written form.
   ☐ filed together with the international application in computer readable form.
   ☐ furnished subsequently to this Authority in written form.
   ☐ furnished subsequently to this Authority in computer readable form.
   ☐ The statement that the subsequently furnished written sequence listing does not go beyond the disclosure in the international application as filed has been furnished.
   ☐ The statement that the information recorded in computer readable form is identical to the written sequence listing has been furnished.

4. The amendments have resulted in the cancellation of:

   ☐ the description, pages:
   ☒ the claims, Nos.: 11

☐ the drawings, sheets:

5. ☒ This report has been established as if (some of) the amendments had not been made, since they have been considered to go beyond the disclosure as filed (Rule 70.2(c)):

(Any replacement sheet containing such amendments must be referred to under item 1 and annexed to this report.)
see separate sheet

6. Additional observations, if necessary:

V. Reasoned statement under Article 35(2) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty (N)	Yes:	Claims	1-10
	No:	Claims	
Inventive step (IS)	Yes:	Claims	1-10
	No:	Claims	
Industrial applicability (IA)	Yes:	Claims	1-10
	No:	Claims	

2. Citations and explanations
see separate sheet

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Re Item I

Basis of the report

The examination is being carried out on the following application documents:

Text for the Contracting States:
AT BE CH CY DE DK ES FI FR GB GR IT IE LI LU MC NL PT SE TR

Description, pages:

1-8 as originally filed

Claims, No.:

1-10 with telefax of 05/08/2002

Drawings, sheets:

1/2,2/2 as originally filed

1. The amendments filed with the International Bureau under Article 19(1) introduce subject-matter which extends beyond the content of the application as filed, contrary to Article 19(2) PCT. The amendments concerned are the following:

   "the surfaces of the first and second annular shoulders are equal or greater to 25% of the area of the section of the pipe walls"

   The passage of the description refered to by the applicant suggests only that the annular shoulder of the female part has an area that is not less than 25% of the area of the section of the pipe. No such indications concerning the male part were disclosed in the original filing.

1.1 The report has therefore been established as if the amendments concerning the annular shoulders had not been made.

Re Item V

Reasoned statement under Article 35(2) with regard to novelty, inventive st p or industrial applicability; citations and explanations supporting such statement

1. Reference is made to the following document:

   D3: EP-A-0 767 335 (VALLOUREC OIL & GAS; SUMITOMO METAL IND (JP)) 9 April 1997 (1997-04-09) cited in the application

2. The document D3 is regarded as being the closest prior art to the subject-matter of claim 1, and shows (the references in parentheses applying to this document):

   a pipe integral threaded joint comprising a male element provided on its external surface with two radially spaced, truncated cone-shaped threaded portions with tooth of trapezoidal profile, said two portions being divided by a first annular shoulder lying on a plane orthogonal to the pipe axis and a female element provided on its internal surface with two radially spaced, truncated cone-shaped threaded portions, said two portions being divided by a second annular shoulder lying on a plane orthogonal to the pipe axis, the two threaded portions having the same conicity value, being adapted to screw mutually and reversibly one inside the other in order to produce a contact between said two annular shoulders in an assembled position, each of said male and female elements being provided with two sealing surfaces the first annular shoulder and the second annular shoulder have the shape of an annulus, lying on a plane for its entity without presenting any elevation from the plane and in that the sealing surfaces are respectively placed at each axial end of the threaded portions of each male and female element.

2.1 The subject-matter of claim 1 therefore differs from this known threaded joint in that a first one of said respective two sealing surfaces has a conical shape and a second one has a spherical shape and in that at least one of said male and female elements comprises along its entire surface a circumference cavity between one of said two threading parts and the corresponding annular shoulder, the at least one cavity being adapted to receive the expansion of the lubricating grease of the joint.

2.2 The subject-matter of claim 1 is therefore novel (Article 33(2) PCT).

3. The problem to be solved by the present invention may therefore be regarded as reducing the load of the metal-to-metal sealing area and at the same time avoiding the problem of the lubricant that is hydraulically pressed between the sealing areas, which could affect complete mechanical build-up of the pipe connection.

3.1 The solution to this problem proposed in claim 1 of the present application is considered as involving an inventive step (Article 33(3) PCT) for the following reasons:

3.2 The combination of the features of claim 1 is not rendered obvious by the available prior art.

4. Claims 2 - 10 are dependent on claim 1 and as such also meet the requirements of the PCT with respect to novelty and inventive step.

CLAIMS

1. Pipe integral threaded joint comprising a male element (1) provided on its external surface with two radially spaced, truncated cone-shaped threaded portions (6,7) with tooth of trapezoidal profile, said two portions (6,7) being divided by a first annular shoulder (5') lying on a plane orthogonal to the pipe axis and a female element (2) provided on its internal surface with two radially spaced, truncated cone-shaped threaded portions (8, 9), said two portions being divided by a second annular shoulder (5") lying on a plane orthogonal to the pipe axis, the two threaded portions (6, 7, 8, 9) having the same conicity value and being adapted to screw mutually and reversibly one inside the other in order to produce a contact between said two annular shoulders (5', 5") in an assembled position, each of said male (1) and female (2) elements being provided with two sealing surfaces (12', 13', 12", 13") the first annular shoulder (5') and the second annular shoulder (5") have the shape of an annulus, lying on a plane for its entirety without presenting any elevation from the plane, and in that the sealing surfaces (12', 13', 12", 13") are respectively placed at each axial end of the threaded portions (6, 7, 8, 9) of each male (1) and female element (2) <u>characterised in that</u> a first one of said respective two sealing surfaces (12', 13', 12", 13") has a conical shape and a second one has a spherical shape and in that the surfaces of the first and second annular shoulders (5', 5") are equal or greater to 25% of the area of the section of the pipe walls and in that at least one of said male (1) and female (2) elements comprises along its entire surface a circumference cavity (14, 15) between one of said two threading parts and the corresponding annular shoulder (5', 5"), the at least one cavity (14,15) being adapted to receive the expansion of the lubricating grease of the joint.

2. Joint according to claim 1, characterized in that both the male (1) and female (2) elements comprise along their entire surface the circumference cavity (14,15), placed between one of the two threaded portions and the corresponding annular shoulder (5', 5").

3. Joint according to claim 2, characterized in that the dimension (D') of the external diameter (3) of said female element (2) at the joint is larger than the dimension (D") of the external diameter (4) of the same female element (2) of a

value equal to or smaller than 3%.

4. Joint according to claim 3, characterized in that said corresponding two threaded portions (6,7,8,9) of said male (1) and female (2) elements each have the same conicity value, comprised between 6.25 and 12.5%.

5. Joint according to claim 4, characterized in that said corresponding conical sealing surfaces (13', 12") of said male (1) and female (2) elements have a conicity value comprised between 12.5% and 25%.

6. Joint according to claim 5, characterized in that the profile of the threading tooth has a loading flank (10) with a negative (α) angle and an stabbing flank (11) with a positive (β) angle.

7. Joint according to claim 6, characterized in that said negative (α) angle has a value comprised between 0 and-10 and said positive (β) angle has a value comprised between 10 and 30 .

8. Joint according to claim 7, characterized in that said corresponding two threaded portions (6,7,8,9) of each of the male (1) and female (2) elements is provided with a perfect thread for its entire length.

9. Joint according to claim 7, characterized in that at least one of said two threaded portions (6, 7, 8, 9) of at least one of the male (1) or female (2) elements is provided with a first end (6', 7') with an imperfect thread.

10. Joint according to claim 9, characterized in that the corresponding threaded portion of the other of said at least one of male (1) and female (2) elements is also provided with a second area (8', 9') having an imperfect thread placed at th axially opposite end with respect to said first end.

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REQUEST

The undersigned requests that the present international application be processed according to the Patent Cooperation Treaty.

For receiving Office use only

PCT/EP 01/08188

International Application No.

