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EVAPORATION RATES OF DROPS IN FORCED CONVECTION WITH SUPERPOSED TRANSVERSE SOUND FIELD

POUL S. LARSEN and JOHN W. JENSEN*
Fluid Mechanics Department, Technical University of Denmark, Lyngby

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Abstract—Evaporation rates for single drops of distilled water suspended in dry air in upwards motion and subject to a horizontal standing wave sound field have been measured from photographic records. The range of drop diameter (0.8–2 mm), sound pressure (132–152 dB) and frequency (82–734 Hz) yielded parameter ranges of oscillating Reynolds number $16 < Re < 180$ and Strouhal number $0.3 < S < 2.8$ at convective Reynolds numbers in the range $5 < Re_c < 45$.

Empirical correlations are presented for $S < 1$ and $S > 1$, respectively. The data for these cases show the increase in Nusselt number owing to the sound field to be 5–90% and 8–30%, respectively.

NOMENCLATURE

a, particle amplitude of sound field [m];
c, mass fraction of vapor;
D, diameter of drop [m];
$\mathscr{D}$, binary mass diffusivity [m^2/s];
f, function;
h_{fg}, enthalpy of evaporation [J/kg];
Nu, time average Nusselt number;
Nu_c, steady convective Nusselt number;
Pr, Prandtl number;
Re, oscillating Reynolds number, $U_1 D/\nu$;
Re_c, convective Reynolds number, UD/ν;
S, Strouhal number, D/a;
Sc, Schmidt number;
Sh, Sherwood number;
t, time [s];
T, temperature [°C];
U, convective velocity [m/s];
U_1, velocity amplitude of sound field, $a\omega$ [m/s];
U_i, resulting instantaneous velocity [m/s].

Greek symbols

λ, thermal conductivity [W/m °C];
ν, kinematic viscosity [m^2/s];
ρ, density [kg/m^3];
τ, time constant in drop motion [s];
ω, angular frequency [1/s].

Subscripts

c, convective;
f, liquid;
s, drop interface;
∞, free stream.

1. INTRODUCTION

THE EFFECT of a sound field in a fluid (or equivalently the vibration of a surface) on the flow and on heat and mass transfer in external flows has been studied extensively [1], in particular for cylinders [2–8] and to less extent for spheres [9–14]. The vibration may be inadvertent, as in vibrating machinery, or intentional, as in unit operations, and it usually increases heat and mass transfer.

In the absence of convection the effect of an imposed sound field is to induce a pattern of acoustic streaming adjacent to the surface whenever there exists nonuniform distributions in Reynolds stresses associated with the fluctuating velocity components in an acoustic boundary layer. The steady streaming is a nonlinear effect of second order and is expected to be important only when free or forced convection is moderate or absent.

For the cylinder the theory of flow and heat transfer associated with a sound field alone is well documented [4, 6, 7]. The local effects of superposed free convection interfering with streaming has also been studied [8].

For the sphere the velocity field due to a sound field is more complex and solutions are known for some parameter ranges only [10]. The problem of superposed flows is complex and no analytical solutions appear to be available.

The characteristic parameters of the sphere problem are the oscillating Reynolds number $Re = U_1 D/\nu$ and the Strouhal number $S = D/a$, where D denotes the diameter of the sphere, a the particle amplitude and $U_1 = a\omega$ the velocity amplitude of the sound field of angular frequency ω. Superposing a uniform flow with velocity U, which may be longitudinal or transverse to the motion in the sound field, introduces the convective Reynolds number $Re_c = UD/\nu$ and the direction of flow as additional parameters.

Aside from an early study [9] of the flow pattern associated with acoustic streaming around the sphere experiments have been concerned with the net effect on heat and/or mass transfer. Superposition of free convection perpendicular to the sound field [11] at $S \sim 1$ gave an increase in heat transfer from a heated sphere for $Re > 220$, while there was no measurable effect for superposed forced convection at $Re_c \sim 10^4$.

