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THE EFFECT OF VELOCITY OF STEAM-AIR MIXTURES ON THE HEATING OF GLASS CONTAINERS

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STEAM-AIR MIXTURES at near zero velocity were found by Pflug and Nicholas (7) to be less efficient when used to heat containers of water than either a water bath, water spray or saturated steam. The wide usage of steam-air mixtures in pickle pasteurizers stimulated further work to ascertain the cause of this decreased efficiency. This investigation was undertaken to determine the effect of the velocity of steam-air mixtures on the heating characteristics of water in glass containers.

EXPERIMENTAL METHOD

All tests were conducted using 16-oz. (303) glass jars with lug closure: outside diameter 3.12 in.; height to finish 4.25 in.; overflow capacity 16.5 oz.; fill 16 oz. of distilled water.

The apparatus used in this experiment is shown in Fig. 1. It consisted of a laboratory retort modified by the insertion of an inverted tank with a 10 in. length of 6⅛ in. I.D. pipe, welded in place. The test jar was centered in this pipe by sharpened screws and rested on a strand of 0.020 in. O.D. steel wire stretched across the pipe.

The heating control system is also shown in Fig. 1. The air flow rate was measured by a rotameter and maintained constant by manual control. Steam flow to the system was maintained by a proportioning temperature control system. The sensing thermocouple was located below the jar in the entrance to the pipe and adjacent to the retort temperature measuring thermocouple. The air and steam entered the retort through ports in a four-armed cross at the bottom of the retort. These ports were at right angles to the retort diameter and at a 45° angle from the bottom and imparted a swirling motion to the mixed gases.

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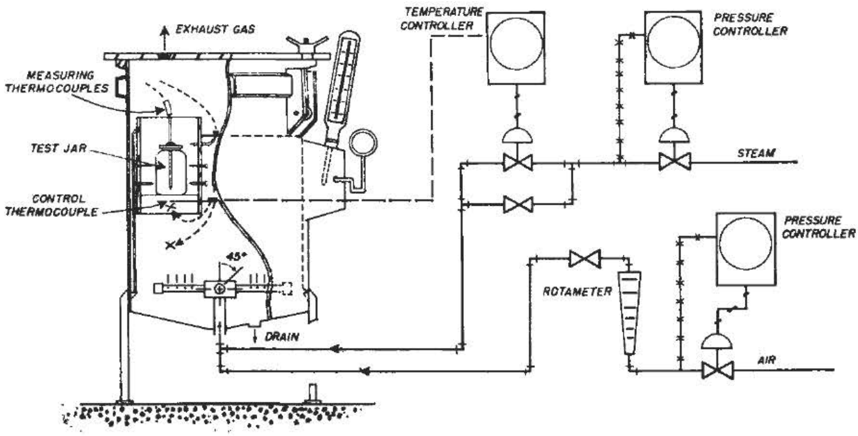

Fig. 1. Schematic diagram of modified laboratory retort and the heating control system.

The lid to the retort was loosely closed during the tests in order to eliminate the various chimney effects that were observed in the systems used in preliminary studies. The retort system was allowed to equilibrate at the test temperatures before each test was started. This procedure and the use of the inner chamber eliminated come-up-time corrections. In some tests the heat load of the retort system was further reduced by insulating the retort with a 2-in. thickness of fiber glass insulation.

The temperatures in the jar were measured with 24-gage copper-constantan thermocouples similar to those used by Pflug and Nicholas (7). The thermocouple junctions were located in the vicinity of the slowest heating point; namely 0.32 in. (0.8 cm.) above the inside bottom of the jar. The steam-air mixture temperatures were measured both above and below the jar and at several other points in the apparatus. Temperatures were measured and recorded by a 12-point (1 min. cycle) temperature recording potentiometer; the smallest chart division was 1°F.