16 JUL 2001 (16.07.01)

International Filing Date

EUROPEAN PATENT OFFICE
PCT INTERNATIONAL APPLICATION

Name of receiving Office and "PCT International Application"

Applicant's or agent's file reference *(if desired) (12 characters maximum)* 2388PTWO/er

Box No. I TITLE OF INVENTION

PIPE INTEGRAL THREADED JOINT

Box No. II APPLICANT ☐ This person is also inventor

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*

DALMINE S.p.A.
Piazza Caduti 6 Luglio 1944, 1
24044 DALMINE (Province of BERGAMO) - ITALY

Telephone No.

Facsimile No.

Teleprinter No.

Applicant's registration No. with the Office

State *(that is, country)* of nationality: IT

State *(that is, country)* of residence: IT

This person is applicant for the purposes of: ☐ all designated States ☒ all designated States except the United States of America ☐ the United States of America only ☐ the States indicated in the Supplemental Box

Box No. III FURTHER APPLICANT(S) AND/OR (FURTHER) INVENTOR(S)

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*

DELLA PINA Giuseppe
Via Buttaro 26
24044 DALMINE (Province of BERGAMO) - ITALY

This person is:
☐ applicant only
☒ applicant and inventor
☐ inventor only *(If this check-box is marked, do not fill in below.)*

Applicant's registration No. with the Office

State *(that is, country)* of nationality: IT

State *(that is, country)* of residence: IT

This person is applicant for the purposes of: ☐ all designated States ☐ all designated States except the United States of America ☒ the United States of America only ☐ the States indicated in the Supplemental Box

☒ Further applicants and/or (further) inventors are indicated on a continuation sheet.

Box No. IV AGENT OR COMMON REPRESENTATIVE; OR ADDRESS FOR CORRESPONDENCE

The person identified below is hereby/has been appointed to act on behalf of the applicant(s) before the competent International Authorities as: ☒ agent ☐ common representative

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country.)*

GERVASI Gemma
NOTARBARTOLO & GERVASI S.p.A.
Corso di Porta Vittoria 9
20122 MILAN - ITALY

Telephone No.
+39 02541799.1

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Teleprinter No.

Agent's registration No. with the Office

☐ **Address for correspondence:** Mark this check-box where no agent or common representative is/has been appointed and the space above is used instead to indicate a special address to which correspondence should be sent.

Continuation of Box No. III FURTHER APPLICANT(S) AND/OR (FURTHER) INVENTOR(S)

If none of the following sub-boxes is used, this sheet should not be included in the request.

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*	This person is:
ROTINI Giuseppe Via XXV Aprile 56/A 24044 DALMINE (Province of BERGAMO) - ITALY	☐ applicant only ☒ applicant and inventor ☐ inventor only *(If this check-box is marked, do not fill in below.)* Applicant's registration No. with the Office
State *(that is, country)* of nationality: IT	State *(that is, country)* of residence: IT

This person is applicant for the purposes of: ☐ all designated States ☐ all designated States except the United States of America ☒ the United States of America only ☐ the States indicated in the Supplemental Box

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*	This person is:
SIGNORELLI Angelo Via Portici Manarini 18E 24060 CHIUDUNO (Province of BERGAMO) - ITALY	☐ applicant only ☒ applicant and inventor ☐ inventor only *(If this check-box is marked, do not fill in below.)* Applicant's registration No. with the Office
State *(that is, country)* of nationality: IT	State *(that is, country)* of residence: IT

This person is applicant for the purposes of: ☐ all designated States ☐ all designated States except the United States of America ☒ the United States of America only ☐ the States indicated in the Supplemental Box

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*	This person is:
	☐ applicant only ☐ applicant and inventor ☐ inventor only *(If this check-box is marked, do not fill in below.)* Applicant's registration No. with the Office
State *(that is, country)* of nationality:	State *(that is, country)* of residence:

This person is applicant for the purposes of: ☐ all designated States ☐ all designated States except the United States of America ☐ the United States of America only ☐ the States indicated in the Supplemental Box

Name and address: *(Family name followed by given name; for a legal entity, full official designation. The address must include postal code and name of country. The country of the address indicated in this Box is the applicant's State (that is, country) of residence if no State of residence is indicated below.)*	This person is:
	☐ applicant only ☐ applicant and inventor ☐ inventor only *(If this check-box is marked, do not fill in below.)* Applicant's registration No. with the Office
State *(that is, country)* of nationality:	State *(that is, country)* of residence:

This person is applicant for the purposes of: ☐ all designated States ☐ all designated States except the United States of America ☐ the United States of America only ☐ the States indicated in the Supplemental Box

☐ Further applicants and/or (further) inventors are indicated on another continuation sheet.

PCT/EP 01/08188

Box No. V DESIGNATION OF STATES *Mark the applicable check-boxes below; at least one must be marked.*

The following designations are hereby made under Rule 4.9(a):

Regional Patent

☒ **AP ARIPO Patent:** GH Ghana, GM Gambia, KE Kenya, LS Lesotho, MW Malawi, MZ Mozambique, SD Sudan, SL Sierra Leone, SZ Swaziland, TZ United Republic of Tanzania, UG Uganda, ZW Zimbabwe, and any other State which is a Contracting State of the Harare Protocol and of the PCT

☒ **EA Eurasian Patent:** AM Armenia, AZ Azerbaijan, BY Belarus, KG Kyrgyzstan, KZ Kazakhstan, MD Republic of Moldova, RU Russian Federation, TJ Tajikistan, TM Turkmenistan, and any other State which is a Contracting State of the Eurasian Patent Convention and of the PCT

☒ **EP European Patent:** AT Austria, BE Belgium, CH & LI Switzerland and Liechtenstein, CY Cyprus, DE Germany, DK Denmark, ES Spain, FI Finland, FR France, GB United Kingdom, GR Greece, IE Ireland, IT Italy, LU Luxembourg, MC Monaco, NL Netherlands, PT Portugal, SE Sweden, TR Turkey, and any other State which is a Contracting State of the European Patent Convention and of the PCT

☒ **OA OAPI Patent:** BF Burkina Faso, BJ Benin, CF Central African Republic, CG Congo, CI Côte d'Ivoire, CM Cameroon, GA Gabon, GN Guinea, GW Guinea-Bissau, ML Mali, MR Mauritania, NE Niger, SN Senegal, TD Chad, TG Togo, and any other State which is a member State of OAPI and a Contracting State of the PCT *(if other kind of protection or treatment desired, specify on dotted line)* ....

National Patent *(if other kind of protection or treatment desired, specify on dotted line):*

☒ AE United Arab Emirates
☒ AG Antigua and Barbuda
☒ AL Albania ....
☒ AM Armenia ....
☒ AT Austria ....
☒ AU Australia ....
☒ AZ Azerbaijan
☒ BA Bosnia and Herzegovina ....
☒ BB Barbados
☒ BG Bulgaria ....
☒ BR Brazil ....
☒ BY Belarus ....
☒ BZ Belize ....
☒ CA Canada
☒ CH & LI Switzerland and Liechtenstein
☒ CN China ....
☒ CO Colombia
☒ CR Costa Rica ....
☒ CU Cuba ....
☒ CZ Czech Republic ....
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☒ DK Denmark ....
☒ DM Dominica
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☒ IS Iceland
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☒ KG Kyrgyzstan ....
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☒ KZ Kazakhstan ....
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☒ LR Liberia
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☒ MG Madagascar ....
☒ MK The former Yugoslav Republic of Macedonia ....
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☒ SD Sudan
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☒ SI Slovenia ....
☒ SK Slovakia ....
☒ SL Sierra Leone ....
☒ TJ Tajikistan ....
☒ TM Turkmenistan ....
☒ TR Turkey ....
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☒ TZ United Republic of Tanzania
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☒ UG Uganda ....
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Check-boxes below reserved for designating States which have become party to the PCT after issuance of this sheet:

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Precautionary Designation Statement: In addition to the designations made above, the applicant also makes under Rule 4.9(b) all other designations which would be permitted under the PCT except any designation(s) indicated in the Supplemental Box as being excluded from the scope of this statement. The applicant declares that those additional designations are subject to confirmation and that any designation which is not confirmed before the expiration of 15 months from the priority date is to be regarded as withdrawn by the applicant at the expiration of that time limit. *(Confirmation (including fees) must reach the receiving Office within the 15-month time limit.)*

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Sheet No. 4.......

