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# Dual enhancement in HTC and CHF for external tubular pool boiling – A mechanistic perspective and future directions

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ABSTRACT

External tubular pool boiling plays an important role in a range of commercial, industrial, and residential applications including refrigeration and air conditioning, distillation, cryogenics, desalination, and steam generation in power plants. Enhancing the pool boiling performance while increasing the operating heat fluxes results in improved cycle efficiency and reduced equipment size. This approach requires dual enhancement in heat transfer coefficient (HTC) and critical heat flux (CHF) simultaneously. An increase in HTC improves the cycle efficiency in refrigeration evaporators, while increasing CHF permits employment of higher operating heat fluxes. As a means of improving the boiling performance over tubular surfaces, a number of enhancement techniques have been reported in literature. A detailed discussion of each enhancement technique and its underlying mechanisms is presented in this paper. The augmentations have been categorized based on the heat transfer enhancement method. Significant enhancements of up to 4–6-fold over plain tubes in the HTC have been reported with High-Flux, Thermoexcel, and Turbo tubes with somewhat limited enhancement in the CHF of 1–1.9-fold. In refrigeration and air-conditioning applications, the augmented tubes have been used in low heat flux ranges and the focus has been primarily on enhancing the HTC. Recently, new mechanistic approaches have been developed to significantly enhance CHF up to 2–3.8-fold and HTC up to 10–40-fold for pool boiling over flat surfaces. Specific experiments conducted on tubular surfaces have shown these mechanistic approaches to be effective for tubular surfaces as well. Based on these studies, a mechanistic approach for providing significant dual enhancement in both HTC and CHF for external tubular boiling is outlined. Such efforts are expected to enable operating the refrigerant evaporators efficiently (high HTC resulting in low wall superheat) at considerably higher heat fluxes (due to enhanced CHF) to dramatically reduce the size of the evaporators with resultant cost savings and environmental benefits as well from the reduced refrigerant inventory.



## 1. Introduction

Tubular surfaces are at the heart of flooded evaporators in large-scale refrigeration and air conditioning units, reboilers in distillation, desalination and chemical applications, and steam generators in power plants. In 2012, the commercial industry used 381 billion kW h of energy for cooling and refrigeration [1], and in 2010 the manufacturing industry used 61 billion kW h of energy for the same purpose [2]. For those two years combined, cooling and refrigeration consumed over 21% of the net energy used by these two sectors. The U.S. Department of Energy reported that air conditioning equipment accounts for a nearly $100 billion, 100 million-unit per year global market [3]. The underlying economic and environmental concerns in these applications provide a strong impetus to reduce the size and enhance the performance of the evaporators.

The International Energy Agency (IEA) predicts that by 2050, air conditioning energy consumption will increase 4.5 times and 1.3 times over 2010 levels for non-Organization of Economic Coordination and Development (OECD) countries and OECD countries respectively [3]. With such large growth expectations, advancement in system efficiency will not only allow for enormous cost and energy savings, but also greatly reduce the environmental impact related to energy production as well as reduced refrigerant inventory in the system.

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**Nomenclature**

| Symbol | Description [Units/Value] |
|---|---|
| $g$ | acceleration due to gravity [m/s$^2$] |
| $R_a$ | arithmetic mean roughness [μm] |
| $D_d$ | bubble departure diameter [m] |
| $N_r$, $N$ | bundle row number [–] |
| $T_{C,I}$ | chiller inlet [°C] |
| $T_{C,O}$ | chiller outlet [°C] |
| *COP* | Coefficient of Performance [–] |
| *CHF* | Critical Heat Flux [W/m$^2$] |
| $\rho_l$ | density of liquid [kg/m$^3$] |
| $\rho_V$ | density of vapor [kg/m$^3$] |
| $\mu_l$ | dynamic viscosity [Ns/m$^2$] |
| $q''$ | heat flux [W/m$^2$] |
| h | heat transfer coefficient [W/m$^2$ °C] |
| *HTC* | Heat Transfer Coefficient [W/m$^2$ °C] |
| $h_0$ | heat transfer coefficient at reference state [W/m$^2$ °C] |
| $h_{fg}$ | latent heat of vaporization [J/kg] |
| $M$ | molecular mass [–] |
| $R_p$ | peak roughness value [μm] |
| $p_r$ | reduced pressure [–] |
| $p$ | pressure [Pa] |
| *Re* | Reynolds number [–] |
| *Pr* | Prandtl Number [–] |
| $T_{sat}$ | saturation temperature of the fluid [K] |
| $C_{pl}$ | specific heat [J/kg-K] |
| A | surface area [m$^2$] |
| $C_s$ | surface factor [–] |
| $\sigma$ | surface tension [N/m] |
| $\Delta T$ | temperature differences for two evaporator temperatures [°C] |
| k | thermal conductivity of the fluid [W/mK] |
| $\alpha_l$ | thermal diffusivity [m$^2$/s] |
| Q | total heat transfer rate [W] |
| $D$ | tube diameter [m] |
| $T_{E,1}$, $T_{E,2}$ | two evaporator temperature scenarios [°C] |
| $X_0$ | void fraction [–] |

