

Jan. 25, 1966

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3,230,636

HEAT TRANSFER METHOD AND MEANS

Original Filed Aug. 28, 1959

2 Sheets-Sheet 1

Fig. 1

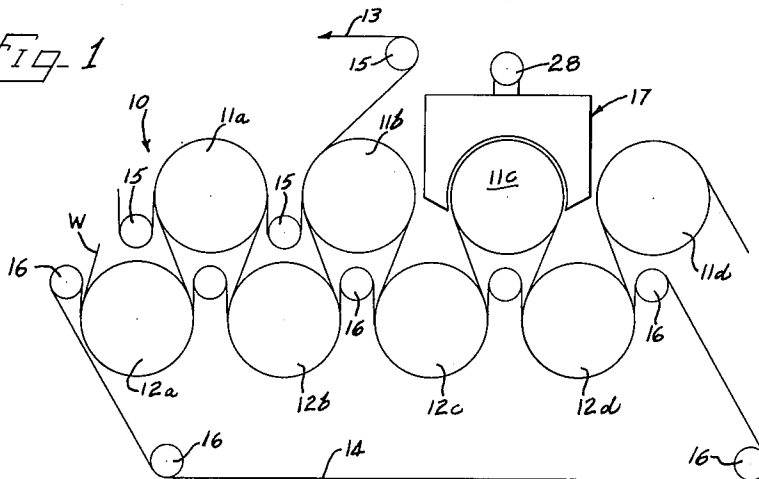


Fig. 3

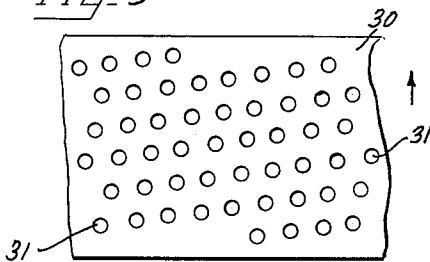


Fig. 4

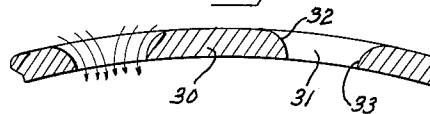
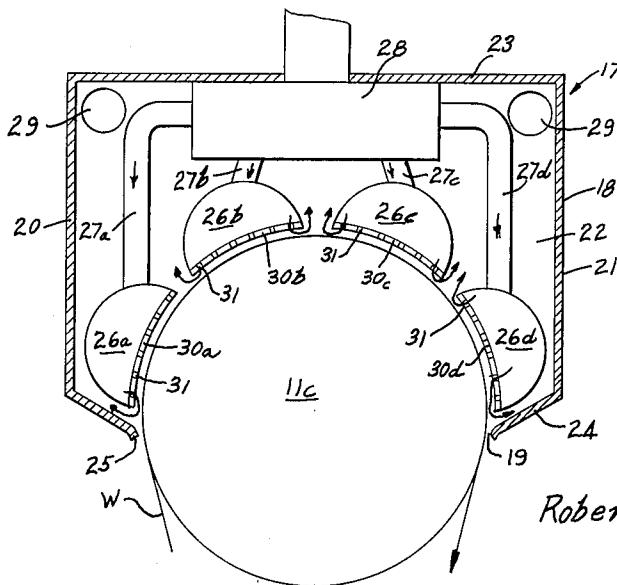


