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Random vibration protection of a double-chamber submerged jet impingement cooling system: A continuous model

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ABSTRACT

High-powered embedded computing equipment using air transport rack (ATR) form-factors is playing an ever-increasing role in aerospace applications. High power and wattage of the electronics and processors require large heat dissipation. Thus more sophisticated, more efficient, and sometimes exotic thermal cooling systems like loop heat pipes or jet impingement systems are demanded. Despite their better thermal performance, these systems are usually more susceptible to mechanically harsh environment where they are deployed. Random vibration is one of the primary excitation sources in aerospace environments where highly efficient cooling systems are used. In this article, a multiple degree of freedom model of an isolated double-chamber jet impingement cooling system is developed, and its response to random vibration is analyzed and compared to that of a traditional single degree of freedom model. Vibration isolation system properties are optimized to minimize the vibration of the internal components of the cooling system, while the whole system enclosure is confined within an allowed rattle space.

1. Introduction

Electronic components are becoming increasingly miniaturized and integrated. This allows for computers and electronic equipment with dense printed circuit boards (PCB) to become more compact in dimensions [13]. The result is an increasing heat flux at both the component and the circuit board levels. Conventional methods for cooling are not sufficiently effective for these systems. Research into more advanced methods such as liquid-cooling and spray-cooling is being done. These methods can provide higher heat dissipation power and can be used in rugged environments if they are ruggedized accordingly. The objective of this paper is to develop a good model of vibration isolation for a liquid-cooling system.

In this paper a double-chamber submerged jet impingement cooling system is used to provide 2 kW of heat dissipation for an electronic equipment. The CAD model of the impingement system is illustrated in Fig. 1. In this system, water is pumped into the chamber housing, and is then pushed through tiny nozzles (with a diameter of 1 mm) of the nozzle plates at two sides of the chamber housing. Thus, the water is accelerated and impinges with higher velocity on the impingement plates at two sides of the cooling system. Impinged water on the impingement plates thus removes heat from the heating sources that are mounted back to back on the impingement plates (heating sources are not shown in Fig. 1). A thermal diagram of the cooling system is depicted in Fig. 2.

The cooling chamber shown in Fig. 1 along with other components is accommodated within an Air Transport Rack (ATR) chassis as shown in Fig. 3. It should be noted that the Heat Rejection Unit (HRU) where the heated water is cooled down is not placed inside the chassis. The whole chassis with the included components is subjected to severe random vibration in three directions (2 lateral and 1 vertical) from the bottom face (base) of the chassis. The main objective of this research is to protect the cooling system shown in Figs. 1 and 3 against the random base excitations. The aim of vibration protection is to prevent any loss in the thermal performance of the cooling system and any mechanical failures. This will be done by designing an optimum passive vibration isolation system.

The traditional optimal design for vibration isolation from random vibration is based on a trade-off choice of damping and stiffness properties of isolation mounts. The traditional design is focused primarily on optimizing the dynamic response of the entire enclosure or chassis, subject to limitations imposed on their maximum vibration travel (rattle space) [15]. However, some devices include lightly-damped components that are extremely responsive over a wide frequency range. These components are not considered

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Nomenclature

σ	standard deviation
S	power spectral density
T	transfer function
k_c	stiffness of the vibration isolation system
c_c	damping coefficient of the vibration isolation system
m_c	chassis mass (including the components inside)
ω_c	undamped natural frequency of the vibration isolation system
ζ_c	damping ratio of the vibration isolation system
x_c	chassis displacement
x_b	base displacement
z	chassis displacement relative to the base
AF	attenuation factor
w_a	absolute displacement of the plate model
w_b	boundary displacement of the plate model
w	deflection of the plate model
a	length of the plate model
b	width of the plate model
ρ	aerial density of the plate model
c	viscous damping coefficient of the plate model
D	flexural rigidity of the plate model
ψ_{mn}	*mn*th eigenfunction of the plate model
q_{mn}	*mn*th time-varying modal amplitude of the plate model
ω_{mn}	*mn*th undamped natural frequency of the plate model
β_{mn}	frequency bandwidth of the *mn*th mode of the plate model
ζ_{mn}	damping ratio of the *mn*th mode of the plate model
m_{mn}	modal mass of the *mn*th mode of the plate model
c_{mn}	modal damping of the *mn*th mode of the plate model
$E[\dots]$	expected (mean) value

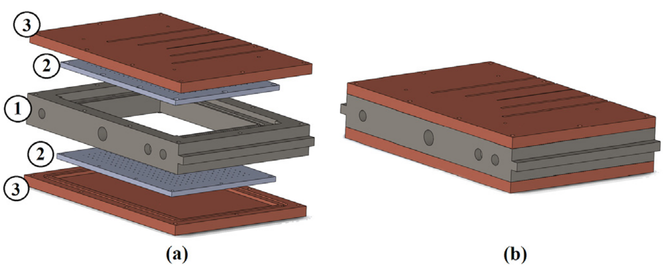

Fig. 1. (a) Exploded view of the assembly of the jet impingement system: 1. chamber housing, 2. nozzle plate, 3. impingement plate. (b) Assembled jet impingement chamber.