Box No. VI PRIORITY CLAIM

The priority of the following earlier application(s) is hereby claimed:

Filing date of earlier application *(day/month/year)*	Number of earlier application	Where earlier application is: national application: country	regional application:* regional Office	international application: receiving Office
item (1) 17 July 2000 (17.07.2000)	MI2000A001614	ITALY		
item (2)				
item (3)				
item (4)				
item (5)				

☐ Further priority claims are indicated in the Supplemental Box.

The receiving Office is requested to prepare and transmit to the International Bureau a certified copy of the earlier application(s) *(only if the earlier application was filed with the Office which for the purposes of this international application is the receiving Office)* identified above as:

☐ all items ☐ item (1) ☐ item (2) ☐ item (3) ☐ item (4) ☐ item (5) ☐ other, see Supplemental Box

* *Where the earlier application is an ARIPO application, indicate at least one country party to the Paris Convention for the Protection of Industrial Property or one Member of the World Trade Organization for which that earlier application was filed (Rule 4.10(b)(ii)):* ....
....

Box No. VII INTERNATIONAL SEARCHING AUTHORITY

Choice of International Searching Authority (ISA) *(if two or more International Searching Authorities are competent to carry out the international search, indicate the Authority chosen; the two-letter code may be used)*:

ISA / ....

Request to use results of earlier search; reference to that search *(if an earlier search has been carried out by or requested from the International Searching Authority)*:

Date *(day/month/year)* Number Country *(or regional Office)*

Box No. VIII DECLARATIONS

The following declarations are contained in Boxes Nos. VIII (i) to (v) *(mark the applicable check-boxes below and indicate in the right column the number of each type of declaration)*: Number of declarations

☐ Box No. VIII (i) Declaration as to the identity of the inventor :

☐ Box No. VIII (ii) Declaration as to the applicant's entitlement, as at the international filing date, to apply for and be granted a patent :

☐ Box No. VIII (iii) Declaration as to the applicant's entitlement, as at the international filing date, to claim the priority of the earlier application :

☐ Box No. VIII (iv) Declaration of inventorship (only for the purposes of the designation of the United States of America) :

☐ Box No. VIII (v) Declaration as to non-prejudicial disclosures or exceptions to lack of novelty :

Box No. IX CHECK LIST; LANGUAGE OF FILING

This international application contains:		This international application is accompanied by the following item(s) *(mark the applicable check-boxes below and indicate in right column the number of each item)*:	Number of items
(a) the following number of sheets in paper form:		1. ☐ fee calculation sheet	
~~request~~ (including declaration sheets)	5	2. ☒ original separate power of attorney	2
description (excluding sequence listing part)	8	3. ☐ original general power of attorney	
claims	2	4. ☐ copy of general power of attorney; reference number, if any:	
abstract	1	5. ☐ statement explaining lack of signature	
drawings	2	6. ☐ priority document(s) identified in Box No. VI as item(s):	
Sub-total number of sheets	18	7. ☐ translation of international application into *(language)*:	
sequence listing part of description *(actual number of sheets if filed in paper form, whether or not also filed in computer readable form; see (b) below)*		8. ☐ separate indications concerning deposited microorganism or other biological material	
Total number of sheets	18	9. ☐ sequence listing in computer readable form (indicate also type and number of carriers (diskette, CD-ROM, CD-R or other))	
(b) sequence listing part of description filed in computer readable form		(i) ☐ copy submitted for the purposes of international search under Rule 13*ter* only (and not as part of the international application)	
(i) ☐ only (under Section 801(a)(i))		(ii) ☐ *(only where check-box (b)(i) or (b)(ii) is marked in left column)* additional copies including, where applicable, the copy for the purposes of international search under Rule 13*ter*	
(ii) ☐ in addition to being filed in paper form (under Section 801(a)(ii))		(iii) ☐ together with relevant statement as to the identity of the copy or copies with the sequence listing part mentioned in left column	
Type and number of carriers (diskette, CD-ROM, CD-R or other) on which the sequence listing part is contained *(additional copies to be indicated under item 9(ii), in right column)*:		10. ☒ other *(specify)*: letter	1

Figure of the drawings which should accompany the abstract: 1	Language of filing of the international application: ENGLISH

Box No. X SIGNATURE OF APPLICANT, AGENT OR COMMON REPRESENTATIVE

Next to each signature, indicate the name of the person signing and the capacity in which the person signs (if such capacity is not obvious from reading the request).

GERVASI Gemma

Milan, 12 July 2001

For receiving Office use only

1. Date of actual receipt of the purported international application:	(16.07.01) 16 JUL 2001	2. Drawings: ☒ received: ☐ not received:
3. Corrected date of actual receipt due to later but timely received papers or drawings completing the purported international application:		
4. Date of timely receipt of the required corrections under PCT Article 11(2):		
5. International Searching Authority (if two or more are competent): ISA /	6. ☐ Transmittal of search copy delayed until search fee is paid	

For International Bureau use only

Date of receipt of the record copy by the International Bureau:

WO 02/06715
PCT/EP01/08188
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PATENT COOPERATION TREATY

From the INTERNATIONAL BUREAU

PCT

To:
GERVASI, Gemma
Notarbartolo & Gervasi S.p.A.
Corso di Porta Vittoria, 9
I-20122 Milan
ITALIE

NOTARBARTOLO & GERVASI
MILANO
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- 1 FEB. 2002

NOTICE INFORMING THE APPLICANT OF THE COMMUNICATION OF THE INTERNATIONAL APPLICATION TO THE DESIGNATED OFFICES

(PCT Rule 47.1(c), first sentence)

Date of mailing (day/month/year) 24 January 2002 (24.01.02)	
Applicant's or agent's file reference 2388PTWO/er	IMPORTANT NOTICE

International application No.	International filing date (day/month/year)	Priority date (day/month/year)
PCT/EP01/08188	16 July 2001 (16.07.01)	17 July 2000 (17.07.00)

Applicant
DALMINE S.P.A. et al

1. Notice is hereby given that the International Bureau has communicated, as provided in Article 20, the international application to the following designated Offices on the date indicated above as the date of mailing of this notice:
KP,KR,US

In accordance with Rule 47.1(c), third sentence, those Offices will accept the present notice as conclusive evidence that the communication of the international application has duly taken place on the date of mailing indicated above and no copy of the international application is required to be furnished by the applicant to the designated Office(s).

2. The following designated Offices have waived the requirement for such a communication at this time:
AE,AG,AL,AM,AP,AT,AU,AZ,BA,BB,BG,BR,BY,BZ,CA,CH,CN,CO,CR,CU,CZ,DE,DK,DM,DZ,EA,EC, EE,EP,ES,FI,GB,GD,GE,GH,GM,HR,HU,ID,IL,IN,IS,JP,KE,KG,KZ,LC,LK,LR,LS,LT,LU,LV,MA, MD,MG,MK,MN,MW,MX,MZ,NO,NZ,OA,PL,PT,RO,RU,SD,SE,SG,SI,SK,SL,TJ,TM,TR,TT,TZ,UA,UG,

The communication will be made to those Offices only upon their request. Furthermore, those Offices do not require the applicant to furnish a copy of the international application (Rule 49.1(a-bis)).

3. Enclosed with this notice is a copy of the international application as published by the International Bureau on
24 January 2002 (24.01.02) under No. WO 02/06715

REMINDER REGARDING CHAPTER II (Article 31(2)(a) and Rule 54.2)

If the applicant wishes to postpone entry into the national phase until 30 months (or later in some Offices) from the priority date, a demand for international preliminary examination must be filed with the competent International Preliminary Examining Authority before the expiration of 19 months from the priority date.

