US 20120312294A1

(19) **United States**

(12) **Patent Application Publication** (10) **Pub. No.: US 2012/0312294 A1**

Grindatto (43) **Pub. Date: Dec. 13, 2012**

(54) **TUBULAR HEAT EXCHANGER, IN PARTICULAR RECEIVING TUBE OF A CONCENTRATING SOLAR PLANT**

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(21) Appl. No.: **13/505,689**

(22) PCT Filed: **Nov. 3, 2010**

(86) PCT No.: **PCT/IT2010/000441**

§ 371 (c)(1), (2), (4) Date: **Jul. 27, 2012**

(30) **Foreign Application Priority Data**

Nov. 3, 2009 (IT) RM2009A000556

Publication Classification

(51) **Int. Cl.**

F28F 1/40 (2006.01)

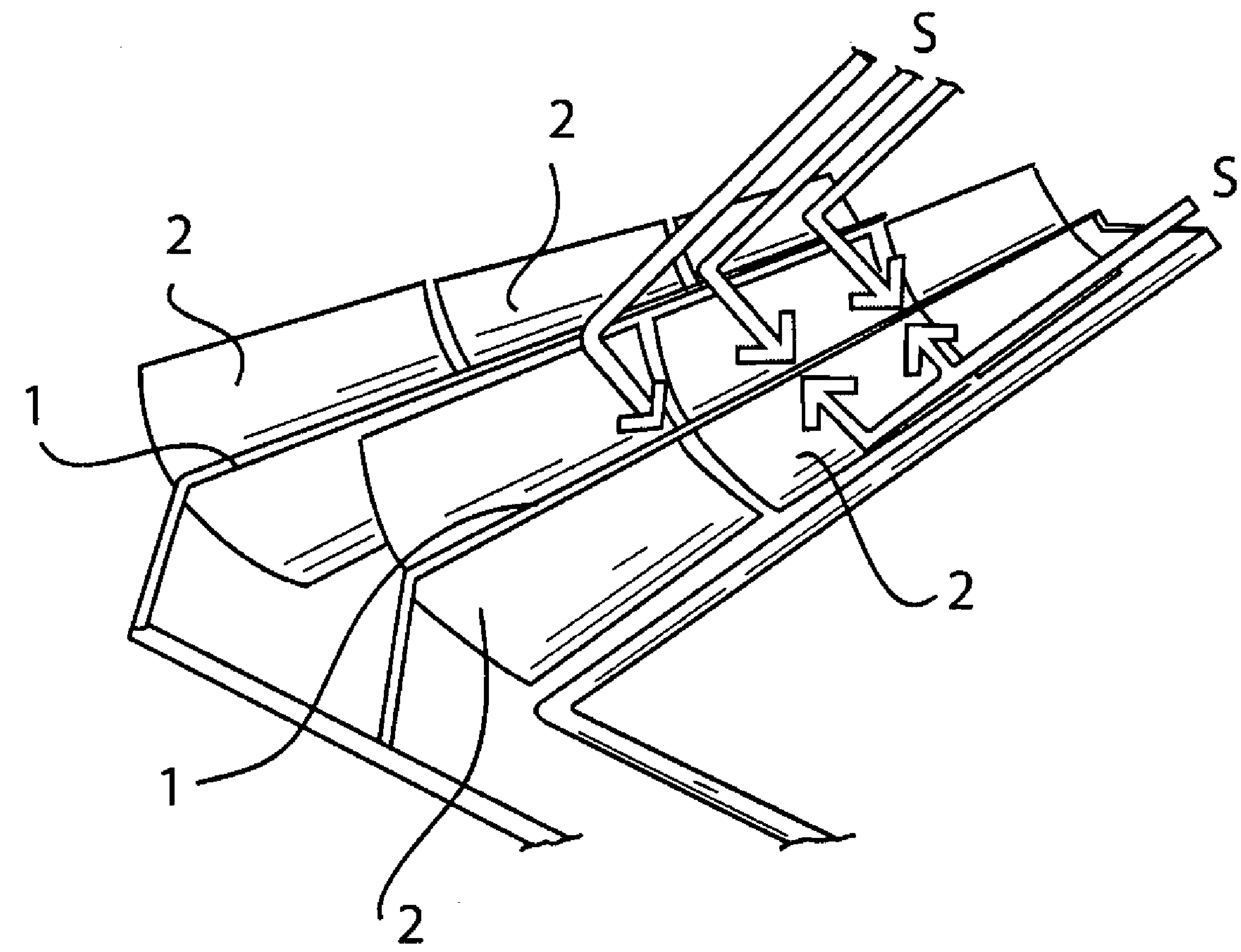
F28F 13/14 (2006.01)

F24J 2/24 (2006.01)

(52) **U.S. Cl.** **126/651**; 165/182

(57) **ABSTRACT**

The present invention concerns a tubular heat exchanger (**10**) between a source of radiant heat located externally with respect to said tubular heat exchanger (**10**) and a heat-carrying fluid flowing inside the exchanger, the radiant heat coming from said source being concentrated on a longitudinal portion of the external surface of said tubular heat exchanger, comprising means (**11**) for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger.