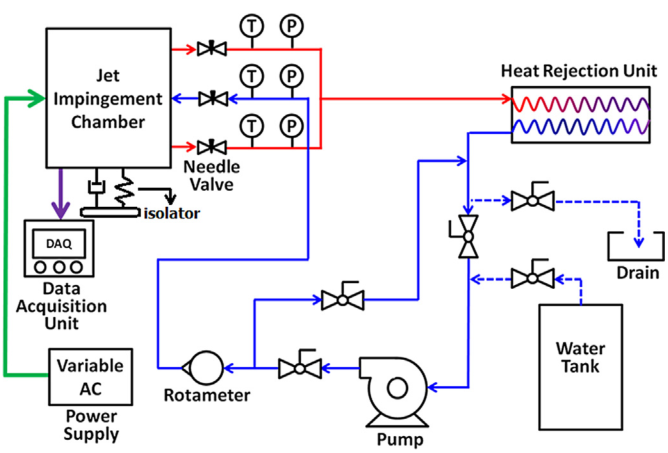

Fig. 2. Thermal diagram of the jet impingement cooling system.

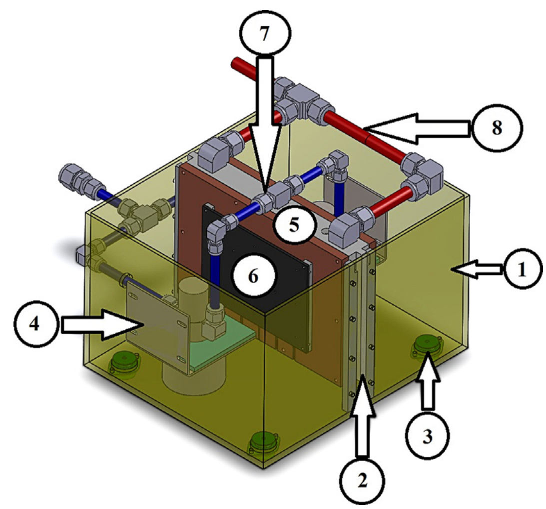

Fig. 3. CAD model of the ATR chassis with all the included components: 1. chassis, 2. wedge locks, 3. isolators, 4. micro pumps, 5. cooling chamber, 6. heating boards, 7. inlet tubing, 8. outlet tubing.

in the traditional vibration isolation system design. Consequently, traditionally designed vibration isolation systems ignoring these internal components are insufficient for maintaining a fail-safe vibration environment for the system.

Although the traditional design may fulfill the shock or vibration requirements in some applications [1,2,5,10,12,21], it does not always result in an optimum design. Therefore, the conventional method of vibration isolation system design has been challenged in the last decade in some areas like electronics, cryogenic coolers and hard disk drives. Veprik and Babitsky added a secondary degree of freedom for a PCB to the model of an isolated rigid electronic box [16]. They tried to optimize the vibration isolator's parameters for a harmonic excitation to minimize the PCB's absolute acceleration or relative deflection, subject to an overall rattle space. Later, Veprik used the same model to optimize the isolator's properties this time for random vibration input [15].

Veprik and his colleagues designed an isolation system for an infrared equipment to minimize the vibration-induced line-of-sight jitter which was caused by the relative deflection of the infrared sensor and the optic system [17]. They added a second degree of freedom for the cold finger of the cryogenic cooler to the main SDOF model of the system to improve the model as a more realistic 2DOF system. Later on, they used this 2DOF model which included the sensitive cantilevered cold finger to design optimum compliant snubbers for impact protection [18]. There has been some research work conducted to enhance vibration protection for workers working with percussion machines taking into account internal components of these machines for a better design of the protection system [3,4,14].

The authors previously designed an integrated linear–nonlinear vibration isolation system for the ATR chassis mentioned above catering for base random vibration input in presence and absence of g-loading [7]. However, their model ignored the presence of the internal components of the cooling system that could be excited by the input vibration. Hence, in their more recent research study, the authors have developed a 3DOF model of the cooling chamber (a different design of the cooling system from the one studied in