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The Synergism Between Heat and mass Transfer Additive and Advanced Surfaces in Aqueous LiBr Horizontal Tube Absorbers

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DISCLAIMER

THE SYNERGISM BETWEEN HEAT AND MASS TRANSFER ADDITIVE AND ADVANCED SURFACES IN AQUEOUS LIBR HORIZONTAL TUBE ABSORBERS

ABSTRACT

Experiments were conducted in a laboratory to investigate the absorption of water vapor into a falling-film of aqueous lithium bromide (LiBr). A mini-absorber test stand was used to test smooth tubes and a variety of advanced tube surfaces placed horizontally in a single-row bundle. The bundle had six copper tubes; each tube had an outside diameter of 15.9-mm and a length of 0.32-m. A unique feature of the stand is its ability to operate continuously and support testing of LiBr brine at mass fractions ≥ 0.62. The test stand can also support testing to study the effect of the falling film mass flow rate, the coolant mass flow rate, the coolant temperature, the absorber pressure and the tube spacing.

Manufacturers of absorption chillers add small quantities of a heat and mass transfer additive to improve the performance of the absorbers. The additive causes surface stirring which enhances the transport of absorbate into the bulk of the film. Absorption may also be enhanced with advanced tube surfaces that mechanically induce secondary flows in the falling film without increasing the thickness of the film.

Several tube geometry's were identified and tested with the intent of mixing the film and renewing the interface with fresh solution from the tube wall. Testing was completed on a smooth tube and several different externally enhanced tube surfaces. Experiments were conducted over the operating conditions of 6.5 mm Hg absorber pressure, coolant temperatures ranging from 20 to 35°C and LiBr mass fractions ranging from 0.60 through 0.62.

Initially the effect of tube spacing was investigated for the smooth tube surface, tested with no heat and mass transfer additive. Test results showed the absorber load and the mass absorbed increased as the tube spacing increased because of the improved wetting of the tube bundle. However, tube spacing was not a critical factor if heat and mass transfer additive was active in the mini-absorber. The additive dramatically affected the hydrodynamics of the falling film and a droplet flow regime was evident for testing at all tube spacings.

The mechanical mixing of the advanced surfaces increased the mass transfer to about 75% of that observed on a smooth tube bundle, tested with heat and mass transfer additive. Testing with heat and mass transfer additive and advanced surfaces demonstrated a synergistic effect which doubled the mass absorbed from that observed with only the advanced surface. The overall film-side heat transfer coefficient for the advanced tube bundles doubled with the addition of 500-wppm of 2-ethyl-1-hexanol.

KEYWORDS

falling-film absorber, LiBr brine, advanced surfaces, heat and mass transfer additive, absorption chillers

INTRODUCTION

The worldwide market recognizes absorption cooling as a quality air-conditioning system capable of supporting comfort cooling for commercial office complexes, medical facilities, department stores, and industrial assembly facilities. Absorption cooling is a favorable option because the equipment is adaptable to the available energy resources. Chillers can be steam-fired, direct-fired, or fired by a waste heat stream (Fallek 1985). Virtually all machines use LiBr and water as the absorption fluids, and the machines have capacities ranging from 5 to 2500 tons of refrigeration capacity.

The capacity of chiller equipment is determined by the evaporator load, which in turn is directly proportional to the mass of assimilated refrigerant in the absorber. Improving the absorber's transport processes leads to an increase in the chillers capacity to support comfort cooling. Hence, the design of the absorber is of paramount importance to the performance of the chiller. Typically, commercial chiller absorbers are of the shell-and-tube configuration with smooth tubes arranged in a horizontal bundle. The absorber is the largest of the system heat exchangers, and its design is based on empirical rules of thumb resulting in heavy, costly, and over-designed equipment. Any improvement in the transport processes would reduce the size and weight of the absorber and make the first-cost of absorption chillers more competitive to vapor compression technology.

It is well known that surface-active agents, added in small quantities to the solution, can significantly enhance the heat and mass transfer processes within the absorber. Kashiwagi (1985) experimentally proved that additives can substantially increase mass transport. The use of enhanced tube surfaces has also been shown to improve the heat and mass transport in falling films (Isshiki, Ogewa, and Sasaki 1991; Inoue 1988). Enhanced tubes, although common in heat exchangers, are not usually used in absorption chillers because of the perceived large increase in manufacturing cost associated with enhanced tubes. The affordable premium for enhanced tubes is small because of the low cost of the heat and mass transfer additive. However, the absorber is often the largest heat exchanger within chiller equipment; and the more compact its design and the more efficient its mass transfer, the less material and the smaller footprint needed to support comfort cooling. Kawamata et al. 1989 tested an integral fin tube which had axial grooves to promote convective motion of the film around the tube.

He claimed a synergistic effect occurred when he tested the tube with heat and mass transfer additive. It is therefore desirable to experimentally investigate the performance of enhanced tubes within the absorber to validate any improvement in the efficiency and reliability of chiller equipment.

Experimental work was conducted to test whether film mixing induced by enhanced surfaces is as effective as the chemical agitation induced by heat and mass transfer additive. The planned approach for the project was intended to yield information on whether enhanced geometry can increase mass transfer to the same level observed with 500 wppm of the heat and mass transfer additive, 2 ethyl 1-hexanol.

EXPERIMENTAL FACILITY

A mini-absorber test stand was used for the acquisition of experimental data. The stand (Fig. 1) is designed for continuous steady-state heat and mass transfer testing of aqueous LiBr solutions. Resistance heaters fire the boiler and are manually controlled to roughly balance the absorber load. Strong solution from the boiler is tempered using a tube-in-tube heat exchanger; it enters the absorber saturated. A right-angle high-vacuum valve with stainless steel nested bellows is used to trim the vapor pressure entering the absorber. A trough and slide-bar attachment drip strong solution over the tube bundle (Fig. 2). The slide-bar is placed in the floor of the trough atop the trough's drip ports. The ports in the trough and slide-bar are vertically aligned and are spaced about 0.003-m apart along the length of the absorber tube bundle. The slide-bar is attached to a remote vacuum feed-through. The feed-through is manually adjusted to move the slide-bar and change the opening of the drip ports while testing. Changing the falling film flow rate or changing the brine's concentration affects the flow distribution falling from the drip ports. The trough, slide-bar and vacuum feed-through attachment are key features for dispensing a uniform drip pattern onto the tube bundle. Coupled heat and mass transfer occurs as the falling film of LiBr brine (the absorbent) absorbs water vapor (the absorbate). The film is cooled by water flowing inside the absorber tube bundle: the flow is cross-countercurrent to the falling film. Weak solution leaving the absorber is pumped from a sump tank and injected back into the boiler to complete the loop.

The horizontal absorber (Fig. 2) is designed for visual inspection of the falling film and has provisions for testing a single-row bundle of tubes. One end of the mini-absorber has a 0.254-m-diameter sight glass: the other end is fitted

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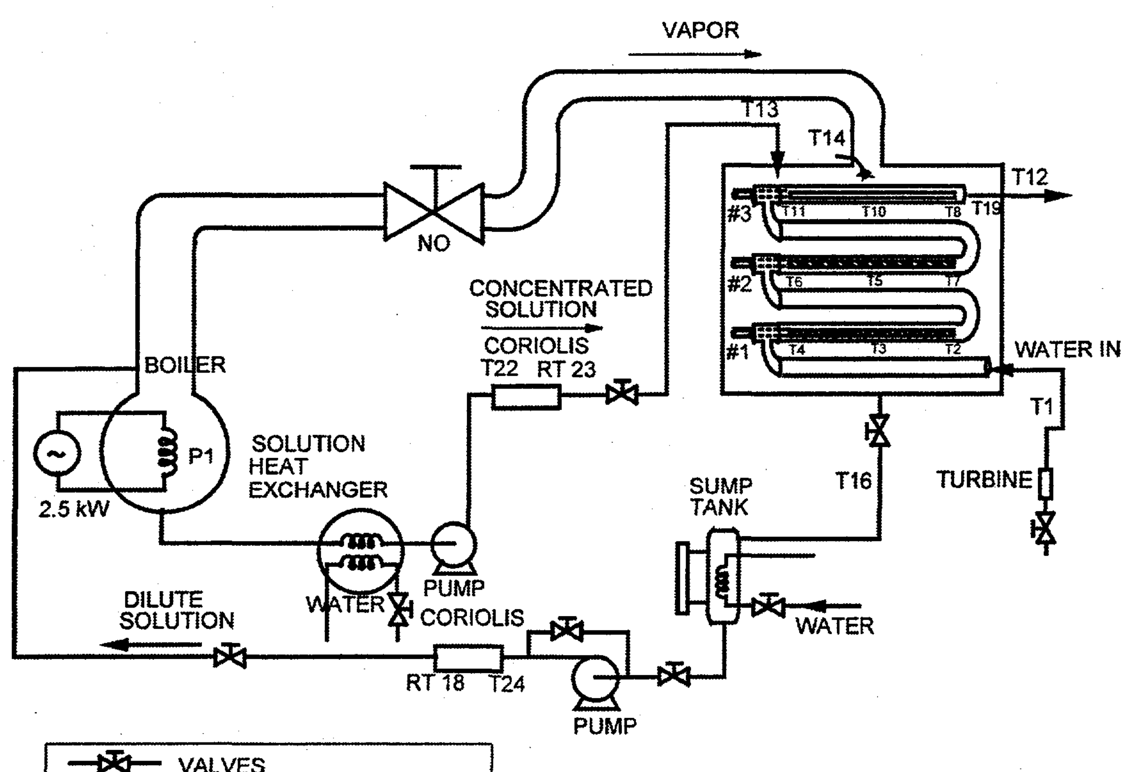

Fig. 1. Refrigeration schematic of the mini-absorber test stand.

with a larger 0.48-m-diameter disk made of 0.02-m-thick borosilicate glass for viewing the hydrodynamics of the falling film.

