HYBRID COOLING SYSTEM FOR INDUSTRIAL APPLICATION

by

C. I. EZEKWE
Department of Mechanical Engineering
University of Nigeria, Nsukka.
(Manuscript received September 4, 1979 and in revised form March 20, 1980)

ABSTRACT

A hybrid cooling System was constructed and tested for glass-ware and plastic-ware production. The unit utilizes water-in-air stream to cool molds in glass and plastic forming processes. The rate of heat transfer between the mold surface and the two component two-phase stream was increased more than five times over that achieved by using the gas (air) phase alone. The enhanced cooling rate yielded speed increase in production by about 65%. The system also reduced the level of noise due to air blast at the press by limiting the use of compressed air. This cooling unit could also be used in iron and steel industries.

NOTATION

m liquid-to-air mass flow ratio
Nu local two component Nusselt number
Pe_1 Peclet number for liquid
r radial coordinate measured from centre of cylinder
R cylinder radius
Re free stream Renolds number
T temperature
T_o cylinder temperature
T_δ temperature at the film-air interface
T_∞ free stream temperature of droplets
TAU Blasius shear polynomial
U_δ Φ-velocity at film-air interface
U_∞ free stream velocity upstream from cylinder
δ film thickness
$\bar{\delta}$ normalized film thickness, δ/R
θ_o $T_0 - T_\infty$
Θ_δ $T_\delta - T_\infty$
ρ_a density of air
$\bar{\rho}$ ratio of density of liquid to that of air
τ external shear stress acting on film due to air flow
Ø angular coordinate measured from leading stagnation point

1. INTRODUCTION

The rate of heat transfer between solid surfaces and a gas stream can be increased substantially by introducing minute water droplets into the gas stream. This increase can be explained qualitatively visualizing the boundary layer over the solid surface. A liquid boundary layer which has a higher heat transfer coefficient will replace the previously existing gas boundary layer. Hence the liquid droplets entrained in the gas stream enhance heat transfer from the body by sensible heating and evaporation of the liquid in the film. This technique is closely related to modern cooling applications such as film, ablation and transpiration cooling. Excess water in the air stream, however, gives rise to pool boiling which adversely affects the rate of heat transfer. A survey of the literature reveals that different heat transfer investigations have been carried out for two-phase. two-component flow over some geometrical surfaces. Actives et al.[1],Smith [2], Hodgson [3], Goldstein et al.[4], Hodgson et al.[5], and Hodgson and Sunderland [6] worked on cross flow over a circular cylinder. All the experimental and analytical studies confirmed the high potential for increasing heat transfer by this method. A thin continuous liquid film was observed on all surfaces directly exposed to water-in-air flow. Thomas and Sunderland [7] determined the heat transfer and liquid film thickness for a wedge-shaped body in air-water stream and arrived at the result that heat transfer rate increased by about twenty times over that for air stream alone.

The current investigation is concerned with the performance characteristics of a unit designed for large- scale application of the hybrid cooling technique. Prototypes were studied in the laboratory before a final assembly was constructed. The assembled unit was operated in an industrial production of plastic and glass-ware to test the viability of the new technology in various industrial manufacture.

2. ANALYSIS OF TWO-COMPONENT HEAT TRANSFER

The heat transfer analysis for the performance of the system follows the method of Hodgson and Sunderland [6] developed for an isothermal cylinder exposed to a crossflow consisting of a water-in-air spray. The analysis considers integral forms of the continuity, momentum and energy equations as applied to the liquid film which forms on the cylinder. All flow is considered incompressible and evaporation from the film is neglected. With all properties (except pressure and temperature) held constant, the continuity and momentum equations are not coupled to the energy equation.

The velocity distribution in the liquid film in the Ø direction is obtained by assuming a velocity profile approximated by a third-degree polynomial. Simultaneous solution of the continuity and momentum equations for the film introducing the velocity profile equation yields a closed form solution for the film thickness which is given by

$$\overline{\delta}^2 = \frac{6m Sin\, \emptyset / Re}{(3\tau/2\, \rho_a U_\infty^2 + (5m\, Sin\, \emptyset\, Cos\, \emptyset/4)} \quad (1)$$

In performing calculations, values for τ are obtained by use of the Blasius series as given by Schlichting [8].