* Present address: NIRO Atomizer A/S, Søborg, Denmark.

Heat transfer from small spherical bodies and evaporation rates of drops in a sound field longitudinal to the forced convection for $S > 1$ and $0 < Re/Re_c < 4$ were found to agree with quasi-steady analysis based on the time average velocity from the two effects [12]. A similar study of the mass transfer from a sublimating sphere [13] for $S \gtrsim 1$, $Re_c > 10^3$ and $0 < Re/Re_c < 1$ showed the increase in Sherwood number to depend on $(Re/Re_c)S^{0.45}$. The sparse data on evaporation rates of drops in a sound field without forced convection [14] suggest $Nu \sim (Re/S)^{1/2}$ as indicated by theory for cylinders [2] for $ReS > 1$ and $Re/S < 1$.

The purpose of the present study has been to investigate the feasibility of augmenting heat and mass transfer by the application of acoustic fields in unit operations such as drying processes involving drop evaporation. The case considered involves a sound field perpendicular to the convective motion at small Reynolds numbers typical of slip velocities encountered in sprays and in the free fall of drops.

2. EXPERIMENT

The evaporating drop was positioned at a pressure node of a 100 mm ID standing wave resonance pipe in a transparent test section where a vertical upwards stream of dry and temperature controlled air could be superposed on the horizontal sound field (Fig. 1). of the water. A timed sequence of still photographs provided data for equivalent drop diameter as function of time. The drops were rotationally symmetrical and nearly spherical. They were therefore approximated as ellipsoids whose volume was calculated from measured values of major and minor diameters on magnified pictures from the film. A sequence of every fifth photograph is reproduced in Fig. 2 along with a plot of the corresponding complete sequence of $(D/D_0)^2$ vs time. Here D_0 denotes the diameter at time zero, set at about 40–60 s after the drop was suspended. The figure also shows the evolution of the ratio of major to minor diameter, ranging typically from 1.08 to 1.03 for drop diameters in the range 1.6–0.8 mm.

The free stream temperature T_∞ ranged from 25 to 50°C, and the calculated contribution to heat transfer from conduction along the support and from radiation was found to be negligible (1–2% for a 1.0 mm drop) compared to that of conduction and convection through the gas phase to the drop, and was therefore ignored.

The conservation of mass for a spherical drop of diameter D and uniform temperature T_s combined with the interfacial energy balance gives for quasi-steady evaporation

$$dD^2/dt = -4\lambda(T_\infty - T_s)Nu/(h_{fg}\rho_f). \quad (1)$$

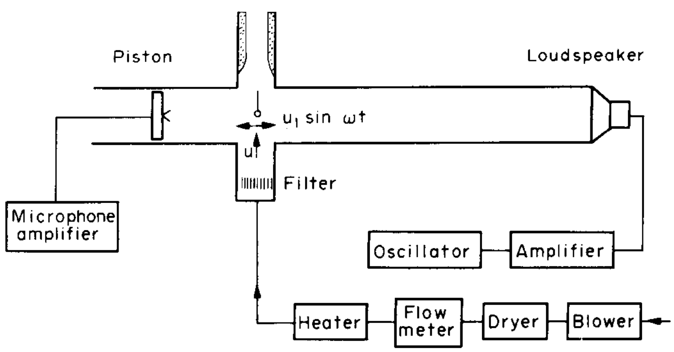

FIG. 1. Experimental set-up.

Velocity distributions of the sound field and the convection field were measured with hot-wire for calibration of microphone and flow meter readings. These calibrations were difficult to carry out because of the low velocities involved. The results were later checked by LDA measurements which also showed that the sound field was harmonic, $U_1 \sin \omega t$, to within a few percent. The velocity distribution across the standing wave pipe at the test section was approximately parabolic, however the variation in U_1 across a large drop at the location of suspension did not exceed 3%.