Fig. 2



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2 Sheets-Sheet 2

Fig-7

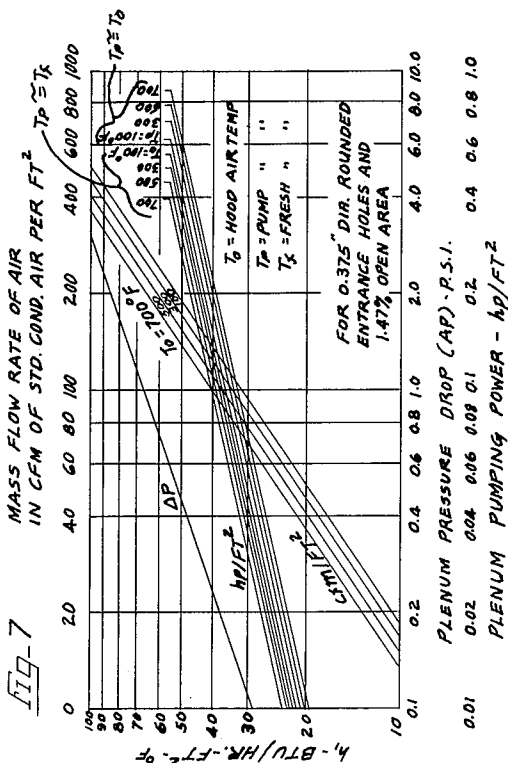


Fig-8

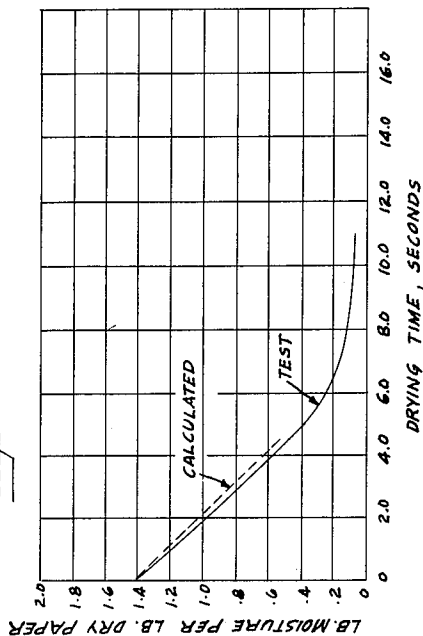


Fig-5

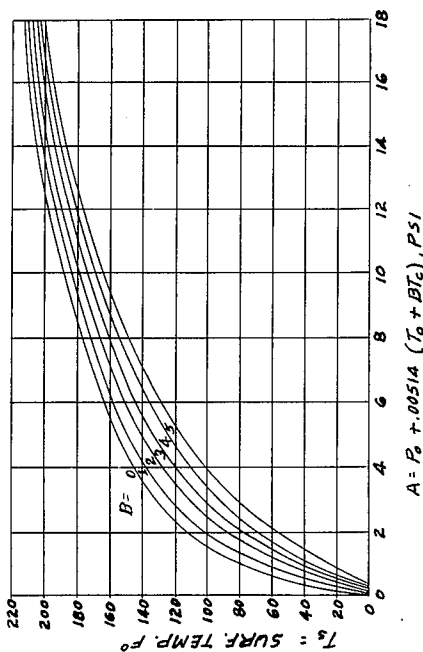
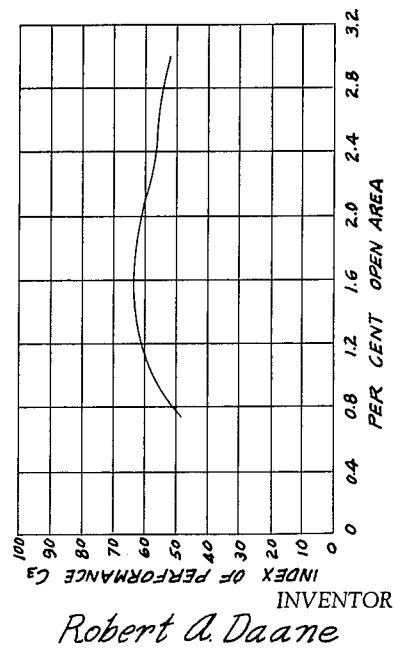


Fig-6



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3,230,636

HEAT TRANSFER METHOD AND MEANS

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Continuation of application Ser. No. 836,683, Aug. 28,
1959. This application June 6, 1963, Ser. No. 285,911
5 Claims. (Cl. 34-122)

This application is a continuation of my application Serial No. 836,683, filed August 28, 1959, and entitled, "Heat Transfer Method and Means," and now abandoned.

The present invention relates broadly to impingement flow techniques, and is more particularly concerned with a novel method and apparatus for directing fluid flow normal to a heated or cooled surface in heat transfer relation therewith and having as an important illustrative application the drying of newly coated paper sheets in a relatively short length run to prevent marring of the coated surface during subsequent passage over cylinder dryers or sheet rolls.

