Feb. 13, 1962 D. BEGGS ETAL 3,021,236

CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

Filed May 28, 1958 10 Sheets-Sheet 1

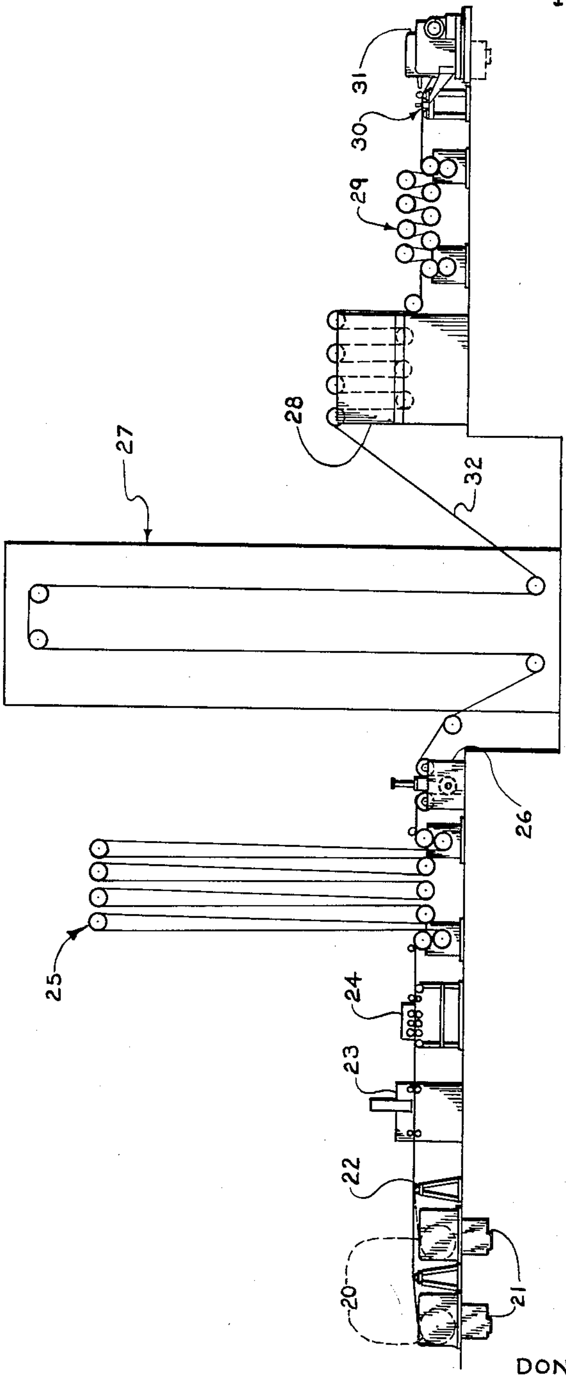

FIG. 1

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CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

Filed May 28, 1958

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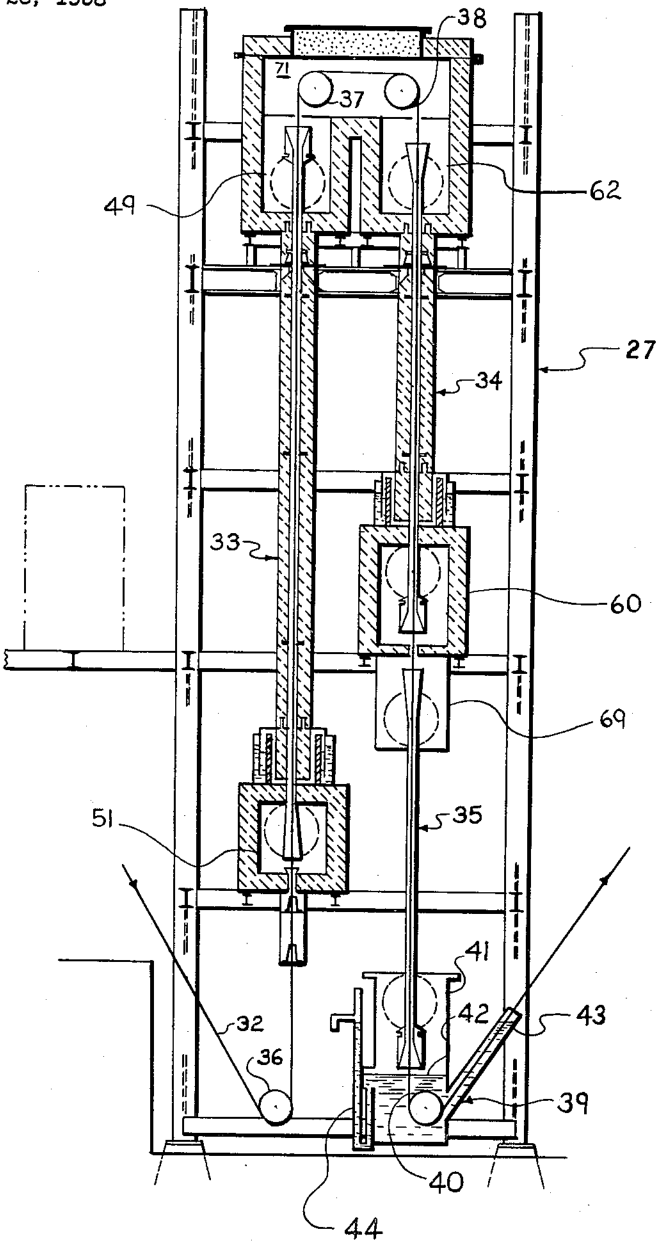

FIG. 2.

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CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

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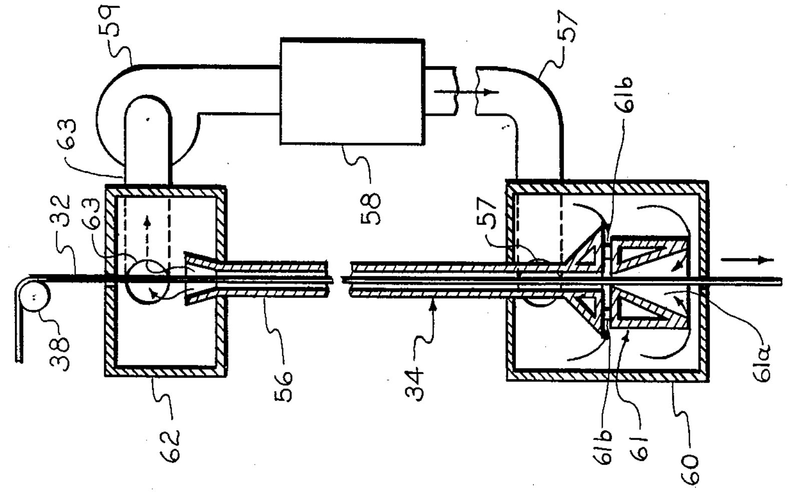

FIG. 4.

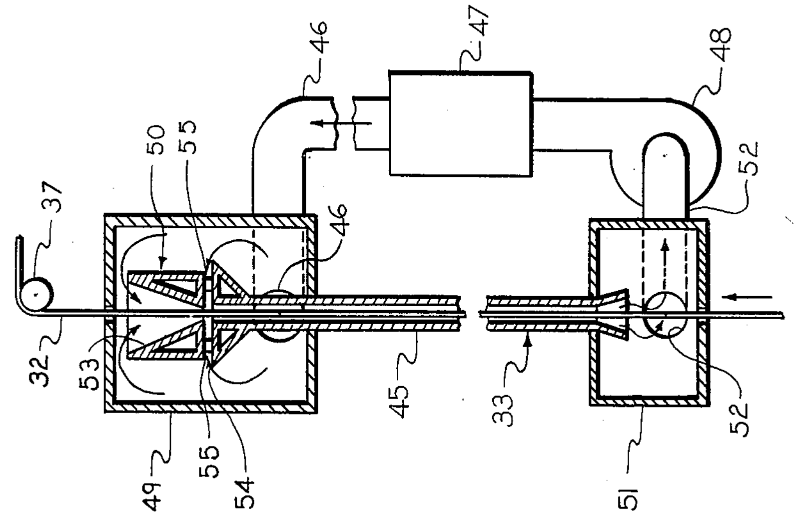

FIG. 3.

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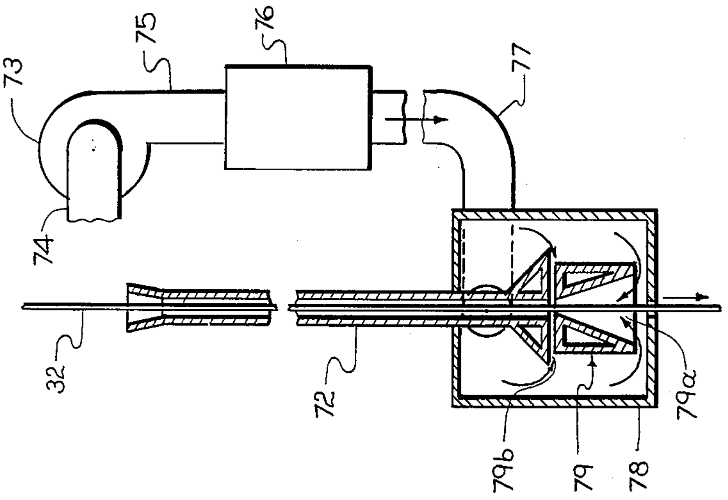

FIG. 6.

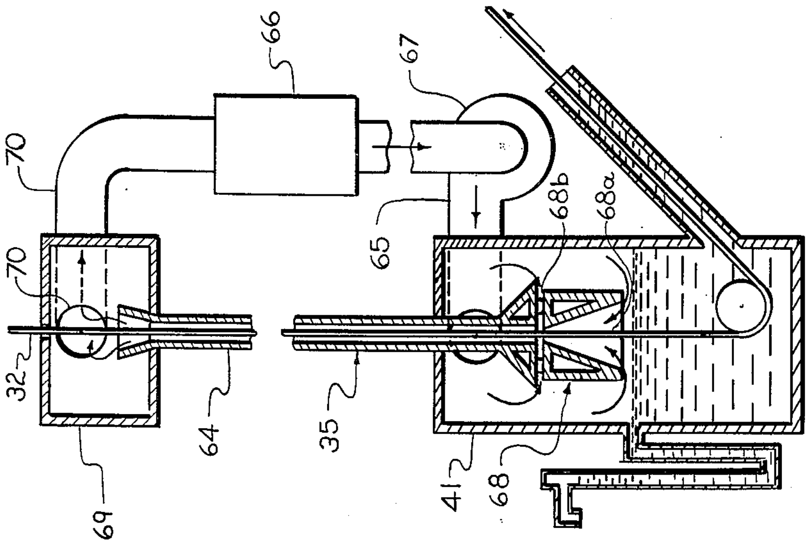

FIG. 5.