The drop of distilled water was suspended from a 20 µm diameter platinum wire bent into a horizontal circular arch of diameter 0.7 mm and attached to the end of a 100 µm dia vertical glass rod. A drop was placed by shooting the necessary amount of liquid towards the suspension wire from a capillary tube syringe. Care was taken to prevent any contamination

Combining the energy and mass balances at the interface gives

$$(T_\infty - T_s)/(c_s - c_\infty) = (h_{fg}\rho_f \mathscr{D}/\lambda)(Sh/Nu). \quad (2)$$

Phase equilibrium at the interface gives, for an ideal gas mixture and using vapor pressure data of water, a relation between c_s and T_s at the prevailing pressure. Combining this relation with (2) determines $T_s = T_s(c_\infty, T_\infty, Sh/Nu)$.

In each run T_s was calculated from measured values of pressure and T_∞, while $c_\infty = 0$ was imposed. Transport properties were evaluated at the film temperature. Note that T_s remains constant as long as Sh/Nu and properties are constant.

The thermal relaxation time of the drop is initially no greater than about 20 s. The time for complete evaporation—the reciprocal of the RHS of (1) multiplied by the square of the initial diameter—is about 200–500 s. After the initial period the quasi-steady

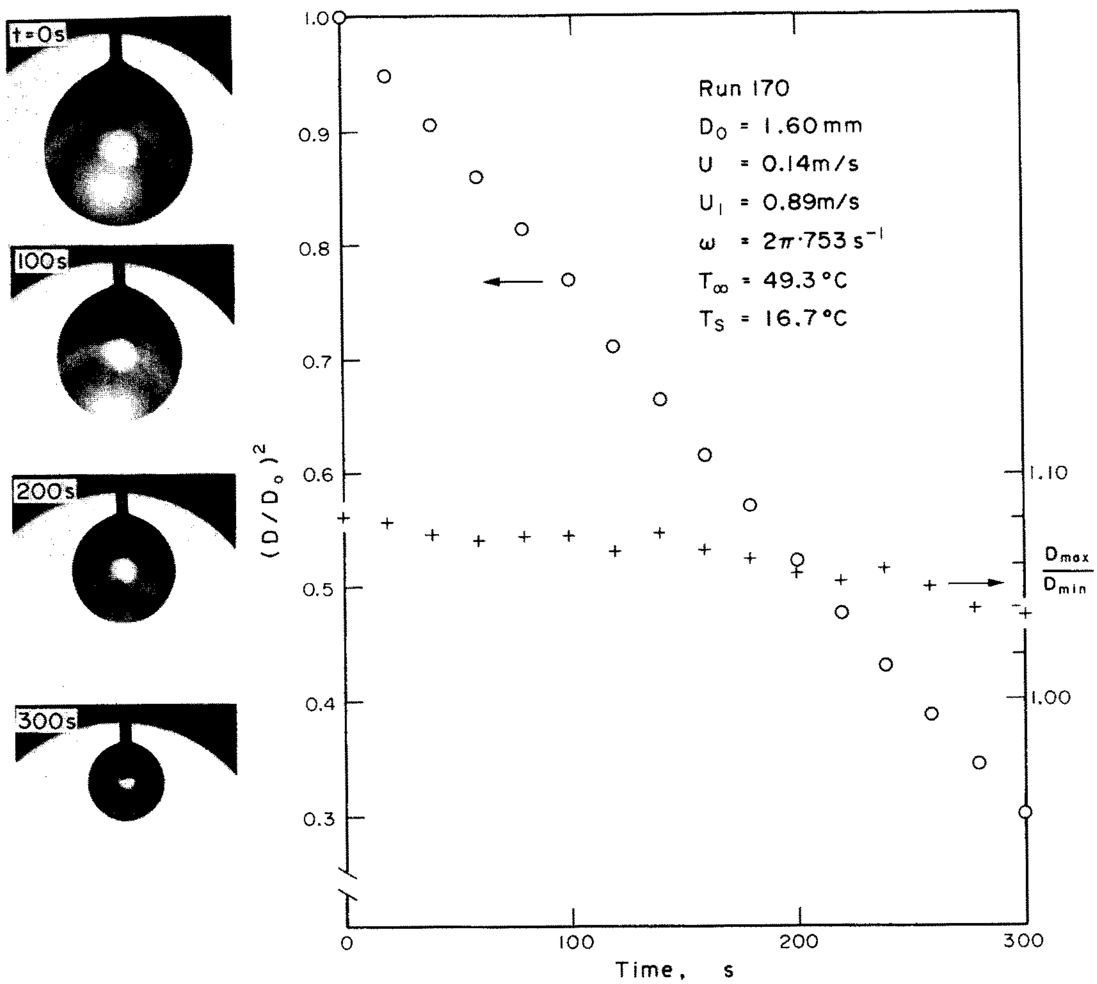

FIG. 2. Sample of experimental data. Equivalent diameter squared and ratio of major to minor diameter vs time. Photographs show every fifth picture in sequence.

formulation is thus valid and (1) yields instantaneous average values of Nu. Here dD^2/dt was calculated at the desired time by analytical differentiation of a second order polynomial fitted by least squares to all of the measured data points D^2 vs t. The RMS-error on $(D/D_0)^2$ was typically about 0.01. Each evaporating drop thus provided several data points at different diameters. Data used for subsequent correlation were selected within the picture sequence from No. 3 to 13 to exclude possible errors from the initial transient approach to quasi-steady state and from the influence of the support on small drops.

The selection of frequencies and intensities of the sound field were such that resonance oscillations of the drop as described in [12] did not occur. However, at the lowest frequency a pendulating motion with a maximum amplitude of about 3% of the drop diameter was observed in some runs. The resulting error on Nu was estimated to be less than 2%, hence the data was not excluded.

3. RESULTS

Forced convection

The experimental procedure was checked in some preliminary runs with forced convection alone. The results (Fig. 3) compare well with the standard correlation of Frössling (see the summary in [15])

$$\begin{aligned} Sh_c &= 2+0.55Sc^{0.33}Re_c^{0.5} \\ Nu_c &= 2+0.55Pr^{0.33}Re_c^{0.5}, \end{aligned} \quad (3)$$

where the analogy between mass and heat transfer has been implied. Since the Schmidt and Prandtl numbers are nearly constant and not too different ($Sc \sim 0.6$, Pr

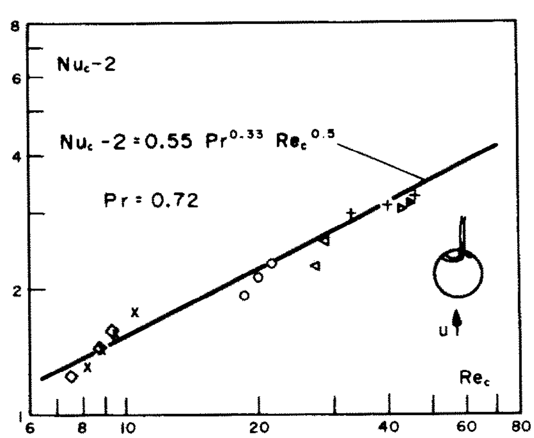

FIG. 3. Drop evaporation in convection.

~ 0.72) we take $Sh/Nu = 1$ in (2), ignoring its weak dependence on Re_c. The effect of free convection is considered to be negligible since $Gr/Re_c^2 < 0.1$ for all runs.

Next we consider evaporation of drops in forced convection with superposed transverse sound field. The acoustic parameter ranges of the data are shown in Fig. 4. In all cases $ReS > 1$ and we consider separately the data for $S < 1$ and $S > 1$.

