

12

EUROPEAN PATENT APPLICATION

Application number: **84100427.8**

Int. Cl.³: **F 28 F 1/42**

Date of filing: **17.01.84**

Priority: **25.01.83 US 460784**

Date of publication of application:
01.08.84 Bulletin 84/31

Designated Contracting States:
BE DE FR GB IT NL SE

Applicant: **GULF AND WESTERN INDUSTRIES, INC.**
1, Gulf + Western Plaza
New York, N.Y. 10023(US)

Inventor: **Kienast, David L.**
19 Dixie Acres Road
Danville Illinois, 61832(US)

Representative: **Weitzel, Wolfgang, Dr.-Ing.**
St. Pöltener Strasse 43
D-7920 Heidenheim(DE)

54 **Finned heat exchanger tube having optimized heat transfer characteristics.**

57 A metal heat exchanger tube which provides for optimized heat transfer characteristics and which is particularly adapted for use in the direct expansion shell and tube type evaporators of mechanical refrigeration systems. The metal heat exchanger tube incorporates integral external and internal fins wherein the dimensional and geometrical proportions of the surface or heat transfer areas of the external and internal fins and the cross-sectional flow area of the heat exchanger tube, in conjunction with the lead angle of the internal helical fins have been correlated in conformance with predetermined mathematical criteria in order to optimize the heat transfer capacities of the tubes, particularly when the tubes are to be employed in direct expansion evaporator of mechanical refrigeration systems.

EP 0 114 640 A2

1 FINNED HEAT EXCHANGER TUBE HAVING
 OPTIMIZED HEAT TRANSFER CHARACTERISTICS

BACKGROUND OF THE INVENTION

1. Field of the Invention

5 The present invention relates to a metal heat
exchanger tube which provides for optimized heat transfer
characteristics and, more particularly, relates to an
integrally finned metal heat exchanger tube which is
particularly adapted for use in the direct expansion shell
10 and tube evaporators of mechanical refrigeration systems.

Heat exchanger elements, such as metal tubes which
are employed for heat transfer purposes and which may
constitute components of direct expansion shell and tube
evaporators for mechanical refrigeration systems, are well
15 known in the art; particularly in configurations wherein the
tubes are plain, in essence, are unfinned and have
essentially smooth bores. Heretofore, in order to improve
upon the heat transfer properties of such metal heat
exchanger tubes, the tubes have, in general, been provided
20 with a plurality of integral internal fins transverse of the
length of the tubes in a parallel spaced or helical pattern,
thereby increasing the internal heat transfer surface area of
the tubes and improving the heat transfer capabilities
thereof. Although such internally finned heat exchanger
25 tubes evidence improved heat transfer characteristics in
comparison with plain or unfinned tubes, in essence, tubes
which do not possess any internal fins, the degree of
improvement in heat transfer capability over unfinned tubes
is still insufficient to achieve the potential optimum heat
30 transfer capacity of such heat exchanger tubes.

Consequently, more recently, finned metal heat
exchanger tubes have been developed for this type of

1 refrigeration technology wherein the addition of external
integral fins has been incorporated into the physical
geometries of the heat exchanger tubes for the purpose of
still further enhancing the heat transfer capacities of the
5 tubes. Numerous analytical investigations and actual
physical experiments have been undertaken in the industry
with regard to correlating the dimensions and configurations
of the heat exchanger tubes and those of the integral
external and internal tube fins in order to attempt to
10 optimize, or at least improve upon, the heat transfer
characteristics of such finned heat exchanger tubes. For
this purpose, extensive mathematical formulae have been
developed in the heat exchanger technology, through the
application of which there are derived metal heat transfer
15 tube configurations, particularly for metal heat exchanger
tubes which are adapted to be employed in the direct
expansion shell and tube evaporators of mechanical
refrigeration systems, and wherein the formulae are
predicated upon relatively predictable parameters, such as
20 the operating conditions of the system, type of heat exchange
fluids being conducted within and externally of the heat
exchanger tubes, and upon the actual external and internal
dimensions and configurations of the heat exchanger tube.

— Although considerable efforts have been expended in
25 the technology in attempting to obtain an optimization of
externally and internally finned heat exchanger tubes in
order to achieve improved heat transfer properties, at best,
the results have only been partially successful in achieving
the desired goals.