The derivation of the energy equation for the liquid film invokes the outlined basic assumptions, and the temperature profile is also approximated by a third- degree polynomial. Substitution of the velocity and temperature profiles into the integral energy equation results in a first-order, linear, ordinary differential equation. This is expressed as

$$(1-\theta_\delta/\theta_0) = \frac{(9/16)\frac{d}{d\emptyset}(U_\delta\overline{\delta}/U\infty)}{(31/70)\frac{d}{d\phi}(U_\delta\overline{\delta}/U\infty) + (3/Pe_1\overline{\delta})} \quad (2)$$

(2)

The local Nusselt number for two – component heat transfer is given by

$$Nu = \frac{-2R}{T_0 - T_\infty}\left(\frac{\delta T}{\delta r}\right)_{r=R} = (3/\overline{\delta})(1-\theta_\delta/\theta_0) \quad (3)$$

Consideration of the hydrodynamic equations which neglect the film inertia, body force, and pressure gradient yields:

$$\frac{d}{d\emptyset}(U_\delta\overline{\delta}/U_\infty) = (12m/\overline{\rho})$$
$$\{\cos\emptyset\left[\frac{TAU + \left(\frac{2}{3}\right)m\sqrt{2Re}\ Sin\,\emptyset\ Cos\,\emptyset}{6\,TAU + 5\,m\sqrt{2Re}\,Sin\,\emptyset\ Cos\,\emptyset}\right]$$
$$+ m\sqrt{2Re}\ Sin\,\emptyset\left[\frac{TAU\ Sin\,\emptyset\ Cos\emptyset - TAU(1-2\,Sin^2\emptyset)}{(6\,TAU + m\sqrt{"Re}\ Sin\,\emptyset\ Cos\,\emptyset)\hat{}2}\right]\} \quad (4)$$

Where TAU = 6.973 $\emptyset - 2.732\ \emptyset^2$ + . . . and

TAU = $\frac{d}{d\emptyset}$(TAU) = 6.973 – 8.196$\emptyset^2$ + . . .(from Schlinchting [8]

With known values of m and Re, Eqns. (1 – 4) provide the means for calculating the local Nusselt number around the cylinder The calculated values have been shown [6] to agree well with the experimental results of Smith [2], Hodgson [3], and Hodgson et al. [5]. At low (less than 0.02) water-to-air mass flow ratios, however, there is a consistent underestimation of the heat transfer by the no evaporating film model used in the present analysis. For high (greater than 0.02) water-to-air mass flow ratios evaporative effects are not significant. A possible explanation for this is that the evaporating surface is subjected to an influx of "cold" liquid in the form of droplets. The rate of the film evaporation will be reduced by these droplets and this effect will increase as the liquid-to-air mass flow ratio increases. For the cooling system under discussion, the operating water-to- air mass flow ratio varies from 0.05 to 0.15.

3. SYSTEM DESIGN AND CONSTRUCTION

The main component of the unit comprises a set of six spray nozzles. Round jet type of commercial atomizing nozzles ($\frac{1}{4}$JN No.46) manufactured by Spray Systems Company, Illinois was selected. The nozzle operating characteristics were verified in the laboratory, the most important of which are shown in Fig.1