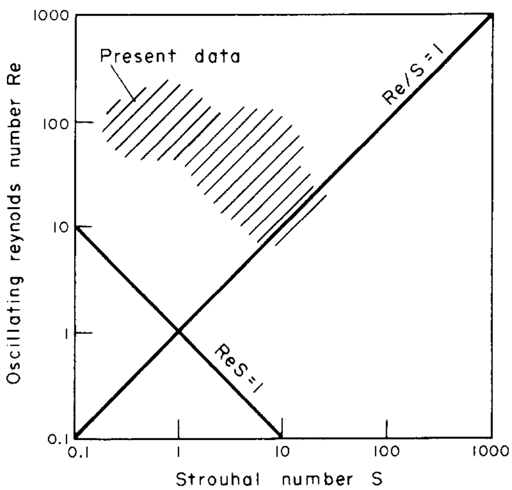

FIG. 4. Acoustic parameter ranges.

Small Strouhal number

For $S \ll 1$ we have an oscillating sound field of large amplitude superposed on the steady forced convection. The resulting flow is expected to be well approximated by quasi-steady convection with an effective instantaneous velocity whose magnitude is the vector sum of the two fields, $U_i = (U^2 + U_1^2 \sin^2 \omega t)^{1/2}$. Introducing this velocity in place of U in Re_c in (3) and integrating over a period of oscillation yields a time average Nusselt number which may be expressed as

$$(Nu-2)/(Nu_c-2) = f, \quad (4)$$

$$f(Re/Re_c) = \overline{U_i^{1/2}}/U^{1/2}$$

$$= \frac{1}{\pi}\int_0^{\pi} [1+(Re/Re_c)^2 \sin^2 \omega t]^{1/4}\, d(\omega t),$$

where Nu_c is associated with the steady convection as given by (3). Note, that $f(0) = 1$ and that $Re/Re_c \gg 1$ yields the asymptotic solution

$$f = \frac{\Gamma(3/4)}{\sqrt{\pi}\Gamma(5/4)}(Re/Re_c)^{1/2} = 0.7628(Re/Re_c)^{1/2}, \quad (5)$$

which is within 5% of the exact value for $Re/Re_c > 5$.

Clearly, (4) represents an upper limit because it assumes no flow reversal, or boundary layer or wake interference. Relative to the fluid the drop is seen to move on a sinusoidal path leaving behind it a concentration and thermal wake whose width is, say, at most on the order of the diameter of the drop. During one half period the relative displacement of translation is $\pi U/\omega$. As this quantity decreases towards the diameter of the drop, or as $\omega D/U = SRe/Re_c$ increases, the wake effect tends to diminish the heat and mass transfer. For fixed, and particularly for large, values of Re/Re_c the data confirm this trend and show f of (4) to decrease with increasing S. At $Re/Re_c = 21.2$, for example, a value of $SRe/Re_c = 8.4$ yielded 92% of the limiting value given by (5).

To include interference effects we expect f in (4) to be

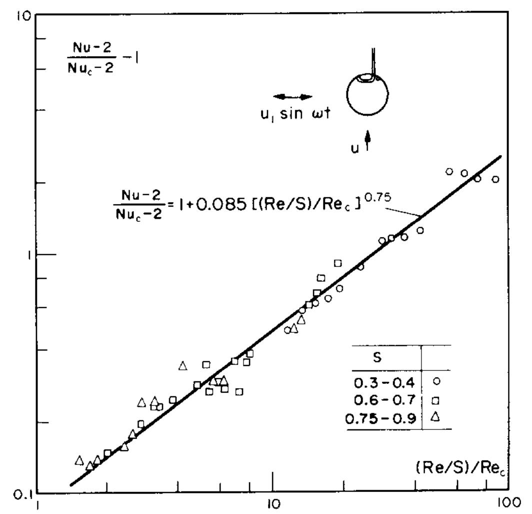

FIG. 5. Drop evaporation in convection with superposed transverse sound field; $S < 1$.

a function of both Re/Re_c and SRe/Re_c. Regression analysis of the data (Fig. 5) yields the empirical correlation

$$(Nu-2)/(Nu_c-2) = 1+0.085[(Re/S)/Re_c]^{0.75}; \quad (6)$$

$$0.3 < S < 0.9, \quad 7 < Re_c < 41.$$

Although some data are close to the upper limit implied by (5) there is insufficient data to yield a correlation that recovers this upper limit as S becomes small. The validity of (6) is therefore strictly limited to the parameter ranges indicated. Note, however, that as $Re/Re_c \to 0$ the effect of the sound field vanishes and (6) yields the forced convection limit.

