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Numerical analysis of heat exchanger designs for passive spent fuel pool cooling to ambient air

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ABSTRACT

Passive cooling of spent fuel pools via natural two-phase convection of a fluid with low boiling point is a promising alternative to active cooling circuits as such a passive heat transfer system would still work in safety–critical situations, such as a station blackout. For ambient air as the ultimate heat sink the heat exchanger design plays a crucial role as driving temperature differences may be low. This paper outlines a numerical investigation on a finned oval tube bundle heat exchanger operated under natural air convection in a chimney. We studied the role of chimney geometry and heat exchanger fin geometry. With respect to the chimney we found that velocity, Nusselt number and heat transfer are enhanced by 161.3%, 31.7% and 62.5% respectively, if chimney height increases from 2 m to 16 m. With respect to the fin design we determined an optimal fin configuration with a fin height of 17 mm, fin spacing of 3 mm and fin thickness of 1.5 mm, which improves the heat transfer performance by 28.7%, the Nusselt number by 28.9% and the fin efficiency by 19.2% at a given temperature difference of 40 K. The final optimized finned tube bundle heat exchanger design achieves a volumetric heat transfer density of $\bar{q}_{vol} = 3.61\frac{kW}{mK}$.

1. Introduction

The storage of spent fuel assemblies in large water pools is a common practice in nuclear power plants. The removal of decay heat generated from the fuel requires cooling. Such is commonly being done via an active cooling circuit comprising a pump, heat exchangers and water as heat transfer fluid. The Fukushima Daiichi accident in 2011 changed the view on the reliability of this kind of cooling systems, since cooling may fail in case of a longer persisting station blackout. A promising alternative approach is a passive heat removal. Passive heat removal systems base on a sustained natural convection of the heat transfer fluid driven by the power of the heat source. The heat has to be transferred to the ambient air. Hence the driving temperature difference is given by the nominal pool water temperature and the ambient temperature.

There are two principles of operation for passive cooling circuits. In a single phase loop the fluid (commonly a liquid) does not undergo phase change and natural circulation is driven by thermally induced density differences. In a two-phase heat transfer system the circulation is a result of the phase change. In a primary heat exchanger, the evaporator, which is located in the spent fuel pool, the heat transfer liquid is evaporated. The coolant flows as steam to a secondary heat exchanger, the condenser, and there transfers the heat to the secondary side, condenses and flows back as liquid to the primary heat exchanger. In both cases the secondary heat exchanger must be located sufficiently above the primary one.

Passive cooling systems are under consideration for the nuclear reactor and the spent fuel pool. In case of passive decay heat removal form the reactor core the temperature difference between the reactor and the environment is commonly high and so is the heat flux. Thus, e.g. Sviridenko (2008) proposed an emergency reactor cooling concept with heat transfer from the reactor to an air-cooled condenser using low temperature heat pipes or thermosiphons. A passive residual heat removal system for a 300 MW nuclear power plant was studied by Wang et al. (2013) using RELAP5. In this study both a heat exchanger placed in water tank and an air cooled heat exchanger surrounded by a chimney were considered as heat sinks. It was found that water cooling performs better than air cooling in the early period while the performance of the water-operated system deteriorates when the water in the coolant reservoir evaporates in the later stage. The air-cooled heat removal system is more effective on long term, since natural convection heat transfer enhances when chimney is used.

Temperature differences between nuclear reactors and the environment are high compared to the spent fuel pool. The natural

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Nomenclature

A	area of convection surface, m^2
A_{fin}	area of fin surface, m^2
A_{fr}	area in front of heat exchanger, m^2
C_{comp}	heat exchanger compactness, m^2/m^3
C_h	chimney height, m
c_p	specific heat, kJ/kgK
D_t	oval tube diameter, mm
g	acceleration due to gravity, 9.81 m/s^2
$\overline{h}$	average heat transfer coefficient, W/m^2K
h_f	fin height, mm
L	flow length, mm
L_{st}	oval tube straight length, mm
$\dot{m}$	mass flow rate, kg/s
N	number of tube rows
$\overline{Nu}$	average Nusselt number based tube diameter
p	pressure, Pa
Pr	Prandtl number
Q	heat transfer, W
$\overline{q}$	average heat transfer rate, W/m^2
$\overline{q}_{vol}$	volumetric heat flux density, kW/m^3K
s_f	inter-fin spacing, mm
t_f	fin thickness, mm
T_{air}	average air temperature, °C
T_f	fin Temperature, °C
T_{in}	air temperature in front of the heat exchanger, °C
T_{pl}	tube pitch longitudinal, mm
T_{pt}	tube pitch transversal, mm
T_{out}	air temperature behind the heat exchanger, °C
T_{tube}	outer tube surface temperature, °C
ΔT_{air}	front-to-behind air temperature difference, K
ΔT_{HT}	base-to-frontal temperature difference, K
V_{inst}	installation volume of a heat exchanger, m^3
x,y,z	Cartesian coordinates
u,v,w	velocity components in x, y, z direction respectively m/s,