It is the applicant's sole responsibility to monitor the 19-month time limit.

Note that only an applicant who is a national or resident of a PCT Contracting State which is bound by Chapter II has the right to file a demand for international preliminary examination (at present, all PCT Contracting States are bound by Chapter II).

REMINDER REGARDING ENTRY INTO THE NATIONAL PHASE (Article 22 or 39(1))

If the applicant wishes to proceed with the international application in the national phase, he must, within 20 months or 30 months, or later in some Offices, perform the acts referred to therein before each designated or elected Office.

For further important information on the time limits and acts to be performed for entering the national phase, see the Annex to Form PCT/IB/301 (Notification of Receipt of Record Copy) and the PCT Applicant's Guide, Volume II.

The International Bureau of WIPO 34, chemin des Colombettes 1211 Geneva 20, Switzerland	Authorized officer J. Zahra
Facsimile No. (41-22) 740.14.35	Telephone No. (41-22) 338.91.11

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Continuation of Form PCT/IB/308

NOTICE INFORMING THE APPLICANT OF THE COMMUNICATION OF THE INTERNATIONAL APPLICATION TO THE DESIGNATED OFFICES

Date of mailing (day/month/year) 24 January 2002 (24.01.02)	IMPORTANT NOTICE
Applicant's or agent's file reference 2388PTWO/er	International application No. PCT/EP01/08188

The applicant is hereby notified that, at the time of establishment of this Notice, the time limit under Rule 46.1 for making amendments under Article 19 has not yet expired and the International Bureau had received neither such amendments nor a declaration that the applicant does not wish to make amendments.

PCT/EP01/08188

PATENT COOPERATION TREATY

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From the INTERNATIONAL BUREAU

PCT

To:

GERVASI, Gemma
Notarbartolo & Gervasi S.p.A.
Corso di Porta Vittoria, 9
I-20122 Milan
ITALIE

NOTARBARTOLO & GERVASI
MILANO
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12 MAR. 2002

INFORMATION CONCERNING ELECTED OFFICES NOTIFIED OF THEIR ELECTION

(PCT Rule 61.3)

Date of mailing (day/month/year) 05 March 2002 (05.03.02)	
Applicant's or agent's file reference 2388PTWO/er	IMPORTANT INFORMATION

International application No.	International filing date (day/month/year)	Priority date (day/month/year)
PCT/EP01/08188	16 July 2001 (16.07.01)	17 July 2000 (17.07.00)

Applicant
DALMINE S.P.A. et al

1. The applicant is hereby informed that the International Bureau has, according to Article 31(7), notified each of the following Offices of its election:

   EP :AT,BE,CH,CY,DE,DK,ES,FI,FR,GB,GR,IE,IT,LU,MC,NL,PT,SE,TR
   National :AU,BG,CA,CN,DE,GB,IL,JP,KP,KR,MN,NO,NZ,PL,RO,RU,SE,SK,US

2. The following Offices have waived the requirement for the notification of their election; the notification will be sent to them by the International Bureau only upon their request:

   AP :GH,GM,KE,LS,MW,MZ,SD,SL,SZ,TZ,UG,ZW
   EA :AM,AZ,BY,KG,KZ,MD,RU,TJ,TM
   OA :BF,BJ,CF,CG,CI,CM,GA,GN,GW,ML,MR,NE,SN,TD,TG
   National :AE,AG,AL,AM,AT,AZ,BA,BB,BR,BY,BZ,CH,CO,CR,CU,CZ,DK,DM,DZ,EC,EE,ES,FI,GD,GE,GH,GM,HR,HU,ID,IN,IS,KE,KG,KZ,LC,LK,LR,LS,LT,LU,LV,MA,MD,MG,MK,MW,MX,MZ,PT,SD,SG,SI,SL,TJ,TM,TR,TT,TZ,UA,UG,UZ,VN,YU,ZA,ZW

3. The applicant is reminded that he must enter the "national phase" before the expiration of 30 months from the priority date before each of the Offices listed above. This must be done by paying the national fee(s) and furnishing, if prescribed, a translation of the international application (Article 39(1)(a)), as well as, where applicable, by furnishing a translation of any annexes of the international preliminary examination report (Article 36(3)(b) and Rule 74.1).

   Some offices have fixed time limits expiring later than the above-mentioned time limit. For detailed information about the applicable time limits and the acts to be performed upon entry into the national phase before a particular Office, see Volume II of the PCT Applicant's Guide.

   The entry into the European regional phase is postponed until 31 months from the priority date for all States designated for the purposes of obtaining a European patent.

The International Bureau of WIPO 34, chemin des Colombettes 1211 Geneva 20, Switzerland Facsimile No. (41-22) 740.14.35	Authorized officer: Zakaria EL KHODARY Telephone No. (41-22) 338.83.38

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JUNTA ROSCADA INTEGRAL PARA TUBOS

CAMPO DE LA INVENCIÓN

La presente invención concierne a una junta roscada integral para tubos, que tienen dimensiones diametrales reducidas, en particular para tubos empleados en la industria de explotación de campos de gas o petróleo. Esos tubos pueden usarse tanto como conductos para el bombeo del gas o del petróleo así como para el revestimiento de los pozos.

ANTECEDENTES DE LA INVENCIÓN

La industria de la explotación de gas natural y petróleo, emplea tubos que tienen una longitud fija, que deben ser conectados a los extremos de otros, a fin de llegar a las enormes profundidades en donde se localizan generalmente las minas de hidrocarburos.

En la técnica de perforación más comúnmente empleada, se perforan pozos que empiezan desde la tierra o desde la superficie del mar, a fin de llegar al campo de petróleo o gas. Esos pozos pueden tener una profundidad de varios miles de metros. Durante la perforación, los pozos se recubren por dentro con tubos metálicos, en toda su longitud.

Los segmentos de tubo metálico, que tienen una longitud de aproximadamente 10 metros, se conectan unos con otros mediante juntas roscadas. Por lo tanto, estos tubos forman una línea tubular que tiene el mismo diámetro en toda su longitud excepto en las juntas, cuyo diámetro externo es generalmente de al menos 25.4 mm (1 pulgada) mayor que el de la línea.

A fin de formar un revestimiento para toda la profundidad del pozo, se usan varias líneas que tienen, debido a la resistencia mecánica y a las características geológicas del sitio, menor diámetro mientras mayor es la profundidad de la línea, a fin de formar una estructura "telescópica". Se comprende que, dado que el diámetro en la parte inferior del pozo varía de acuerdo con la presión y el flujo de los fluidos que se van a extraer, mientras más profundo es el pozo, mayor será su diámetro en la superficie. Como consecuencia desfavorable de lo mencionado anteriormente, el costo de perforación es alto y, además, estos revestimientos para pozos requieren de una gran cantidad de material, siendo entonces muy caros. Un diámetro más pequeño en los pozos significa también menores tiempos de perforación y revestimiento. Por lo tanto es esencial minimizar el diámetro del pozo y el diámetro del tubo de revestimiento, con una cantidad igual de fluidos extraídos.

Una vez finalizada la perforación, dentro del pozo revestido se inserta otra línea tubular que debe bombear el gas o el petróleo hacia afuera del campo subterráneo. Esta línea, que sigue la profundidad total del pozo y que puede alcanzar por lo tanto una longitud de varios miles de metros, se forma también conectando tubos que tienen también una longitud de aproximadamente 10 metros, mediante las juntas mencionadas anteriormente. Usualmente, también este segundo tipo de línea tiene el mismo diámetro en toda su longitud excepto en las juntas, cuyo diámetro externo es en general mayor, como sucede en el caso previo.