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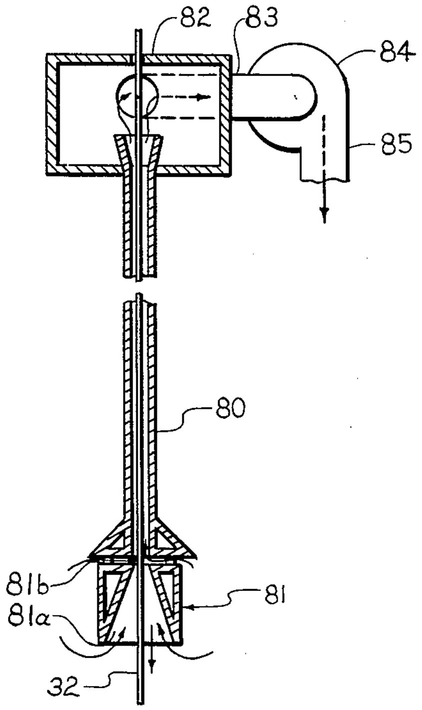

FIG. 7

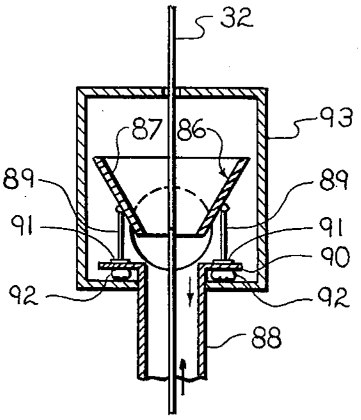

FIG. 8

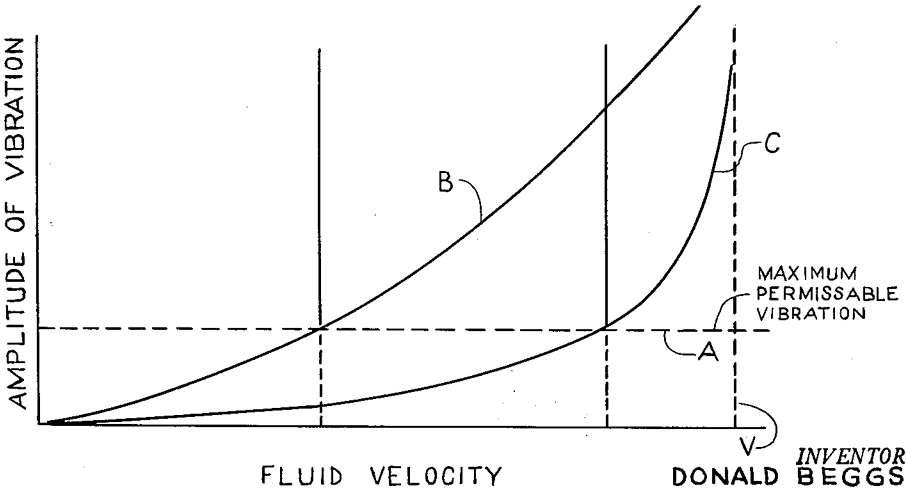

FIG. 9

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CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

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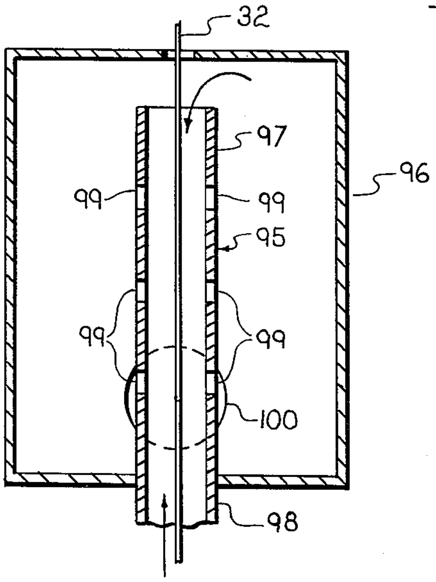

FIG_10_

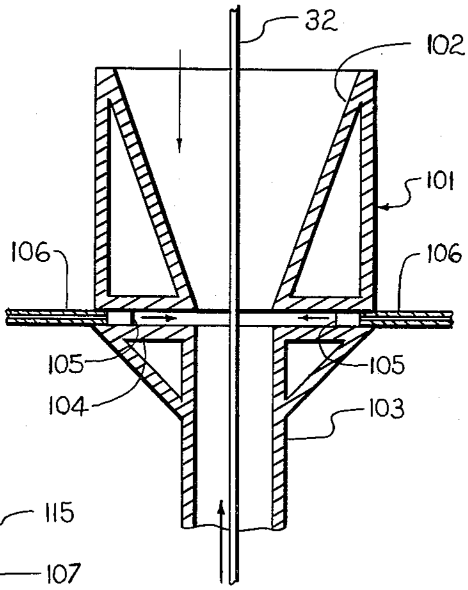

FIG_11_

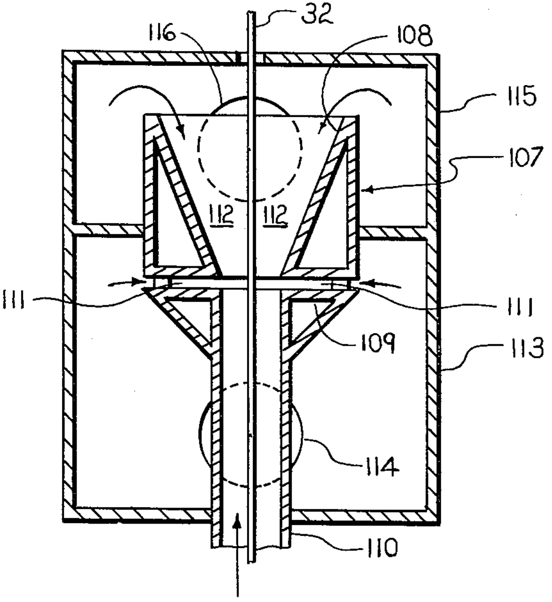

FIG_12_

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CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

Filed May 28, 1958 10 Sheets-Sheet 7

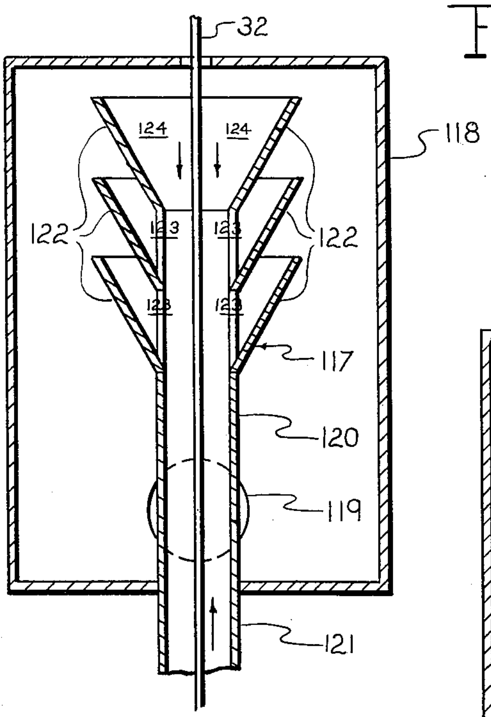

FIG_13_

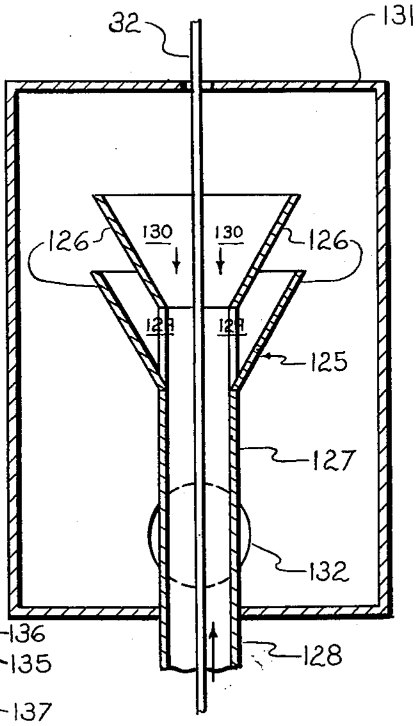

FIG_14_

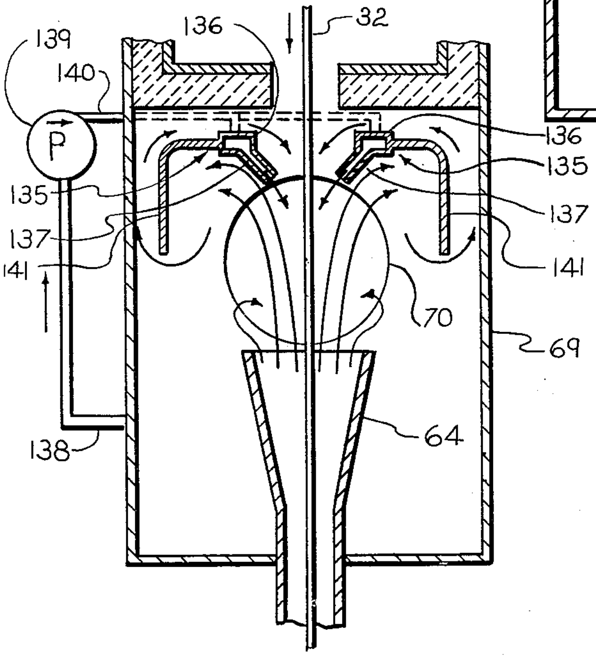

FIG_15_

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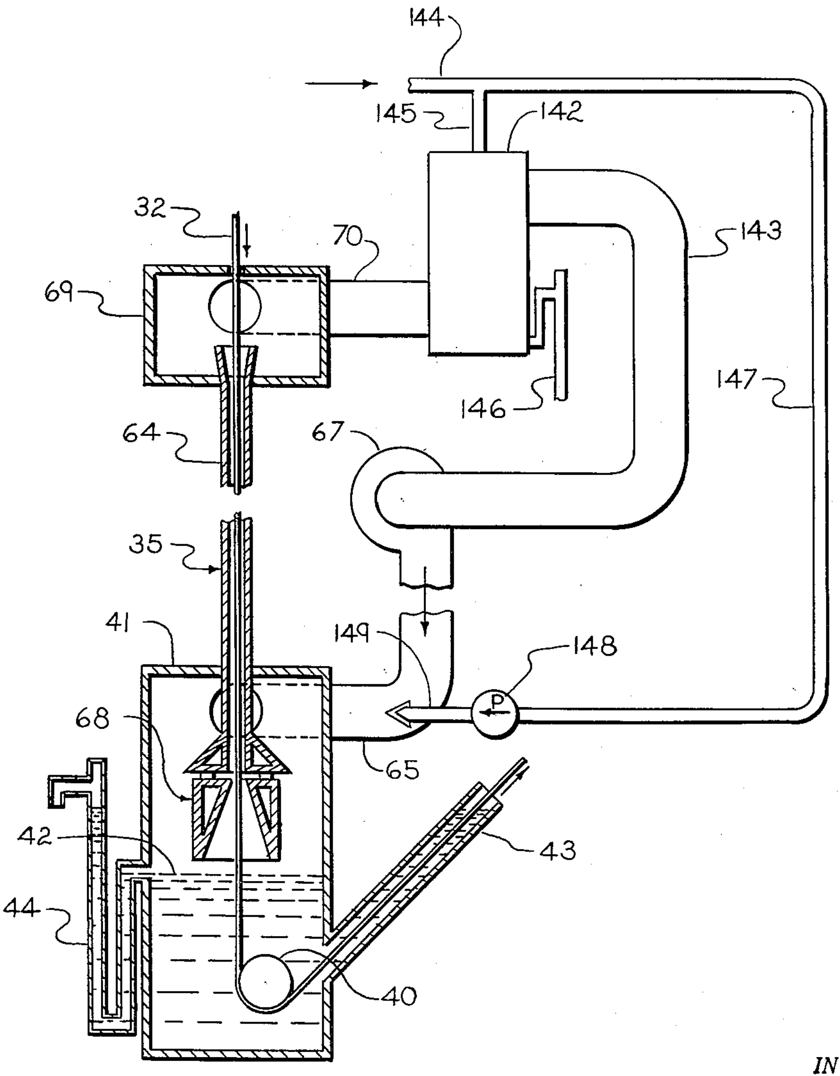
FIG. 16