Large Strouhal number

Before considering the data for superposed forced convection it is worthwhile to summarize the nature of the sound induced flow pattern for $S > 1$ in the absence of convection.

The viscous AC boundary layer (Stokes layer) is of thickness $(ReS)^{-1/2}$ and gives rise to the secondary steady streaming [10] shown in Fig. 6. These axisymmetric flows around the sphere may be qualitatively inferred from the corresponding two-dimensional flows around the cylinder.

For $ReS < 1$ (Fig. 6a) the viscous streaming layer is thick and fluid approaches the sphere in the directions of the oscillating motion. Its contribution to heat transfer may be calculated using the velocity field of steady streaming from [10] and the asymptotic procedure described in [6]. The thermal boundary layer is

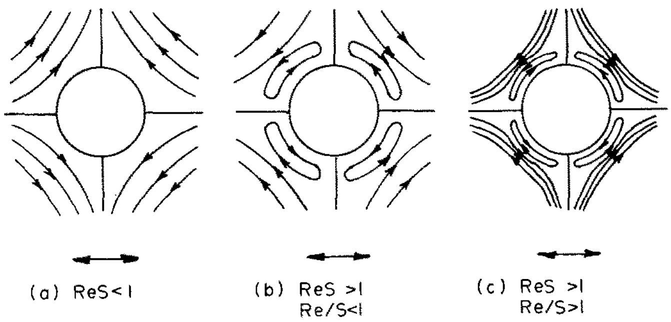

FIG. 6. Flow pattern of steady streaming around sphere for $S > 1$.

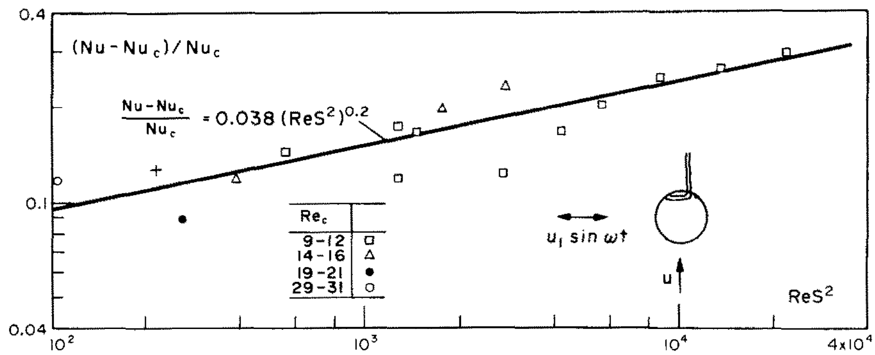

FIG. 7. Drop evaporation in convection with superposed transverse sound field; $S > 1$.

assumed to be thin (large Prandtl number approximation) and the solution to the temperature field shows the Nusselt number to depend on $Re^{2/3}$. However, since $S > 1$ and $ReS < 1$ implies $Re \ll 1$ this contribution will be negligible compared to that of pure diffusion.

For $ReS > 1$ and $Re/S < 1$ (Fig. 6b) there is a thin inner recirculating streaming layer and an outer streaming layer of large extent. The fluid leaves the sphere in the directions of the oscillating motion. Assuming the thermal boundary layer to be thick compared to the inner layer (small Prandtl number approximation) and using the velocity distribution from [10] the integral boundary layer solution gives to first order $(90/\pi^3)^{1/2}$ $(Pr\,Re/S)^{1/2}$ for the Nusselt number associated with streaming. Except for the constant this result is identical to that of the cylinder [6].