The tube bundle (Fig. 2) is attached to an U-brace, which can be horizontally adjusted to center the bundle under the dripper. Vertical adjustment is accomplished by using the vertical slots in the U-brace. One can easily and quickly remove the bundle and a setup a different tube surface. Figure 3 shows the technician, J. Atchley, setting up a new bundle. The racks are made of Teflon and are designed to maintain the tube-to-tube spacing and the level of all tubes. The racks were made the same depth as the dripper to help center the assembly (Fig. 2). The return bends of the bundle are made with O-ring compression fittings that are rated for vacuum. These fittings were made by modifying standard 90° elbow and tee sections. Three sets of Teflon racks and fittings were made for testing at different tube spacings. Once the bundle is set up, coolant lines are connected using flexible vacuum hoses (Fig.2).

A purge tube is located in the lower portion of the absorber to pull noncondensables from the test section. The purge tube is connected to a nitrogen cold trap that assists a 1.4 L/s pump in pulling vacuum on the entire system. The purge tube, cold trap, and pump are also used to increase the concentration of the strong solution. The stand is started at a low concentration of about 60 wt% LiBr to get all solution lines hot. A vacuum pump pulls water vapor from the absorber and holds it frozen in the cold trap. Purging of water vapor continues until a set concentration of strong solution entering the absorber is reached. This technique helps minimize the danger of crystallization, especially for testing at 62 wt % LiBr. To reduce the concentration, distilled water is pulled under vacuum into the boiler and the system is operated for about 1 hour to equilibrate concentration throughout all lines.

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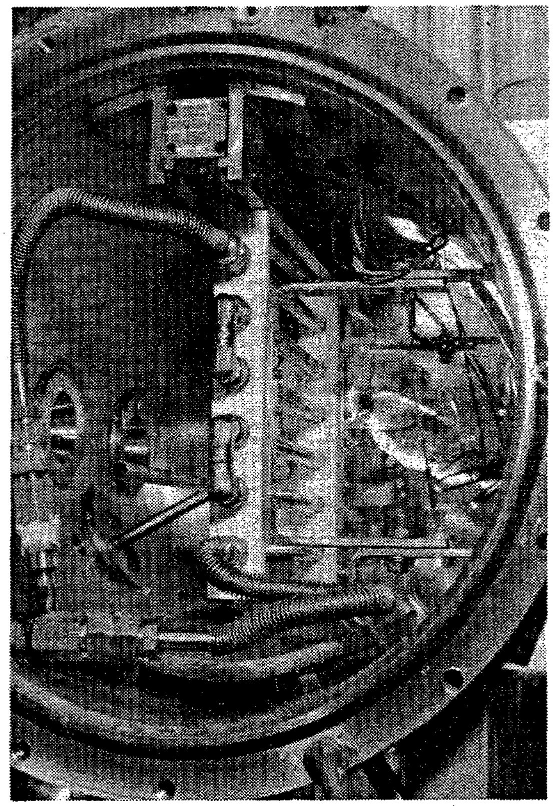

Fig. 2. Side-view of mini-absorber looking through Lexan view port.

The test stand is instrumented at various locations for the measurement of temperature, pressure, density, flows, and boiler power. An instantaneous water heater of 24 kW capacity tempers cooling water entering the absorber. Coriolis mass flow meters measure the mass flow rate and the density of the solution entering and leaving the absorber. A power transducer measures the heat input to the boiler; current draw is monitored with an analog current meter. A turbine meter is used to measure the coolant flow rate. Absorber pressure is measured using a transducer with a calibrated range of 0 to 10 mm Hg. Two temperature controllers regulate heat to the solution lines entering and exiting the absorber to guard against potential crystallization at concentrations ≥ 60 wt %. Temperatures are measured using sheathed thermocouples inserted through compression fittings into the refrigerant and the solution lines.

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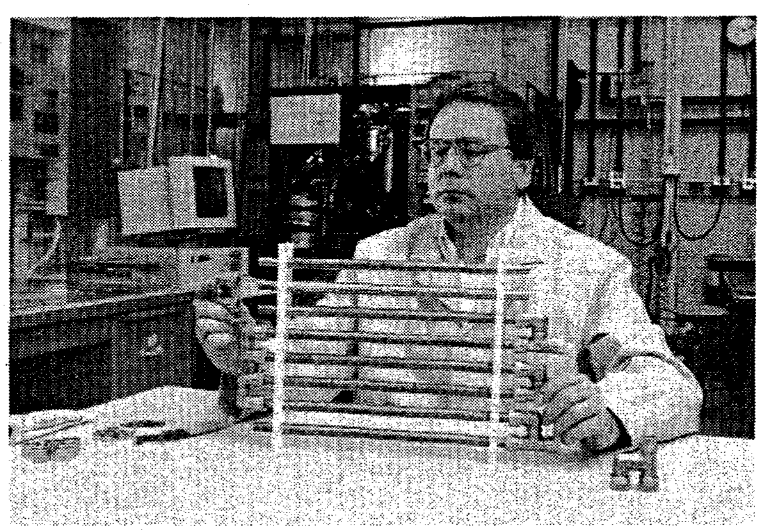


Fig. 3. J. Atchley completes the setup of a tube bundle for testing in the mini-absorber.

The effect of noncondensables was minimized through extensive leak-checking with a mass spectrometer tuned for helium leak detection. Vacuum pumps held the test stand under vacuum during leak checking. Individual parts of the stand were bagged, and the bag was pressurized with helium to check for any leakage into the stand. As a final check, the mini-absorber, generator, sump tank, and connecting plumbing were evacuated to 100 μm using vacuum pumps and nitrogen cold traps. The pumps were valved off from the system, and the pressure increase within the system was recorded vs time for a duration of about 9 days. Outgasing from the inner walls of the rig continued for about 160 hours. Afterwards, the pressure leveled off proving the system leak tight.

LABORATORY APPROACH

Initially, the loop was charged with deionized water. Tubes were first tested according to Wilson's method to validate the use of the Dittus-Boelter correlation for predicting the water-side heat transfer coefficient. We tested with a falling film of distilled water flowing over the tube bundle. A solution heat exchanger tempered the shell-side water, and magnetic pumps controlled water flow over the tube bundle. During these tests, care was taken to keep the shell-side entering water temperature constant. Results correlated well, according to Wilson's method; and hence, several tests were conducted at falling film-to-coolant temperature differences typical of the temperature gradient expected in aqueous LiBr absorption experiments. The results are discussed by Miller, et al (1998) and proved that the water-side convective coefficient can be accurately predicted using the Dittus-Boelter correlation with the Prandtl number raised to the 0.4 power.

We tested a broad spectrum of surface geometries having extended surfaces on the shell-side to characterize the effect of film mixing on absorber performance. Wolverine donated three styles of patented tubes: the Turbo-C, the KoroChil, and two Trufin tubes having 40 and 19 fins per inch. Two of these surfaces, the Wolverine KoroChil and Turbo-C tubes, also had tube-side enhancements for promoting turbulence in the coolant. York donated the Hitachi Thermoexcel-A, and Trane donated a proprietary tube. ORNL manufactured an axial-grooved tube and a spirally fluted tube. ORNL also procured integral pin-fin tubes with two different fin pitches. Sample pictures and the salient features of these advanced surfaces can be found in Miller et al. (1998).

Each tube surface was bright-dipped just before testing. The enhanced surfaces were chemically cleaned to remove oxides, oil, grease, and dust from the outside surface. Pickling of the tubes was done in a pickling solution usable for cleaning copper. Tubes were vapor-degreased using perchlorethylene, water-rinsed, and then dipped in a dilute solution of hydrochloric acid for 1 to 2 min. The tubes were then bright-dipped for 1 to 2 min in a solution containing chromic and sulfuric acids and again rinsed in water. They were dipped again in the hydrochloric acid bath for about 1 min to remove any remaining residue. Finally, the tubes were rinsed in hot water, blown dry, and baked at 150°F for about 1 hour. The tubes were also handled with cotton gloves to minimize any body grease contacting their surfaces. The cleaning procedure and handling precautions eliminated any fouling that might confound measurements of either the coolant-side or the film-side heat transfer coefficient.

The test stand was charged with aqueous LiBr having about 168 wppm of molybdate ($LiMoO_4$) corrosion inhibitor. The solution was supplied by Foote Mineral Corporation and is typical of solution used in industry. In previous testing, solution doped with lithium nitrate inhibitor did little to prevent corrosion damage to the test loop.

DATA REDUCTION

The absorber load is defined as the heat transfer rate (Q) from falling film to cooling water. Given the load and the log-mean-temperature-difference, the average overall heat transfer coefficient was calculated by the following equation :

$$Q = (UA)_o \Delta T_{lm} , \tag{1}$$

where

$$\Delta T_{lm} = \frac{(T_{s,eq} - T_{c,e}) - (T_{s,e} - T_{c,i})}{\ln \dfrac{(T_{s,eq} - T_{c,e})}{(T_{s,e} - T_{c,i})}} , \tag{2}$$

When the temperature differences are summed across the thermal resistance of the falling film, the tube wall, and the coolant, the local temperature drop at any cross section of the tube would be as follows:

$$(T_{if} - T_c) = (T_{if} - T_o) + (T_o - T_{in}) + (T_{in} - T_c) \tag{3}$$

Knowing that the heat transfer at any cross section of the tube is constant, the equations of heat transfer for the respective thermal resistance can be substituted into Eq. (3), and the average overall heat transfer coefficient can be solved for and expressed as follows:

$$\frac{1}{(UA)_o} = \frac{1}{h_s A_o} + \frac{\ln(D_o/D_{in})}{2\pi k_p L} + \frac{1}{h_c A_{in}} , \tag{4}$$

The thermal conductance of the test tube is known, and Wilson's method was used to determine the convective coefficient (h_c) of the coolant. Therefore, the film-side heat transfer coefficient (h_s) can be back-calculated from Eq. (4).

EXPERIMENTAL RESULTS—SMOOTH TUBE BUNDLE

Laboratory testing of a smooth tube bundle was conducted with and without heat and mass transfer additive under conditions observed in commercially available single-stage equipment. Testing with additive was conducted with about 500 wppm of 2-ethyl-1-hexanol. Testing was conducted with saturated solution entering the absorber at concentrations of 60 and 62 wt % LiBr. The absorber pressure was fixed for all testing at 6.5 mm Hg, while the coolant flow, coolant temperature, and falling film mass flow were varied to allow us to observe each parameter's effect. The tube-to-tube spacing was an additional test parameter used only for the smooth tubes. Smooth tube bundles were tested for the above-stated range of conditions for tube wall-to-tube wall spacings of 0.0191, 0.0127, and 0.0286 m. In some equipment, the tube wall-to-tube wall spacing is tighter than 0.0127 m.