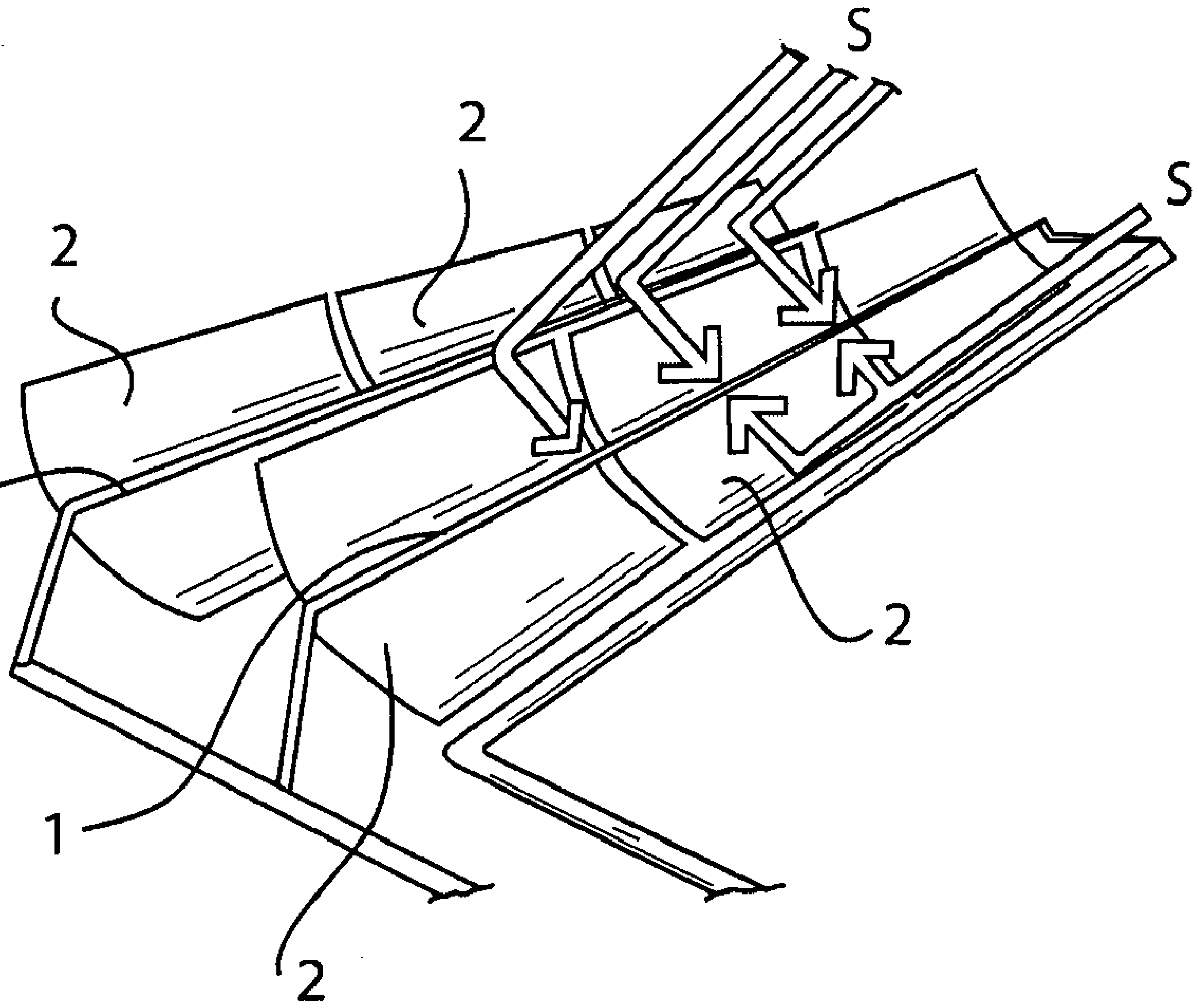


Fig.1

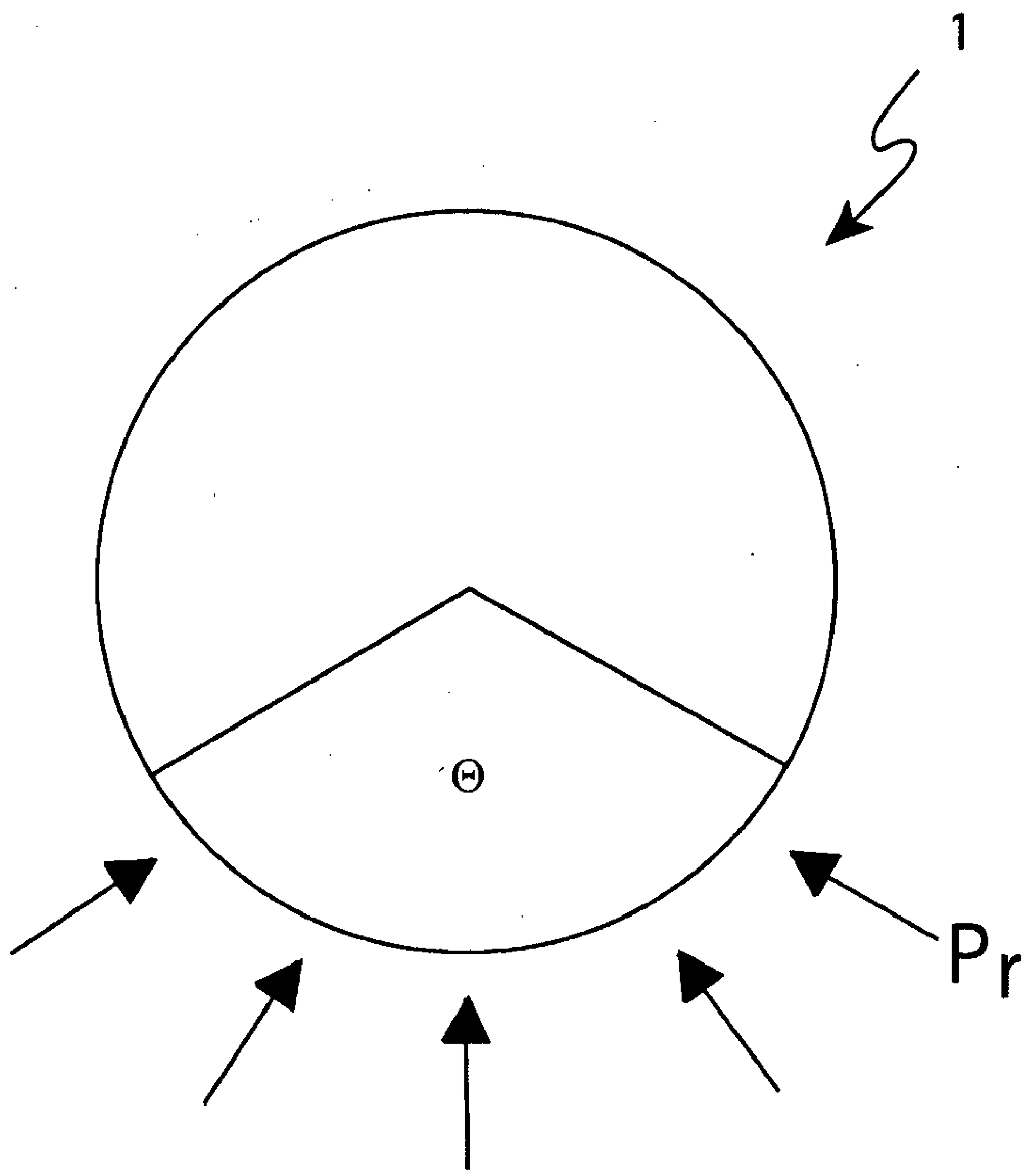


Fig.2

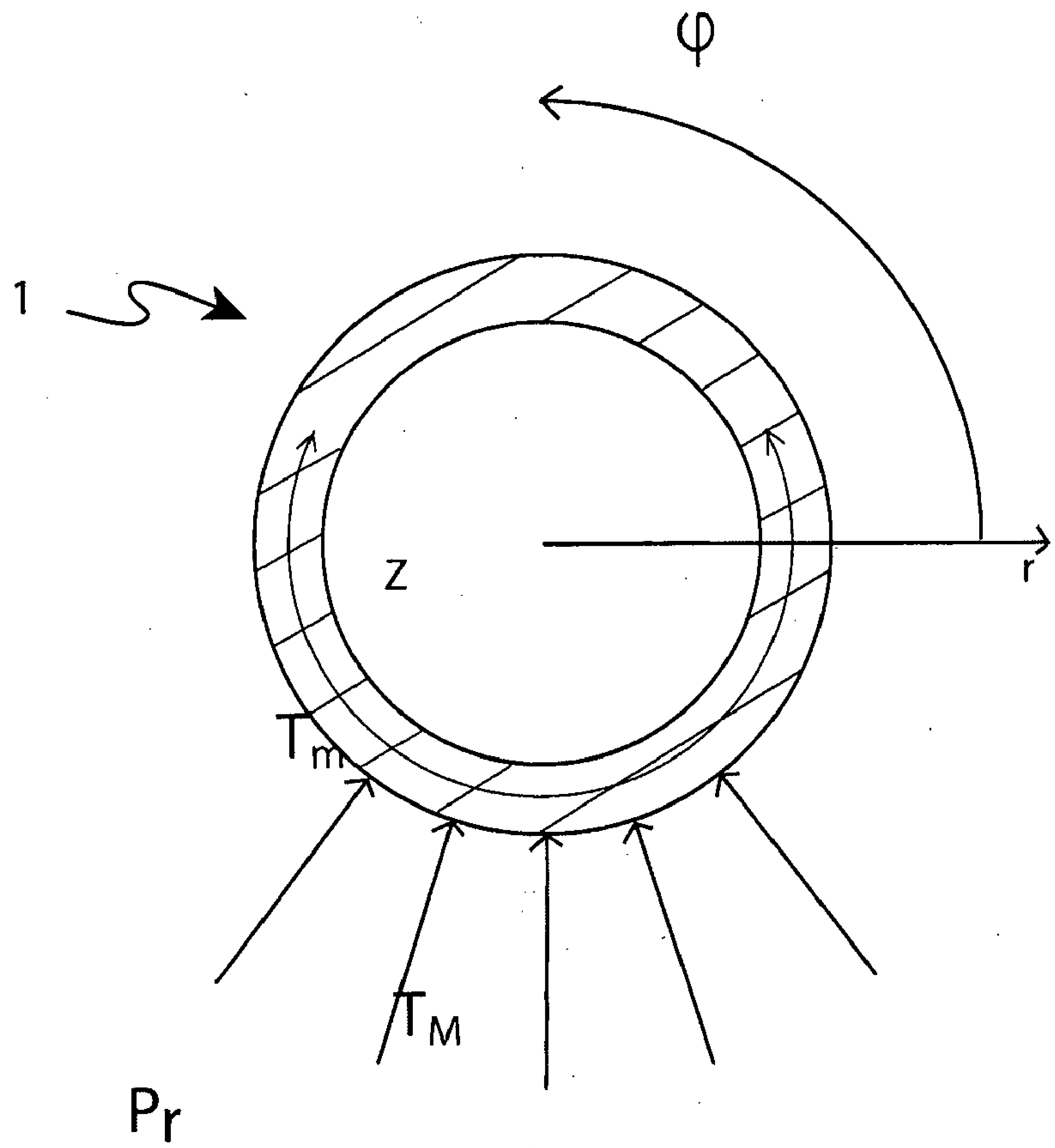


Fig.3

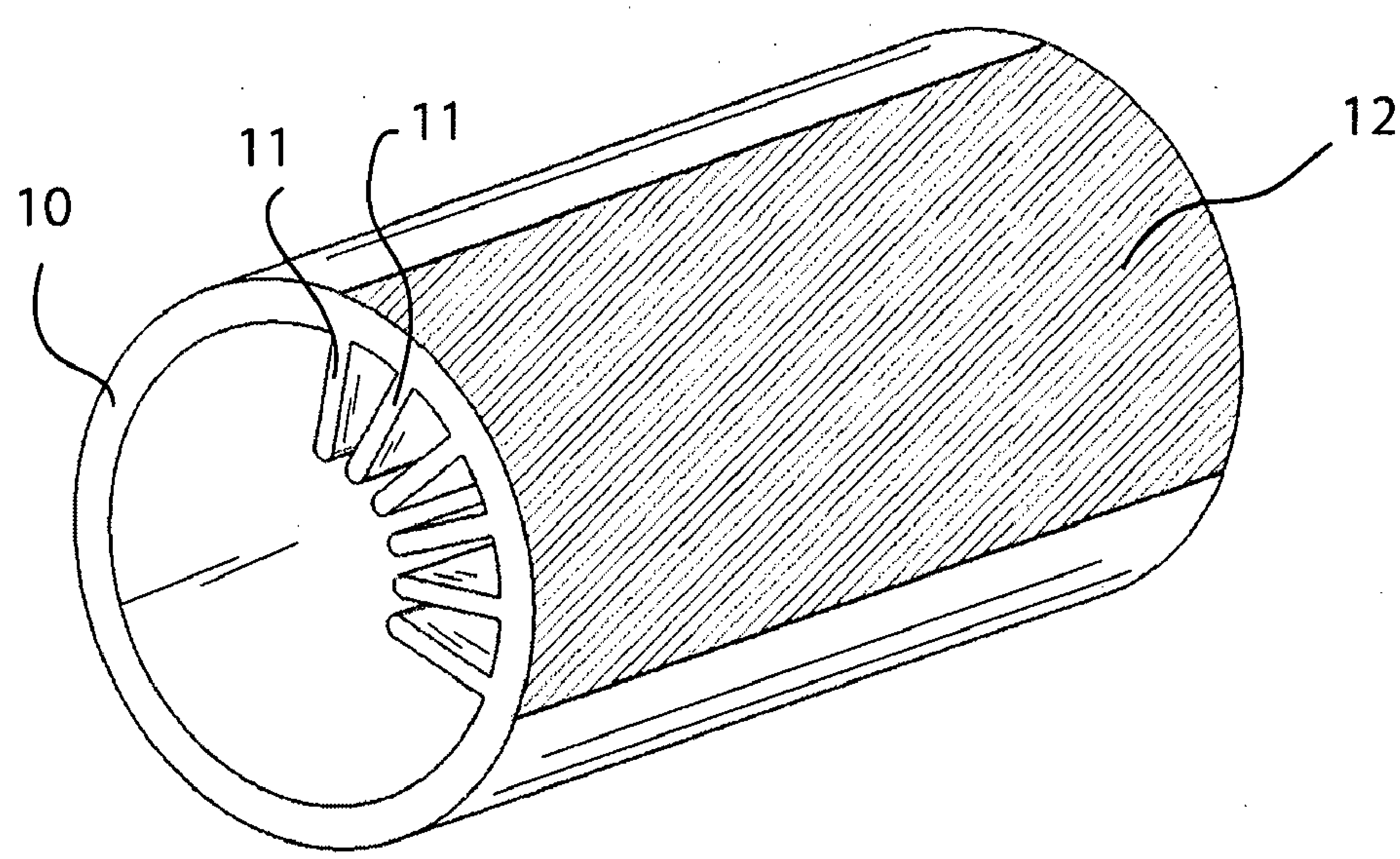


Fig.4

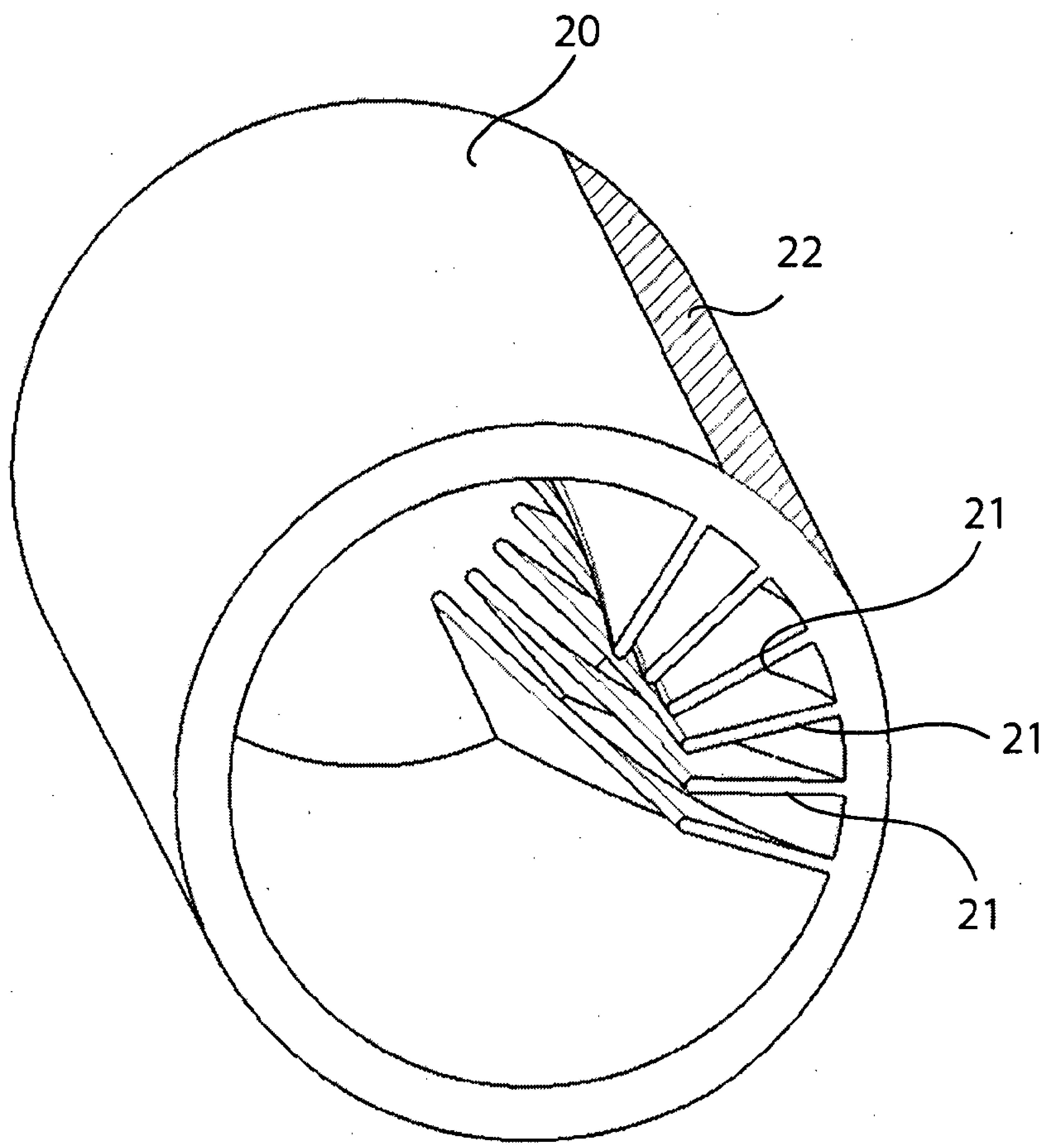


Fig.5

