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An experimental study of a vibrating screen as means of absorption enhancement

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Abstract

An absorption enhancement technique and testing results are presented here. The technique increases the mixing rates at the solution–vapor interface. The second qualifying quality is that, unlike additives, this technique may not require periodic replenishment of organic compounds depleted by thermal instability effects. The results shown here cover a range of solution Reynolds numbers from 20 to 300, with mixing scales from 0.2 to 1 mm, and mixing frequencies from 20 to 100 Hz for solution flow over a vertical flat plate. Analysis shows that significant increases of the absorption rate are possible. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Absorption enhancement; Lithium bromide absorption machines; Active enhancement

1. Introduction

From the standpoint of thermodynamic theory, absorption cycles, inherently heat activated, are the equivalent of a power plant and of a vapor compression cycle combined. Consequently, when compared to vapor compression technology, absorption technology requires more heat exchangers and larger heat rejection rates on site than a simple vapor compression machine. Then, absorption machines tend to have higher initial costs than compression ones. Unless the initial costs are offset by lower activating energy costs or by considerations of dependability or availability of activating energy, vapor compression units may on the whole constitute a more cost-effective option to consumers.

In a competitive setting, a viable route for absorption technology is to reduce the size of the heat exchangers to meet lower cost and space requirements. Since most absorption units have low environmental impact, negligible noise level and superior part load performance, newer, smaller machines could become a feasible strategy to enhance the direct conversion of thermal energy into space conditioning streams. The route to more compact machines lies on enhancement of the heat and mass transfer processes that make up the cycle.

Absorption enhancement has long been one of the targets of designers and researchers. Over the years, this search has yielded a number of patents and articles in additives and in enhanced surfaces [1–3]. Additives have been proven to enhance absorption and generation both in the laboratory and in actual absorption machines [1,4–6], but they require periodic replenishment, due to thermal stability considerations. Enhanced surfaces do increase absorption, but their adoption is not universal due to performance over cost considerations. Indeed, high performance levels for both techniques have been reported in laboratory testing [7,8]. Yet, testing of both additives and enhanced surfaces in actual machines has uncertain and widely divergent results, largely unpublished. Perhaps an all encompassing statement not too far from the experience gathered up to this point by all researchers is that capacity improvements in the order of 30% for additives and in the order of 10–15% for enhanced surfaces over smooth tube absorbers optimized with recirculation are possible.

So, the adoption of enhancement techniques for absorption is not generalized, with some manufacturers

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Nomenclature

A	area, m^2
c	specific heat, kJ/kg K
g	acceleration of gravity, m/s^2
h	convective heat transfer coefficient, kW/m^2 K
$\mathring{m}$	mass flow rate, kg/s
p	pressure, Pa
$\mathring{Q}$	heat flow, kW
Re	Reynolds number
T	temperature, °C
U	overall heat transfer coefficient, kW/m^2 K
x	screen position, m

Greek symbols

δ	falling film thickness, m
Δ	screen vibration amplitude, m
ε	effectiveness
Γ	mass flow rate per plate unit width, kg/h m
μ	solution viscosity, Pa s/m^2
Θ	subcooling, K
ρ	density, kg/m^3
σ	standard deviation

Subscripts

a	absorber
amb	ambient
chw	chilled watercooling water
cw	cooling water
chw	chilled water
e	evaporator
g	gain
i	inlet
m	denotes mass transfer
o	outlet
s	solution
w	water

using them more consistently than others. For machines to truly become smaller and lighter, the high viscosity and low molecular diffusivity of lithium bromide solutions call for active enhancement. Previous attempts to incorporate active enhancement have met with varied success. Absorbers enhanced via rotating disks that produce a thin absorbent film projected to have large absorption rates were described in [9]. This concept led to the development of a rotating machine, Interotex, of small dimensions and testing results recently reported in the literature [10]. The concept is based on rotating heat exchangers at high speed, and using the consequent rotational forces to enhance heat and mass transfer and to circulate the working fluids.