The Effect of Coolant Temperature

Increasing the coolant temperature caused a near-linear drop in the load and in the mass absorbed (Fig. 4). The result was observed whether we tested with or without heat and mass transfer additive. As the outdoor temperature increases, the coolant temperature leaving a cooling tower increases, which causes a pinch point in the approach temperatures between the coolant and the strong solution entering the absorber. This pinch point increases the partial pressure of the absorbate within the liquid-vapor interface. The driving potential drops, which in turn decreases the mass absorbed and the absorber load as coolant temperature increases (Fig. 4). Conversely, the more vapor that is absorbed, the larger the load. The absorber load, then, is a good indicator of the absorber's performance.

The addition of 500 wppm of 2-ethyl-1-hexanol caused a dramatic increase in the mass absorbed. For testing at 29.4°C coolant, the absorbed mass increased by a factor of 2.5 of that observed with no additive. Inoue (1988) measured similar performance boosts while testing a smooth tube bundle with 250 wppm of 2-ethyl-1-hexanol. His mini-absorber had six rows and eight tiers of 0.019-m outside diameter smooth tubes of 0.25-m length.

The Effect of Tube Spacing—*Testing without Heat and Mass Transfer Additive*

In these experiments, we tested the smooth tube at tube wall-to-tube wall spacing of 0.0127, 0.0191, and 0.0286 m. Test results showed the absorber load and the mass absorbed increased as the tube spacing increased because of the improved wetting of the tube bundle (Fig. 4). We observed all six tubes to be fully wet at the largest tube spacing; however, at the tighter spacing, we observed several dry spots on the lower four tubes of the bundle.

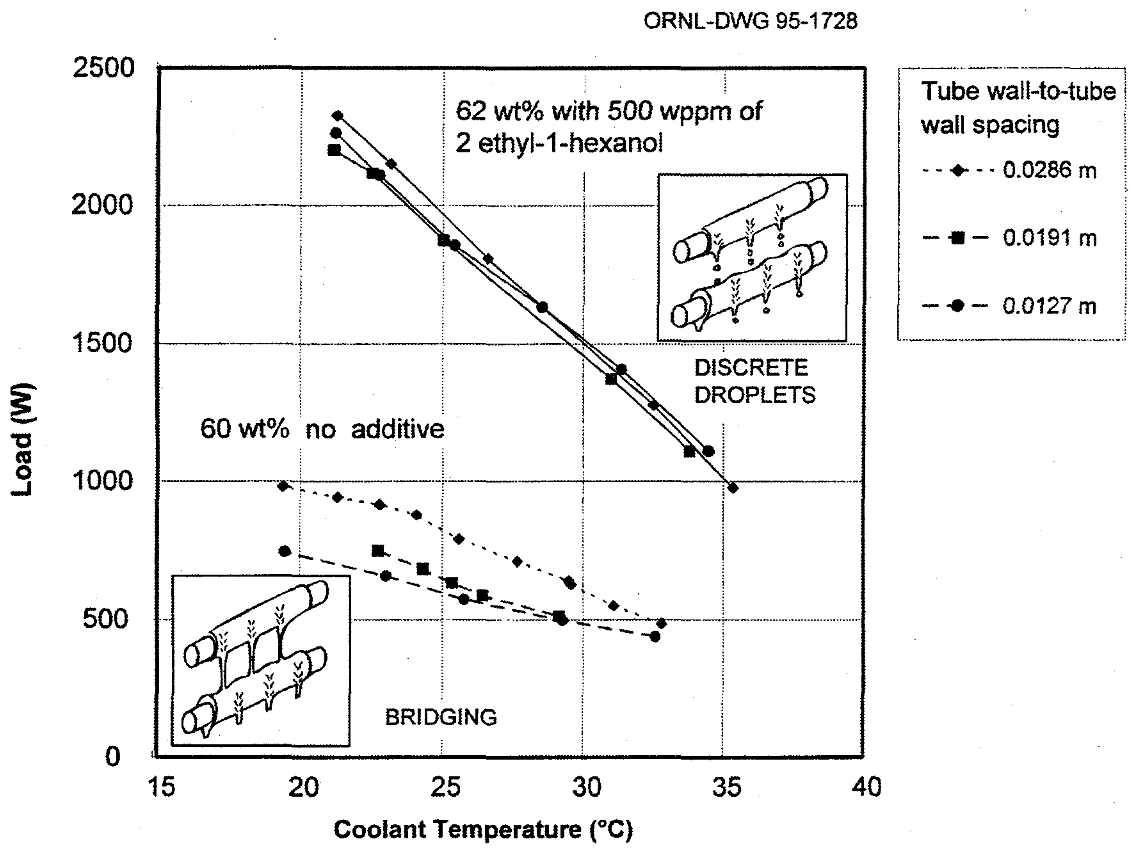
As the tube spacing was decreased, we observed the flow pattern between tubes to transition from a droplet to a bridging pattern, as characterized in Fig. 4. Similar findings were seen by Mitrovic (1986) for vapor condensation on a vertical row of horizontal tubes. Mitrovic observed that the region of discrete droplets usually appears with lower flow rates and larger tube spacing. With increasing flow and/or tighter spacing, the pattern bridges from tube to tube and finally transitions into a sheet flow. Yung, Lorenz, and Ganic (1980) observed that for low-viscosity liquids like water, the length of the droplet just prior to detachment could be predicted by a Taylor instability analysis. A droplet of higher viscosity than LiBr brine was observed by Fujita (1993) to have a narrow tail form after it detached from the bottom surface. The trailing fluid would then break up into several smaller droplets.

High-speed video taken of the largest tube spacing showed droplets with their tails just spanning the gap between tubes. At the tighter tube spacing of 0.0127 and 0.019 m, we observed that the flow from each tube did not form droplets; rather, it flowed as a stream from tube to tube. As we increased the spacing from 0.0127 to 0.0286 m, we observed the film to form droplets that splashed onto the lower tube. This splashing action helped wet the tubes, and as result, at the largest tube spacing of 0.0286 m we observed the lower tubes in the bundle to be fully wetted.

The Effect of Tube Spacing—*Testing with Heat and Mass Transfer Additive*

The inclusion of about 500 wppm of 2-ethyl-1-hexanol dramatically affected the hydrodynamics of the falling film. The tube bundles, regardless of the tube spacing, fully wetted when tested with the heat and mass transfer additive. A droplet flow pattern was observed on the underside of all the absorber tubes, even when testing at the tightest tube spacing of 0.0127 m. Bridging of the film between tubes was not observed. Therefore, with the bundle fully wetted and with a droplet flow regime evident for all tube spacings, the load and the mass absorbed are not strongly affected by tube spacing (Fig. 4). Hence, tube spacing is not a critical design factor if heat and mass transfer additive remains active in the absorber.

Photographic evidence of the flow, observed at the largest tube spacing of 0.0286 m, was documented for the film falling around the tubes and also dripping from the underside of the tubes (Fig. 5). A camera with a shutter speed of 200 frames/s was used to depict the hydrodynamics without and with heat and mass transfer additive. Figure 5a shows a still frame (photo taken at 1/200 s) of the film as smooth laminar; no additive was present, and the tube bundle was fully wetted. Note that there is only one drip port in the field of view and that the droplet with its attached tail has just splashed onto the lower tube. Adding 500 wppm of 2-ethyl-1-hexanol to the vapor side of the absorber caused the film to instantly undulate, with wavy rivulets flowing around the tube perimeter (Fig. 5b). The rivulets oscillated laterally along the tube in a fashion similar to that observed by Cosenza and Vliet (1990). Testing also showed local spots with a film thickness much reduced from that observed for laminar flow with no additive, although this is difficult to see in Fig. 5a and 5b. The additive agitates the solution at the liquid-vapor interface and induces a mixing within the film which some researchers believe is due to a Marangoni convection. Others believe the agitation is caused by a chemical catalytic mechanism that reduces the interface activation energy required for absorption. The chemical mechanism is very difficult to determine because of the dynamic complexities and the dearth of experimental data. Both theories do reason, however, that the additive rests on the liquid-vapor interface and therefore induces strong surface tension forces. The additive affects the interaction between gravity and surface tension forces and causes the frequency and number of droplets falling from the underside of a given absorber tube to increase (Fig. 5c). The still frame in Fig. 5c shows that the detachment length is reduced from that observed with no heat and mass transfer additive. Also, the droplet spacing along the absorber tubes decreased dramatically and therefore the number of droplets per unit length increased (Fig. 5a vs Fig. 5c).



Coolant Reynolds No.	5700
Absorber Pressure	6.5 mm Hg
Falling Film Flowrate	0.0091 kg/s
Absorber Surface area	8.36E-02 m^2

Fig. 4. The effect of the coolant temperature and the tube wall-to-tube wall spacing on the load supported by a 6-tube bundle of smooth tubes.