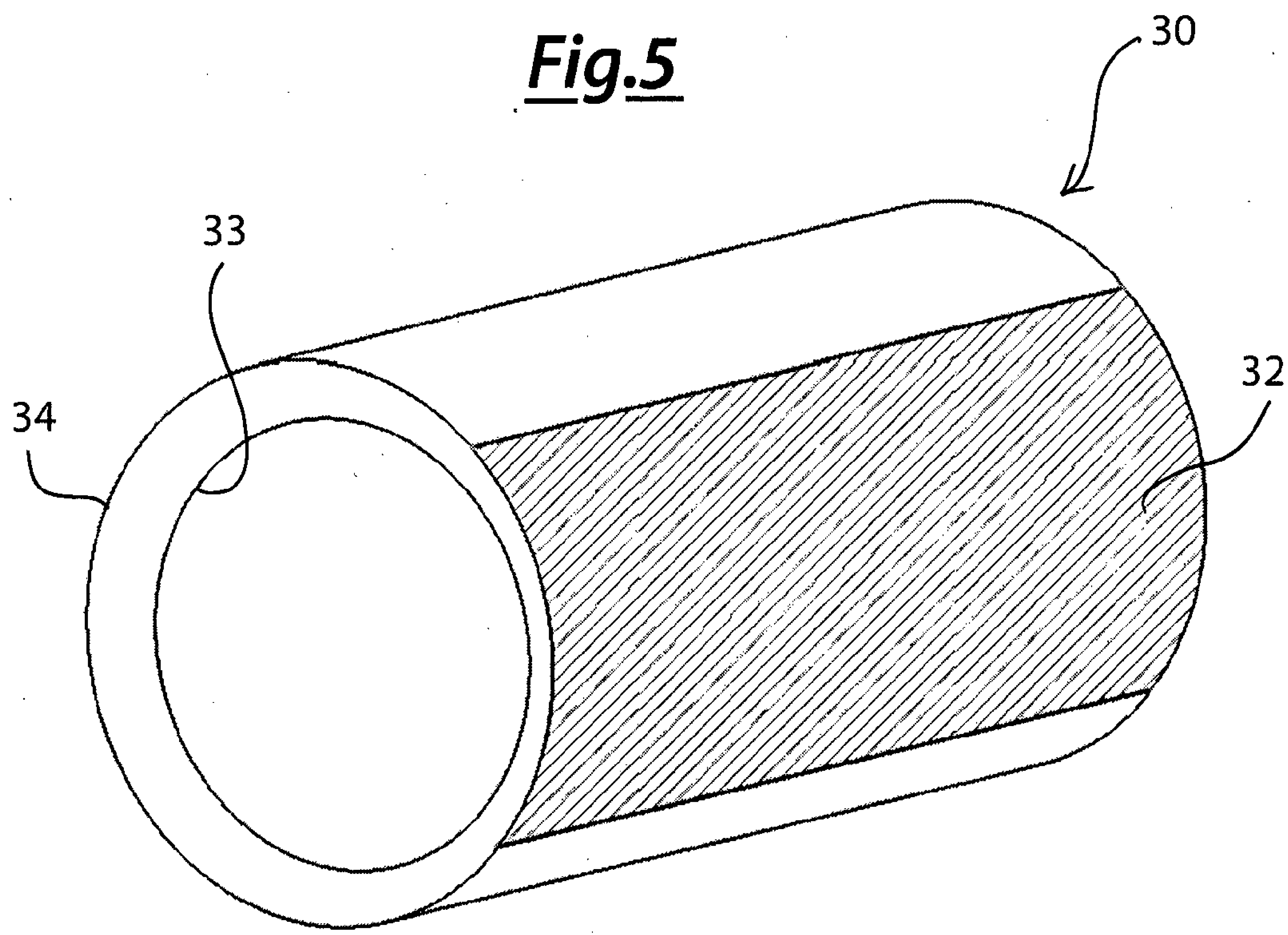


Fig.6

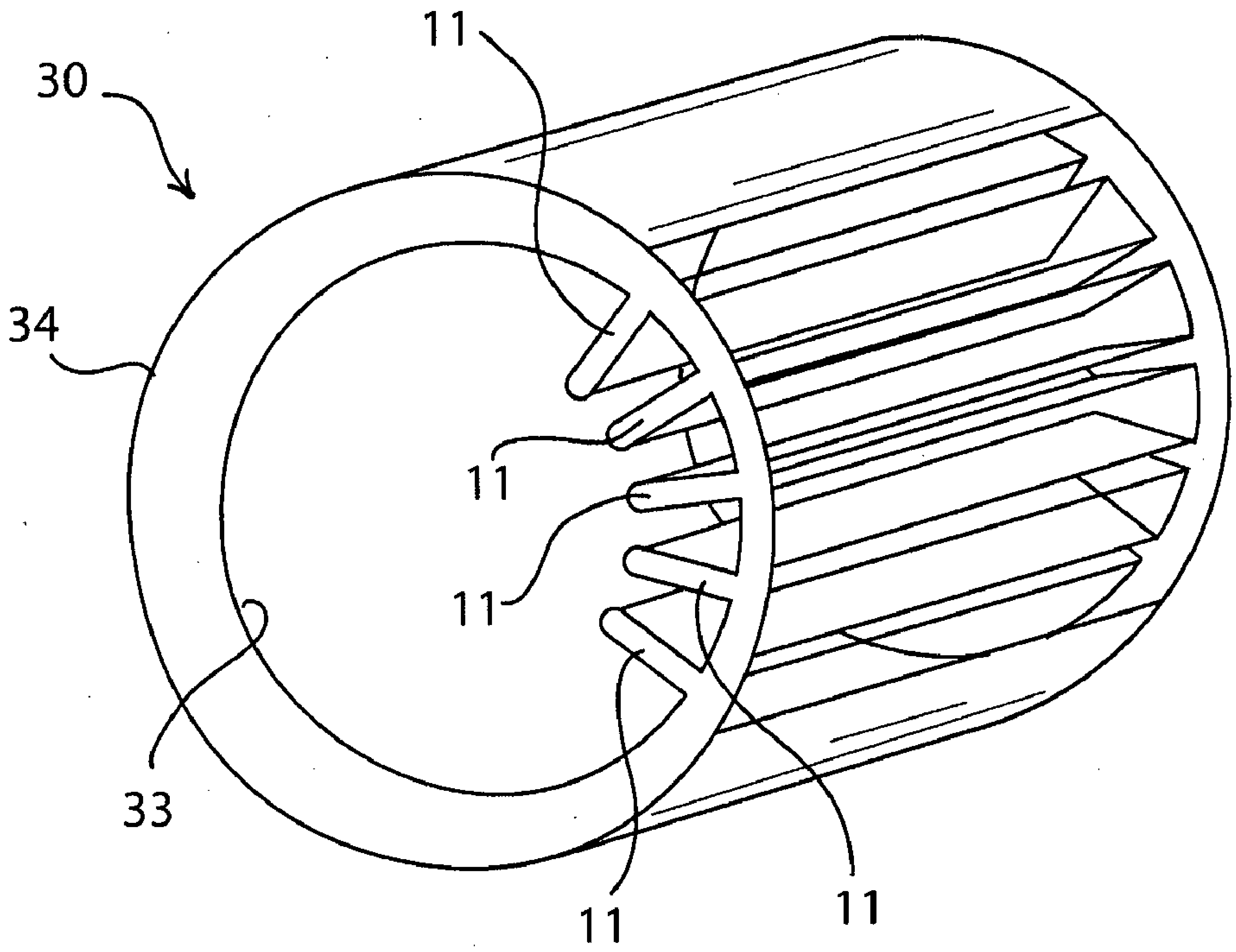


Fig.7

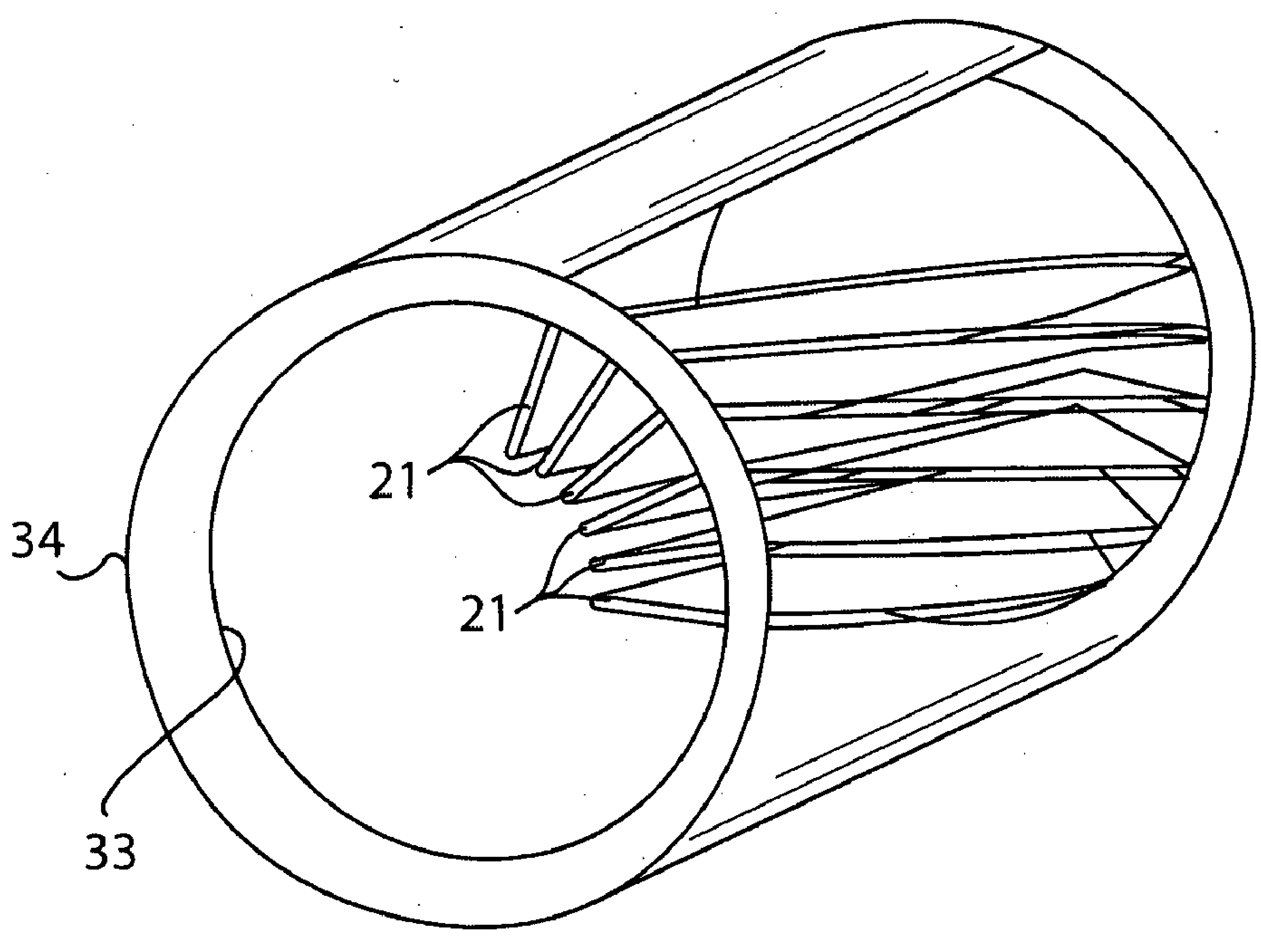


Fig.8

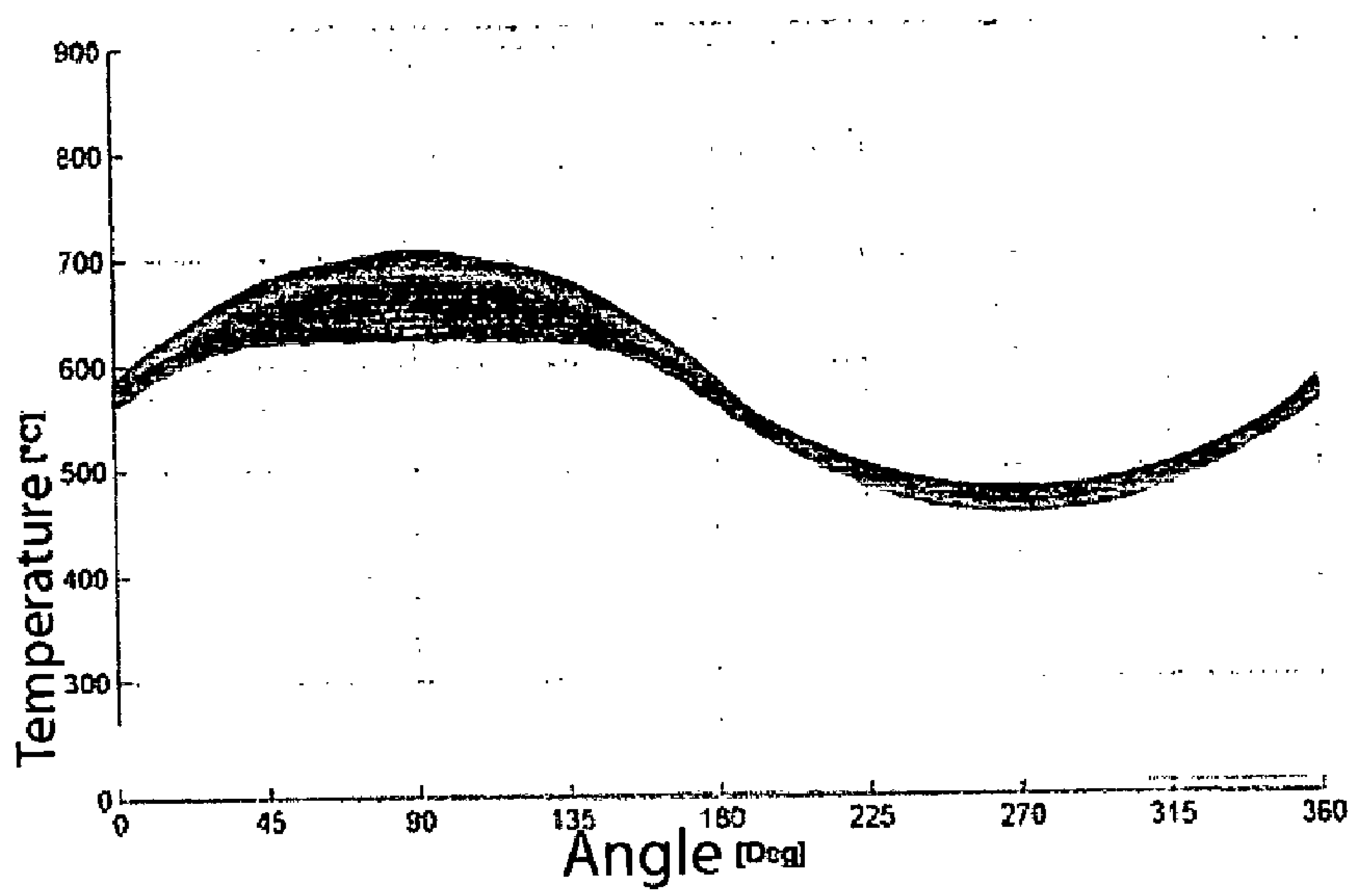

900
800
700
600
500
400
300
0
Temperature [°C]
0
45
90
135
180
225
270
315
360
Angle [Deg]