The gas at the control point thermocouple was assumed saturated with water vapor (steam) and to behave as an ideal solution; therefore,

$$\frac{V_s}{V_t} = \frac{V_t - V_a}{V_t} = \frac{P_s}{P_t} = \lambda$$

where

P_s = partial pressure of steam (2)
P_t = partial pressure of system, one atmosphere here
V_a = volume of air
V_s = volume of steam
V_t = total volume

hence

$V_t = V_a/(1\text{-}\lambda)$.

The volume rate of flow was measured by a rotameter and corrected to retort temperature using the ideal gas law. Appropriate corrections were made for temperature and pressure of the gas at the rotameter (3). This flow was substituted in the above relation to obtain the total gas volume flow rate. This latter flow rate was divided by the calculated annular area between jar and pipe to give the mean annular velocity. Mean annular velocities from near zero to over 8 feet per second were studied.

The initial temperatures of the water in the jars was 95.6 ± 1°F. Three steam-air mixture temperatures were studied; 165, 180 and 195°F. The mean variation in retort temperature within runs was ± 0.5°F.; among runs was ± 1.0°F. for the uninsulated series. The variation for the insulated series was approximately twice as much due to the decreased heating load on the control system.

The heating data were analyzed by plotting the temperature difference between product and heating medium vs. time on semilogarithmic paper according to the conventional method described by Ball and Olson (1). These curves were found to approach a straight line. The f_h and j were then determined for the linear portion of the graph.

RESULT AND DISCUSSION

The results, in terms of the effect of the velocity of steam-air mixture on the heating parameters f_h and j of pint jars of water, are shown graphically in Figs. 2 and 3. The results may be summarized as follows: (1) f_h decreased (faster heating) with increasing velocity; (2) f_h at near zero velocities increased when the system was insulated; (3) f_h decreased with increasing heating medium temperature; (4) the average lag factors j for all tests was 1.34.

The efficiency of steam-air mixtures at low velocities is very poor — note large f_h values in Fig. 3; however, their efficiency at high ve-

Fig. 2. Scatter diagram of heating rate parameters f_h and j.