En ambos casos mencionados anteriormente los tubos se conectan mediante juntas roscadas que pueden ser integrales, y en este caso un extremo del tubo tiene una rosca macho, y el otro extremo tiene una rosca hembra, o pueden ser del tipo manguito de unión, y en este caso ambos extremos del tubo tienen una rosca macho, conectándose mediante un manguito de unión con rosca hembra por ambos lados. Usualmente, la presencia de la junta involucra un incremento del diámetro externo de la línea, en donde se encuentran las juntas; mientras mayor es el diámetro externo de la junta, en comparación con el tubo, mayores son las dimensiones generales de la línea y del pozo de sondeo.

Después de que las compañías petroleras requirieron minimizar los costos de explotación de petróleo y gas, se llevaron a cabo notables esfuerzos a fin de reducir el diámetro de los pozos y por lo tanto el diámetro de los tubos.

A fin de reducir el diámetro externo de la línea, y por lo tanto los costos de perforación y la cantidad de material requerido, se usaron juntas roscadas que tenían dimensiones diametrales reducidas, que podían ser divididas en tres tipos diferentes, de acuerdo con el rendimiento requerido y las dimensiones máximas permitidas. Un primer tipo, que es llamado a menudo "semi-alojado", es una junta de acoplamiento cuyo diámetro externo no excede más del 6 % del diámetro externo del tubo. Un segundo tipo, usualmente llamado "casi alojado", es una junta integral cuyo diámetro externo es aproximadamente de 2 a 3 % mayor que el diámetro externo del tubo. Un último tipo, llamado "alojado" es una junta integral cuyo diámetro externo corresponde al diámetro externo del tubo.

La selección entre los diferentes tipos de juntas depende de la carga existente en la línea de tubos, de la presión que actúe interna y/o externamente, de su longitud y de sus dimensiones diametrales máximas permitidas con respecto al diámetro del pozo.

Si el diámetro de la junta se reduce, es necesario encontrar nuevas soluciones capaces de compensar la menor resistencia estructural. De hecho, en las partes cercanas a las juntas, la eficacia es necesariamente menor que en el tubo, debido a que los elementos estructurales tales como la rosca, los obturadores y los salientes están colocados en el espesor de la pared del tubo y esto involucra una sección reducida en las partes críticas del elemento macho o hembra.

Una minimización de las causas de ruptura de los tubos es esencial debido a que esas rupturas, principalmente después de que los tubos han sido colocados en el subsuelo, y que por lo tanto es casi imposible que los operarios reemplacen una junta rota, puede tener consecuencias económicas extremadamente serias y puede causar problemas ambientales, particularmente si el campo de petróleo o gas contiene elementos dañinos.

Por lo tanto, en el pasado se ha hecho mucho para mejorar las juntas y para alcanzar un nivel de eficacia óptimo, para equilibrar las diferentes necesidades, que a veces contrastan entre sí, de dimensiones mínimas, resistencia estructural máxima y hermeticidad contra la salida y/o entrada de los fluidos. De hecho, los tubos sufren varias fuerzas, especialmente de compresión, tracción, flexión y de la presión producida por fluidos externos y/o por fluidos que circulan dentro de los tubos.

Las juntas deben tener también una resistencia óptima al enroscado y conexión.

Los problemas estructurales y de obturación se

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empeoran a menudo por la temperatura del fluido, por su corrosividad o por las condiciones ambientales de las áreas de excavación. En el pasado se han propuesto varias soluciones a fin de satisfacer las condiciones mencionadas anteriormente.

La Solicitud de Patente Internacional WO-A-93/18329 concierne a una junta con dimensiones diametrales reducidas, que tiene un saliente central. Ese saliente está provisto, tanto sobre el elemento macho como sobre el elemento hembra, con una proyección y una cavidad que son paralelas al eje del tubo, que tienen superficies homólogas que se acoplan a fin de bloquear los dos elementos de la junta. Sobre las proyecciones del saliente se proporcionan dos superficies obturadoras. El saliente divide dos partes roscadas radialmente descentradas, que tienen forma de cono o forma de cono y cilíndrica.

Esta junta es muy eficiente pero tiene una estructura particularmente compleja que involucra costos de producción muy altos.

La Solicitud de Patente Europea EP-A-767335 concierne a una junta con dimensiones diametrales reducidas, provista de un saliente central y dos obturadores metálicos colocados en el extremo de los elementos macho y hembra. El saliente divide dos partes roscadas radialmente descentradas, que tienen forma de cono y cilíndrica.

Esta estructura de junta es más simple que la estructura previa, pero siempre es relativamente compleja debido a la rosca en forma de cono y cilíndrica. Además, los dos obturadores que se encuentran en los extremos de las juntas, sellan herméticamente su interior, presurizando así la grasa lubricante para el enroscado, con posibles efectos negativos en la hermeticidad de la junta.

BREVE DESCRIPCIÓN DE LA INVENCIÓN

Un primer objeto de la presente invención es superar las desventajas, mencionadas anteriormente, de las juntas de la técnica anterior, mediante una nueva junta del tipo "casi alojado" que, aunque tiene las mismas dimensiones de las juntas de la técnica anterior, no involucra los problemas mencionados anteriormente. Un objeto particular de la presente invención es proporcionar una junta integral que tenga dimensiones diametrales reducidas y menores costos de producción, a la vez que asegure una alta resistencia y hermeticidad operativa.

Un objeto adicional de la presente invención es proporcionar una modalidad de esta junta, que simplifique su instalación.

Estos objetos se consiguen mediante una junta integral para tubos que, de acuerdo con la reivindicación 1, comprende un elemento macho cuya superficie externa está provista de dos porciones roscadas en forma de cono truncado, radialmente separadas, en donde las dos porciones están divididas por un primer saliente anular que se encuentra sobre un plano ortogonal al eje del tubo, y un elemento hembra cuya superficie interna está provista de dos porciones roscadas con forma de cono truncado, radialmente separadas, en donde las dos porciones están divididas por un segundo saliente anular que queda sobre un plano ortogonal al eje del tubo, en donde cada uno de los elementos macho y hembra están provistos de dos superficies obturadoras colocadas respectivamente sobre los extremos opuestos con respecto a las porciones roscadas, las dos porciones roscadas respectivas son capaces de enroscarse en forma mutua y reversible, una dentro de la otra, a fin de producir un contacto entre los dos salientes anulares, la junta está caracterizada porque las dos porciones roscadas, correspondientes, de cada elemento macho y hembra, tienen el mismo valor de conicidad, y porque la primera de las dos superficies obturadoras, respectivas, de los elementos macho y hembra, tiene una forma cónica, y una segunda superficie tiene una forma esférica.

Gracias a esta modalidad, la junta permite una fácil instalación de la línea de tubos, con riesgo reducido de daño a las juntas, a la vez que asegura una resistencia óptima de la linea de tubos en las juntas.

Modalidades preferidas de la presente invención se describen con detalle en las reivindicaciones dependientes.

BREVE DESCRIPCIÓN DE LOS DIBUJOS

Caracteristicas y ventajas adicionales de la presente invención serán evidentes a partir de la siguiente descripción detallada de una modalidad preferida, aunque no exclusiva, de una junta integral, proporcionada como un ejemplo no limitativo, con referencia a los siguientes dibujos, en donde:

La figura 1 muestra una sección transversal a lo largo de un plano axial de la junta de acuerdo con la invención, en donde los dos elementos se encuentran en una posición desunida;

La figura 2 muestra la junta de la figura 1 con los dos elementos en una posición ensamblada;

La figura 3 muestra una vista agrandada, de un detalle de la rosca de la junta de la figura 1;

La figura 4 muestra una sección transversal a lo largo de un plano axial de otra modalidad de la junta de acuerdo con la invención, en donde los dos elementos están en una posición desunida;

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La figura 5 muestra la junta de la figura 4, con los dos elementos en una posición ensamblada.