Fig.9

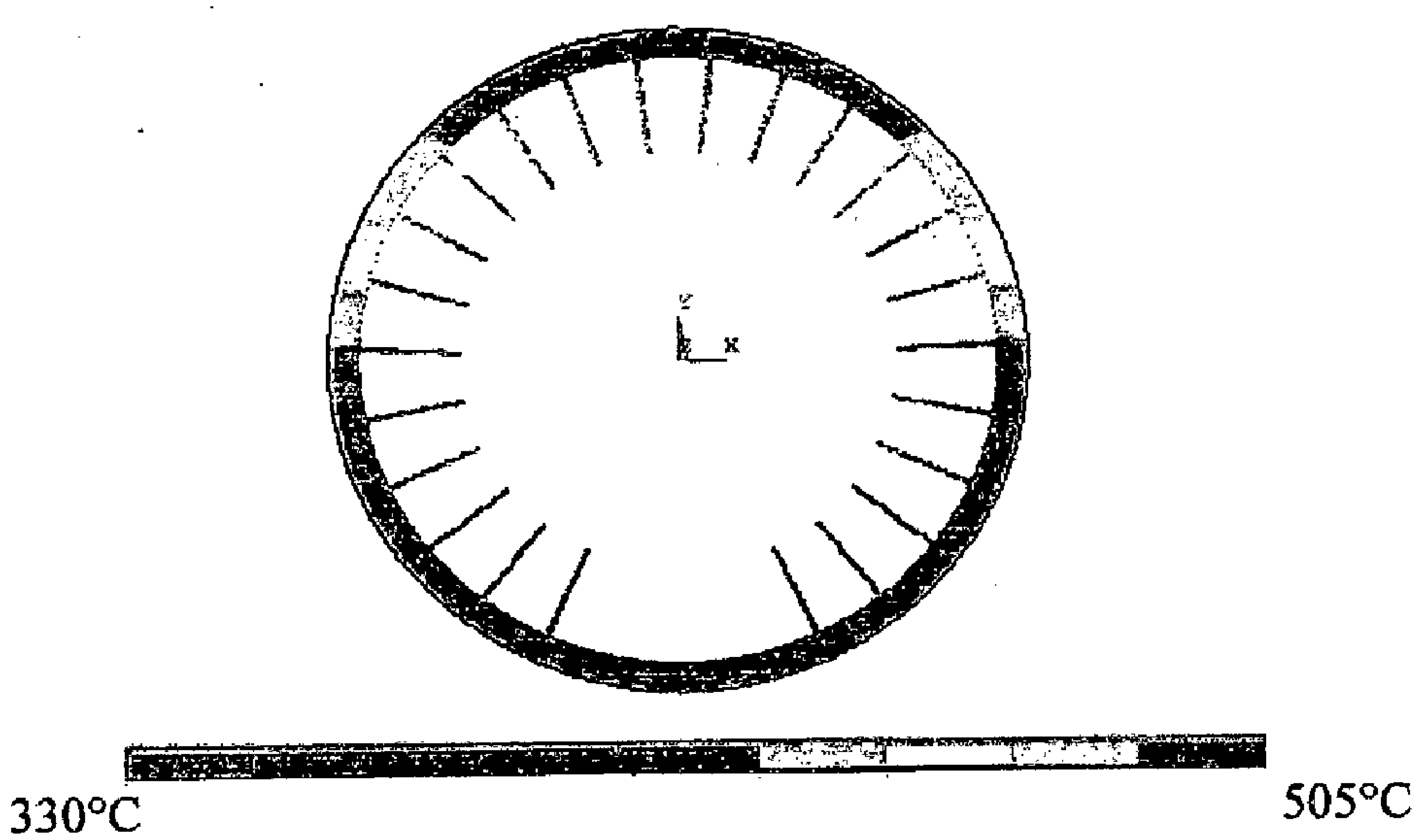

330°C
505°C

Fig.10a

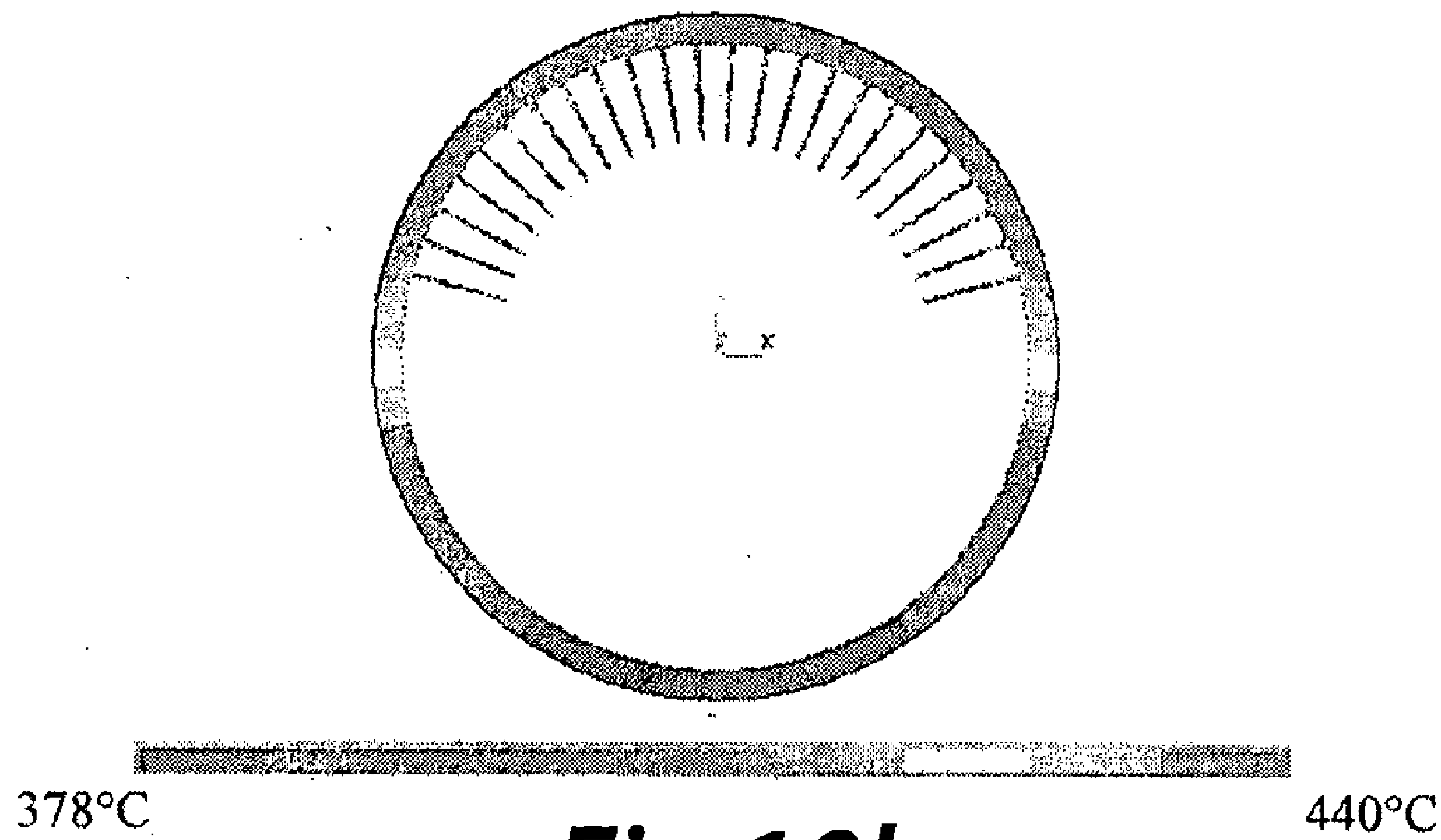


Fig.10b

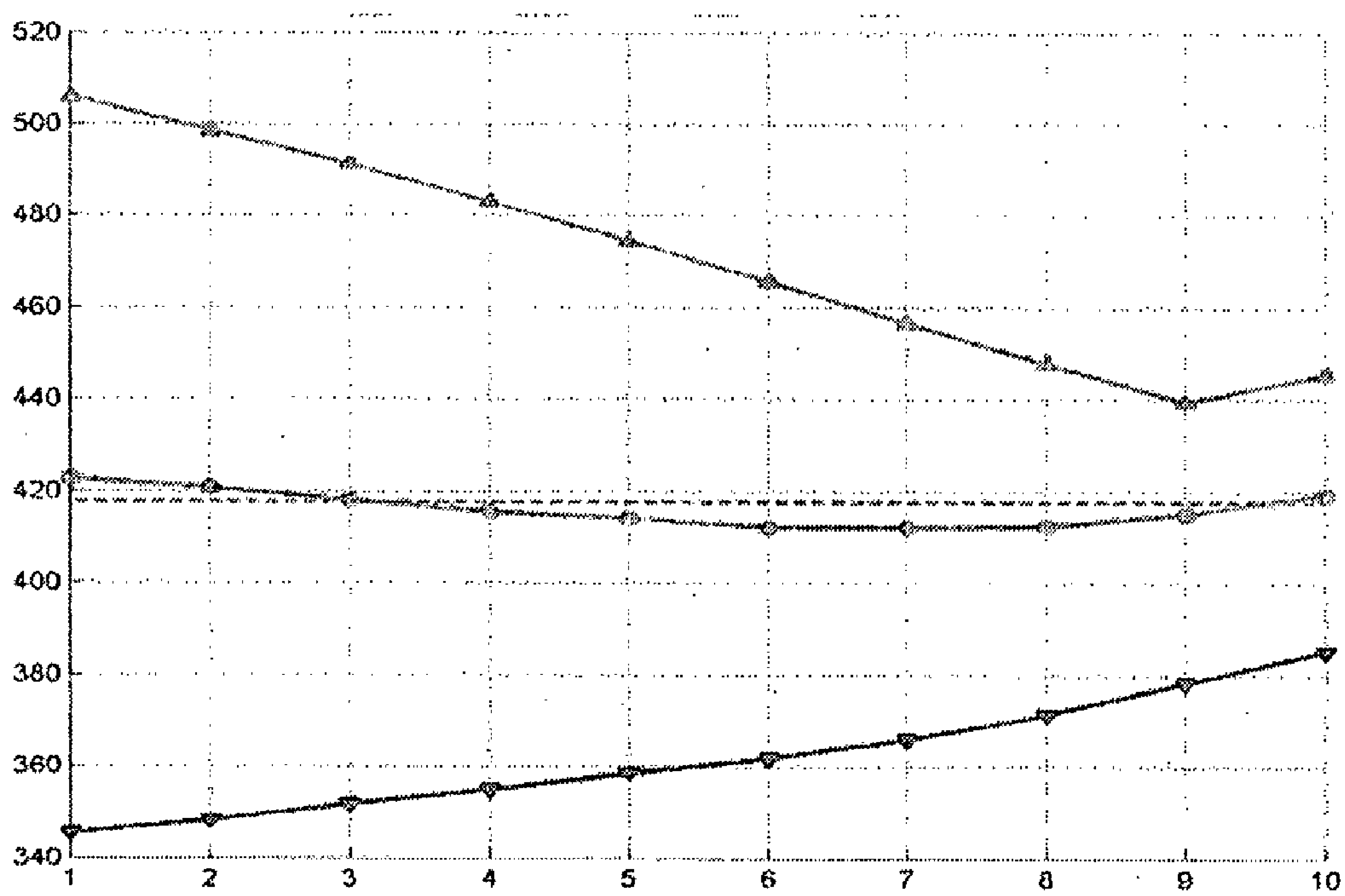


Fig.11

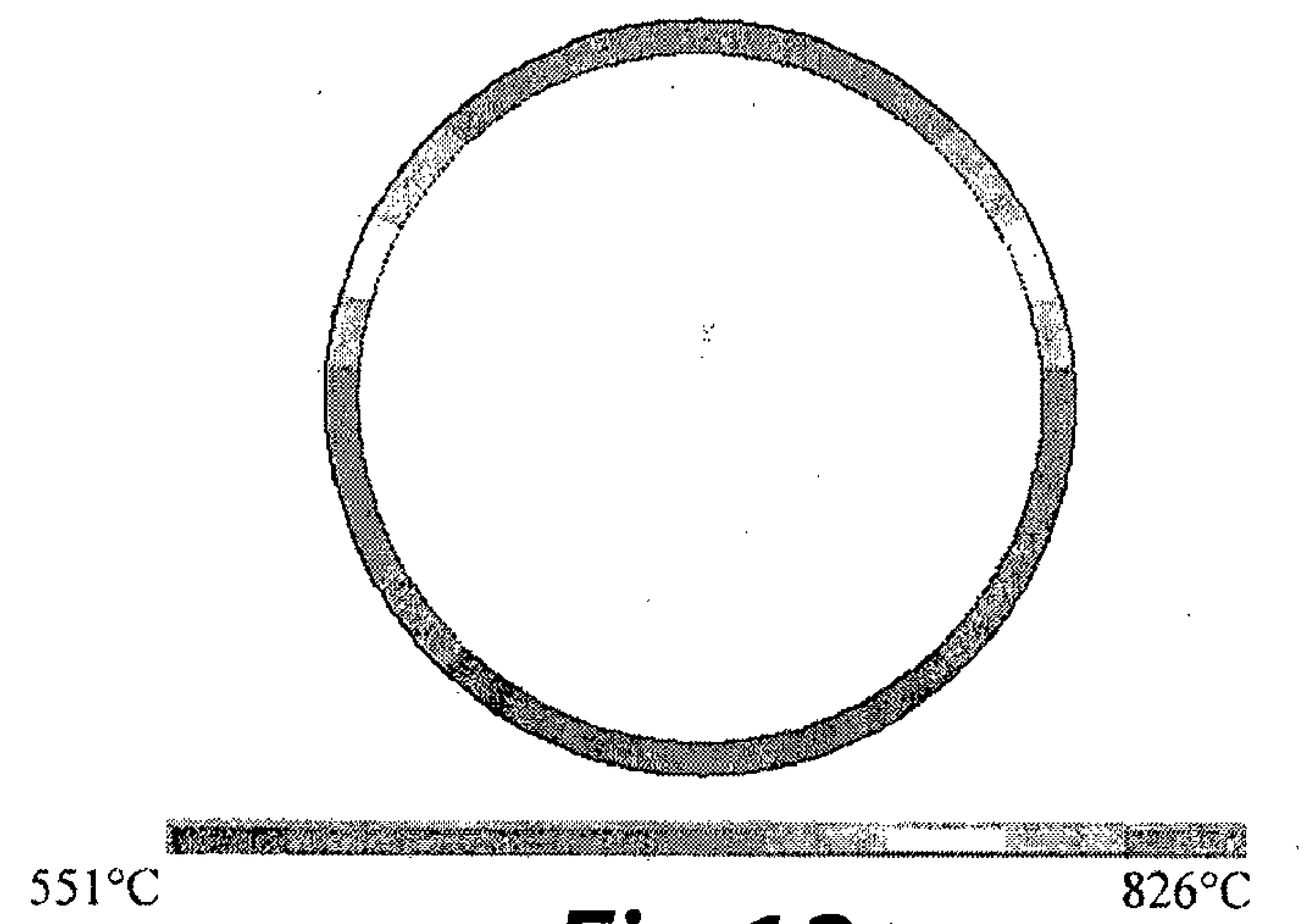


Fig.12a

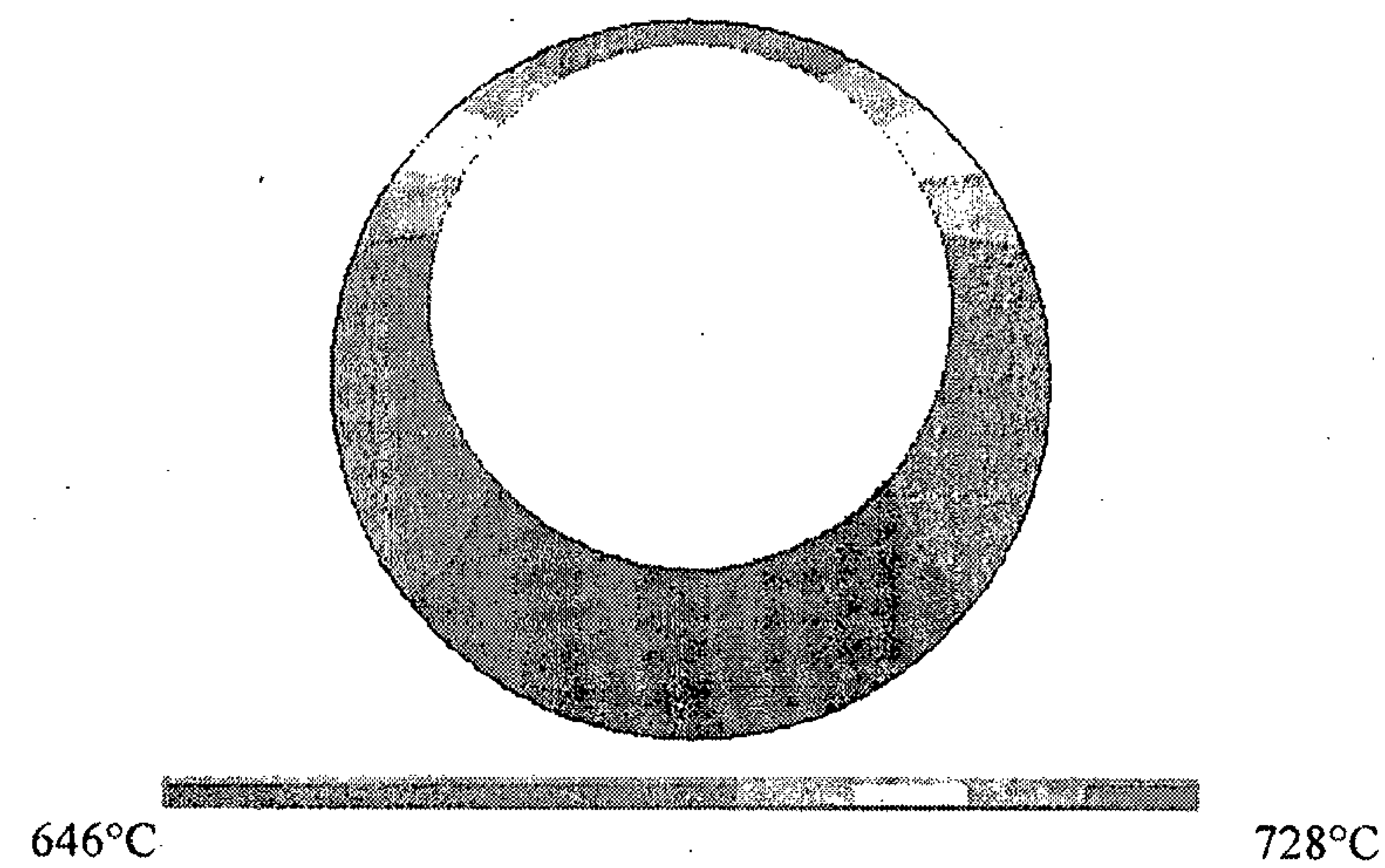


Fig.12b

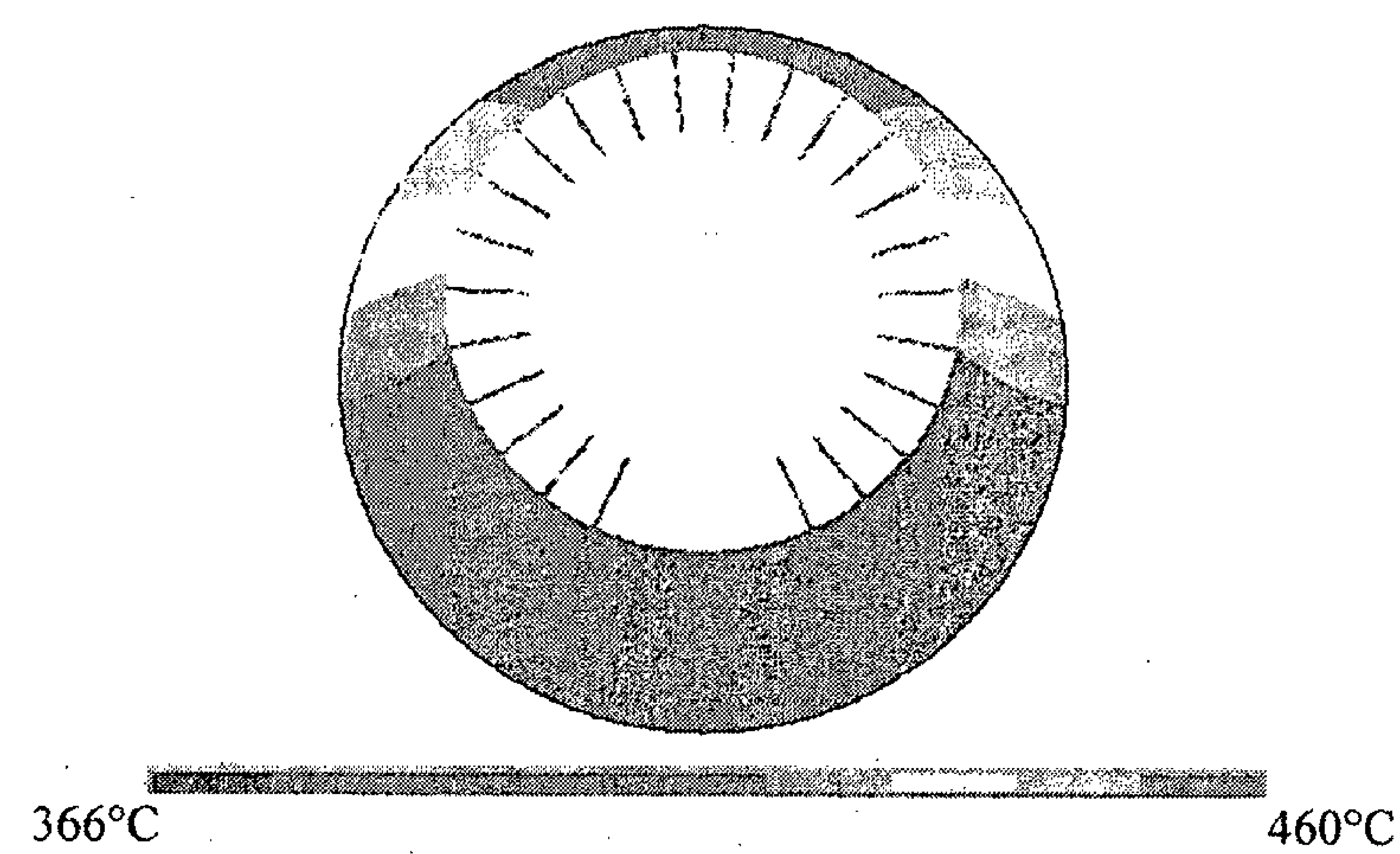


Fig.13a

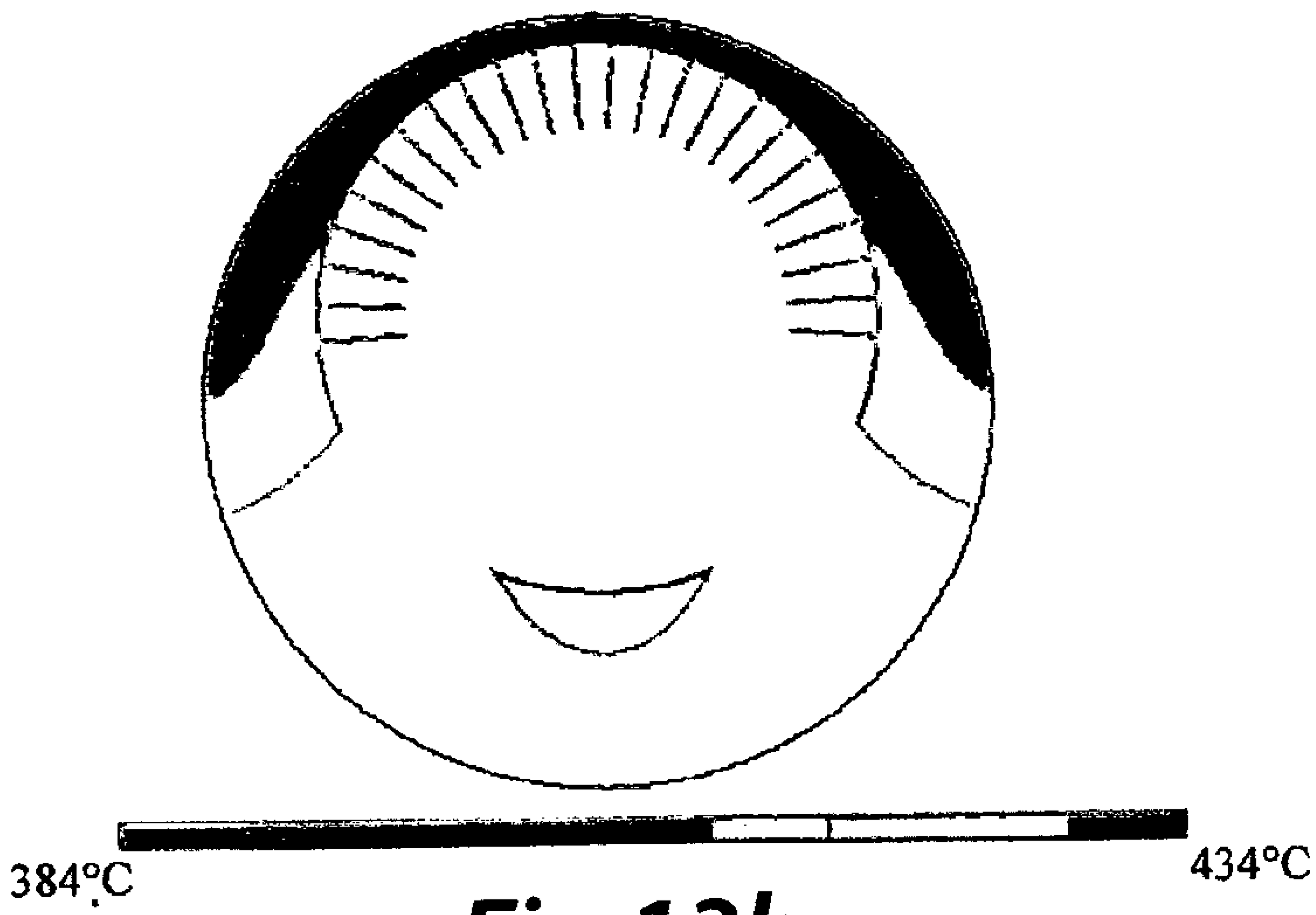


Fig.13b

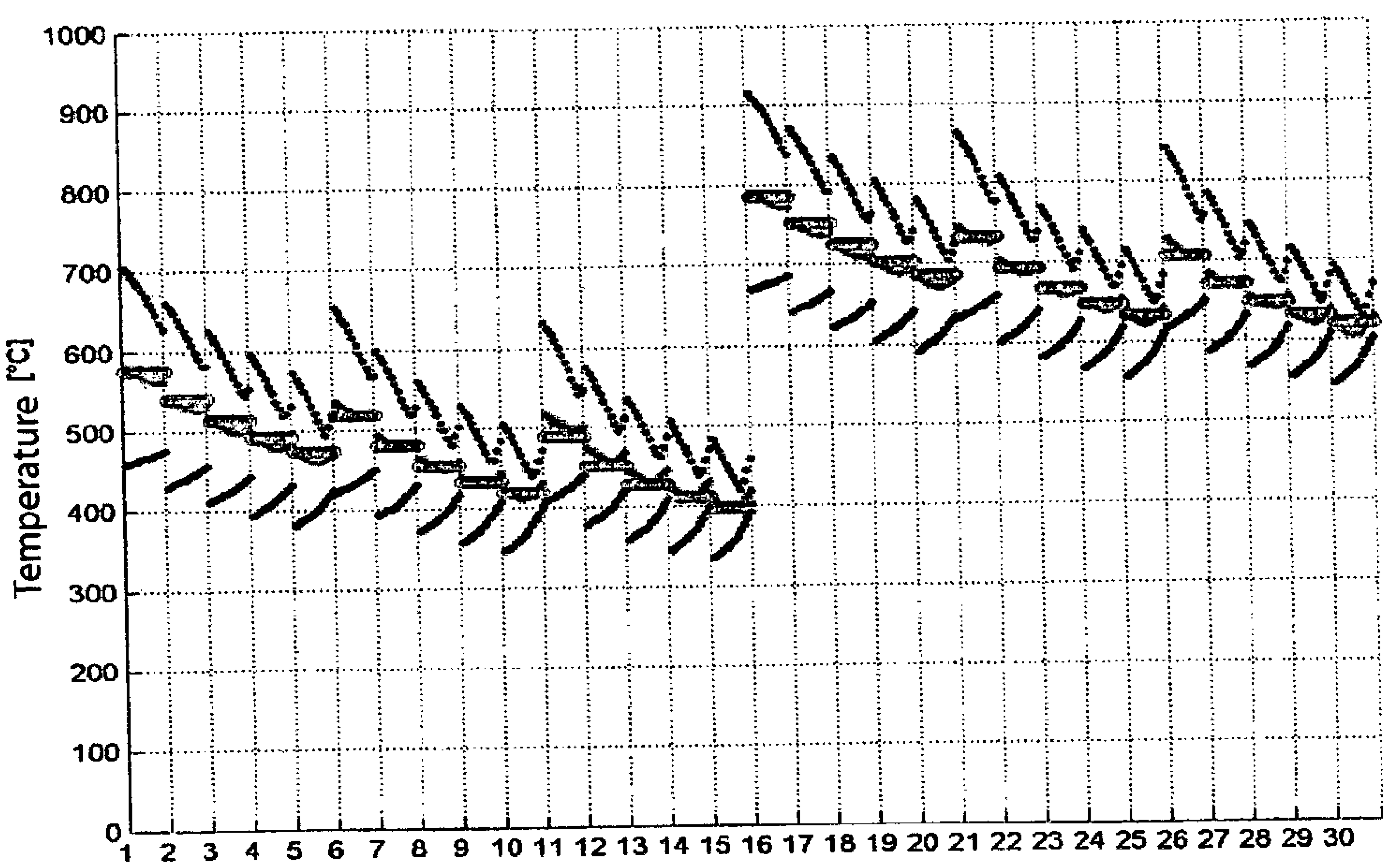


Fig.14