30 2. Discussion of the Prior Art

Basically, in calculating the geometrical
dimensions and/or physical criteria in the design of

1 externally and internally finned metal heat exchanger tubes
 having potentially optimized heat transfer characteristics,
 particularly tubes which are to be employed in the direct
 expansion shell and tube evaporators of mechanical
 5 refrigeration systems, various operating and physical
 parameters are taken into consideration. These parameters
 may be summarized as follows:

	Ao	=	External tube area per foot of length	(ft ² /ft)
10	Ai	=	Internal tube area per foot of length	(ft ² /ft)
	Aix	=	Internal tube cross sectional flow area	(ft ²)
	φ	=	Severity factor Ref. "Heat Transfer Characteristics of Helical-Corrugated Tubes For Intake Boiling of R-12", Withers, J.G. and Habdas, E.P. Wolverine 15 Tube Div., UOP. Presented at 47th National Meeting of AIChE	
	θ	=	Lead angle of internal integral fin reference longitudinal tube axis	Degrees
	ΔP	=	Refrigerant Pressure Drop	(psi)
20	Q	=	Heat Transfer Rate	(BTU/hr)
	A	=	Total heat exchanger external area	(ft ²)
	MTD	=	Mean effective temperature difference	(°F)
25	U	=	Overall heat transfer coefficient	(BTU/hr ft ² °F)

An extensive discussion of finned metal heat
 30 exchanger tubes of the type disclosed in U.S. Patent No.
 3,826,304 is set forth by James G. Withers and
 Edward P. Habdas in Paper No. 87d presented at the 47th
 National Meeting of the American Institute of Chemical

1 Engineers, New Orleans, LA, March 11-15, 1973, entitled "Heat
Transfer Characteristics of Helical-Corrugated Tubes for
Intube Boiling of Refrigerant R-12". Although the article
describes the intended optimization of internally ridged
5 (finned) heat exchanger tubes, notwithstanding the complex
theoretical calculations involved, no criteria can be
ascertained which would readily lead to or support the
attainment of tube dimensions or geometries providing
optimized performance characteristics in the employment of
10 the tubes in the direct expansion shell and tube evaporators
of mechanical refrigeration systems within the normal
operating ranges of such systems. Consequently, although
various design methods have been developed with respect to
the provision of externally and internally finned heat
15 exchanger tubes, which may present performance improvements
over plain or unfinned tubing for use in direct expansion
evaporators, the prior art heat exchanger tubes and design
methods are not in the optimum range for maximum heat
transfer. Thus, externally and internally finned tubes have
20 been designed for use in direct expansion evaporators in
which the heat transfer capacity of these tubes is limited by
the geometrical relationships of the external and internal
fin surface areas and the internal flow cross-section of the
-- tube, without taking into consideration the lead angle of the
25 internal helical fins and any correlation of these tube
dimensions. Consequently, these tubes are not designed for
operation within the optimum range for maximum heat transfer.

Other design methods for heat exchanger tubes
employed for intube boiling of refrigeration systems are not
30 suitable for optimization of the tube configurations with
respect to maximum heat transfer. Specifically, in these
methods, the so-called severity factor of the tubing is not

1 dependent upon the lead angle of the internal helical fins of
the tubes, whereas extensive investigation pursuant to the
present invention indicate that the heat transfer capacity is
an important function of the lead angle of the internal
5 helical fins of the tubes. Moreover, prior art finned tubes
have severity factors which are outside of the "optimum
range" for the particular inventive application, and
previously described heat exchanger tube design methods are
not applicable to optimization of direct expansion evaporator
10 applications. Moreover, the geometrical relationships of
presently known and commercially available externally and
internally finned tubes, particularly with respect to the
correlation among the surface areas of the external and
internal fins and the flow cross-sectional areas of the
15 tubes, fall outside the optimum range for maximum heat
transfer capacity of the tubes.

Other design data currently employed in the
technology is adapted for tubes having either plain
(unfinned) or slightly knurled outer surfaces, and wherein it
20 can be ascertained that the addition of external fins to the
tubes significantly improves their heat transfer capacities.
Such design methods, in general, do not take into
consideration the geometrical or physical interrelationships
of the heat exchanger tube dimensions and, in many instances,
25 the methods are not adapted for superheating applications,
which is most likely encountered in the operation of direct
expansion evaporators.

Among various currently known finned heat exchanger
tubes, a number of these come into consideration with respect
30 to the inventive concept, although none of the prior art
tubes are designed for or adapted to optimization of the
extent of the heat transfer of the tubes.