It is an important aim of the present invention to provide a method of impinging air flow against the surface to heat or cool the same, and which is not restricted to a particular surface but has diverse applications in the treatment of textile sheets, paper webs, plastic films, granular materials, fluid media and other surfaces.

Another object of this invention lies in the provision of a heat transfer method particularly effective in drying operations and wherein a plurality of relatively small diameter air streams are impinged against the body to be dried from a direction normal or perpendicular thereto.

Another object of this invention is to provide drying apparatus for use in the papermaking art, and which comprises means defining a plurality of circumferentially spaced plenum chambers relatively closely radially spaced from a paper web during wrapping travel about a cylindrical dryer, each plenum chamber having as one wall thereof a perforated plate facing the wrapping web and providing a plurality of relatively small diameter impinging air streams effective to reduce the web moisture content.

Still another object of the present invention lies in the provision of nozzle means for a dryer head and taking the form of a plate member having a plurality of rounded entrance holes spaced therethrough and of a particular diameter and open area as compared with the plate total area to assure maximum performance efficiency in the drying operation.

A still further object of the instant invention is to provide a drying method in which a plurality of spaced air streams of relatively small diameters are directed normal to a surface containing moisture to be removed therefrom and in which the moisture bearing air streams are directed in relatively smooth exit flow paths without substantial interference with the impinging air streams.

Other objects and advantages of the invention will become more apparent during the course of the following description, particularly when taken in connection with the accompanying drawings.

In the drawings, wherein like numerals designate like parts throughout the same:

FIGURE 1 is a somewhat diagrammatic view of a dryer section of a paper machine illustrative of a particular environment for the present invention;

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FIGURE 2 is a vertical sectional view of drying means constructed in accordance with the principles of this invention and positioned adjacent a conventional drying cylinder;

FIGURE 3 is a fragmentary plan view of nozzle means embodying the concepts of this invention;

FIGURE 4 is a sectional view of the nozzle means of FIGURE 3, and showing in detail the rounded entrance openings therethrough;

FIGURE 5 is a graph of surface temperature for free surface moisture drying in which combined air and cylinder drying are employing as in FIGURE 2;

FIGURE 6 is a graph plotting index of performance against percent open area, and showing that heat transfer performance is generally at a maximum when the perforations in the nozzle means of FIGURES 3 and 4 is about 1.5 percent of the total area of the nozzle means;

FIGURE 7 is a graph plotting air mass flow rate, pressure drop and theoretical pumping power against the heat transfer coefficient, and illustrative of the performance of the drying means of this invention; and

FIGURE 8 is a graph plotting moisture content in a particular sample tested against drying time and showing the evaporation rates obtained by practice of this invention.

It has long been generally recognized that impingement flow, or flow directed normal to a heated or cooled surface is an efficient means of obtaining high rates of heat transfer between a fluid and a solid body. This is particularly true for heat transfer from or to a gas such as air. In recent years, this means of heat transfer has begun to be used for drying of paper, particularly in the drying of coated paper where the problem arises of having to dry the newly coated sheet in a relatively short length of sheet run, so that it can be passed over subsequent cylinder dryers or sheet rolls without marring the coated surface.

Various drying devices have been proposed for use with paper making machines, and one known form comprises generally clamshell-shaped headers supporting slotted nozzle means spaced from the paper sheet as it is wrapped about suitable rollers. Another form of drying device suggested comprises a plurality of generally vertically aligned and horizontally spaced slender perforated tubes arranged to sweep conditioned air against the sheet during travel through a so-called conditioning chamber housing drying cylinders.