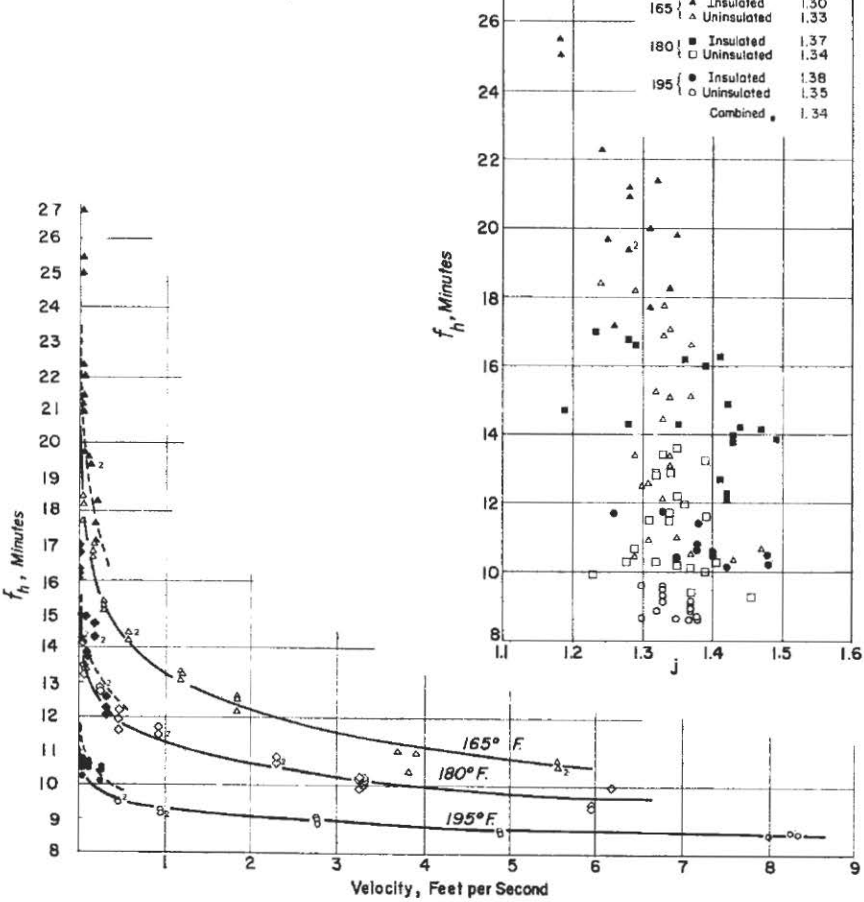

Fig. 3. Heating rate parameter (f_h) of pint jars of water as a function of velocity of steam-air mixtures at atmospheric pressure.

locities is very good, as shown in Table 1, where the f_h of steam-air at the maximum velocity tested are compared to the f_h values Pflug and Nicholas (7) obtained using a water spray. These comparisons point out that at the same temperature heating with steam-air mixtures is nearly as effective as water spray only provided the velocity of the steam-air mixture is sufficiently high.

The heating characteristics of pint jars of water in steam-air mixtures at near zero velocity (Fig. 3) agree with the data reported by Pflug and Nicholas (7) within limits of experimental error.

The differences in the heating rates at 165, 180 and 195° expressed as the ratio of the change in f_h per 15°F. heating medium temperature rise are tabulated (Table 1) as a function of the heating medium velocity. Three factors that may contribute to this increase in the rate of heating (decrease in f_h) with increasing temperature are: (1) an increase in the thermal diffusivity of the water and a decrease in viscosity at the higher heating medium temperature, affecting both water inside the jar and water film on the outside of the jar, (2) an increase in the convective flow inside the jar produced by the initially larger temperature difference between the heating medium and the product in the jar and (3) an increase in the convection heat transfer coefficient with increasing steam-air mixture temperature. The first two factors are not functions of the heating medium per se; therefore, the observed decrease in f_h with increased heating medium temperature can be only partially an effect of the steam-air heating medium.

TABLE 1—Comparison of f_h at maximum test velocities of steam-air mixtures with water spray

Temperature of heating medium	Water spray(7)	Steam-air mixtures	
°F.	f_h, min.	f_h, min.	Velocity ft. per sec.
165	10.0	10.6	5.5
180	9.7	9.7	6.0
195	9.5	8.6	8.0

It seems probable that the major effect is due to an increase in the convection heat transfer coefficient of steam-air mixtures with increasing temperature since the reduction in f_h with a 15°F. increase in heating medium temperature observed by Pflug and Nicholas (7) for water bath, water spray, and saturated steam heating averaged only 5 percent compared to a reduction in f_h from 8 to 22 percent, depending on velocity (Table 2), with the temperature of steam-air mixtures. It seems probable from these comparisons that from 50 to 75 percent of the decrease in f_h with increasing heating medium temperature is due to the steam-air heating medium.

TABLE 2—Ratio of the change in f_h per 15°F. heating medium temperature rise as a function of heating medium velocity

Velocity ft. per sec.	Ratio	
	$1-\frac{f_{h\,180}}{f_{h\,165}}$	$1-\frac{f_{h\,195}}{f_{h\,180}}$
0.2	0.21	0.22
0.5	0.18	0.20
1.0	0.15	0.17
2	0.12	0.15
4	0.10	0.12
6	0.08	0.10

The transfer of heat from a steam-air mixture is a typical example of heat transfer from a mixture of a vapor and non-condensable gas (see Kern, 4). When a mixture of a vapor and a non-condensable gas comes in contact with a surface that is below the dew point, a film of condensate collects on this surface and a film of non-condensable gas and vapor develops adjacent to this condensate film. The rate of sensible heat flow from the fluid to the surface will be a function of the resistances of these films and the temperature difference driving force. At the same time the rate of heat transfer by vapor movement to the surface followed by condensation will depend on the resistance of these films to mass transfer or diffusion.