The multiple approaches for enhancement adopted in the process industries have been summarized recently, [11]. To this excellent work we can add that in absorption, among passive techniques, incorporation of additives in the charge of the machine is the most common approach. Active enhancement techniques are still in the laboratory stage, although advanced prototypes have been built. Additives are thought of acting mostly at the vapor–liquid interface of absorbing solutions, mixing the interface by decreasing the time and motion scales in the liquid side [12].

The approach for enhancement in the present work hinges on using active enhancement on plate absorbers, still relying on gravity and conventional pumps for fluid circulation. A way to increase the interface mixing in plate absorbers, akin to the mode of action of additives but more intensely, is to employ a vibrating screen. The motion of a vibrating screen [13,14] mixes the vapor–liquid interface. The motion communicated to the solution is also projected to reduce the heat transfer resistance across the film [14]. Both actions should result in increased mass transfer and decreased thermal resistance, or in other words, in absorption enhancement. In this paper, we describe the experimental apparatus used to test the concept and we present a selection of the data, together with a discussion of the potential of the enhancement technique.

2. Experimental apparatus

The process diagram for the apparatus is shown in Fig. 1. A batch process is used, where the solution and the refrigerant are contained in tanks s1 and r1. The

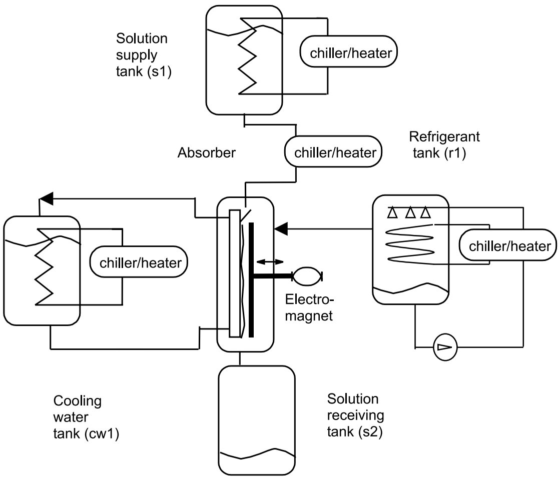

Fig. 1. Schematic of experimental apparatus.

solution is fed to the absorber via a speed-controlled pump. It is possible to adjust the concentration and temperature of the solution in tank s1 via direct access to add water or salt and a heater and chiller coil. An inline heater and coil can be used to trim the solution inlet temperature to the absorber. As the solution flows into the absorber, refrigerant vapor from tank r1 is absorbed. The heat to vaporize the refrigerant in r1 is furnished by a chiller-heater, and the heat rate can be extracted from the flow rate and temperature difference of the heat transfer fluid. The solution diluted with refrigerant flows to tank s2. A constant temperature bath (cw1) is used as the source of cooling water.

Vapor pressure, inlet and exit solution temperatures, density and concentration are monitored. Of special importance for the heat load determination, the chilled water flow rate and its temperature rise are also measured. The temperature sensors are Pt100 RTDs calibrated to ±0.03°C against a Hg thermometer. The solution flow sensors are mass flowmeters accurate to ±0.2% of reading, whereas the coolant and chilled water flowmeters are accurate to ±1% of reading. The pressure transducer is of the capacitance type, with an accuracy of ±0.15% of reading. Concentration is determined by sampling and measuring the density of the solution, with an accuracy of ±0.05% of salt by weight.

All the set variables are automatically controlled and their values maintained within narrow limits during a test. Tests typically run for 20 min, with a stabilization period preceding each test. Steady state is verified by observation of key variables (inlet and exit temperatures, absorber vapor pressure) displayed vs. time in a computer screen. The key variable for mass transfer repeatability is the pressure in the absorber. Even if all temperatures and flows are kept within narrow limits, small pressure variations may drastically influence absorption. This is the case, because the driving force for absorption in lithium bromide absorbers is very small, and any changes in the absorber pressure will affect this driving force significantly. The pressure is maintained by controlling the chiller-heater that activates the evaporator of the apparatus. Hence, constant pressure can be maintained throughout a test and from test to test. Under typical conditions, the following variables are controlled with the standard deviations noted, Table 1. Since Table 1 is based on the experimental data selected randomly, it is clear that the repeatability of initial conditions in the present set-up is accurate to 0.2% or better in all cases.