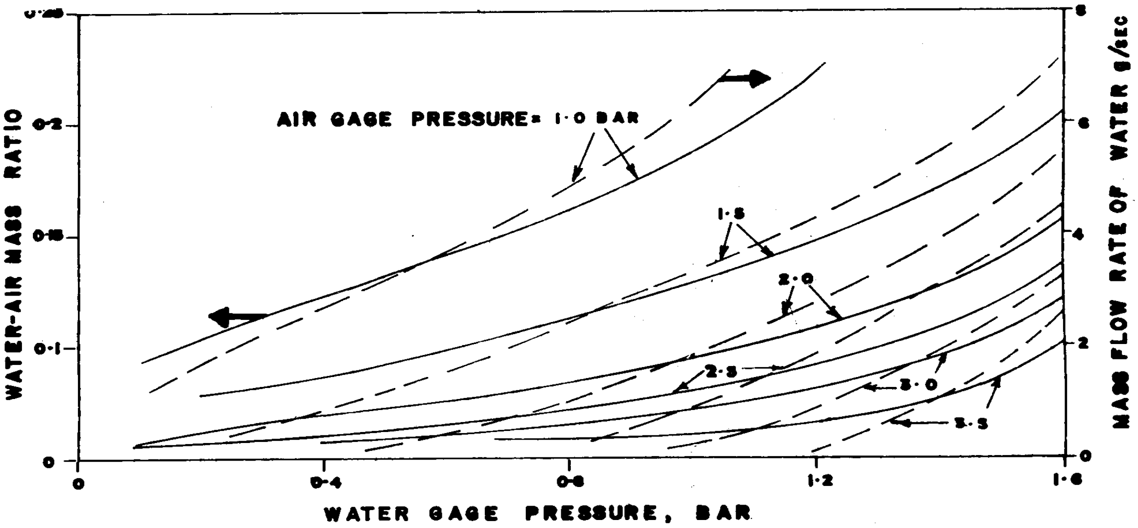

Fig. 1 spray Nozzle flow characteristics

Figures 2 and 3 show an assembly of the whole unit constructed on a frame- work of iron pipes and copper tubings. The system utilizes solenoid valves, needle valves, and pressure gauges to control the spray characteristics. The overalls size of the assembly corresponds with the physical dimensions of the forming machine. The unit is designed for rigid installation beneath the press table of the glass and plastic forming machine which will permit direct impingement of the two-phase two-component stream on the mold surface.

4. PERFORMANCE TESTS

Figure 4 shows the photograph of an automatically operated commercial forming machine with the test unit mounted underneath the press table and fig.5 shows the physical set-up of the press table. Molds used were cylindrical in shape and were of the stainless steel type with a feeding temperature of about 1450^{0}C. The industrial tests were conducted using micron sized liquid water and air in the flow stream. Spray characteristics depended upon air velocity, water-air mass ratio, operating pressure, and the distance of the nozzles from the mold surface. Droplets on the order of 0.13-mm diameter and liquid-to-air mass ratio of 0 to 0.20 were used. Stream velocities varied from 20 to 30 m/s and the stream initial temperature was held fairly constant at 35 ^{o}C. Data are reported for variations in mold surface temperatures measured using Land pyrometer.