1 Thus, Lord et al. U.S. Patent 4,118,944 disclose an
internally finned heat exchanger tube wherein the fin
configuration is selected so as to restrict the temperature
drop of the refrigerant in the tube to within a preselected
5 range as the refrigerant flows therethrough. The dimensions
of the finned tubing disclosed in Lord et al. clearly
indicates, both as to the configuration of the helical
internal fins, and the lack of any external fins which may be
integrally formed with the tube that the heat exchanger tubes
10 disclosed therein would not be suitable for optimization of
the maximum heat transfer range, particularly when the tube
is to be employed in the direct expansion evaporator of a
mechanical refrigeration system.

 Withers Jr., et al. U.S. Patent 3,847,212 disclose
15 an externally finned metal heat transfer tube which includes
helical ridging (finning) on the inner diameter of the tube
so as to provide for improved heat transfer capabilities.
However, review of the calculations and physical dimensions
and geometry of this heat exchanger tube construction clearly
20 evidences that there is no correlation in evidence between
the surface or heat transfer areas of the external and
internal tube fins, the flow cross-sectional area of the heat
exchanger tube and the lead angle of the internal helical
fins which would provide for optimization of the maximum heat
25 transfer capacity of the tube in a manner analogous to that
contemplated by the present invention. In essence, the heat
exchanger tubing disclosed in Withers Jr., et al. does not
provide for optimum maximum heat transfer capability,
particularly when the tubes are to be employed in direct
30 expansion shell and tube evaporators for mechanical
refrigeration systems.

1 Similarly, Thorne U.S. Patent 3,881,382, Rieger
U.S. Patent 3,768,291 and Goodyer U.S. Patent 2,432,308 each
disclose externally and internally finned metal heat
exchanger tubes. However, as in the above-discussed
5 instances, none of these tubes evidence nor suggest geometric
and dimensional interrelationships among the external and
internal fins, the flow cross-sectional area of the tube and
the lead angle of the helical interior fins which would
provide for optimization of the heat transfer capacity of
10 such tubes to thereby render these highly efficient when
employed in direct expansion evaporators, particularly
evaporators utilized for mechanical refrigeration systems.

SUMMARY OF THE INVENTION

 Accordingly, in order to obviate the limitations
15 and drawbacks encountered in metal heat exchanger tubes
designed and constructed pursuant to the prior art, and
particularly heat exchanger tubes which are designed for
utilization in the direct expansion shell and tube
evaporators of mechanical refrigeration systems, pursuant to
20 the present invention the metal heat exchanger tubes
incorporate integral external and internal fins wherein the
dimensional and geometrical proportions of the surface or
heat transfer areas of the external and internal fins and the
- cross-sectional flow area of the heat exchanger tubes, in
25 conjunction with the lead angle of the internal helical fins
have been correlated in conformance with predetermined
mathematical criteria in order to optimize the heat transfer
capacities of the tubes, particularly when the tubes are to
be employed in direct expansion evaporator of mechanical
30 refrigeration systems.

 The inventive heat exchanger tube design and
construction is based on actual experimental test data
gathered from direct expansion coolers in refrigeration

1 systems incorporating various correlated combinations of the
external and internal finned heat exchanger surface areas,
cross-sectional flow areas of the tube, and the lead angle of
the internal fins, which will lead to optimized heat transfer
5 characteristics.

Specifically, the Bo Pierre boiling and ΔP
equations which were published during the 1950's and which
are referred to in the article by James G. Withers and Edward
P. Habdas, Paper No. 87d, entitled "Heat Transfer
10 Characteristics of Helical-Corrugated Tubes for Intube
Boiling of Refrigerant R-12", presented at the 47th National
Meeting of the American Institute of Chemical Engineers, New
Orleans, LA, March 11-15, 1973, have been inventively
modified to account for the lead angle of the internal
15 helical tube fins and the hydraulic diameter of an internally
finned tube. In calculating the optimum physical parameters
for the heat exchanger tube pursuant to the invention, these
modifications have been added to the known general heat
transfer equation $Q = U \times A \times MTD$. A relationship has been
20 found which allows for the coupling of the modified heat
transfer and ΔP equations within the general equation, as
expressed in terms of the physical dimensions of the heat
exchanger tube as set forth hereinabove. This relationship
remains valid for values of the internal cross-sectional flow
25 area of the heat exchanger tube, or the hydraulic diameter,
which are optimal over the normal operating range of direct
expansion evaporators of mechanical refrigeration systems as
currently employed in the industry.