The kernel of the absorber is the assembly plate-vibrating screen, illustrated pictorially in Fig. 2. The absorbing LiBr solution flows down a water-cooled plate. In front of the plate, a vibrating screen is used to mix the solution. The reasons for mixing the solution as a prolongation of the additive effect are detailed in [12]. The mixing enhances the transport of water vapor in the solution, and, in the theoretical models, it also enhances the transport of heat into the absorber plate. The screen has a wire diameter of 0.2 mm and an unobstructed area of 70%.

The screen position is defined by the coordinate x, with origin in the flat plate, Fig. 3. As the electromagnet activating the screen is energized, a vibratory motion is communicated to the screen. The position x of the screen defines the equilibrium position at rest. The amplitude of motion, Δ, is measured from the rest position, and it is clear that $\Delta/2$ must be smaller than x. The equilibrium position, activating frequency and amplitude of motion can be varied within limits, which are in

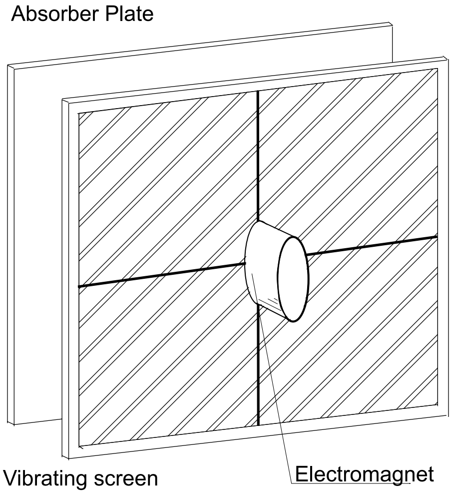

Fig. 2. Schematic of vibrating plate mechanism.

Table 1
Typical inlet conditions and degree of controllability of each

Variable[a]	Mean	Standard deviation, σ	σ as percentage of mean
Solution inlet temperature, °C	47.99	0.06	0.125
Solution inlet concentration, %	62.012	0.042	0.07
Solution inlet mass flow rate, kg/h^2	361.2	0.1	0.03
Cooling water inlet temperature, °C	31.99	0.03	0.09
Refrigerant pressure, mmHg	5.99	0.01	0.16

[a] Statistics corresponding to 10 values chosen randomly. Values corresponding to $Re = 200$.

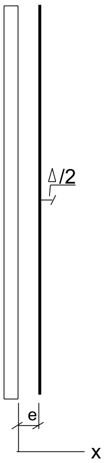

Fig. 3. Screen position coordinates.

turn somewhat dependent on each other. For instance, at low amplitude ($\Delta = 0.2$ mm), the frequencies can be varied from 20 to 100 Hz. At larger amplitudes ($\Delta = 1.0$ mm) the top frequency is only 67 Hz. The accessible frequencies are limited by the power requirements of the magnet. The position of the screen is determined via a non-contact pick-up sensor, and the acceleration from the reading of a piezometric accelerometer. The frequency is dialed in an inverter. The amplitude is determined from the frequency and acceleration values. The equilibrium position is known to be within ±0.08%, whereas the amplitude and frequency are known to be within ±4% and within ±0.002%, respectively, of their dialed values.

3. Performance parameters

Assessing the merits of an enhancement technique in an experimental absorber is difficult. This is so, for although it is possible to characterize the absorber by parameters defining its heat and mass transfer characteristics, the variable the designer is ultimately interested in is the cooling capacity that the absorber can maintain. Hence, we discuss here the heat transfer coefficients as correlated qualitatively with absorber capacity. We focus mainly here on capacity measurements at constant absorber pressure, facilitating the direct comparison of absorption mass fluxes. A dimensionless magnitude, namely absorber effectiveness, is presented for all data acquired at 60 Hz and at the 6 mmHg test pressure. A study including correlations of heat and mass transfer coefficients that apply to this absorber can be found elsewhere [15].