The aim of this paper is to highlight the current mechanistic understanding of tubular pool boiling enhancement techniques and provide guidance for future directions in the field. While previous works have provided more targeted reviews in different applications, a broader scope has been applied here to fully encompass the existing knowledge on tubular boiling mechanisms. In recent literature on the development of flat surface enhancements, it was demonstrated that understanding and exploiting underlying boiling mechanisms yielded the most significant improvements in heat transfer performance [4]. The present work provides a step towards applying this approach to tubular surfaces by first reviewing the current tubular enhancement techniques and providing guidance to direct the development of dual enhancement surfaces for tubular boiling.

### 1.1. Impact of external pool boiling enhancement at the system level

External boiling remains at the core of many applications including chemical industries and refrigeration/air conditioning systems. Performance improvements in heat transfer coefficient (HTC), critical heat flux (CHF), and coefficient of performance (COP) are used as indicators for quantifying the efficacy of the enhanced surfaces in refrigeration and air conditioning as discussed in this section. Further, the system performance improvements also result in reduced equipment sizes with cost benefits, space savings, and reduction in refrigerant inventory in the evaporator.

COP is a measure of overall refrigeration/air conditioning system efficiency determined by the ratio of the cooling effect in the evaporator to the power input. The evaporator temperature in a flooded evaporator remains almost constant at $T_{E,1}$ (ignoring slight variations due to the height of the tube bundles), while the chilled water temperature varies by a small amount from $T_{C,I}$ to $T_{C,O}$ as illustrated in Fig. 1. The average temperature difference between the evaporator temperature and the average chilled water temperature is indicated by $\Delta T_{E\text{-}C}$. This temperature difference is the sum of the two temperature differences, one on the water side and the other on refrigerant side. In comparison, the temperature drop across the separating metal wall is generally negligible. The water side HTC is relatively high resulting in the refrigerant-side temperature difference as the largest factor. In other words, the refrigerant side presents the dominant thermal resistance.

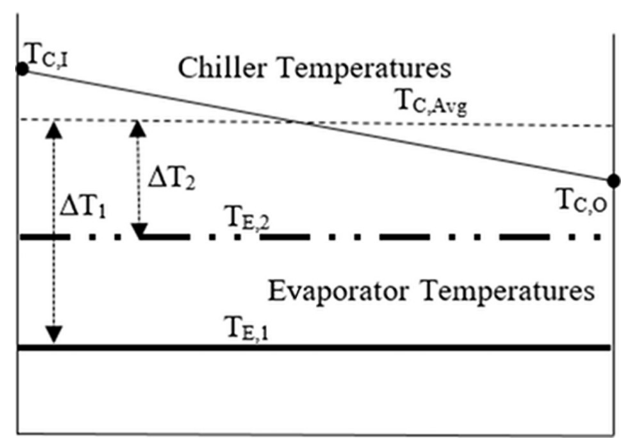


**Fig. 1.** Effect of higher evaporator temperature on the available temperature difference; $T_{C,I}$, $T_{C,O}$, $T_{C,Avg}$ – Chiller inlet, outlet, and average temperatures; $T_{E,1}$, $T_{E,2}$ – Two evaporator temperature scenarios; $\Delta T_1$, $\Delta T_2$ – temperature differences for the two evaporator temperatures.

In relation to Fig. 1, a higher evaporator temperature $T_{E,2}$ is desirable as it leads to a higher pressure in the evaporator and consequently a lower compressor power input and a higher COP. However, a higher evaporator temperature results in a lower available $\Delta T_{E\text{-}C}$. Since it presents the dominant thermal resistance, a higher value of HTC on the refrigerant side is desirable for improving the performance. Webb and Apparao [5] have presented an excellent analysis on the performance of flooded refrigerant evaporators employing enhanced tubes. They developed an analytical model to predict the performance of a 250-ton R-11 chiller operating at full load conditions. The reference evaporator bundle contains 180 tubes which are 3.43 m (135 in.) in length. Three metrics are compared: increased evaporator temperature, reduced evaporator size, and increased heat duty. When considering increased evaporator temperature, the water flow rate, water temperature, and evaporator heat duty are held constant at 136,510 kg/h, 11 °C, and 763 kW respectively. Considering only external surface enhancements, evaporator temperature was increased by 2.3 °C when using a porous outside tube due to a 1.6-fold increase in the bundle average HTC on the refrigerant side. In the example shown in Fig. 1, this