More specifically, optimal interrelationships have
30 been found between the external and internal heat transfer
surface areas of the tube fins, the internal cross-sectional
flow area of the heat exchanger tube, and the lead angle of

1 the internal helical tube fins. Thus, specific optimal
operating ranges have been found, pursuant to the
invention, at lead angles of between about 30 to 60° for the
internal helical tube fins measured relative to the
5 longitudinal axis of the tube, with such geometrical
relationships not at all having been heretofore contemplated
or employed in prior art heat exchanger tube structures.

Accordingly, it is a primary object of the present
invention to provide for a finned metal heat exchanger tube
10 of the type described which optimizes the heat transfer
characteristics due to its physical parameters.

A more specific object of the present invention
resides in the provision of a metal heat exchanger tube
having integral external and internal helical fins wherein
15 the physical dimensions of the external and internal tube
fins, the lead angle of the internal fins, and the cross-
sectional flow area of the tube are correlated with each
other to provide for optimum heat transfer capacities,
particularly when the tube is to be employed in the direct
20 expansion shell and tube evaporator of a mechanical
refrigeration system.

BRIEF DESCRIPTION OF THE DRAWINGS

Reference may now be had to the following detailed
description of a finned metal heat exchanger tube, in which
25 the tube is particularly adapted to provide for optimized
heat transfer characteristics when employed as a component in
direct expansion evaporators for mechanical refrigeration
systems; taken in conjunction with the accompanying drawings;
in which:

30 Figure 1 illustrates a longitudinal view, partly in
section, of an externally and internally finned heat
exchanger tube pursuant to the invention; and

1 Figure 2 is a cross-sectional view taken along line
2-2 in Figure 1.

DETAILED DESCRIPTION

5 Referring now in detail to the drawings, a metal
heat exchanger tube 10 having a cylindrical wall construction
12 incorporates, integrally formed therewith, external fins
14 and internal fins 16.

10 As illustrated, the external fins 14, which are
integrally formed with the cylindrical tube wall 12, may be
of a generally helical configuration. Similarly, the
internal fins which protrude into the flow passageway 18 of
the heat exchanger tube 10 are also of a helical
configuration.

15 In order to optimize the heat transfer capacity of
the heat exchanger tube, particularly when the tube is to be
employed in a direct expansion evaporator of a mechanical
refrigeration system, extensive experimentation and actual
testing pursuant to the invention has been undertaken in
order to derive an optimum heat exchanger tube design based
20 on the physical and dimensional interrelationship of the
external area A_o of the external fins 14 for each foot of
length of heat exchanger tubing (ft^2/ft), the area of the
internal fins A_i for each foot of tube length (ft^2/ft), the
internal cross-sectional flow area A_{ix} of the heat exchanger
25 tube 10 (ft^2), and the lead angle θ of the internal helical
fins measured relative to the longitudinal axis of the heat
exchanger tube (degrees). Through suitable correlation of
the dimensional interrelationships of these heat exchanger
tube design parameters, extensive experimental test data has
30 indicated that the thermal performance of shell-and-tube type
direct expansion evaporators for mechanical refrigerator
systems can be predicted within predetermined bounds so as to

1 allow for a heat exchanger tube design which considers the
operating conditions of the cooler and provides an optimized
heat transfer performance over the most likely employed range
of operating conditions for such evaporators.

5 Basically, the physical design criteria for the
heat exchanger tube 10 takes into consideration the operating
conditions of the cooler; in effect, wherein

- $\dot{m}$ = refrigerant mass flowrate per tube lbm/hr
- ΔX = refrigerant quality change
- Refrigerant
- 10 - SST = refrigerant saturated exit temperature from the tube.

The design for the heat exchanger tube is adapted
for use when the heat exchanger tubes are utilized to boil
15 and superheat the refrigerant flowing within the tubes
(approximately 8 to 10°F superheat).

In essence, the heat exchanger tube 10, based on
the foregoing operating conditions of a cooler which is
employed in the direct expansion evaporators of mechanical
20 refrigeration systems, employs dimensional parameters in the
design of the heat exchanger tubes, based on each unit of
tube length (L) as measured in feet. These dimensional
parameters are as follows:

The outside heat transfer area A_o (ft²/ft) for the
25 heat exchanger tube 10, which is measured as the total heat
transfer surface A_o of the external fins 14 for each foot of
tube length L.

The internal heat transfer area of the tube 10
which, in effect, is the total surface area A_i (ft²/ft) of
30 the internal fins 16 for each foot of tube length L, the lead
angle θ of the internal fins, in degrees, measured relative
to the longitudinal axis of the heat exchanger tube 10;

1 and the cross-sectional flow area A_{ix} (ft^2) of the
heat exchanger tube 10.