The described slot arrangement and means of the same general character has not, however, been characterized by efficient performance as measured by the heat transfer coefficient obtainable for a given expenditure of air blower horsepower. Additionally, known nozzle slot arrangements have the disadvantages of interfering with the impingement flow and generally require a relatively complex system of air removal ducts between the slots. Further, in any arrangement in which slot-shaped nozzles formed by sheet metal vanes projecting into the space between the drier and the impingement air supply plenum to permit spent air removal with a flow between and parallel to the slot, introduces difficulties in the practical operation of a paper drier since the sheet metal nozzles are not generally dimensionally stable and readily trap sheet fragments should a paper break at the drier occur. Projection of the nozzles into the impingement

air space is of course necessary in order to make the lateral passages for exhaust flow sufficiently large so that excessive velocity does not result when the drier is of the width required on modern paper machines, while at the same time keeping the desired ratios between nozzle throat width, nozzle tip distance from the sheet, and number of nozzles per inch.

Referring now to the drawings, there is shown somewhat diagrammatically in FIGURE 1 a dry end section 10 of drying apparatus of a paper machine with which the instant invention is of important application. However, it will be readily apparent during the course of the description now to follow that the drying means and method herein disclosed has numerous applications other than for paper machine driers, and is productive of advantageous results in the heating or cooling of any surface by means of air. Exemplary of such additional applications are textile drying, drying of plastic films, use of the apparatus and method with slasher driers, potato and cereal drying, milk drying and related processes which can clearly benefit by the inventive concepts herein disclosed, since the heat transfer behavior in the named applications is essentially the same as that existing in the drying of paper sheets or webs.

The dry end section 10 shown in FIGURE 1 comprises a first row of horizontally aligned drying cylinders or drums 11a-d of which four are shown in the exemplary embodiment illustrated, and a second row of horizontally aligned drying cylinders 12a-d staggered with respect to the drying drums 11 of the first row. A web of paper W passes alternately about the drying cylinders 11 and 12, and is maintained in contact with the surfaces of the drying cylinders 11a and 11b by a felt 12, and against the surfaces of the drying drums 12a-d by a felt 14. A suitable number of rollers 15 are arranged to guide the felt 13, and rollers 16 guide the felt 14. The drying arrangement described will be recognized as of essentially conventional construction, and in performance of the drying cycle, the cylinder or drum 11d may be a sweat drier supplied with relatively cold water to cause condensation on the surfaces thereof from surrounding moist atmosphere, the condensate being picked up by the web W traveling over this drier to restore a portion of the moisture removed by any over-drying. However, it is of course understood that all applications do not require use of the sweat drier 11d.

Impingement air drying means provided by this invention, and having the advantages earlier noted, is designated generally by the numeral 17, and it may be seen from FIGURES 1 and 2 that such means are located in relatively close wrapping relation to the drying cylinder 11c. Identical means could of course additionally be employed in association with the drier drum or cylinder 12d, and in such an application the felt 14 would be removed from the latter drum.

The impingement air drying means 17 comprises a housing or insulated hood cover 18 having a length preferably coextensive with the width of the web W and provided with a bottom opening 19 therein, so that when the housing 18 is suitably supported it overhangs the cylinder 11c approximately to the distance shown. Stated otherwise, the housing 18 extends a sufficient distance about the cylinder or drum 11c to be coextensive with the wrap of the web W during travel along the circumferential portion of the rotatable drum 11c.

The housing 18 may be provided to provide a pair of generally upright opposed side walls 20 and 21, a pair of opposed end walls 22 (one of which is shown), a generally flat top wall 23, and a downwardly inclined bottom wall 24 which may be flanged at 25 to provide a confined path of web travel during on-running and off-running wrapping travel along the drum 11c.

The housing 18 supports therewithin in any suitable manner a plurality of circumferentially spaced supply

plenums 26a-d, four of which are shown in the exemplary form illustrated. Each supply plenum 26a-d may be of generally semicylindrical shape, and the plenum means 26a-d communicate, respectively, with conduit means 27a-d leading to a supply duct 28, which in turn connects with a suitable source of heated or cooled air. To exhaust moisture bearing air from the housing 18, a plurality of exhaust openings 29 are provided, these openings being somewhat diagrammatically shown and communicating with an exhaust fan or the like (not shown).