For $ReS > 1$ and $Re/S > 1$ (Fig. 6c) both inner and outer streaming layers are thin. The fluid leaves the sphere as rather well defined jets in the directions of the oscillating motion. Although the velocity fields are not available for this case we expect in analogy to the cylinder case that the Nusselt number due to streaming will depend on $(Re/S)^{1/2}$.

Now, the acoustic parameters of the present experimental data for $S > 1$ correspond to the case $ReS > 1$ and $Re/S \geqslant 1$, and the superposed convection corresponds to $1 < Re/Re_c < 7$ and $0.5 < (Re/S)/Re_c < 2.7$. Here Re/Re_c is the ratio of Reynolds numbers of oscillation and convection, while $(Re/S)/Re_c$ is the ratio of Reynolds numbers of streaming and convection. The increase in heat transfer due to the sound field was at most 30% of the forced convection contribution. For this reason and because the forced convection is expected to significantly interfere with the streaming and dominate the flow pattern the data was examined for a correlation of the relative increase in heat transfer in terms of acoustic parameters. The parameter group $(Re/S)^{1/2}$ resulting from the theory mentioned above did not appear to correlate the data. Regression analysis (Fig. 7) yielded the empirical correlation

$$(Nu - Nu_c)/Nu_c = 0.038(ReS^2)^{0.2}; \quad 100 < ReS^2 < 20\,000, \tag{7}$$

where Nu_c for convection is given by (3). The limited data for $ReS^2 < 100$, where S approaches unity, show considerable scatter and do not satisfy (7)'

4. DISCUSSION

It is of interest to compare the present results for transverse oscillation to the findings of [12] for longitudinal oscillation for the case $S < 1$. For $Re/Re_c > 1$ some features of boundary layer or wake interference are common. Following [12], equation (47), it should be possible to correlate the data for the quasi-steady case by the relation

$$(Nu - 2)/(\overline{Nu} - 2) = \overline{U_i^{1/2}}/\overline{U_i}^{1/2}, \tag{8}$$

where an over bar denotes time average and, for the present study, the instantaneous velocity is given by $U_i = U[1 + (Re/Re_c)^2 \sin^2 \omega t]^{1/2}$.

The numerator of (8)—Nu being the convective Nusselt number from (3) based on the time average velocity U_i—is not a measured quantity but merely a convenient computed reference. The RHS of (8) is a monotonically decreasing function of Re/Re_c approaching the asymptotic value $(\pi/4)/[\Gamma(5/4)]^2 = 0.956$ for $Re/Re_c \gg 1$. The present data, however, show also a marked dependence on S, particularly at large values of Re/Re_c in the range $2 < Re/Re_c < 21$.

In case of longitudinal oscillation [12] employs $U_i = U|1 + (Re/Re_c)\sin\omega t|$ and the RHS of (8) takes a minimum value of $8^{1/2}/\pi = 0.900$ at $Re/Re_c = 1$ and then increases asymptotically to the foregoing value of 0.956 as $Re/Re_c \gg 1$. The data of [12], which follow the latter relation closely, cover the region $0 < S < 0.8$, but are limited to the relatively low range $0 < Re/Re_c < 4$. Also, the motion of the sphere relative to the air is rectilinear with flow reversal and symmetrical boundary layer or wake interference.

In summary, the quasi-steady model (3) for $S < 1$ appears to yield correct bounds for the attainable increase in heat and mass transfer. The data shows that the upper bound is approached to within 10% when $S \sim 0.3$. At low values of Re/Re_c the dependence on S is probably negligible as found for a longitudinal sound field [12].

For $S > 1$, $Re/S \geqslant 1$ the superposed convection coincides in direction with the outer streaming layer on the upstream side of the sphere but opposes it on the downstream side. The Stokes layer is probably active within the convective boundary layer on the upstream side driving some outer streaming in a layer of same thickness as that of the convective boundary layer. The separate additive effect of the sound field on heat and mass transfer may be expected to depend on the Stokes layer parameter ReS and the streaming layer parameter Re/S, although the dependence found empirically in (7) does not lend itself to ready interpretation.