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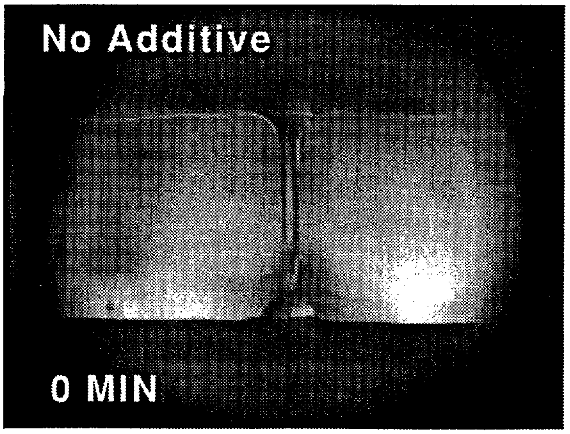
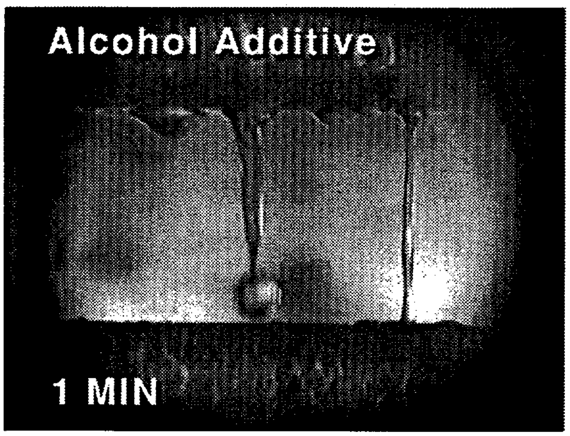
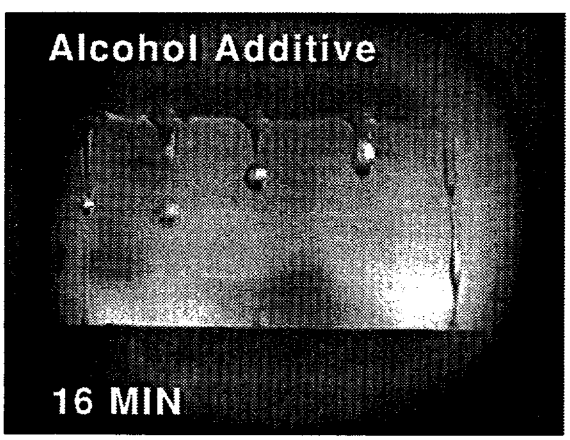


(a.) smooth laminar falling film



(b.) wavy rivulets observed instantly after adding additive on vapor side



(c.) number of droplets increased with inclusion of additive

Fig. 5. The hydrodynamics of the falling film was photographed with a camera having a shutter speed of 200 frames per sec.

Close inspection of Fig. 5b and 5c shows that the film hydrodynamics transition from the strong surface tension induced wavy-rivulet flow (Fig. 5b) to a smoother, almost laminar flow (Fig. 5c). The transition is at present unexplainable. As a caveat, the load and the mass absorbed, which were continuously monitored, did not drop, proving that noncondensables were absent. Also, the sump tank, located at the exit of the mini-absorber (Fig. 1), was purposely run dry to avoid pooling the alcohol in the sump.

The Effect of the Coolant Mass Flow Rate

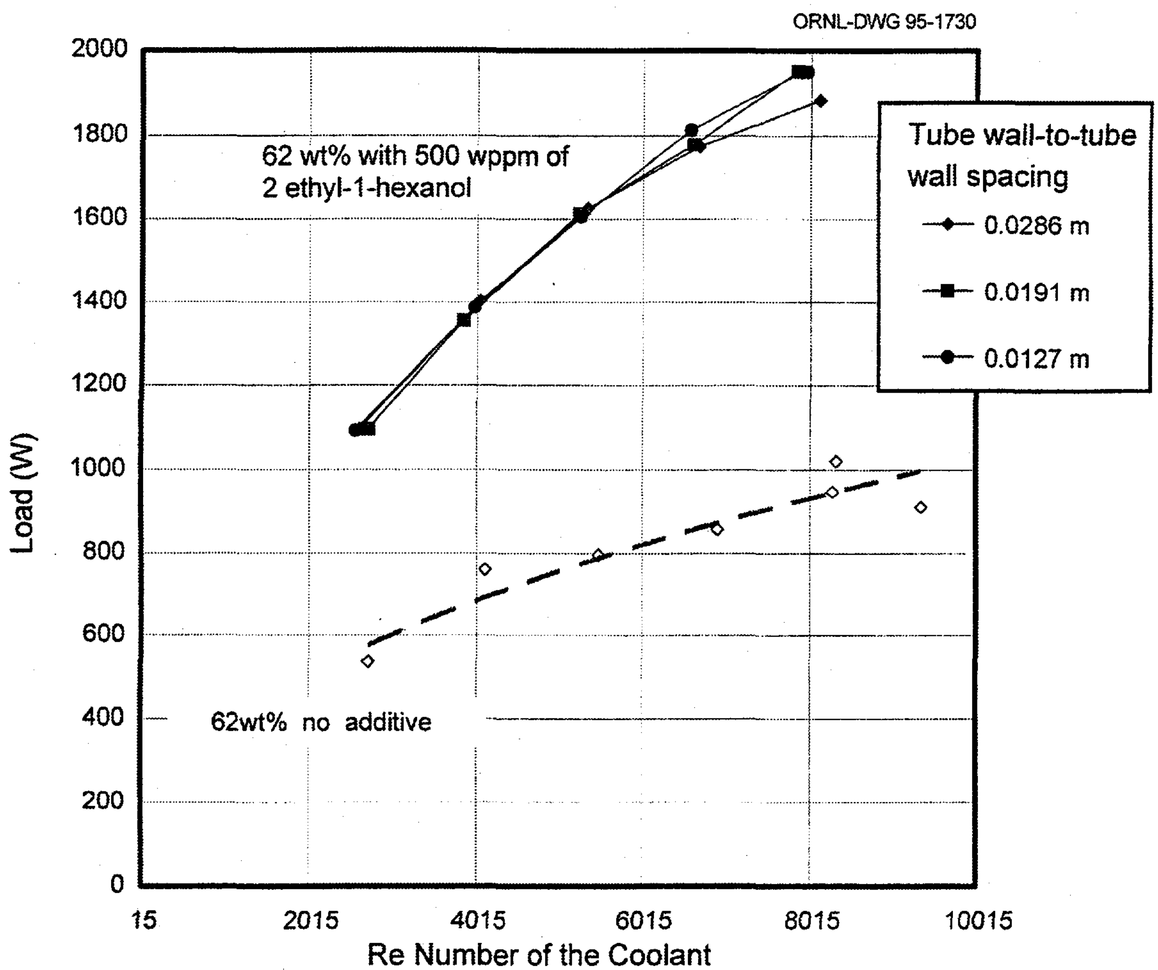
A domestic single-stage 200-ton chiller would typically operate at 6.3 mm Hg absorber pressure and 29.4°C coolant temperature. LiBr brine would leave the generator at about 64 wt % LiBr but would enter the absorber at a concentration of about 62 wt % as a result of recirculating design. The coolant flow per tube would be fully turbulent, having a Reynolds number of about 42,000 (i.e., 0.593 L/s per tube). We could not run the coolant at that high a flow through the mini-absorber but conducted tests to determine the effect of the coolant flow rate.

Increasing the coolant flow rate increased the water-side capacitance, which in turn supported more mass transfer into the falling film (Fig. 6). Results for testing at 62 wt % LiBr show a continued improvement in performance to Reynolds numbers greater than 8000. The driving force for absorbing the absorbate vapor into the absorbent falling film is the difference between the partial pressure of water just within the liquid-vapor interface and the pressure of the ambient absorbate vapor. This driving force is affected by the heat flow from the film to the coolant, which in turn is affected by the coolant flow rate as well as the coolant temperature and the concentration of brine entering the absorber.

Testing with additive resulted in a larger increase in the load and the mass absorbed as compared to testing without additive. Increasing the Reynolds number of the coolant from 3000 to 8000 almost doubled the smooth-tube-bundle's performance (Fig. 6). We tried to test at higher water flows; however, some of the return bends, which were O-ring vacuum compression fittings, came off the absorber tubes because of the coolant's increased static pressure at higher flow rates. As a result, we limited the higher coolant flow tests to Reynolds numbers less than 9000. Therefore, the load and mass absorbed have not reached maximum potential because the coolant-side flow is not fully turbulent and its heat transfer resistance is not minimized.

The Effect of the Falling Film Mass Flow Rate—*No Heat and Mass Transfer Additive*

Increasing the falling film flow rate showed increases in load. However, its effect is not as pronounced as that observed for coolant flow or for coolant temperature. Full wetting of the tube bundle was observed at a minimal flow of 0.007 kg/s. For testing as function of the falling film mass flow rate, the absorber pressure was 6.5 mm Hg, and the Reynolds number of the coolant was set at 5700. Coolant temperature was set at 29.4°C.



Coolant Temperature	29.4 °C
Absorber Pressure	6.5 mm Hg
Falling Film Flowrate	0.0091 kg/s
Absorber Surface area	8.36E-02 m^2

Fig. 6. The effect of the coolant flow rate on the load supported by the 6-tube bundle of smooth tubes tested with and without heat and mass transfer additive.

The subcooling measured at the exit of the mini-absorber appears to have a maximum inflection point for the range of film flows depicted in Fig. 7. The lowest flow tested had the thinnest film and the longest exposure to vapor. The subcooling was therefore lower because the film would saturate more completely than at the higher and thicker film flows. Note, however, that the absorber load and mass absorbed were also less than that observed at the higher flows. At the largest falling film flow tested, we observed an increase in the mass absorbed due to the more intense splashing action from tube to tube. The increased agitation between tubes explains the reduced subcooling. Tests at all three concentrations of LiBr and at all three tube spacings have similar inflection points. The physical properties of the brine, the tube-to-tube spacing, and the film flow rates must be considered in the design of the absorber. The design should minimize subcooling by delivering the maximum possible film flow. But if film bridging from tube to tube occurs, then the subcooling will increase. Therefore, an optimum exists between the film flow rate and the tube spacing that minimizes the subcooling and avoids the bridging of film between tubes.

The Effect of the Falling Film Mass Flow Rate—*Testing with Heat and Mass Transfer Additive*

Testing with 500 wppm of 2-ethyl-1-hexanol reduced the subcooling of the brine leaving the absorber to about 2°C (Fig. 7). The tube spacing had little if any effect on tests with heat and mass transfer additive. Further, sensitivity to the flow regime is not as evident as that observed during smooth tube testing without heat and mass transfer additive.