Each supply plenum 26 supports nozzle means 30a-d, which, in accordance with the principles of this invention, takes the form of a plate having a plurality or array of spaced holes or perforations 31 therein of a particular configuration, diameter and number in order to obtain maximum drying efficiency from the impingement air drying means 17. The perforated plates 30 close the mouth portion of each supply plenum 26, and assure that the air streams issuing therefrom impinge against the paper web W normal thereto. Each plate 30 is of generally arcuate shape when viewed in end, or stated otherwise, each plate or nozzle member 30 has a curvature corresponding to that of the cylinder or drum 11c.

Referring now also to FIGURES 3 and 4, it will be observed that each opening 31 in the perforated plate 30 has a rounded entrance portion 32 which connects with a generally straight walled discharge portion 33. In tests performed to date, which will be described hereinafter, it has been established that a desirable range of diameters for the openings 31 is between $\frac{1}{4}$ and $\frac{5}{8}$ inch measured at the discharge portion 33, and an optimum diameter at this location is $\frac{3}{8}$ inch. A desirable radius on the rounded entrance portion 32 has been found to be $\frac{3}{16}$ inch. Primarily for structural reasons the perforated plates 30 each have a thickness of approximately $\frac{1}{4}$ inch.

While the radius about which the rounded or flared entrance portion 32 of the nozzle is generated may be varied, it is of advantage to maintain a distinct relationship between the ratio of the diameter of the nozzle opening or orifice to the radius about which the rounded entrance portion 32 is struck. With a $\frac{3}{8}$ -inch diameter nozzle opening and a $\frac{3}{16}$ -inch radius of the rounded entrance portion 32, the ratio of the nozzle opening or orifice diameter to the radius of the rounded entrance portion or flare at the inlet of the orifice equals one-half, and it has been found that it is advantageous to maintain this ratio regardless of changes in diameter of the nozzle opening as will hereinafter more clearly appear as this specification proceeds.

Moreover, there is distinct advantage in maintaining a fairly definite ratio between the radius of the rounded entrance opening or flare of the nozzle opening and the thickness of the plate 30. As previously stated a thickness of approximately $\frac{1}{4}$ of an inch is a practical thickness for the plate and the ratio of the radius of the rounded entrance portion or flare 32 to the thickness of the plate, as herein described, is $1\frac{1}{2}$ and it has been found advantageous to maintain this ratio between 1 and $1\frac{1}{2}$.

In tests which have been conducted it has been demonstrated that a nozzle plate employing $\frac{3}{8}$ -inch diameter holes with a $\frac{3}{16}$ -inch radius rounded entrance is productive of a coefficient of discharge for air flow through the holes which is markedly higher than the coefficient of discharge obtained when square edged holes are provided. Specifically, rounded entrance holes as herein disclosed produce a coefficient of discharge, at jet velocities of from 10,000 f.p.m. to 20,000 f.p.m., which is 0.96, as contrasted with a value of 0.79 for $\frac{3}{8}$ inch straight through holes. These values were obtained employing the following formula:

$$C_d = \frac{G'}{\sqrt{2g\rho\Delta P}} = 0.96$$

Where G =Mass flow rate of air, lb./sec. ft.² based on hole area.

g =32.2 ft./sec.².

ρ =Density at exit pressure, lb./ft.³.

ΔP =Pressure drop, lb./ft.².

The relatively low value of C_d for the square edged holes means an actual loss in performance when this type hole is used. While the low value of C_d might be thought to be merely an indication of more contraction of the flow stream emerging from the square edged hole, if this were the case the same performance as with rounded entrance holes could be achieved by merely increasing the hole size of the square edged holes. However, this was actually tried with plates provided with square edged holes having the same total air flow versus pressure drop curves as corresponding rounded entrance hole plates. The square edged hole plates were tested for heat transfer performance, and except in those cases where very large percentages of open area were provided, the index of performance was smaller for the square edged holes by at least 10 percent.