To determine the mass evaporated we measure the heat delivered to the evaporator by the chiller/heater by means of the chilled water temperatures and mass flow rate as follows:

$$\overset{\circ}{Q}_{\mathrm{e}} = \overset{\circ}{m} c_{\mathrm{w}} (T_{\mathrm{chw,i}} - T_{\mathrm{chw,o}}). \tag{1}$$

In addition, the refrigerant tank receives additional heat input from the surroundings that can be significant. This heat input evaporates the refrigerant that must be absorbed, and consequently must be considered. The evaporator was carefully calibrated by determining the heat removed by the chiller at constant refrigerant temperature as the outdoor temperature varied. The following equation was found to fit the heat gain data well:

$$\overset{\circ}{Q}_{\mathrm{e,g}} = 1.163 \times (13.576 \times (T_{\mathrm{amb}} - T_{\mathrm{r}}) - 7.9266). \tag{2}$$

Once the heat load is determined as the sum of the quantities given by (1) and (2), the refrigerant mass evaporated can be found dividing the heat rate by the heat of vaporization corresponding to the vapor pressure of the absorber.

Another parameter frequently used is the overall heat transfer coefficient, given by:

$$U = \frac{\overset{\circ}{Q}_{\mathrm{a}}}{A \dfrac{((T_{\mathrm{cw,i}} - T_{\mathrm{s,o}}) - (T_{\mathrm{cw,o}} - T_{\mathrm{s,i}}))}{\ln \dfrac{(T_{\mathrm{cw,i}} - T_{\mathrm{s,o}})}{(T_{\mathrm{cw,o}} - T_{\mathrm{s,i}})}}}, \tag{3}$$

with

$$\overset{\circ}{Q}_{\mathrm{a}} = \overset{\circ}{m}_{\mathrm{cw}} c_{\mathrm{cw}} (T_{\mathrm{cw,o}} - T_{\mathrm{cw,i}}). \tag{4}$$

The solution side heat transfer coefficient can then be obtained from the following expression:

$$\frac{1}{h_{\mathrm{s}}} = \frac{1}{U} - \frac{1}{h_{\mathrm{cw}}}, \tag{5}$$

where the coolant heat transfer coefficient is maintained as high as possible, and calculated to be in the order of 8300 $\mathrm{W/m^2K}$, and the wall resistance is negligible.

Yet another parameter of interest is the subcooling, defined as a temperature difference proportional to the departure from vapor pressure equilibrium between the solution and the vapor in the absorber, namely:

$$\Theta = T_{\mathrm{s}}(x_{\mathrm{o}}, p) - T_{\mathrm{s,o}}, \tag{6}$$

where $T_{\mathrm{s}}(x_{\mathrm{o}}, p)$ indicates the solution equilibrium temperature at the exit concentration and vapor pressure, and $T_{\mathrm{s,o}}$ is the measured solution temperature. Large subcooling values indicate that the solution has been cooled well below its equilibrium temperature, and that the mass absorption did not proceed at a sufficient rate to eliminate or decrease the subcooling. A direct consequence of subcooling is the concept of absorber effectiveness, defined as [16]

$$\varepsilon_{\mathrm{m}} = \frac{\mathring{m}_{\mathrm{actual}}}{\mathring{m}_{\mathrm{max}}}. \tag{7}$$

In Eq. (7), the actual mass absorbed is determined from experimental data. The maximum mass absorbed is calculated for solution entering the absorber at the given experimental conditions, and leaving at the measured exit temperature, but in equilibrium with the measured vapor pressure.

Other parameters of interest are the solution Reynolds number, calculated as

$$Re = \frac{4\Gamma}{\mu_{\mathrm{s}}} \tag{8}$$

and the solution film thickness, approximated with the Nusselt expression as

$$\delta = \left(\frac{3\mu_{\mathrm{s}}\Gamma}{\rho_{\mathrm{s}}^2 g}\right)^{1/3}. \tag{9}$$

4. Enhancement

To establish absorber performance with no enhancement, the screen was withdrawn and experimental runs were conducted for varying Reynolds numbers. Then, the screen position was adjusted to a distance from the plate corresponding to the average film thickness as given by Eq. (9), with visual observation confirming that the solution was in contact with the screen. The amplitude of vibration was varied, as well as the activating frequency. Selected results are presented below, along with a brief discussion of the observed trends.