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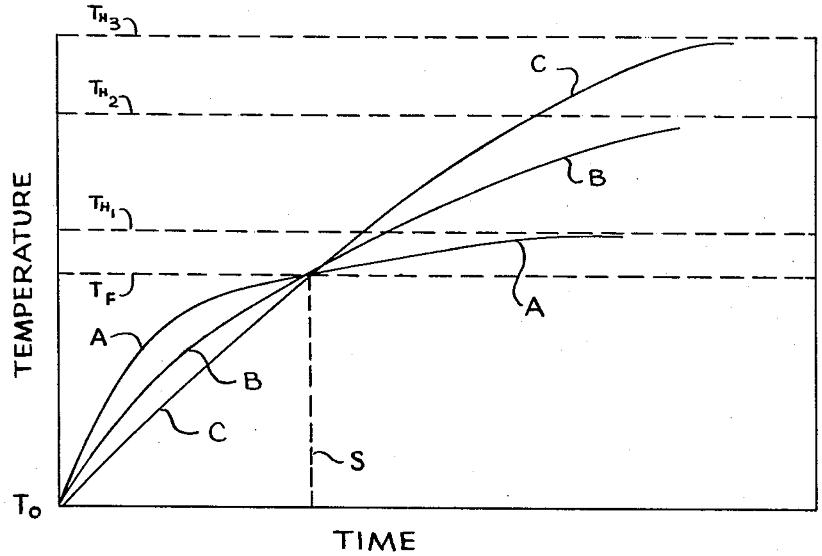


FIG_17_

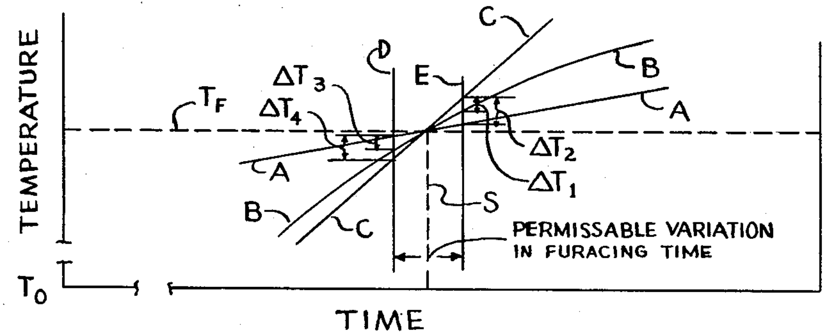
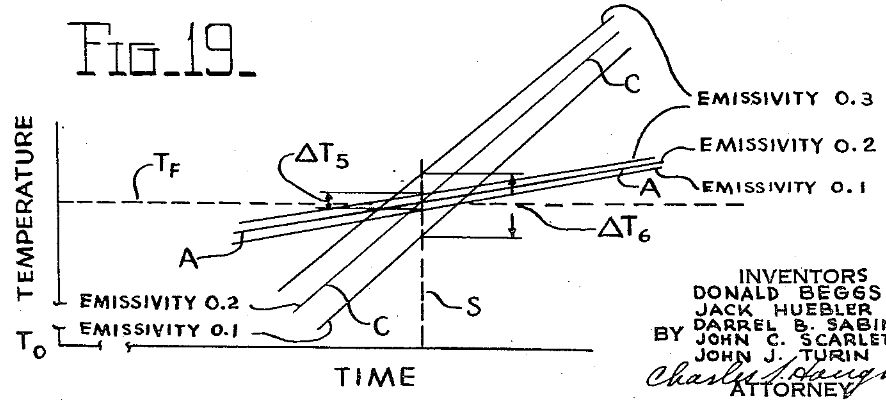


FIG_18_

FIG_19_



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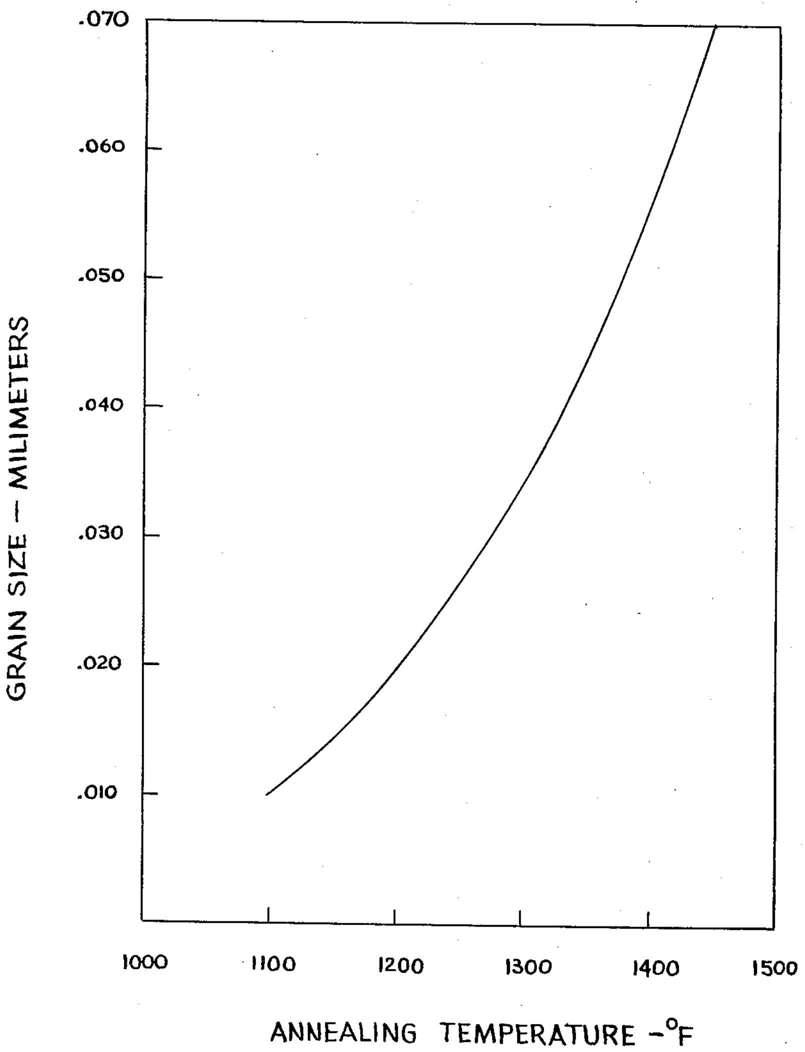

FIG_20_

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Patented Feb. 13, 1962

3,021,236

CONVECTIVE HEAT TRANSFER FURNACE AND METHOD

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Filed May 28, 1958, Ser. No. 738,912

13 Claims. (Cl. 148—13)

This invention relates to a convective heat transfer furnace and method, and, more particularly, to a furnace and method wherein strip work is heated rapidly to a desired temperature, and a heated compressible fluid is circulated in contact therewith, at a rate sufficiently high that convective heat transfer predominates over radiative heat transfer. This application is a continuation-in-part of application Serial No. 530,858, filed August 26, 1955, now abondoned.

A recent development in the art of heat treatment involves rapid heating of work, usually metal, during a short residence time in a heating zone maintained at a temperature substantially above the desired work temperature. Rapid heating of this type achieves numerous advantages including not only economic advantages such as substantial reduction in the floor space that is required to process any given tonnage per hour and smaller furnacing apparatus requirements, but also improved metallurgical results, such as better grain characteristics.

A brief consideration of the factors controlling radiant and convective heat transfer provides a theoretical basis for the experimentally observed fact that radiant heat transfer predominates in a furnacing operation of the type described in the preceding paragraph, which is hereinafter referred to as "high thermal head furnacing." In high thermal head furnacing, which usually is conducted as a continuous operation, the temperature difference between work discharged from the heating chamber and the walls of the chamber is ordinarily in excess of 500° F., and often as much as 1000° F. It is known in the art that the rate of radiant heat transfer from the walls of the chamber to the work passing therethrough is a direct function of the difference between the fourth power of the absolute temperature of the walls and the fourth power of the absolute temperature of the work, while the rate of convective heat transfer from the atmosphere in the furnace to the work is a direct function of the difference between the first powers of these absolute temperatures, assuming the atmosphere temperature to equal the furnace wall temperature. It has been shown in a specific typical instance (see Industrial and Engineering Chemistry, volume 40 No. 6, pages 1995 and following), that at about 2000° F., approximately 80 percent of the heat transferred to work is by radiative heat transfer, and only about 20 percent is by convective heat transfer.

While excellent results have been achieved in many instances using high thermal head furnacing, numerous difficulties have had to be overcome. What has been denominated the "geometry of radiative heating" is responsible for substantial temperature variations in the work, unless suitable provisions are made to compensate therefor. A brief consideration of the high thermal head furnacing of plate stock, even in a heating chamber that is generally circular in cross-section, provides a specific example of one reason why the geometry of radiative heating causes temperature variations in the work. During such furnacing, the longitudinal edge surfaces of the work are heated by radiative heat transfer from the walls of the furnace laterally adjacent the edges, and upper and lower surfaces adjacent the edges are heated by radiative heat transfer from the walls of the furnace thereabove and therebelow. In contrast, the surfaces of the work near the center thereof are heated only by radiative heat transfer from the walls of the furnace thereabove and therebelow. It has been found in actual practice that, as a result of the foregoing fact, temperature differences between the center of a plate and either edge thereof may be as much as 200° F.

It is known that the rate of radiative heat transfer is a direct function of the emissivity of the surface of the material being heated and it has been found that the emissivity of a surface depends not only upon the material, but also upon the surface condition thereof. For example, a carbonaceous surface deposit may double the emissivity of a polished brass surface, and roughening of the surface may have a like effect.

The rate at which work is heated during high thermal head furnacing as ordinarily extremely high. While rapid heating has advantages as discussed above, serious control problems arise if it is attempted to heat treat strip work rapidly in apparatus where radiative heat transfer predominates. A major reason for the extreme difficulty of control with strip work is that no practical way is known to measure strip temperature during such heating. Since temperature differences from point to point in the strip, resulting from the geometry of radiative heating, from variations in emissivity and gauge of the strip, from variations in residence time, or from other such factors cannot be measured, there is no practical way to eliminate the temperature differences during such furnacing. Apparatus according to the invention, however, in one embodiment, makes it possible to heat strip work rapidly, and substantially to eliminate temperature differences from point to point on the strip.