Work performed to date has also demonstrated that a desirable spacing of each perforated plate 30 from the web W during its wrap about the drum 11c is one inch. This provides sufficient clearance to allow spent air removal without excessive lateral velocity or pressure drop, even when the span of the perforated plate sections 30 or the distance between the exhaust ports 29 is as large as 24 inches. In addition, the one inch spacing leads to an optimum hole size and hole spacing which are small enough so that the possibility of spotty drying is not present. Further, the spacing of one inch also appears to be desirable from the standpoint of possible sheet wrap, in the event of a paper break, on the cylinder drier 11c over which the air drier 17 is placed. While the one inch spacing is desirable for the reasons stated, the spacing can of course be altered, however, it has been found when the one inch clearance is increased to two inches, a loss of between 4 and 6 percent in heat transfer coefficient is produced, using the same air velocity, the same ratio of clearance to hole size, and the same percentage of open area.

Hole size or diameter as measured across the straight through portion 33 also has an effect upon index of performance, the manner of obtaining this index being referred to in detail hereinafter. With respect to hole size, it was found that the index of performance reaches a maximum value when the hole size is between $\frac{1}{16}$ inch and $\frac{3}{8}$ inch, and in order to permit some increase in the one inch spacing of the plate from the web without loss in performance, the larger hole size of $\frac{3}{8}$ inch is now preferred.

The effect of open area is, however, a much more influential parameter upon the index of performance. Percentage of open area refers to a ratio of hole or open area to total area of the perforated plate 30, and the effect of variations in the percentage open area is clearly demonstrated in FIGURE 6. As shown therein, a rather sharp increase in index of performance occurs at the point at which the open area is approximately 1.5 percent although this curve shows that satisfactory performance is obtained in the range of 1.4% to 1.6%. The values plotted in the graph of FIGURE 5, were obtained utilizing 0.375 inch diameter rounded entrance holes in a 0.25-inch plate spaced one inch from the surface.

The index of performance of the impingement air system was obtained in the following manner. In general, in forced convection, heat transfer is related to flow by:

$$h_1 = C_1 \left(\frac{C_p u^m}{k} \right) \frac{k}{n^m} G^n$$

where.

h_1 =Heat transfer coefficient from air to sheet surface, B.t.u./hr./ft.² F.^o (per unit area of impingement hood coverage).

C_1 =A coefficient which depends on the shape and size of the impingement system.

C_p =Specific heat of fluid B.t.u./lb. F.^o.

u =Fluid viscosity, lb./hr. ft.

k =Conductivity of the fluid, B.t.u./hr. ft.² F.^o.

G =Mass flow rate, lb./hr. ft.² (based on total surface area covered by the impingement system).

m, n =Constants.

An impingement air system involves flowing air at high velocity through nozzles or orifices with an eventual zero pressure recovery of the kinetic energy of the jet, so that the pressure drop (or impingement supply air plenum pressure) is given by:

$$\Delta P = C_2 G^2 \nu \text{ PSI}$$

where ν =specific volume of the air, ft.³/lb.

and C_2 is a dimensional constant which depends on the nozzle design and on the ratio of nozzle area to total impingement system coverage area.

For the relatively small changes in absolute pressure of the air in these systems, the pumping power (not corrected for duct losses and pump efficiency) will be approximately:

$$\text{hp.} = \text{unit power} = 7.27 \times 10^{-5} G \nu \Delta P \text{ hp./ft.}^2$$

where the 7.27×10^{-5} is a units conversion factor.

This can be combined with the two equations above to give:

$$h_1 = C_3 (\text{hp.})^{1/3}$$

The factor, C_3 , is a combination of constants C_1 , C_2 , and the fluid property terms. For a given fluid, in the present case air at a fixed temperature, C_3 becomes the index of performance of an impingement air system.

Referring now again to FIGURE 3, it may be seen that the holes or openings 31 in each plate 30 are preferably arranged in groups of three when the plate is viewed in plan, and the groups of openings desirably form an equilateral triangle which is slightly askew from the sheet travel direction indicated by the arrow applied to FIGURE 3. As earlier noted, the preferred spacing of the perforated plates 30a-d from the circumference of the drum 11c is approximately one inch, and investigations have shown that there is no particular advantage in a relatively closer spacing between the nozzle tips and the cooled or heated surface. Specifically, the average heat transfer coefficient from the air to sheet surface varied only within the narrow range of 50 to 53 B.t.u./hr. ft.² F.^o when the nozzle throat diameter was varied from 0.31 to 0.125 inch and the distance of the plate from the surface varied from 0.375 inch to 1.50 inches. More specifically, the value of the heat transfer coefficient was 50 with a nozzle throat of 0.31 and a plate distance of 0.375, a value of 51 obtained when the nozzle throat was 0.62 inch and the plate distance 0.75, a heat transfer coefficient of 53 was found when the nozzle throat was 0.94 inch and the plate distance 1.125 inches, and the coefficient was 50 when the nozzle throat was 0.125 inch and the plate distance 1.50 inches. As indicated, the heat transfer coefficient is expressed in B.t.u./hr. ft.² F.^o.