We show in Fig. 4 the observed trend for low Reynolds numbers. At zero frequency, as the Reynolds number (i.e., the flow) increases, so does absorption. The activation of the screen produces a noticeable increase in absorbed fluxes, and it peaks at frequencies in the order of 40 Hz. Further frequency increments have little effect on enhancement.

In Fig. 5, we show the mass flux for larger Reynolds numbers than in the previous case. We see here that as Reynolds increases at zero frequency, absorption increases to a point: at larger Reynolds numbers absorption actually decreases. This is so, since an increased film thickness entails a hotter interface due to higher heat transfer resistance across the film, with a consequently reduced potential for mass transfer. Screen vibration has a much less pronounced effect here than at low Reynolds numbers, in fact, at low frequencies the effect appears negative at Reynolds of 200 and 300. Evidently, the small screen motion amplitude barely influences absorption.

Increasing the amplitude of the screen motion from 0.2 to 0.6 mm, at low Reynolds numbers, increased absorption. For instance at Reynolds equal to 60, peak values of 8.4 kg/h m^2 (compared with 7.12 kg/h m^2 with 0.2 mm amplitude, and 5.77 kg/h m^2 with no screen) were obtained. Within the given frequency amplitude and range, an optimal Reynolds number for enhancement exists. As the Reynolds number is increased from 60 to 300, the mass flux peaks between Reynolds equal 80 and *Re* equal 100, at 11.4 kg/h m^2.

Further increases in amplitude lead to yet higher absorption mass fluxes, Fig. 6. A maximum flux of 14.7 kg/h m^2 was measured at Reynolds equal to 80, with an activation frequency of 60 Hz and 1 mm amplitude. It is possible to speculate that at Reynolds equal to 300, larger peaks at higher values of activating frequencies may be obtained, reproducing the trend registered for Reynolds numbers of 40 and 80. The decay in absorption rates reflected in this figure shows that at high Reynolds numbers the solution is either not stirred enough or the wall heat transfer coefficient is not enhanced enough to fully realize the potential of interface

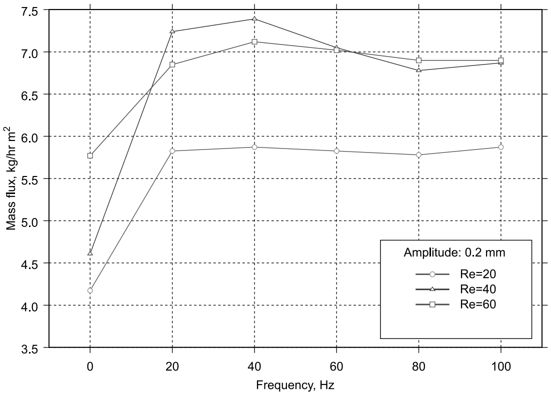

Fig. 4. Absorbed mass flux for low amplitude and small *Re* numbers.

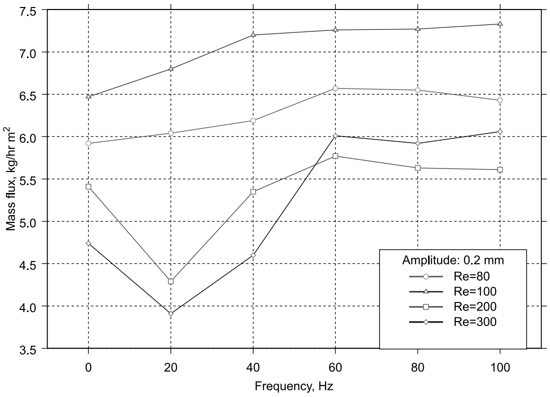

Fig. 5. Absorbed mass flux for low amplitude and high *Re* numbers.