Domestic single-stage chillers have an absorber tube bundle with about 250 tubes. Generally, tube spacing is very tight; however, with additive present, the absorber performance is not affected by the tightly packed bundle. It would be of keen interest to monitor the subcooling leaving these tightly packed tube bundles to determine the life of the additive, especially since it is difficult to control the additive's location within the machine. Chillers use a continuous purge to collect and vent noncondensables from the absorber. Since the pressure of the additive is less than that of water vapor, it is very possible that maintenance practices accidentally purge the additive from the system over time. Further, the additive may degenerate over time after repeated exposures to the higher temperatures in the generator. For either case, the loss of additive would cause serious drops in the performance of the chiller. With a decreased amount of additive, the tightly packed absorber bundle would be strongly affected by the tube spacing, as we have seen from the results of testing with no heat and mass transfer additive. Hence, advanced tube surfaces that mechanically enhance wetting of the tube bundle would help improve long-term reliability of chiller equipment.

EXPERIMENTAL RESULTS—ADVANCED SURFACES

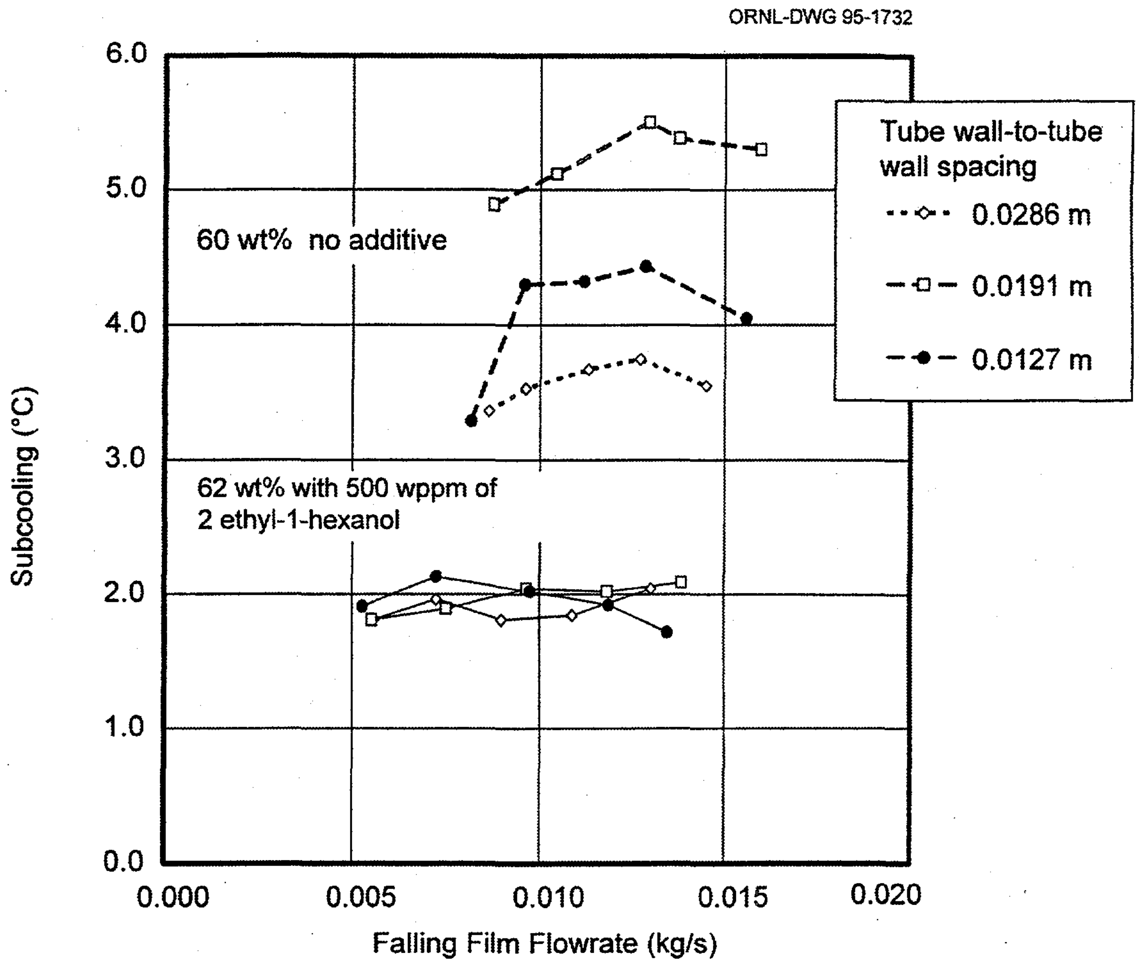
Absorption theory for laminar LiBr-water films predicts that the thermal diffusion of the film should be about two orders of magnitude greater than the mass diffusion; in other words, heat will be transferred readily across the solution film, while mass (water vapor) absorbed at the film surface will tend to remain at the surface, inhibiting absorption of additional water vapor. Mixing of the film can enhance the absorption process because the dominant concentration gradient occurs near the interface. Film mixing will induce secondary flows that increase the mass transfer. The mixing can be achieved by tube surfaces that mechanically induce secondary flows in the solution film without increasing the thickness of the film. Several tube geometries were identified and tested to determine the possible enhancements achievable through film mixing.

Wolverine's Turbo-C tube is enhanced on both the inside and outside surfaces. It was designed for condensing of refrigerants and other light hydrocarbons. The KoroChil is a corrugated tube that was designed for steam condensers, desalting plant condensers, and power plant feedwater heaters. The KoroChil tube showed excellent wetting of all the tubes, and it was also among the top five tubes in performance. The Trufin is an integral extended surface tube having helical fins extruded from the surface. Trufin is designed for the heating or cooling of gases.

The Thermoexcel-A tube is constructed from seamless copper tubing with thin integral fins that have been machined with axial grooves. Kawamata et al. (1989) found that the tube produces a synergistic effect with the presence of the alcohol additive n-octanol. The intent of the extended surface design is to stimulate the film's convective motion as it falls around the tube.

The grooved tube surface has six ridges machined atop the tube to promote mixing of the falling film around the tube. A bottom ridge was included to help develop a uniform drip pattern. The ridges are 0.66 mm high and 1.7 mm thick. The dimensions were gleaned from the published work of Davies and Warner (1969), who experimentally determined the optimum surface roughness for promoting the gas absorption of CO_2 into distilled water running down an inclined plate. Davies and Warner showed the gas absorption to be fastest when the roughness induced wake interference type oscillations in the flowing water. These authors observed that as water flowed down over the ridges, vortices developed around each ridge. For a given groove separation and height, the vortices oscillated, giving the surface a flickering, lattice-like appearance. If the groove spacing was too tight, the vortex shedding across the ridge was hindered: conversely, if the spacing was too large, the vortex was less. In either case, gas absorption was reduced.

The fluted-tube surface was also selected that is similar to a constant-curvature tube tested by Isshiki, Ogewa, and Sasaki (1991). The helix angle of the flute is 22°, and the fluted tube has 24 starts per turn. The fluted surface was selected as a good candidate surface that would pull the film along the axial length of the tube as it fell around the tube. By Young's equation for interfacial tension, the interfacial tension will induce a pressure difference in the liquid film and cause the film to flow from the peak regions to the valley regions of the surface. The film will thin near the peaks, and more rapid absorption will occur there. Gregorig (1967) pioneered this type of film-condensing surface.



Coolant Reynolds No.	5700
Coolant Temperature	29.4°C
Absorber Pressure	6.5 mm Hg
Absorber Surface area	8.36E-02 m^2

Fig. 7. Chemical agitation by heat and mass transfer additive causes the subcooling, measured at the absorber exit, to drop and shows the improvement in mass transfer performance of the 6-tube bundle of smooth tubes.

The Thermek pin-fin surface was selected because of its previous outstanding performance in vertical tube experiments. The Thermek tubes we tested had spine pitches of 5 and 3 mm, respectively. Each tube had 26 fins per turn. and the fin height was 3 mm. Testing of the Thermek tubes showed them to be poor for horizontal application. The tighter spine-pitch of 3 mm channeled the film flow and caused the film to thicken between spines. The spines also restricted any lateral movement of the film as it flowed around the tube.

The Effect of Coolant Temperature

We compared the advanced surfaces against the baseline of the smooth tube over a range of coolant temperatures (Figs. 8 and 9). Testing was conducted with the strong solution fixed at 62 wt % LiBr. The absorber pressure was set at 6.5 mm Hg, and the coolant flow rate was controlled to a Reynolds number of 5700. Here the smooth tube results are described for testing with and without heat and mass transfer additive. The results of the advanced surfaces are shown in Figs. 8 and 9 without the effect of heat and mass transfer additive. The solution flow was set at 0.011 kg/s.

The Wolverine Turbo-C and the KoroChil tubes supported the largest load—about 50% more load than did the smooth tube tested with no heat and mass transfer additive. Next best were the Excel-A, the proprietary Trane, the fluted, and the grooved tubes; all clustered together, they supported about 25% greater load than the smooth tube. The Trufin with 40 fins per inch followed closely, supporting 20% more load than the smooth tube. The Thermek with 5-mm pitch was next to last, and the loading for the Trufin with 19 fins per in. was no better than that of the smooth tube.

The mass of absorbed vapor (Fig. 8) follows closely the loading. Wolverine's Tubro-C and KoroChil absorbed about 50% more vapor than did the smooth tube having no heat and mass transfer additive. However, the Wolverine tubes supported only about 75% of the mass absorbed by the smooth tube (Fig. 8) when the smooth tube was tested with 500 wppm of heat and mass transfer additive (2-ethyl-1-hexanol). Hence, the enhancement induced by the mechanical mixing of the advanced surfaces is not as effective as the enhancement induced by the chemical agitation of the additive.

Increasing the outdoor temperature causes the load to drop linearly for all tested tube surfaces. It is seen that as the coolant temperature increases, the smooth tube tested with heat and mass transfer additive continues to support the largest load and mass absorbed (Fig. 8). However, as the coolant temperature increases, the load and mass absorbed for the smooth tube with additive appears to be converging to the clustering of all the advanced surfaces. The heat and mass transfer driving forces drop with increasing coolant temperature. Neither the chemical agitation nor the mixing induced by the advanced surfaces are able to counteract the reduction of the heat transfer effects from the interface. At a coolant temperature of 35°C, the mass absorbed for the Wolverine Turbo-C and KoroChil tubes is within about 12% of that absorbed on a smooth tube with heat and mass transfer additive.