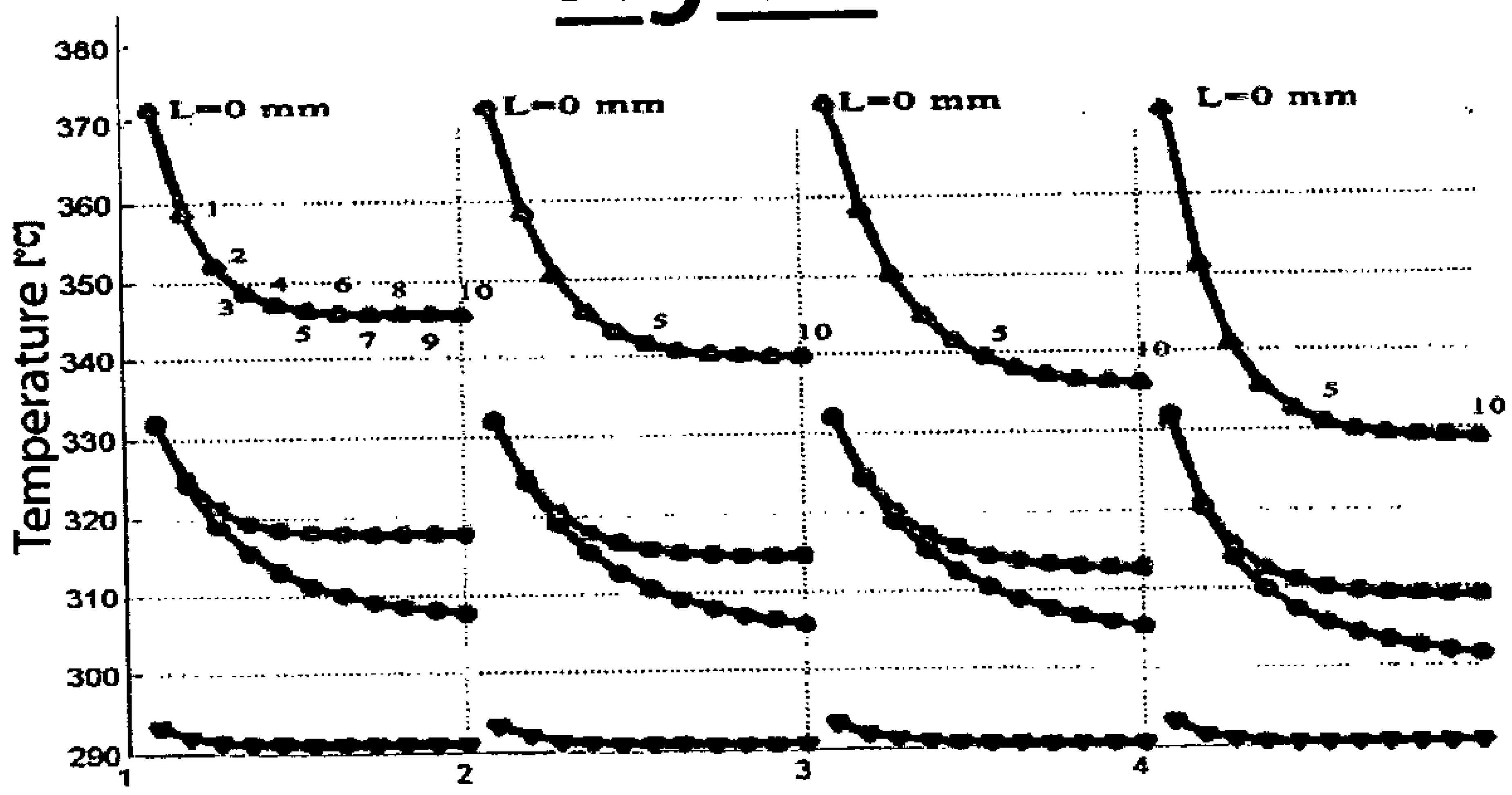


Fig.15

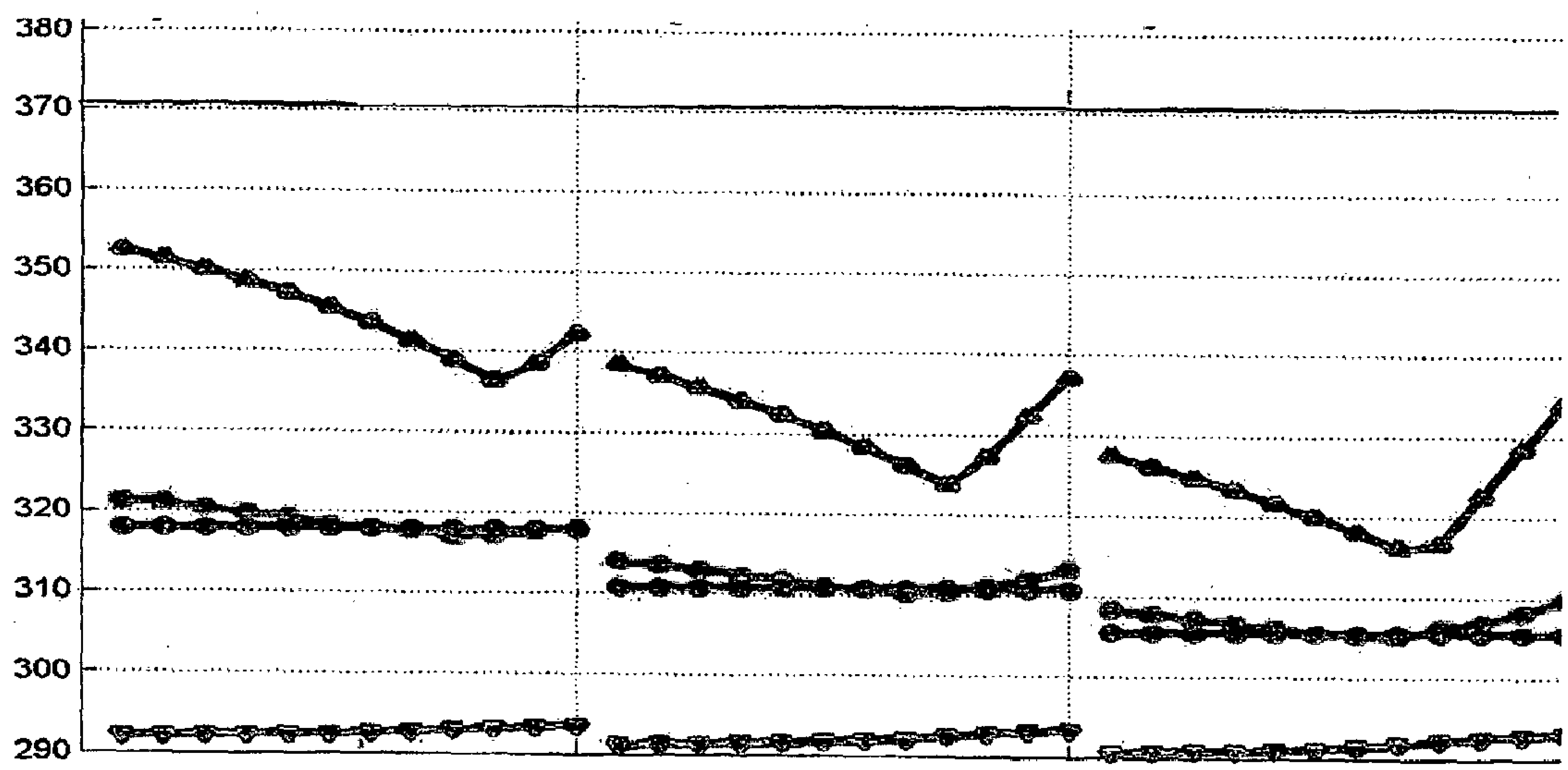

380
370
360
350
340
330
320
310
300
290

Fig.16

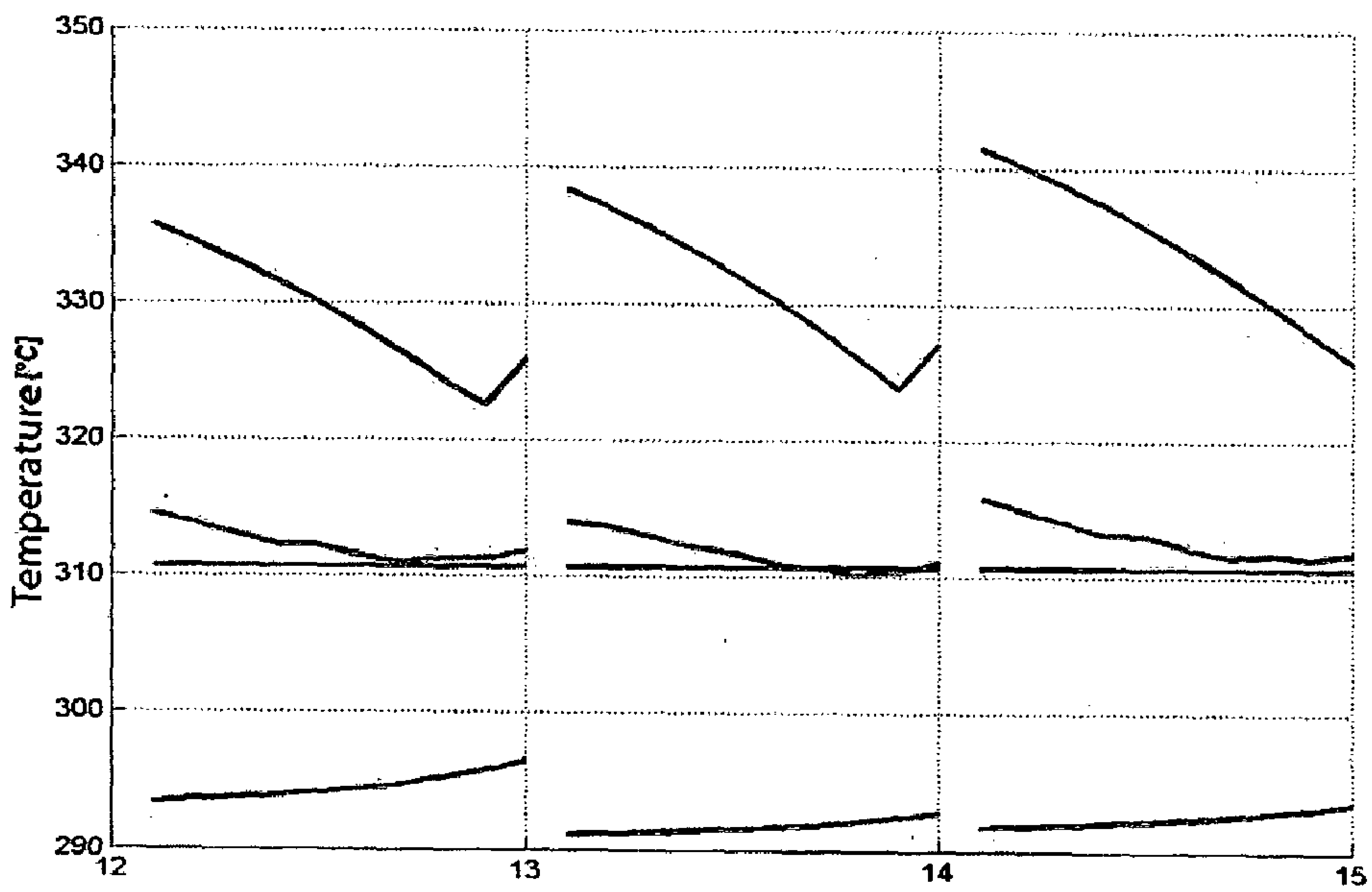

350
340
330
320
310
300
290
Temperature[°C]
12
13
14
15

Fig.17

TUBULAR HEAT EXCHANGER, IN PARTICULAR RECEIVING TUBE OF A CONCENTRATING SOLAR PLANT

[0001] The present invention concerns a tubular heat exchanger, in particular a receiving tube of a concentrating solar plant.