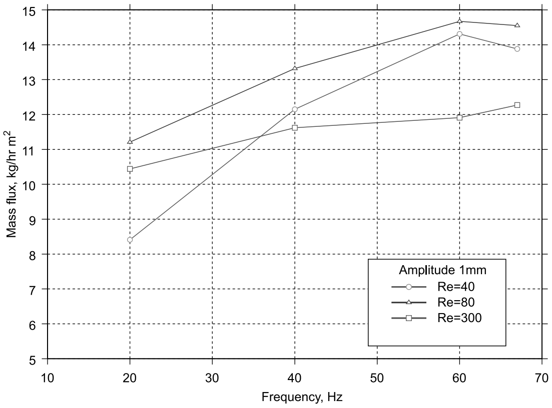

Fig. 6. Mass flux for large amplitudes.

mixing. We address this important question in the following section.

5. Discussion

Three matters are addressed here, each considered of value to those involved in absorber design. A long-standing question addresses whether overall heat transfer coefficients can indeed be employed to calculate the absorber capacity. The second one addresses the value of the data presented above. For the data to be fully representative of screen enhancement, the coolant-side heat transfer resistance must be negligible. The third matter concerns diagnosis of absorber operation via subcooling and effectiveness determinations.

In practice, the capacity of an evaporator is assumed to be proportional to the absorber heat transfer coefficient U. This proportionality was tested for two cases for all frequencies, at constant absorber vapor pressure. The Pearson correlation between mass flux and overall heat transfer coefficient was found to be strong, 0.93 for Reynolds equal to 20, and 0.98 for Reynolds equal to 80. It is possible to conclude that, at least at constant pressure and within the ranges of these experiments, the overall heat transfer coefficient and mass flux correlate well.

It is also important to determine whether the overall heat transfer coefficient and the solution-side coefficient correlate well. If affirmative, this means that the coolant-side coefficient is high enough to warrant reliable detection of absorption-side improvements. Two cases were considered. The first one corresponds to a series of intermediate mass flux values ($Re = 20$, amplitude $= 0.6$ mm), whereas the second corresponds to the highest absorption values measured in the present work. ($Re = 80$, amplitude 1.0 mm). In both cases, the correlation between U and h_{s} is noteworthy, for it is practically one. Hence, it is possible to conclude that the coolant-side heat transfer coefficient is indeed very large compared to the solution side. From this standpoint, improvements in absorption due to enhancement applied to the solution side should be readily detected in the present apparatus.

With the validity of our measurements established, we now focus on subcooling and effectiveness. Positive subcooling at the absorber exit indicates that absorption did not reach equilibrium, and that some of the absorbing solution could have absorbed more refrigerant than it actually did. The subcooling is a measurement of the balance between heat transfer rates and mass transfer rates: large subcoolings indicate effective heat transfer operations as compared to the mass transfer.

The subcooling trends are clear: for a given frequency, the subcooling (Fig. 7) increases with Reynolds, and then tends to decrease. At the high Reynolds number of interest for this study, and for the given experimental apparatus, the subcooling increases with amplitude. In other words, larger amplitudes tend to be more effective in augmenting the heat, rather than the mass transfer rates. It is remarkable that the subcoolings peak at a Reynolds number of about 200: beyond this limit the falling film thickness corresponds to this Reynolds, and the screen vibration reduces the subcooling. Yet, consideration of subcooling alone does not allow to diagnose whether mass transfer is improving or heat transfer is deteriorating.

The effectiveness, defined in Eq. (7), affords a complementary view of absorber performance. The effectiveness is shown in Fig. 8. This figure confirms the insights gained over years of practical experience, and offers new ones. Practitioners have long known that the best absorbers with no enhancement (typically in the form of splashing due to recirculation) operate with low Reynolds numbers, equivalent to thin films. Thus, the line for no screen displays decreasing effectiveness from

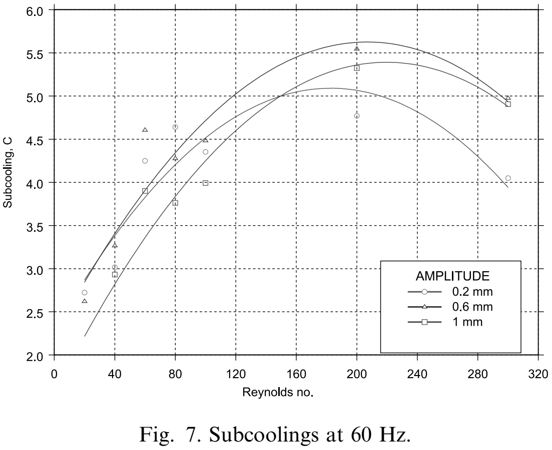

Fig. 7. Subcoolings at 60 Hz.