We compared the subcooling leaving the various enhanced tube bundles for the loading measured during testing as a function of the coolant temperature. The thermodynamic state of the brine leaving the absorber is of particular interest to designers; it gages the mass absorbed as compared to the maximum mass that could be absorbed. Too much subcooling shows poor performance. It also indicates the degree of possible improvement. The greater the subcooling, the larger the amount of mass that the absorber can support through improved design. Hence, the level of subcooling shows the deviation from ideal performance.

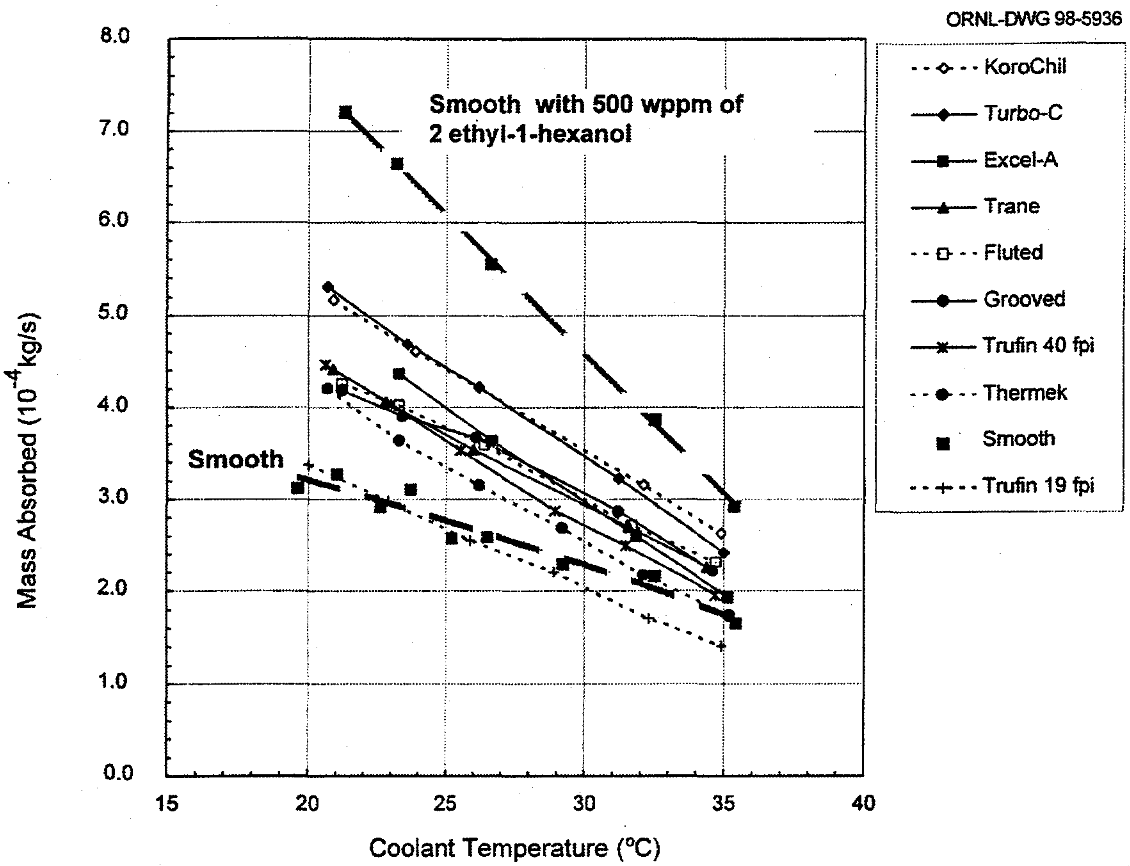
The subcooling for the smooth tube tested with no heat and mass transfer additive (large solid square in Fig. 9) varied from a low of 3.5°C for an entering coolant temperature of 35°C to a high of 6°C for an entering coolant temperature of 20°C. For a given load, the axial-grooved and KoroChil tubes had the least amount of subcooling of all the tubes tested. These two tube surfaces did the best job of mixing the film. The Turbo-C, Excel-A, Trane proprietary, fluted, and Trufin tube surfaces all had higher levels of subcooling. These advanced surfaces were not as effective in mixing the absorbate into the bulk of the film.

Testing with 500 wppm of 2-ethyl-1-hexanol reduced the subcooling measured for the smooth tube to about 2°C (Fig. 9). The superiority of chemical agitation is clearly proven over that of mechanical mixing for the given tested surfaces. Also, the subcooling remained about 2°C even as the coolant temperature varied from 20 to 35°C. Testing without the additive showed a stronger effect of the coolant on the subcooling. This again shows that the selected surfaces do not mix the film as well as does the chemical additive.

The Effect of the Falling Film Mass Flow Rate—*No Heat and Mass Transfer Additive*

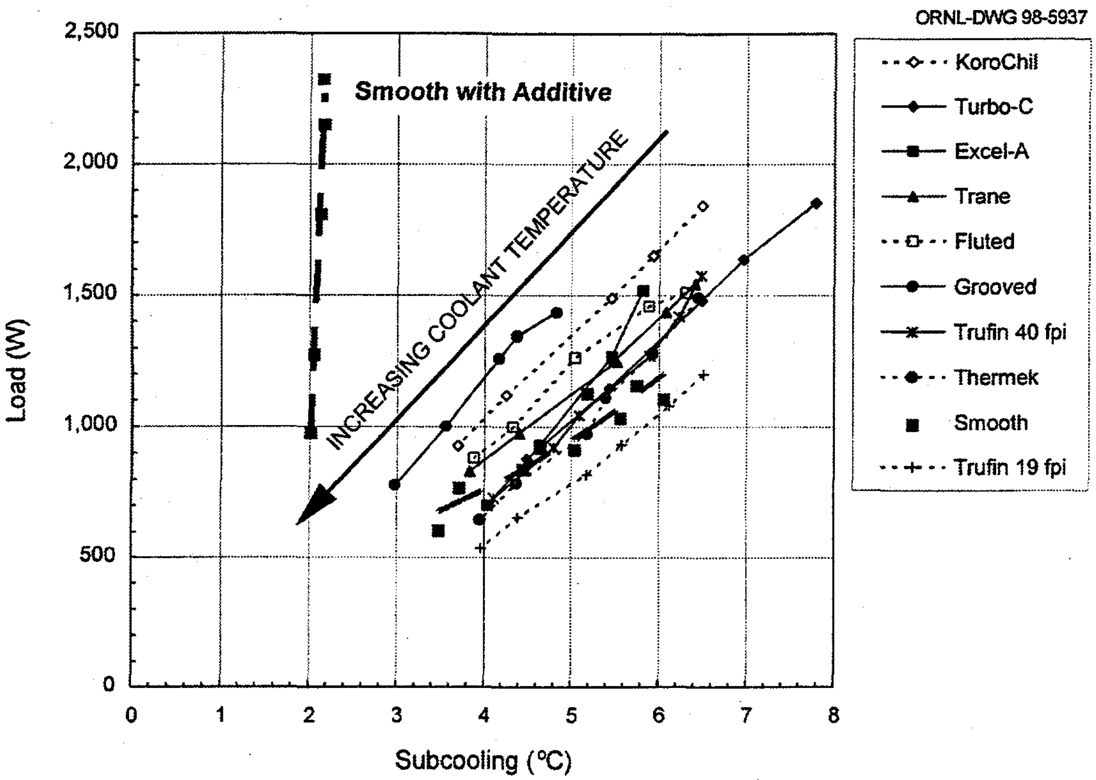
Testing continued with the advanced surfaces compared against the baseline of the smooth tube over a range of falling film mass flow rates (Figs. 10 and 11). The strong solution was fixed at 62 wt % LiBr, and the absorber pressure was set at 6.5 mm Hg. The coolant flow rate was controlled to a Reynolds number of 5700. Here both the smooth tube and the advanced tube test results are described for testing with and without heat and mass transfer additive.

Wolverine's Turbo-C and KoroChil tubes supported the largest load and therefore absorbed the greater mass (Fig. 10) for testing without heat and mass transfer additive. The higher loading and mass absorbed for both the Turbo-C and KoroChil tubes are due in part to each tube's internal enhancements. We verified the effect by testing the axial-grooved tube with straight bore at the higher coolant Reynolds number of 8300. This made the coolant-side resistance for the axial-grooved tube nearly equal to the resistance for the Turbo-C and KoroChil tubes and eliminated any confounding variables. With all other control variables fixed, the axial-grooved tube supported a load that was about 10% greater than the load for the KoroChil tube (1245 vs 1121 W, respectively). The additional testing shows that the axial-grooved, Excel-A, and Trane proprietary tubes would perform alongside the



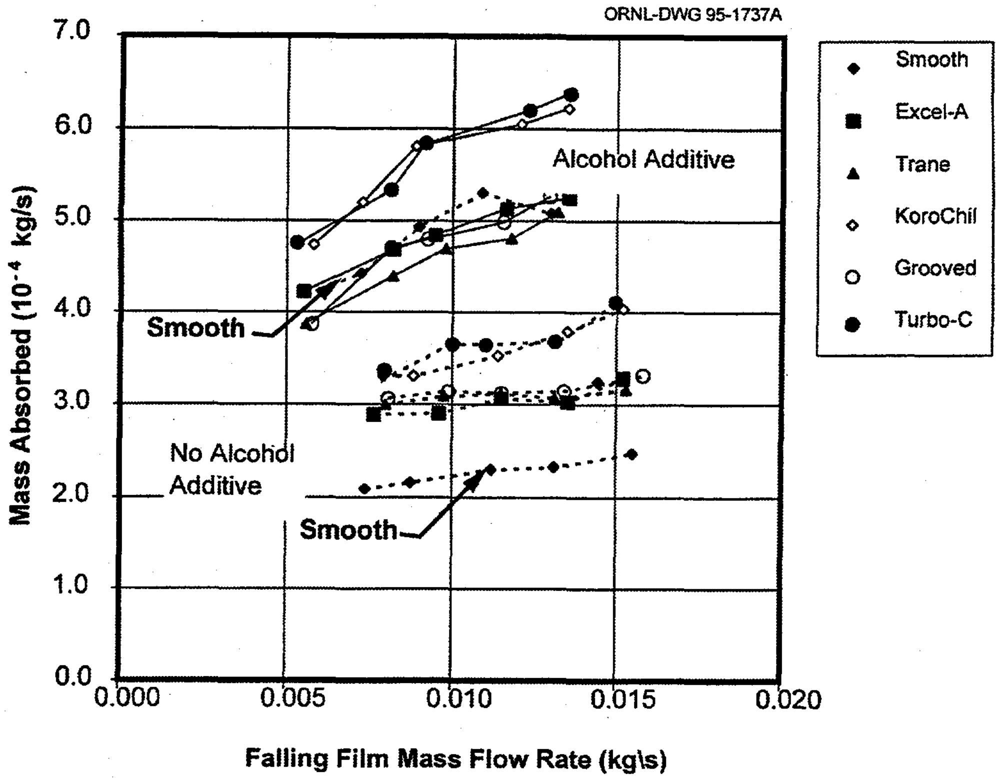
Tube Spacing	0.0286 m
Coolant Reynolds No.	5700
Absorber Pressure	6.5 mm Hg
Entering Concentration	62 wt% LiBr
Falling Film Flow Rate	0.0110 kg/s

Fig. 8. Increasing the coolant temperature causes a near linear drop in the mass absorbed by the advanced and the smooth tube surfaces.