The correlation of data in [13] for the increase in mass transfer due to longitudinal oscillations for $1.2 < S < 4.9$, $0 < Re/Re_c < 1$ and $Re_c > 10^3$ involves the parameter $(Re/Re_c)S^{0.45}$ which, however, does not correlate the present data. It appears that there are distinct parametric differences between the effects of longitudinal and transverse sound fields. Also, it may be expected that differences between data for solid and liquid spheres arise due to internal circulation of the liquid driven by the convection.

The present data shows, for limited parameter ranges, the extent to which the heat and mass transfer to single spheres in forced convection may be augmented by the application of a transverse sound field. When applying the results to freely falling spherical drops it must be ascertained that the frequency is high enough that the drops do not follow the motion of the sound field. The fraction of the full amplitude attained is given by [16] $\omega\tau/[1+(\omega\tau)^2]^{1/2}$, where, in the Stokes range, the time constant is $\tau = D^2\rho_f/(18\nu\rho)$. Also, the extent to which the supporting wire affects the flow inside and outside the drop needs to be ascertained.

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VITESSE D'EVAPORATION DES GOUTTES EN CONVECTION FORCEE

Résumé—On mesure à partir d'enregistrements photographiques les vitesses d'évaporation pour des gouttes isolées d'eau distillée suspendues dans l'air sec en mouvement ascendant et soumises à un champ sonore. Les domaines des diamètres des gouttes (0,8–2 mm), de la pression sonore (132–152 dB), de la fréquence (82–734 Hz) correspondent à des nombres de Reynolds d'oscillation $16 < Re < 180$, à des nombre de Strouhal $0,3 < S < 2,8$ et à des nombres de Reynolds de convection $5 < Re_c < 45$.

On présente des formules empiriques pour $S < 1$ et $S > 1$. Les données pour ces cas montrent que l'accroissement du nombre de Nusselt en fonction du champ sonore peut être respectivement dans chaque cas 5–90% et 8–30%.

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Zusammenfassung—Es wurden aus Fotografien Verdampfungsgeschwindigkeiten einzelner in trockener, aufwärts strömender Luft suspendierter Tropfen destillierten Wassers unter dem Einfluß eines horizontalen Schallwellenfeldes bestimmt. Verändert wurden der Tropfendurchmesser im Bereich 0,8–2 mm, der Schalldruck im Bereich 132–152 dB und die Frequenz im Bereich 82–734 Hz. Es ergaben sich oszillierende Reynolds-Zahlen $16 < \mathrm{Re} < 180$, Strouhal-Zahlen $0,3 < S < 2,8$ bei konvektiven Reynolds-Zahlen $5 < Re < 45$. Für $S < 1$ beziehungsweise $S > 1$ werden empirische Beziehungen angegeben. Die Versuchsergebnisse für diese Fälle zeigen eine durch das Schallwellenfeld verursachte Zunahme der Nusselt-Zahl von 5–90% beziehungsweise 8–30%.

СКОРОСТЬ ИСПАРЕНИЯ КАПЕЛЬ ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ПОПЕРЕЧНОМ ЗВУКОВОМ ПОЛЕ

Аннотация — Исследовалась скорость испарения отдельных взвешенных капель дистиллированной воды в объеме сухого воздуха при направленном вверх движении и при одновременном воздействии горизонтального звукового поля. Для измерений использовались фотографические снимки. При диаметре капель от 0,8 до 2 мм, давлении звука от 132 до 152 дб и частоте колебаний от 82 до 734 гц диапазон пульсационного критерия Рейнольдса составлял $16 < R < 180$ и числа Струхаля $0,3 < S < 2,8$ при значениях конвективного числа Рейнольдса в пределах $5 < R_c < 45$. Эмпирические соотношения представлены соответственно для $S < 1$ и $S > 1$. Экспериментальные данные, полученные в этих случаях, показывают, что благодаря звуковому полю увеличение числа Нуссельта составляет соответственно 5–90% и 8–30%.