[0002] More precisely, the invention concerns a receiving tube for a concentrating solar plant, also known as solar tube or solar boiler, realised with features reducing its average temperature and homogenizing the axial and circumferential temperature gradients originating because of the irradiation produced by the solar power being concentrated, by means of mirrors, usually but not necessarily parabolic mirrors (so called "parabolic through" or Fresnel technology or other technology concentrating sun energy and anyway any case indicated in the following with the expression concentrating thermodynamic solar plant) on a portion of the surface of the sane solar tube or receiver. The invention also allows for obtaining an homogenization of the thermodynamic properties of the heat-carrying fluid flowing inside the solar tube.

[0003] As it is known, the functioning of a concentrating solar plant provides for the amplification of the solar irradiation by concentrating it by means of concentrating mirrors, generally but not necessarily having a parabolic shape, conveying the irradiation on a receiving tube or solar tube, in order to increase the thermal power collected by the tube and favoring it warming up to high temperature, i.e. the increasing of its enthalpy or the change of state of a heat-carrying fluid (i.e. a fluid carrying out energy) flowing inside it or anyway a thermodynamic transformation favouring thermal chemical transformations in the fluid (chemical reformulation of the heat-carrying fluid by means of thermal energy coming from the irradiated tube).

[0004] The heat-carrying fluid can be subsequently used for a series of applications exploiting such a thermodynamical state, such as for example applications linked to the conversion of thermal energy to electric power by means of transformation plants downstream the solar tube using gas turbines and vapour turbines or combinated cycle plants; applications linked to the water desalting for producing water to be used for irrigation, drinking, industry; applications connected to reforming processes of pure or mixed light hydrocarbons (methane, propane, butane, liquefied petroleum gas, methanol, etc. . . .) intended for the enrichment of fuels in fuel fractions having a higher value (hydrogen or lighter hydrocarbons) or for the decomposition or reformulation of fuels from the original chemical formulation to simpler molecular forms.

[0005] Since the applications connected to the conversion of thermal energy to electric power are those generally making use of this technology (concentrating thermodynamic solar plant), in the following reference is made, for illustrative non limitative purposes of the invention, to them in particular.

[0006] The technology allowing for using sun energy for producing electric power, in the following simply called "thermodynamic solar power", is based on the concentration of sun radiation on a receiving tube positioned at the focus of a parabolic strings of mirrors and inside which a heat-carrying fluid flows (chosen for example amongst diathermic oil, molten salts, etc.) heating up to a high temperature (in current applications 500-600° C.), such thermal power being provided, possibly by means of an intermediate fluid through a thermal energy storage section, for being converted into mechanical-electrical power by means of a motor thermodynamic cycle, such as for example the cycles of Rankine, Him, Joule or Ericsson, wherein the heat-carrying fluid or the intermediate fluid represent the high temperature source.

[0007] The concentration ratio obtained by the parabolic mirrors (at present usually variable between 50 and 150) allows for the tube heating up to a high temperature (with significant circumferential and axial temperature gradients in the solar tube) in order to increase temperature or enthalpy of the heat-carrying fluid with respect to the values that are possible without concentrating the energy of the sun. It is therefore possible to extend the possibility of using the energy of the sun to applications that would not be possible without concentration, in particular, as already cited, to the thermodynamic conversion powered by sun energy or to total or partial fuel reforming or other applications needing a high temperature.

[0008] According to the known prior art about solar tubes, it is provided for the use of metallic tubes of uniform thickness (circumferentially ed axially) able to resist to the working pression of the heat-carrying fluid, smooth on the internal surface in order to limit pressure drop, the external surface being covered with suitable materials increasing the absorption coefficient of the solar radiation, allowing for a behaviour as a black body. Around the metallic tube a co-axial tube is provided, suitably thermally insulated from the solar tube and made of a material that is transparent to the solar radiation and absorbs the re-radiated energy from the metallic tube, so that a great part of the re-radiated energy from the solar tube is retrieved. The interspace defined between the metallic tube and the surrounding glass tube is kept under vacuum, with the consequence that also the following must be provided: a static sealing system between the metallic tube and the glass tube; suitable (metallic) elastic bellows allowing for absorbing the great axial expansion of the tube following its temperature and axial temperature gradients increase; a joint working also as a sealing allowing for rotation of parts of the solar tube with respect to others, rotation connected with the need of continuity of great length of the solar tube, the need of bringing to the soil the solar tube itself, quick disconnection or division of the tube or parts thereof. A solar tube, in fact, depending on the power needed, can extend for kilometers in terms of length and, therefore, needs portions where it is easy to realise a disconnection of parts.

[0009] The evolution of the thermodynamic solar energy technology (and of the technology of its possible applications) foreste an increase in the dimensions (diametre) of the solar tube, in order to allow for a greater surface collecting the thermal energy and, therefore, a greater power capacity of the plant, for the same tube length (and thus for the same overall extension); a further increase of the sun energy concentration ratio from that available on aplanar surface and that, concentrated by parabolic mirrors, available on the surface of the solar tube; and consequently an increase of the materials and heat-carrying fluids working temperature, with a consequent greater severity of the working conditions of the mechanical sealing system, and of the problem of longitudinal and circumferential thermal distorsions of the tube, of sealing between the metallic tube and the transparent tube, of sealing between adjacent portions of tube, i.e. with connection and disconnection devices of different parts of the plant, and of the structural stability of the treatments of the surface of the tube to improve its behaviour as a black body. The use of gas (for example nitrogen, carbon dioxide, water steam, helium, carbon dioxide), as heat-carrying fluid, in order to overcome the problem of the solidification of presently used heat-carrying fluids (in particular molten salts) at a temperature below 200-250° C., with the consequence of the need for an increase of the pressure of the heat-carrying fluid, in order to compensate the low density, with a consequent further worsening of the working conditions of the sealing system. Moreover, in order to favor an increase of the thermal power carried by the heat-carrying fluid (as a consequence of the increase of the diametre of the tubes or the increase of the concentration ratio) it will be needed to increase the temperature of the heat-carrying fluid at the outlet of the solar tube and this will imply an increase of the working temperature of the materials of the tube and a worsening of the circumferential and axial temperature gradients.

[0010] The solutions according to the prior art, also in the light of the technical evolution, present a series of working problems.

[0011] Since the receiving tube is irradiated only on a semi-cylindrical surface, great circumferential temperature gradients develop on the tube. Such gradients are critical for the static sealing system between the metallic tube (inside which the heat-carrying fluid flows) and the glass tube or the tube of a material that is transparent to sun light and absorbs the light re-radiated from the tube.

[0012] Further, since the heat-carrying fluid flowing inside the receiving tube gradually increases its temperature, consequently temperature gradients develops along the axis of the receiving tube and around the circumference of the same, causing important differential expansions that are critical for the thermo-mechanical stress and for the structural preservation of the treatments of the surface, due to the differential expansions around the circumference and the axis of the tube.

[0013] The replacement of presently used heat-carrying fluids (oils or molten salts) with gas, will involve a debbrease of the convective heat transfer coefficient, causing, at the same temperature of the heat-carrying fluid, an increase of the temperature of the tube, unless an increase of the working pressure of the heat-carrying fluid (gas) is provided, but this would need much higher thickness of the tube with an increase in particular in weight, costs, deflection of solar tube and overall deformations. The gas speed increase inside the tube (consequent to the increase of the mass flow) in fact will compensate only partly the lowering of the heat transfer coefficient. For this reason, in the applications concerning the gas the problem of increasing the heat transfer coefficient between gas and metal wall will be present, in order to realise a lowering of the temperature difference between gas and metal walls, the thermal power irradiated by the sun and suitably concentrated on the solar tube being constant, as will be explained in the following, other conditions being the same (for example but, not only: geographical position of the site, solar tube dimensions, concentrating system dimensions, solar tube position with respect to the focus of the parabolic mirrors).

[0014] Moreover the increase in the temperature of the tube, for the same temperature of the heat-carrying fluid, and also the presence of circumferential and axial temperature gradients due to the non uniformity of the concentrated solar irradiation and the temperature of the heat-carrying fluid increasing more and more flowing inside the solar tube, make critical the conservation of the structure of the materials used for covering the surface of the solar tube exposed to the radiation with the aim of reducing the sun radiation reflection and increasing the absorbed portion.

[0015] Further, since the overall solar tube is realised by assemblying different materials with surface treatments and housing systems differently reacting to temperature changes and to heat-carrying fluids passing through, this subjects the solar tube to differential extensions of the different parts composing it, which can generate deformations of the structure, as much greater as is the temperature of the tube.

[0016] At last, also the weight if the solar tube represent, at higher temperature, a critical element, since it increases the deflection connected to the deformation due to gravity force and, therefore, the differential deformations, particularly critical, as already said, on the external surface of the tube, where said surface treatments reducing the thermal power the tube radiates towards the exterior are realised.

[0017] In this context it is entered the solution according to the present invention, with the aim of providing for a receiving tube for a concentrating solar plant, manufactured so to overcome the problems of the tubes according to the prior art and due to the thermal state of the metallic tube and to its circumferential and axial temperature gradients and, therefore, the problem of the sealing between the metallic tube and the glass tube and also the problems connected to the thermal stratification of the heat-carrying fluid and also problems connected with the mechanical stability of the surface treatments.

[0018] These and other results are achieved according to the present invention by proposing a receiving tube for a concentrating solar plant having structural features (thickness of the tube suitably circumferentially changing) and/or elements (fins, grooves, inserts, all with radial extension on the side of the heat-carrying fluid) locally changing with a specific aim the (convective and conductive) circumferential and axial thermal resistance (realising changes in the same circumferential and axial direction of the conductive and convective thermal resistance). Moreover, such elements favor the homogenization of the heat-carrying fluid flowing inside the tube.

[0019] With reference to the cited elements (fins, grooves, inserts, all extending radially on the side of the heat-carrying fluid), their introduction inside or outside the tubes, with the aim of increasing the thermal exchange between the fluids exchanging energy, is prior art. In fact, it is known that such elements increase the turbulence of a fluid and, therefore, the exchanged thermal energy, thus realising a reduction in the dimensions of the heat exchanger, for a set thermal power to be exchanged. The field of internal combustion alternative engines, to which most of the prior art is referred, majes use of such elements in cases when exchange fluids are usually exhaust gases, air, cooling liquids. In the cited field, in fact, the size reduction is a fundamental element for the insertion of the heat exchanger, such a size reduction being realised by increasing the heat transfer coefficient. A further need is requie in prior art applications concerning the reduction of pressure drop. EP0677715A1, for example, discloses fins arranged in a V shape, intensifying the turbulence and preventing the deposition of particles that would foul the exchanger on the internal side of fluid (exhaust gases of a internal combustion alternative engine being cooled). A similar solution is presented, always for thermal exchangers operating with engine exhaust gases, in patents DE19540683A1, DE19654367A1 and DE19654368A1 where such elements are realised with a low cost technology. Similar inserts are disclosed in EP1061319A1 and DE10227084A1, with reference also to other kinds of thermal exchangers (for example engine radiators) having a uniform distribution of heat along the surface: the aim of the solutions according to these patents being being always that of intensifying the exchanged thermal power (per surface unit of the heat exchanger), thus decreasing the size of the exchanger and realising more compact solutions.

[0020] DE102005029321A1 and US2010/0139631A1 disclose a further improvement in the arrangement of inserts, with the aim of realising an optimum between the exchanged thermal power (for the same surface of the heat exchanger) and the pressure drop of the fluid, tending to intensify as a consequence of the positioning of inserts. The characteristic of these patents consists in intensifying the number of structural elements in the flow direction, thus realising an intensification of the exchanged thermal power: in the area of the fluid inlet in the tube a reduced number of structural elements is preferably arranged, limiting the pressure drop. In this area, in fact, the high difference of temperature between the two fluids make it possible to obtain a great transfer of thermal energy, thus the elements could be only partially useful in order to intensify the exchanged thermal power, but a decrease in their number favours the decrease of pressure drop. More and more the fluid enters inside the tube, the exchanged thermal power tend to lower, since the difference of temperature increase. An intensification in the number of structural elements (in the direction of the fluid) compensate such a reduction, causing an increase of the convective heat transfer coefficient. With the aim of achieving an increase of the exchanged thermal power (with the intensification of the number of structural elements along the direction of the flow) and the decreasing of pressure drop (compared to a solution providing for a uniform distribution of structural elements), DE102005029321A1 and US2010/0139631A1 disclose different constructive solutions, where the structural elements are differently spaced and angled with respect to the flow direction.

[0021] FR2682747A1 discloses a tubular heat exchanger presenting a non cylindrical thermal exchange surface, that is a thermal surface with irregular shapes, possibly repeated on angular sectors, with the aim of increasing the thermal exchange surface and intensifying the exchanged thermal power. Such internal surface does not have structural elements (fins) but superficial grooves (irregularities). The application referred to is that of the condensation of a fluid and the aim of these irregularities of the surface is that of promoting the intensification of the turbulence and/or the passage from a laminar flow to a turbulent flow. Such grooves favour the formation of drops, acting as promoters of the condensation and, reducing the thickness of the metallic tube, reduce the radial thermal resistance, increasing the convective heat transfer coefficient. The resulting heat exchanger is more efficient and has lower realisation costs.

[0022] A similar solution is disclosed in US005409675A, in which the internal section of the tubular exchanger is shaped with lobes having a set pitch and shape, increasing the thermal exchange surface on the side of the internal fluid realising at the same time a reduction of the pressure drop. The specific application is for a heat exchanger inside which pirolisis transformations for producing olefines are performed. In the flow direction, the section of the heat exchanger changes, getting bigger in the flow direction, and the internal surface keeps the lobe size such to define an increase of the volume available for the reacting gas.

[0023] US005655599A discloses a solution according to which the internal surface of a tubular exchanger are provided with fins in order to increase the thermal energy exchanged due to convection and irradiation between a gas (also subject to combustion) flowing inside and the external surface of the tube. The shape of the fins extending realising a sensible reduction of the passage surface for the gas inside is optimised to intensify the radiant thermal exchange between the gas and the exposed surface, thus extending the application of fins of the thermal exchange surfaces (usually referred to the convection) to the case of radiant exchange. In this solution it is further provided for the fins being realised along elicoidal lines subjecting the internal fluid to a rotation around the axis of the tube (twisting). The pitch of such elicoidal lines can be changed along the tube.

[0024] The use of fins provided inside tubular thermal exchangers is also disclosed in DE102004027208A1. According to this patent, fins are arranged internally in order to increase the temperature of the tube heat exchanger and prevent the condensation of water steam that is comprised in exhaust gases deriving from combustion processes, delaying the formation of acidic condensates. In this way it is possible to use materials that are less resistant to corrosion and less expensive. In the claims solution, gas arriving from the combustion process reaches the tubular heat exchanger on the external surface, with a perpendicular direction with respect to the axis of the tube exchanger. It is thus realised a first portion of the external surface of the tube, reached by the gas flow, in correspondence of which exhaust gas stagnates ("upstream region"), increasing its pressure, and a second portion of the external surface of the tube, diametrally opposed with respect to the first, in fluodynamic wake, characterised by low pressure ("downstream region"). When the combustion gas reaches the exchange tube (inside which a fluid flows) it is realised a different difference of temperature between the gas and the metal wall, in the region reached by the flow (upstream) and in that diametrally opposed (downstream). In the latter region, considering the lower pressure of the gas, the difference of temperature can take the gas below the dew-point of acidic condensates. The different arrangement of the fins promotes an increase in the difference of temperature between fluid and gas thus preventing the possibility of formation of acidic condensates.