In the heat treatment of cold worked brass strip, it is known to be extremely difficult to achieve uniform grain size by high thermal head furnacing. Brass grain size after annealing, for example, is known to be a function of annealing temperature, other factors being equal. It is believed that variations in emissivity, in thickness, or in both, whether across a width or from point to point along the length of the strip, are responsible for sufficiently wide temperature variations during such furnacing to cause the non-uniform grain size. In addition, the inability to compensate adequately for temperature differences caused by the geometry of radiative heating, or even slight variations in residence times, is believed to contribute to the non-uniform grain size.

The instant invention is based upon the discovery of a convective heat transfer furnace and method enabling the rapid heating, predominantly by convective heat transfer, of continuous strip work. Cold worked brass strip, for example, can be heat treated continuously according to the invention to substantially uniform grain size, indicating substantially uniform temperatures and times at temperatures for all parts of the strip.

It is, therefore, an object of the invention to provide improved apparatus for the rapid heating of continuous strip work.

It is a further object of the invention to provide an improved method for the heat treatment of continuous strip work.

Other objects and advantages will be apparent from the description which follows, reference being had to the accompanying drawings, in which—

FIG. 1 is a diagrammatic representation of a continuous strip processing line including heat treating apparatus according to the invention;

FIG. 2 is a view in vertical section of apparatus according to the invention for rapid heating of continuous strip;

FIG. 3 is a vertical sectional view showing a pre-heat section of the apparatus of FIG. 2, and including an inlet section which minimizes vibrational effects on continuous strip work caused by the introduction of a compressible fluid flowing at high velocity adjacent thereto during heat transfer between the work and the fluid;

FIG. 4 is a vertical sectional view showing the heating portion of the apparatus of FIG. 2;

FIG. 5 is a vertical sectional view showing the cooling portion of the apparatus of FIG. 2;

FIG. 6 is a view in vertical section similar to FIGS. 3–5 showing a modified heating or cooling apparatus according to the invention;

FIG. 7 is a vertical sectional view similar to FIG. 6 showing a further modified form of cooling apparatus;

FIG. 8 is a vertical sectional view of a compressible fluid inlet portion of a heating or cooling chamber according to the invention, and showing a modified form of inlet for minimizing vibrational effects on continuous strip work caused by the introduction of a compressible fluid flowing at high velocity adjacent thereto during heat transfer between the work and the fluid;

FIG. 9 is a diagram showing the advantage of inlets according to the invention for minimizing vibrational effects on continuous strip work;

FIG. 10 is a vertical sectional view showing a modified inlet according to the invention;

FIG. 11 is a view in vertical section showing a still further modified inlet;

FIG. 12 is a vertical sectional view of still another inlet;

FIG. 13 is a view in vertical section of a further modified inlet;

FIG. 14 is a vertical sectional view of an additional inlet;

FIG. 15 is a view in vertical section on an enlarged scale of the compressible fluid discharge portion of the cooling apparatus shown in FIG. 5, and showing details of mechanism for preventing undesired flow of compressible fluid from the cooling apparatus to heating apparatus;

FIG. 16 is a vertical sectional view of a modified form of cooling apparatus according to the invention showing details of means for rapid cooling of strip work;

FIG. 17 is a time-temperature diagram showing heating curves for strip work heated by a compressible fluid flowing at two different high velocities in apparatus according to the invention, and by radiative heat transfer in accordance with the prior art;

FIG. 18 is a diagram similar to FIG. 17, but on an enlarged scale, showing the portions of the three heating curves near the desired final strip temperature;

FIG. 19 is a diagram similar to FIG. 18, but with only two of the three curves represented and showing in addition, the effect of variations in emissivity of the strip work; and

FIG. 20 is a diagram showing average grain size to be expected in 70–30 brass strip work as a function of annealing temperature if all other factors affecting grain size remain constant.

Referring now in more detail to the drawings, and particularly to FIG. 1, a specific continuous strip processing line includes a pair of pay-off reels **20**, serviced by elevators and loaders **21**, a pullover roll **22**, a stitching unit **23**, a cleaner **24**, a looping tower indicated generally at **25**, a tensiometer **26**, a heat treating unit indicated generally at **27**, a pickling unit **28**, an exit looping tower indicated generally at **29**, a shear indicated generally at **30**, and a reel re-winder **31**. Continuous strip work **32** is shown in all parts of the processing line.

Referring now to FIG. 2, the heat treating unit **27** in the strip processing line of FIG. 1 comprises a rapid convective pre-heating apparatus indicated generally at **33**, a rapid convective heating apparatus indicated generally at **34**, and a rapid convective cooling apparatus indicated generally at **35**. Strip **32** entering the unit **27** passes around a roll **36**, vertically upwardly through the preheating apparatus **33**, around a roll **37**, around a roll **38**, and vertically downwardly through the rapid convective heating apparatus **34** and the rapid convective cooling apparatus **35**. Strip discharged from the cooling apparatus **35** passes around a roll **40** through a liquid atmosphere seal indicated generally at **39**, and then is led out of the unit to the looping tower **28**.

The liquid atmosphere seal **39** comprises a chamber **41** which is both a plenum chamber, as subsequently described in more detail, and a liquid seal container. A liquid **42**, which is preferably water, fills the lower portion of the chamber **41**, surrounding the roll **40**, and is forced upwardly by compressible fluid pressure in the chamber **41** into a strip outlet **43** and an overflow **44**.

In the heat treatment of brass strip, temperatures below about 800° F. are sub-critical in that the surfaces of the strip are not damaged by passing over rolls while at such temperatures, and grain size is substantially unaffected. At temperatures higher than about 800° F., however, zinc is vaporized from brass strip, and tends to deposit upon available surfaces, for example, the surface of a roll. Therefore, if brass strip passes over or around a roll while at a temperature higher than about 800° F., the zinc vaporized tends to build up in volcanic-like deposits on the surface of the roll. Such deposits, after a short period of operation, cause permanent damage to strip passing thereover.

The foregoing result may be avoided by passing the strip work through the furnace in a straight line, eliminating the passage over or around rolls, but practical limitations on furnace size limit the maximum production rate of a straight line furnace. In a furnace of any given physical size the production rate may be increased by making two passes through the length, but this requires reversal of direction of the strip around rolls and, as pointed out above, zinc deposits on the roll surfaces if they are located at a point where the temperature is higher than about 800° F.

In the apparatus **27** this dilemma is overcome by the utilization of the pre-heat chamber **33** extending upwardly, in which the strip **32** is pre-heated to a temperature not higher than 800° F. so that its direction is changed by the rolls **37** and **38** without the danger of zinc deposit, and the strip **32** then moves downwardly through the heating apparatus **34** and cooling apparatus **35**, where it is heated to the desired temperature and cooled without contacting a roll. The use of the pre-heating chamber **33** thus provides for a maximum length of heat applying distance and time and, thus, maximum production for a furnace of any given height.

However, the pre-heating apparatus **33** can be eliminated, if desired, and the heating apparatus **34** and cooling apparatus **35** operated in precisely the same way as will be described below to achieve identical results, except, of course, at a lower rate of strip travel and production.

The specific rapid convective pre-heating apparatus **33** (FIG. 3) comprises a heating chamber **45**, which, in the specific embodiment of the invention shown, is of the duct or conduit type, and external duct work **46** including a fluid heater **47** and a blower **48** for circulating a compressible fluid through the heating chamber **45**. The compressible fluid moves upwardly through the duct **46** in the direction of the arrow, and is discharged therefrom into a plenum chamber **49**, and thence flows through an inlet indicated generally at **50** and downwardly through the heating chamber **45** contrary to the direction of movement of the strip **32** therethrough. Compressible fluid discharged from the chamber **45** enters a return plenum chamber **51** and passes from there through a return duct **52** to the low pressure side of the blower **48**.

The inlet **50**, in the specific embodiment of the invention shown in FIG. 3, is a trough-shaped member **53** which converges in the direction of fluid flow toward the heating chamber **45**, and is supported symmetrically with respect to the work, to the heating chamber **45**, and to a flange **54** of the heating chamber **45**. Support for the trough-shaped member **53** is provided by laterally spaced, vertical braces **55** welded or otherwise rigidly attached both to the trough-shaped member **53** and to the passage provided by its open end and through the minor passages formed by slots between the braces **55**. The braces **55** may consist of slotted plates, spaced arms or similar structures providing for structural mounting and the ingress of fluid in lesser quantity into the chamber **45**.

It has been found, however, that the benefit of the particular inlet **50** is retained even though the member **53** is supported in a cocked relationship with respect to either or both the work **32** and the flange **54**, or off center.

Referring now to FIG. 4, the specific rapid convective heating apparatus **34** is substantially identical with the pre-heat chamber **33**, except that it is positioned in inverted relationship so as to provide countercurrent flow of compressible fluid relative to the strip work **32**. The apparatus **34** comprises a heating chamber **56**, ducts **57** for circulating a compressible fluid heated in a heater **58** through the chamber **56**, and a blower **59**, for circulating the compressible fluid. The compressible fluid moves downwardly through the duct **57** in the direction of the arrow, and is discharged therefrom into a plenum chamber **60**, and thence flows through an inlet indicated generally at **61** upwardly through the chamber **56**. Fluid enters the inlet **61** both through its open converging end passage **61***a* and its minor passages or slots **61***b*. Compressible fluid discharged from the chamber **56** enters a return plenum chamber **62** and passes from there through a return duct **63** to the low pressure side of the blower **59**. The inlet **61** is identical in its structural details and operation with the inlet **50** of FIG. 3.

Referring now to FIG. 5, the specific rapid convective cooling apparatus **35** is similar in construction to the pre-heat apparatus **33** and also to the rapid convective heating apparatus **34**, comprising a cooling chamber **64**, ducts **65** for circulating a compressible fluid cooled in a heat exchanger **66** through the chamber **64** and a blower **67** for circulating the compressible fluid. The compressible fluid is blown through the duct **65** in the direction of the arrow, and is discharged therefrom into the plenum chamber portion of the chamber **41**, and thence flows through an inlet indicated generally at **68** upwardly through the chamber **64**. The fluid enters the inlet **68** through its open converging end **68***a* and through its secondary openings **68***b*. Compressible fluid discharged from the chamber **64** enters a return plenum chamber **69** and passes from there through a return conduit **70** to the heat exchanger **66** and then to the low pressure side of the blower **67**. The inlet **68** is identical in its structural details and operation with the inlet **50** of FIG. 3.

In the specific heat treating unit **27** it has been found to be advantageous to employ direct heating in the heaters **47** and **58**, and direct cooling in the heat exchanger **66** to maintain the desired compressible fluid temperatures. Direct heating can be accomplished in the heaters **47** and **58** by combustion therein, in contact with the recirculated fluid, of gas or oil with a desired quantity of air, and produces flue gas as the fluid. Direct cooling in the heat exchanger **66** can be accomplished by spraying water into the heat exchanger, or flowing water therethrough, for example over Berl saddles, in contact with the compressible fluid therein. It has also been found to be advantageous to operate the unit **27** so that there is a small flow of compressible fluid from the plenum chamber **60** of the apparatus **34** into the plenum chamber **69** of the apparatus **35**, as well as from the plenum chamber **62** of the apparatus **34** into a transfer chamber **71** (see FIG. 2), and thence into the plenum chamber **49** of the pre-heat chamber **33**. When the unit **27** is operated in such way there is no opportunity for the compressible fluid in the apparatus **34** to be cooled either by the compressible fluid in the cooling apparatus **35** or in the pre-heat apparatus **33**.