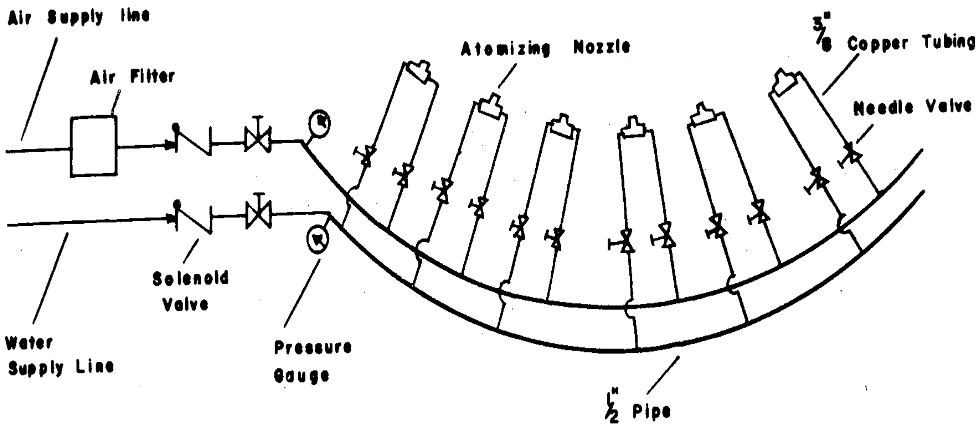

Fig.2 schematic diagram of the cooling unit

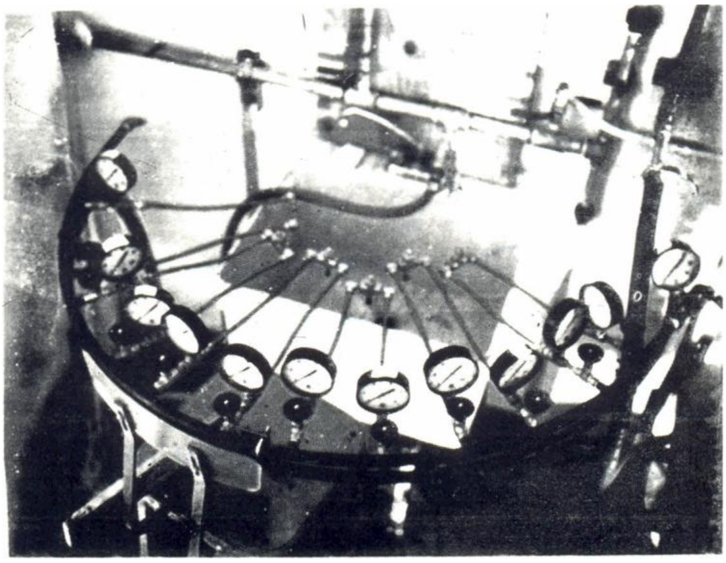

Fig. 3 photograph of the laboratory set - up of the cooling system

5. RESULTS AND DISCUSSIONS

For glass forming, mold temperatures dropped by an average of 200^0C from loading position to Take-out position when mist cooling was applied at four stations as compared to an average drop of 127^0C obtained when only air cooling was used in ten stations. The mold surface temperature values compare favorably with analytical results. Similar result was obtained for plastic- ware production. The rate of heat transfer was strongly influenced by water- air mass ratio in the stream which impinged on the mold surface. The mass ratio of about 10% gave the best performance. Table 1 shows the rates of cooling obtained with different water-air mass ratios. The cooling rate was also influenced by the air velocity and droplet size. Higher air velocities favour increased rates of heat transfer as illustrated in fig.7. Smaller water droplet sizes add to increased heat transfer rate probably because they enhance evaporative cooling. Quantitative data of the water-droplet particle size could not be obtained due to the cost of