Kusak (5) has found that for steady-state condensation of vapors from a non-condensing gas that apparent film coefficient for a system may be satisfactorily correlated using the equation:

$$h_a = a \left[\frac{GD}{\mu LD}\right]^{1/3} e - cM_a$$

where

- a, c = experimental constants
- D = diameter of condenser, ft.
- e = Napierian base 2.718 . . .
- G = mass velocity of fluid, lb/hr.(ft.^{2} of cross section)
- h_a = fictitious gas conductance including all conductances between bulk of gas and condenser surface at L feet from entrance Btu/(hr.)(ft.^{2})(°F.)
- L = length of condenser, ft.

M_a = mole fraction air in the gas-vapor steam
μ = absolute viscosity, lb./(hr.)(ft.)

Since Kusak has shown that the gas film is the principal resistance, it is, therefore, not surprising that h is a function of flow rate G in the same manner as for non-condensing fluids. Steady-state forced convection heating of such fluids outside tubes may be satisfactorily correlated using the equation (6):

$$\frac{h_m D_o}{K} = b_1 \left[\frac{D_o G}{\mu}\right]^n \left[\frac{C_p \mu}{K}\right]^m$$

where

b_1 = experimental dimensionless constant
C_p = specific heat of film Btu/(lb. fluid)(°F.)
D_o = outside diameter, ft.
G = mass velocity of fluid, lb/(hr.)(sq. ft.^{2} of cross section)
h_m = mean surface coefficient of heat transfer Btu/(hr.)(ft.^{2})(°F.)
K = thermal conductivity of film, Btu/(hr.)(ft.)(°F.)
m, n = dimensionless experimental exponents
μ = absolute viscosity of film, lb/(hr.)(ft.)

In both equations, h, the convection heat transfer coefficient, varies directly with G^n; therefore, an increase in the velocity of heating medium past the container should produce an increase in the rate of heating. This phenomenon, in general, explains the relationship shown in Fig. 3; f_h decreases (rate of heating increases) with increased velocity. The effect of velocity on the decrease in f_h with increasing heating medium temperature can be partially explained by an analogy between heat and mass transfer presented by Kern (4) where he indicates that the quantity of vapor transferred in a system increases with the sensible convection heat transfer coefficient h, which is, in turn, a function of velocity.

The diffusivity of a condensing gas diffusing through a non-condensing gas was calculated by an equation (13.31) recommended by Kern (4) and the diffusivity in sq. ft. per hr. for 165, 180 and 195°F. was found to be 1.15, 1.20 and 1.26, respectively. The diffusivity increased 4.8 percent from 165 to 180° and 4.6 percent from 180 to 195°F. This change in diffusivity at least partially explains the decrease in f_h with increasing heating medium temperature. The saturation pressure also increases with temperature, hence increasing the total diffusion potential. In addition, this phenomena may be interrelated with factor (2) above.

Insulating the retort caused the f_h of the jars to increase. Since at velocities other than zero insulating the retort cannot affect velocity, the effect must be due to some other factor. In the uninsulated retort, due to the increased load, there is considerably more steam flow into the retort for a given velocity than for the insulated retort. Since the velocity past the jar is the same, a considerable quantity of steam will have condensed in the bottom of the retort; some of this condensate may be present in the form of fog, altering the heat capacity and heat transfer characteristics of the steam-air mixture.

There is no known relationship between f_h and j for convection heating as there is for conduction heating products with finite film resistances (see Ball and Olson (1)). Pflug and Nicholas (7), in their study of the effect of heating medium, found evidence which suggests that f_h and j may be either interrelated or functions of a common parameter. There are subtle suggestions of the dependence of f_h and j by the data shown in Fig. 2.