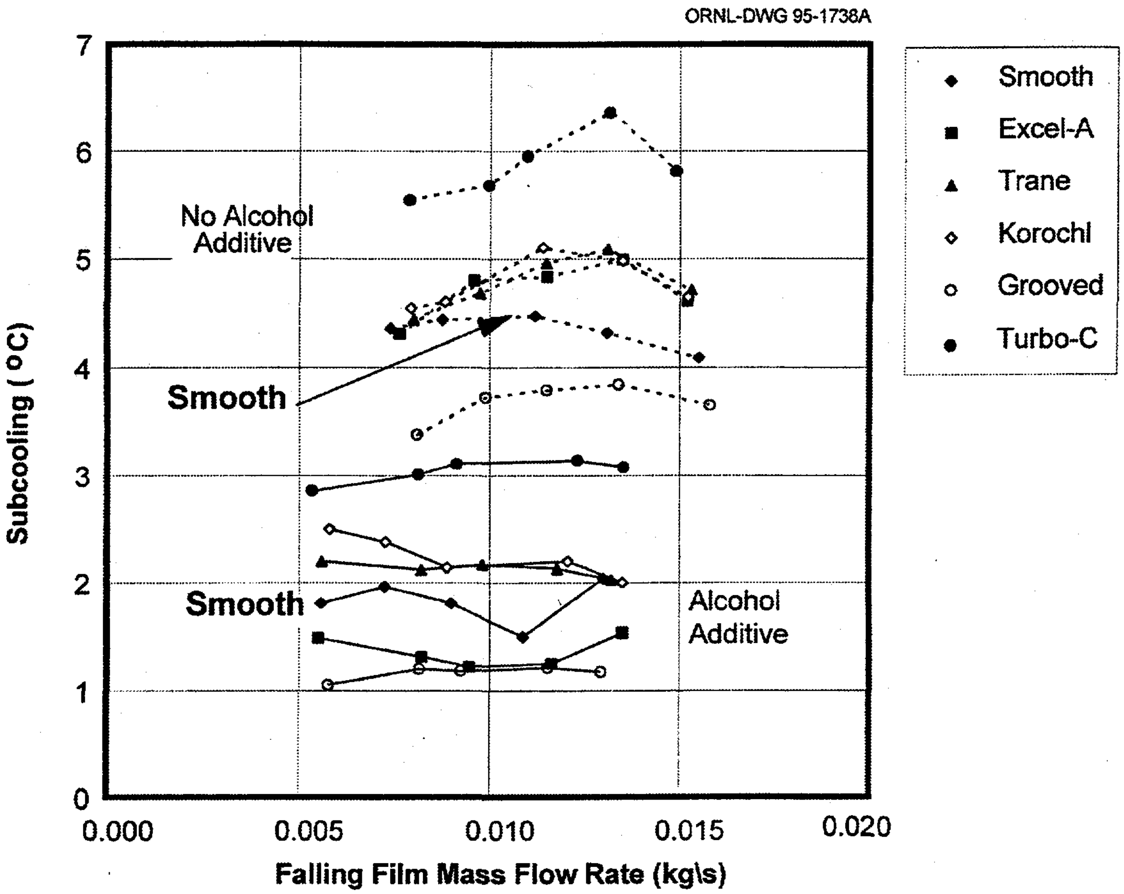
Tube Spacing	0.0286 m
Coolant Reynolds No.	5700
Absorber Pressure	6.5 mm Hg
Entering Concentration	62 wt% LiBr
Falling Film Flow Rate	0.0110 kg/s

Fig. 9. The axial grooved and KoroChil tubes did the best job of mixing the film; however, when compared to chemical agitation, the advanced surfaces are not as effective as the smooth tube bundle with 500 wppm of additive.



Coolant Reynolds No.	5700
Entering Concentration	62 wt% LiBr
Absorber Pressure	6.5 mm Hg
Tube Wall-to-Wall Spacing	0.0286 m

Fig. 10. The smooth tube and advanced tube bundles were tested with and without 500 wppm of 2 ethyl-1-hexanol.



Coolant Reynolds No.	5700
Entering Concentration	62 wt% LiBr
Absorber Pressure	6.5 mm Hg
Tube Wall-to-Wall Spacing	0.0286 m

Fig. 11. Testing with 500 wppm of 2 ethyl-1-hexanol reduced the subcooling of LiBr brine leaving the mini-absorber.

KoroChil and Turbo-C tubes if they also had internal enhancements. Hence, a designer could use internally enhanced absorber tubes and downsize the coolant pumps for a lower flow rate. However, the tubes must not have excessive coolant-side pressure drop, which is true of Wolverine's Turbo-C tube but not of Wolverine's KoroChil tube. The chiller would support the design loading at a reduced power draw thereby improving the efficiency of the machine.

Of all the tube geometries tested, the KoroChil fully wetted much faster than the other tubes: we were able to fully wet all six of the KoroChil tubes within only 2 hours of pre-absorption testing. The Turbo-C, Excel-A, Trane proprietary, fluted, and axial-grooved tubes fully wetted after about 12 hours of continuous pre-absorption testing. We thought the KoroChil would not perform well; however, flow visualizations show that the constant curvature that is characteristic of this tube pulls the solution over the tube's surface in a direction transverse to the path of the falling film flow. High-speed video revealed that the film was evenly distributed over the entire tube surface and that there was no preferred drip location from the bottom of each corrugation. Isshiki, Ogewa, and Sasaki (1991) had tested constant-curvature surfaces with concave geometry rather than the convex (dome-shaped) geometry of the KoroChil. Their research showed significant improvement in heat and mass transfer coefficients compared with their own smooth tube testing. These researchers reasoned that the constant curvature of the surface helped spread the film over the surface with a near-optimal film thickness. Too thin a film would saturate quickly, while too thick a film would cause a higher film-side heat transfer resistance, with a consequent reduction in performance in both cases.

For the advanced surfaces, we observed the solution to drip from different places along the bottom of an absorber tube. A droplet would grow in size until the balance of inertial forces exceeded surface tension forces. The droplet appears to expand until its weight causes it to release and fall. The droplet leaves a long narrow tail that remains attached to the upper tube. The expanded droplet leads what appears to be a stream tube to the lower tube. Stop-action photos show what looks like an inverted wine glass just before the expanded drop impacts the lower tube (similar to Fig. 5a). The droplet shatters and releases its contents. We believe that the splashing improved wetting of the tube bundles for both smooth and advanced tube surfaces. Further, we believe that the absorber design should promote the splashing action because the largest absorption rates may occur where the droplet shatters onto the top of each successive tube in the absorber bundle.

Of further interest in Fig. 11 is the inflection point occurring at maximum subcooling for all of the advanced surface tested without heat and mass transfer additive. Trends are very similar to those discussed for the smooth tube and show again that the design of the absorber should consider the physical properties of the brine, the tube-to-tube spacing, and the film flow rates. We did not see any film bridging on the advanced surfaces; however, we also tested the advanced surfaces only at the largest tube spacing. We believe, after viewing the high-speed video, that the surface roughness has little effect on the flow regime between tubes if heat and mass transfer additive is not present.