It is also possible to introduce any desired compressible fluid, other than flue gas, or another atmosphere, which may be oxidizing, neutral or reducing, into the pre-heat apparatus **33**, the heating apparatus **34** and the cooling apparatus **35** by supplying the desired compressible fluid, and any necessary make-up, to each circulating system. In such case, when the atmosphere is not flue gas, indirect heating should be employed in the heaters **47** and **58**, for example by means of radiant tubes, or electric heating elements, in order to avoid contamination of the compressible fluid. Either direct or indirect cooling can be utilized in the heat exchanger **66** provided that suitable precautions are taken in the former instance to avoid contamination of the compressible fluid by the coolant. For example, when water is used as a direct coolant, the water can be recirculated and indirectly cooled to avoid the introduction thereinto of oxygen or other material that might be present as absorbed gas in the liquid water and whose presence as a contaminant might be undesirable in the system. It may then be desirable to dehydrate the compressible fluid if water is an undesirable constituent therein.

In each of the rapid convective heat transfer devices shown in detail in FIGS. 3–5, compressible fluid is circulated in a closed system. This represents a preferred arrangement of apparatus according to the invention. However, as shown in FIG. 6, an open system can also be employed to provide the desired compressible fluid flow through a rapid convective heat transfer chamber **72**. In the specific apparatus shown in FIG. 6, a blower **73** draws a compressible fluid into a conduit **74** and discharges the compressible fluid through a duct **75** into a heat exchanger **76**, and thence through a duct **77** into a plenum chamber **78**. The compressible fluid flows from the plenum chamber **78**, in the manner previously described, into the chamber **72** through an inlet indicated generally at **79**, which is structurally identical with the inlets **50**, **61** and **68** (FIGS. 3, 4 and 5), having a major open converging end **79***a* and secondary or minor openings **79***b*. The compressible fluid supplied to the duct **74** can be air, or any other desired fluid provided from a source (not illustrated). The heat transfer chamber **72** can be a pre-heater, heating apparatus, or cooling apparatus, depending upon whether the compressible fluid is heated or cooled in the heat exchanger **76**, as described.

Referring now to FIG. 7, still another type of open compressible fluid circulation system is shown. The apparatus of FIG. 7 comprises a cooling chamber **80** having a compressible fluid inlet indicated generally at **81**, and identical in its structural details with the inlets previously discussed, having a major, open, converging end **81***a* and secondary openings **81***b*. Compressible fluid is drawn from a plenum chamber **82**, through a duct **83** by a blower **84**, and discharged from the blower **84** through a duct **85** to atmosphere. Withdrawal of compressible fluid from the plenum chamber **82** creates a partial vacuum therein which induces a flow of air from the atmosphere through the inlet **81** and the cooling chamber **80**. Because there is no possibility for controlling either the composition or the temperature of the compressible fluid drawn into the chamber **80**, it is possible only to use this arrangement for rapid convective cooling, and only when ambient air temperatures are sufficiently low for this purpose. The inlet **81** makes possible extremely high compressible fluid velocities, without disruption of strip stability, and thereby enables extremely rapid convective cooling of the strip.

A slightly modified inlet similar to those previously discussed is indicated generally at **86** in FIG. 8. The inlet **86** comprises a trough-shaped member **87**, converging in the direction of fluid flow toward a heating or cooling chamber **88** through which the work **32** passes. The member 87 is supported relative to the chamber 88, in generally symmetrical relationship to the work, and to the chamber, and held by spaced arms 89. The arms 89 are supported relative to a flange 90 of the chamber 88 by nuts 91 and wing nuts 92. As in the cases of the earlier described inlets, fluid enters the chamber 88 through the major opening through the member 87 and through secondary openings between the spaced arms 89. In the specific embodiment shown in FIG. 8 the inlet 86 is positioned in a plenum chamber 93. The benefit of the particular inlet 86, like that of the inlets previously discussed, is retained even though the member 87 is supported in a cocked relationship with respect to either or both the work 32 and the chamber 88, or off center.

The various inlet devices discussed above in conjunction with FIGS. 3–8 are particularly advantageous in connection with the so-called strip instability and "flutter" problem. Specifically, in the rapid convective heat transfer furnacing of strip work, it has been found to be difficult to achieve, in the heating and cooling chambers, the compressible fluid velocities necessary to raise the convective heat transfer coefficient sufficiently that the ratio of convective heat transfer to radiative heat transfer is at least 2:1. Instead, it has been found that, as the compressible fluid velocity is increased, strip work in a heating chamber such as a duct, becomes unstable mechanically and begins to vibrate. The vibration at first is not a true resonant vibration because it seems to have no regular frequency, but as wind velocity is increased still further, the amplitude of random vibration increases, until the condition is reached when the vibration amplitude is sufficient that the work strikes the walls of the heating chamber or duct thus irreparably marring the surface of the work.

In some instances, as the compressible fluid velocity is increased, the strip is forced into contact with one of the walls of the heating chamber, or into a twisted condition sometimes in contact with both opposite walls. These positions are limiting conditions, obviously useless for the present purpose since it is impossible to transfer heat by fluid convection to the surface of a strip in contact with a wall, or to transfer heat uniformly to the surface of the strip when the velocity contour of the convection fluid within the duct has been rendered non-uniform by the erratic position of the strip work within the duct.

It is furthermore recognized that even if the limitations of movement introduced by the proximity of the walls were removed, and the compressible fluid velocity through the heating chamber increased, for any specific tension and strip thickness, there would occur a true resonant vibration. This resonant vibration, called flutter, is usually characterized by the vibration amplitude rising almost without limit as the fluid velocity is slightly increased beyond a certain critical maximum. Flutter causes such large destructive forces to come into play as to cause violent rupture of the strip work. When it is possible to stabilize the strip sufficiently in a duct so that the convection fluid velocity may be increased to the velocity which causes flutter, the ultimate limitation in convective heat transfer to the strip is attained at a velocity slightly smaller than that inducing flutter. The unstable vibrations discussed above, however, in ordinary installations, prevent the utilization of compressible fluid velocities even approaching the convection fluid velocity capable of causing resonant flutter.

The vibration condition in previously known strip heating chambers is represented by curve B of FIG. 9, which is a plot of amplitude of vibration of strip work passing through a duct-type heating chamber against fluid velocity in such chamber. It will be noted that curve B passes through the origin of the plot, but has a relatively constant positive slope, indicating that amplitude of vibration is a direct, almost linear function of fluid velocity through the duct, at least up to dotted line A, which represents a maximum permissible amplitude of vibration for any particular duct, being a vibration amplitude arbitrarily selected for practical reasons equal to one-half the duct thickness. Curve B is typical for heating chambers having straight or flared fluid inlets, including Venturi trough and bell flares, but with no means for equalizing pressure on opposite sides of the strip, as subsequently discussed in detail.

Amplitude of unstable vibration of work in strip form as a function of fluid velocity in a duct-type heating chamber provided with one of the inlets shown in FIGS. 3–8, and described above in connection therewith, is represented by curve C of FIG. 9. It will be observed that curve C, throughout the range of permissible fluid velocities, is displaced substantially to the right of curve B, indicating that at any permissible maximum amplitude of unstable vibration, a substantially increased convection fluid velocity is permissible without an increase in vibration amplitude. Therefore, using any of the inlets shown in FIGS. 3–8, higher fluid velocities through the duct heating chamber are permissible. Since the effective convective heat transfer coefficient is a direct function of fluid velocity, and since the amount of heat transferred by convective heat transfer is a direct function of effective convective heat transfer coefficient, a higher ratio of convective heat transfer to radiative heat transfer, other factors being equal, is possible in furnacing apparatus provided with the inlets shown in FIGS. 3–8, than in such apparatus provided with conventional inlets.

Curves B and C of FIG. 9 represent amplitude of unstable vibration as a function of fluid velocity through a heating or cooling chamber for conventional inlets and for inlets as shown in FIGS. 3–8, respectively, other variables being held constant. In FIG. 9 the dotted vertical line V, represents the convective fluid velocity in contact with the strip required to cause the resonant "flutter" variation. It has been found, however, that the fluid velocity, V, required to cause resonant flutter vibration in strip work is also a complex function of the tension applied to the strip passing through a duct heating chamber, being increased as the tension is increased.

Once the problem of unstable pre-flutter vibration has been eliminated by employing one of the inlets of the present invention, the fluid velocity required to induce flutter in a specific instance may be increased by raising the tension force applied to the strip work 32, passing through the heat treating unit 27.

In another aspect, therefore, the invention contemplates the provision of means, such as the tensiometer 26 shown in FIG. 1, for maintaining the work 32 under high tension as it passes through a heating or cooling chamber. The maximum permissible tension that can be applied to heated work under any given set of conditions depends upon many factors. For example, in the case of brass strip, it has been found that the work is likely to be unsatisfactory, for example because of permanent mechanical distortion or gauge inaccuracy, if the work is elongated more than about 1/10 of 1 percent during the course of a furnacing operation such as an annealing. In any event, however, in one embodiment, the invention contemplates the provision of means, such as the tensiometer 26, for maintaining the strip work 32, as it passes through a heating or cooling chamber, under high tension ranging from about 50 percent to the limit of 100 percent of the maximum tension permissible under the conditions prevailing.

Although it is not fully understood why the inlets shown in FIGS. 3–8 reduce the amplitude of vibration and provide mechanical strip stability at any given fluid velocity below that inducing flutter as shown in FIG. 9, it is desired to present a theoretical explanation for this phenomenon in order to make as full and complete as possible a disclosure of this aspect of the invention. The following theoretical explanation, therefore, is presented solely for the purpose of further illustrating and disclosing, and is in no way to be construed as a limitation upon the invention. It will be observed in FIGS. 3 and 4, for example, that the strip work **32** passing through the heating chamber **45** or **56** may be considered to be a diaphragm separating such heating chamber into two portions. Whenever a slight pressure differential occurs between the two sides of the diaphragm (strip work), there is a tendency for the diaphragm to move in the heating chamber to balance the pressures. Such movement, however decreases the cross-sectional area of the low pressure side of the diaphragm, thus tending instantaneously to increase the velocity of compressible fluid there, and still further decrease the pressure. The further pressure decrease causes still further movement of the diaphragm, with the result that any pressure inequality starts a chain of dynamically unstable consequences which exaggerates the effect of the original difficulty.

It is believed that the effectiveness of the inlets shown in FIGS. 3–8 in stabilizing the strip and reducing the amplitude of vibration at any given fluid velocity smaller than that causing flutter demonstrates that pressure inequalities between the two sides of the diaphragm are most likely to be caused by the dynamic conditions prevailing at the fluid inlet to the heating or cooling chamber.

Each of these inlets provides a plurality of converging passageways for compressible fluid from a plenum chamber or from atmosphere to a heating or cooling chamber. Each inlet also provides at least one major passage and minor alternate passages from the plenum chamber or from atmosphere to the interior of the heating or cooling chamber near the fluid inlet end thereof. The minor passages are, in essence, pressure stabilizers, in the sense that compressible fluid from the plenum chamber can flow therethrough to either side of the diaphragm (strip work) with the result that the effects of pressure inequality between the two sides of the diaphragm are counteracted by flow of fluid from the plenum chamber or from atmosphere to the low pressure side of the diaphragm, with resulting strip stabilization and elimination of sub or pre-flutter vibration, because no strip movement is required to equalize pressures on both sides.