DESCRIPCIÓN DETALLADA DE LA INVENCIÓN

Con referencia a las figuras mencionadas anteriormente, la junta de conformidad con la presente invención comprende dos elementos o tubos, a decir el elemento macho 1 y el elemento hembra 2. La junta está formada por una parte interna 20 en donde fluyen los fluidos, por ejemplo gas natural o petróleo, u otro fluido análogo, bajo presión, y una parte externa 30 que puede ser llenada también con diferentes fluidos o líquidos, usualmente también presurizados. El diámetro externo 3 del tubo 2 en el área de la junta, tiene el diámetro máximo en donde una dimensión D' es ligeramente mayor que el diámetro externo 4 del tubo 2 en el área central lejana a la junta, con una dimensión D". Las dos áreas del tubo, mencionadas anteriormente, que tienen diferentes diámetros 3 y 4 se conectan en una forma gradual. La dimensión D' es mayor que la dimensión D" en un valor igual o menor que 3 %.

El elemento hembra 2 de la junta tiene una rosca interna dividida en dos porciones, o escalones, 8 y 9, con una generatriz cónica con cierto ángulo respecto al eje del tubo. Esas dos partes 8 y 9 están radialmente descentradas y la rosca puede ser perfecta en toda su longitud o puede tener una porción imperfecta. Esa modalidad se muestra en las figuras 1 y 2.

En otra modalidad de la presente invención, mostrada en las figuras 4 y 5, las porciones roscadas 7 y 8 pueden tener respectivamente extremos con una rosca imperfecta. Esas dos porciones 8 y 9 tienen la misma conicidad, con valores comprendidos entre 6.25 y 12.5 %. Este intervalo resulta ser óptimo debido a que, por una parte, valores menores involucrarían roscas demasiado largas y un acoplamiento difícil, y por otra parte, valores mayores involucrarían un número reducido de dientes y por lo tanto una capacidad de soporte insuficiente de la rosca.

En el área de conexión entre la parte roscada y la superficie interna del tubo, lejos de la junta, el elemento hembra está provisto de un área 12" que tiene una superficie de forma cónica, con respecto al eje del tubo. La conicidad de esta superficie se encuentra comprendida entre 12.5 % y 25 % a fin de asegurar una hermeticidad buena con la superficie de contacto, correspondiente, del elemento macho, reduciendo así el tiempo de deslizamiento durante el paso de enroscado.

Este intervalo resultó ser óptimo con respecto al valor de conicidad de la rosca y pudo reducir la influencia negativa de las cargas de tracción. En el área de conexión entre las dos porciones roscadas 8, 9, el elemento hembra está provisto de un saliente 5" que queda substancialmente sobre un plano perpendicular con respecto al eje de la junta, cuya área no es menor que el 25 % del área de la sección del tubo.

Entre el saliente 5" y el principio de la parte roscada 8 se proporciona una cavidad 15 que se extiende a lo largo de toda la circunferencia interna del elemento hembra 2. La cavidad asegura un tanque de expansión para la grasa usada para lubricar la junta, que está presente en las dos porciones roscadas 8, 9 y es arrastrada por el empuje generado por el deslizamiento de los elementos 1 y 2 durante el enroscado. Esa solución limita el desarrollo de una presión excesiva de la grasa, causada por la presencia de un obturador metálico doble en los extremos de la junta, con un esfuerzo reducido subsiguiente de la junta.

En su extremo el elemento hembra 2 está provisto de una superficie esférica 13" que, al hacer contacto con una área cónica 13' con la que queda confrontado después de que se enrosca con el elemento macho 1, asegura la hermeticidad de la junta contra la presión externa.

El perfil del diente de la rosca es del tipo "gancho", con un flanco de carga 10 que tiene un ángulo α negativo, con valores comprendidos entre 0 y -10° y un flanco de entrada 11 que tiene un ángulo β positivo con valores

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comprendidos entre 10 y 30°. Estos intervalos de valores tienen ventajas notables que mantienen una fácil instalación de la junta. Un flanco de carga con ángulo negativo permite un acoplamiento eficiente de los dos elementos de la junta y reduce el peligro de un posible retiro de la junta debido a cargas de tracción demasiado altas. Un flanco de entrada con un ángulo positivo pero no demasiado grande, permite una cooperación eficiente de la rosca, a la resistencia contra las cargas de compresión.

En el área de la superficie externa cercana a la porción roscada del elemento hembra, el elemento macho 1 está provisto de una rosca colocada en una forma perfectamente recíproca, que tiene porciones de forma exactamente análoga con respecto a la porción roscada del elemento hembra.

El elemento 1 está provisto de dos porciones roscadas 6, 7 separadas por un saliente 5' y por una cavidad circunferencial 14, entre el saliente 5' y el comienzo de la porción roscada 7. La presencia de las dos cavidades 14 y 15 para la expansión de la grasa, permite limitar el incremento de presión de la grasa después del enroscado de la junta, evitando así un esfuerzo excesivo en la junta y mejorando su funcionalidad.

El área de conexión entre la superficie externa del elemento 1 y el comienzo de la porción roscada 7 tiene

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una superficie de forma cónica 13' con una conicidad comprendida entre 12.5 y 25 %. Esta superficie ejerce presión contra la superficie 13" después del enroscado de la junta, y las dimensiones y tolerancias se seleccionan de manera tal que el contacto entre metal y metal garantice la hermeticidad necesaria para evitar la entrada del liquido o fluido que se encuentra bajo presión afuera de la junta.

En forma análoga, en su extremo, el elemento macho 1 tiene una superficie esférica 12' que, después del enroscado con el elemento hembra 2, ejerce presión contra la superficie cónica 12' del elemento hembra. También, en esta segunda área se genera una presión por el contacto metal-metal entre los dos elementos, la cual asegura la hermeticidad necesaria contra la presión del fluido que se encuentra dentro del tubo. La selección de dos áreas en los extremos de la junta, en donde se proporcionan obturadores metálicos que tienen, en forma respectiva, superficies esférica y cónica correspondientes, de conformidad con la invención, hace que la junta sea menos sensible a las cargas de presión y da por resultado que sea óptima para tubos delgados. En efecto, considerando la delgadez de los extremos provistos de las superficies obturadoras 12' y 13', la presión, respectivamente interna o externa al tubo, que actúa en los extremos mencionados anteriormente, causa fácilmente su flexión. Por lo tanto, una superficie obturadora esférica puede mantener un contacto óptimo, a diferencia de un obturador en forma de cono truncado, que, en este caso, debido a la rotación impuesta por la flexión del extremo, no guarda el contacto sobre toda la parte obturadora.

La forma de la rosca del elemento macho 1 es la misma que la que se mencionó anteriormente para el elemento hembra 2. Ventajosamente, la rosca tiene un perfil perfecto.

En la otra modalidad de conformidad con la presente invención, y mostrada en las figuras 4 y 5, un extremo de una o dos de ambas porciones roscadas 6, 7 del elemento macho, por ejemplo el área 6′ cerca del saliente 5′, puede tener una rosca con un perfil imperfecto. La parte correspondiente 8 del elemento hembra sobre el lado del saliente 5″ que queda frente al área 6″ tiene una rosca perfecta. El área 8′ en el extremo de la porción roscada 8 opuesta axialmente al área 6′, es decir la que se encuentra cerca a la superficie obturadora 12″, tiene también una rosca con un perfil imperfecto. Sin embargo, esta área puede estar provista también de una rosca perfecta.

El mismo tipo de procedimiento puede aplicarse también a las porciones roscadas 7 y 9, en donde la porción 7 esté provista de una rosca perfecta sobre el lado del saliente 5′, y la parte 9 esté provista de un área 9′ que tenga una rosca imperfecta sobre el lado del saliente 5″. En combinación o en forma alternativa a esta modalidad, la parte

7 puede estar provista también de un área 7' que tenga una rosca imperfecta sobre el lado de la superficie obturadora 13' y la parte 9 puede estar provista de una rosca perfecta sobre el lado de la superficie obturadora 13".

A partir de lo mencionado anteriormente está claro que las ventajas de la junta de conformidad con la presente invención, que, aunque tienen dimensiones diametrales diferentes a las del tipo "semi- alojado" de la técnica anterior, y únicamente ligeramente mayores que las del tipo "alojado", asegura un funcionamiento y eficiencia de trabajo óptimos.