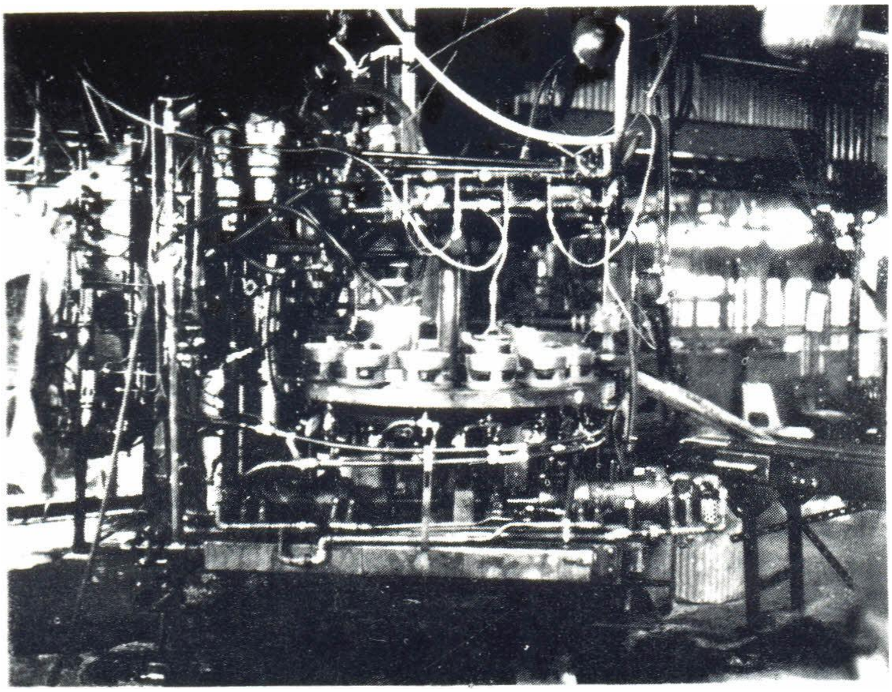

Fig. 4: Photograph of glass and plastic forming machine with hybrid cooling unit

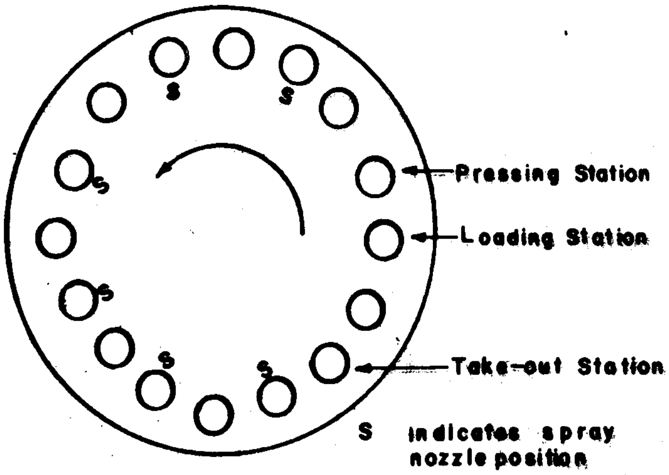

Fig.3: Photograph of the laboratory set - up of the cooling system

Table I. Variation of measured cooling rate with water-air ~ass ratio (Re = 5 x 10)

Water-air ma,as ratio	Heat Flux KJ/Sec.:-m^2
0.00	15.46
0.025	53.02
0.05	64.18
0.10	76.80
0.15	74.36
0.20	70.21

the instrumentation. However, the approximate data supplied by the nozzle manufacturers lie between 100 and 200 microns. It was therefore important to control each of the contributing variables simultaneously in order to obtain high heat transfer rates.

Using air cooling, the highest speed of glass forming operation was 15 drops per minute. With hybrid cooling, production rate of 25 drops per minute was achieved. This maximum speed was limited by the glass pull on the glass tank. At that