Application of Data to Commercial Conditions

At the present time, there appears to be no way that these heating data can be used to predict the heating rate of food products in a steam-air mixture in commercial pasteurizing machines, since it is not possible to evaluate their temperature or velocity. The temperature of the steam-air mixture changes markedly from the inlet of the pasteurizer to the point most distant from the inlet. The temperature of the steam at the pipe orifice will be approximately 212°F. As the steam moves away from the nozzle it mixes with air so that by the time it reaches the bulb of the control device it is at the desired temperature which in commercial practice may be anywhere from 150° to 205°F.

General assumptions regarding the temperature pattern can be made, assuming the heating medium moves in a straight line from the steam orifice to the control point and on to some third point. If the container of food being heated is located between the sensing element of the control device and the steam nozzle, it can be assumed that the heating medium temperature to which the container is exposed will always be between the control point temperature and steam jet temperature.

If, on the other hand, the container of food is located between the sensing element of the control device and point 3, the temperature will always be below the control point temperature and will be somewhere between the control temperature and the temperature at point 3. This variable temperature heating pattern may be contrasted with saturated vapor heating where the temperature is for practical purposes constant throughout the system.

The velocity factors can be determined from a consideration of the typical pasteurizer unit. The commercial steam-air pasteurizer consists of a rectangular tunnel about 100 ft. long, 7 ft. wide and 2 ft. deep, open at both ends; there is a chain conveyor in the bottom and the steam distribution headers run lengthwise.

The units are usually made in sections 10 ft. long with canvas baffles between sections to keep drafts from blowing the steam-air mixture out of this open box. The velocity of the steam-air mixture past the container is a criterion of the steam jet design, the quantity of steam issuing from the jet, the degree of fullness of the pasteurizer and the location of the container in reference to the steam jet. The steam-air mixture velocity therefore is a function of heat load; a half full pasteurizer will have a lower velocity than a full pasteurizer and the velocity in a nearly empty pasteurizer will approach zero.

The results of this study indicate certain guiding principles for steam-air pasteurizer operation: (1) The heating medium temperature should be as high as proper product processing conditions will permit. (2) The steam distribution system should be designed to deliver steam uniformly to all points of the heating section at widely varying steam flow rates. (3) The location of the temperature control sensing element is very important since it is probably necessary to control temperatures over a range rather than at a point. (4) Although increasing the design velocity of the steam-air mixture may not be possible, the operators of steam-air pasteurizers, however, should be aware that the velocity is lower in a nearly empty pasteurizer than it is in a full pasteurizer.

SUMMARY

The effect of the velocity of steam-air mixtures on the f_h of jars of water has been determined at 165, 180 and 195°F. for velocities ranging up to 8 ft. per sec. The f_h decreases with increasing velocity and with increasing heating medium temperature. The j value remains approximately constant.

LITERATURE CITED

1. Ball, C. Olin and F. C. W. Olson (1957). *Sterilization in Food Technology.* McGraw-Hill Book Co., New York, N. Y., 654 pp.
2. Combustion Engineering, Inc. New York (1940). Steam Tables 3rd Ed., 38 pp.
3. Fischer and Porter Co., Halboro, Penn. (1947). Theory of the flowrater. Catalog section 98 A.
4. Kern, Donald Q. (1950). *Process Heat Transfer.* McGraw-Hill Book Co., New York, N. Y., 871 pp.
5. Kusak, Loyd James (1958). The condensation of vapors from non-condensing gases. Dissertation for Ph.D. degree, Cornell University (unpublished).
6. McAdams, W. H. (1954). *Heat Transmission* 3rd Ed. McGraw-Hill Book Co., New York, N. Y., 532 pp.
7. Pflug, I. J., and Nicholas, R. C. (1961). Heating rates at pasteurization temperatures as affected by heating medium and product in glass containers. Mich. Agr. Expt. Sta. Quart. Bul. 44: 153-165.

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