Careful study of FIGS. 3–8 reveals that the various inlets which have been found to be advantageous have in common the feature that they provide a plurality of paths for compressible fluid flow from the plenum chamber exterior of the heating or cooling chamber into the interior of the heating or cooling chamber. In the inlets shown in FIGS. 3–8 compressible fluid has one flow path interior of a trough-shaped member, which path, itself, converges in the direction of fluid flow, as previously described, and also has at least one other path which converges, relative to the first path, such other converging path or paths being defined by the space between the downstream extremity of the trough-shaped member and the upstream extremity of a flange on the heating or cooling chamber.

Optimum results have been achieved, experimentally, using an inlet providing a plurality of converging paths as shown in FIGS. 3–8 for compressible fluid flow from the exterior to the interior of a heating or cooling chamber, when there has been actual flow of compressible fluid along a plurality of converging paths. With inlets of the types shown in FIGS. 3–8 it has further been found that optimum results, in the form of strip stability, as previously discussed in detail, have been achieved when the ratio of volume of fluid flowing through the converging major inlet passages to volume of fluid flowing through the slots or minor passages has been from about 1½:1 to about 3:1, most desirably from about 2:1 to about 2½:1.

It will be apparent from the foregoing discussion that numerous inlets other than those shown in FIGS. 3–8 can be utilized to increase strip stability by equalizing pressures on either side of strip work in a heating or cooling chamber through which a compressible fluid travels at high velocity in a direction generally parallel to the strip surfaces. One such inlet is indicated generally at **95** in FIG. 10. The inlet **95** is shown in a plenum chamber **96**, and comprises an extension **97** of a heating chamber **98**. Openings **99** in the extension **97** provide minor passages for the flow of compressible fluid which converge with respect to a major fluid flow passage into the open end of the extension **97**, as indicated by the arrow. The openings **99** may be continuous or interrupted peripherally, not necessarily being symmetrically arranged with respect to the strip work. The compressible fluid is supplied to the plenum chamber **96** through a supply duct **100**. The inlet **95** can be used in conjunction with a plenum chamber in accordance with the showings of FIGS. 3–6, or with atmosphere as the plenum chamber in accordance with the showing of FIG. 7, and whether fluid flow is induced by supplying compressible fluid under pressure to the plenum chamber **96** or by drawing a vacuum on a plenum chamber surrounding the fluid discharge end (not illustrated) of the chamber **98**.

Still another embodiment of an inlet according to the invention is indicated generally at **101** in FIG. 11. The inlet **101** comprises a trough-shaped member **102** similar to the member **53** shown in FIG. 3, and which converges in the direction of fluid flow toward a heating chamber **103**. The member **102** is supported symmetrically with respect to the work **32**, to the heating chamber **103**, and to a flange **104** of the heating chamber **103** by members **105** welded or otherwise rigidly attached both to the trough-shaped member **102** and to the flange **104**. Fluid flow in the direction of the vertical arrow is effected through the trough-shaped member **102** in any of the ways previously discussed. Additional fluid flow from the lateral openings and following paths converging into the heating chamber is induced, for example, by supplying compressible fluid under pressure to conduits **106**, preferably from a common header or plenum chamber (not illustrated). The conduits **106** communicate with a space between the lower portion of the trough-shaped member **102** and the flange **104**, so that the fluid supplied therethrough flows into the chamber **103** in generally the same manner as in the inlets of FIGS. 3–8, except that the rate of such flow can be regulated by varying the fluid pressure in the conduits **106** by any suitable means (not illustrated). The trough-shaped member **102** of the inlet **101** can, if desired, be open to atmosphere, in which case it is useful only as a cooling chamber, or it can be positioned in a plenum chamber and used either for heating or cooling of strip work.

An additional inlet indicated generally at **107** in FIG. 12 is similar to the inlet **101**, comprising a trough-shaped member **108** positioned in spaced relationship with a flange **109** of a heat transfer chamber **110** in the manner previously described. Minor fluid flow conduits **111** which provide paths that are convergent with respect to major fluid flow paths **112** through the trough-shaped member **108** on either side of the strip work **32** are enclosed within a plenum chamber **113** to which a compressible fluid is supplied under pressure from a duct **114**. The upper portion of the trough-shaped member **108** is positioned in a plenum chamber **115** to which a compressible fluid is supplied through a duct **116**. By suitable control of the relative pressures of the compressible fluids supplied to the plenum chamber **113** through the duct **114**, and to the plenum chamber **115** through the duct **116**, the ratio of volume of compressible fluid flowing through the passages **112** to that flowing through the passages **111** can be regulated as desired.

Still another modified inlet is indicated generally at **117** in FIG. 13, where it is shown in a plenum chamber **118** to which a compressible fluid under pressure is supplied through a conduit **119**. The inlet **117** comprises an extension **120** of a heating or cooling chamber **121**. A plurality of conical fin members **122** extend angularly upwardly therefrom, on opposite sides of slots

123 in the wall of the extension 120. The slots 123 provide a plurality of minor fluid flow paths which converge with respect to major paths 124 through which fluid flows in the direction of the arrows.

An inlet indicated generally at 125 in FIG. 14 is generally similar to the inlet 117 of FIG. 13, except that only two conical fins 126 are provided near the end of an extension 127 of a heat exchange chamber 128, so that one minor fluid flow path through slots or openings 129 converging with respect to major fluid flow paths 130 is provided on either side of the strip work 32. In the specific embodiment of the invention shown, the inlet 125 is positioned in a plenum chamber 131 to which a compressible fluid under pressure is supplied through a duct 132.

The various inlets shown in FIGS. 10–14 represent modifications of a preferred species of inlet according to the invention wherein there are a plurality of converging fluid flow paths. Each of these inlets is suitable for use in a closed system as shown in FIGS. 3–5, or in an open system as shown in FIGS. 6 and 7, in place of the inlet specifically shown in these various figures.

It will be apparent, however, that compressible fluid inlets other than those specifically shown and discussed also can be used if they provide means for equalizing the pressure between the two sides of the strip work without causing appreciable lateral strip movement in the rapid convective heating or cooling chamber. In order to be effective pressure equalization must, however, be rapid so that a static pressure communication between the two sides of strip in a heating chamber is effective for strip stabilization if pressure differences are rapidly eliminated, and any resonant condition in the static interconnection avoided. When such pressure equalization is accomplished by flow modulation of a compressible fluid through a plurality of converging paths into the interior of a heating or cooling chamber, the necessary rapidity of pressure equalization is achieved, and any problem of resonance is eliminated. It is principally for this reason that the preferred inlets provide a plurality of converging compressible fluid flow paths.

As has been discussed above, it is usually preferred, in the rapid convective heating apparatus 34, and also in the rapid convective cooling apparatus 35, that the compressible fluid flow countercurrent to the strip 32. When the unit 27 is operated in this way, relatively cold compressible fluid moving at high velocity adjacent the surfaces of the strip 32 in the cooling chamber 64 has a substantial tendency to continue such movement and to pass from the cooling chamber 64, through the upper portion of the plenum chamber 69 and into the plenum chamber 60 of the apparatus 34 where it cools the compressible fluid supplied to the heating chamber 56 of the apparatus 34, thus substantially reducing the effectiveness thereof.

In a preferred embodiment, apparatus according to the invention includes buffer jets indicated generally at 135 in FIG. 15 for preventing admixture of relatively cold compressible fluid in plenum chamber 60 from the plenum chamber 69 with hot compressible fluid used for heating (see FIG. 2). The buffer jets 135 comprise conduit portions 136 extending across the upper portion of the plenum chamber 69, and generally parallel to the strip work 32. Spout members 137 are structurally integral with the conduit portions 136, and are directed downwardly at an angle of approximately 40° with respect to the strip work 32, in the embodiment of the invention shown. Compressible fluid is withdrawn from the plenum chamber 69 through a pipe 138 and discharged by a pump 139 through a pipe 140 to each of the conduit portions 136. The compressible fluid is discharged through the nozzles 137 against, and across the width of, the strip work 32 in streams which oppose the flow of compressible fluid from the cooling chamber 64 into the plenum 60 (FIG. 2). These opposing fluid streams from the spouts 137 force cold compressible fluid away from the surfaces of the strip 32, and cause a localized fluid circulation around shroud plates 141, as indicated by the arrows, in addition to a general compressible fluid movement into the duct 70. The buffer jets 135, therefore, prevent cooling of the compressible fluid supplied to the plenum chamber 60 for heating of strip work in the apparatus 34, and enable the operation of the apparatus with a general compressible fluid flow from the plenum chamber 60 into the plenum chamber 69.

A preferred furnace arrangement for annealing of brass strip is shown in FIG. 2 wherein the heating and cooling are accomplished in a single vertical pass of the strip to avoid marking the strip due to roll pickup problems. As shown more clearly in FIG. 16, the strip work 32 enters a water seal 42 prior to passing over a submerged roll 40 therein and thence through duct 43. It is well known that hot strip has a tendency to warp and buckle when quenched in water; to avoid such distortion, it is necessary to cool the strip to about 250° F. to 300° F. prior to quenching in water. The cooling to 250° to 300° F. cannot be at too drastic a rate, or distortion will result; yet, if the cooling rate is too conservative, or slow, the furnace cooling chamber becomes excessively long (or has too low a capacity in weight of strip cooled per hour).

It has been found by experiment that even the relatively high convective cooling rates made possible by the increased fluid velocities resulting from this invention are conservative from a strip distortion point of view. To achieve a maximum cooling rate to the desired temperature, about 250° F. to 300° F., without encountering strip distortion, it is preferred to use a two-phase convective cooling system where a liquid such as water is supplied as a fine mist to the cooling, compressible fluid used in the cooling chamber.

When water alone is sprayed onto hot strip, heat transfer coefficients of the order of 1000 to 2000 B.t.u. per sq. ft. of strip surface per hour per degree F. temperature difference between the strip and water temperatures are attained. When convective heat transfer cooling is employed, coefficients of 20 to 30 are about the maximum attainable short of encountering resonant flutter in the strip.

Experiments show that brass strip work will tolerate a cooling coefficient of 50 to 200, when cooled from red heat to about 250° F., without buckling or distortion.

The two-phase, or mist, convection cooling system will effect cooling coefficients of 50 to 200, producing a highly efficient and compact convection cooling section. Such coefficients can be obtained when water mist is injected into a flue gas stream and the mixture is passed longitudinally through the cooling chamber at velocities below that which causes resonant flutter. By maintaining the liquid content of the cooling compressible fluid relatively low, preferably about 0.1 to 0.5 pound of mist per pound of compressible fluid, and circulating a sufficient weight of mixture per weight of strip cooled to maintain a portion of the mixture in liquid form for a substantial portion of the cooling chamber (preferably the compressible cooling fluid leaving the cooling chamber should contain a substantial portion of mist) very rapid and uniform cooling is attained without distortion, and at heat transfer coefficients intermediate those heretofore obtainable in production of distortion free strip work.