[0025] The working principle of the proposed solutions making the man improvement with respect to the prior art, is described in the following on the base of a physical-mathematical approach describing the processes occurring in a solar tube irradiated by concentrated and non concentrated sun energy.

[0026] It is known that the mechanisms of forced convection between a fluid flowing in a tube are described by the equation:

$$Nu=kRe^{0.8}Pr^{0.4} \tag{1}$$

wherein the Nusselt number (Nu) is

$$Nu = \frac{h \cdot D}{k} \tag{2}$$

wherein h is the gas-wall convective heat transfer coefficient, D the diametre of the tube, k the internal thermal conductivity of the tube and wherein the Reynolds number Re and the Prandtl number Pr are given by:

$$Re = \frac{\rho \cdot u \cdot D}{\mu} \quad Pr = \frac{v}{k \cdot c_p} \tag{3 a-b}$$

wherein ρ, μ, v and c_p are respectively the fluid density, dynamic viscosity, kinematic viscosity, specific heat at constant gas pressure. The fluid speed u, together with the cross section of the tube and with its density, defines the gas mass flow within the solar tube.

[0027] Equations (1), (2) and (3) are valid for both gas and liquid; between these two possibilities, the option of considering a gas involves a heat transfer coefficient lower of about one order of magnitude.

[0028] The thermal energy reaching the solar tube and exchanged with the heat-carrying fluid can be represented by the equation:

$$Q_{IRR} = q_{irr} \cdot S \cdot = h \cdot S_{int} \cdot (T_p - T_{fl}) \tag{4}$$

wherein:

Q_{irr} is the thermal power reached by the external surface of the tube;

g_{irr} is the specific thermal flow reached by the external surface of the tube and deriving from the solar irradiation amplified by the amplification ratio of the solar mirrors;

S· is the irradiated external surface of the tube;

S_{int} is the internal surface of the tube in contact with gas;

$(T_p - T_{fl})$ is the difference of temperature between the internal wall of the solar tube (lapped by the fluid) and the heat-carrying fluid.

[0029] The term on the right of the equation (4) is constant and depends only by the exposed surface (external diameter of the solar tube and by the concentration ratio of sun light achieved by the concentrating mirrors and by the insulation features of the site) thus, the equation (4) can be used in the form:

$$(T_p - T_{fl}) = \frac{Q_{irr}}{h \cdot S_{int}} \tag{5}$$

In order to evaluate the difference of temperature between the fluid and the wall.

[0030] It is thus possible to see that when the heat transfer coefficient lowers, the temperature of the solar tube internal wall tends to increase, involving an increase of the temperature of the external surface of the solar tube (shell) according to the equation:

$$T_m = T_p + \frac{Q_{irr}}{\ln\frac{R_{inr}}{R_{est}} \cdot \frac{1}{2 \cdot \pi \cdot L \cdot k}} \tag{6}$$

If Q_{irr} increases, T_m tends to increase in turn. T_{fl} tends to increase also more and more the fluid proceeds inside solar tube, thus obtaining T_m getting bigger in an axial direction.

[0031] The same power transferred to the heat-carrying fluid must pass due to conductive behaviour all the thickness of the tube, thus increasing the temperature of the solar tube external surface. This causes, according to the equations (5) and (2), a temperature increase of the internal surface of the tube (and of the shell) in the direction of the axis of the boiler itself. A temperature gradient is caused along the axis of the solar tube generating a conductive thermal flow moving along the solar tube from sections at higher temperature to sections at lower temperature. This thermal flux flow sto the part of the sections of the tube having a lower temperature. For this reason, the heat-carrying flow has a heating gradient along the axis of the solar tube being higher in the first portions of the boiler and tends to get uniform in the last portions of the boiler. This phenomenon causes, as already seen, a thermal temperature distribution along the solar tube axis with consequent differential extensions.

[0032] In case the irradiated power is limited to a portion of the external surface of the solar tube, situation occurring in solar tubes of concentrating thermodynamic solar plants concentrating the solar irradiation and referred to in the present invention, circumferential temperature gradients are generated causing circumferential thermal distorsions added to the longitudinal distorsions along the axis of the tube.

[0033] The fact that only half of the surface of the solar tube is irradiated causes a further phenomenon represented by the fluid thermal stratification within the tube. Heating being mostly obtained in a portion of the tube, in fact, causes locally a more intense thermal energy transfer, causing a local change of fluid density and, therefore, of the other connected thermodynamic and physical properties. This phenomenon is more evident in case the heat-carrying fluid is gas; and therefore, tends to amplify a non optimal functioning of the solar tube exactly related to the expected technical evolution. Such thermal stratification is also unfavourable in the applications of the solar tube for processes of hydrocarbon reforming or for other chemical reactions in which homogeneity of the heat-carrying fluid (reactant) is an important condition.

[0034] The present invention aims at solving the problems relating to circumferential and axial temperature gradients occurring inside the solar tube by means of the introduction of protruding elements (fins, grooves, inserts, all extending radially on the side of the heat-carrying fluid) on the irradiated portion of the surface, on the internal side of the tube. A result induced by this invention is also that of lowering the mean temperature of the tube in its working conditions. moreover, the present invention takes part also in connection with the solution of the problem of mechanical sealing between glass tube (or another transparent material for the direct solar radiation) and the solar tube and with the protection of the integrity of surface treatments realised on the solar tube external surface.

[0035] The operating principle at the base of the positioning of said protruding elements (fins, grooves, inserts, all extending radially on the side of the heat-carrying fluid) is obtained by the local change (obtained by said devices as proposed) of the convective and conductive thermal resistance, that are respectively:

$$R_{convettiva} = \frac{1}{h \cdot S_{int}} \quad R_{conduttiva} = \frac{w}{k \cdot A}$$

being w the thickness of the tube and A the surface exposed to conductive thermal exchange. For such a reason, the decreasing of the convective thermal resistance requires for increasing the product $h \cdot S_{int}$ intervening on h and S_{int}, whereas reducing the conductive thermal resistance requires a lowering of the thickness and/or an increase of the section for the passage of thermal energy (in the conductive thermal flow direction). To these reductions obtained locally by means of said protruding elements (fins, grooves, inserts, all extending radially on the side of the heat-carrying fluid) for the convective side and/or changing the thickness of the tube in a circumferential direction for the conductive side corresponds locally an increase of the conductive thermal flow in a circumferential (and axial) direction and, therefore, a homogenization of the temperature circumferential thermal gradients. If such devices are repente periodically along all the tube, an equivalent effect is obtained also on the homogenization of the axial thermal gradients. The changing thickness of the tube in a circumferential direction and/or the insertion of said protruding elements (fins, grooves, inserts, all extending radially on the side of the heat-carrying fluid), as confirmed by the equations above of the conductive thermal resistance, realising the same homogenising effect on the temperature gradients. The two techniques, when applied together, amplify such a homogenising effect.

[0036] The purpose of the present invention is therefore that of realising a receiving tube of a concentrating solar plant allowing for overcoming the limits of the solutions of the prior art and for obtaining the previously described technical results.

[0037] Further aim of the invention id that said receiving tube can be manufactured with substantially low costs, as far as both production costs and managing costs is concerned.

[0038] Not least aim of the invention is that of realising a receiving tube of a concentrating solar plant that is substantially simple safe and reliable.

[0039] It is therefore a first specific object of the present invention a tubular heat exchanger between a source of radiant heat located externally with respect to said tubular heat exchanger and a heat-carrying fluid flowing inside the exchanger, the radiant heat coming from said source being concentrated on a longitudinal portion of the external surface of said tubular heat exchanger, characterised in that it comprises means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger.

[0040] According to the invention, said means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger comprise a plurality of protruding or recessive elements with respect to the internal surface of the tubular heat exchanger, positioned mainly in correspondence of said longitudinal portion of the external surface of the tubular heat exchanger on which the radiant heat coming from said source is concentrated.

[0041] Alternatively, according to the invention, said protruding or recessive elements are positioned along the cylinder generatrixes of the tubular heat exchanger, or are positioned along curves that do not follow the cylinder generatrixes of the tubular heat exchanger.

[0042] Preferably, according to the invention, the number and/or the surface of said protruding or recessive elements change in a circumferential and axial direction.

[0043] Moreover, according to the invention, the shape of said protruding or recessive elements can change in a circumferential and axial direction.

[0044] Further, always according to the invention, said means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger comprise a thickness of the walls changing in a circumferential direction, between a minimum value in correspondence of said longitudinal portion of the external surface of the tubular heat exchanger on which the radiant heat coming from said source is concentrated, and a maximum value in correspondence of the longitudinal portion of the external surface of the tubular heat exchanger opposed to that on which the radiant heat coming from said source is concentrated.

[0045] It is moreover a second specific object of the present invention the use of a tubular heat exchanger as previously defined in a concentrating solar plant.

[0046] It is then a third specific object of the present invention the use of a tubular heat exchanger as previously defined in a hydrocarbon reforming plant for producing hydrogen or mixtures of methane and hydrogen or of hydrogen other hydrocarbons lighter than the originary hydrocarbon.

[0047] The present invention will be now described, for illustrative non limitative purposes, according to its preferred embodiments, with particular reference to the figures of the enclosed drawings, wherein:

[0048] FIG. **1** shows a schematic view of a portion of a concentrating parabolic solar plant;

[0049] FIG. **2** shows a schematic section view of a solar tube, and represents the portion of the cylindric surface of said tube reached by the concentrated irradiation;

[0050] FIG. **3** shows a schematic section view of a solar tube, and represents the genesis of circumferential temperature gradients;

[0051] FIG. **4** shows a perspective view of a portion of solar tube according to a first embodiment of the present invention;

[0052] FIG. **5** shows a perspective view of a portion of solar tube according to a second embodiment of the present invention;

[0053] FIG. **6** shows a perspective view of a portion of solar tube according to a third embodiment of the present invention;

[0054] FIG. **7** shows a portion of solar tube according to a fourth embodiment of the present invention;

[0055] FIG. **8** shows a portion of solar tube according to a fifth embodiment of the present invention;

[0056] FIG. **9** shows a diagram of the temperature distribution over the solar tube, having a set thickness and irradiation of the emisurface, as a function of the angle on the circumference, for a heat-carrying fluid with a convective heat transfer coefficient of 55 $W/m^{2\circ}$ C. and a fluid temperature of 290° C.;

[0057] FIGS. **10***a* and **10***b* show the distribution of the thermal zones over the circumferential section of two solar tubes according to the present invention, respectively having equispaced fins over all the circumference or in correspondence of a portion thereof, for a heat-carrying fluid having a heat transfer coefficient of 55 $W/m^{2\circ}$ C. and a fluid temperature of 290° C.;

[0058] FIG. **11** shows the trends of the maximum, mean and minimum temperature of the solar tube for changes of the circumferential disposition of fins in a solar tube according to the present invention, having fins equispaced, for a heat-carrying fluid having a heat transfer coefficient of 55%/$m^{2\circ}$ C.;

[0059] FIGS. **12***a* and **12***b* show the distribution of the thermal zones over the circumferential section of two solar tubes according to the present invention, respectively over a tube having constant thickness and over a tube having a changing thickness along its circumference, for a heat-carrying fluid having a heat transfer coefficient of 55 $W/m^{2\circ}$ C. and a fluid temperature of 290° C.;

[0060] FIGS. **13**a and **13**b show the distribution of the thermal zones over the circumferential section of two solar tubes according to the present invention, with the section changing in the same way around the circumference, respectively with equispaced fins over all the circumference or in correspondence of a portion thereof, for a heat-carrying fluid having a heat transfer coefficient of 55 $W/m^{2\circ}$ C. and a fluid temperature of 290° C.;

[0061] FIG. **14** shows on an overall diagram the results of the test performed for a heat-carrying fluid having a heat transfer coefficient of 55 $W/m^{2\circ}$ C., the abscissa indicating each tested configuration, changing number, height and disposition of fins and the ordinate showing the corresponding maximum, minimum and mean temperature values;

[0062] FIG. **15** shows on a diagram the results of the test performed for a heat-carrying fluid having a heat transfer coefficient of 520 $W/m^{2\circ}$ C., the abscissa indicating each tested configuration, changing the number and height of fins (equispaced between one another) and the ordinate showing the corresponding maximum, minimum and mean temperature values;

[0063] FIG. **16** shows on a diagram the results of the test performed for a heat-carrying fluid having a heat transfer coefficient of 520 $W/m^{2\circ}$ C., the abscissa indicating each tested configuration, keeping fixed the features of the fins and changing their number and position inside the tube and the ordinate showing the corresponding maximum, minimum and mean temperature values; and

[0064] FIG. **17** shows on a diagram the results of the test performed for a heat-carrying fluid having a heat transfer coefficient of 520 $W/m^{2\circ}$ C., abscissa indicating each tested configuration, keeping the features of the fins fixed and changing the position of the zone of the tube having a higher thickness, and the ordinate showing the corresponding maximum, minimum and mean temperature values.

[0065] With reference to FIG. **1**, it is schematically shown a portion of a concentrating parabolic solar plant, in which two solar tubes **1** are represented and a plurality of parabolic mirrors **2**, and the solar rays S deviated by parabolic mirrors **2** and concentrated over the solar tubes **1**.

[0066] FIG. **2** shows a schematic section view of a solar tube **1**, and indicates with θ the angle subtending the cylindrical surface portion of said tube reached by the concentrated irradiation (schematised with the arrows indicated with P_1).

[0067] With reference to FIG. **3**, it is shown a further shematic section view of a solar tube **1**, and the parameters used in the previously described formula are represented.

[0068] With reference to FIG. **4**, a solar tube according to a first embodiment of the invention is indicated with the reference numeral **10**, and is provided with a plurality of fins **11** in it, arranged in correspondence of the portion **12** of the external surface of the tube irradiated by the solar radiation concentrated by the parabolic mirrors, equispaced between each other in a circumferential direction and positioned along the length of the solar tube. The presence of fins causes a lowering of the convective thermal resistance in the irradiated area, with advantages as far as it concerns the temperature of the tube.

[0069] Further, the presence of fins in a portion of solar tube allows for locally lowering the thermal resistance of the wall of the tube, thus locally promoting the transit of the thermal flow with a lowering of the circumferential thermal gradients and of temperature.

[0070] With reference to the embodiment shown in FIG. **5**, besides the changes of the convective thermal resistance and conductive thermal resistance in the wall of the tube **20** in a circumferential and axial direction, this second embodiment produces a mixing of the heat-carrying fluid reducing the thermal stratification and also an intensification of the convective heat transfer coefficient on the side of the heat-carrying fluid.

[0071] According to this embodiment of the invention, fins **21** are positioned in a non linear way inside the tube **20**, in correspondence of the irradiated portion **22**. Fins, therefore, can be realised by following any curve belonging to the internal cylindrical surface of the tube.

[0072] Amongst the embodiments of the present invention, all those solutions where fins are introduced having different geometry (height, thickness, length) changing in a circumferential and axial direction according to any condition; i.e. with such a shape locally creating or intensifying the non axial fluid motion (for example a swirl motion or elicoidal movement within the tube) with the further advantage, beside homogenization, of further locally intensifying convective heat transfer coefficient.

[0073] FIG. **6** shows a portion of a solar tube according to a third embodiment of the present invention; the change of thickness of the tube in a circumferential direction realises in the same direction a change in the conductive thermal resistance (a lowering of the thickness increase) with advantage on the reduction of circumferential thermal gradients. The embodiment shown in FIG. **7** shows the surface of the cylinder representing the internal surface **33** of the tube **30** eccentric with respect to the external (cylindrical) surface **34**, in order to define a lower thickness of the wall of the tube **30** in correspondence of the irradiated area **32**. The changes in thickness can be obviously realised also by modifying the external surface **34** of the solar tube **30**. By distributing the thickness of the tube **30**, possibly integrating the previously described solutions, around the circumferential direction localised changes are realised of the overall (conductive and convective) thermal resistance to the passage of the thermal flow.