Thus, inventively, the following geometrical
relationships have imparted optimized heat transfer
5 characteristics to the heat exchanger tube 10, when
maintained within the following parameters:

$$4.60 \leq \frac{A_i}{\sqrt{A_{ix}}} \leq 6.20$$

10 $30^\circ \leq \theta \leq 60^\circ$

$$1.5 \leq \frac{A_o}{A_i} \leq 5$$

15 Moreover, the following tube geometries, utilizing
the above dimensional parameters, have been found to be an
optimization over prior art heat exchanger tube designs:

20 $4.25 \leq \frac{A_i}{\sqrt{A_{ix}}} \leq 6.20$

$$40^\circ \leq \theta \leq 50^\circ$$

$$1.5 \leq \frac{A_o}{A_i} \leq 5$$

25

The present invention distinguishes with respect to
prior art heat exchanger tube designs in that the dimensional
proportions of A_o , A_i , A_{ix} , and θ are uniquely employed in a
30 manner which will optimize the heat transfer capacity of the
heat exchanger tube 10, which is of particular significance
when employed in the direct expansion shell and tube
evaporator of a mechanical refrigeration system.

35

1 In effect, an optimal interrelationship has been
found between A_o , A_i , and A_{ix} which will vary with θ .
Moreover, the optimal operating range for each heat exchanger
tube has also been shown to vary with θ . Consequently, set
5 forth herein is the physical and dimensional correlation
between A_o , A_i , A_{ix} and θ which is applicable over the
optimum operating range for each heat exchanger tube; for
example, the optimum operating range for a heat exchanger
tube having $\theta = 45^\circ$ is somewhat different from the optimum
10 operating range for a heat exchanger tube having $\theta = 60^\circ$.
The same relationship between A_o , A_i and A_{ix} which is
applicable for the heat exchanger tube having $\theta = 45^\circ$ has
been found to be applicable for a tube having $\theta = 60^\circ$.

15 In summation, the invention sets forth a novel
geometrical interrelationship for the various dimensional
parameters of a heat exchanger tube which differs from those
commercially available, inventively utilizing a simplified
mathematical computation and design method which is not
contemplated in the prior art.

20 While there has been shown and described what is
considered to be a preferred embodiment of the invention, it
will of course be understood that various modifications and
changes in form or detail could readily be made without
departing from the spirit of the invention. It is therefore
25 intended that the invention be not limited to the exact form
and detail herein shown and described, nor to anything less
than the whole of the invention herein disclosed as
hereinafter claimed.

30

35

1 WHAT IS CLAIMED IS:

1. A metal heat exchanger tube providing for optimized heat transfer characteristics, said metal tube comprising an integral external fin structure; and an
5 integral internal helical fin structure defining a predetermined helix lead angle measured relative to the longitudinal central axis of the tube, the dimensions of the external and internal fin surface areas for each unit of tube length, the internal cross-sectional flow area of said tube,
10 and said internal fin lead angle being geometrically correlated within predetermined parameters to optimize the heat transfer characteristics of said tube.

2. A heat exchanger tube as claimed in claim 1, wherein said external and internal fin area dimensions
15 define, respectively, the external and internal heat transfer area for each unit of tube length.

3. A heat exchanger tube as claimed in claim 2, wherein said tube is geometrically correlated in that the ratio of the heat transfer area of said internal fins
20 relative to the square-root of the internal cross-sectional flow area of said tube is within the range of about 4.60 to 6.20; the ratio of heat transfer area of said external fins relative to the heat transfer area of said internal fins is within the range of about 1.5 to 5.0; and said helix lead
25 angle of the internal fins is within the range of about 30° to 60°.

4. A heat exchanger tube as claimed in claim 3, wherein the ratio of the heat transfer area of said internal fin relative to the square-root of the internal
30 cross-sectional flow area of said tube is within the range of about 4.25 to 6.20; and wherein the helix lead angle of the internal fins of said tube is within the range of about 40° to 50°.

1 5. A heat exchanger tube as claimed in claim 1,
wherein said tube is utilized for boiling and superheating a
refrigerant conducted through said tube.

5 6. A heat exchanger tube as claimed in claim 5,
wherein said refrigerant is superheated to a temperature of
about 8 to 10°F.

7. A heat exchanger tube as claimed in claim 1,
wherein said tube comprises a component of a direct expansion
evaporator.