While the mechanism of mist convection cooling may be somewhat controversial, it is believed that the high velocity compressible fluid prevents collection of insulating steam pockets on the strip surface, and that the latent heat of vaporization of the droplets of mist adjacent the strip acts to maintain local fluid temperatures somewhat lower than otherwise attained. It is noted, however, that the degree of additional cooling of the circulating compressible fluid by vaporized mist is not sufficient by itself to explain the higher cooling rates attained.

In FIG. 16, the compressible fluid within the duct 70 is admitted to a heat exchanger 142 so that the fluid flows upwardly through the heat exchanger to a fluid discharge near the top of the exchanger and passes through a duct 143 to the low pressure side of the blower 67. Water or other vaporizable liquid that is sufficiently free of absorbed contaminants, as described, is supplied to a pipe 144 from any suitable source (not illustrated), and passes from thence through a pipe 145 into the upper portion of the heat exchanger 142 through which the liquid flows in direct contact with, and countercurrent to, the compressible fluid. An overflow pipe 146 is provided near the bottom of the heat exchanger 142, and below the compressible fluid inlet to the heat exchanger. A desired liquid level is thus maintained in the bottom of the exchanger 142 to provide a gas seal, and excess liquid is discharged through the overflow pipe 146 and either discarded, or cooled and recirculated to the pipe 144. Water or other coolant from the pipe 144 is also passed through a pipe 147 to a pump 148 and sprayed as a fine mist by an atomizing nozzle 149 into the stream of compressible fluid in the duct 65. The mist of water or other coolant is carried through the duct 65, the plenum portion of the chamber 41, and the cooling chamber 64 by the stream of compressible fluid. The strip work 32 can be immersed in the atmosphere seal 42 without appreciable thermal shock when the mist of liquid coolant is employed.

The advantage of heating work in strip form in apparatus according to the invention, which provides a sufficiently high rate of flow of compressible fluid adjacent the strip work that the ratio of heat transferred by convective heat transfer to heat transferred by radiative heat transfer is at least 2:1, will be apparent from a brief consideration of FIGS. 17–19. Curves A, B and C of FIG. 17 show the temperature of strip work, from an inlet temperature of T_0 to a final temperature of T_f, as a function of furnacing time, the total time being S (represented by the dotted line). Curve A shows such time-temperature relationships when the compressible fluid is circulated through the heating chamber at a velocity just less than that required to cause flutter of the strip work, and the temperature of the compressible fluid is T_{H1}. Curve B is similar to curve A, but represents the lower limit of operation of apparatus according to the invention, where the velocity of heated compressible fluid has been reduced to an extent such that the ratio of heat transferred by convective heat transfer to heat transferred by radiative heat transfer is 2:1. When the velocity of compressible fluid flow has been decreased, as in the situation represented by curve B, it is necessary to raise the fluid temperature to a temperature T_{H2} higher than T_{H1} in order to heat the strip work to the temperature T_f in time S, because the overall coefficient of heat transfer has been lowered. When the compressible fluid velocity is lowered still further below that necessary to effect heating as represented by curve B, the overall coefficient of heat transfer to the strip work is still further reduced, so that a still higher fluid temperature T_{H3} is required to heat the work from T_0 to T_f in time S. This situation is represented by curve C of FIG. 17, which is a typical curve for rapid, high thermal head heating of strip work in apparatus known prior to the instant invention.

By reference to FIG. 18, where the portions of curves A, B and C adjacent their intersections with the dotted line T_f are shown on an enlarged scale, it will be seen that the temperature differences to be expected in strip work as a result of variations of residence time in the furnace are substantially higher in high thermal head furnacing where radiative heat transfer predominates than in furnacing operations conducted in apparatus according to the invention. Substantially smaller temperature differences result from residence time variations when the ratio of convective to radiative heat transfer is at least 2:1, as represented by curve B, and the differences are even less under the limiting condition at a fluid velocity just short of that at which resonant flutter occurs, as represented by curve A. Since it is impossible to control residence time of the strip in a furnace absolutely, apparatus according to the invention has the advantage over high thermal head furnacing apparatus that variations in residence time affect final work temperature less. For example, if the furnacing apparatus is designed to provide a residence time of S plus (E minus S) or, minus (S minus D), the improved temperature control (over conventional high thermal head furnacing) achieved by operating at a compressible fluid velocity sufficient to give a ratio of heat transferred by convective heat transfer to heat transferred by radiative heat transfer of at least 2:1 is represented by the increment ΔT_1 or ΔT_3. Similarly, the improved temperature control achieved by operating apparatus according to the invention at a compressible fluid velocity just short of that required to cause resonant flutter is represented by the increment ΔT_2 or ΔT_4.

In a specific instance it has been determined that an increase by 10 percent in residence time of brass strip in a furnace will cause an increase in temperature of the strip of approximately 50° F. in a high thermal head furnace, but of only about 25° F. when, according to the invention, a compressible fluid velocity such that the ratio of heat transferred by convection to heat transferred by radiation is approximately 3:1 is used. Variations in gauge of the strip work being heat treated have the same general effect as variations in residence time. A 10 percent decrease in gauge is approximately the equivalent of a 10 percent increase in residence time, and a 10 percent increase in gauge is approximately equivalent to a 10 percent decrease in residence time. Thus, in FIG. 18, ΔT_1, ΔT_2, ΔT_3, and ΔT_4 represent the variations in temperature to be expected as a consequence of gauge variations, if the furnace residence times remain constant.

Variations in strip emissivity cause even greater variations in strip discharge temperature from a high thermal head furnacing apparatus than do variations in gauge of the strip or residence time. FIG. 19 shows on an enlarged scale the portions of curves A and C adjacent their intersections with the dotted line representing T_f, and, also, companion curves representing temperatures if the strip emissivity increases from 0.2 (curves A and C) to 0.3, or decreases to 0.1. As has been stated, such variations in emissivity must be anticipated. As shown in FIG. 19, when apparatus according to the invention is operated with a compressible fluid velocity just short of that which would cause resonant flutter, a substantially decreased temperature variation (ΔT_5) in discharge strip temperature results from emissivity variations between 0.1 and 0.3 than with conventional high thermal head furnacing (ΔT_6). The relationships for curve B of FIG. 17 are not represented in FIG. 19 in order to avoid confusion; the band of temperatures to be expected from variations in emissivity from 0.1 to 0.3 would be narrower than that shown with curve C, but somewhat broader than that shown with curve A. In a specific instance it has been determined that emissivity variations between 0.1 and 0.3 in strip work will result in temperature variations in a high thermal head furnacing apparatus of more than 350° F., while, in apparatus according to the invention, operated to give a ratio of heat transferred by convection to heat transferred by radiation of about 3.0, the temperature variation will be less than 100° F.

It will be observed from FIG. 19 that the temperature range resulting from emissivity variations decreases in magnitude with increases in furnacing time and strip temperature. This is true for the range with curve A, as well as for the range with curves B and C. If furnacing were continued for an infinite time, the three curves would become identical, and the strip work temperature at all points would equal the furnace temperature. Decreased temperature variations as a result of differences in strip emissivity in apparatus according to the invention are achieved not only because radiation is a smaller factor in heating the strip work, but also because heating is conducted so that a much closer approach to temperature equilibrium in the strip work is achieved in apparatus according to the invention than in high thermal head furnacing. It will be observed from FIG. 17 that the shapes of the three curves A, B and C are generally the same, except that they approach different limits, T_{H1}, T_{H2} and T_{H3}, respectively. The proximity of strip temperature to equilibrium conditions after time S (or at discharge) depends upon temperature head (T_{H1} minus T_f, T_{H2} minus T_f, or T_{H3} minus T_f, respectively), the smaller the temperature head the closer the approach to temperature equilibrium in the strip. However, a high compressible fluid velocity is required to heat the strip work in time S if the fluid is at a low temperature.

In one aspect, therefore, the invention contemplates a method for the continuous heating of strip work by heat transfer between a compressible fluid and the strip while the compressible fluid is forced to flow, in a direction generally axial of the strip, at a velocity sufficiently high that the ratio of heat transferred by convection to heat transferred by radiation is at least 2:1. It is preferred that the temperature difference, in degrees F., between the compressible fluid and strip work after heating thereby, be not more than about 30 percent of the number of degrees that the strip work is heated, and most preferred that such temperature difference be not more than about 25 percent. It is also preferred to use a compressible fluid velocity that is not only sufficient to give a ratio, as indicated, but also sufficient to accomplish the necessary heating in the available time at a relatively low fluid temperature, as indicated. In this way, a close approach to temperature equilibrium in the strip at the time of discharge thereof from the heating apparatus is achieved.

The following example is presented solely for the purpose of further illustrating and disclosing the invention, and is in no way to be construed as a limitation thereon.

Example

A furnace of the type shown in FIG. 2 with a duct heating chamber having a cross-section area of 1.5 square inches was used to anneal samples of cartridge brass work in strip form, 0.005 inch thick, and having different histories and different grain sizes. The brass strip, as received, had a grain size, in millimeters, ranging in various samples from 0.010 to 0.090, and was reduced in various samples from 83 percent to 40 percent by cold rolling prior to furnacing. Each sample of strip, after reduction, was subjected to furnacing for fifteen to thirty seconds, with flue gas at temperatures from 1050° F. to 1450° F. passed through the heating chamber at velocities ranging from 120 to 150 lineal feet per second. The annealed work was then examined visually under a microscope, and grain size estimated.

The results of these tests are plotted as a curve shown on FIG. 20 which indicates generally the relationship between annealed grain size and the final temperature reached by the strip prior to emerging from the heating chamber. To one skilled in the art of annealing cartridge brass by conventional slow methods of heating, often a matter of hours, this grain size vs. strip temperature relationship has been found to be of extreme interest and is believed to be new in the art in that higher strip temperatures were used for the grain size produced than heretofore believed possible. It will be observed from FIG. 20 that average grain size, in millimeters, can be expected to vary as a function of annealing temperature from about 0.01 at 1100° F., to about 0.02 at 1200, 0.034 at 1300, 0.056 at 1400 and 0.07 at 1450° F., when heating from cold to final temperature in 30 seconds or less total time.

It has been determined that emissivity of brass strip can be expected to average about 0.2, but to vary between about 0.1 and 0.3. It has been demonstrated that it is not feasible to provide uniform grain size by continuous, rapid radiative heat transfer annealing. For example, if brass strip were pre-heated to 800° F. and then heated by radiative heat transfer to an average of 1200° F. annealing temperature, even at the relatively low rate of 12,000 pounds per hour of 0.025" thick by 26" wide strip in a heating chamber 24 feet long, portions thereof having an emissivity of 0.3 would be heated approximately to 1320° F., while portions thereof having an emissivity of 0.1 would be heated only to about 1040° F. Thus, the total temperature variation to be expected in the strip work would be 280° F. It will be seen by reference to FIG. 20 that at least a four-fold variation in grain size would be expected to result from such temperature variations.

In apparatus according to the invention, convective heat transfer has been made to predominate over radiative heat transfer to such an extent that continuous rapid annealing of brass strip to uniform grain size is made possible, notwithstanding variations in strip surface emissivity.