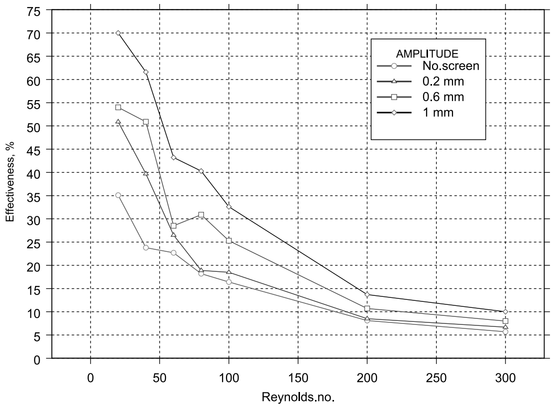

Fig. 8. Absorber effectiveness, 60 Hz.

35% to 5.7% for Reynolds from 20 to 300. Not surprisingly, most designs resort to small Reynolds numbers (i.e., thin films) to obtain large effectiveness. Screen vibration increases effectiveness throughout, but the effect is more pronounced at small Reynolds numbers. Effectiveness also increases with screen amplitude of vibration. Remarkably, for a Reynolds of 300, the effectiveness nearly doubles over its unenhanced value for amplitudes of 1 mm. At large Reynolds numbers, the absorber is operating at low values of effectiveness, which shows that heat transfer rates are increased at a faster pace than mass transfer rates. This diagnosis was not clear from the subcooling data.

A noteworthy aspect from Fig. 8 consists of the projected mass fluxes that could be achieved in this experimental apparatus. The inverse of the effectiveness (Eq. (7)) is a multiplying factor of the actual mass absorbed, yielding the maximum that could be absorbed for the heat transfer effected. Thus, with no enhancement and Reynolds equal to 300, the maximum value could be 17 times the observed value. With the screen at maximum amplitude and 60 Hz, the top value could be 10 times the detected value, or 119 kg/h m^2. It is our understanding that this mass flux would be in the order of 10–12 times larger than that obtained in current horizontal-tube absorber designs. All indications, taking the effectiveness and subcooling results together, are that mass transfer needs to be accelerated to reach higher effectiveness. Whether the interface temperature is too high or vapor absorption is obstructed in some way by the screen is not clearly known at this stage.

6. Conclusions

The outcome of this work is the demonstration of the screen effectiveness in enhancing absorption. In the absence of the screen, the effectiveness of the plate absorber is relatively low: around 35% for low Reynolds numbers. This effectiveness is inferior to that observed in commercial shell and tube absorbers, for which we project values in the order of 40–50%. The values obtained with screen vibration, in the order of 70%, show potential. There is little, if any, documented experience operating absorbers at high Reynolds numbers, in the order of 200–300. In this range, the screen doubles the operating effectiveness. It is certainly for large Reynolds numbers that this technique shows the most promise, and it is there where most of the potential seems to be present.

The effectiveness values at high Reynolds numbers are low; the observed trends indicate that to reduce the subcooling, it would be necessary to increment mass transfer rates via increased frequencies and perhaps amplitudes. The screen wire diameter (0.2 mm) and free area (70%) are important in that too small a free area may actually reduce absorption due to surface tension effects and obstruction, whereas too wide a spacing may not mix the entire interface thoroughly. Hence, the screen could be an important factor in future work.

Although absorption flux values higher than the ones measured at high Reynolds number seem possible, the screen power consumption remains a concern. It is clear that viscous forces arising in the solution oppose the screen motion. Hence, a recuperative system with a spring or set of springs storing some of the energy communicated to the screen may yield, adequately tuned, lower energy consumption than in the presently used apparatus. The prize of such design could be an absorber taking only one tenth of the area currently used to serve a given evaporator.

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