10 8. A heat exchanger tube as claimed in claim 1,
wherein said external fin comprises a continuous helical fin.

9. A method of forming a metal heat exchanger tube
providing for optimized heat transfer characteristics,
comprising forming an integral external fin on said tube; and
15 forming an integral internal helical fin within said tube
defining a predetermined helix lead angle measured relative
to the longitudinal central axis of said tube, wherein the
dimensions of said external and internal fin surface area for
each unit of tube length, the internal cross-sectional flow
20 area of said tube and of said helix lead angle are
geometrically correlated within predetermined parameters to
thereby optimize the heat transfer capability of said tube.

10. A method as claimed in claim 9, wherein said
external and internal fin dimensions define, respectively,
25 the external and internal heat transfer areas for each unit
of tube length.

11. A method as claimed in claim 10, wherein said
tube is geometrically correlated so that the ratio of the
heat transfer area of said internal fins relative to the
30 square-root of the internal cross-sectional flow area of said
tube is within the range of about 4.60 to 6.20; the ratio of
the heat transfer area of said external fin relative to the

1 heat transfer area of said internal fin is within the range
of about 1.5 to 5.0; and said helix lead angle of the
internal fin is within the range of about 30° to 60°.

5 12. A method as claimed in claim 11, wherein the
ratio of the heat transfer area of said internal fin relative
to the square-root of the internal cross-sectional flow area
of said tube is within the range of about 4.25 to 6.20; and
wherein the helix lead angle of the internal fin of said tube
is within the range of about 40° to 50°.

10 13. A method as claimed in claim 9, wherein said
tube is utilized for boiling and superheating a refrigerant
conducted through said tube.

15 14. A method as claimed in claim 13, wherein said
refrigerant is superheated to a temperature of about 8 to
10°F.

15 15. A method as claimed in claim 9, wherein said
tube comprises a component of a direct expansion evaporator.

16. A method as claimed in claim 9, wherein said
external fin is formed as a continuous helical fin.

20

25

30

35

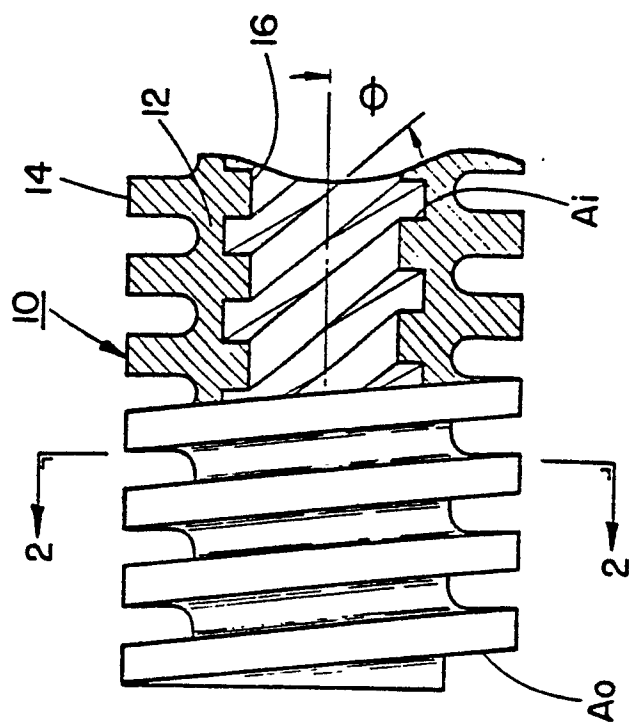


FIG. 1

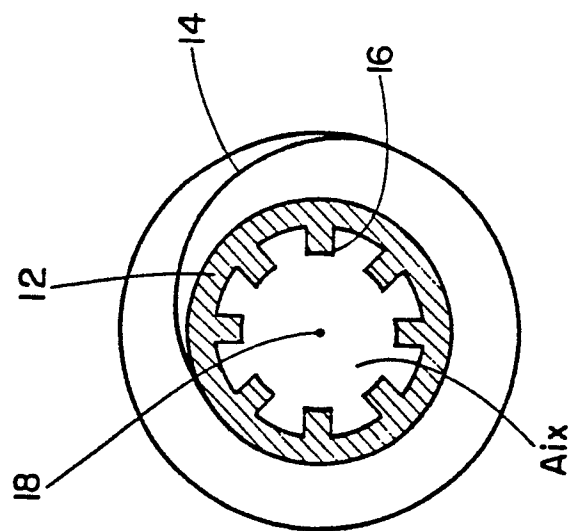


FIG. 2