[0074] The embodiments shown with reference to FIGS. **7** and **8** represent solutions in which the presence of fins (straight or curved) is associated to the realisation of a thickness of the wall of the tube circumferentially changing.

[0075] Cases shown in FIGS. **4**-**8** can also be provided with fins changing their thickness in the direction of the radius and along their directrixes (straight or having any shape). The portions of solar tube reported can be repeated to reproduce the overall length of the solar tube or spaced out or alternated (portions with straight fins followed by portions with fins following any curve).

[0076] Further embodiments of the present invention are represented by cross section of the heat-carrying fluid according to any other shape, for example with cross sections realised by means of eccentric cylindrical holes partially intersecating or elliptical cross sections or any other shape realising changes in the conductive thermal resistance around the circumferential direction. With respect to the same principle, the section of the tube can also be variable along the axis of the tube thus realising, for example, frustoconical holes.

[0077] The solutions shown in FIGS. **4-8** can be realised in a solar tube separately or simultaneously; these further configurations being object of further embodiments of the present invention.

[0078] The validity of the present invention is demonstrated in the following on the base of a series of physico-mathematical elaborations making use of methodologies for the calculation of circumferential thermal levels and of exchanged flows, evaluated according to analytical methods and through the theory of finite elements. The calculations were made for temperature levels, thermal gradients and thermal distorsions connected to the thermo-mechanical state.

[0079] The theoretical predictions were made with reference to two levels of the convective heat transfer coefficient; the first characterised by a high level (520 W/m^{2}° C.) and closer to the applications of thermodynamic solar power using gas as the heat-carrying fluid; the second characterised by a lower level (55 W/m^{2}° C.) closer to industrial applications (reforming, desalting, or other). In this way, the simulations made cover a wide possibility of variation of the most important parametre for the performance of the tube or solar boiler.

[0080] A parametric analysis has been done as a function of:

[0081] a) fins height (5, 10, 15, 20,25 mm);

[0082] b) number of fins (8, 12, 16, 20, 24);

[0083] c) angle of disposition of the fins around the circumference (from 310° to 90° with angular interval of 20°-310°, 290°, 270°, 250°, 230°, 210°, 190°, 170°, 150°, 130°, 110°, 90°-, centered around the direction of the sun rays (point of higher temperature of the tube). Purposely in the analysis angular intervals of disposition of the fins were considered greater than those involved by the irradiation on the external surface. This was made in order to outline the effect of concentration of the fins in the irradiated portion only;

[0084] d) temperature of the heat-carrying fluid (290° C., 500° C.).

[0085] The physico-mathematical elaborations allowed to obtain the following figures.

[0086] FIG. **9** shows a diagram of the temperature distribution over the solar tube, having a set thickness and irradiation on the emisurface, in a circumferential direction, for a heat-carrying fluid having a convective heat transfer coefficient of 55 W/m^{2}° C. and a fluid temperature of 290° C. The upper curve makes reference to the temperature distribution without fins, the lower one to the presence of fins with a set thickness, height and angular frequence. The presence of fins concentrate in a restricted angular sector makes the temperature trend approximatively constant around the circumferential angle. It is also evident the advantage of the fins on the decreasing temperature mean value of the tube.

[0087] FIGS. **10***a* and **10***b* show the distribution of the thermal zones over the circumferential section of two solar tubes according to the present invention, respectively having fins equispaced over all the circumference or only over a portion of the circumference, as shown in the figure, for a heat-carrying fluid having a heat transfer coefficient of 55 W/m^{2}° C. and a fluid temperature of 290° C. FIGS. **10***a* and **10***b* show therefore the thermal level on the thickness of the tube. In the case of FIG. **10***a*, the minimum temperature is of 330° C., the maximum of 505° C., the temperature intervals in the thermal levels are 9, equispaced. All the geometric parametres of the tube and irradiation conditions on the emisurface are definite for calculation. In FIG. **10***b* the minimum temperature is of 378° C., the maximum of 440° C., the temperature intervals in the thermal levels are 9, equispaced. The advantage of the increase of the minimum temperature, the lowering of the maximum temperature and, therefore, of the reduction of circumferential temperature gradients is evident.

[0088] The figure makes also evident the beneficial effect of concentrating the fins on the side of the irradiated surface. In fact it is outlined the reduction of the temperature mean value of the solar tube and the reduction of the circumferential thermal gradients. The solutions shown for example in FIGS. **10***a* and **10***b* refer to fins arranged in the axial direction of the solar tube.

[0089] FIG. **11** shows the trends of the maximum, mean and minimum temperature of the solar tube obtained by introducing the same number of fins differently spaced over angular sectors. The case indicated with "1" on the abscissa corresponds to an angular distribution over all the circumference; the following cases ("2", "3", "4", . . .) refer to a reduction of the angular sector. Data concerning the boundary conditions are those of FIGS. **10***a* and **10***b*, for a set geometry of the tube and fins.

[0090] FIGS. **12***a* and **12***b* show the temperature distribution obtained for a tube having constant thickness (**12***a*) and for a tube having variable thickness (**12***b*) around the circumference. In particular, the thermal level over the thickness of the tube of FIG. **11***a* shows a minimum temperature of 551° C. and a maximum temperature of 826° C., the temperature intervals in the thermal levels are 9, equispaced. In FIG. **12***b* the thermal level over the thickness of the tube shows a minimum temperature of 646° C.; a maximum of 728° C., the temperature intervals in the thermal levels are 9, equispaced. The minimum thickness of the tube is the same as for FIGS. **10***a* and **10***b* due to needs of resistance to the internal pressure. The advantages in terms of reduction of the circumferential gradients are evident.

[0091] FIGS. **13***a* and **13***b* show the distribution of the thermal zones over the circumferential section of two solar tubes according to the present invention, for a tube with eccentric hole, the size of the cross section of the heat-carrying fluid being the same in the two cases, with the same number of fins, respectively equispaced over all the circumference (FIG. **13***a*) or over part of the circumference (FIG. **13***b*). The thermal boundary conditions are those of FIGS. **10***a* and **10***b*. The minimum thickness of the tube is that of FIGS. **10***a*, **10***b*, defined by the need of resistance to the set internal pressure of the heat-carrying fluid. In the case of FIG. **13***a* the thermal level presents a minimum temperature of 366° C. and a maximum of 460° C., the temperature intervals in the thermal levels are 9, equispaced. In the case of FIG. **13***b* the minimum temperature is of 384° C. and the maximum of 434° C., the temperature intervals in the thermal levels are 9, equispaced. A comparison of the data of the figure makes it evident the advantages relating to thermal gradients: the case of FIG. **13***b* shows a higher minimum temperature, a lower maximum temperature, a reduction of the thermal gradients compared to the case of FIG. **13***a*.

[0092] FIG. **14** shows the results of the analysis of sensitivity made on the data of table 1. The convective heat transfer coefficient on the side of the heat-carrying fluid is of 55 W/m^{2}° C. with the heat-carrying fluid at 290° C. FIG. **14** presents a synthesis of all the cases.

[0093] In particular, the abscissa reports: cases **1-6**: fins height: 5 mm; cases **7-11**: fins height: 10 mm; cases **12-16**: fins height: 15 mm. Each segment **1-2**, **3-4**, etc. . . . represents the situation obtained for a certain number of fins (**1-2**: eight fins; **2-3**: 12 fins; **3-4**: 16 fins; **4-5**: 20 fins; **5-6**: 24 fins), for a set fluid temperature (cases **1-2**, **2-3**, **3-4**, . . . **15-16** Tfluid=290° C.; cases **16-17**, **17-18**, **18-19**, etc. . . . Tfluid=500° C.); changing the angular interval over which the fins are positioned. Each segment **1-2**, **2-3**, **3-4**, . . . **30-31** is comprised of 12 points reporting the dituation from 230° to 90° angular.

[0094] The ordinate reports the temperature of the fins. The lower curves represent the minimum temperature of the tube, the upper curves represent the maximum temperature of the tube and the intermediate thin curves represent the mean temperature of the tube (in each interval **1-2**, **2-3**, **3-4**, . . .): values are referred to the disposition of the same number of fins as a function of the angle. The intermediate thick curves represent the average temperature calculated on the base of an analytical formulation.

[0095] The advantages due to the present invention are evident.

[0096] The increase in the number of fins reduces the average temperature of the tube. For example from case 1 (8 fins) to case 5 (24 fins) the average temperature passes from 580° C. to 480° C.

[0097] The increase of the length of the fins gives its contribution in a selective way (the temperature reduction is much more sensible passing from 5 mm (cases **1-2-3-4-5**) ro 10 mm (cases **6-7-8-9-10**) than from 10 mm to 15 mm (cases **11-12-13-14-15**).

[0098] All the other parametres remaining unchanged, the arrangement of the fins over a restricted angular arc produces a beneficial effect on the temperature of the tube.

[0099] FIG. **15** shows the results due to the presence of fins in a case with a fluid having high heat transfer coefficient (520 W/m^{2}° C.). In figure are reported the trends of maximum temperature (upper curve), minimum temperature (lower curve) and average temperature (intermediate upper curve), beside the average temperature calculated in analytical form (intermediate lower curve), as a function of the height of the fins (the first three sections starting from the left refer to cases with 24 fins in total increasing heights of the fins (1 mm, 2 mm, 3 mm)) and of the number of fins (the fourth section refers to a case with 40 fins of 2 mm height). From the study it can be seen that: a so high heat transfer coefficient implies temperature of the cold wall (non radiated area) closet o the fluid temperature; the maximum temperature passes from 370° C. in absence of fins down to a temperature of 330° C. obtained using 40 fins 6 mm long and 2 mm thick, cutting by half the circumferential temperature gradient with respect to the solution in absence of fins.

[0100] FIG. **16** introduces the further possibility of arranging the fins concentrating them in an angular sector. The three sections of the figure refer to the cases of 12, 24 and 40 fins. For each of these cases **12** configurations were studied with as many distribution angular intervals of the fins themselves: from the right to the left passing from 310° to 90° with an interspace of 20°. The convective heat transfer coefficient heat-carrying fluid wall is of 520 W/m^{2}° C. The height and thickness of the fins are of 5 mm and 2 mm respectively.

[0101] The efficacy of the proposed solutions can be verified. The maximum temperature passes from 370° C. without fins down to the temperature of 328° C. obtained using 24 fins 5 mm long and 2 mm thick, distributed in an arc of 150°, thus reducing the temperature gradient of about 60%. The temperature of the cold wall (non radiated zone) is close to the fluid temperature (290° C.). The maximum temperature passes from 370° C. in absence of fins down to the temperature of 323° C. obtained using 24 fins 5 mm long distributed over an arc of 150° (the minimum value in FIG. **11**, right section) thus reducing the temperature gradient of 70%.

[0102] With reference to FIG. **17** it is shown the situation observed with an eccentric hole (variable thickness of the tube along the circumferential direction), for a set number of fins (24), with a fin distribution angle varying from 310° to 130°. The figure shows three sections representing three different construction solution all providing for an eccentric hole in the solar tube: on the left with a minimum thickness on the side of the solar irradiation, on the right with a minimum thickness on the side of the non radiated surface, on the center with a constant thickness (no hole eccentricity).

[0103] It can be observed, as for all the preceding simulations, that the temperature of the cold wall (non radiated zone) is close to the fluid temperature (290° C.); the maximum temperature passes from 370° C. without fins down to the temperature of 323° C. obtained using 24 fins 5 mm long and distributed over an arc of 150° (minimum value of FIG. **12**, right section) thus reducing the temperature gradient of 70%.

[0104] The technological and operative improvement of the solar tube according to the present invention is evident also for those aspects concerning the deformation on the cross section of the solar tube and on the longitudinal direction (deflection).

[0105] For example, but without limitation of validity, some results are reported which were obtained for a tube having the same features of above, and 4 m long. The tube can be considered a module of a much longer construction (typical length in the applications of thermodynamic solar power are thousands of metres).

[0106] With reference to the deformations on the cross section, table 1 reports the variations on the inner diameter of the solar tube (ΔDEXT), for the two situations considered, with reference to convective heat transfer coefficient between heat-carrying fluid and wall and with a linear thermal extension coefficient α of 12e-6 m/° C. in case of a high heat transfer coefficient (520 W/m^{2}K), the presence of fins reduces the deformations of 10%, whereas for a low thermal exchange coefficient (55 W/m^{2}K) the contribution to the reduction of the deformation of the section is much more sensible (about 50% in the presence of fins, compared to the case without fins).

TABLE 1

Values	Results
Fin height = 5 mm Number of fins = 24 H_conv = 520 W/(m^{2} K)	TMAX = 323° C.; TMIN = 292° C.; ΔDEXT = 0.34 mm
Fin height = 0 mm Number of fins = 0 H_conv = 520 W/(m^{2} K)	TMAX = 371° C. - TMIN = 274° C. - ΔDEXT = 0.37 mm
Fin height = 5 mm Number of fins = 24 H_conv = 55 W/(m^{2} K)	TMAX = 503° C. - TMIN = 407° C. - ΔDEXT = 0.51 mm
Fin height = 0 mm Number of fins = 0 H_conv = 55 W/(m^{2} K)	TMAX = 823° C. - TMIN = 549° C. - ΔDEXT = 0.73 mm

[0107] The same advantage is observed for the tube deflection caused by the longitudinal thermal gradients. The presence of an irradiated cylindrical emisurface, compared to the same surface non radiated, causes the generation of thermal gradients along the axis of the tube causing a deformation of the tube in the direction of the axis (deflection). Table 2 shows the advantages connected with the presence of fins, for the same conditions defining the thermal problem. It can be seen a sensible lowering of the deflection predicted around 50%.

TABLE 2

T fluid	Type of prediction	fMAX without fins [mm]	fMAX with fins [mm]
290° C.	Analytical form	25.8	10.7
290° C.	FEM	28	13
290° C.	FEM	34	15
585° C.	Analytical form	19	10.7

[0108] The present invention has been described for illustrative non limitative purposes, according to its preferred embodiments, but it is to be understood that any variation and/or modification can be made by the skilled in the art without escaping from the relative scope of protection, as defined by the enclosed claims.

[0109] In particular, the proposed solution can be applied to all the tubes used as heat exchangers between an external source of heat non homogeneously distributed around the tube and a fluid flowing inside the tube.

1. Tubular heat exchanger between a source of radiant heat located externally with respect to said tubular heat exchanger and a heat-carrying fluid flowing inside the exchanger, the radiant heat coming from said source being concentrated on a longitudinal portion of the external surface of said tubular heat exchanger, characterised in that it comprises means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger.

2. Tubular heat exchanger according to claim **1**, characterised in that said means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger comprise a plurality of protruding or recessive elements with respect to the internal surface of the tubular heat exchanger, positioned mainly in correspondence of said longitudinal portion of the external surface of the tubular heat exchanger on which the radiant heat coming from said source is concentrated.

3. Tubular heat exchanger according to claim **2**, characterised in that said protruding or recessive elements are positioned along the cylinder generatrixes of the tubular heat exchanger.

4. Tubular heat exchanger according to claim **2**, characterised in that said protruding or recessive elements are positioned along curves that do not follow the cylinder generatrixes of the tubular heat exchanger.

5. Tubular heat exchanger according to any of claims **2-4**, characterised in that the number and/or the surface of said protruding or recessive elements change in a circumferential and axial direction.

6. Tubular heat exchanger according to any of claims **2-5**, characterised in that the shape of said protruding or recessive elements change in a circumferential and axial direction.

7. Tubular heat exchanger according to any of previous claims, characterised in that said means for reducing the temperature gradient along the cross section of the walls of the tubular heat exchanger comprise a thickness of the walls changing in a circumferential direction, between a minimum value in correspondence of said longitudinal portion of the external surface of the tubular heat exchanger on which the radiant heat coming from said source is concentrated, and a maximum value in correspondence of the longitudinal portion of the external surface of the tubular heat exchanger opposed to that on which the radiant heat coming from said source is concentrated.

8. Use of a tubular heat exchanger as defined according to any of claims **1-7** in a concentrating solar plant.

9. Use of a tubular heat exchanger as defined according to any of claims **1-7** in a hydrocarbon reforming plant for producing hydrogen or mixtures of methane and hydrogen or of hydrogen in presence of other hydrocarbons.

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