Greek symbols

α	thermal diffusivity, m/s^2
β	thermal expansion coefficient, 1/K
ε	turbulence dissipation rate, m^2/s^3
η	fin efficiency
μ	dynamic viscosity, kg/ms
μ_T	turbulent viscosity, Ns/m^2
ν	kinematic viscosity, m/s^2
ρ	density, kg/m^3
λ_{air}	thermal conductivity of air, W/mK
λ_S	thermal conductivity of fin material, W/mK
ω	turbulent frequency, s^{-1}

convection in a passively cooled spent fuel pool tank was simulated by the means of CFD by Merzari and Gohar (2012). The porous medium approach was used to investigate the heat transfer in the fuel tank. Likewise the behaviour of a passively cooled spent fuel pool was analysed by Ye et al. (2013) for different cases. The simulations could prove that the cooling system removes the decay heat from the spent fuel pool effectively. A porous-medium model was used in a similar manner by Hung et al. (2013) to analyse heat transfer capability of spent fuel pools. Different spent fuel configurations were considered for the pool under full-core discharge and the assumption of failing external cooling system. Without an external cooling system local boiling occurs on the spent fuel surface for all configurations. An experimental analysis of a loop-type passive residual heat removal system was described by Xiong et al. (2014), where ammonia was the working fluid and air represented the heat sink. In the framework of this study different parameters were varied, e.g. the air velocity, hot water inlet temperature and volumetric filling ratio of the heat pipe. The same authors evaluated a similar setup, which consisted of a large-diameter and long-length evaporator, where water was used as a working fluid (Xiong et al., 2015). For both the parametric sensitivity analysis suggested a significant effect of the pool water temperature on the heat pipe performance, followed by air velocity, air temperature and water flow rate. An extensive review on single phase natural circulation loops for solar thermal systems and nuclear thermal hydraulics was given by Basu et al. (2014). The authors presented different scaling methodologies and modelling approaches. Thereby also unconventional topics, like nanofluids, natural circulation loops for marine reactors and system dynamics issues were addressed. A recent study on a passive residual heat removal heat exchanger submerged in an in-containment refuelling water storage tank was carried out by Lu et al. (2016). In the experiments a C-shaped heated rod bundle was used to mimic the heat exchanger. The velocity distribution in the water storage tank was measured qualitatively by particle image velocimetry technique and the temperature distribution was measured by thermocouples in different positions. An increasing heat transfer coefficient appears after 4000 s, when the heat transfer mechanism changes from single phase natural convection to sub-cooled boiling. The authors proposed correlations to predict the heat transfer coefficient for this scenario.

In most applications ambient air represents the final heat sink, independent of the component to be cooled and the operation mode of the cooling system. The heat transfer to air can in principle occur under natural convection and forced convection. However, as we consider passive heat removal we also assume only natural convection at the secondary side of the heat exchanger to air. Natural convection occurs due to density differences induced by the heat-up of air. Beside passiveness, natural convection designs are furthermore simple, cost effective and require minimum maintenance (Senapati et al., 2016). However, a big drawback is the low heat transfer rate. To overcome this disadvantage, extended heat exchanger surfaces are common practice. The finned tube heat exchanger is the most common device for natural convection heat transfer, used e.g. in electronics cooling, cooling systems for air conditioning and refrigeration, gas turbines and compressors. Due to small temperature differences between spent fuel pool water and ambient air, special care is needed in the design of a passive heat removal system (Fuchs et al., 2015). Since the temperature and velocity distribution are influenced mutually in natural circulation, heat transfer performance is difficult to predict (Hanjalic and Vasic, 1993). This task is approached in the framework of this study by a numerical investigation of finned oval shaped tube bundle heat exchanger cooled by air.

Most commonly heat exchanger tubes have a round shape. Nevertheless, recent studies suggested that oval tube shapes are beneficial over annular tube shapes. Kumar et al. (2016) performed an extensive study of the thermal hydraulic characteristics of air cooled fin and tube heat exchangers. Different tube designs were analysed (annular tube with 24 mm diameter and oval tubes with 30 × 10 mm, 30 × 15 mm and 30 × 20 mm size). They observed that the flow separation on the tube surface occurred later for oval tubes, which resulted in a smaller wake region. The results indicate a slight increase in heat transfer coefficient and a strong decrease of pressure drop with decreasing ellipticity of the tube. Similar findings were reported by Bhuiyan and Islam (2016). A numerical and experimental study on heated oval shaped cylinder was performed by Li et al. (2014). The axis ratios of the oval shaped tubes were varied and heat transfer performance was analysed. The best heat transfer performance occurred for a minor-to-major axis ratio of 1:2. For finned tube heat exchangers an