It is to be seen from the foregoing data that close clearance between the impingement nozzles and the heated or cooled surface is not an important requirement for obtaining an optimally designed impingement air heat transfer system. However, the importance of the preferred one inch spacing is that the spent air can be removed from the space between the perforated plate and the heated or cooled surface with a flow of the spent air in any direction parallel to the perforated plate, without

interfering appreciably with the impingement flow. The path of exhaust air flow is shown in FIGURE 2 by arrows, and it is to be noted that said flow is essentially parallel as described. Accordingly, the smooth perforated plate 30 of this invention provides no interference with impingement flow, which is the case when slots are employed since they necessarily must be placed at 90° to the direction of sheet travel, for reasons of drying uniformity across the width of the sheet. Removal of the spent air when a slotted arrangement is used, accordingly requires a complex system of air removal ducts between the slots, or a configuration of sheet metal nozzles projecting into the space between the drier and the impingement air supply plenum, in order that spent air can be removed with a flow in between and parallel to the slots. As earlier noted, this introduces many practical difficulties and creates conditions interfering with efficient paper machine operation.

The spacing of the nozzle openings 31 along the plate 30 in a triangular array provides a nozzle pattern in which lines drawn along the bottom of the plate 30 transversely of the direction of web travel along the drier drum 11c to intersect one nozzle opening 31 will at all times intersect more than one nozzle opening, providing a uniform distribution of air through the plate 30 and thereby resulting in a uniform drying rate of the web across its entire width. This is particularly important when freshly coated webs are being dried and the coating material is still more or less fluid, making uniform drying treatment essential.

In FIGURE 7 of the application drawings there are shown curves which clearly illustrate the improved performance of the impingement air drying means of this invention. The curves presented therein plot heat transfer coefficient against air mass flow rate per square foot of drier surface covered by the impingement plates (in units of c.f.m., of 70° F., atmospheric air), pressure drop in p.s.i. (for flow through the perforated plate and to the hood exhaust ports 29) and theoretical pumping power in hp./ft.² as given hereinabove. The curves of FIGURE 7 are readily employed in connection with the graph of FIGURE 5, showing surface temperature for free surface moisture drying using a combined air and cylinder drying, and utilization of the graphs of FIGURE 5 and FIGURE 7 permits prediction of drying rates and supplies information allowing a particular air system to be sized for specified requirements.

Drying tests have been conducted to demonstrate the efficient evaporation rates obtained by the apparatus of this invention. For this purpose, 12-inch by 4-inch newsprint samples were dried by means of hot air impinging thereon through perforated plates 30 essentially as shown in the instant drawings. The samples had not been dried and rewetted, but were in the form as received from customary press sections. Dry air at measured temperature and velocity was impinged on the sheet samples for short periods of time, which was controlled by an electrically timed solenoid valve. The results of one test are shown in FIGURE 8, which plots drying time in seconds against the weight of moisture per pound of dry paper. The weight of newsprint was 0.0107 pound per square foot, the air flow was 112 c.f.m. "standard condition" air per square foot, and the air temperature was 300° F. The results obtained were at least equal to those obtained by the other drying means. However, it may be noted from the graph of FIGURE 8 that the fall-off in drying rate is less with impingement air drying as herein disclosed, when compared with hot surface drying.

A preferred impingement plate configuration has been illustrated and described, and as well as the hole configuration, percent open area, hole diameter, and spacing of the plate from the heated or cooled surface. However, while in work performed to date the described relationships have proven to be productive of substantially improved results, it is of course apparent that various

changes and modifications may be effected without departing from the novel concept of the present invention.