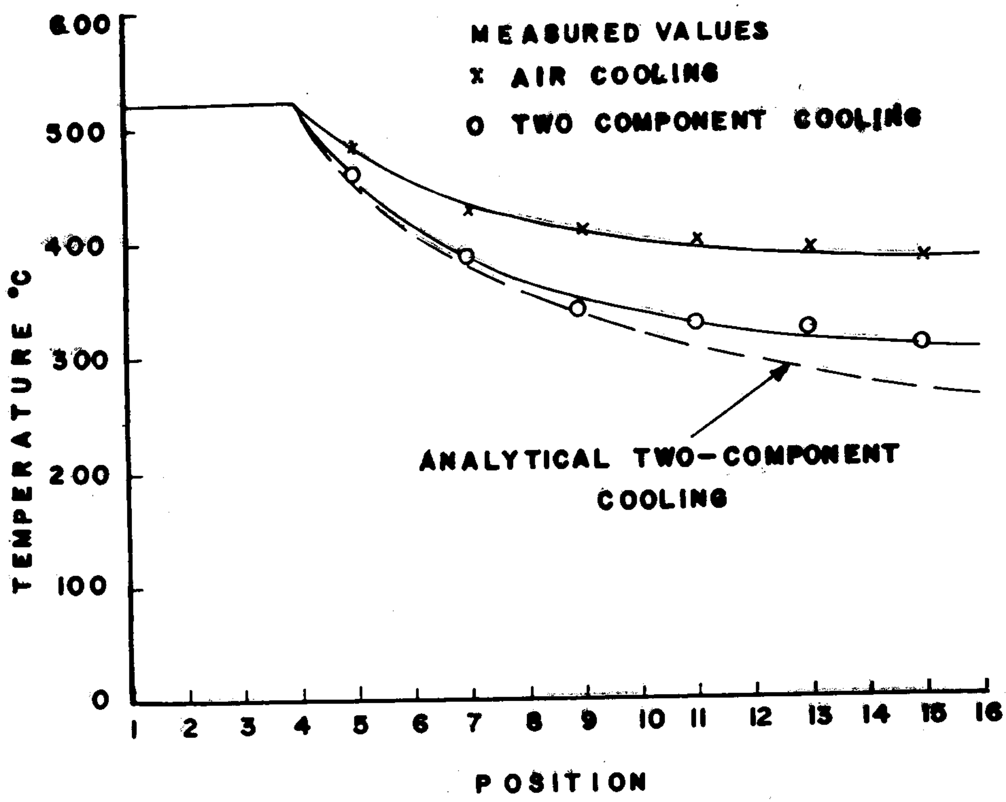

Fig. 6 mold surface temperature history

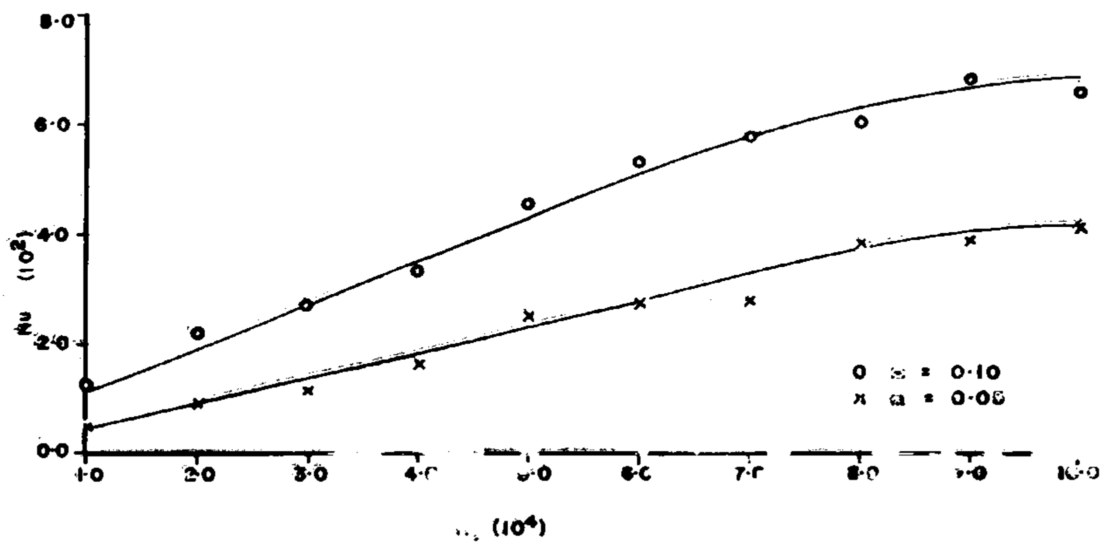

Fig. 7. Influence of stream velocity on rate of cooling

speed, only four out of the six spray cooling stations were in operation which indicated that additional speed increase was possible with more glass pull.

6. Conclusions

The applicability of the hybrid cooling system in an industrial-scale manufacture has been established. The system is easily operated and has significant advantages over the conventional air cooling units. The enhanced heat transfer rate resulting from the use of the system yielded speed increase in glass-ware and plastic-ware production by about 65%. The unit can be adapted for use in various industrial establishments such as iron ran and steel factories. The new system has also the advantage of reducing the level of noise due to air blast at the press by limiting the use of impressed air. This is one of the major safeguards for personnel in industries.

References

I. Acrivos,A., Ahern,J.E., and Navy.A.R. Jr., Research Investigation of Two-Component Heat Transfer. Marguardt Corp., Aerospace Research Laboratories, Wright-Patterson AFB, Ohio Report ARL64-116, 1964.
2. Smith, J. E., Heat Transfer Studies of Water-Spray Flows, Northern Research and Engineering Corp, Aerospace Research Laboratories, Wright-Paterson AFB, Ohio, Report ARL66-0091, 1966.
3. Hodgson, J. W., Investigation of the Heat Transfer from an Isothermal Cylinder Exposed to Cross flow of a Water-Air Spray, Ph.D. Thesis, Georgia Institute of Technology, Atlanta, Ga., 1967.

4. Goldstein, M. E., Yang, W.J., and Clark, J.A., Boundary Layer Analysis of Two-Phase (Liquid-Gas) Plate, Trans. ASME, J. Heat Transfer,Vol.89C 1965, pp.185-193.

5. Hodgson, J.W., Saterbak,R.T., and Sunderland, J,E., Experimental Investigation of Heat Transfer from Spray-Cooled Isothermal Cylinder, Trans. ASME, J. Heat Transfer,Vol.90C,1968, pp.457-463.

6. Hodgson, J.W., and Sunderl and,J.E Heat Transfer from a Spray-Cooled Isothermal Cylinder, Ind. Eng. Chem. Fundam., Vol.7,No.4,1968,pp.567-572.

7. Thomas,W.C.,and Sunderland, J.E., Heat Transfer between a Plane Surface and Air Containing Suspended Water Droplets,Ind.Egg.Chem.Fundam.,Vol.9 No.3,1970,pp 368-374.

8. Schlichting, H., Boundary Layer Theory, McGraw-Hill Book Company, Inc., New York, fourth Edition, 1960.