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REIVINDICACIONES

1. La junta roscada integral para tubos, que comprende un elemento macho (1) provisto sobre su superficie externa de dos porciones roscadas en forma de cono truncado, radialmente separadas (6, 7), las dos porciones (6, 7) están divididas por un primer saliente anular (5′) que queda sobre un plano ortogonal con respecto al eje del tubo, y un elemento hembra (2) provisto sobre su superficie interna de dos porciones roscadas con forma de cono truncado, radialmente separadas (8, 9), las dos porciones están divididas por un segundo saliente anular (5″) que queda sobre un plano ortogonal al eje del tubo, cada uno de los elementos macho (1) y hembra (2) está provisto de dos superficies obturadoras (12′, 13′, 12″, 13″) colocadas respectivamente en cada extremo axial las cuales están opuestas a las porciones roscadas (6, 7, 8, 9) correspondientes, las dos porciones roscadas correspondientes (6, 7, 8, 9) están adaptadas para enroscarse en forma mutua y reversible, una dentro de la otra, a fin de producir un contacto entre los dos salientes anulares (5′, 5″), caracterizada porque las dos porciones roscadas respectivamente (6, 7, 8, 9) de cada elemento macho (1) y elemento hembra (2) que tiene el mismo valor de conicidad y en que una primera de las dos superficies obturadoras respectivas (12′, 13′, 12″, 13″) del elemento

macho y elemento hembra tiene una forma cónica y una segunda tiene una forma esférica.

2. La junta de conformidad con la reivindicación 1, caracterizada porque al menos uno de elemento macho (1) y elemento hembra (2) comprende, a lo largo de toda su superficie, la cavidad circunferencial (14, 15), colocada entre una de las dos porciones roscadas y el saliente anular correspondiente (5', 5"), la cavidad (14, 15) está adaptada para recibir la expansión de la grasa lubricante de la junta.

3. La junta de conformidad con la reivindicación 2, caracterizada porque cada uno de elemento macho (1) y elemento hembra (2) comprende, a lo largo de toda su superficie, la cavidad circunferencial (14, 15), colocada entre una de las dos porciones roscadas y el saliente anular correspondiente (5', 5").

4. La junta de conformidad con la reivindicación 3, caracterizada porque la dimensión (D') del diámetro externo (3) del elemento hembra (2) en la junta, es mayor que la dimensión (D") del diámetro externo (4) del mismo elemento hembra (2) de un valor igual o menor que 3 %.

5. La junta de conformidad con la reivindicación 4, caracterizada porque las dos porciones roscadas correspondientes (6, 7, 8, 9) de los elementos macho (1) y hembra (2) tienen, cada una, el mismo valor de

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conicidad, comprendido entre 6.25 y 12.5 %.

6. La junta de conformidad con la reivindicación 5, caracterizada porque las superficies obturadoras, cónicas, correspondientes (13', 12") de los elementos macho (1) y hembra (2) tienen un valor de conicidad comprendido entre 12.5 % y 25 %.

7. La junta de conformidad con la reivindicación 6, caracterizada porque el perfil del diente de la rosca tiene un flanco de carga (10) con un ángulo (α) negativo y un flanco de conexión (11) con un ángulo (β) positivo.

8. La junta de conformidad con la reivindicación 7, caracterizada porque el ángulo (α) negativo tiene un valor comprendido entre 0 y −10° y el ángulo (β) positivo tiene un valor comprendido entre 10 y 30°.

9. La junta de conformidad con la reivindicación 8, caracterizada porque las dos porciones roscadas correspondientes (6, 7, 8, 9) de cada uno de los elementos macho (1) y hembra (2) está provista de una rosca perfecta en toda su longitud.

10. La junta de conformidad con la reivindicación 8, caracterizada porque al menos una de las dos porciones roscadas (6, 7, 8, 9) de al menos uno de los elementos macho (1) o hembra (2) está provista de un primer extremo (6', 7') con una rosca imperfecta.

11. La junta de conformidad con la reivindicación 10, caracterizada porque la porción roscada correspondiente, del otro de al menos uno de los elementos macho (1) y hembra (2) está provista también de una segunda área (8', 9') que tiene una rosca imperfecta colocada en el extremo axialmente opuesto con respecto al primer extremo.

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RESUMEN DE LA INVENCIÓN

La presente invención se refiere a una junta roscada integral para tubos, formada por un elemento macho (1) y un elemento hembra (2) colocados respectivamente sobre la superficie externa, y sobre la superficie interna, de dos partes con forma de cono truncado, de la rosca (6, 7, 8, 9), que tienen el mismo valor de conicidad y que están separadas radialmente, ambas divididas respectivamente por salientes anulares (5, 5') que son paralelos y ortogonales con respecto al eje del tubo. Están provistos de dos superficies obturadoras (12, 12', 13, 13'), una de los cuales tiene una forma cónica y la otra una forma esférica. Después de enroscar los dos elementos (1, 2) los dos salientes anulares (5, 5') quedan en contacto. Cada elemento macho (1) y hembra (2) comprende, respectivamente a lo largo de su superficie externa e interna, una cavidad circunferencial (14, 15) colocada entre una parte de la rosca y el saliente anular (5, 5') a fin de permitir la expansión de la grasa lubricante.

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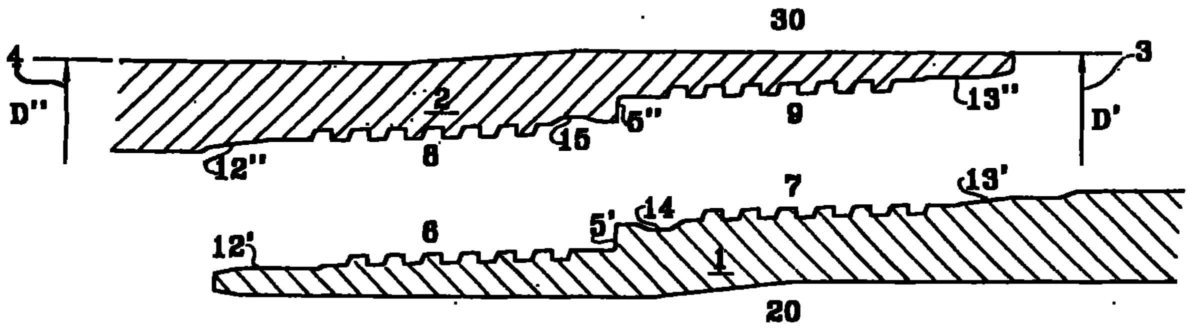

FIG. 1

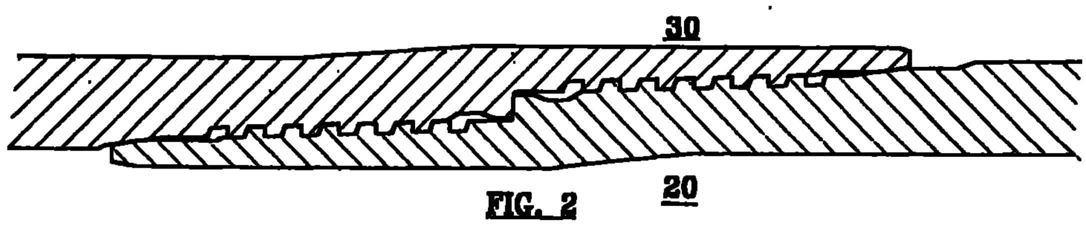

FIG. 2

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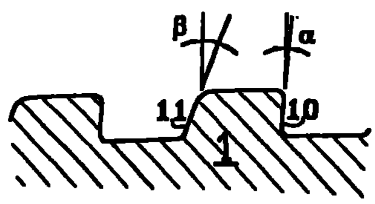

FIG. 3

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91.