In the claims:

1. An apparatus for drying a fibrous web traveling about a rotatable drying cylinder, comprising a hood supported adjacent the drying cylinder and extending about a portion of the circumference thereof, at least one plenum chamber contained within said hood, an air duct leading into said hood and plenum chamber, an air outlet leading from said hood, for withdrawing spent moisture bearing air from said hood and from the fibrous web passing about the drying cylinder, a plate conforming to the periphery of said drying cylinder and spaced from said drying cylinder a distance in the range of between one and two inches and forming a bottom wall of said plenum chamber and having a plurality of nozzle openings leading therethrough, each of said nozzle openings consisting in a perforation through said plate and having an inwardly flared entrance throat in communication with said plenum chamber and a generally cylindrical wall leading therefrom for impinging air onto the web passing about said drying cylinder, said nozzle openings providing approximately a 1.5% open area in said plate, the diameters of said cylindrical wall portions of said nozzle openings being substantially $\frac{3}{8}$ inch and the radii about which the flares of said flared throats are generated being $\frac{3}{16}$ inch.

2. An apparatus for drying a fibrous web traveling about a rotatable drying cylinder, comprising a hood supported adjacent the drying cylinder and extending about a portion of the circumference thereof, at least one plenum chamber contained within said hood, an air duct leading into said hood and plenum chamber, an air outlet leading from said hood, for withdrawing spent moisture bearing air from said hood and from the fibrous web passing about the drying cylinder, a plate conforming to the periphery of said drying cylinder and spaced from said cylinder a distance of at least one inch and not over two inches and forming a bottom wall of said plenum chamber and having a plurality of nozzle openings leading therethrough, each of said nozzle openings being no longer than the thickness of the plate and consisting in an unimpeded perforation through said plate and having a diameter in the range between $\frac{1}{4}$ inch and $\frac{5}{8}$ of an inch and having an inwardly flared entrance throat in communication with said plenum chamber, the radius about which the flare of said flared entrance throat is generated being substantially $\frac{3}{16}$ of an inch, said nozzle openings having a generally cylindrical wall leading from said flared entrance throat for impinging air onto the web passing about said drying cylinder, and said nozzle openings providing substantially a 1.5% open area in said plate.

3. An apparatus for drying a fibrous web traveling about a rotatable drying cylinder, comprising a hood supported adjacent the drying cylinder and extending about a portion of the circumference thereof, at least one plenum chamber contained within said hood, an air duct leading into said hood and plenum chamber, an air outlet leading from said hood, for withdrawing spent moisture bearing air from said hood and from the fibrous web passing about the drying cylinder, a plate conforming to the periphery of said drying cylinder and spaced from said cylinder a distance of at least one inch and not over two inches and forming a bottom wall of said plenum chamber and having a plurality of nozzle openings leading therethrough, each of said nozzle openings being no longer than the thickness of the plate and consisting in an unimpeded perforation through said plate and having a diameter in the range between $\frac{1}{4}$ inch and $\frac{5}{8}$ of an inch and having an inwardly flared entrance throat in communication with said plenum chamber, the radius about which the flare of said flared entrance throat is generated being substantially $\frac{3}{16}$ of an inch, said nozzle openings having a generally cylindrical wall leading from said flared entrance throat for impinging air onto the web passing about said drying cylinder, said nozzle openings being

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arranged in a generally equilateral triangular array, all of the sides of the triangles of which triangular array are angular both with respect to the direction of travel of the web and with respect to a line extending perpendicular thereto, and providing a nozzle pattern uniformly distributed about said plate and having an open area of substantially 1.5% of the total area of said plate.

4. A drying apparatus in accordance with claim 2 wherein the ratio of the diameter of each discharge orifice to the radius of the rounded entrance throat of each orifice is at least $\frac{1}{2}$.

5. A drying apparatus in accordance with claim 2 wherein the ratio of the radius of the rounded entrance throat of each nozzle opening, to the thickness of said plate is one or greater and is not more than $1\frac{1}{2}$.

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