It has also been found that high thermal head furnacing cannot be adapted to the type of apparatus contemplated by this invention and give the degree of control desired. For example, even when a high velocity compressible fluid is used in a heating chamber for heating brass strip to about 1200° F. according to the invention, it is not known to be possible to achieve sufficiently close temperature control of the work when the temperature head is more than about 300. The term "temperature head" is used herein, and in the appended claims, to refer to the temperature difference, in degrees F., between work discharged from a heating or cooling chamber and the temperature of compressible fluids supplied thereto. Preferred results in the form of more uniform discharge temperature for the work are achieved when the temperature head is not greater than 250, and optimum results are achieved in the most desired instance when the temperature head is not greater than 200. Uniformity of final temperature of brass strip work may be estimated on the basis of finished grain size. In general, the smaller the temperature head, the longer the furnacing time required to heat the work to the desired temperature. Therefore, a temperature head of at least 25 is usually preferred, and at least 50 is most desired.

It will be apparent that various changes and modification can be made from the specific details discussed and shown in the attached drawings without departing from the spirit of the invention. As a specific example, compressible fluid velocities in apparatus according to the invention have been discussed as having an upper limit just below that velocity which causes resonant flutter. Because flutter is believed to be a resonant condition, however, it is to be expected that strip stability would be achieved throughout a range of fluid velocities higher than those at which resonant flutter occurs. In order to achieve such higher velocities, it would be necessary to stabilize the strip, for example by means of removable heavy plates on either side thereof, as fluid velocities are increased through and beyond those which cause resonant flutter. The plates or other stabilizing means could then be discharged, for example from a heating chamber, and operation conducted, as previously described, at fluid velocities within a range beyond the first resonant flutter point. It has been found experimentally to be possible to achieve compressible fluid velocities in apparatus according to the invention sufficiently high that the ratio of heat transferred by convective heat transfer to heat transferred by radiative heat transfer is as high as 6:1 for brass, and 20:1 for aluminum before resonant flutter occurs. Even higher ratios could be achieved using fluid velocities beyond the first resonant flutter point, or on materials having extremely low emissivity (mirror-like surfaces).

We claim:

1. A method for heat treating metal strip work which

comprises passing the work through an enclosed heating zone and then through a separate and enclosed cooling zone, causing a heated compressible fluid to flow through the heating zone generally longitudinally of the work, at a velocity sufficiently high that the ratio of heat transferred to the work by convective heat transfer to heat transferred by radiative heat transfer is at least 2:1 and at a temperature which is higher than the temperature at which the strip enters the heating zone by from 1.0 to 1.3 times the number of degrees the strip is heated therein, and causing a cool compressible fluid to flow at high velocity through the cooling zone generally longitudinally of the work, and wherein the work is unsupported during its travel from entering the heating zone to leaving the cooling zone, but is maintained under tension by forces applied exteriorly of said zones.

2. A method for heat treating cold worked metal strip work to effect grain recrystallization and consequent annealing, which method comprises passing the work in a substantially vertical direction through an enclosed heating zone and then through a separate and enclosed cooling zone, causing a heated compressible fluid to flow within the heating zone in heat transfer relationship with the work to effect heating thereof, in a heating region, from below a minimum temperature at which the work is subject to marking due to roll pick-up and to a temperature sufficiently high to cause grain recrystallization and consequent annealing, and cooling the work, in a cooling region of the cooling zone, to below the minimum temperature at which the work is subject to marking by roll pick-up, by causing a cool compressible fluid to flow at high velocity within the cooling zone in heat transfer relationship with the work and by maintaining a body of liquid water in contact with the work adjacent the strip discharge end of the cooling region, and wherein the work is unsupported during its travel from entering the heating region to leaving the cooling region, but is maintained under tension by forces applied exteriorly of said regions.

3. A method for heat-treating cold worked metal strip work to effect grain recrystallization and consequent annealing, which method comprises passing the work in a substantially vertical direction through an enclosed heating zone, and then through a separate and enclosed cooling zone, causing a heated compressible fluid to flow within the heating zone in heat transfer relationship with the work to effect heating thereof, in a heating region, from below a minimum temperature at which the work is subject to marking due to roll pick-up and to a temperature sufficiently high to cause grain recrystallization and consequent annealing, and causing a cool compressible fluid to flow at high velocity within the cooling zone in heat transfer relationship with the work and effecting cooling thereof, in a cooling region, to below the minimum temperature at which the work is subject to marking by roll pick-up, and wherein the work is unsupported during its travel from entering the heating region to leaving the cooling region, but is maintained under tension by forces applied exteriorly of said regions.

4. A method for heat treating cold worked metal strip work to effect grain recrystallization and consequent annealing, which method comprises heating the work from a temperature of T_0 to T_f by passing it in a substantially vertical direction through an enclosed heating zone and then through a separate and enclosed cooling zone, causing a heated compressible fluid at a temperature not more than 1.3 times (T_f-T_0) higher than T_0 to flow at high velocity through the heating zone generally longitudinally of, and in heat transfer relationship with, the work to effect heating thereof, in a heating region, from below a minimum temperature at which the work is subject to marking due to roll pick-up and to a temperature sufficiently high to cause grain recrystallization and consequent annealing, and causing a cool compressible fluid to flow at high velocity through the cooling zone generally longitudinally of, and in heat transfer relationship with, the work and effecting cooling thereof, in a cooling region, to below the minimum temperature at which the work is subject to marking by roll pick-up, and wherein the work is unsupported during its travel from entering the heating region to leaving the cooling region, but is maintained under tension by forces applied exteriorly of said regions, the velocity of the heated compressible fluid being sufficiently high that the ratio of heat transferred to the work by convective heat transfer to heat transferred by radiative heat transfer is from about 2:1 to about 20:1.

5. A method for heat treating cold worked metal strip work to effect grain recrystallization and consequent annealing, which method comprises passing the work in a substantially vertical direction through an enclosed heating zone and then through a separate and enclosed cooling zone, causing a heated compressible fluid to flow through the heating zone in heat transfer relationship with the work to effect heating thereof, in a heating region, from below a minimum temperature at which the work is subject to marking due to roll pick-up and to a temperature sufficiently high to cause grain recrystallization and consequent annealing, and causing a cool compressible fluid to flow at high velocity through the cooling zone generally longitudinally of, and in heat transfer relationship with, the work and effecting cooling thereof, in a cooling region, to below the minimum temperature at which the work is subject to marking by roll pick-up, by directing a primary compressible fluid stream into a fluid inlet end of, through, and from a fluid outlet end of the cooling zone on each side of the strip work, and directing a secondary compressible fluid stream into each of the primary streams at the fluid inlet end of the zone so that a mixture of the primary and secondary streams flows through, and from the zone at high velocity, and wherein the work is unsupported during its travel from entering the heating region to leaving the cooling region, but is maintained under tension by forces applied exteriorly of said regions.

6. A method according to claim 5 wherein the metal strip is maintained at a tension between 50 percent and 100 percent of that which will cause a strip elongation of $\frac{1}{10}$ of 1 percent under the conditions prevailing.

7. A method according to claim 5 wherein the volume ratio of fluid passing in said primary streams to fluid passing in said secondary streams is from 1.5:1 to 3:1.

8. A method for heat treating cold worked metal strip work to effect grain recrystallization and consequent annealing, which comprises passing the work through an enclosed heating zone and then through a separate and enclosed cooling zone, controlling the temperature at which the work enters the heating zone to one below the minimum at which the work is subject to marking due to roll pick-up, causing a heated compressible fluid to flow within the heating zone in heat transfer relationship with the work and to heat the work to a temperature sufficiently high to cause grain recrystallization and consequent annealing, controlling the fluid velocity within the heating zone to one sufficiently high that the ratio of heat transferred to the work by convective heat transfer to heat transferred by radiative heat transfer is at least 2:1, controlling the fluid temperature within the heating zone to one higher than the temperature at which the strip enters the heating zone by from 1.0 to 1.3 times the number of degrees the strip is heated therein, and causing a cool compressible fluid to flow at high velocity within the cooling zone and cooling the work to a temperature below the minimum at which the work is subject to marking by roll pick-up, and wherein the work is unsupported during its travel from entering the heating zone to leaving the cooling zone, but is maintained under tension by forces applied exteriorly of said zones.

9. A method as claimed in claim 8 wherein the work is pre-heated to a sub-critical temperature before it enters the heating zone.

10. A method as claimed in claim 8 wherein the work is further cooled after leaving the cooling zone.

11. A method as claimed in claim 8 wherein a body of liquid water is maintained within the cooling zone in contact with the work adjacent the strip discharge end of the zone.

12. Apparatus for continuous rapid heat treating of metal work in strip form comprising a vertically extending heating chamber having continuous, closed sidewalls and open ends for the passage of work therethrough, an adjacent, aligned, vertically extending cooling chamber having continuous closed sidewalls and open ends, for the passage of work therethrough one of the open ends of said cooling chamber being positioned in spaced, aligned relationship relative to one of the open ends of said heating chamber, vertically aligned cooperating rolls exterior of said heating and cooling chambers for conveying strip work through said heating chamber and then through said cooling chamber along a generally vertical work path, the work being unsupported at all points within said chambers, inlet means for introducing a heated compressible fluid into said heating chamber for circulation in heat transfer relationship with work therein, means for withdrawing compressible fluid from said heating chamber, means for circulating withdrawn compressible fluid and for returning it to said inlet means, means for heating the circulating compressible fluid, inlet means for introducing a cool compressible fluid into said cooling chamber for circulation in heat transfer relationship with work therein, and means for withdrawing compressible fluid from said cooling chamber, and effective to prevent the flow of compressible fluid from said cooling chamber into said heating chamber.

13. Apparatus for continuous rapid heat treating of metal work in strip form comprising a vertically extending heating chamber having continuous, closed side walls and upper and lower open ends for the passage of work therethrough, an adjacent, aligned, vertically extending cooling chamber having continuous closed side walls and upper and lower open ends for the passage of work therethrough, the upper open end of said cooling chamber being positioned in spaced, aligned relationship relative to the lower open end of said heating chamber, a water chamber vertically aligned relative to the lower open end of said cooling chamber, vertically aligned cooperating rolls exterior of said heating chamber and one being contained within said water chamber for conveying strip work through said heating chamber and then through said cooling chamber along a generally vertical work-path, the work being unsupported at all points within said heating and cooling chambers, inlet means for introducing a heated compressible fluid into said heating chamber for circulation in heat transfer relationship with work therein, means for withdrawing compressible fluid from said heating chamber, means for circulating withdrawn compressible fluid and for returning it to said inlet means, means for heating the circulating compressible fluid, inlet means for introducing a cool compressible fluid into said cooling chamber for circulation in heat transfer relationship with work therein, and means for withdrawing compressible fluid from said cooling chamber and effective to prevent the flow of compressible fluid from said cooling chamber into said heating chamber.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,021,236

February 13, 1962

Donald Beggs et al.

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 1, line 18, for "Serial No. 530,858" read -- Serial No. 530,868 --; column 5, line 4, after "the", second occurrence, insert -- flange 54. Fluid enters the inlet 50 through the major --; column 8, line 36, for "variation" read -- vibration --.

Signed and sealed this 5th day of June 1962.

(SEAL)
Attest:

ERNEST W. SWIDER
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DAVID L. LADD
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