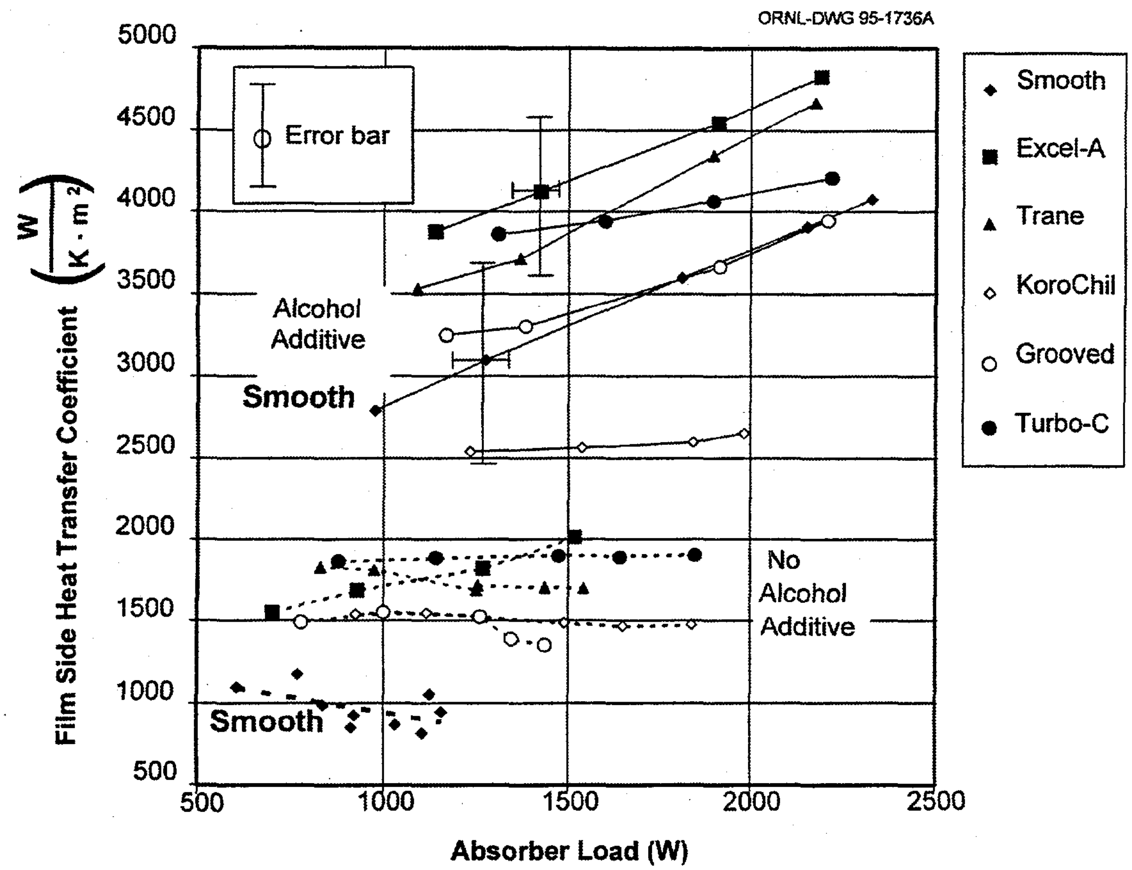
The Effect of the Falling Film Mass Flow Rate—*Testing with Heat and Mass Transfer Additive*

The 2-ethyl-1-hexanol additive causes the film to have several very thin spots interspersed between the wavy rivulets flowing around the perimeter of the tubes. These thinner spots reduce the local film-side resistance, which in turn increases the mass absorbed, as do the convective motions of the wavy rivulets (Fig. 10). The mass absorbed by the Turbo-C and KoroChil tubes almost doubled with the inclusion of 500 wppm of 2-ethyl-1-hexanol additive (Fig. 10). The mass absorbed by these Wolverine tubes exceeded the that absorbed by the smooth tube tested with additive by about 10 to 25%. In part, the internal enhancement of the Wolverine tubes accounts for the increase, and it is difficult to determine any synergy between the additive and surface geometry. The axial-grooved, the Trane proprietary and Excel-A tube bundles all performed the same as the smooth tube bundle.

The subcooling drops as the absorbed mass increases (Figs. 10 and 11). The results in Fig. 11 show that the smooth tube tested with heat and mass transfer additive had only about 2°C of subcooling as compared to about 4.5°C of subcooling for testing with no additive. The Thermoexcel-A and the axial-grooved tube bundles had the lowest subcooling of the tested advanced surfaces, just above 1°C; they are approaching optimal performance. The axial-grooved tube was designed to pool solution between the top two ridges. The solution would then be plowed under as it flowed over the ridges and around the tube. High-speed video shows pooling atop each grooved tube. The design helps prolong the exposure time of the film to vapor; therefore, the tube tended to have less subcooling than the other surfaces. The Thermoexcel-A surface has thin integral fins machined with axial grooves. Apparently, the surface design does not impede the lateral convective motion induced by the heat and mass transfer additive and a synergistic effect occurs similar to that observed in the independent testing of Kawamata et al. (1989).

The Synergism of Advanced Surfaces with Heat and Mass Transfer Additive

A simplistic look at Fig. 10 can lead one to view a synergistic effect of heat and mass transfer additive with Wolverine's Turbo-C and KoroChil tubes. Stated earlier, these tubes had internal enhancements which confounds a direct comparison between them and the tested surfaces having only a smooth internal bore. The straight bore tubes with advanced surface were tested at a higher coolant Re Number to match the coolant-side heat transfer resistance measured for the KoroChil tubes. The result showed similar loading and mass absorption for the KoroChil and the axial-grooved tube bundles. In practice, the coolant flow in the absorber tubes is fully turbulent; however, the coolant flow through the mini-absorber was limited to a maximum Re Number of about 9000 because of the design of the return bends and because the accuracy of the measured load drops as Re Number increases.



Coolant Reynolds No.	5700
Entering Concentration	62 wt% LiBr
Absorber Pressure	6.5 mm Hg
Tube Wall-to-Wall Spacing	0.0286 m

Fig. 12. A synergistic effect of heat and mass transfer additive and advanced surface is observed on the heat transfer coefficient.

We therefore plotted the film-side heat transfer coefficients as a function of the loading (Fig. 12). This eliminates the confounding resistance variable on the coolant-side and enables a fair comparison between the smooth and advanced tube bundles. The coupled heat and mass transfer process is film-side limited; in other words, mass transfer is controlled by the heat transfer from the film's interface. An increase in the film-side heat transfer coefficient above that observed for the smooth tube bundle tested with heat and mass transfer additive would show a synergistic effect of heat and mass transfer additive with advanced surface geometry.

For the smooth tube bundle, 500 wppm of 2-ethyl-1-hexanol additive caused the film-side heat transfer coefficient to more than triple that observed on the smooth tube tested with no additive (Fig.12). The large jump is attributed to the convective motions induced by the additive that improved wetting and helped induce a more uniform flow pattern around the tubes within the bundle.

The top five candidate surfaces, tested without heat and mass transfer additive, significantly increased the overall film-side heat transfer coefficient. The advanced surfaces mixed absorbate into the film and also helped spread the film over the tube's surface, causing the film-side heat transfer coefficient to nearly double the value measured for a smooth tube tested without heat and mass transfer additive. The maximum measured film-side heat transfer coefficient for the smooth tube was 1000 $W/K{\cdot}m^2$, measured at a loading of 900 W. The Turbo-C, Excel-A, and Trane proprietary tubes had heat transfer coefficients of about 1900 $W/K{\cdot}m^2$ for loading about 50% higher than that of the smooth tube. The best film-side heat transfer coefficient measured on the advanced surfaces was about 2000 $W/K{\cdot}m^2$, which is about 50% to 60% of that measured on the smooth tube with additive (Fig. 12).Again, the results show that the advanced surfaces do not mix the film nor spread the film as well as does the chemical agitation of 500 wppm of 2-ethyl-1-hexanol.

For testing with 500 wppm of 2-ethyl-1-hexanol, the Thermoexcel-A, the Trane proprietary and the Turbo-C surfaces show a synergistic effect and further boosted the film-side coefficient about 20% above that measured for the smooth tube also tested with additive. The results show a reordering of the best tube performers. Rather than the Turbo-C and KoroChil tubes, the Thermoexcel-A, Trane proprietary and Turbo-C tubes are the best performers when tested with heat and mass transfer additive. Interestingly, the heat transfer coefficient for the KoroChil tube was less than that measured for the smooth tube (Fig. 12). The KoroChil tube had not previously been tested as an absorber tube. While the corrugation helped wet the tube better than any other surface tested (Gregorig 1967), the tube's curvature is not optimally designed to spread the film over the surface. In other words, the film is too thick causing a higher film-side resistance. The KoroChil's performance in Fig. 10 is therefore strongly attributed to the coolant-side enhancement and the tube is actually penalized on the film-side (Fig. 12).

CONCLUSIONS

Testing with 500 wppm of 2-ethyl-1-hexanol caused the film-side heat transfer coefficient for a smooth tube to more than triple that observed on the smooth tube tested with no additive. The additive caused the film to better wet the smooth tube surface, and because of the local thinning of the film and the strong undulating wavy rivulets which drive convective motion, the heat and mass transfer was enhanced.

The advanced surfaces tested in this study do not induce the secondary flows seen by the chemical agitation of 500 wppm of 2-ethyl-1-hexanol. Hence, the enhancement generated by the mechanical mixing of the advanced surfaces is not as effective as the enhancement induced by the chemical agitation of the additive. The Wolverine Turbo-C and KoroChil tubes supported only about 75% of the mass absorbed on the smooth tube when the smooth tube was tested with heat and mass transfer additive.

The Thermoexcel-A, Trane proprietary and Turbo-C tubes showed a synergistic effect with additive that increased the heat and mass transfer by about 20% of that measured for the smooth tube tested with additive. The synergistic effect is rather small, limiting an affordable cost premium for advanced absorber design using the tested tube surfaces.

A single-stage chiller typically has about 250 smooth absorber tubes tightly bundled in several rows and tiers. In some equipment, the tube wall-to-tube wall spacing is tighter than 0.0127 m. Testing with additive showed that decreasing the tube spacing did not cause bridging between tubes. The tube bundles, regardless of the tube spacing, fully wetted when tested with the heat and mass transfer additive. A droplet flow pattern was observed on the underside of all the absorber tubes, even when testing at the tightest tube spacing of 0.0127 m. Therefore, the load and the mass absorbed were not strongly affected by tube spacing. Hence, tube spacing is not a critical design factor for the liquid-side if heat and mass transfer additive remains active in the absorber. Tube spacing is important, however, for design on the vapor side.

Over time, if the additive degenerates or escapes through the system's continuous purge, film bridging could occur and further degrade performance. The results observed for different tube-wall-to-tube wall spacing are then applicable. An optimum would exist between the film flow rate and the tube spacing that would minimize the subcooling and avoid the bridging of the film between tubes.

ACKNOWLEDGMENTS

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NOMENCLATURE

A	tube bundle surface area (m^2)
D	diameter (m)
h	convective heat transfer coefficient ($kW/K{\cdot}m^2$)
k	conductivity ($kW/K{\cdot}m$)
L	length of absorber tube (m)
T	temperature (°C)
U	overall heat transfer coefficient ($kW/K{\cdot}m^2$)
Q	absorber load (W)

Subscripts

c	coolant, coolant side
e	exit
eq	equilibrium
i	inlet
if	interface
in	inner radius or diameter
lm	log-mean
o	outer radius or diameter
p	pipe
s	bulk solution, falling film

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