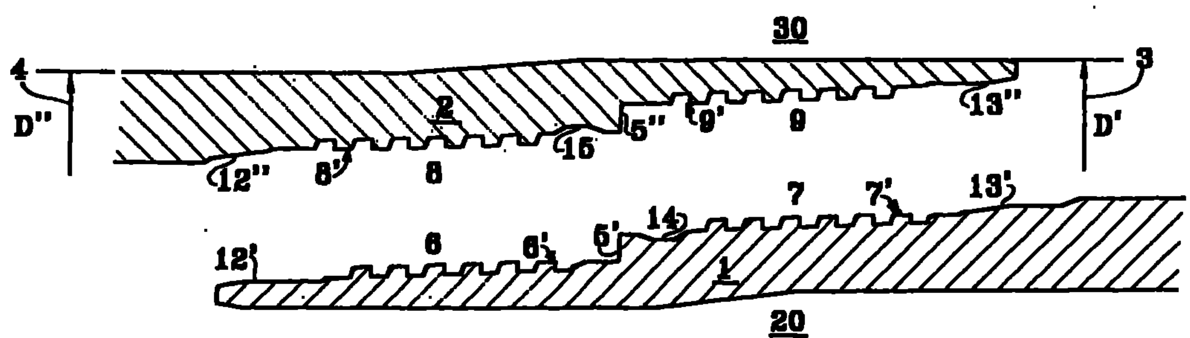

FIG. 4

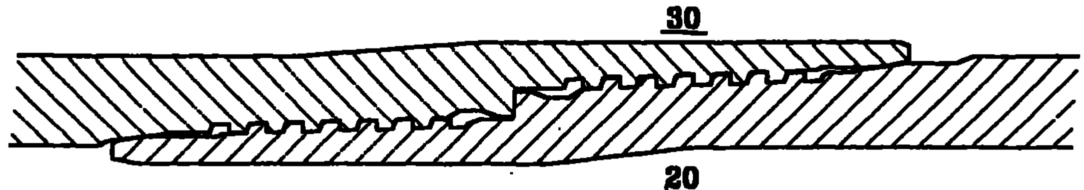

FIG. 5

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REIVINDICACIONES

1. La junta roscada integral para tubos, que comprende un elemento macho (1) provisto sobre su superficie externa de dos porciones roscadas en forma de cono truncado, radialmente separadas (6, 7) con diente de perfil trapezoidal, las dos porciones (6, 7) están divididas por un primer saliente anular (5') que queda sobre un plano ortogonal con respecto al eje del tubo, y un elemento hembra (2) provisto sobre su superficie interna de dos porciones roscadas con forma de cono truncado, radialmente separadas (8, 9), las dos porciones están divididas por un segundo saliente anular (5") que queda sobre un plano ortogonal al eje del tubo, las dos porciones roscadas (6, 7, 8, 9) tienen el mismo valor de conicidad y están adaptadas para enroscarse en forma mutua y reversible, una dentro de la otra, a fin de producir un contacto entre los dos salientes anulares (5', 5") en una posición ensamblada, cada uno de los elementos macho (1) y hembra (2) está provisto de dos superficies obturadoras (12', 13', 12", 13") el primer saliente anular (5') y el segundo saliente anular (5") tienen la forma de un ánulo y quedan sobre un plano en su totalidad, sin presentar elevación alguna desde el plano, y las superficies obturadoras (12', 13', 12", 13") están colocadas respectivamente en cada extremo axial de las porciones

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roscadas (6, 7, 8, 9) de cada elemento macho (1) y elemento hembra (2), caracterizada porque una primera de las dos superficies obturadoras respectivas (12′, 13′, 12″, 13″) tiene una forma cónica, y una segunda de esas superficies tiene una forma esférica, y las superficies del primer y segundo salientes anulares (5′, 5″) son iguales o mayores al 25 % del área de la sección de las paredes del tubo, y al menos uno de los elemento macho (1) y hembra (2) comprende, a lo largo de toda su superficie, una cavidad circunferencial (14, 15) entre una de las dos partes roscadas y el saliente anular correspondiente (5′, 5″), la al menos una cavidad (14, 15) está adaptada para recibir la expansión de la grasa lubricante de la junta.

2. La junta de conformidad con la reivindicación 1, caracterizada porque ambos elementos macho (1) y hembra (2) comprenden, a lo largo de toda su superficie, la cavidad circunferencial (14, 15), colocada entre una de las dos porciones roscadas y el saliente anular correspondiente (5′, 5″).

3. La junta de conformidad con la reivindicación 2, caracterizada porque la dimensión (D′) del diámetro externo (3) del elemento hembra (2) en la junta, es mayor que la dimensión (D″) del diámetro externo (4) del mismo elemento hembra (2) de un valor igual o menor que 3 %.

4. La junta de conformidad con la

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reivindicación 3, caracterizada porque las dos porciones roscadas correspondientes (6, 7, 8, 9) de los elementos macho (1) y hembra (2) tienen, cada una, el mismo valor de conicidad, comprendido entre 6.25 y 12.5 %.

5. La junta de conformidad con la reivindicación 4, caracterizada porque las superficies obturadoras, cónicas, correspondientes (13', 12") de los elementos macho (1) y hembra (2) tienen un valor de conicidad comprendido entre 12.5 % y 25 %.

6. La junta de conformidad con la reivindicación 5, caracterizada porque el perfil del diente de la rosca tiene un flanco de carga (10) con un ángulo (α) negativo y un flanco de conexión (11) con un ángulo (β) positivo.

7. La junta de conformidad con la reivindicación 6, caracterizada porque el ángulo (α) negativo tiene un valor comprendido entre 0 y -10° y el ángulo (β) positivo tiene un valor comprendido entre 10 y 30°.

8. La junta de conformidad con la reivindicación 7, caracterizada porque las dos porciones roscadas correspondientes (6, 7, 8, 9) de cada uno de los elementos macho (1) y hembra (2) está provista de una rosca perfecta en toda su longitud.

9. La junta de conformidad con la reivindicación 7, caracterizada porque al menos una de las

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dos porciones roscadas (6, 7, 8, 9) de al menos uno de los elementos macho (1) o hembra (2) está provista de un primer extremo (6', 7') con una rosca imperfecta.

10. La junta de conformidad con la reivindicación 9, caracterizada porque la porción roscada correspondiente, del otro de al menos uno de los elementos macho (1) y hembra (2) está provista también de una segunda área (8', 9') que tiene una rosca imperfecta colocada en el extremo axialmente opuesto con respecto al primer extremo.

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REQUISITOS NECESARIOS PARA PRESENTACION Y AGILIZACION DEL TRAMITE

		USUARIO	RADICADOR
PATENTE DE INVENCIÓN X	**MODELO DE UTILIDAD**	( )	( )
- Indicación de que se solicita la concesión de una patente.		( )	( )
- Datos de identificación del solicitante o de la persona que se presenta la solicitud.		( )	( )
- Descripción de la invención.		( )	( )
- Dibujos de ser estos pertinentes.		( )	( )
- Comprobante de Pago de las tasas establecidas.			
COMPLETA ( )	**INCOMPLETA**	( )	( )

DISEÑO INDUSTRIAL ( )

	USUARIO	RADICADOR
- Indicación de que se solicita el registro de Diseño Industrial	( )	( )
- Datos de identificación del solicitante o de la persona que presenta - la solicitud	( )	( )
- Representación gráfica y fotográfica del Diseño Industrial o muestra del Material que incorpora el diseño.	( )	( )
Comprobante de pago de las tasas establecidas	( )	( )
COMPLETA ( ) **INCOMPLETA**	( )	( )

ESQUEMA DE TRAZADO ( )

	USUARIO	RADICADOR
- Indicación de que solicita el registro de un Esquema de trazado.	( )	( )
- Datos de identificación del solicitante o de la persona que presenta La solicitud	( )	( )
Representación gráfica del esquema de trazado	( )	( )
Comprobante de pago de las tasas establecidas	( )	( )
COMPLETA ( ) **INCOMPLETA**	( )	( )

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